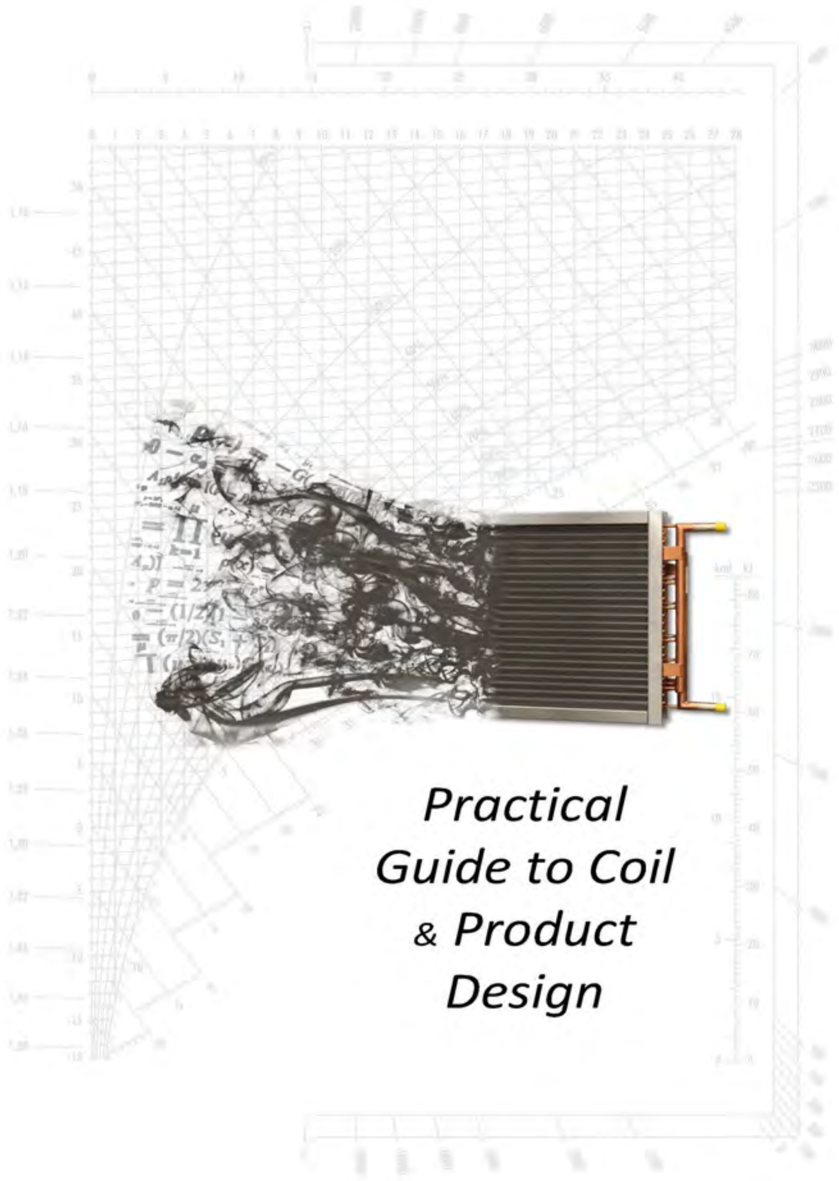


Practical Guide to Coil & Product Design

3RD EDITION

ABSTRACT

The purpose of this document is to provide a foundation for heat exchanger (coil) and coil related product design and selection and to provide a knowledge base for the technology used by EAS's selection software; currently known as CoilCalc; encompassing generalised thermodynamics, fluid dynamics, aerodynamics and acoustics.

Compiled by : Shaun M. Nash

January 2025

CONTENTS

Throughout this document, the pictograms below are used to underline points or important notions

	Important information
	Good to know - Tricks
	Beware
	To be avoided
	Mandatory action

INTRODUCTION

The author wishes to clarify that very little of the content of this design manual is 'original work', but a compilation of data and information from a wealth of sources such as the internet, heat exchanger design manuals, industry relevant literature etc., plus 50 years of personal experience.

The document has been collated to hopefully provide a simple to understand and coherent source of coil and product related information specifically related to HVACR industry sector.

As EAS does not have an official 'Design Handbook', this compilation may serve as an alternative.

Novices to this industry may find the document useful as an introduction into this business, whilst others may find the information and data useful as a 'one stop shop'.

TYPES OF HEAT EXCHANGER

Tube and plate fin heat exchangers, otherwise known as **Coils** in our industry sector, are just one type of heat exchanger available for transferring energy between two flow streams. Other common types of heat exchangers, primarily for liquid-to-liquid heat transfer are …

- Shell and tube heat exchangers
- Brazed or welded plate heat exchangers
- Coaxial tube-in-tube heat exchangers

However, tube and fin coils are best suited for transferring heat between a liquid and a gas due to the high surface area available on gas side compared with the low internal surface area available on the tube side. This suits the large disparity between the internal and external heat transfer coefficients associated with gases and liquids.

HEAT EXCHANGER APPLICATIONS

Coils are used in a wide range of applications …

- HVAC, refrigeration and heat pumps …
    - Duct mounted steam, hot/chilled water and DX/condenser coils
- Process heating & process cooling …
    - High temperature exhaust gas cooling
    - Process gas pre-heating
    - Oil coolers
    - Ammonia air cooled condensers
    - Power transformation dry air liquid coolers
- Closed circuit condenser desuperheaters
- Closed circuit cooler anti-plume dry coils
- Off-shore sea water cooling applications

.. to name a few.

PERFORMANCE CHARACTERISTICS OF TUBE AND FIN HEAT EXCHANGERS

The driving force behind heat transfer is temperature difference and if this difference is diminished the heat flux will reduce and thus to achieve a desired capacity, more surface is required. More surface is typically achieved by reducing the fin pitch or increasing the rows deep.

However, the geometry of the tube and fin heat exchanger also has a direct bearing upon the coil's ability to transfer heat. Generally, the closer (more compact) the configuration the greater the ability to transfer energy and the closer the bulk surface temperature of the fin is to the bulk temperature of the fluid flowing in the tubes. But remember, compact geometries are not suitable for all applications and can exhibit detrimental performance characteristics for certain types of applications.

Clearly, the surface temperature of the fin governs the limit to which the temperature of the air can be raised or lowered. Thermodynamically each type of geometry, fin pitch and material make-up of a given coil will constitute a minimum allowable ‘approach temperature’. The approach is defined as the difference between the air temperature at a given point and that of the inside tube bulk fluid temperature.

It is practically impossible for the air temperature to exactly reach the fluid temperature when there is a physical barrier separating the cross-counter flow mediums. This barrier is made up from the respective medium boundary layers, tube and fin material resistances, bond contact between tube and fin material and internal and/or external fouling factors.

Under controlled laboratory conditions, what may be referred to as ‘thermal saturation’ can be achieved where approach temperatures of as little as 1.75K have been measured. But such conditions are not realistic in practice and thus wider temperature differences will prevail in ‘real’ applications.

Thus, EAS limits this governing constraint to around 2.5K in their selection software to avoid misuse and misunderstanding of typical coil applications.

In fact, it is the responsibility and discretion of the Engineer performing the design and selection to understand the true coil application and allow a selection to be accepted if the applications were to compromise the coil performance.

Approach temperatures achievable will depend upon the operating coil face velocity, fin pitch, materials of construction, coil orientation and location in its ultimate operating environment plus the operating temperatures and differences. All these constraints will affect the coil behaviour and performance and govern when effective thermal saturation is reached.

In view of the above and the knowledge and experience that these considerations entail, it is wise to set the approach temperature to 3 or 4 K to ensure predictable and reliable performance.

Another important consideration is when the fin surface is operating in a ‘wet’ state. Again, the behaviour is affected and must be duly considered.

It is often misunderstood, but when a coil reaches ‘thermal saturation’, any amount of additional surface will not solve the under-performance issue. In fact, for a given fan/motor/pressure drop scenario, more surface (rows or fins) will increase the total static pressure drop and thus reduce the delivered air volume through-put and hence make matters worse !!

The only way to improve the capacity from a thermally saturated installation is to either increase the air volume or alternatively install a more efficient heat exchanger – if they are available !!

Another consideration often misunderstood, resulting in application misuse and applies to most coil manufactures, is that all performance predictions are in line with certain European or International standards. In our case, tests in accordance with EN1216 have enabled us to conduct coil performance tests and fully understand the behaviour. This has helped the development of mathematical models to be formulated and included within design software.

Clearly, laboratory tests and conditions are set up to be ‘ideal’, however ‘real’ applications are often far from this assumption !!

Therefore, it is essential that Engineers make allowances for the application and location of the coil. Generally, this is achievable by using the Surface Margin/Reserve option within the design software to force an over-sized coil to be selected.

Over-surfacing will help counteract the effect upon the performance of a coil in typical applications where the airflow is often ‘non ideal’ and the pattern is disturbed. Furthermore, items of plant and machinery are often placed too close to the coil and the location in ducting or AHU’s can cause temperature stratification implications within the air stream. All these factors will result in the coil under-performing compared with the ‘ideal’ predictions.

As a manufacturer of only one of the components of what will become a complete system, we cannot know or are often not informed of the ‘whole picture’ and thus cannot judge how to adjust the design for whatever ramifications these factors may impose.

If we perform the design task without the above knowledge or information, the resulting selections could well be undersized. A simple analogy would be requesting the manufacture of an item with a drawing that has certain critical dimensions missing !

Failure to consider the real implications upon an application of a coil in its final environment constitutes neglect in a crucial area of the design process.

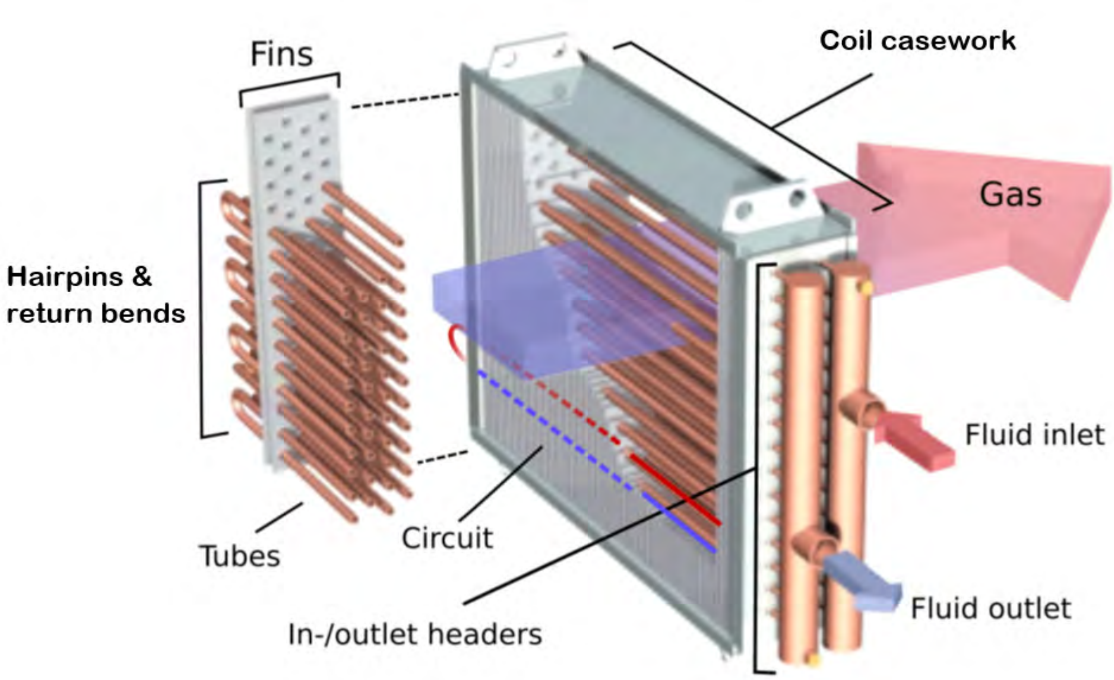
COIL COMPONENT PARTS

Coils comprise the following major components …

- **Tubes** – a variety of diameters and a variety of materials
- **Fins** – a variety of hole pattern geometries manufactured from a variety of materials with a variety of fin pitches
- **Inlet and outlet manifolds** (headers) - materials typically matching the tube material in a variety of sizes and configurations/orientations
- **Casework** material - to suit the application and/or tube and fin materials

The following is a sketch showing the general assembly of the above components …

MAJOR COIL COMPONENTS AND THEIR GENERAL ARRANGEMENT



MANUFACTURING PROCESS

The following 4 processes are conducted in parallel to provide the components for assembly …

- Tubes are cut to length and if the application/coil size warrants it, bent through 180° to form 'hairpins'
- Sufficient fins are punched and cut to size to form the finned matrix block
- The encasing sheet metal is cut to size, folded and the tube plates are prepared with flared clearance holes to accommodate the tubes
- The inlet and outlet header sub-assemblies are cut to size and the end caps and connections fittings TIG welded or brazed, whilst the header legs/stubs are welded/brazed into position to match the circuitry pattern

COIL ASSEMBLY

The above parts can then be assembled as follows …

- The sheet metal casework component parts are assembled and stacked with sufficient fins
- Straight tubes and/or hairpins are inserted into the fin pack to match the circuitry pattern
- The tubes are mechanically expanded in diameter using a hardened steel 'bullet' to ensure an 'interference fit' between the tube and fin's collar to ensure a good bond and thus good heat transfer
- Return bends are brazed or welded into the tube ends to complete the desired circuitry

- The inlet and outlet header sub-assemblies are offered up to the matrix block and the header stubs brazed or welded to the appropriate tubes to complete the coil
- The coil is partially pressurised with dry air and then immersed in a water tank and fully pressurised in accordance with the Pressure Vessel Directive – 2014/68/EU to test for any leaks
- Any leaks identified are repaired and the pneumatic pressure test repeated
- The connections are plugged with plastic caps to avoid the ingress of debris and ensure the internal surfaces of the coil dry and clean
- EAS labels are affixed in accordance with the PED
- A pressure test certificate is generated in accordance with the design requirements in addition to any documents of conformity and material certification as dictated by the PED

COIL RELATED PRODUCTS

Besides coils being supplied as discrete heat exchangers for specific applications, they can be incorporated into a sheet metal enclosure and fitted with fans, motors and legs and thus transform into products such as …

- Dry air liquid coolers, otherwise known as …
  - Dry coolers
  - Radiators
  - Fin fan coolers
  - Blast coolers
- Air cooled condensers
- Liquid air coolers – products using brines to cool air for cold storage applications
- Evaporators – DX or pump circulated refrigerant products for cold storage applications
- Hybrid adiabatic dry coolers and condensers using …
  - Water spray nozzle technology
  - Evaporative cooling pad technology
- CO_2 gas coolers

INDUCED DRAFT & FORCED DRAFT

Many of the products described above can be manufactured in two varieties …

- Induced draft (sucking) – fans mounted on the top of a horizontal flatbed product
- Forced draft (blowing) – fans mounted on the underside of the flatbed product

In essence, the two variations of the same product would perform very similarly, however, there can be some situations where one of the configurations is preferrable.

Induced Draft

In the HVAC sector, dry cooler and air cooled condenser operated at moderate temperatures and thus by far the most common arrangement is to mount the fans above the heat exchanger and induce the air through the coil and then discharge the warmed air at high speed through the fan(s).

In this case the fan/motor arrangement is subject to the higher air leaving temperature but providing the temperature does not exceed, typically 60 to 70°C, then many motors can comfortably accommodate such conditions without affecting the motor or its bearings.

This arrangement also benefits from relatively high discharge air speed, which can aid the dispersion of the hot air stream into the surrounding stagnant ambient air and thus avoid air recirculation effects.

Similarly, for low temperature DX evaporators or brine coolers, high speed discharge air streams can be beneficial and provide decent ‘air throws’ to assist air circulation in cold stores etc.

As with all practically designed equipment, where space is a premium, induced draft arrangements; where the air is sucked through the coil and then accelerated through the smaller area of the fan orifice; can often suffer from ‘dough-nut’ shaped air distribution profiles with, on occasions, reverse airflow in the corners of the rectangular coil. Clearly such scenarios do not lend themselves to optimum thermal performance. So the product design is often a compromise between size against thermal performance.

Forced Draft

It has been argued that using the forced draft concept of blowing air through the coil, where the fan discharge plenum is pressurised, provides a more even air velocity distribution across the heat exchanger face area lending itself to better or more stable performance. However, the arrangement of the fans and the air volume delivered can often detrimentally affect this ideal concept.

Assuming that an induced and forced draft variant of a given fan delivers the same air volume … *which many manufacturers claim* … the product performance should be similar, but a forced draft fan/motor arrangement is subjected to the lower air inlet (ambient) air temperature, thus higher air density. This factor alone results in the fan delivering a greater air mass flow than for an induced scenario, because the air density at the fan is greater than when the fan handles the warmer air temperature and lower density of an induced draft arrangement. Remember … a fan is a constant volume machine … *see Fan Technology*

Furthermore, the air velocity leaving the product/coil is somewhat lower than for the comparable induced draft variant and this can be beneficial in medium temperature 'preparation rooms' for meats or flowers where the workers are not subjected to uncomfortable draughts.

In the case of high temperature process dry coolers, where the coolant temperatures may be above 100°C and thus the cooling air stream can be heated to temperatures well above acceptable 'Air Over Motor' (AOM) temperature for standard AC or EC motors, having the fan(s) only be subjected to the much lower ambient AOM temperature, can be a more practical and cost effective solution.

COIL CODE

The following describes the format of EAS's coil code, of which the 'grey' portion is usually only referred to, whilst the material and connection portion is included for completeness.

				Fin Height		*Fin Length*		*Rows*		*Circuits*		*Fin Pitch*		*Conn Pos*								
F	**V**	**A**	**-**	**1200**	**-**	**2500**	**-**	**06**	**-**	**010**	**-**	**2.5**	**-**	**A1**	**:**	**CU**	**/**	**AL**	**/**	**GALV**	**-**	**DN100**

Geometry
A - 60 x 30
B - 30 x 30
C - 40 x 35
Z - 25 x 22

Application
V - Air Heating
K - Air Cooling
C - Condenser
D - Evaporator - DX
S - Steam
G - Gas Cooler
P - Pump Circulated
F - Flooded

Tube Size
F - Ø15 mm
T - Ø10 mm (9.52)

Range
1.5
..
..
10.0

Pos.
A1
A2
A3
A4
A5
A6
A7
A8
B1
B3
B5
B6
B7
B8

Tube
CU
AL
SS304
SS316
Ti

Fin
AL
ALEP
ALMg
ALHy
CU
CUSN
SS304
SS316

Casework
GALV
ALZN
ALMG
SS304
SS3016

Conns
DN15
DN20
DN25
DN32
DN40
DN50
DN65
DN80
DN100
DN125
DN150

The coil code describes the geometry and fin block size but does not include any casework style details or overall dimensions. Such data would be included on an approved drawing which is the basis for manufacture.

PRODUCT CODE

All EAS's product variants e.g. flat-bed, vertical, V and H type dry coolers and air cooled condensers product designations conform to the following ...

- Fin pitch
- Number of circuits
- Type of product : D= Dry Cooler C= Condenser
- Unit position : H= Horizontal V= Vertical
- Air delivery : I = induced draft F= forced Draft
- Number of lines of fans
- Number of rows deep
- Number of fans
- Fan speed designator : ED/ES, HD/HS, SD, MD/MS, LD/LS, XD or EC
- Module Length : A = 500, B = 600, C = 700 .... Z = 2800
- Module width : F-Range : T = 720, S = 1080, M = 1200, L = 1440, W = 2160, X = 2400
  V-Range : M = 2x1200, L = 2x1440, W = 2x1800, X = 2x1920, Z = 2x2400
  VFW-Range : S = 2x1800, M = 2x2100, Z = 2x2400
- Fan manufacturer : 'Z' = Ziehl Abegg, 'E' = EBM, 'S' = Solar Palau, 'W' = WingFan
- Fan diameter : 050 = 500, 063 = 630, 071 = 710, 080 = 800, 091 = 910, 100 = 1000, 150 = 1500
- Fin geometry : A = 60x30-15, C = 40x35-15, D = 40x35-12, Z= 25x22-10
- Unit arrangement : F= Flat, V= Vee, FSW = Industrial F, VFW = Industrial V

MODULE WIDTHS

Module	Coil Finned Height - mm			
Width Code	F-range	FSW-range	V-range	VFW-range
T	720	-	-	-
S	1080	-	-	2 x 1800
M	1200	-	2 x 1200	2 x 2100
L	1440	-	2 x 1440	-
W	-	2160	2 x 1800	-
X	-	2400	2 x 1920	-
Z	-	-	2 x 2400	2 x 2400

MODULE LENGTHS

Module Length Code	Length mm
A	500
B	600
C	700
D	800
E	900
F	1000
G	1100
h	1200
J	1300
K	1400
L	1500
M	1600

Module Length Code	Length mm
N	1700
P	1800
Q	1900
R	2000
S	2100
T	2200
U	2300
V	2400
W	2500
X	2600
Y	2700
Z	2800

	# Poles	50 Hz Synchronous Speed	Typical Motor Speed	AC Motor Speed Designator	Delta/Star Designator
	2	3000	2800	~	~
	4	1500	1440	E	ED/ES
	6	1000	960	H	HD/HS
	8	750	710	M	MD/MS
Special	10	600	575	S	SD/-
	12	500	450	L	LD/LS
	16	375	350	X	XD/XS

Products fitted with EC motors use the generic motor speed designator EC.

However, squirrel cage IEC AC induction motors run at speeds somewhat less than their synchronous speed due to electrical and mechanical inefficiencies.

The synchronous speed is governed by the 'number of poles' of the motor and the supply frequency, either 50 or 60 Hz. For a 60 Hz supply the above speeds typically increase by 20%.

Historically in the Flex coil days, when only AC motors were readily available, products were offered with fans driven by AC motors which conformed to the above speed permutations.

The first letter E, H, M, S, L, X was allocated to 4 through 16 pole options.

The one speed option which is considered 'special' and applies only to FSW range, is the 10 pole motor variant.

Most AC motors ... but not all ... can be designed to provide 2 speeds from a given number of poles. The high speed when the 3 phase supply is connected in Delta and a Low speed when connected in Star. This is where the second letter designator ... D or S is derived from ... giving ED/ES or HD/HS etc. high/low speed options.

Such two speed (Delta/Star) AC motors are often available for AC compact fans from EBM, Ziehl or S&P. However, for fans fitted with WingFan impellers and IEC induction motors, the Star/Delta connection and speed options may not be available .. all depending upon the supply voltage, frequency and shaft power demands.

LEGACY PRODUCTS

FLEX COIL A/S

Flex coil a/s (*former trading name for Evapco Europe A/S referred to as EAS*) used an alternative designation system for their products ranges. Briefly, all dry cooler product codes began with 'V' (*first letter of the Danish word for water i.e. Vand*) and all condenser ranges began with 'C' (*first letter of Condenser*).

Some of the old product ranges that existed were the A, B, F, T, X, L, M, V, K, Q & Z and depending upon whether they were dry coolers or condensers became for example, VQ/CQ, VZ/CZ.

The latter product examples Q & Z were the current standard ranges in product prior to EAS's introduction of the new V & F-ranges and subsequent re-designation of all product variances.

These legacy products did not use multiple fan diameters or variable finned widths, or lengths as introduced in the current F & V-ranges, but the 'Q' signified the use of a Ø630 mm compact fan and coil module size of 1080 x 1300 mm, whilst the 'Z' signified the use of a Ø910 mm compact fan and module size of 1080 x 1875 mm. In both cases, a single or double line of fans could be accommodated up to a finned length of 12000 mm. Furthermore, these products were typically offered with a standard fin pitch of 2.1 mm.

Product	Application	Range	Airflow	Module Size	Std Fin Pitch	Impeller Ø	Fan/Motor
V - Dry Cooler / C - Condenser	Commercial Flatbed	**A**	Induced Draft	1080 x 1300	2.1	650	-
		B				660	Inverted Box MultiWing + Motor
		M				?	-
		T				627	MultiWing + Motor
		Q				630	Ziehl Compact
		X		1080 x 1875		895	MultiWing + Motor
		Z				910	Ziehl Compact
	Industrial Flatbed	**L**		2160 x 1850	2.3	1000	Inverted Box MultiWing + Motor
		SW				985	MultiWing + Motor
	Industrial Vee	**V**		2 x 1800 x 1220	2.1	1000	Inverted Box MultiWing + Motor
		F		2 x 1800 x 2000		1500	MultiWing + Motor
		FZ		2 x 2100 x 1200		800	Ziehl Compact
	Special	**IS**		2160 x 2300	2.1	1730	MultiWing + Motor
		IT					
		IV	Forced Draft	2 x 1800 x 2000			
Unit Cooler	-	**K**	Induced Draft	1 : 660 x 660 2 : 900 x 900	5.0	500	Ziehl / Nicotra Compact

EAS NEW DEVELOPMENTS

The further development of the CoilCalc product selection software during the 2010's and implementation of the 'real time' calculation of the operating air volume for any defined fan characteristic and coil pressure drop permutation, yielded the ability to optimise coil and fan combinations resulting in a list of suitable selections, from which the Engineers could choose the most cost effective, quietest, lowest hydraulic resistance etc. solution.

In principle, EAS can design any product variant with any fan module size incorporating any of the available coil geometries in any of the available tube & fin materials.

A fairly recent trend, fuelled by the recent energy crisis, is the popularity of localised heat pump installations for district heating applications. Such systems use ambient air as the 'low energy level' heat source and amplify if via the refrigeration system and plate heat exchangers, to provide hot water.

Energizers are required to extract the energy via the cooling of the ambient air.

In essence an Energizer is simply a 'reverse' dry cooler, where instead of heating up the ambient air to cool down the hot process fluid, the idea here is to cool down ambient air and extract the energy, whilst heating the water/glycol solution passing through the tubes.

The system described uses a single phase secondary circuit connected to typically, an evaporator plate heat exchanger.

A second type of Energizer uses CO_2 DX evaporating coils in the 'dry cooler', where again the air is cooled whilst the CO_2 is evaporated. This concept dispenses with the secondary single phase aspect of the system circuit and improves system efficiency.

A third variant is to use pump circulated ammonia (NH_3) as the evaporating refrigerant in the Energizer.

All three systems require some form of defrosting system, either hot gas or reverse cycle defrosting or the embedding of a hot glycol circuit into the heat exchanger to clear the frost or ice build-up.

GLOBALISATION : ECO-AIR PRODUCT LINE

Instigated in 2016 and finally launched during 2018, EVAPCO Inc. introduced the concept of the 'Full Spectrum' product portfolio, where the eco-Air range of dry cooler and air cooled condensers … including hybrid adiabatic variants … were introduced to complement their existing cooling tower and closed circuit cooler and condenser ranges. Now the product portfolio encompassed fully dry to fully wet alternatives to better fulfil the changing needs of the global customer base.

The eco-Air product line was fundamentally based upon a rationalised range of EAS's F & V product line including the precooling pad (EASiPad) and water spraying (EASiSpray) features to provide hybrid/adiabatic features to enhance the thermal performance. Furthermore, the sophisticated electronic control system developed by EAS was also adopted for 'Generation 1' of the eco-Air product line, albeit that migration to a more generic PLC is under development.

ORIGINS

In our industry, which has origins in the HVAC sector, a **heating coil** is classified on the basis that the 'air is being **heated**', whilst the typically hot water inside the tubes is cooled. Conversely, a **cooling coil** cools the air whilst the water in the tubes is heated. Thus, the definitions usually relate to the processing of the fluid in contact with the external finned surface of the heat exchanger i.e. the airside of the process.

However, in the 'process' industry sector the fluid stream that is under consideration is often what is inside the tubes and Process Engineers will often require a fluid to be cooled and thus in 'our' terminology, this requires a heating coil … the air is heated as the process fluid is cooled to the required conditions.

Furthermore, for example, EAS consider a dry cooler's fluid inlet temperature to be the higher of the two given temperatures, however, from the Process Engineer's perspective, his process inlet temperature is the dry cooler's fluid leaving temperature i.e. the lower of the two temperatures, because his process is heating the fluid to a higher temperature, whilst the outdoors dry cooler, cools the fluid back down to the design inlet temperature.

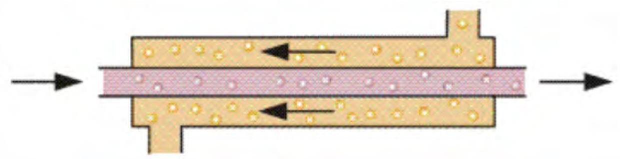
So, misinterpretation of provided data can often occur and care should be taken to understand the application in question.

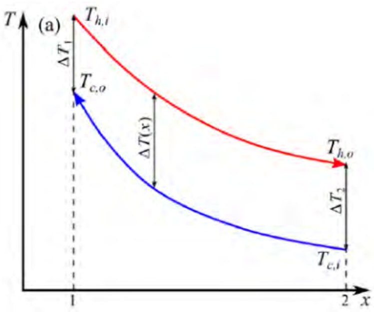
HEAT EXCHANGER TERMINOLOGY

In the heat exchanger or coil industry sector, similar to all business areas, there is terminology that is used that is specific to the industry. The following attempts to clarify the meanings of the most common terms used.

COUNTER FLOW

Counter flow or contra flow is the preferred arrangement to maximise the heat transfer from one fluid stream to the other.

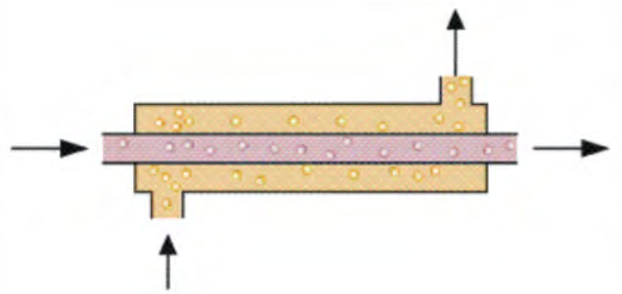


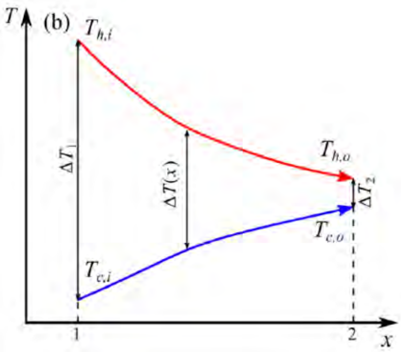


In this arrangement the hot and cold fluid streams flow in opposite directions, ensuring that the temperature difference between the streams is maximised along the length of the tube, leading to the best possible LMTD (log mean temperature difference).

PARALLEL FLOW

Parallel flow or co-current flow is less desirable and limits the potential heat transfer possible compared with the counter flow arrangement.



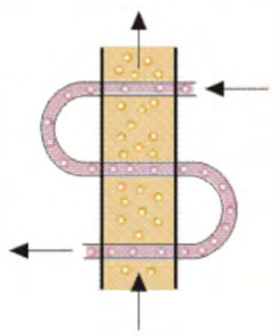


In this arrangement the hot and cold fluid streams flow in the same direction, which limits the heat transfer capabilities and, in some cases, can dramatically diminish the operating temperature difference and thermal performance potentially attainable.

On occasions, when the temperature differences are reasonably close, the hot fluid leaving temperature will limit the required cold fluid's leaving temperature resulting in a severe under performance.

CROSS-COUNTER FLOW

Both counter & parallel flow options typically relate to tube-in-tube, shell & tube heat exchangers, whilst tube and fin heat exchangers fall under the category of **cross flow** heat exchangers.



Furthermore, this heat exchanger division can be further sub-divided into cross-parallel and cross-counter flow.

The figure to the right shows the preferred cross-counter flow arrangement, where the tubes are perpendicular to the air stream and represent the cross flow aspect, whilst the tube circuitry is connected in such a manner that the tube fluid flow is in counter flow.

Clearly, the above figures depicting the temperature profiles for counter and parallel flow also relate to cross-parallel and cross-counter flow.

The most desirable configuration is thus the cross-counter flow arrangement.

OPERATING TD

The operating TD (temperature difference) for single phase fluid heaters or coolers is defined as the difference between the fluid inlet temperature and air inlet temperature, which equates to the maximum thermodynamic temperature difference available for heat transfer.

In the case of very large TDs a fluid coil or indeed a dry cooler's capacity is proportional to the TD, however, for typical operating temperatures the capacity is more closely approximated when the TD is evaluated as the difference between the 'mean fluid temperature' and air inlet temperature.

In the case of a condenser, the operating TD is the difference between the condensing temperature and the air inlet temperature. Generally, the capacity of the condenser is proportional to this TD.

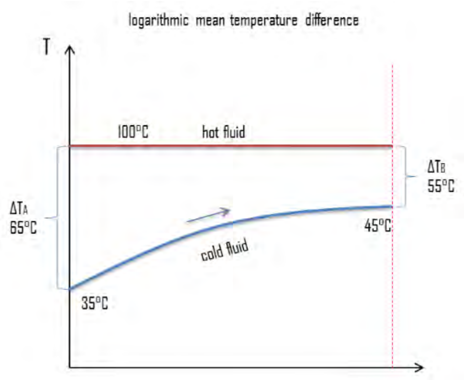
For condensers using zeotropic refrigerants (*refrigerants exhibiting a temperature glide*), consideration of the impact of the refrigerant pressure drop temperature penalty in addition to the temperature glide, may affect this proportionality behaviour.

For evaporators, the operating TD is the difference between the air inlet temperature and evaporating temperature and again, for small differences, the capacity is proportional to the TD.

However, care must be taken when using azeotropic refrigerants (*refrigerant with no glide*), because the refrigerant pressure drop temperature penalty can impact upon the available temperature difference available between the air leaving temperature and the refrigerant inlet temperature.

LOG MEAN TEMPERATURE DIFFERENCE - LMTD

The LMTD is a logarithmic average of the temperature difference between the hot and cold feeds at each end of the heat exchanger and governs the driving force for heat transfer.



When the heating and cooling process lines are not parallel or follow the tendency shown in the figure, then the mean temperature difference calculated from the arithmetic average does not correctly describe the 'real' average and thus the logarithmic method better represents the behaviour.

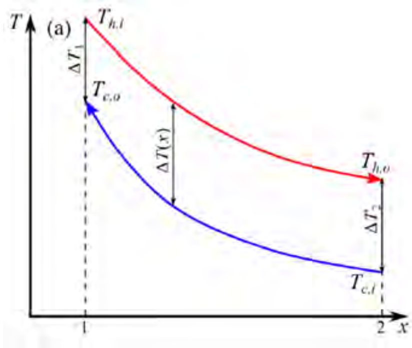
APPROACH TEMPERATURE

The approach temperature is defined as the smallest of the temperature differences between the two fluid streams.

In the figure to the right, this would be **Δ**T_1 … the hot fluid inlet temperature, $T_{h,i}$ minus the cold fluid leaving temperature, $T_{c,o}$.

However, it could be **Δ**T_2 if the cold fluid inlet temperature was somewhat higher or the hot fluid leaving temperature was lower, thus reducing the TD.

For practical purposes, it would be wise to design a heat exchanger with an approach temperature >= 2K. Although it is technically feasible to achieve a closer approach, the negative impact upon the LMTD and resulting increase in surface … and cost … is often not warranted.

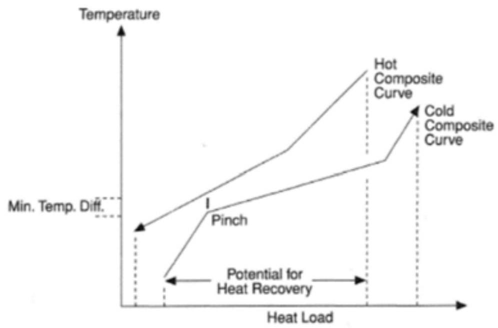


PINCH TEMPERATURE

Strictly speaking the pinch temperature difference, is defined as the smallest temperature difference between the two fluid streams.

Clearly, on occasions this may also be the approach temperature, but in a typical process shown on the right, there is a point where the two fluid stream temperature difference is the smallest and this is referred to as the pinch temperature.

The pinch temperature governs the 'real' performance capability of a given heat exchanger and as suggested above regarding the approach temperature definition, should practically not be less than 2K, without penalising the performance and surface requirements.



THERMAL SATURATION

Thermal saturation is not strictly a thermodynamic term but used by the author to describe a situation when the approach temperature, or indeed the pinch temperature, has reached its limit i.e. < 2K and the heat exchanger can no longer improve its thermal performance without something changing.

Take for example a dry cooler which is unable to reach the design capacity. A layperson's conclusion might be to increase the coil surface by adding more rows (surface) to the heat exchanger because it is not performing.

Investigation subsequently indicates that the air leaving temperature has reached the thermal saturation limit of within 2K of the hot fluid entering temperature.

Clearly, increasing the surface … rows deep … will not have any positive effect, in fact, it would only make things worse !!

The increase in the rows deep would increase the air resistance that the fan is working against and thus decrease the cooling air volume achievable from that fan. Now as the air temperature rise has already reached the maximum achievable, then a lower air volume along with the same air temperature rise will result in ***even less capacity****.*

The only solution to this problem is to increase the air volume, which would likely involve changing the impeller pitch angle, blade profile, number of blades or motor speed (if possible), all of which would increase the impeller absorbed kW, which may well overload the existing motor. So perhaps new motors would be needed.

Nevertheless, the increased air volume will reduce the air temperature rise – avoiding thermal saturation – improving the LMTD and provide the ability to increase the thermal performance without increasing the surface.

SURFACE MARGIN / SAFETY FACTOR

When performing a coil calculation it is often wise to provide some 'safety' in the calculation to allow for unknow circumstances that may detrimentally impact the thermal performance.

Installation implications causing non-ideal air flow and/or velocity distribution, operating conditions that may exceed the specified design conditions, internal or external fouling that may, over time, reduce the thermal performance etc. are all reasons to be cautious.

Clearly, imposing an additional **surface margin**, **surface reserve**, **design margin** or **safety factor** will increase the surface area necessary to meet the design capacity and will also increase the coil's cost, but at least the coil should perform.

The magnitude of the **surface margin** is often subjective and clearly should be a value greater than 1.0.

On occasions the Client or specification will request the inclusion of say, a 20% surface reserve implying the use of a **surface margin** of **1.2** resulting in an increase in coil surface are of +20% and thus 20% more rows deep than 'theoretically' calculated.

COMMERICAL FACTOR

The word **Commercial Factor** is often considered taboo and not discussed with End Users.

In today's world it is not commercially wise to design a coil with more surface than is necessary. If so, it is likely that a competitor will offer a solution that is slightly smaller (less surface) and thus cheaper, hence secure the order.

Many years ago, before the days of software to perform building load calculations or other methods for predicting capacity requirements, the tools available to estimate duty requirements were then 'bumped up' by as much as +20% to allow for 'unknowns'. Thus design duty requirements were often greater than were actually needed.

Often coil manufacturers underrated their equipment by up to -15% knowing full well that the design duty was +20% more than required and thus their undersized coils would still have 5% margin, compared with the design figures !!

However, during the 1980s & 90s, as the technical software industry began to evolve, capacity prediction software became more accurate and duty requirements were no longer inflated by 20% … but coil manufactures did not adjust their underrating commercial factors.

Furthermore, more 3rd party testing establishments sprang up around America & Europe and witness testing of equipment became more commonplace. This was followed by CTI & Eurovent which eventually led to coil manufactures being exposed as 'massaging their data.

So today the use of commercial factors is frowned upon and liable to result in warranty cases claiming under performance.

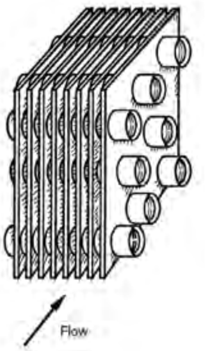
Coils are designed with software (CoilCalc) that use verified thermodynamic data in accordance with perhaps American or European performance testing standards e.g. EN1216. However, like all Standards, conditions are ideal and thus provide 'best case' predictions to advantage the manufacturer in question.

Knowing the 'real' performance and thus minimum required surface area necessary to meet a given duty requirement, the application of a 'hidden' commercial factor to slightly underpredict the necessary surface (rows), can provide a slight 'commercial edge' and 'tilt the table' towards securing an order … if you don't get caught with an underperforming coil !!

Coils can be mounted vertically with a horizontal air flow, horizontally with a vertical air flow or on an angle such as coils fitted to Vee type products or perhaps wall mounted fan coil units.

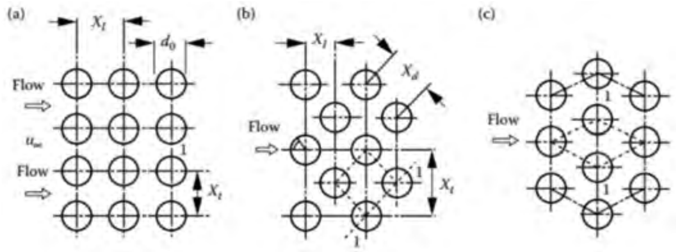
Consequently, terms such as finned height may appear oblique when the coil is mounted horizontally ... it may appear more logical to refer to this dimension as the finned width.

However, for the purpose of the following terminology, we shall assume that the coil is vertical … standing upright ... with a horizontal air flow. So our example of finned height is indeed the vertical distance of the fin pack.



GEOMETRY OR TUBE PATTERN

Refers to the arrangement of the tubes and the matching holes in the fins.



Typically, the tubes can be arranged ‘in-line’ with one another (**a**) or ‘offset’. This latter option can either be ‘staggered’ (**b**) or an ‘equilateral’ pattern (**c**).

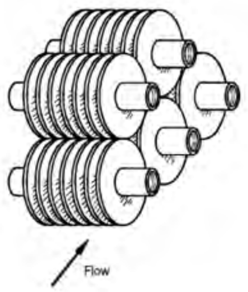
The offset (staggered) arrangement is invariably preferred due to the improved external heat transfer coefficient compared with in-line alternatives. However, for lower temperature cooling applications where frosting and ice accumulation may become an issue, the in-line geometry option may be favoured to increase the time between defrosting.

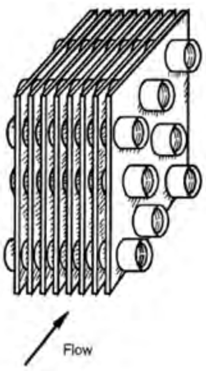
A staggered geometry can be configured into a number of formats, some of which perform better than others, but in reality, the tubes have to be interconnected using return bends to form the required circuitry pattern to achieve cross-counter flow and even a ‘free draining’ attribute. So, minimising the number of return bend sizes results in two offset patterns …

- Staggered : Tube pitch, X_t = 2 x Row pitch, X_l
  - A-fin : 60 x 30 mm requiring return bends with 60 mm & 42.43 mm centres
- Equilateral : Tube pitch, X_t = Diagonal pitch, X_d
  - C-fin : 40 mm equilateral requiring return bends with 40 mm & 69.28 mm centres
  - Z-fin : 25 mm equilateral requiring return bends with 25 mm & 43.3 mm centres

PRIMARY & SECONDARY SURFACE

The heat transfer surface is the surface area of the exchanger core that is in direct contact with the fluids and through which heat is transferred by conduction. That portion of the surface that is in direct contact with both the hot and cold fluids and transfers heat between them is referred to as the **primary** or direct surface. To increase the heat transfer area, appendages (fins) may be intimately connected to the primary surface to provide an extended, **secondary** or indirect surface.



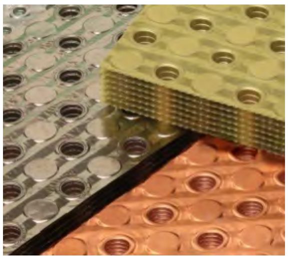


In our industry, these extended surface elements are referred to as fins. Thus, heat is conducted through the fin material and convected (and/or radiated) from the fin (through the surface area) to the surrounding fluid, or vice versa, depending upon whether the fin is being cooled or heated.

Consequently, the addition of fins to the primary surface reduces the thermal resistance on that side of the process and thereby increases the total heat transfer from the surface for the same temperature difference.

These secondary surfaces or fins may also be introduced primarily for structural strength purposes or to provide thorough mixing of a highly viscous liquid.

FINS

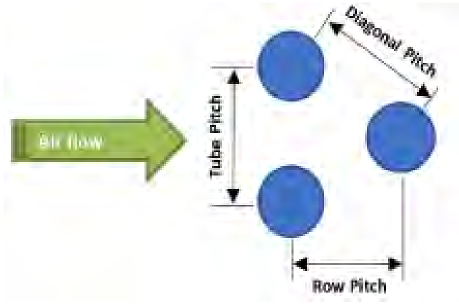


Fins refer to the extended 'secondary' surface surrounding the 'primary' tube surface and is manufactured from level wound rolls/coils of 'fin strip' or 'foil' in a variety of materials conforming to the shape and pattern in the previous section.

Typically aluminium is used but hydrophilic and hydrophobic coated alternatives are available along with aluminium magnesium, copper, copper tinned and stainless 304/316. Technically titanium is an option, however, this is only considered for extremely hostile environments.

TUBE PITCH

Refers to the centre-to-centre distance between adjacent tubes perpendicular to the air flow direction. Tube pitch is also refered to as the *Transverse Pitch*.



ROW PITCH

Refers to the centre-to-centre distance between adjacent tubes parallel to the air flow direction. Other references to this parameter are *Lateral Pitch, Longitudinal Pitch* or the *Tube Stagger*.

DIAGONAL PITCH

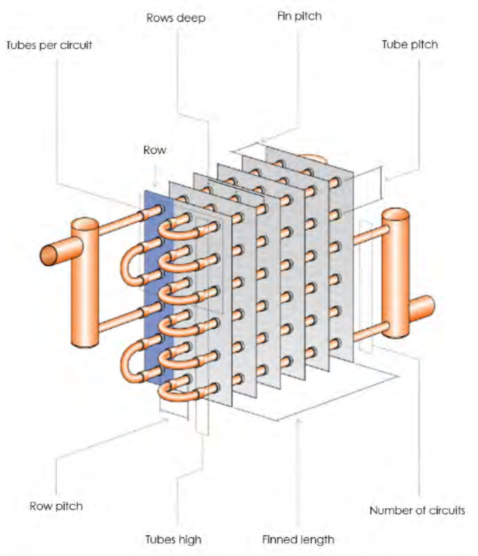
Refers to the diagonal centre-to-centre distance between adjacent tubes in sequential rows and only relates to staggered tube patterns.

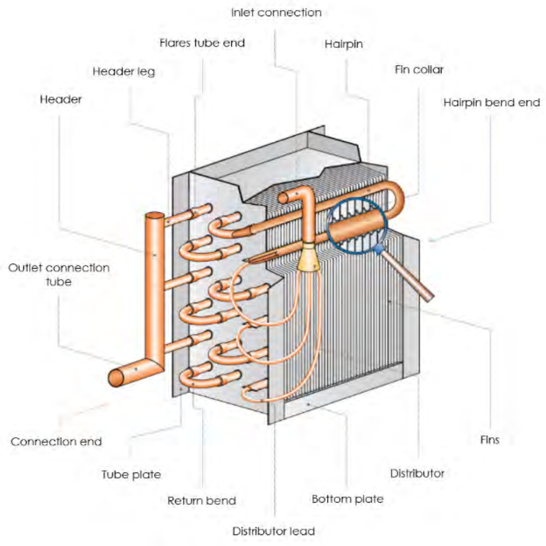
COIL BLOCK / MATRIX BLOCK

Refers to the finished assembly comprising the expanded tubes and fins and casework assembly if applicable.

The Coil Block may be referred to as the *Fin block*, *Coil matrix* or *Heat Exchanger Core*.

The following exploded views highlight the component parts of typical coils …





TUBES HIGH

Refers to the number of horizontal tubes in the height of the coil in each row. As a consequence of the tube geometries/patterns we use, the number of tubes in each row is the same.

However, competitor geometries can have a different number of tubes in alternate rows, depending upon how the fins are punched and slit.

FINNED HEIGHT

The finned height of a coil is governed by the number of tubes in the height and the tube pitch related to the geometry selected.

The vertical distance over the fins is referred to as the …

Finned Height = Tubes High x Tube Pitch

For example, a C-fin geometry with 20 tubes high, which uses a 40 mm equilateral pattern, would have a finned height of 20 x 40 = 800 mm.

The finned height will always be a multiple of the tube pitch dimension.

Clearly, for a horizontally mounted dry cooler, utilising this same coil, the distance over the finned height may be referred to as the ***Finned Width***.

FINNED LENGTH

Refers to the distance over the finned area of the coil block along the length of the tubes.

To confuse matters further, a vertical coil with a horizontal air flow, perhaps for mounting in a ducting system, may have its finned length referred to as the *Duct Width, Coil Width* or *Finned Width.*

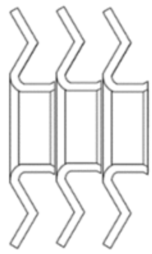
FIN DEPTH

Refers to the depth of the coil block in the direction of the air flow and is calculated by …

Fin Depth = Number of Rows x Row Pitch

For example, a C-fin geometry which is 4 rows deep uses a 40 mm equilateral pattern with a row pitch of 34.64 mm, would have a fin depth of 4 x 34.64 = 138.56 mm

FIN PITCH



Refers to the distance at which the extended surface/fins are spaced and is EAS's preferred terminology relating to the positioning of the fins.

Fin pitch is defined as the distance between the centrelines of two adjacent fins and usually presented in millimetres e.g. 2.0 mm or 3.2 mm.

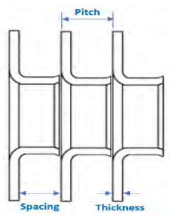
On occasions the fin pitch is presented in terms of the *fins per meter (FPM)* which translates into a fin pitch by taking the reciprocal of the FPM x 1000 …

Fin Pitch = 1000 / FPM

The old Imperial method of referring to the fin pitch is referred to as ***FPI*** or ***Fins per Inch*** and can be converted into FPM by FPI x 1000 / 25.4 or into the Fin Pitch …

Fin Pitch = 25.4 / FPI

FIN SPACING



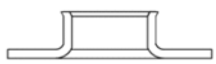
Refers to the gap or spacing in between two adjacent fins.

Thus,

Fin Spacing = Fin Pitch – Fin Thickness

The term 'Fin spacing' is often misused when in fact, the Fin Pitch is being referred to.

COLLAR HEIGHT



Refers to the height of the extruded material which forms the 'collar', which will both be in contact with the tube and create the fin spacing (gap between the fins), when the fin pack is stacked prior to the tube expansion.

Depending upon the design of the fin die and whether the underside of the fin collar has a recess to accommodate the mating fin's flared collar when the fins are stacked, the collar height may equal the fin pitch.

However, the collar is adjusted to ensure that the desired fin pitch is achieved.

ROWS DEEP

Refers to the *Number of Rows* of tubes in the direction of the air flow.

HANDING – INLET/OUTLET POSITIONS

EAS use a letter followed by a number to indicate where the coil connections are located with respect to the air flow direction.

So for example **A1** indicates that the coil is vertically mounted (horizontal air flow) and both the inlet & outlet connections are positioned on the right hand side .. looking at the coil in the direction of the air flow.

A2 is similar to A1 but both the inlet & outlet connections are positioned on the left hand side of the coil.

A3 & A4 indicate that the connections are on opposite ends of the coil, A3 suggesting the inlet is on the RHS, whilst A4 indicates a LHS inlet.

EAS, like many competitors, have a standard for where the inlet connection is physically mounted in the inlet header and likewise for the outlet header. The premise is that the inlet is located at the bottom of the inlet header, to assist forcing the air out of the coil as it is filled … usually from the inlet connection. This infers that the outlet connection will be positioned at the top of the outlet header.

However, Evapco Inc. have chosen an alternative arrangement, where they prefer to locate the inlet connection in the top of the inlet header and thus the outlet connection at the bottom of outlet header.

The rationale for this is to provide some continuity with their air cooler condenser variants of their eco-Air product line. Clearly, all condensers have their hot gas inlet either at the top of the inlet heard or centrally mounted. The condensate drain connection must be located at the very bottom of the outlet header.

Consequently, the Handing designations were extended to allow for this reversal of the connection positions and avoid confusion. Therefore, **A5** & **A6** are equivalent to A1 & A2, but with the inlet connection at the top and outlet at the bottom of the headers.

For a fuller explanation and inclusion of horizontally mounted coils and their handings, refer to the section below : Coil Orientation Designations

SAME / OPPOSITE END CONNECTIONS

Generally, both the inlet and outlet connections supplied with a coil are mounted on the same end/side of the coil. This arrangement lends itself to most installations and applications, for example Air Handling Units (AHU) or duct mounted coils, where positioning the connections at the same end, eases installation, coil removal and maintenance upon the coil.

An exception is steam coils, where the EAS preferred vertical tube orientation results in the steam inlet connection being located at the top of the coil, whilst the condensate outlet connection is placed at the lowest position to assist the condensate drainage from the coil.

On occasions the design requirement and/or location of the coil necessitates opposite end connections, usually associated with large products, perhaps 12 meters long, which requires a low fluid pressure drop, then 3 passes may meet the design limit where 4 passes would exceed the allowable pressure drop.

FACE AREA

The face area of a coil is defined as …

FA = H / 1000 x L / 1000 *where,* *FA = face area, m^2*
H = finned height, mm
L = finned length, mm

Often this is the same cross-sectional area as the ducting into which the coil will be mounted. However, in the case of an air handling unit; and depending upon the AHU design concept; the coil finned size may indeed match the internal dimensions of the AHU or may be somewhat smaller if the coil slides into the AHU coil housing.

FACE VELOCITY

The face velocity is a term used to indicate the nominal air speed/velocity at the coil inlet or through the coil and is defined as …

$$\mathbf{FV = \dot{V} / FA}$$

where, *FV = face velocity, m/s*

$\dot{V}$ *= air volume, m³/s*

FA = face area, m²

The entering face velocity for a coil is in fact the average approach velocity to the coil calculated from the inlet air volume, which will be different from the leaving face velocity … if the temperatures are different. *This also assumes that the velocity profile across the duct/coil is uniform, which is often far from reality.*

This is justified on the basis that the coil experiences a constant mass flow rate. Therefore, if there is a difference in temperature between the inlet and outlet conditions; *which is clearly the case with a heat exchanger, otherwise there is no purpose for the coil*; the air density will change and thus the air volume will change.

It can be argued that face velocities should be calculated from the average air volume based upon the average air temperature across the coil, however often, 'standard air' volumes are considered and thus the face velocity is an indication of air velocity through the ducting/coil/system.

External heat transfer coefficients are often presented against face velocities for convenience purposes, but in reality, the Reynolds number would be a more accurate parameter.

Similarly, coil air pressure drop characteristics are often plotted against the face velocity, which is adequate for most purposes. But again, Reynolds number would be more correct.

INTERSTITIAL VELOCITY

Although the face velocity is a good indication of the air velocity through a coil … and is simple to calculate … the air speed 'inside' the tube and fin matrix/tube bundle is not constant.

As the air passes through the fin block the air is forced to divide into multiple air streams as it passes between the fins and over the tubes and then again, as it enters the diagonal pitch. Thus, there is a sequential acceleration and deacceleration of the air speed as the multiple air streams re-join and divide once again.

The resultant uneven velocity, known as the interstitial velocity, assists in promoting induced turbulence and thus a higher airside heat transfer coefficient, had the velocity remained constant.

Usually, the heat transfer correlation uses the highest interstitial velocity as a basis for predicting the heat transfer coefficient.

ASPECT RATIO

On occasions the size of the coil is not given and only the air volume and duty or perhaps temperatures provided. Under such circumstances the finned height and length need to be determined to begin the calculation process.

It is safe to say that a 'square' coil is more expensive to manufacture than a 'rectangular' coil with the same face area, due to the greater number of tubes in the finned height and thus the number of return bends, header length and number of brazed or welded joints.

Therefore, a 3 : 2 (or 1.5 : 1) aspect ratio is a reasonable starting point to begin the design exercise.

Furthermore, reasonable design face velocities of 3.0 m/sec for heating coils and 2.5 m/sec for cooling coils are again a good starting point.

Based upon these assumptions, the appropriate finned height can be calculated, which when rounded to the nearest number of 'physical' tubes high for the geometry chosen, will provide the design fin height. Thereafter, the finned length can be calculated from the design face velocity chosen.

- Application – heating coil
- Air volume is given as $\dot{\mathbf{V}}$ m^3/s
- Assumed aspect ratio - 1.5 : 1. **Ratio = 1.5**
- Assume face velocity **FV** = 3.0 m/s
- Face area, **FA =** $\dot{\mathbf{V}}$ **/ FV** , m^2
- Let **X = (FA / Ratio)$^{0.5}$ x 1000** , mm
- Choose the geometry to be used, which will define the Tube Pitch, **TP** i.e. A – 60 mm, C – 40 mm, Z – 25 mm
- Tubes high, **T = Integer(X / TP)** … *rounded to the nearest tubes high based upon tube pitch for the geometry selected*
- Finned height, **H = T x TP** , mm
- Finned length, **L = FA / H x 10^6** , mm

Now, the finned height and length are known, which provides the assumed 3.0 m/s face velocity with the air volume initially given … problem solved !

Example

Let's assume we have a duct mounted cooling coil application where we know the air volume is 6000 m^3/h and thus to avoid water carry-over issue we shall impose a maximum face velocity of 2.3 m/s. So what is the required fin height & length ?

Now, 6000 m^3/h equates to 6000 / 3600 = 1.67 m^3/s

The required coil face area to meet the desired 2.3 m/s face velocity is thus 1.67 / 2.3 = 0.724 m^2

Assuming an aspect ratio of 1.5 the estimated coil height, X = (0.724 / 1.5)$^{0.5}$ x 1000 = 695 mm

A coil of this size would generally be serviced by utilising our Z-fin, where the tube pitch, TP is 25 mm, so the actual fin height must be based upon the number of tubes that will fit this physical fin height.

So, Integer(695 / 25) = Integer(27.8) = 27 tubes high and thus the fin height, H = 27 x 25 mm = **675 mm**

Finally, the finned length = 0.724 / 675 / 10^6 = **1072 mm**

Clearly, one may wish to adjust the fin length to say 1100 or 1075 mm or it may be concluded that 27 tubes high is not a 'good' number regarding circuitry, especially if this cooling application required a DX evaporator. Thus perhaps 28 tubes in the height might be preferable. If so, the actual fin height should be recalculated and then the associated fin length recalculated to ensure that the 2.3 m/s face velocity is not breached.

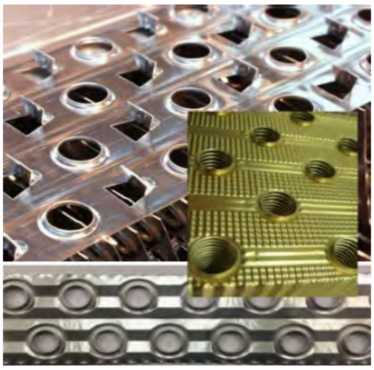
ENHANCED TUBES

Traditionally tubes are circular with a smooth internal surface, however, especially for refrigeration evaporator or condenser applications (two phase processes), internal heat transfer coefficient improvements are gained using internally enhanced/turbulated tubes.

Such internally enhanced tubes employ a riffled or grooved internal surface that tends to accelerate the onset of nucleate boiling or condensation. However, some of the advantages are diminished by the increased pressure drop implications, which have a negative impact upon the performance.

For single phase fluid applications such as oil coolers, internally enhanced tube walls provide little or no advantage and thus 'turbulator inserts' can be used, such as a 'twisted tape' or corrugated extrusions, which help to disturb the stream-lined laminar flow characteristics associated with the high viscosity fluids, which in turn boost the often low heat transfer coefficients.

TURBULATED FINS



The secondary surface indicated above shows simple 'flat fins', however such fins can have a variety of shapes and profiles which promotes air turbulence within the tube bundle and results in improved heat transfer.

Turbulated fin surfaces have diverse designs which are often referred to as – corrugated, rippled, wavy, sinusoidal, louvres, slit or flaps.

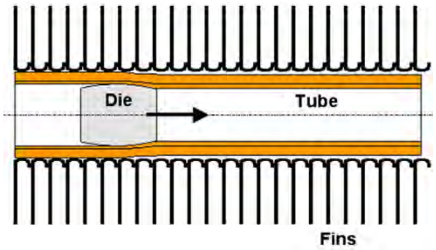
The use of turbulators is always a compromise, yes, they improve performance or reduce the surface requirements, but they increase the airside pressure drop, which may have detrimental effects upon the air volume that a fan can deliver.

Furthermore, such surfaces trap dust, grime and debris which degrade the surface performance and can further increase the airside pressure drop.

EXPANSION

To ensure a good thermodynamic contact between the tube and fin material; after the tubes are inserted into the fin bundle through the extruded fin collars; the tubes need to be expanded to increase their outside diameter to provide an 'interference fit' between the tube and fins.

Expansion can be achieved in a number of ways …



- Mechanical expansion – using steel carbide 'bullets' whose outside diameter is greater than the inside diameter of the tube.
  These bullets are forced through the tube to plastically deform it to a larger diameter. In doing so the fin collar is elastically deformed providing an excellent contact and minimal 'bond resistance'.
  For EAS's Ø15 mm tube, the finished expanded diameter is 15.65 mm and for Ø3/8" tube, 10.15 mm.
- Hydraulic expansion – this involves using hydraulic water pressure inside sealed tubes to expand the outside diameter to achieve the contact between the tube and fin. This method does not offer the same level of predictability and consistency of finished expanded outside diameter, which may be considered as a disadvantage.
- Pressurised bladder expansion – used for high volume production where length restrained rubber bladders are inserted into the tubes and pressurised, expanding the tubes.

The method of expansion employed is dictated by the machinery chosen for manufacture, however the results are similar.

HOW TO CONSIDER THE PLASTICALLY DEFORMED TUBE MATERIAL

When a material is plastically deformed its properties change, so how should the material be considered for strength calculation purposes?

- Expanding the tube plastically (past yield, $Rp_{0.2}$) imposes a finite hoop strain: the material work-hardens (dislocation density rises), raising yield strength and often increasing R_m (UTS) slightly
- Cold work reduces uniform and total elongation (A%), and reduces fracture toughness and ductility-dependent behaviour (necking, ductile tearing)
- Residual stresses are produced and these can influence fatigue and stress-corrosion cracking behaviour
- If expansion produces no wall thinning, the cross-sectional area for static stress is unchanged - but the material's stress-strain curve changes
- Conservatively use the original annealed values for strength calculations

To estimate the degree of cold working, check the imposed hoop strain …

Original OD = 15.00 mm
Expanded OD = 15.65 mm
Engineering hoop strain ≈ $\Delta D / D = (15.65 - 15.00) / 15.00 = 0.0433 = 4.33\%$

Converting to the true (logarithmic) hoop strain:

$\varepsilon\_true = \ln(1 + \varepsilon\_eng) = \ln(1.0433) = 0.0424 = 4.24\%$

A plastic hoop strain of ~4.2% is substantial (many yield points are <0.5%) indicating plastic strain and significant strain hardening.

FLARED/SWAGED HOLES

There are several methods of coil block/matrix block/fin pack construction and the one favoured by EAS is a semi-floating design where the tube plates and divider plates are punched with formed/flared/swaged holes to avoid the tubes from bearing their weight on plain cut (sharp) edges.

These flared holes offer a parallel sided portion of the swage which ensures that the tube is not in contact with any sharp edges. Furthermore, the expanded O.D. of the tube is slightly smaller than the I.D. of the swaged hole. This allows the expanded tube and fin bundle to move within the tube and divider plates when subject to thermal expansion.

Generally, most of the tube plate holes are swaged. However, large clearance holes are punched, in place of the swaged hole, when tube holes are located close to a folded edge. Again, these clearance holes are sufficiently large in diameter so that there is no possibility of contact with the tube material.

Alternatively, there is a school of thought that advocates expanding the tubes into 'clusters' of swaged holes, whilst all other holes are over-sized clearance holes. The theory here is that these 'clusters' support the tube and fin bundle, whilst the clearance holes allow for tube movement resulting from thermal expansion.

One final difference of opinion relating to the assembly of the coil component parts is the orientation of the swaged holes.

So the $64000 question is 'Should the swaged holes protrude outwards (as shown above) at both ends of the coil or should they be orientated to assist the 'tubing up' process and thus point inwards at the end where the tubes are inserted and outwards where the tubes break through the last tube plate' ?

EAS favours the latter option, especially for copper tube coils, since there is a propensity for the outward pointing flared hole to 'grab' the tube and cause scoring along the tube length … a potential for future failure.

NUMBER OF CIRCUITS

The Number of Circuits is the number of separate flow paths that the incoming volumetric flow rate is divided into to ensure that the design tube velocity is achieved, and the required hydraulic resistance (*pressure drop*) is not exceeded.

NUMBER OF TUBES PER CIRCUIT

Also referred to as the ***Number of Passes***, is defined as …

Number of Tubes per Circuit (No. Passes) = Tubes High x Rows Deep / Number of Circuits

For two phase refrigerant condensers and evaporators, the N*umber of Tubes per Circuit* should generally be the same for each circuit. However, for single phase fluid coils, having a mixture of circuits with a different number of passes is allowable, albeit that the aim is to equalise the number of passes in each circuit.

As an example, a coil with 16 tubes high and 4 rows deep with 8 circuits equates to 8 tubes per circuit in each circuit. But if the design required 10 circuits to perhaps keep the fluid pressure drop below the Client's limit, then 16 x 4 / 10 = 6.4 tubes per circuit in a coil with a total of 64 tubes.

In practice, the coil would be circuited (*interconnection of the tubes*) such that it would have 8 circuits, each with 6 tubes and 2 circuits, each with 8 tubes giving (8 x 6T) + (2 x 8T) = 64 … where all the tubes are used.

NUMBER OF PASSES

Refer to the above definition for the ***Number of Tubes per Circuit***.

Traditionally, the industry has been split in terms of the usage of 'passes' or 'circuits'. Some manufacturers have chosen to use the '# of circuits' in their coil codes, whilst others have chosen to use '# of passes'.

Both terminologies can result in the same circuitry pattern if the total number of tubes in the coil divided by the # of passes is a whole number. If not, then there is a degree of 'guesswork' without reference to the 'as built' circuitry pattern.

In the above example where a coil has 10 circuits and the # of passes is thus theoretically 6.4, then more detailed information is required to clarify the mix of 6 pass and 8 pass circuitry.

Generally, EAS prefers to use the terminology 'number of circuits' and 'tubes per circuit' and the number of circuits is a part of the EAS coil code.

BLANK TUBES

Defines the tubes in a coil block that are *Unused, Blank or Blind* and not included in the circuitry, usually because the total number of tubes in the coil block (*Tubes High x Rows*) is an odd number and the normal requirement for '*same end connections*' demands an 'even' number of tubes in each circuit.

For example, a coil with 13 tubes high and 5 rows deep with 10 circuits results in 6.5 tubes per circuit/passes. If 'same end connections' are required, only 64 of the 65 tubes can be utilised in the circuitry, so there will be one Blank tube, resulting in (8 x 6T) + (2 x 8T) + 1 Blank.

Generally, when mixed circuitry is allowable, there will only ever be a maximum of 1 Blank is any coil whether it has same or opposite end connections.

NUMBER OF SECTIONS

A coil can be divided into more than one section where each may have its own inlet and outlet connections, which again may be of similar sizes or may be different. Furthermore, the sections of the coil may be of difference sizes.

In some industry sectors, what we know as a 'section' can be referred to as a 'circuit' … especially in the refrigeration industry sector, where a refrigeration circuit infers a complete system comprising the compressor, condenser, evaporator and expansion device. So when a coil serves two or more independent refrigeration systems, we need to split the coil into discreet sections, each handling an individual system.

In our terminology, one 'circuit' can comprise a single tube e.g. inlet and outlet at opposite ends of the coil, such as a vertical tube steam coil or a number of interconnected tubes, say 4 tubes per circuit … also referred to by Evapco Inc. as 4 **passes**.

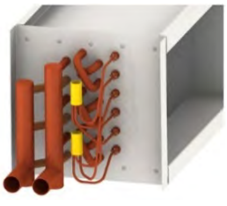
An example of this is a DX evaporator required to deliver 33%, 67% & 100% capacity. This could be achieved by splitting the coil into 3 equal sections. However, this scenario can be achieved by only using two sections, the first providing 33% and the second section providing 67% capacity. Thus with the appropriate valves and pipework design either 33% or 67% capacity is reach with either section activated, whilst when both sections are active, 100% capacity is met.

Coil sections can be arranged in different ways to fulfil functionality requirements …

- Split in face

- Split in rows deep

- Interlaced



Fluid heating and cooling coils typically use the first two methods, whilst DX evaporating cooling coils would often use either split in face or interlaced sections, the latter to provide a more predictable performance behaviour.

See DX Evaporator section for further details

NUMBER OF SETS OF CONNECTIONS

Usually, the volume flow rate of the fluid apportioned to each section dictates the physical size/diameter of the required fitting. This in turn, dictates the diameter of the connection tube and thus header pipe diameter.

If the resultant size/diameter of the required header pipe is too large to be accommodated withing the space available, one option is to use two or more sets of connections on a common header pipe which allows the header pipe diameter to be reduced.

For example, a coil requiring 1 x 4" connection and matching 4" header pipe can also be fulfilled by using a 2½" header pipe fitted with 2 x 2½" connection fittings …

1 section : 1 x 4" inlet/outlet

1 section : 2 x 2½" inlet/outlet

Both coils might be fed with the same volume flow rate, but the coil to the right with the 2 sets of connections allows a smaller header pipe of 2½" to be used because each connection is only handling 50% of the total flow rate. If sized correctly, the overall fluid pressure drop for both scenarios will be similar.

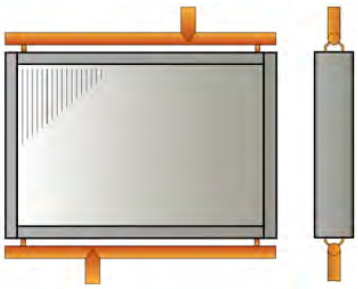
A variation to the above single section with two sets of connections is to split the coil into 2 discreet sections, each with its own inlet & outlet connection …

2 sections : 2 x 2½" inlet/outlet

This coil solution would behave and function identically to the above coil with 2 sets of connections on a common header. However, from a PED perspective, now we have a coil with 2 discreet sections and thus half the internal volume per section. Therefore, depending upon the maximum allowable pressure (PS), maximum allowable temperature (TS), Fluid Group and application of the coil, will affect the PED Category calculation. Usually in such cases, the category will be lower and may even fall within the scope of a PED § Article 4/3 categorisation.

The coils discussed so far are typical fluid heating and cooling coils fitted with same sized inlet and outlet header pipe/connections and to a lesser extent, refrigerant condenser coils, albeit that the latter variant would typically have a smaller condensate outlet header/connection size than the larger inlet hot gas header/connection.

STEAM COILS



1 section : 1 inlet/outlet

Steam coils are an oddity in so far that typically they are designed with vertical tubes and opposite end connections and thus the headers, instead of being vertical, are horizontally mounted at the top and bottom of the coil. In essence the coil is turned through 90° compared with other types of coils.

Other variants of steam coils are horizontal tube options using either hairpins or alternatively a 'transfer header' to provide a 'same end connection' solution. Additionally, another opposite end connection solution with horizontal tubes can be achieved by mounting the fin block on an incline inside the casework. These horizontal tube steam coil solutions are usually avoided.

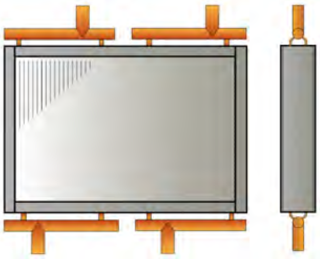
Steam coils are in essence condensers, however, due to the nature of their application involving the condensing of dry saturated steam into a saturated liquid (condensate), often operated cyclically (On/Off), they lend themselves to a vertical tube arrangement.

The above sketch of a 2 row, single section steam coil shows horizontal headers with the inlet & outlet connection tubes mounted as some location along the header. This is not necessarily typical and the header(s) can be extended horizontally to provide the connections. Thus, the steam enters the end of the header horizontally and may exit in a similar manner at the bottom of the coil. So the connection orientation is dictated by the Client's requirements.

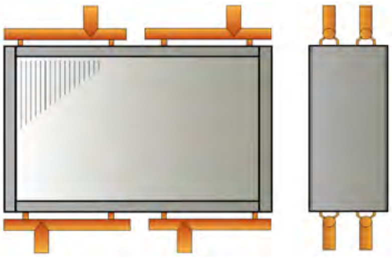
Related to the above comments, we impose restrictions on the maximum header pipe length and also the number of rows that a header can distribute steam to …

- Maximum finned height/header length : 1500 mm
- Maximum rows per header : 2 rows

These restrictions will automatically require the coil to be split into discreet sections, which favourably influences the discreet section PED categorisation.



As an example, a 4 row deep vertical steam coil with a finned height of 2500 mm, thus header length > 1500 mm, will result in 4 sections. At least 2 sections are required with a header length of 1250 mm and 2 sections in the depth because only 2 rows can be fed by a single header. Thus a total of 4 sections.



In the case of an odd number of rows e.g. 5 rows, then if the header length is <1500 mm the coil would have 3 inlet headers and 3 outlet headers … one header to serve rows 1 & 2, a second header to serve rows 3 & 4 and a third header serving row 5.

This arrangement of multiple headers/connections is often considered a 'plumbers nightmare' and may affect whether the Client wants to accept our solution, but our experience with especially steam pressures above 3 barg, have prompted us to invoke the header length and rows per header restrictions.

The reason for imposing a maximum of 2 rows feeding each header stems from ensuring that the header legs experience similar thermal related stresses.

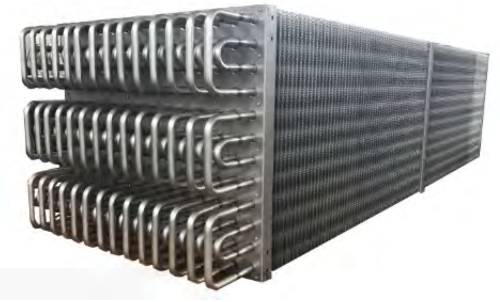
A one row coil will use identical length straight legs and a 2 row, two similar length bent legs, whilst a 3 row would need to accommodate two bent and one straight leg. In such a case, for a given temperature rise, the straight leg would linearly expand slightly less than the slightly longer bent legs, thus imposing uneven stresses upon the tube-to-tube but more importantly, the tube-to-header welded joints. Over time and subject to the typical ON/OFF cyclic mode of operation of steam coils, fatigue can become an issue, resulting in stress fractures and/or pin hole leaks.

RETURN BENDS

Return bends are used to interconnect the tubes in the coil block to form the circuits in line with the circuitry pattern. The sometimes complex circuitry patterns are either generated by software or drawn by hand.

Return bends with different centre-to-centre dimensions are available for each tube pattern and tube material/thickness to accommodate the required pattern.

CROSSOVERS



On occasions, special circuitry patterns call for bends that perhaps span more than 3 rows and must be specially manufactured.

Crossovers are often used on pump circulated evaporators where the circuitry aligns with the rows deep resulting in a vertically up-feed arrangement. When the application requires a 'double serpentine' interconnection, then multi-layered crossovers can be used.

TRIPOD BENDS

Although rarely used by EAS, tripods are a simple means to increase or decrease the number of tubes in a circuit, perhaps halfway through the circuit path.

Such composite bends can be used in condenser applications when there is a need; during the latter stages of condensation; to increase the condensate velocity inside the tube by merging two circuits into a single circuit.

As the refrigerant vapour condenses the liquid portion occupies a much smaller proportion of the tube and the velocity reduces. The lower liquid velocity reduces the internal heat transfer coefficient, so if the velocity can be increased; by reducing the number of circuits; then the improved velocity and heat transfer coefficient can improve the efficiency of the coil.

HAIRPINS

Refers to a straight length of tube which has been bent 180° to form a 'hairpin', that is in effect, two straight lengths with an integral return bend.

Hairpins minimise the number of return bends required to be brazed or welded to an otherwise 'straight tube' coil block. Their use also reduces the labour time associated with 'tubing up' or the 'lacing' of the coil block.

HEADERS

If a coil has more than one circuit per section, then header sub-assemblies are manufactured comprising larger diameter pipes that are sealed with end caps and accommodate one or more connections tubes (*see Header Legs*) that serve as the fluid inlet or outlet connection.

The header assembly in the figure shows a horizontally mountable fabrication with header legs plus two capped plus threaded bosses to accommodate either air venting valves or for monitoring the fluid pressure. Furthermore, the connecting tube shows a 'fabricated Tee' arrangement which has a 'plain tail' i.e. no threaded or flange fitting is attached.

Coils manufactured with stainless steel tubes obviously use stainless steel headers and in this case the header material can conform with either DIN/EN standard material sizes and wall thicknesses or alternatively can conform to the ASME Schedule standards, which allows for Schedule 10s and Schedule 40 wall thicknesses, which exceed the DIN thicknesses. Such materials are used when the PED strength calculations fail with the thinner 2.0 mm DIN wall thicknesses.

Headers fitted to condenser coils are referred to as the 'Hot Gas Inlet' and 'Condensate Outlet'. In the case of evaporators, the outlet header is referred to as the 'Suction Outlet' ... as the interconnection refrigerant pipework leads directly to the 'suction' (inlet) connection of the compressor.

L-TYPE 90° MITRE JOINT

The simplest method to fabricate a header with a connection inlet/outlet at right angles (90°) is to cut the pipe at 45° and either braze or TIG weld the resultant 90° assembly. However, this joint has strength limitations and when considering PED category coils, must often be changed to a fabricated Tee or an extruded or hydroformed Tee piece.

FABRICATED/EXTRUDED/HYDROFORMED TEE JOINTS

For high pressure and/or high temperature applications where the above mitred joint may not be strong enough, either a fabricated Tee or the use of a bought-out extruded/hydroformed Tee must be provided when designing the header assembly

HEADER LEGS/STUBS

The header tube/manifold has '*header stubs*' or '*header legs*' matching the number of circuits, which are brazed or welded to the coil tubes that provide the beginning and end of the circuit pattern when offered up to the expanded coil block.

END CAPS

End caps seal the header assembly at the ends and can be either 'plate type' flat discs, which can be welded or for copper headers, occasionally brazed into position. However, EAS prefer to TIG weld the copper end cap onto the copper header.

In the case of stainless steel or titanium header assemblies, if the operating pressure warrants it, the end caps might need to be 'domed' in shape and conform with EN/DIN or ASME standards to match the header material of construction.

CONNECTIONS/NOZZLES

Generally, in the HVAC & R 'coil' industry sector, the term 'connections' is in more common use than the terminology 'nozzles', which relates more to shell & tube heat exchangers.

The type of attachment methodology provided on the coil inlet and outlet can be any of the following …

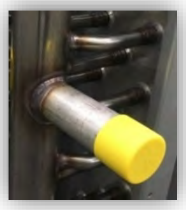
- Plain tails – no fittings
- BFW – bevelled for welding
- Threaded – male or female, parallel or tapered
- Flanged – loose, weld-neck, slip-on, screw-on
- Victaulic
- and more …

The connection fitting, when required, is welded or brazed to the connection tube/nozzle, which in turn is welded/brazed to the header/manifold.

Typically, <u>small</u> water heating or chilled water coils and DX & condenser coils are provided with plain tail connections, albeit that refrigerant coils will be sealed and filled with a positive charge of dry air/nitrogen, usually 1 barg.

For water or single phase fluid applications, the plain tails will either be brazed or welded directly to the supply and return pipework or may be coupled using compression fittings or similar.

Larger water/single phase fluid coils might be supplied with threaded fittings, up to 2" (DN 50), whilst larger connections sizes (2½" to 4" – DN 65 to 100) are usually supplied with flanged fittings.



On occasions, the connection tubes/nozzles can have a smaller diameter than the main header, either to minimise pressure drop in long headers or perhaps required from a manufacturing perspective, where the number of header legs would cause a welding access issue with a smaller header diameter.

AIR VENTS & DRAINS

When a water or single-phase fluid coil is filled with liquid there is a likelihood that air can become trapped somewhere in the headers or circuitry, which can result in underperformance of the heat exchanger if not expelled.

Over time, as the fluid circulates, any trapped air will migrate to the highest point in the system. For a coil this is usually the top of the headers and thus a fitting will be located at this high point. The fitting is often a simple female threaded boss and matching male plug. Removal of the plug allows the air to escape.

Automatic air vents are devices that automatically open and expel any air and can be attached to the female threaded boss to eliminate the manual process of venting the coil.

If air vents are fitted, then usually matching drain fittings are located at the bottom of the header(s) to allow any residual fluid, which may not be able to be drained from the inlet/outlet connection fittings, if not located at the very bottom of the header.

Usually, the drain fitting is a female threaded boss and matching male plug.

Air vents and drains are typically ¼" for small coils with connections sizes <= 1½" and ½" for connections sizes 2" and above. However, for fully drainable coils, eco-Air products or if the Customer specifies, ¾" if fitted.

FROST POCKET

A frost pocket is a short length of small diameter tube (copper or stainless steel) located at right angles to the header and fitted with a plugged threaded boss to accommodate a temperature sensor.

To avoid frost damage from freezing water or similar inside of the tubes, the temperature sensor can detect the onset of this condition and activate a mechanism to avoid any damage.

SCHRADER VALVE

The Schrader valve (also called American valve) is a type of pneumatic tyre valve used on virtually every motor vehicle in the world today.

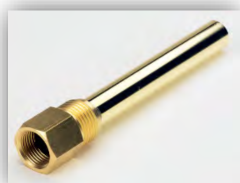
The Schrader company, from which it was named, was founded in 1844 by August Schrader. The original Schrader valve design was invented in 1891 and patented in the United States in 1893.

The Schrader valve consists of a valve stem into which a valve core is threaded. The valve core is a poppet valve assisted by a spring. A small rubber seal located on the core keeps the gas from escaping through the threads.

¼" Schrader valves are used in the refrigeration industry to allow easy charging or refilling of systems

EAS use Schrader valves for similar purposes on DX and condenser coils/products and as a quick self-sealing fixing on other coils for pressure testing purposes.

DRY POCKET

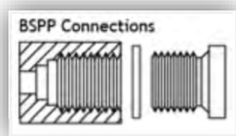


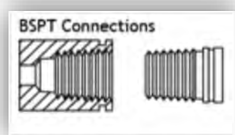
Dry pockets are usually mounted in the outlet header of a coil or more usually a dry cooler's heat exchanger, and when fitted with a temperature sensor, monitors the fluid leaving temperature.

The dry cooler's fluid outlet temperature is also the 'process inlet temperature' and usually the controlling temperature for the system.

A typical dry pocket is shown, which in this case is brass but can also be stainless steel 304 or 316. A female threaded boss is provided in the header and the dry pocket screwed into position so that typically, the end of the pocket is aligned with the centre line of the header. Therefore, the temperature sensor mounted inside the dry pocket measures the bulk temperature of the circulating fluid.

BSP(M/F/T) SIZES





The BSP (British Standard Pipe) thread form is based upon on trade size rather than the actual diameter and thus can lead to confusion.

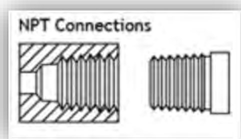
There are two types of BSP threads..

- Parallel (BSPP) also referred to as G or Rp
- Tapered (BSPT) also referred to as R or Rc

Both threads have the same pitch, 55° angle and rounded peaks and valleys (Whitworth thread form)

Trade Size	Threads per inch	Pitch		Major Diameter		Minor Diameter		Gage Length	
		Inch	mm	Inch	mm	Inch	Mm	Inch	mm
1/8	28	0.0357	0.907	0.383	9.728	0.3372	8.565	0.1563	3.97
1/4	19	0.0526	1.337	0.518	13.157	0.4506	11.445	0.2367	6.012
3/8	19	0.0526	1.337	0.656	16.662	0.5886	14.95	0.25	6.35
1/2	14	0.0714	1.814	0.825	20.955	0.7336	18.633	0.3214	8.164
3/4	14	0.0714	1.814	1.041	26.441	0.9496	24.12	0.375	9.525
1	11	0.0909	2.309	1.309	33.249	1.1926	30.292	0.4091	10.391
1 ¼	11	0.0909	2.309	1.65	41.91	1.5336	38.953	0.5	12.7
1 ½	11	0.0909	2.309	1.882	47.803	1.7656	44.846	0.5	12.7
2	11	0.0909	2.309	2.347	59.614	2.2306	56.657	0.625	15.875
2 ½	11	0.0909	2.309	2.96	75.184	2.8436	72.227	0.6875	17.463
3	11	0.0909	2.309	3.46	87.884	3.3436	84.927	0.8125	20.638
4	11	0.0909	2.309	4.45	113.03	4.3336	110.073		

NPT/NPS SIZES



NP (National Pipe) threads is also an American trade standard thread and not directly related to the actual diameter. Similar to the British variant, the thread form can be tapered (NPT) or parallel (NPS).

This thread form is sometimes referred to as MPT, MNPT or NPT (M) for male external threads and FPT, FNPT or NPT (F) for female internal threads.

A thread sealant should always be used to achieve a leak free joint, except for NPTF.

Both threads have the same pitch, 60° angle and have flat peaks and valleys (Sellers thread form)

Trade Size	Threads per inch	Pitch		Major Diameter (O.D)	
		Inch	mm	Inch	mm
1/8	27	0.03704	0.94082	0.405	10.29
1/4	18	0.05556	1.41122	0.54	13.72
3/8	18	0.05556	1.41122	0.675	17.15
1/2	14	0.07143	1.81432	0.84	21.34
3/4	14	0.07143	1.81432	1.05	26.67
1	11 ½	0.08696	2.20878	1.315	33.4
1 ¼	11 ½	0.08696	2.20878	1.66	42.16
1 ½	11 ½	0.08696	2.20878	1.9	48.26
2	11 ½	0.08696	2.20878	2.375	60.33
2 ½	8	0.125	3.175	2.875	73.03
3	8	0.125	3.175	3.5	88.9
4	8	0.125	3.175	4.5	114.3

BSP V'S NPT THREADS

NPT/NPS threads are common in America & Canada, whilst BSP threads are widely used elsewhere around the world.

Whilst American and British outside pipe diameters differ slightly, either thread can be cut onto a pipe of its respective trade size. However, threads should not be swapped if used on high pressure systems.

Trade Size	Pitch (Threads per Inch)	
	NPT/NPS	BSP
1/8	27	28
1/4	18	19
3/8	18	19
1/2	14	14
¾	14	14
1	11 ½	11
1 ¼	11 ½	11
1 ½	11 ½	11
2	11 ½	11
2 ½	8	11
3	8	11
3 ½	8	11
4	8	11
5	8	11
6	8	11

COMPENSATORS

Compensators are usually used as ‘flexible’ connections between a coil’s connections to the supply and return pipe-work. Such devices are fitted to either allow for thermal expansion/contraction of the heat exchanger or pipework or to allow for movement e.g. when a dry cooler is mounted on AVMs (anti-vibration mounts) and able to move, but the supply and return pipework is fixed.

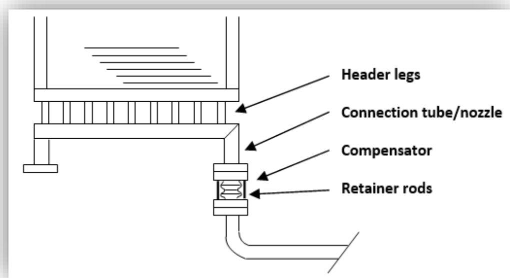
Recommendations …

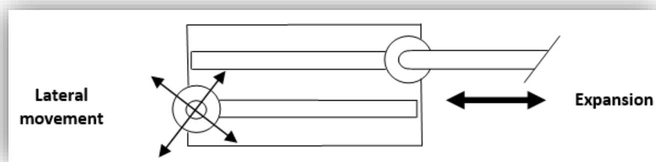
- It is the responsibility of the end-user to ensure that the coil connections/nozzles are protected against direct axial, angular or radial forces imposed by any interconnecting pipe-work or indeed via product movement if supported on spring -type AVMs (anti-vibration mounts).
- It is strongly recommended that flexible compensators fitted with restrictor bolts or indeed restricted hoses are used on both the inlet and outlet connections/nozzles to Flex coil products. Such compensators should be designed for lateral and radial forces to eliminate any pipe-work related movement from thermal expansion or transmitted vibrations in the case of some types of reciprocating compressors.
- All interconnecting pipe-work should be independently supported (not hung from the product) to ensure that no vertical loads are transfer to the product manifold assemblies.
- The load imposed upon any single ‘copper’ coil-to-header connecting tube (leg) should not exceed **+/-4 Newtons**. Thus any individual header assembly must not be exposed to a load greater than .. 4 x # circuits (# *header legs*) Newtons.
- The maximum allowed expansion of the compensator can be calculated by the following equation if the ‘spring constant’ is know for the compensator chosen.

Maximum allowable expansion = # Circuits x 4 / Spring Constant (mms)

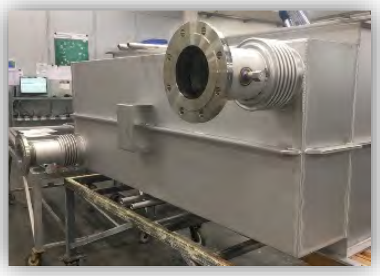
Example …

A dry cooler with 30 circuits is installed with lateral compensators with a spring constant of 32 N/mm. It therefore follows that the maximum allowable expansion of the compensator is 30 x 4 / 32 = 3.75 mm





In the case of condensers, compensators or stainless steel braided 'anaconda' hoses can be used to dampen the high frequency pulses generated by the reciprocating compressor from reaching the coil and causing 'tube fretting' withing the support sheet metal casework , resulting in leaks.



However, a novel use of compensators is to manage thermal expansion and the associated movement of the heat exchanger tube & fin matrix when mounted in a gas tight casework construction. If the coil connection tubes were just welded to the casework as they broke-out through the casework, then over time fatigue fractures would occur.

One solution is to use 'external slip-over' compensators to allow for movement but not transfer any stresses to the welds.

DISTRIBUTOR & LEADS

Distributors and leads are associated with refrigerant DX (*Direct or Dry eXpansion*) evaporators with more than one circuit per section where they and are fitted with a refrigerant distribution system comprising a short inlet connection tube attached to a distributor body (often brass) that accommodates a number of capillary or distributor leads matching the number of circuits of the coil.

This inlet arrangement differs from a single phase cooling coil due to the nature of the liquid/gas mixture that exits the expansion device and is fed to the coil. A conventional inlet header would result in bad refrigerant distribution … gravity would favour the liquid portion.

The distributor leads can typically be Ø4, 5, 6 or 8 mm in diameter depending upon the refrigerant mass flow rate that they need to handle. The lead length is governed by the coil section finned height, with leads usually not being less than 500 mm or greater than 1500 mm.

The distributor leads are sized to create a pressure drop to ensure equal refrigerant liquid/flash gas distribution to each circuit. However, there are various 'schools of thought' regarding what the 'ideal' pressure drop should be. For example, Danfoss suggest a pressure drop of 0.75 bar, whilst Alco suggest 1.0 bar and Sporlan suggest 10 psi (~0.7 bar).

The lead selection charts plot the diameter required against the necessary length for a variety of refrigerants at differing evaporating temperatures and liquid inlet temperatures. Then the 'trick' is to choose the most suitable diameter.

Fortunately, the capacity range for each lead diameter is deemed to be around +/- 25%, so there is scope for movement, besides the fact that typical DX coil operation conditions are rarely constant because the thermal load fluctuates.

The connection tube and distributor lead material matches the coil tube material. For stainless steel and aluminium coils, the distributors are fabricated by EAS in a similar material. However, in the case of copper tube coils, the distributor is a 'bought out' item and manufactured in brass.

For special applications, stainless steel coils can be fitted with brass distributor bodies if the material is suitable for the refrigerant in question and issues with refrigerant distribution are known to exist. In such cases, the stainless distributor leads are silver soldered into the brass distributor body at one end, whilst and hollow stainless plugs attached to the other end are then TIG welded into the coil tubes.

Such an arrangement may be used if TIG welding the stainless leads into a stainless distributor creates a contraction of the internal lead hole diameter, which would likely cause uneven refrigerant distribution and unstable evaporator operation.

TUBE PLATES/TUBE SHEETS/END PLATES

For a typical vertically mounted coil (horizontal airflow), there are three common terminologies to describe the left hand side and right hand side vertical sheet metal plates that make-up the coil casework. These plates/sheets contain the flared/swagged or clearance holes designed to accommodate/support the heat exchanger tubes.

- Tube plates
- Tube sheets
- End plates

Usually, the customer dictates the casework material and on occasions the required thickness too. Otherwise, the application, size and weight of the coil will dictate the most suitable material and thickness.

One exception are coils with stainless steel tubes, where due to the acid passivation of the welds, galvanised steel tube plates would be adversely affected and corroded. Therefore, all stainless steel tube coils are supplied with stainless steel tube plates.

TOP & BOTTOM PLATES

Most, but not all, coils have top & bottom plates that complete the rectangular casework assembly that encapsulates the tube and fin matrix.

PERFORATED BOTTOM PLATE

Cooling coils whose finned surface temperature is below the dew point of the incoming air stream will generate condensed water on the fins. This condensate will run down the fins and accumulate on the bottom plate of the coil casework.

To avoid a water agglomeration issue, often the coil casework bottom plate is perforated with holes to allow the water to pass through this platework.

If for example, the coil is fitted into an AHU then the cooling coil section may include its own integral drain tray/pan to carry the condensed water away.

DIVIDER PLATES

As standard, if a coil's finned length exceeds 1100 mm, divider plates will be incorporated into the casework design to support the fin & tube pack along its length.

Clearly, depending upon the coil application, overall size, tube & fin material, row depth and orientation, divider plates may or may not be included.

INTERMEDIATE DIVIDER PLATES

Intermediate divider plates are usually associated with dry cooler or condenser products, where major divider plates define the fan sections, whilst intermediate dividers are inserted to support the coil block when fan section length typically exceeds 1100 mm.

END PLATES ONLY

Refer to coils which are not supplied with top & bottom plates, but only the two end tube plates. Such coils are often small and often fitted into fan coil units or small heating or cooling units

DRAIN TRAY/PAN

Cooling coils that generate condensed water on the fins must provide a mechanism to capture and carry away the condensate. In such cases a drain pan/tray can be included within the casework design.

Drain trays can be integral water capturing devices that replace the bottom plate of the coil. However, alternatives are sloping drain pans, removable slide-in trays (horizontal or sloping), made from the same material as the casework or perhaps stainless steel (304 or 316).

Furthermore, the drain tray can have drain connections mounted horizontally in the ends of the tray or vertically centred or off-set in the bottom of the tray.

Besides capturing the condensed water from the fins, the drain tray is usually extended to catch the drips from the return bends and headers.

For duct mounted cooling coils, where the coil block is subject to either positive or negative pressures, the drain tray design incorporates a section that is exposed to the positive or negative ductwork pressure and a section exposed to atmospheric pressure and as such, has two condensate drain connections.

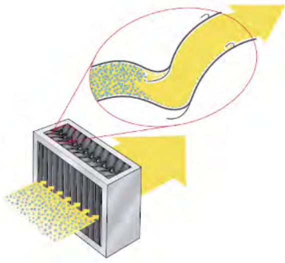
INTERMEDIATE DRAIN TRAY/PAN

When the finned height of a cooling coil exceeds 1200 mm the quantity of water condensing upon the fins is a major consideration and often an intermediate drain tray is fitted at the mid-point of the coil, as shown in the figure, to capture the water from the upper half of the finned surface and avoid water agglomeration and the potential for 'water carry over'. This construction involves splitting the fin block horizontally into two fin packs, which sandwich the intermediate drain tray.

Vertical drainpipes are fitted to this intermediate drain tray to allow drainage of water into the main drain tray mounted at the base of the coil.

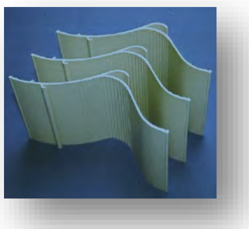
ELIMINATORS

Eliminators are used to remove air borne water droplets from an air stream from travelling downstream. They can be either plastic or stainless steel profiles, which have a complex shape to capture the water droplets.



The full height eliminator profiles or 'blades' are mounted vertically and evenly spaced across the coil's width (finned length).

Such eliminator sections can either be an integral feature of the cooling coil and incorporated within the coil casework or confined into a separate encased section which is attached to the coil during final installation.



The integral design captures any carried-over water into the cooling coil's drain tray, whilst a separate eliminator section would have its own drain tray assembly.

In the case of large coils, *see above figure in Intermediate drain tray section*, which are fitted with intermediate drain trays, the eliminator section is split vertically to drain into the intermediate tray. However, special designs may involve not splitting the coil, but allowing water carry-over and capturing the downstream water droplets in a horizontally split eliminator section with intermediate drain trays.

COVER BOXES

Cover boxes fitted to a coil; also referred to a 'cladding'; are sheet metal enclosures encapsulating the header end and/or the return bend end of the coil. These boxes can be simple pop riveted constructions or if the coil uses stainless casework, can be a fully welded construction providing a gas tight solution.

Standard coil casework constructions inherently leak air through the clearance swagged holes in the tube plates that support the tube and fin coil block. If air leakage is a concern, cover boxes can be fitted to reduce, but not eliminate air leakage.

If an air/gas tight construction is required, then fully welded cover boxes must be fitted.

Cover boxes can also be insulated i.e. filled with Rockwool, Armaflex or expanded foam to either prevent heat loss or eliminate condensation on the headers.

TRANSFORMATION SECTIONS

Sheet metal transformation sections convert the coil face dimensions to either smaller of larger square, rectangular or circular duct sizes over a specified length. Such transformations are only offered for relatively small internal static pressures or where we can offer reinforcement that can withstand the stresses imposed.

AIR SPIGOTS

These are usually quick assembly circular ductwork attachments that are attached to a rectangular plate and mounted to the inlet, outlet or both faces of the coil to provide a matching connection to the client's ductwork.

REINFORCED CASEWORK

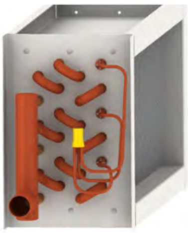
When coils are mounted in ductwork systems, they may be subject to internal positive or negative static pressures. Usually, such pressures are small, and the standard casework material thickness is sufficient to avoid any over stressing of the construction.

However, large coils that comprise large areas of unsupported sheet metal, and subject to relatively low internal pressures, may deform too much. In these cases, reinforcement is added to the coil casework and cover boxes to avoid any deformation and damage.

As an example, a 1.5 m^2 unsupported area of sheet metal subject to a 3000 Pa static pressure is exposed to a uniformly distributed force of 1.5 x 3000 = 4500 N which equates to it having to support a load of approximately 450 kgs. Thus a 1.5 mm or even a 3.0 mm thick galv. sheet will unduly deflect and would need reinforcement profiles.

Flat bar profiles e.g. 40 x 10 mm or 60 x 12 mm are stitch welded to the outside of the casework construction to ensure that the construction is not over stressed.

DX EVAPORATORS



Direct or dry expansion (DX) evaporators are cooling coils utilising a two phase evaporating refrigerant to cool the air stream.

DX coils; unlike single phase cooling coils; are fitted with an inlet distributor and distributor (capillary) leads to handle the entering liquid/flash gas mixture (*see Distributor & lead section above*) and a conventional outlet (suction) header to return the typically superheated gas to the compressor.

LIQUID INLET

For an evaporator, the liquid inlet usually refers to the inlet connection of the DX coil. This, smaller of the inlet & outlet connections, is either directly attached to the expansion device or via a short length of piping.

TEV/EEV

The warm, high pressure liquid refrigerant leaving the condenser flows through the TEV (thermostatic expansion valve) or EEV (electronic expansion valve) and drops in pressure from the condensing pressure to the evaporating pressure.

This essentially adiabatic process causes the refrigerant to traverse the saturated liquid line and enter the two-phase refrigerant 'vapour dome', ending up at typically a 15 – 20% dryness fraction (quality) i.e. 15-20% of the liquid 'flashes off' into a gaseous state.

The TEV/EEV is controlled by a sensor mounted on the upper 180° of the suction outlet connection pipe of the coil. Its location allows it to measure the superheated suction gas temperature leaving the DX coil. If the measured amount of superheat differs from the superheat setting of the valve, typically 3 to 7K, the valve will modulate to adjust the refrigerant mass flow rate.

Superheat at the DX coil suction is needed to ensure that only gas enters the compressor. Should any slugs of unevaporated liquid enter the compressor, then significant damage may ensue.

SUCTION OUTLET

Refers to the outlet connection of an evaporator, where the superheated gas/vapour, following evaporation, is collected and fed back to the compressor.

The term 'suction' originates from the fact that the outlet piping from the evaporators is connected to the inlet of the compressor.

Compressor terminology refers to the low pressure inlet port as the 'suction', whilst the outlet high pressure port is known as the discharge or hot gas discharge port.

INTERLACED SECTIONS

Often DX evaporators are required to have more than one refrigerant section, perhaps as part of the same refrigerant system or perhaps a singe coil servicing two separate refrigerant systems, perhaps even using different refrigerants.

Such cases can be served by building the coil with two or more discrete sections, each with their own liquid inlet connections and suction outlet headers.

The individual coil sections can either be mounted above one another; known as **split in face**, or can be arranged to intertwin with one another, known as **interlaced**.

Considering a simple 2 equal section coil operating at 50% or 100% capacity, the former arrangement results in the deactivated upper or lower half of the coil, passing untreated air, whilst the active section passes cooled air. Clearly, the treated and untreated air streams would mix downstream to provide the desired leaving temperature.

This arrangement may well function adequately, but basically when operating at 50% load, only 50% of the coil surface is providing any useful work.

If however, the 2 sections have their circuits interlaced, then when one section is deactivated, the active section's circuits span the whole face area of the coil and although only 50% of the circuits and tubes are providing cooling, the whole of the coil's surface area is available to do work. Thus this arrangement is a more efficient means of providing variable capacity control.

PUMP CIRCULATED EVAPORATORS

An alternative to DX evaporation is a pump circulated system. This differs from a DX system insofar that in place of a TEV/EEV is a low pressure vessel fitted with a float valve operated pressure reduction valve, where the low pressure liquid/vapor mixture resides.

The pump circulated coil is fed with 100% liquid from the vessel via a liquid pump. Usually the liquid is overfed to the inlet liquid header, mounted horizontally and located at the bottom of the coil. As the liquid is pumped through the coil, passing vertically upwards through the vertically arranged circuits, a portion of the liquid evaporates and the mixture exisits via the horizontal outlet header. The quality of the refrigerant leaving the coil is dictated by the pump rate, often PF = 1.5 or 2.0 for Ammonia systems.

The exiting refrigerant returns to the low pressure vessel where the liquid portion is recirculated through the coil, whilst the evporated gas is drawn off the top of the vessel and returns along the suction line to the compressor.

CONDENSERS

A condenser coil handles the high pressure superheated hot gas from a refrigeration system's compressor and initially de-superheats (cools) the gas to saturation and then condenses the refrigerant into a saturated liquid. If the coil has sufficient surface area, the liquid may be marginally sub-cooled.

HOT GAS INLET

Is defined as the inlet (larger) connection of a condenser coil. Usually, the pipework from the compressor's discharge port runs directly to the condenser, so the compressor outlet equates to the condenser inlet and handles the high pressure, superheated discharge refrigerant gas.

As the state of the refrigerant at this stage of the cycle is a high pressure gas, the pipework is sized to handle a large volume of gas without creating too much pressure drop, which would adversely affect the condensing temperature.

Typical design discharge gas velocities in pipe runs are in the range **10 – 18 m/s** where the lower limit ensures oil entrainment and eliminates fouling issues, whilst the upper limit affects pressure drop, noise and the economic aspects. Furthermore, ASHRAE design guidelines suggest a maximum pressure drop equating to temperature drop of **0.02 K/m**.

Generally, if in doubt, oversize the connection to ensure there is no pressure drop penalty.

CONDENSATE OUTLET / LIQUID DRAIN

Refers to the outlet connection of a condenser where the condensed liquid exits the coil on route via the liquid line to the system's liquid receiver or perhaps directly to the expansion device.

As the condensate (liquid) is somewhat smaller in volume than the volume of the entering hot gas, the size of the condensate connection can be smaller than the hot gas inlet connections size.

Traditionally, the condensate connections were assumed to be one size smaller than the hot gas inlet connection, however, often the inlet and outlet sizes are kept the same to ease manufacture.

Although keeping the sizes similar is not detrimental to pressure drop or performance, it can be argued that it is not the most economical solution !

The condensate connection size is often based upon a liquid pipeline velocity of <0.5 m/s. However, there is a 'school of thought' that states that this threshold velocity (sewer velocity) should not be exceeded when the outlet connection tube is **only half full**. This is to ensure that liquid cannot 'backup' into the condenser coil and cause the condenser to underperform. Thus, the preferred sewer velocity is **~0.25 m/s**.

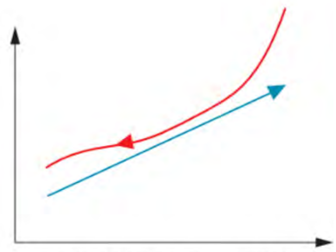
The condensate outlet/drain is also a term that can refer to a cooling coil's condensate drain tray/pan connection, which routes the water away to the sewer.

CO_2 GAS COOLERS

Gas coolers play a similar role to condensers in a refrigeration system, but rather than processing a subcritical hot gas, desuperheating it and then condensing it, the supercritical CO_2 gas is cooled and rejects its heat as a supercritical fluid. Also *refer to Section CO2 (R744) Gas Coolers*

Supercritical fluids exhibit unusual behaviour and are often referred to as existing as a 4th phase, where the previous 3 states are solid, liquid & gas.

The critical pressure for CO_2 is ~73 barg equating to ~31°C.



Heat rejection - transcritical CO_2

Typical gas cooler operating conditions are 90 to 120 barg with an entering gas temperature up to 125°C. Often the gas is required to be cooled to just a few degrees above the cooler's air entering temperature. Thus the cooling profile of the gas is significant and traditional design approaches do not lend themselves to such equipment design.

Owing to the pressures involved, thick wall copper or K65 high tensile copper must be used for the coil tubes and either K65 headers or stainless steel headers.

Evapco Inc.'s eco-Air gas cooler range uses 5/8" stainless tubes with ANSI Schedule 80 or 160 header material to comply with ASME 31.5

FULLY DRAINABLE COILS

Fully drainable coils generally relate to single phase fluid coils or dry coolers, typically water filled, that need to be able to fully drain during times when ambient temperatures fall below zero and frost damage is a cause for concern.

In the case of large dry coolers that must use water in the process, wintertime creates issues if the process is off-line … often during Christmas holidays.

Unless a system is frost protected by using glycol or similar antifreeze fluids or winterisation recirculating heating systems, the water may freeze, and tubes or headers will split or fracture.

Therefore the ability to open the air vents and drain connections and fully drain the coil is essential.

Surface tension effects of water in long, even slightly inclined small diameter smooth tubes result in a degree of water retention, which may result in frost damage if this water freezes or encapsulates slugs of water.

Furthermore a horizontal coil with more than one pass (tubes per circuit) has a tendency to slightly sag and thus opening the vents and drains will not evacuate all the water.

To overcome this issue, all return bends are replaced with secondary headers, which also have air vents and drain connections fitted, plus the whole product or coil is placed on a small incline (1%), so that gravity will overcome the surface tension effects and enable the coil to fully drain.

Additionally, it is always recommended to use air pressure to assist the evacuation.

FROST DAMAGE

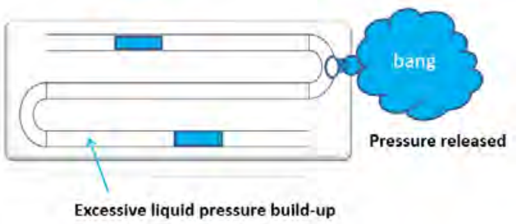
There is a misconception that coils or dry coolers that have failed exhibiting frost damage is because the water has frozen … *and as everyone knows*, ice has a greater volume than water, so clearly the tubes or bends split because of this phenomenon !!

Well, although in a fully filled and sealed container, the above scenario would hold true, the real life situation with a duct mounted coil or dry cooler exposed to temperatures close to or below zero, often experience 'cold spots' in certain areas of the coil face. The water inside the tubes in these vicinities will tend to freeze causing an 'ice plug' to form and over time, grow.

However, an ice plug, if restrained … *by the tube wall* … does not grow circumferentially, but axially/longitudinally along the length of the tube where the resistance is far less.

Most water based process systems are not totally enclosed and pressurised, so the occurrence of a single ice plug in a fully filled coil/dry cooler will not create an issue … because as the plug grows it will push the water away.

However, if two ice plugs in the same circuit form simultaneously and grow axially, then the water trapped between the ice plugs quickly become pressurised. Now as water is non compressible, the water pressure inside the trapped section of the tube rises steeply and the weakest portion of the tube, return bend or hairpin or perhaps even the header leg, will fail, immediately relieving the pressure.



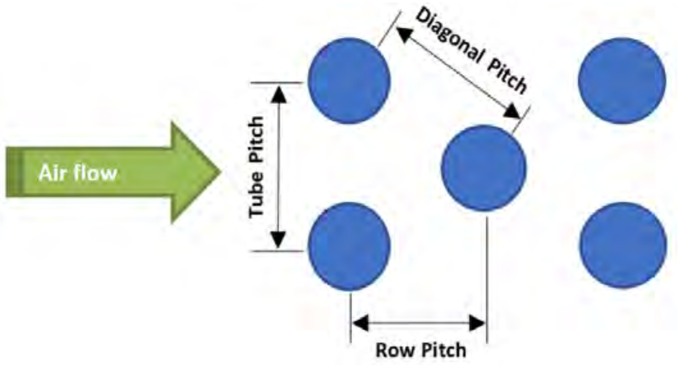
The weakest area of a copper coil's circuit is often the 'heat affected zone' following the brazing process, thus the annealed copper in the vicinity of the brazed joints. However, on occasions it can be the hairpin bend that will fail.

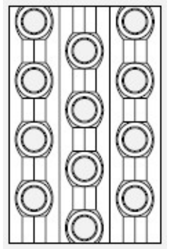
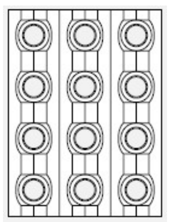
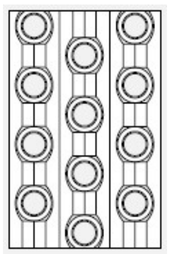
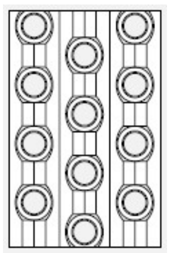
Clearly there are exceptions to the above and thus other reasons for failure, so each case should be considered on its own merit.

EAS has developed a frost protection coil design for vertically mounted coil used in ducted systems or AHUs, which is a derivative of the 30 year old ThermoGuard system, whose patent has now expired. In the days of Flex coil, the name FlexGuard was adopted for this system.

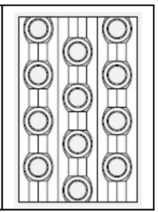
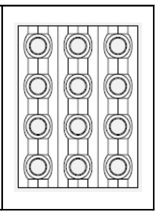
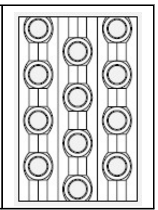
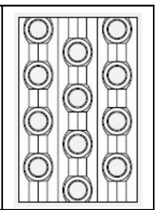
Nevertheless, such systems are rare these days and frost protection is usually applicable to water based flat-bed dry cooler systems, mentioned above, which adopt the sloped, fully drainable coil design approach.

GEOMETRIES

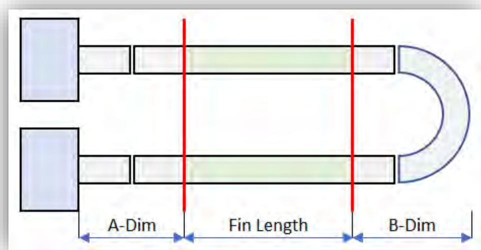


Pattern		Geometry		
		Tube Pitch	Row Pitch	Diagonal Pitch
A : 60x30-Ø15 *Staggered*		60.00	30.00	42.43
B : 30x30-Ø15 *In-line*		30.00	30.00	N/A
C : 40x35-Ø15 *Staggered*		40.00	34.64	40.00
Z : 25x22-Ø9.52 *Staggered*		25.00	21.65	25.00

TUBE MATERIAL OPTIONS

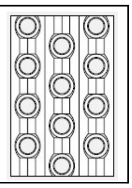
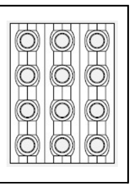
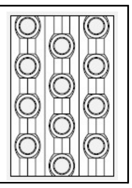
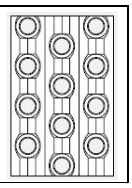
		Geometry			Tube Material					
					Outside Diameter - mm		Wall Thickness - mm			
		Tube Pitch	Row Pitch	Diagonal Pitch	Nominal	Expanded	CU	AL	304/316	TITAN
A : 60x30-Ø15 *Staggered*		60.00	30.00	42.43	15.00	15.65	0.38 0.75	1.25	0.60	0.60
B : 30x30-Ø15 *In-line*		30.00	30.00	N/A	15.00	15.65	0.38 0.75	-	-	-
C : 40x35-Ø15 *Staggered*		40.00	34.64	40.00	15.00	15.65	0.38 0.75	-	0.60	-
Z : 25x22-Ø9.52 *Staggered*		25.00	21.65	25.00	9.52	10.15	0.28 0.5	-	-	-

DIMENSIONAL ALLOWANCES



Tube Ø	Tube Matl	Geometry	A-Dim	B-Dim: Hairpins	B-Dim: Straight tube + R/bend
3/8"	Copper	Z	75	40	40
15 mm	Copper	A	90	50	50
		C		60	60
	SS304/316	A	90	50	65
		C		60	75
	Aluminium	A	110	50	70
	Titanium	A	130	100	100

FIN MATERIAL OPTIONS

Geometry		Fin Material															
		AL		ALEP		ALMg		ALHy		CU		CUSN		304		316	
		Thk - mm	Pitch	Thk - mm	Pitch	Thk - mm	Pitch	Thk - mm	Pitch	Thk - mm	Pitch	Thk - mm	Pitch	Thk - mm	Pitch	Thk - mm	Pitch
A : 60x30-Ø15 *Staggered*		0.12	1.5 - 4.0	0.12	1.5 - 4.0	0.12	1.5 - 4.0	0.18	1.5 - 6.0	0.12	1.5 - 4.0	0.12	1.5 - 4.5			0.12	1.5 - 4.7
		0.18	1.5 - 6.0	0.20	1.5 - 6.0	0.25	1.5 - 6.0			0.15	1.5 - 6.5			0.18	1.5 - 6.5	0.18	1.5 - 6.5
		0.25	1.5 - 7.0							0.25	1.5 - 7.0						
B : 30x30-Ø15 *In-line*		0.12	1.5 - 3.4	0.12	1.5 - 3.3	0.12	1.5 - 3.0	0.18	1.5 - 3.8	0.12	1.5 - 3.4	0.12	1.5 - 3.0	-	-	-	-
		0.18	1.5 - 3.8	0.20	1.5 - 3.5	0.25	1.5 - 4.0			0.15	1.5 - 4.0						
		0.25	1.5 - 5.0							0.25	1.5 - 4.5						
C : 40x35-Ø15 *Staggered*		0.12	1.5 - 3.0	0.12	1.5 - 4.0	0.12	1.5 - 3.0	0.12	1.5 - 3.0	0.12	1.5 - 3.0	0.12	1.5 - 3.0	0.15	15 - 5.0	0.12	1.5 - 3.0
		0.18	1.5 - 4.0	0.20	1.5 - 7.0	0.25	2.6 - 5.0	0.18	1.5 - 4.0	0.15	1.5 - 4.0					0.18	1.5 - 5.0
		0.25	1.5 - 5.0					0.25	1.5 - 5.0								
Z : 25x22-Ø9.52 *Staggered*		0.1	1.5 - 3.0	0.12	1.5 - 3.2	0.12	1.5 - 3.0	0.1	1.5 - 3.2	0.1	1.5 - 3.0	-	-	-	-	-	-

FIN SURFACE AREA – A_O, A_I

The total surface area of a coil can be calculated from the normalised parameter, A_o for a given geometry and fin pitch, where …

A_T = A_o x Fin Height / 1000 x Fin Length / 1000 x Rows m^2

Furthermore, (*see later*), calculation of the overall heat transfer coefficient involves the use of A_i, A_m & A_o for a given geometry and fin pitch.

Geometry	A			C		Z	
Wall Thk - mm	0.38	0.6	1.25	0.38	0.6	0.28	0.5
A_i - $m^2/m^2/row$	0.780	0.757	0.689	1.169	1.135	1.237	1.181
A_m - $m^2/m^2/row$	0.800	0.378	0.344	0.585	0.567	1.272	0.591
Fin Thk - mm	0.12					0.1	
Fin Pitch - mm	A_o - $m^2/m^2/row$						
2.0	27.48			30.86		19.39	
2.5	22.15			24.93		15.77	
3.0	18.60			20.99		13.37	
3.5	16.06			18.17			
4.0	14.15			16.05			
5.0	11.49			13.09			
6.0	9.71			11.12			

Stainless steel tubing is one of the most versatile metal alloy materials used in manufacturing and fabrication. The two common types of tubing are seamless and welded. Deciding between welded v's seamless tubing primarily depends on the application requirements of the product. In choosing between the two keep in mind that first the tubing must be compliant with your project specifications and that secondly, it must meet the conditions for which the tubing will ultimately be used.

TUBING V'S PIPING

Though both the words tube and pipe are often used interchangeably, largely because both are hollow shaped, there are important distinctions between the two when determining welded vs. seamless tubing needs. Tubes are measured by the outside diameter (OD) and wall thickness. A pipe, on the other hand, is measured by its inside diameter (ID). In terms of functionality, tubing is generally used in structural and aesthetic applications whereas piping is used for transporting fluids, liquids, and gases.

SEAMLESS TUBE MANUFACTURE

Knowing that distinction can also help in determining which tubing is best for a given application, welded or seamless. The method of manufacturing welded and seamless tubing is evident in their names alone. Seamless tubes are as defined – they do not have a welded seam. The tubing is manufactured through an extrusion process where the tube is drawn from a solid stainless steel billet and extruded into a hollow form. The billets are first heated and then formed into oblong circular moulds that are hollowed in a piercing mill. While hot, the moulds are drawn through a mandrel rod and elongated. The mandrel milling process increases the moulds length by twenty times to form a seamless tube shape. Tubing is further shaped through pilgering, a cold rolling process, or cold drawing.

WELDED TUBE MANUFACTURE

A welded stainless steel tube is produced through roll-forming strips or sheets of stainless steel into a tube shape and then welding the seam longitudinally. Welded tubing can be accomplished either by hot forming or cold forming processes. Of the two, cold forming results in smoother finishes and tighter tolerances. However, each method creates a durable, strong, steel tube that resists corrosion. The seam can be left beaded or it can be further worked by cold rolling and forging methods. The welded tube can also be drawn; similar to seamless tubing; to produce a finer weld seam with better surface finishes and tighter tolerances.

CHOOSING BETWEEN WELDED & SEAMLESS

There are benefits and drawbacks in choosing welded v's seamless tubing.

SEAMLESS TUBING – EN10216-2

By definition, seamless tubes are completely homogenous tubes, the properties of which give seamless tubing more strength, superior corrosion resistance, and the ability to withstand higher pressure than welded tubes. This makes them more suitable in critical applications in harsh environments, but it comes with a price.

Benefits

- Stronger
- Superior corrosion resistance
- Higher pressure resistance

Applications

- Oil and gas control lines
- Chemical injection lines
- Below sea safety valves

- Chemical processing plant steam and heat trace bundles
- Fluid and gas transfer

WELDED TUBING – EN10217-5

Welded tubing is generally less expensive than seamless tubing due to the simpler manufacturing process in creating welded tubing. It is also readily available, like seamless tubing, in long continuous lengths. Standard sizes can be produced with similar lead times for both welded and seamless tubing. Seamless tubing costs can be offset in smaller manufacturing runs if less quantity is required. Otherwise, though custom-sized seamless tubing can be produced and delivered more quickly, it is more costly.

Benefits

- Cost-efficient
- Readily available in long lengths
- Fast lead times

Applications

- Architectural applications
- Hypodermic needles
- Automotive industry
- Food and beverage industry
- Marine industry
- Pharmaceutical industry

COSTS OF WELDED V'S SEAMLESS

Costs of seamless and welded tubing are also related to such properties as strength and durability. Welded tubing's easier manufacturing process can produce larger diameter tubing with thinner wall sizes for less. Such properties are more difficult to produce in seamless tubing. On the other hand, heavy walls can be achieved more easily with seamless tubing. Seamless tubing is often preferred for heavy wall tubing applications that require or can withstand high pressure or perform in extreme environments.

Note : *As a point of reference, seamless tubing can be substituted for welded tubing, but welded tubing should never be substituted for seamless tubing*.

TEMPERATURE & PRESSURE LIMITS

The following are guideline nominal upper limits for both pressure and temperature for different coil applications.

It should be born in mind that EAS’s **PED Design Tool** allows for more detailed strength calculations to be performed, so conditions outside of those detailed below can be assessed in the correct fashion.

	Tube Material		
		PS - barg	TS - °C
_V / _K	0.28/0.38 mm Copper	10	110
	0.5/0.75 mm Copper	10	175
	Aluminium	10	150
	Titanium	10	300
	Stainless Steel	20	300
_S	0.75 mm Copper	3	143
	Stainless Steel	7	170
_D / _C	0.28/0.38 Copper	Refrigerant Dependant	125
	Stainless Steel		125
_V / _K / _S	SS Chamber Coils	6	165

Fin Material	
0.1/0.12 mm Aluminium	120
>= 0.18 mm Aluminium	150
ALEP/ALHy	150
ALMg	225
Copper	225
Stainless Steel	300

Temperature limits for materials are generally related to when the material begins to lose its mechanical strength, which will affect what pressure can be handled or may impact upon the ‘bond contact’ related to the tube and fin expansion process.

COPPER

Generally, copper should not be exposed to temperatures above 225°C, otherwise oxidisation can take place, which over time may resulting in failure and /or excessive corrosion.

STAINLESS STEEL

Austenitic stainless steels (304/316) should not be exposed to temperatures >525°C in view of the change to its granular structure and the severe weakening that results.

TITANIUM

Titanium and its alloys have melting points higher than those of steels, but maximum useful temperatures for structural applications generally range from as low as 400 °C to the region of approximately 600 °C, dependent upon composition.

Titanium begins to lose its strength when heated above 430 °C and is not as hard as some grades of heat-treated steel; it is non-magnetic and a poor conductor of heat and electricity.

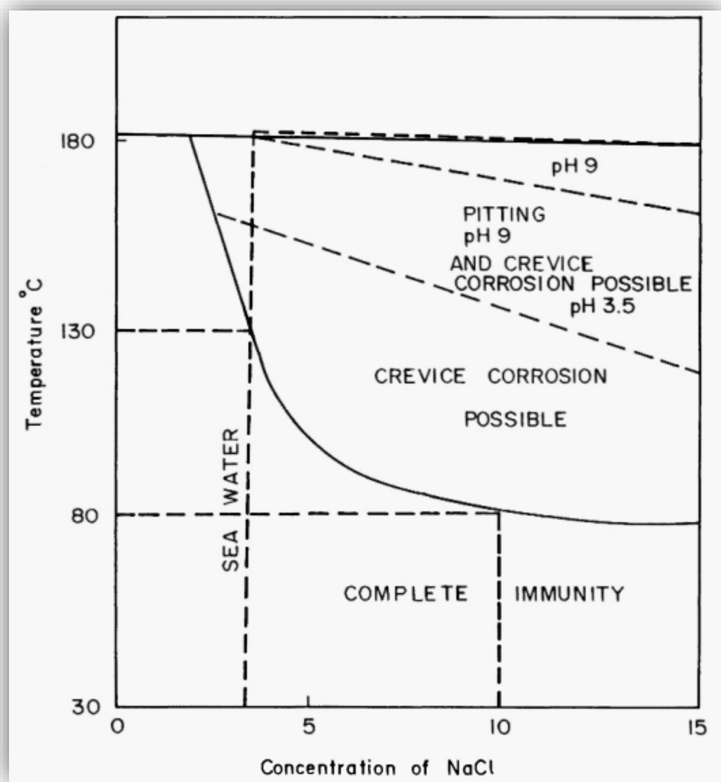
When alloyed with other metals such as aluminium or vanadium, titanium becomes dramatically stronger than many steels. In terms of tensile strength, the best titanium alloys beat low to medium grade stainless steels. However, the highest grade of stainless steel is stronger than titanium alloys.

Maximum service temperature of titanium alloys is mostly limited by the creep and oxidation resistance. Typically, titanium alloys are specified for a maximum service temperature of 600°C. However, they are usually used for temperatures around 540°C because the parts have a service time of several thousand hours.

Because of its high strength and creep resistance, commercially pure titanium can remain stable at temperatures up to approximately 300°C.

Titanium is resistant to a sea-water environment up to a temperature of 130°C. However, at high NaCl concentrations and low pH brines, pitting and crevice corrosion can occur even as low as 80°C.

However, for typical offshore related applications where sea water temperature perhaps rarely exceed 35°C, titanium is immune to corrosion and erosion and can withstand velocities well above 20 m/sec.



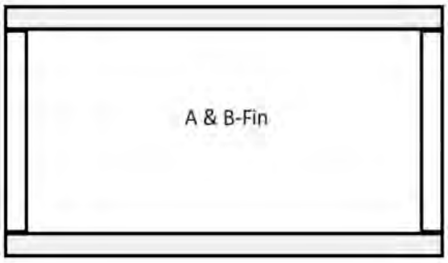
K65 – HIGH TENSILE COPPER

Recently, a high tensile copper material has become popular for CO_2 gas cooler applications where pressures often approach 120 barg. This variant of copper tube material has a variety of designations …

- K65
- CuFe2P
- C19400
- CW107C

Currently, EAS only uses this material for the header assemblies up to 2 1/8" (Ø54 mm) along with domed end caps and Tee pieces, all rated at 120 barg.

STANDARD CASEWORK CONSTRUCTIONS



Full length top & bottom plates

Full height tube & return bend plates

- Galvanised & AluZink casework have - as standard - pop riveted corners & divider plates
- SS304/316 & Aluminium casework have welded corners & divider plates
- Generally a **1.0 mm** gap between the fin block and the top & bottom plates is provided to ease tubing up and expansion. However, this can be removed if necessary

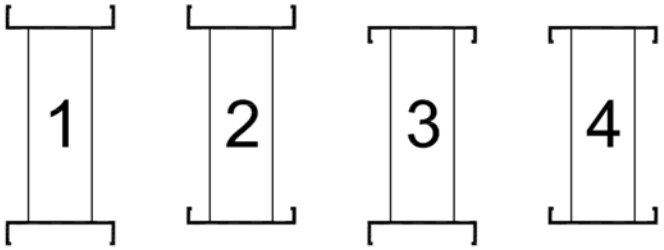
ALTERNATIVE CONSTRUCTIONS

Casework constructions to suit the application can be diverse …

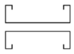
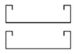
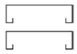
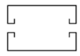
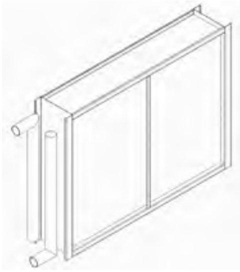
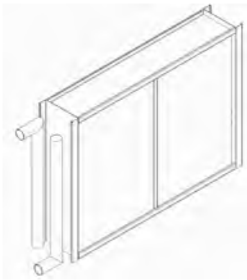
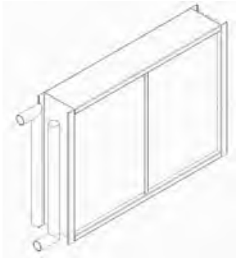
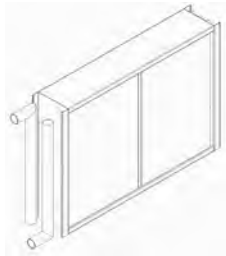
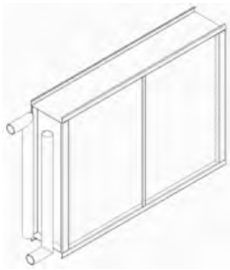
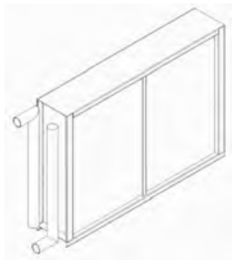
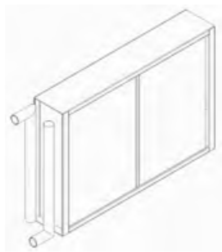
- Tube plates/sheets only (*no top & bottom plates*) : fan coil units & air conditioners
- Duct mounted : ventilation systems
- Slide-in : air handling units
- Reinforced : pressurised applications
- Fitted with a drain tray
  - Fixed – integral
  - Removable
  - Sloped
- Intermediate drain tray
- Cover boxes : header & return bend ends
- Transition sections
- Circular spigots

CASEWORK STYLES

Besides the two casework concepts, there are four common construction alternatives, albeit that there are many other permutations, such as constructions without ‘top & bottom’ plates or perforated bottom plates and of course cooling coils fitted with drain trays and coils with cover boxes to enclose the header assemblies and/or return bends.



Combining the two casework constructions and four styles we arrive at 8 basic casework permutations …

		Style			
		1	**2**	**3**	**4**
					
Construction	**1**				
	2				

CASEWORK MATERIAL THICKNESSES

0.28/0.38 Cu or 1.25 AL Tube with AL or Cu Fins

Fin Thk	Galv/AlZn/SS304/SS316			ALMg		
	L <= 2900 and Rows <= 8			**L <= 2900 and Rows <= 8**		
	H <= 1200	1200 < H <= 1800	H > 1800	H <= 1200	1200 < H <= 1800	H > 1800
0.1 / 0.12 0.18 0.25	**1.5**	**1.5**	**1.5**	**2.0**	**2.5**	
		2.0	**2.0**			
	L <= 4000 and Rows <=12			**L <= 4000 and Rows <=12**		
0.1 / 0.12 0.18	**1.5**	**1.5**	**2.0**	**2.5**		
0.25	**2.0**	**2.5**	**2.5**			
	L > 4000 or Rows > 12			**L > 4000 or Rows > 12**		
	H <= 2880			H <= 2880		
0.1 / 0.12	**2.0**			**2.5**		
0.18 0.25	**2.5**					

0.75 Cu, 0.6 SS or Ti Tube with AL, Cu or SS Fins

Fin Thk	Galv/AlZn/SS304/SS316			ALMg		
	L <= 2900 and Rows <= 8			**L <= 2900 and Rows <= 8**		
	H <= 1200	1200 < H <= 1800	H > 1800	H <= 1200	1200 < H <= 1800	H > 1800
0.12 0.18 / 0.25	**1.5**	**1.5**	**2.0**	**2.5**		
	L <= 4000 and Rows <=12			**L <= 4000 and Rows <=12**		
0.12 0.18 / 0.25	**2.0**	**2.0**	**2.5**	**2.5**		
	L > 4000 or Rows > 12			**L > 4000 or Rows > 12**		
	H <= 2880			H <= 2880		
0.12 0.18 / 0.25	**3.0**			**3.0**		

CASEWORK MATERIAL WEIGHT

	Galv	**AlZn**	**SS304 - V2A** SS304 - 1.4301 SS304L - 1.4306	**SS316 - V4A** SS316 - 1.4401 SS316L - 1.4404	**AL/ALMg**	**Cu**
Density - kg/m^3	7860	7860	7930	7960	2710	8930
Thickness - mm	**Weight - kg/m^2**					
1.25	9.83	9.83	9.91	9.95	3.39	11.16
1.50	11.79	11.79	11.90	11.94	4.07	13.40
2.00	15.72	15.72	15.86	15.92	5.42	17.86
2.50	19.65	19.65	19.83	19.90	6.78	22.33
3.00	23.58	23.58	23.79	23.88	8.13	26.79
4.00	31.44	31.44	31.72	31.84	10.84	35.72

TUBE PLATES/SHEETS

To support the tube and fin block, the sheet metal platework; through which the tubes pass; referred to as the ‘tube plates’ or ‘tube sheets’ are punched with swaged/flared holes. These swaged holes are calibrated to ensure that the tubes ‘sit’ on a parallel portion of the formed hole and will not make contact with any raw, cut edges.

When the tubes are mechanically expanded to provide an interference fit with the fin collar, they are deliberately <u>not</u> expanded into the tube plates. The resultant small clearance is a design feature to allow the tubes to thermally expand lengthways without ‘stressing’ the tube plates … had the tubes been in intimate contact with the swaged holes.

EAS’s machinery limits the creation of swagged/flared holes to …

- Ø15 mm   <= 2.5 Stainless steel & Galvanised steel
- Ø9.52 mm   <= 2.0 Galvanised steel & <= 1.5 Stainless steel

If thicker tube plate material is specified, clearance holes are punched and internal thinner gauge flared hole plates are pop riveted or stitch welded to the thicker tube plates.

However, Evapco Inc. take another viewpoint and because of their hydraulic expansion technique, do indeed expand some of the tubes into the tube plate swaged holes. Furthermore, their tube plates have ‘clusters’ of swaged holes and clearance holes, so not all the tubes are expanded into the tube plates. This is apparently for thermal expansion reasons.

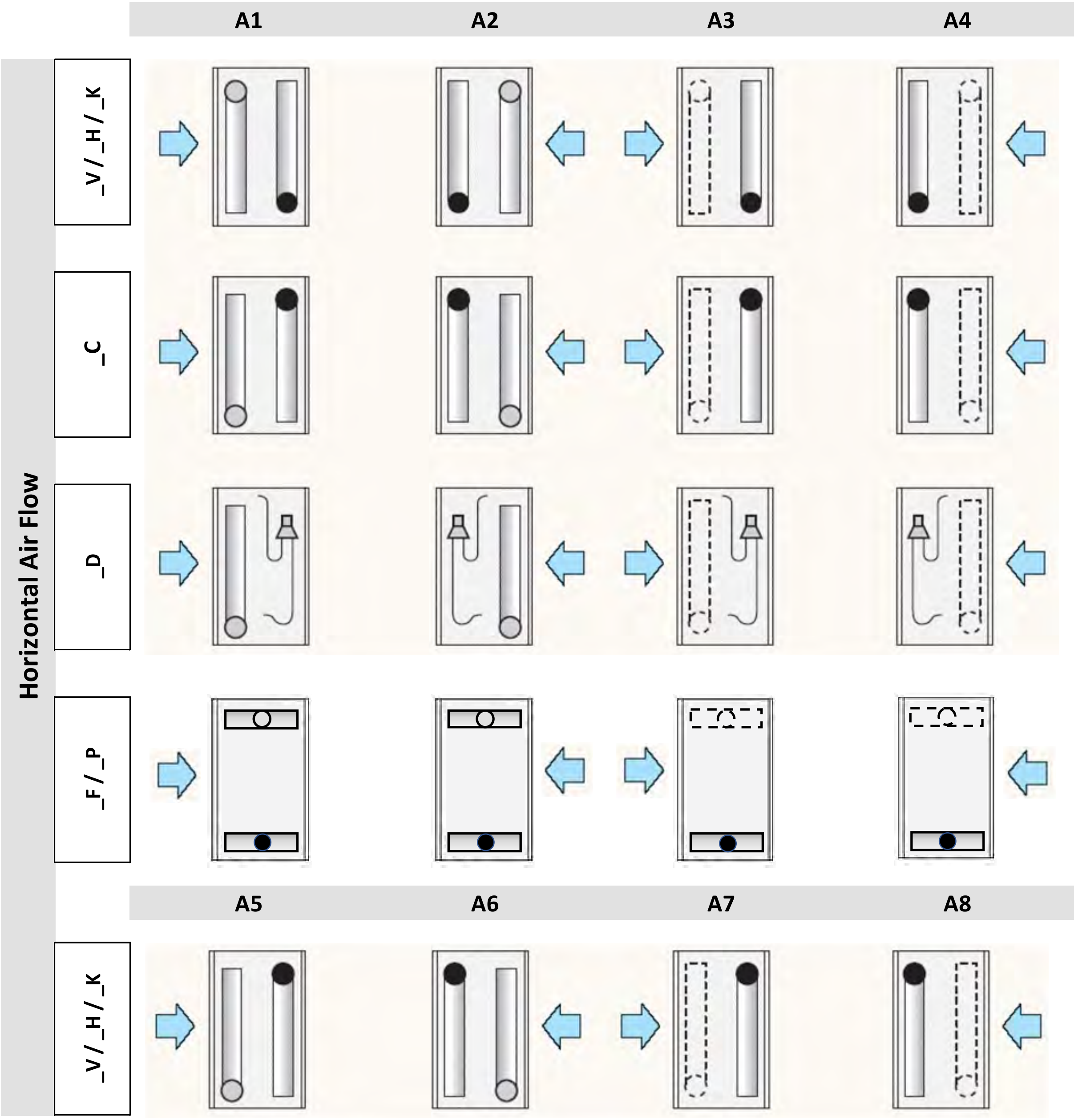
Incidentally, some manufacturers provide 100% clearance holes in their tube plates and support the fin block via longitudinal steel rods that pass through additional formed holes in the fin material that match additional holes in the tube plates. This construction is often referred to as a ‘floating coil’.

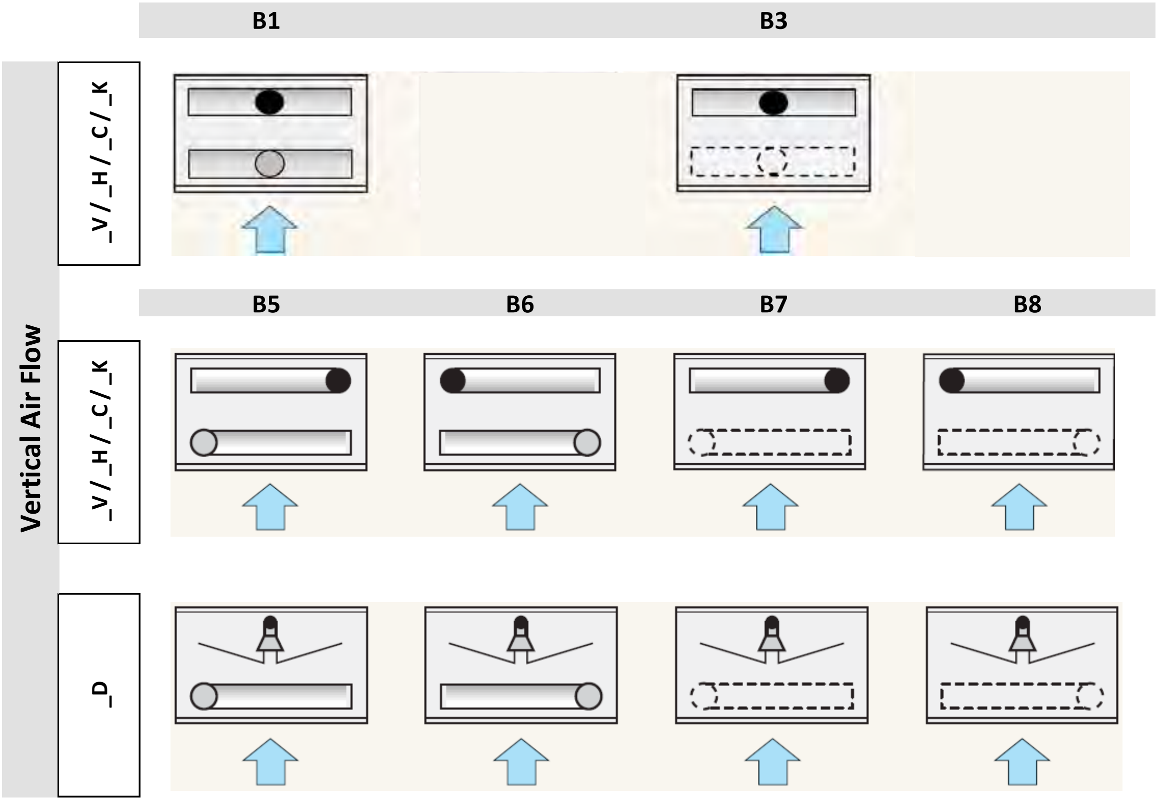
COIL ORIENTATION DESIGNATIONS

The standard coil orientations (also known as Handing) and air flow directions are provided below, however, there can be other variants that meet customer specific requirements.

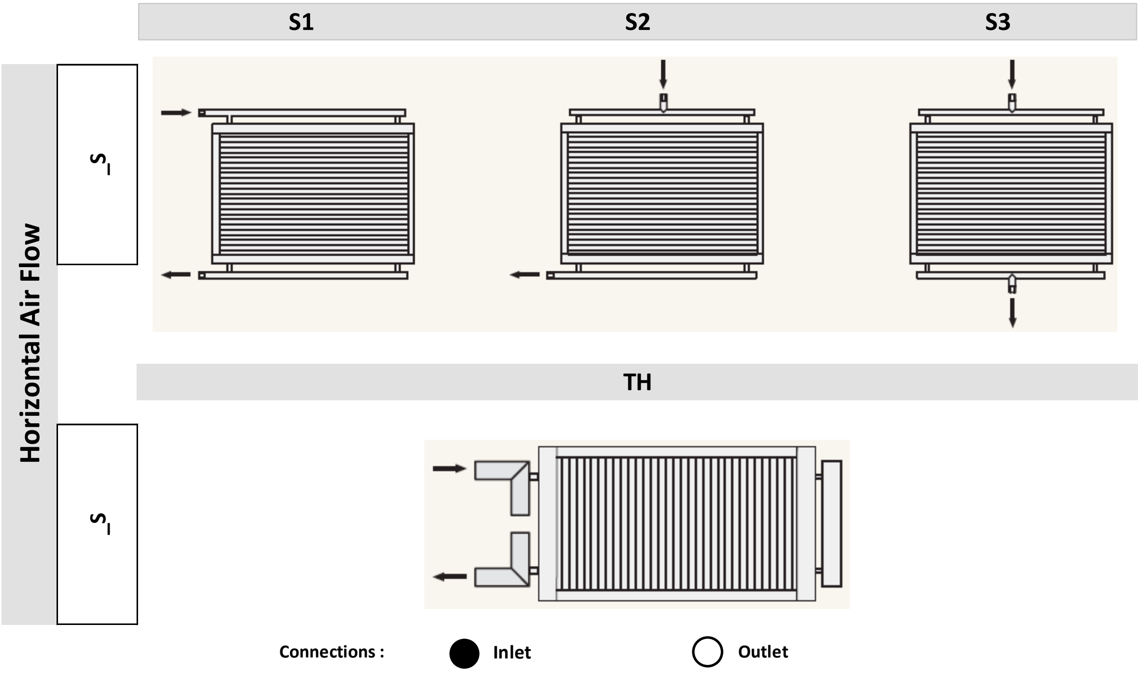
The reference point when considering the coil's orientation/handing is to consider that the observer is looking towards the face of the coil in the direction of the air flow and the connection positions relate to this orientation …

HORIZONTAL AIR FLOW / VERTICAL COIL ORIENTATION OPTIONS

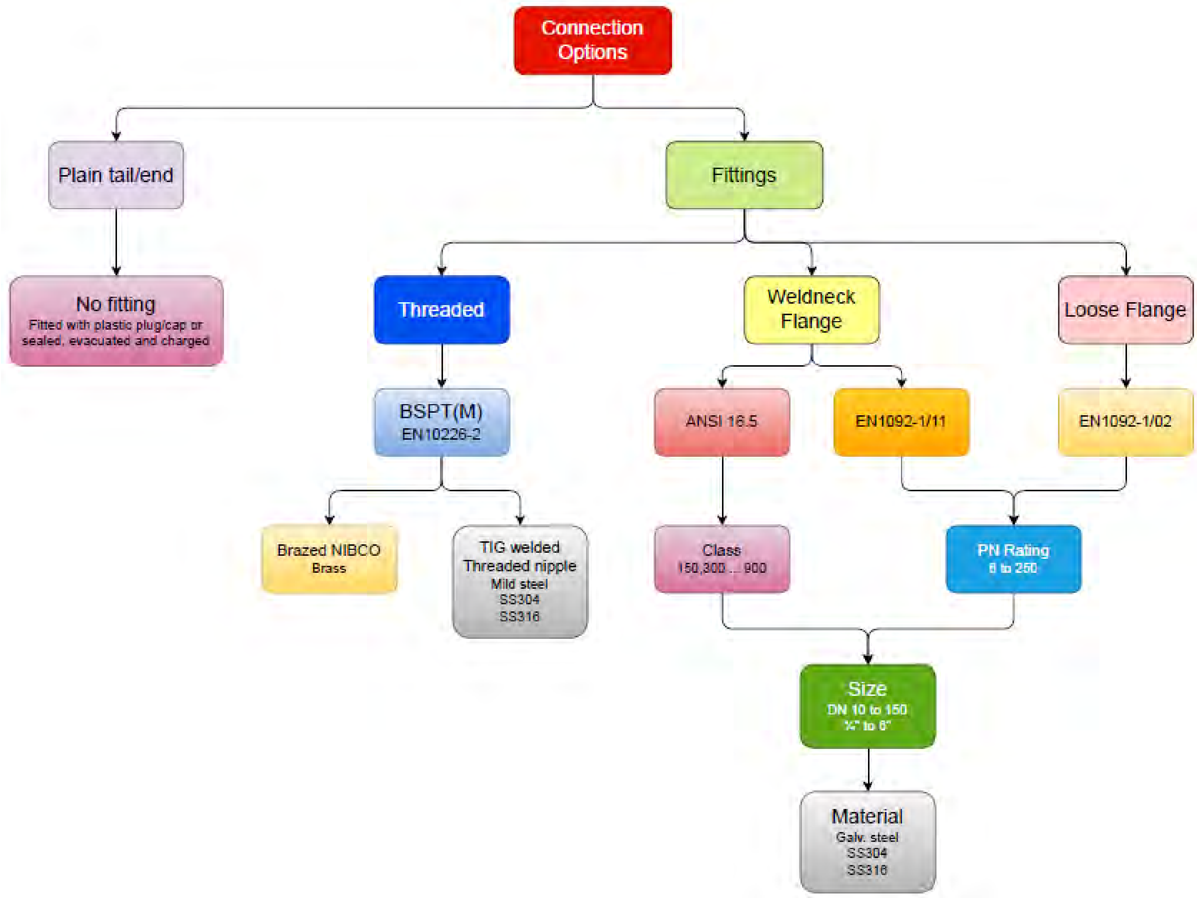




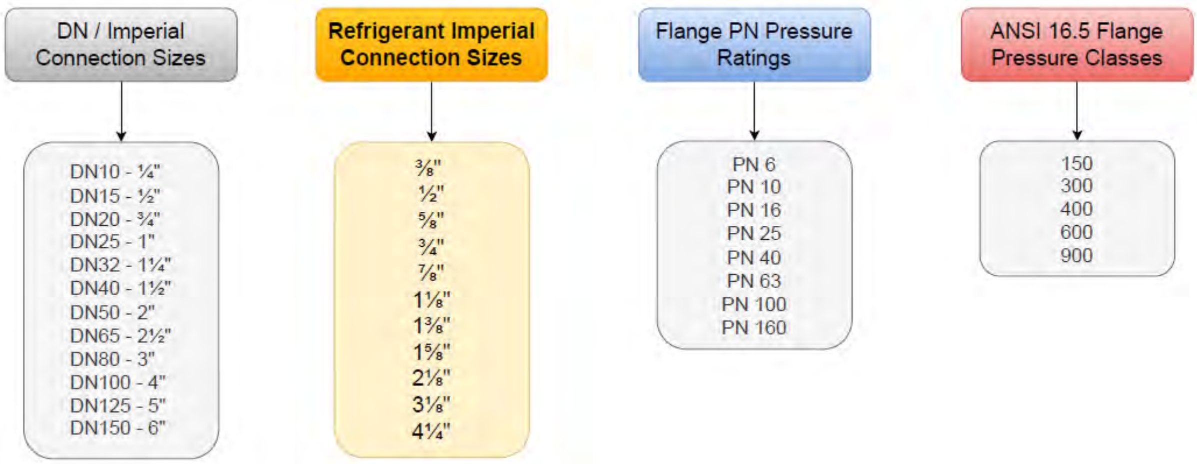
Vertical Coil - Horizontal Air Flow	A1 : RHS same end connections. Inlet at the bottom, outlet at the top A2 : LHS same end connections. Inlet at the bottom, outlet at the top A3 : Opposite end connections. RH inlet at the bottom, LH outlet at the top A4 : Opposite end connections. LH inlet at the bottom, RH outlet at the top A5 : RHS same end connections. Inlet at the top, outlet at the bottom A6 : LHS same end connections. Inlet at the top, outlet at the bottom A7 : Opposite end connections. RH inlet at the top, LH outlet at the bottom A8 : Opposite end connections. LH inlet at the top, RH outlet at the bottom
Horizontal Coil - Vertical Air Flow	B1 : Same end connections. Inlet at the top, outlet at the bottom B3 : Opposite end connections. Inlet at the bottom, outlet at the top B5 : Same end connections. Inlet at the RH top, outlet at the LH bottom B6 : Same end connections. Inlet at the LH top, outlet at the RH bottom B7 : Opposite end connections. Inlet at the RH top, outlet at the LH bottom B8 : Opposite end connections. Inlet at the LH top, outlet at the RH bottom


S1
S2
S3
TH
Horizontal Air Flow
S_
S_
Connections :
Inlet
Outlet

CONNECTION OPTION FLOWCHART



CONNECTION SIZES & PRESSURE RATINGS



CIRCUITRY

BACKGROUND

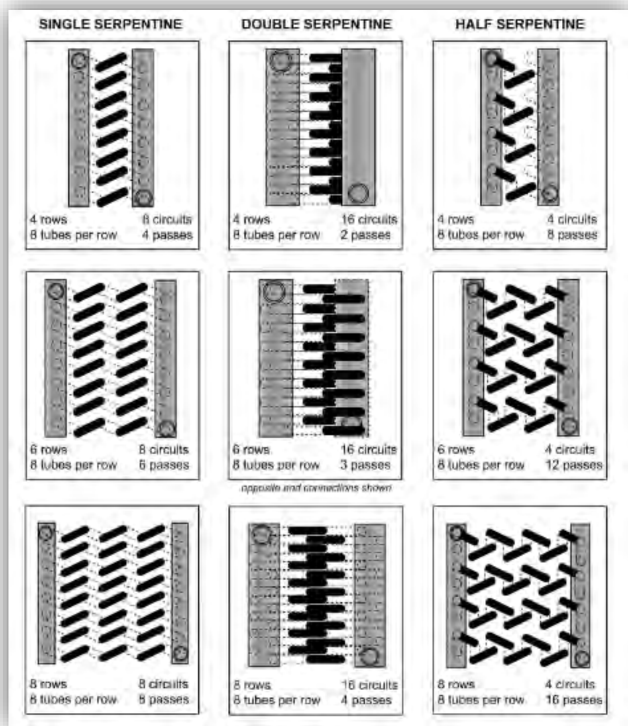
Circuitry is more of an ‘art’ than a science. Certainly there are underlying rules that should be adhered to, but with the wealth of coil manufacturing companies in different countries, a variety of ‘schools of thought’ have evolved over time, which affect the final pattern.

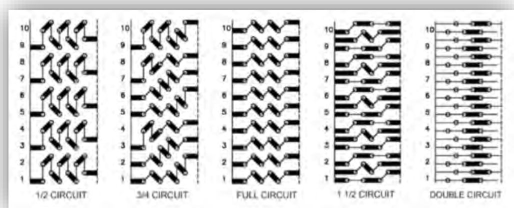
OTHER TERMINOLOGY

Some manufacturers use standardised circuitry patterns for different row combinations and furthermore, use different terminology other than number of circuits or number of passes.

References to single, double or half serpentine obliquely relate to the number of circuits or number of passes but relate more to the number of rows deep of the coil in question.

Furthermore, the same concept is also referred to as ½, ¾, full, 1½ & double circuits by some manufacturers.





Although this alternative terminology is commonly used by a number of manufacturers, EAS complies with the following ‘rules’ …

GENERIC RULES FOR CIRCUITRY PATTERNS

1. Design for Cross Counter Flow - Air flow & Fluid flow in opposite directions
   - Fluid enters on the air leaving side of the coil and exists on the air entering side

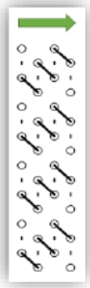
   Exception
   - *1 Row deep coil cannot achieve cross counter flow but is purely Cross Flow*
   - *2 Row deep coils are allowed to have a vertical zig zag pattern*
2. Minimum Number of Passes (Tubes per Circuit) = Integer[Rows Deep / 2 - 0.001] + 1 … for Same End connections
   - o Example : 7 row deep coils require a minimum of 4 Passes, if Same End connections
   - o Would require minimum of 5 Passes, if Opposite End connections

   Exception
   - *5 & 6 Row Deep coils can have 2 Passes (tubes per circuit)*
     - o *5R A Fin – Cu & SS & AL*
     - o *6R A Fin – Cu & SS … not AL*
     - o *5R C Fin – Cu & SS*
     - o *6R C Fin – Cu & SS … special X-Overs*
3. Single phase fluid coils (_V, _K) can utilise a mixed number of passes e.g. [4 x 6T] + [8 x 8T]
4. Circuitry for single phase fluids (e.g. water) must drain naturally
   - Pattern should progress upwards from inlet to outlet with no 'valleys or traps'

Exception

- *Vee type products with inclined coils should have a pattern progress downwards from inlet to outlet*
- *On rare occasions, Customer may allow a pattern with valleys/traps, however, counter flow must be maintained*

5. All Evaporators and Condensers/Gas Coolers (_C, _D, _P, _F) must have an equal number of passes in each circuit
   - Number of 'Blank'/'Unused tubes' should not exceed the number of rows deep
6. Circuitry for Condensers must flow downwards and in counter flow from 'hot gas inlet' to 'liquid condensate' outlet
7. Circuitry for DX Evaporators generally flows upwards from the liquid inlet on the air leaving face to the superheated vapour suction outlet on the air entering face



   - Ensure that the last pass or last few passes feed upwards to prevent any liquid slugging, typically <= 3 tubes high
   - Allow zigzag pattern for > 3 rows
   - Endeavour not to exceed 180 mm circuit height when using zigzag pattern

   Exception

   - *Application may allow for 'zig zag' pattern, but ensure counter flow is maintained & pattern finishes upwards*
   - *Low temperature applications <-20°C may require to be fed downwards to ensure oil is returned to the compressor*
8. Circuitry for Pump Circulated & Flooded Evaporators (_P, _F) must be ‘up fed’ from the lower horizontal liquid header to the upper horizontal suction header
   - Number of circuits is a factor (or multiple) of the Rows Deep

HEADERS DETAILS

HEADER SIZES

	Copper - BSP Thread		Copper - Flange		Aluminium		DIN - Stainless		ASME Sch 10		ASME Sch 40		ASME Sch 80	
Size	OD	Wall	OD	Wall	OD	Wall	OD	Wall	OD	Wall	OD	Wall	OD	Wall
DN15 - ½"	22.00	1.00	18.00	1.00	22.00	2.00	21.30	2.00	21.34	2.11	21.34	2.77	21.34	3.73
DN20 - ¾"	28.00	1.20	22.00	1.00	28.00	2.00	26.90	2.00	26.67	2.11	26.67	2.87	26.67	3.91
DN25 - 1"	35.00	1.50	28.00	1.20	35.00	2.00	33.70	2.00	33.40	2.77	33.40	3.38	33.40	4.55
DN32 - 1¼"	42.00	1.50	35.00	1.50	42.00	2.00	42.40	2.00	42.16	2.77	42.16	3.56	42.16	4.85
DN40 - 1½"	48.00	1.50	42.00	1.50	48.00	2.00	48.30	2.00	48.26	2.77	48.26	3.68	48.26	5.08
DN50 - 2"	64.00	2.00	54.00	1.50	60.00	2.00	60.30	2.00	60.33	2.77	60.33	3.91	60.33	5.54
DN65 - 2½"	76.10	2.00	76.10	2.00	76.00	3.00	76.10	2.00	73.02	3.05	73.02	5.16	73.02	7.01
DN80 - 3"	88.90	2.00	88.90	2.00	90.00	3.00	88.90	2.00	88.90	3.05	88.90	5.49	88.90	7.62
DN100 - 4"	108.00	2.00	108.00	2.00	108.00	3.00	114.30	2.00	114.30	3.05	114.30	6.02	114.30	8.56
DN125 - 5"	133.00	3.00	133.00	3.00			139.70	2.00	143.00	3.40	143.00	6.55	143.00	9.53
DN150 - 6"	159.00	3.00	159.00	3.00			168.30	3.00	168.27	3.40	168.27	7.11	168.27	10.97
DN200 - 8"									219.08	3.76	219.08	8.18	219.08	12.70

	Cu - CD/DX		CuFe2P - K65	
Size	OD	Wall	OD	Wall
3/8"	9.52	0.76	9.52	0.65
1/2"	12.70	0.85	12.70	0.85
5/8"	15.88	0.90	15.88	1.05
3/4"	19.05	0.81	19.05	1.30
7/8"	22.22	1.15	22.22	1.50
1 1/8"	28.58	1.27	28.58	1.90
1 3/8"	34.93	1.45	34.93	2.30
1 5/8"	41.28	1.72	41.28	2.70
2 1/8"	53.98	2.24	53.98	3.50
2 5/8"	66.68	2.77	-	-
3 1/8"	79.38	1.65	-	-
4 1/4"	107.95	2.50	-	-

HEADER INTERNAL VOLUMES

	Internal Volume per Meter - Litre/m						
DN15 - ½"	0.314	0.201	0.254	0.235	0.230	0.196	0.151
DN20 - ¾"	0.515	0.314	0.452	0.412	0.396	0.344	0.279
DN25 - 1"	0.804	0.515	0.755	0.693	0.610	0.558	0.464
DN32 - 1¼"	1.195	0.804	1.134	1.158	1.053	0.965	0.827
DN40 - 1½"	1.590	1.195	1.521	1.541	1.433	1.313	1.140
DN50 - 2"	2.827	2.043	2.463	2.489	2.358	2.165	1.905
DN65 - 2½"	4.083	4.083	3.848	4.083	3.518	3.088	2.734
DN80 - 3"	5.661	5.661	5.542	5.661	5.385	4.770	4.261
DN100 - 4"	8.495	8.495	8.171	9.555	9.196	8.213	7.417
DN125 - 5"					14.568	13.252	12.067
DN150 - 6"					20.475	18.638	16.816
DN200 - 8"					35.153	32.277	29.462

0.050	0.053
0.095	0.095
0.156	0.149
0.239	0.213
0.312	0.290
0.533	0.482
0.806	0.722
1.125	1.011
1.924	1.733
2.935	-
4.546	-
8.324	-

COPPER SIZE VARIATIONS

UNITED STATES, CANADA, AND BRAZIL

Common wall-thicknesses of copper tubing in the U.S., Canada and India are "Type K", "Type L", "Type M", and "Type DWV"

- Type K has the thickest wall section of the three types of pressure rated tubing and is commonly used for deep underground burial, such as under sidewalks and streets, with a suitable corrosion protection coating or continuous polyethylene sleeve as required by the plumbing code. In the United States, it usually has green-coloured printing. This pipe designation is also used in the Refrigeration Industry.

- Type L has a thinner pipe wall section and is used in residential and commercial water supply and pressure applications. In the United States, it usually has blue-coloured printing.
- Type M has an even thinner pipe wall section and is used in residential and commercial low-pressure heating applications. In the United States, it usually has red-coloured printing.
- Type DWV has the thinnest wall section and is generally only suitable for unpressurized applications, such as drain, waste, and vent (DWV) lines. In the United States, it usually has yellow or light orange coloured printing, common sizes being 1½”, 1¼” and 2-inch copper tube size.

Types K and L are generally available in both hard drawn straight sections and in rolls of soft annealed tubing, whereas type M and DWV are usually only available in hard drawn straight sections.

In the North American plumbing industry, the size of copper tubing is designated by its nominal diameter, which is 1/8” less than the outside diameter. The inside diameter varies according to the thickness of the pipe wall, which differs according to pipe size, material, and grade: the inside diameter is equal to the outside diameter, less twice the wall thickness.

The North American refrigeration industry uses copper pipe designated ACR (air conditioning and refrigeration field services) pipe and tubing, which is sized directly by its outside diameter (OD) and a typed letter indicating wall thickness. Therefore, one-inch nominal type L copper tube and 1 1/8” inch type D ACR tube are exactly the same size, with different size designations. ACR pipe is manufactured without processing oils that would be incompatible with the oils used to lubricate the compressors in the air conditioning system.

Except for this difference between ACR (types A and D) and plumbing (types K, L, M, and DWV) pipes, the type only indicates wall thickness and does not affect the outside diameter of the tube. Type K ½”, type L ½” and type D 5/8” ACR all have the same outside diameter of 5/8”.

EUROPE

Common wall-thicknesses in Europe are "Type X", "Type Y", and "Type Z", defined by the EN 1057 standard.

- Type X is the most common and is used in above-ground service, including drinking water supply, hot and cold water systems, sanitation, central heating, and other general purpose applications.
- Type Y is a thicker walled pipe, used for underground works and heavy duty requirements, including hot and cold water supply, gas reticulation, sanitary plumbing, heating and general engineering.
- Type Z is a thinner walled pipe, also used for above-ground service, including drinking water supply, hot and cold water systems, sanitation, central heating and other general purpose applications.

In the plumbing trade, the size of copper tubing is measured by its outside diameter in millimetres. Typical sizes are 15 mm, 18 mm, 22 mm, 28 mm, 35 mm, 42 mm, 54 mm, 66.7 mm, 76.1 mm and 108 mm outside diameters.

CONNECTION TYPES

DEFINITIONS

There are many abbreviations used to identify pipe sizes and these differ from Imperial to Metric pipe sizes and differ depending upon the pipe material i.e. stainless steel or copper.

In America NPS or ‘Nominal Pipe Size’ is often incorrectly named ‘National Pipe Size’ due to the confusion with NPT, which refers to ‘National Pipe Thread’.

To confuse matters further, the use of NPS does not conform to the American Standard (ASME B36.10M & B36.19M) pipe designations, where NPS means 'National Pipe Thread Straight'.

Size		Outside Diam.	
DN	NPS	mm	inch
15	½"	21.34	0.840
20	¾"	26.67	1.050
25	1"	33.40	1.315
32	1¼"	42.16	1.660
40	1½"	48.26	1.900
50	2"	60.33	2.375
65	2½"	73.02	2.875
80	3"	88.90	3.500
100	4"	114.30	4.500
125	5"	143.00	5.630
150	6"	168.27	6.625
200	8"	219.08	8.625
250	10"	273.05	10.750

Connection(s) is the generic terminology used to describe how or what 'couplings' may or may not be factory fitted to the coil or equipment's inlet and outlet piping or manifolds/headers to serve as an interface to the customer's pipework or system.

The type of connection(s) supplied with a coil or product depends upon both the application and customer requirements.

Generally, the scope of the DN & Imperial pipe sizes, to which the connection couplings are designed to match, is given in the table. Furthermore, the outside diameter for ASME scheduled steel pipes is also given. *Note that the outside diameter differs slightly for the DIN standard equivalent sizes.* *See also Header Sizes*

Although smaller BSP/DN sizes do exist, for our industry sector, ½" (DN15) is the practical smallest connections size. However, some refrigerant coil pressure and charging fittings, such as Schrader valves are often ¼".

Typically, fluid heating and cooling coils are provided with male threaded (BSPM) fittings up to 2" (DN50) and flanged connections for 2½", 3" & 4" (DN 65, 80 & 100).

Evaporators and condensers are usually supplied with 'plain tails' i.e. no fittings, just the plain header connection tube.

For refrigeration systems, the interconnecting pipework is brazed or welded directly to the coil/product 'plain tails'. However, on rare occasions, special refrigeration fittings are requested.

Steam coils would often be supplied with flanged connections, however for low pressure steam applications, threaded connection may be specified.

PLAIN TAILS

In its simplest form, no 'fittings' are supplied, and a 'plain tail' or 'plain end' is provided, which will typically be 100 to 150 mm in length and has a similar diameter to the coil header diameter … but not always.

On occasions, BFW preparation is specified for steel plain tail connections. BFW is an abbreviation for 'bevelled for welding' and involves machining a 45° bevel on the end of the connection tube in readiness for welding to the customer's site pipework.

Refrigerant coils usually have their inlet and outlet connection tubes sealed with a blanking end cap to enable the coil to be evacuated and charged with dry nitrogen/air to ensure that the coil is internally dry when the seal is broken just prior to brazing/welding into the refrigeration system.

THREADED – BSP/DN

Threaded, otherwise known as screwed connections, have their origins in the water distribution industry sector and are generally referred to in Imperial sizes, albeit that there is an equivalent European metric designation … see above.

To complicate matters, threaded fittings can either have 'parallel' (BSP) or 'tapered' (BSPT) threads. Furthermore, the thread can be external/male (BSPM) or internal/female (BSPF).

The most common screwed connections are BSPM with parallel threads, but the customer should specify exactly what is required.

As mentioned above, generally screwed connections are offered up to and including 2" (DN50). Sizes greater than this would usually be offered with flanged fittings for the following reason.

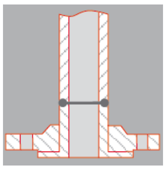
The torque required to fully tighten/seal a large diameter screwed connection is easily capably of over stressing a copper coil's header and tube brazed joints, resulting in leakage. Therefore, fitting a flange coupling minimises this risk.

However, for AHU applications, where coils are designed to slide into an AHU assembly, the removable coil cover-plates are more manageable if the coils are fitted with screwed couplings and then the 'screw-on' flanges re-fitted, providing the more *desirable flanged connection. So, 2½, 3 & 4” connections are indeed a hybrid fitting with a threaded nipple on to which is screwed a ‘screw-on’ flange.*

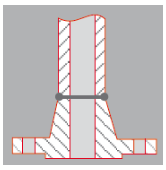
FLANGED

Flanged coupling or fittings can either be DIN (European) or ANSI (American) and are available in a variety of types …

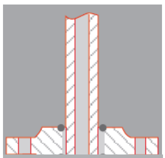
- Loose/spinning/lap joint flange
  Used in conjunction with a ‘lap joint stub end’ and are similar to slip-on flanges with the exception of a radius at the intersection of the flange face and the bore to accommodate the flanged portion of the stub end. Used in applications where the joint must be frequently disassembled or where bolt alignment is important



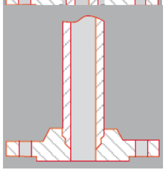
- Weld-neck flange
  Designed with a hub on the backside tapering to a diameter matching the mating pipe and bored to match the pipe’s internal diameter. Used in more severe applications than loose flanges



- Slip-on or lap flange
  Designed to slip over the outside diameter of the pipe to which it will be welded. They are attached to the pipe by fillet welds at the hub and at the end of the pipe inside the flange. Not usually used for high stress applications



- Screw-on flange
  Threaded in the bore to match the external thread on the pipe. The threads are tapered to create a seal between the flange and pipe. Normally used on low pressure, non-cyclic applications or in applications where welding is hazardous



Each type of flange is available in the sizes given above and a variety of materials. Furthermore, both DIN & ANSI flanges have a variety of pressure ratings, which affect the thickness of the flange and/or the number of bolts used to secure it.

The face of the flange that ‘mates’ with the ‘counter flange’ on the interconnecting pipework, can have a flat face or can be grooved or more typically can be ‘raised face’. The American ANSI B16.5 RF is an example and often stipulated for off-shore and oil & gas industry applications.

EN 1092/DIN STANDARD FLANGES

Traditionally in Europe, bolted flanged connections have been referred to as DIN standard flanges, which was indeed the case as the governing standard was a German DIN standard. Furthermore, ISO 7005 considered both DIN and ASME standard flanges.

However, over time, the European Union has consolidated international and national standards into EN standards and now flanges are covered by EN 1092 and EN 1759.

Generally, flange connections often requested fall under EN 1092 - Part 1 for steel flanges and Part 4 for Aluminium Alloy flanges.

The pressure ratings for EN 1092 flanges are PN 2.5, 6, 10, 16, 25, 40, 63 & 100 are nominal pressure ratings in barg at ambient temperatures and it is important to appreciate that at elevated temperatures, the tensile strength of any material will reduce, and this will lower the ‘real’ pressure handling capability. So, both temperature and pressure are the major considerations when determining whether a flange is ‘fit for purpose’.

Just because the maximum allowable pressure (PS) is 10 barg **does not** confirm that a flange rating of PN10 is strong or indeed safe enough for the application.

Type No	Description
01	Plate flange for welding
02	Loose plate flange with weld-on plate collar or for lapped pipe end
04	Loose plate flange with weld-neck collar
05	Blank flange
11	Weld-neck flange
12	Hubbed slip-on flange for welding
13	Hubbed threaded flange
21[a]	Integral flange
32[b]	Weld-on plate collar
33[b]	Lapped pipe end
34[b]	Weld-neck collar

[a] Flange type 21 is an integral part of some other equipment or component.

[b] Ancillary components type numbers 32 and 33 are for use with type 02 flanges and type number 34 for use with type 04 flanges.

NOTE Type numbers have been made non-consecutive to permit possible future additions.

EN Flanges have a variety of flange face finishes, which can be smooth, grooved or ‘raised face’, which is the most common design that we encounter.

The method of attachment of the flange to the piping or connection tube of the coil is catered for via alternative flange designs that are given Type numbers, the most common being …

- Type 11 : Weld-neck flange
- Type 01 : Plate flange for welding : *not used with Cat II to IV coils*
- Type 02 : Loose/spinning flange
- Type 13 : Hubbed threaded flange : *screw-on flange*

ASME/ANSI FLANGES

ASME B16.5 is an American standard for Pipe Flanges and Flanged Fittings that covers flanges sizes from NPS ½” to 24”. (*NPS – Nominal Pipe Size which also relates to BSP – British Standard Pipe size*).

In this standard, flanges are classified based on their pressure-temperature rating, which is also known as a flange class.

As ASME is an American Standard, all temperatures are given in °F and pressures are in psig (pound per square inch gauge). Furthermore, psig is shortened to lb, which is the English symbol for pound. Additionally, sometimes lb is replaced by #.

Pressure-temperature rating

Pressure-temperature rating is the maximum allowable non-shock gauge pressure at the specific temperature for a given material.

As per ASME B31.3, ratings are maximum allowable working gauge pressures at the given temperature and for the applicable material and pressure class.

Temperature Rating

Piping materials such as carbon steel, stainless steel & alloy steel have different mechanical and chemical properties. The same material can handle different amounts of stress at a different temperature. Based on the ability of a material to handle the stresses at a given temperature, ASME B31.3 has devolved the maximum allowable stress value for a material at a specific temperature.

The purpose behind establishing the temperature rating is to calculate the adequate wall thickness of the pipe, flange and flanged fittings so that they can withstand the stresses due to pressure and other loads.

Pressure Rating

The pressure rating is the safe working or maximum operating pressure with respect to the working temperature. It depends upon the material’s Stress-Strain characteristics.

Flange Rating

ASME B16.5 lists the Pressure-Temperature ratings for flanges. These ratings are established via hydro testing of the flanged fittings to the bursting and by adding a factor of safety of 3.0 at the rated working pressure and ambient temperature.

ASME/ANSI B16.5 define these temperature-pressure ratings by using the following formula …

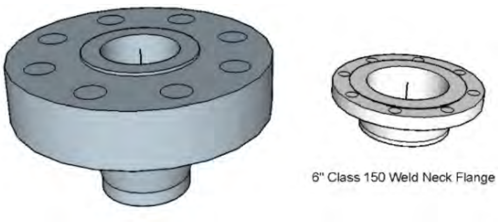
PT = (Pr x SI) / 8750

where, PT = rated working pressure (psig) for specified material at temperature T
Pr = pressure rating as per Class in psig
SI = stress in psig for specified material at temperature T

Flange Class

ASME has developed a Flange Class related to temperature and pressure rating. There are 7 divisions Class 150#, 300#, 400#, 600#, 900#, 1500# & 2500#.

The higher the flange ratings the heavier the flange and the greater ability to withstand higher pressures and temperatures. When the temperature goes up the maximum allowable pressure goes down, and vice versa. See the image showing the comparison of 6” Class 150 and Class 2500 weld-neck flange.



6" Class 2500 Weld Neck Flange

6" Class 150 Weld Neck Flange

ANSI flange ratings

The pressure classes of flanges are commonly referred to in terms of "pounds" rather than "pounds per square inch." In describing the pressure class of flanges, the terms "pound" and "class" may be considered interchangeable.

Flanges pressure rating in psig for common carbon steel ASME flanges (A105 / A350 LF2 / A350 LF3 / A350 LF6)							
Temperature (F°)	150#	300#	400#	600#	900#	1500#	2500#
-20 to 100	285	740	985	1480	2220	3705	6170
200	260	680	905	1360	2035	3395	5655
300	230	655	870	1310	1965	3270	5450
400	200	635	845	1265	1900	3170	5280
500	170	605	805	1205	1810	3015	5025
600	140	570	755	1135	1705	2840	4730
650	125	550	730	1100	1650	2745	4575
700	110	530	710	1060	1590	2655	4425
750	95	505	675	1015	1520	2535	4230
800	80	410	550	825	1235	2055	3430
850	65	320	425	640	955	1595	2655
900	50	230	305	460	690	1150	1915
950	35	135	185	275	410	685	1145
1000	20	85	115	170	255	430	715

ASME / ANSI 150 pressure rating

As an example, a class 150 flange suggests that the safe working pressure for this flange at the rated temperature and defined material is 150 pound per square inch.

But the same flange can be referred to in the following ways …

- Class #150 flange
- 150 pound flange
- Class 150 flange pressure rating
- 150 pressure rating flange
- 150 lb flange

Furthermore, as an example assume that we need an ASTM A105 flange (carbon steel) for use at a pressure of 1,200 psig (82.7 barg) at 500°F (260°C).

Reference to the ASME pressure-temperature table for carbon steel indicates we need a flange with 600# rating to meet the requirement.

Highlighted in Green is a Class 600# flange which can withstand 1205 psig pressure at 500 °F.

VITAULIC COUPLING

Is a mechanical piping system that offers an alternative to welding, threading and flanging for joining two pipe ends. This coupling is occasionally specified for single phase fluid dry coolers and fluid heating or cooling coils.

The grooved pipe joining systems uses a roll grooving technique to join pipes and pipe joining components.

A groove is either machined on the end of two pipes or a grooved nipple is butt welded to the pipe end to prepare the pipe's engagement with the coupling housing and gasket. The gasket creates a pressure responsive seal on the outside diameter of the pipe, unlike standard compression joints, where pressure acts to separate the seal.

The gasket sealing is enhanced as the coupling housing is tightened onto the pipe end.

The coupling provides simplified assembly involving three basic concepts: a pressure responsive gasket that creates a leak-tight seal, couplings that hold the pipe together and fasteners that secure the couplings.

JOINTING PROCESSES

Generally non-ferrous materials are brazed or silver soldered, whilst ferrous materials such as high tensile black steel or stainless steels or indeed titanium are traditionally welded.

BRAZING & SOLDERING

Brazing usually refers to a jointing process when a filler rod is used which is not the same material as the base metal and which melts above >450°C.

Soldering, by definition is the same principle, but at temperatures below <450°C. Soldering is not a process performed during the manufacture of EAS heat exchangers

The most common brazing rods used for typical HVAC brazing are 0%, 5% and 15% plus 38% or indeed 55%.

The percentage is the percentage of silver content in the rod. The only real reason to use lower content silver is the cost implications.

Usually, for 'copper to copper' joints, CopperPhos rods (Cu/P/Ag - *Copper with phosphorous and silver*) are used which do not need flux or even benefit from it. The phosphorus content allows the rod to self-flux the copper eliminating the need for acidic based fluxes which when overused, can contaminate the system and cause more harm than good.

Flux is indeed required when joining brass to copper with >=15% silver rods. Such rods are often coated in flux, eliminating the need to apply a wet flux to the joint.

The silver in the rods increase the 'ductility' of the filler and allows it to flow at a slightly lower temperature. This results in a better joint and reduces the propensity of cracking due to thermal expansion, contraction or even vibration. The increased silver also allows the solder to remain strong when filling slightly larger gaps due to ill-fitting copper parts.

WELDING

There are variety of welding methods such as …

- Oxy-acetylene gas welding
- Electric arc stick welding
- MIG
- TIG

MIG is generally used for welding black steel and sheet metal or high tensile steel headers, whilst TIG is used for aluminium and stainless steel sheet metal, headers and tubes plus titanium tubes and headers.

Unusually, compared with many competitors, EAS TIG welds copper header mitred joints, fabricated Tee connections and end caps.

Furthermore, black steel fittings e.g. threaded nipples and stainless steel weld neck flanges etc. are commonly TIG welded to copper header connection tubes.

A common misunderstanding regarding the welding of dissimilar materials e.g. copper and black steel, is corrosion related issues. As the two dissimilar materials are in intimate intermolecular contact, without the presence of a dielectric, there is no propensity for corrosion at the welded joint.

FUNDAMENTAL DEFINITIONS

This section describes and/or explains the terminology specifically used in the coil industry, which may differ from the interpretation in other market sectors.

However, for clarity, a brief introduction to the Système international d'unités (SI system) of units.

Fundamentally the SI system uses units of measurement often named after a person, typically a scientist, physicist, chemist, engineer etc. Therefore such units, for example *Kelvin, Pascal, Newton, Watt, Joule* are denoted by abbreviations of their names that begin with a capital letter such as .. *K, Pa, N, W, J* respectively.

Denominations of the basic unit such as fractions … 1/10, 1/100, 1/1000 or multiples such as … 10, 100, 1000 etc. are denoted by deci, centi, milli and deca, hecto, kilo respectively, which are designated by small letters such as *d, c, m* & *da, h, k*.

However, just to confuse matters still further, all multiples above 1000 (k) are denoted by a capital letter e.g. Mega (M - 10^6), Giga (G - 10^9), Tera (T - 10^{12}).

So, we can have scenarios such as …

- kg - kilogram - 1000 grams
- MW - MegaWatt - 10^6 Watts
- mPa - milliPascal - 10^{-3} Pascals
- Tb - Terabyte - 10^{12} bytes

BASIC UNITS OF MEASUREMENT

Apart from the kilogram, the Système international d'unités (SI) has several other base units of measurement, all now derived from physical processes. These include …

- **Time** - The base unit of time is the second, which was once defined as 1⁄86,400 of an average day. However, fluctuations in the Earth's rotation made it difficult to measure precisely. It is now defined as being the time taken for a Caesium atom to vibrate 9,192,631,770 times.
- **Length** - The metre was first defined as one ten-millionth of the distance from the equator to the north pole and converted into a one meter long platinum bar held in Paris. Now it defined as the length travelled by light in a vacuum in 1/299,792,458th of a second.
- **Temperature** - The basic unit of temperature is the Kelvin, which is defined as being 1/273.16th of the temperature of water at 0.01°C - the *triple point*.
- **Electric current** - The ampere is defined in terms of the current required to generate a certain force between two wires set at one metre apart.

Two other units are included in the list: the amount of a substance in a given sample is measured in **moles** and luminous intensity is measured in **candelas**, but these units of measurement and certainly the latter item, are not relevant to our industry.

A list of the more common elements. Atomic mass given in g/mol (ref. ^{12}C)

Atomic No.	Element	Symbol	Atomic Mass
1	Hydrogen	H	1.0079
2	Helium	He	4.0026
3	Lithium	Li	6.9410
4	Beryllium	Be	9.0122
5	Boron	B	10.8110
6	Carbon	C	12.0107
7	Nitrogen	N	14.0067
8	Oxygen	O	15.9994
9	Fluorine	F	18.9984
10	Neon	Ne	20.1797
11	Sodium	Na	22.9897
12	Magnesium	Mg	24.3050
13	Aluminum	Al	26.9815
14	Silicon	Si	28.0855
15	Phosphorus	P	30.9738
16	Sulfur	S	32.0650
17	Chlorine	Cl	35.4530
19	Potassium	K	39.0983
18	Argon	Ar	39.9480
20	Calcium	Ca	40.0780
21	Scandium	Sc	44.9559
22	Titanium	Ti	47.8670
23	Vanadium	V	50.9415
24	Chromium	Cr	51.9961
25	Manganese	Mn	54.9380
26	Iron	Fe	55.8450
28	Nickel	Ni	58.6934
27	Cobalt	Co	58.9332
29	Copper	Cu	63.5460
30	Zinc	Zn	65.3900
31	Gallium	Ga	69.7230
32	Germanium	Ge	72.6400
33	Arsenic	As	74.9216
34	Selenium	Se	78.9600
35	Bromine	Br	79.9040
36	Krypton	Kr	83.8000
42	Molybdenum	Mo	95.9400
45	Rhodium	Rh	102.9055
46	Palladium	Pd	106.4200
47	Silver	Ag	107.8682
48	Cadmium	Cd	112.4110
50	Tin	Sn	118.7100
51	Antimony	Sb	121.7600
53	Iodine	I	126.9045
54	Xenon	Xe	131.2930
55	Cesium	Cs	132.9055
56	Barium	Ba	137.3270
74	Tungsten	W	183.8400
77	Iridium	Ir	192.2170
78	Platinum	Pt	195.0780
79	Gold	Au	196.9665
80	Mercury	Hg	200.5900
82	Lead	Pb	207.2000

GRAVITATIONAL FORCE

In chemistry and often at a fundamental level, Earth's gravitational field exerts forces upon any object within its reach, which dictates the magnitude of the gravitational acceleration.

Earth's gravitational acceleration, **g** ≈ 9.81 m/s^2, however this figure changes slightly depending upon the location on the Earth's surface.

MASS & WEIGHT

Mass (**m**) and weight (**w**) are often considered to be interchangeable, but mass is measured in kilograms (kg) and according to Einstein '*can neither be created nor destroyed*' and is an intrinsic property of the object and has a definitive magnitude.

Alternatively, weight is actually a force, measured in Newtons and represents the object's resistance to free fall in a gravitational field. Thus the 'weight' of an object is really **m** x **g**, but in a constant gravitational field the weight of an object is proportional to its mass and referring to the object's mass as being also its weight is not usually problematic!

In reality when you stand on your home scales and 'your weight' shows 70 kg, the scales are really measuring the force that your body mass is exerting i.e. 70 x 9.81 = 686.7 N, but this is converted back to kilograms for convenience, thus displaying your weight as 70 kg.

However, due to the Earth's change in gravitational field around the world your 'weight' of 70 kg would show more than or less depending upon your location, even though your mass remains the same at 70 kg.

In our industry when performing load and strength calculations, it is the force (weight) in Newtons that is of importance.

For approximate calculations use **g ≈ 10 m/s^2**, so a 10 kg mass exerts a force of ≈100 N.

MASS - KILOGRAM

The kilogram is the only base unit of measurement that has continued to be defined by a manufactured object (*until recently*), a platinum-iridium alloy cylinder located in *Pavillon de Breteuil, Paris*.

Its existence can be traced to the decision by Louis XVI to support a new system of weights and measures, one that would be independent of vagaries such as the length of a monarch's arm. This metric system was put forward in 1791 with the kilogram originally being defined in terms of the mass of a one litre (1 dm^3) of water at 0°C.

For the next 100 years, the kilogram was redefined on a number of occasions, while a gradually increasing number of nations joined France in adopting the metric system, stimulated by growth in international trade and the need to standardise the weights of manufactured goods. Eventually, the international metre convention was signed in 1875 by 17 nations.

Since then, there have been many efforts to redefine the kilogram in terms of a fundamental measurement – the proposal being defined in terms of an electric current.

Recent work carried out by the late *Bryan Kibble* at the National Physical Laboratory, conceived the basic concepts of a device that will replace the physical standard kilogram located in Paris.

The '*Kibble balance'* works by measuring the electric current that is required to produce an electromagnetic force equal to the gravitational force acting upon a mass. A second stage allows the electromagnetic force to be determined in terms of a fundamental constant known as the *Planck constant, h = 6.62607015 x 10^{-34} kg.m^2/s (also J.Hz^{-1} or J.s)* which will, in future, be used to define a kilogram. These machines will provide the standard for weighing objects.

Incidentally, now that the definition is explicitly linked to the Planck constant, then the kilogram is effectively defined in terms of the second and the metre … see the units of Planck's constant.

This new definition was accepted on 16th November 2018 and officially came into force on 20th May 2019 … along with the new definition of the Avogadro number.

In chemistry and often at a fundamental level of physics, quantities of gases or liquids are referred to in terms of their molar mass or mole fraction.

By definition, a mole (symbol **mol**) of any substance is defined as the amount of that substance that contains exactly 6.022 140 76 x 10^{23} elementary entities, defined by Avogadro's number.

It is common chemical engineering practice to use the kilomole (kmol), which was numerically identical to the kilogram-mole (until the 2019 redefinition of SI units, which redefined the mole to be very nearly equivalent to but no longer exactly equal to the gram-mole, defined as the number of entities in 12 g of ^{12}C), but whose name and symbol adopt the SI convention for standard multiples of metric units – thus, kmol means 1000 mol.

Concepts

The mole corresponds to a given count of particles. Usually the particles counted are chemically identical entities, individually distinct. For example, a solution may contain a certain number of dissolved molecules that are more or less independent of each other.

However, in a solid the constituent particles are fixed and bound in a lattice arrangement, yet they may be separable without losing their chemical identity. Thus the solid is composed of a certain number of moles of such particles. In yet other cases, such as diamond, where the entire crystal is essentially a single molecule, the mole is still used to express the number of atoms bound together, rather than a count of molecules. Thus, common chemical conventions apply to the definition of the constituent particles of a substance, in other cases exact definitions may be specified. The mass of a substance is equal to its relative atomic (or molecular) mass multiplied by the molar mass constant, which is almost exactly 1 g/mol.

Depending upon the nature of the substance, an elementary entity may be an atom, a molecule, an ion, an ion pair, or a subatomic particle such as a proton. For example, 10 moles of water (a chemical compound) and 10 moles of mercury (a chemical element) contain equal amounts of substance, and the mercury contains exactly one atom for each molecule of the water, despite the two substances having different volumes and different masses.

The mole is widely used in chemistry as a convenient way to express amounts of reactants and amounts of products of chemical reactions. For example, the chemical equation $2H_2 + O_2 \rightarrow 2H_2O$ can be interpreted to mean that for each 2 mol dihydrogen (H_2) and 1 mol dioxygen (O_2) that react, 2 mol of water (H_2O) are formed.

Before the 2019 redefinition of the SI base units, the mole was defined as the amount of substance of a system which contains as many elementary entities as the number of elementary entities in 12 grams of carbon-12 [^{12}C - the most common isotope of carbon].

The term gram-molecule was formerly used to mean one mole of molecules, and gram-atom for one mole of atoms. For example, 1 mole of $MgBr_2$ is 1 gram-molecule of $MgBr_2$ but 3 gram-atoms of $MgBr_2$.

Molar mass

The molar mass of a substance is the ratio of the mass of a sample of that substance to its amount of substance. The amount of substance is given as the number of moles in the sample. For most practical purposes, the numerical value of the molar mass expressed with the unit gram per mole (g/mol) is the same as that of the mean mass of one molecule of the substance expressed with the unit **Dalton**. For example, the molar mass of water is 18.015 g/mol.

The number of moles of a substance in a sample is obtained by dividing the mass of the sample by the molar mass of the compound. For example, 100 g of water equates to 100 / 18.015 = ~5.551 mol of water.

Molar fraction

The molar fraction or mole fraction of a substance in a mixture (such as a solution) is the number of moles of the compound in one sample of the mixture, divided by the total number of moles of all components.

For example, if 20 g of NaCl is dissolved in 100 g of water, the amounts of the two substances in the solution will be (20 g)/(58.443 g/mol) = 0.34221 mol and (100 g)/(18.015 g/mol) = 5.5509 mol, respectively; and the molar fraction of NaCl will be 0.34221/(0.34221 + 5.5509) = 0.05807.

In a mixture of gases, the partial pressure of each component is proportional to its molar ratio.

FLUID - DEFINITION

A fluid is referred to in this document by the English 'physics' definition, which assumes that it can be a gas, a liquid, a vapour or indeed plasma. This definition differs from its interpretation in other languages, where a fluid infers a 'liquid only' state.

TEMPERATURE

Temperature denotes the degree of hotness or coldness of a system and is typically measured in degrees Celsius (*named after the Swedish astronomer Anders Celsius*). This scale is also referred to as degrees 'centigrade' and was based two known fixed parameters, water freezing at 0°C and boiling at 100°C.

The absolute temperature scale using this system is known as the Kelvin scale (*named after the English physicist William Thomson, later known as Lord Kelvin*) and defines absolute zero as 0K or -273.15°C. Thus, on the Kelvin scale, water freezes at +273.15 K and boils at +373.15 K.

Temperatures given in centigrade or Celsius are denoted by °C, whilst absolute temperatures are followed only by K … there is no ° symbol used.

Historically, both Celsius and Fahrenheit (*a Dutch-German-Polish physicist*) were separately working upon defining a temperature scale during the early 1700s. Fahrenheit proposed a scale based upon the melting point of an ammonium chloride brine solution defining 0°F and the average human body temperature defining 100°F, albeit that over the years the scale has been adjusted and now equates to water boiling at 212°F and freezing at 32°F.

Furthermore, similar to the Celsius/Kelvin scenario, the absolute scale in degrees Fahrenheit is known as the Rankine scale (*named after the Glaswegian physicist William John Macquorn Rankine*), where absolute zero is defined as 0°R *[0°Ra]* or -459.67°F.

The Fahrenheit and Celsius temperatures scales converge at -40°C/-40°F and the conversion performed using …

°F = °C x 1.8 + 32 or **°C = (°F - 32) / 1.8**

DRY BULB TEMPERATURE

Is defined as the temperature measured by a thermometer/sensor freely exposed to air but shielded from radiation and moisture. It indicates the amount of heat in the air and is directly proportional to the mean kinetic energy of the air molecules. As described above, it is measured in °C or K.

WET BULB TEMPERATURE

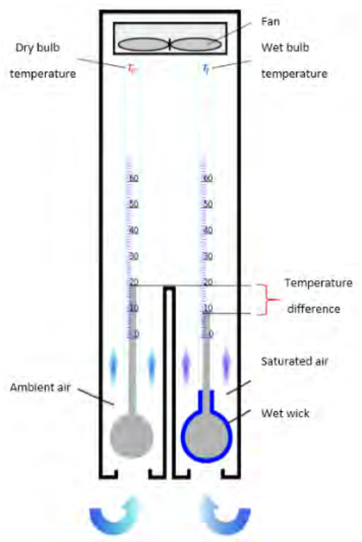
Is the temperature read by a thermometer or sensor which is covered with a water soaked 'wick' over which air is passed.

Traditionally, before the availbility of modern day semi-conductor humidistats, a 'sling psychrometer' was used, where 'twirling' the dual dry & wet thermometers around created a forced convective air stream rather than the fan indicated in the illustration below.

At 100% relative humidity (*see Relative Humidity section*), when the air is fully saturated, the dry bulb temperature equals the wet bulb temperature.

If the air sample is not saturated, then it can absorb moisture and in doing so is cooled by evaporation. The degree of evaporative cooling becomes greater the dryer the air sample and thus governs the depression in the wet bulb temperature compared with the dry bulb temperature.

When thermal equilibrium is reached, the rate of heat transfer from the warmer air to the wick; resulting in a decrease in the wet bulb temperature; equates to the rate of heat transfer required to evaporate the water from the wick into the air stream.



Since the transport mechanism that controls the convective heat transfer between the air and water also controls the moisture transfer between air and water, there exists a relationship between the heat and mass transfer coefficients, which are quantified by the ***Lewis Number, Le***.

By definition, the Lewis Number, Le is a dimensionless number defined as the ratio of thermal diffusivity to mass diffusivity, which can also be expressed as …

Le = Sc / Pr

where, Sc = Schmidt number defined as the ratio of momentum diffusivity (kinematic viscosity) and mass diffusivity
Pr = Prandlt number

Fortunately, for ambient moist air systems at normal working temperatures and pressures, Le ~ 1.0 and this practically means that the wet bulb line and adiabatic saturation lines coincide.

On a psychrometric chart, the plotted wet bulb lines and specific enthalpy lines are often shown as coincident. In reality, for air conditioning applications, the deviation is rather small and thus the coincident assumption is acceptable. However, for high temperature, high humidity applications, care should be taken.

DEPRESSED DRY BULB TEMPERATURE

Is the term used to denote the reduced (*depressed*) dry bulb temperature resulting from the thermodynamic cooling effect (*evaporative cooling*) when water is absorbed into an air sample whose relative humidity is < 100%.

Saturated air (100% RH and where the dry bulb and wet bulb temperatures are equal), by definition, is unable to absorb water and thus does not exhibit a depression in the dry bulb temperature.

This process is adiabatic (*constant enthalpy*) and on a psychrometric or Möllier chart (*see Psychrometrics section*) follows the wet bulb temperature line.

SWITCH POINT TEMPERATURE

Is a term used to denote the temperature at which an otherwise dry cooling application 'switches' to an adiabatic (wet) mode. This temperature is also often referred to as the ***depressed dry bulb temperature***.

Below the switch point temperature, the product will operate completely dry and use no water. However, as the ambient dry bulb and wet bulb temperatures change throughout the day or season, a condition may be reached where the dry bulb temperature is too high for the product to meet the cooling requirement.

Introducing water into the incoming air stream via precooling pads or a water spray system (*see sections Precooling pads & Water spray system*) results in an evaporative cooling process which depresses the dry bulb temperature, widening the operating TD and creating the ability to meet the cooling load.

DEW POINT TEMPERATURE

Is defined as the temperature to which atmospheric air must be cooled to become saturated with water vapour. When cooled further, the airborne water vapour (*existing as superheated steam at its partial pressure*), will condense to form liquid water, otherwise known as 'dew'.

If air comes into contact with a surface whose temperature is below the dew point of the localised air, water will condense on the surface.

The dew point of moist air is directly related to the humidity ratio (absolute humidity) and pressure as defined by …

$$P_{ws}(t_d) = (p \times W) / (0.621945 + W)$$

where, t_d = *dew point, °C*
$P_{ws}(t_d)$ = *saturation vapour pressure @* t_d *, Pa*
p = *barometric pressure, Pa*
W = *humidity ratio, kg/kg*

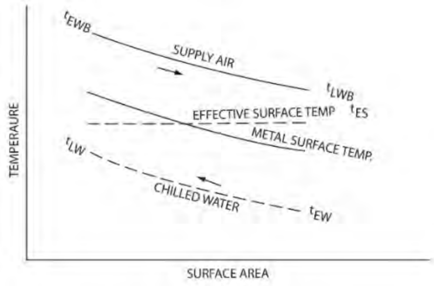
A simplified approximation for the dew point for relative humidities > 50% is …

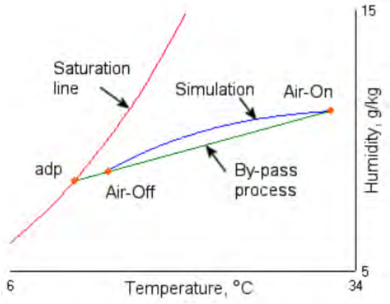
$$t_d \approx t - (100 - \%RH) / 5$$

APPARATUS DEW POINT

Is defined as the 'effective' surface temperature of a coiling coil.

It is also the temperature for a fixed flow rate at which both sensible and latent heat gains are removed at the required rates.





This parameter is usually derived from a psychrometric chart as the projection of the line joining the air inlet and outlet conditions projected to intersect with the saturation line.

The intersection point is the apparatus dew point (**ADP**).

The closeness of the air leaving condition to the ADP is governed by the bypass factor.

BYPASS FACTOR

The bypass factor describes the percentage of air that is not cooled to the ADP.

The air that is bypassed remains unchanged from the entering coil conditions. The bypass factor is a function of the airflow, number of rows, surface temperature, fin pitch, finned height and various other construction attributes of the coil.

The bypass factor can be calculated using the enthalpy, dry bulb temperature or humidity ratio as follows …

$$BF = (X_{in} - X_{out}) / (X_{in} - X_{APD})$$

where, BF *= bypass factor*
X_{in} , X_{out} , X_{ADP} *= dry bulb, enthalpy or humidity ratio*

Although the bypass factor is a traditional coil characteristic, it is not directly utilised in EAS's coil calculation methodology, which uses a complex cooling curve algorithm to establish the air leaving conditions from the coil. However, knowledge of the air conditions and ADP allows the bypass factor to be retrospectively calculated.

HUMIDITY

HUMIDITY RATIO OR MOISTURE CONTENT

Is defined as the ratio of the mass of water vapour to the mass of dry air in a given sample and is denoted by the symbol **W** and measured in units of kg/kg.

A sensible heating or cooling process involves no change in the humidity ratio or moisture content, whilst a cooling process involving condensation results in a reduction of the humidity ratio and thus moisture content of the cooled air stream.

SPECIFIC HUMIDITY

Is defined as the ratio of the mass of water vapour to the total mass of the moist air sample.

In terms of the humidity ratio …

$$\gamma = W / (1 + W)$$

RELATIVE HUMIDITY

Is defined as the ratio of the actual water vapour partial pressure in moist air at the dew point pressure and temperature to the reference saturation water vapour partial pressure at the dry bulb pressure and temperature.

More simply, it is the amount of water vapour present in air expressed as a percentage of the amount needed for saturation at the same temperature.

Relative humidity is denoted by the symbol **Ø** or more often, **%RH**

Dry air is defined as 0%RH, whilst saturated air is defined as 100%RH.

- *A sensible heating process will result in a constant humidity ratio (moisture content) but a reduction in the relative humidity.*
- *A sensible cooling process will also result in a constant humidity ratio (moisture content) but an increase in the relative humidity.*
- *A latent cooling process (reduction in dry bulb temperature and production of condensation) will result in a reduced humidity ratio (moisture content) and an increase in the relative humidity.*

Water vapour above 100°C and at atmospheric pressure exists in a superheated state and consequently the characteristics of the relative humidity scale change drastically.

Below 100°C it is possible to achieve 100% relative humidity at any temperature, but above 100°C, the maximum possible relative humidity plunges rapidly as the temperature increases. For example, at 150°C the maximum %RH achievable is 20%, whilst at 175°C, this drops to 10%. At 200°C the maximum is 5.9% and at 370°C (close to the critical point for water) achieves a mere 0.48%.

VOLUME

Volume is defined as the quantity of three dimensional space enclosed by a closed surface and is denoted by the symbol **V** and measured in cubic metres, m^3 in the SI system.

The metric system includes the litre, **L** otherwise known as dm^3.

1 litre = 1 dm^3 = 1000 cm^3 = 0.001 m^3

1 m^3 = 1000 litres

VOLUMETRIC FLOW RATE

Volumetric flow rate is defined as the volume of fluid which passes per unit time and is denoted by the symbol $\dot{\mathbf{V}}$ and measured in m^3/s. This parameter is more usually referred to as the air volume.

The air volume change throughout a system is proportional to the absolute temperature change in degrees K. Therefore, for a coil the air volumes entering and leaving are different if there has been a temperature change and the air mass flow rate is constant.

AIR-SIDE

The air-side volumetric flow rate is usually provided in m^3/s or m^3/hr and occasionally L/s.

1 m^3/s = m^3/hr / 3600

TUBE-SIDE

The tube-side volumetric flow rate is usually provided in L/s or L/hr.

1 L/s = L/hr / 3600

Note that many fluid and thermodynamic equations expect the volumetric flow rate in m^3/s

m^3/s = L/s / 1000

DENSITY

The density of a substance is defined as the mass per unit volume and denoted by the symbol **ρ** and measured in kg/m^3.

Thus, Density = mass / volume

$\boldsymbol{\rho = m / V}$ *where,* ρ *= density,* kg/m^3

m = mass, kg
V = volume, m^3

Specific gravity or relative density is the ratio of the density of the substance to the density of a reference substance. For liquids, water is often the reference substance where its density relates to 4°C (maximum density), whilst for gases, usually standard atmospheric pressure and room temperature of 20°C is the reference.

STANDARD AIR

The International Standard Atmosphere (ISA) defines dry air density at 101.325 kPa and 15°C as 1.225 kg/m^3, which most of Europe accepts.

$\boldsymbol{\rho_{air} = 1.225}$ **kg/m^3**

Typically, air consists of the following …

Components in dry air		Volume ratio = Molar ratio, compared to dry air		Molar Mass	Molar mass in air	
Name	Formula	[mol/mol_{air}]	[vol %]	[g/mol] [kg/kmol]	[g/mol_{air}] [kg/$kmol_{air}$]	[wt %]
Nitrogen	N_2	0.78084	78.084	28.013	21.873983	75.52
Oxygen	O_2	0.20946	20.946	31.999	6.702469	23.14
Argon	Ar	0.00934	0.934	39.948	0.373114	1.29
Carbon dioxide	CO_2	0.00033	0.033	44.010	0.014677	0.051
Neon	Ne	0.00001818	0.001818	20.180	0.000367	0.0013
Helium	He	0.00000524	0.000524	4.003	0.000021	0.00007
Methane	CH_4	0.00000179	0.000179	16.042	0.000029	0.00010
Krypton	Kr	0.0000010	0.0001	83.798	0.000084	0.00029
Hydrogen	H_2	0.0000005	0.00005	2.016	0.000001	0.000003
Xenon	Xe	0.00000009	0.000009	131.293	0.000012	0.00004
Average molar mass of air					28.9647	

The Universal Gas constant **R_o = 8.3145 J/K/mol** and by definition the gas constant for air is defined by …

R_{air} = Ro / molar mass

Thus, R_{air} = 8.3145 x 1000 / 28.96 = **287.1 J/kg/K**

The air density can be calculated using the Ideal Gas Laws (*see section Perfect gas equations*), simplified to …

PV = RT, where in terms of density the equation becomes …

ρ_{air} = P / (R x T) *where,* *ρ_{air} = air density, kg/m^3*
P = absolute pressure, Pa
T = absolute temperature, K
R = gas constant for dry air 287.1, J/kg/K

Therefore, at standard atmospheric pressure of 101.325 kPa and 15°C (288.15 K)

ρ_{air} = 101325 / (287.1 x 288.15) = **1.225 kg/m^3**

However, the traditional air density used in American literature is 1.204 kg/m^3 (based upon 0.075 lb/ft^3 at 70°F and usually rounded to 1.2 kg/m^3) and referred to as NTP (normal temperature & pressure). In fact, both figures are indeed correct but in view of the definition temperature, the densities have a slightly different magnitude.

A simple equation to predict the air density at different temperatures, assuming standard atmospheric pressure is …

ρ_t = 1.225 x (273 + 15) / (273 + t) = 1.225 x 288 / (273 + t)
ρ_t = 352.8 / (273 + t) *where,* *ρ_t = air density, kg/m^3*
t = temperature, °C

NORMAL AIR

On occasions, particularly in the air compressor industry sector, air volumetric flow rate is given in Nm^3/s or Nm^3/hr.

In essence this figure describes a mass flow rate because the 'N' signifies **Normal** air density which is often referred to 0°C and 1 atm (101.325 kPa). As a consequence, the air density is defined as 1.292 kg/m^3, so the mass flow rate is determined by the specified volume flow rate multiplied by 1.292, giving a mass flow in kg/s or kg/hr.

For such a system the mass flow is a constant and thus the volume flow can be calculated at the elevated process temperature once the density is calculated at the operating temperature and pressure.

To confuse matter, there are two Standards relating to Normal Cubic Meters …

- *DIN 1343 : 0°C & 1.01325 barA – this is more common understanding of Nm^3/h or Nm^3/s*
- *ISO 2533 : 15°C & 1.01325 barA*

So it is important to clarify which definition applies to the air volume provided.

MOIST AIR

Moist or air with a relative humidity > 0% and thus contains water vapour, has a lower density than dry air at the same temperature and pressure.

This may appear odd to the layperson, on the basis that obviously water is denser (heavier) than air, so if air contains water, then should it not be heavier ?

The water vapour portion of humid air is not liquid water with a nominal density of 1000 kg/m^3 at say 20°C, but actually exists as superheated steam at its partial pressure.

A simple expression to derive the density of water vapour is …

$$\rho_w = 0.0022 \times p_w / T$$

where, ρ_w = *water vapour density , kg/m^3*
p_w = *water vapour partial pressure, Pa*
T = *absolute temperature, K*

If one consults ASHRAE Fundamentals 2017 Chapter 1, tables for the properties of water at saturation details both the Absolute Pressure p_{ws} (saturation partial pressure) and saturated vapour specific volume v_g and at 20°C, P_{ws} = 2339.2 Pa and v_g = 57.76 m^3/kg. Thus, the density $\rho_{water\ vapour}$ = 0.0173 kg/m^3. So, the density of water vapour is significantly lower than dry air.

Thus, if we consider that in a given volume, let's say 1 cubic meter (m^3), containing dry air, then by definition the mass would be 1.225 kg. Now, if we were to saturate this volume of air with water vapour (create 100% RH moist air) and maintain the same temperature and pressure, then this would involve removing a portion of the 'heavier' dry air to replace it with the 'lighter' water vapour (superheated steam at its partial pressure). Thus, the same total volume of 1 m^3 contains a smaller mass of dry air plus the replacement volume of water vapour, which has a lower mass. Hence the total sample has a lower mass too. Thus, for this given 1m^3, the density is lower.

ALTITUDE

Altitude is referred to as the height above sea level and is the basis for the definition of barometric pressure and 'standard atmosphere', which is based upon 15°C and sea level.

The table to the right indicates how the temperature and pressure of atmospheric air changes with altitude.

The ASHRAE Fundamentals 2017, Chapter 1 equations are provided below …

$$t = 15 - 0.0065 \times Z$$

$$p = 101.325 \times (1 - 2.25577 \times 10^{-5} \times Z)^{5.2559}$$

where, t = *temperature, °C*
p = *pressure, kPa*
Z = *altitude, m*

Altitude m	Temperature °C	Pressure kPa
-500	18.2	107.478
0	15.0	101.325
500	11.8	95.461
1000	8.5	89.875
1500	5.2	84.556
2000	2.0	79.495
2500	-1.2	74.682
3000	-4.5	70.108
4000	-11.0	61.640
5000	-17.5	54.020
6000	-24.0	47.181
7000	-30.5	41.061
8000	-37.0	35.600
9000	-43.5	30.742
10000	-50.0	26.436

Clearly, the perfect gas laws (*see section Gas Laws*) can be applied to arrive at the density at altitude if needed for thermal calculations.

SPECIFIC VOLUME

The specific volume of a substance is defined as the ratio of the substance's volume to its mass and denoted by the symbols **sv** and measured in m^3/kg. Furthermore, it is the reciprocal of density.

Thus, Specific Volume = volume / mass or the reciprocal of the density, 1 / ρ

sv = V / m *where,* *sv = specific volume, m³/kg*

m = mass, kg

V = volume, m³

SPECIFIC HEAT

Specific heat is defined as the amount of heat per unit mass required to raise the temperature by 1°C. It is denoted by the symbol C_p and measured in kJ/kg/K.

This property is temperature dependent, but for the purpose of our industry section the following values are sufficient.

For air …

C_p = 1.006 dry air, kJ/kg/K *Some literature suggests a value of 1.005*

C_p = 1.006 + 1.86 x W_{avg} moist air, kJ/kg/K

where, *W_{avg} = average moisture content, kg/kg*

1.86 = specific heat capacity of water vapour, kJ/kg/K

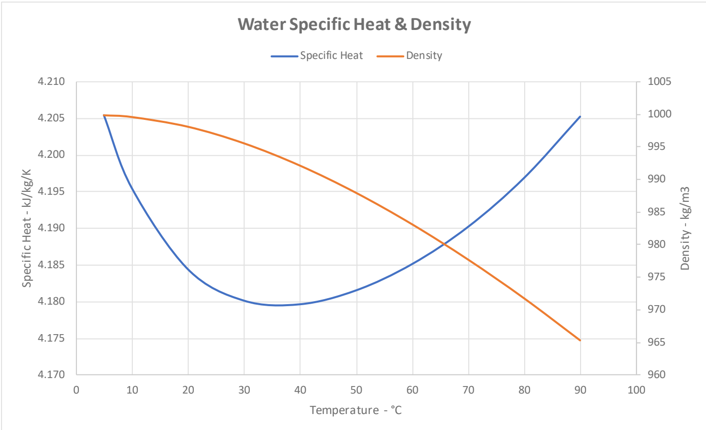
For water the graph shows the change in the specific heat that has a minimum value at around 30-40°C. Otherwise …

Typical chilled water temperatures **C_p = 4.20 kJ/kg/K**

Typical dry cooler hot water temperatures **C_p = 4.18 kJ/kg/K**

However, at higher water temperatures the specific heat value increases, yet again, to 4.2 kJ/kg/K

see below …



Data derived using NIST RefProp 10.0

THERMAL CONDUCTIVITY

Is defined as the quantity of heat that is transmitted through a unit thickness of substance due to a temperature gradient, under steady state conditions. It is denoted the symbol **k** and is measured in W/m/K.

This property is also temperature dependent and increases in value as the temperature increases.

The values shown are those agreed by Evapco Inc. and used in the selection software.

Material	Thermal Conductivity W/m/K
Copper	385.0
Aluminium	221.7
AlMg	191.5
Stainless	16.2
Titanium	20.0

Regarding the materials of construction of a coil, the thermal conductivity of tube material and associated wall thickness act as a barrier to the heat transfer between the primary and secondary fluids.

Furthermore, the thermal conductivity of the fin material (extended surface) impacts even more upon the thermal performance of the coil. As an example, a copper tube/aluminium fin heat exchanger of typically 4 rows deep would require perhaps 10 rows deep if the tube and fin materials were stainless steel.

In the case of both the fluid on the inside of the tube and the outside air, the thermal conductivity affects the magnitude of the Prandtl & Nusselt numbers (*see sections Prandtl number - Pr & Nusselt number*) which in turn affect the internal and external heat transfer coefficients. Again, the higher the value the better the heat transfer.

Over time a number of ways have been devised to designate viscosity grades of the lubricants, thermal & hydraulic oils. There are SAE (Society of Automotive Engineers) grades for gear oils and crankcases (engines), AGMA (American Gear Manufacturers Association) grades for gear oils, SUS (Saybolt Universal Seconds), cSt (kinematic viscosity in centistokes), and absolute viscosity.

To add to the confusion, two measures of temperature (Fahrenheit and Celsius) can be applied to most of these, not to mention that viscosity might be presented at either 40°C (104°F) or 100°C (212°F).

While all of these have served useful purposes to one degree or another, most lubrication practitioners settle on and use one method as a basis for selecting products. To the novice, the number of options can be confusing, particularly if the primary lubricant supplier does not associate one of the prominent viscosity systems to the product label.

To complicate matters, machinery designers must define the lubricant viscosity in such a way that the equipment user understands clearly what is needed without having to consult outside advice.

This points to the need for a universally accepted viscosity designation - one that can be used by lubrication practitioners, lubricant suppliers and machinery design engineers simultaneously with minimal confusion.

In 1975, the International Standards Organization (ISO), in unison with American Society for Testing and Materials (ASTM), Society for Tribologists and Lubrication Engineers (STLE), British Standards Institute (BSI), and Deutsches Institute for Normung (DIN) settled upon an approach to minimize the confusion. It is known as the International Standards Organization Viscosity Grade, **ISO VG** for short.

Perhaps the most important thermophysical property of an oil is its viscosity, so ..

WHAT IS VISCOSITY ?

Viscosity is the measure of the oil's resistance to flow (shear stress) under certain conditions. To simplify, the oil's viscosity represents the measure for which the oil wants to stay put when pushed (sheared) by moving mechanical components.

There are two viewpoints of the resistance to flow that the designer is interested in. One is the measure of how the fluid behaves under pressure, such as a pressurized hydraulic line. This property is called **absolute viscosity** (also known as **dynamic viscosity**) and is measured in centipoises (cP).

The other consideration is how the fluid behaves only under the force of gravity. This is called **kinematic** viscosity and measured in centistokes. The two are related through the specific gravity of the fluid. To determine the dynamic viscosity in centipoise (cP) of a fluid it is necessary to multiply the kinematic viscosity in centistokes (cSt) of the fluid times the specific gravity of the fluid or measure it directly using an absolute viscometer.

Dynamic viscosity (cP or mPa.s) = Kinematic viscosity (cSt or mm^2/s) x Density (kg/m^3) / 1000

On a side note, if you are using in-service oils, it is probably worth measuring the viscosity in absolute units. The measure in centistokes can be misleading because the specific gravity of lubricants changes with age, generally moving up. It is possible to find yourself exceeding an absolute viscosity limit for a machine but still have a kinematic measure that indicates you are OK.

So, viscosity is a measure of the fluid's resistance to flow. Water has a low viscosity of 1 cSt whereas honey has a very high viscosity, perhaps 1,000 cSt.

Viscosities are defined or assigned using a laboratory device called a viscometer. For lubricating oils, viscometers tend to operate by gravity rather than pressure. Think of a kinematic viscometer as a long glass tube that holds a volume of oil. The measure of the fluid's viscosity is the measure of the amount of time that it takes for the designated amount of oil to flow through the tube under very specific conditions.

This is similar to the amount of time it takes a specific volume of fluid at a specific temperature to drain through a funnel. As the fluid gets thicker - a function of its increasing resistance to flow - then it takes progressively longer to move through the tube (funnel). Water goes through in one second. The same amount of honey takes a thousand seconds (hypothetically).

We know that if we raise and lower the temperature of a fluid, there is often a correlating change in the fluid's resistance to flow. The fluid gets thicker at lower temperatures and it gets thinner at higher temperatures.

Given all of these variables and details, several organizations decided to come up with a way to characterize lubricating oils so that members of their respective organizations would have a uniform and simple way to communicate, educate and ultimately protect their interests.

PURPOSE OF THE ISO VG SYSTEM

The purpose of the ISO system of classifying viscosity grades is to establish a viscosity measurement method so that lubricant suppliers, equipment designers and users will have a common (standardized) basis for designating or selecting industrial liquid oils.

Different approaches were thoroughly considered before the ISO Technical Committee (TC23) settled on an approach that is logical and easy to use. There were a few important criteria to keep in mind from the beginning, such as:

- Referencing the oil/lubricants at a nominal temperature for industrial systems.
- Using a pattern that conforms to uncertainties imposed by dimensional manufacturing tolerances.
- Using a pattern that had some sense of repeatability up and down the scale.
- Using a pattern that used a small, easily manageable number of viscosity grades.

The reference temperature for the classification should be reasonably close to average industrial service experience. It should also relate closely to other selected temperatures used to define properties such as viscosity index (VI), which can aid in defining a lubricant.

A study of possible temperatures indicated that 40°C (104°F) was suitable for the industrial-lubricant classification as well as for the lubricant-definition properties mentioned above. This ISO viscosity classification is consequently based on kinematic viscosity at 40°C (104°F).

For the classification to be used directly in engineering design calculations in which the kinematic viscosity of the lubricant is only one of the parameters, it was necessary that the viscosity grade width (range of tolerance) be no more than 10 percent on either side of the nominal value. This would reflect an order of (centre point) uncertainty in calculations similar to that imposed by dimensional manufacturing tolerances.

This limitation, coupled with the requirement that the number of viscosity grades should not be too large, led to the adoption of a system with gaps between the viscosity grades.

This classification defines 20 viscosity grades in the range of 2 to 3200 square millimetres per second (1 mm^2/s = equals 1 cSt) at 40°C (104°F). For petroleum-based liquids, this covers approximately the range from kerosene to cylinder oils.

Each viscosity grade is designated by the nearest whole number to its midpoint kinematic viscosity in mm^2/s at 40°C (104°F), and a range of +/- 10 percent of this value is permitted. The 20 viscosity grades with the limits appropriate to each are listed in Table 1.

The classification is based on the principle that the midpoint (nominal) kinematic viscosity of each grade should be approximately 50 percent greater than that of the preceding one. The division of each decade into six equal logarithmic steps provides such a system and permits a uniform progression from decade to decade.

The logarithmic series has been rounded off for the sake of simplicity. Even so, the maximum deviation for the midpoint viscosities from the logarithmic series is 2.2 percent.

Table 2 pulls together some popular viscosity measurement methods into one table.

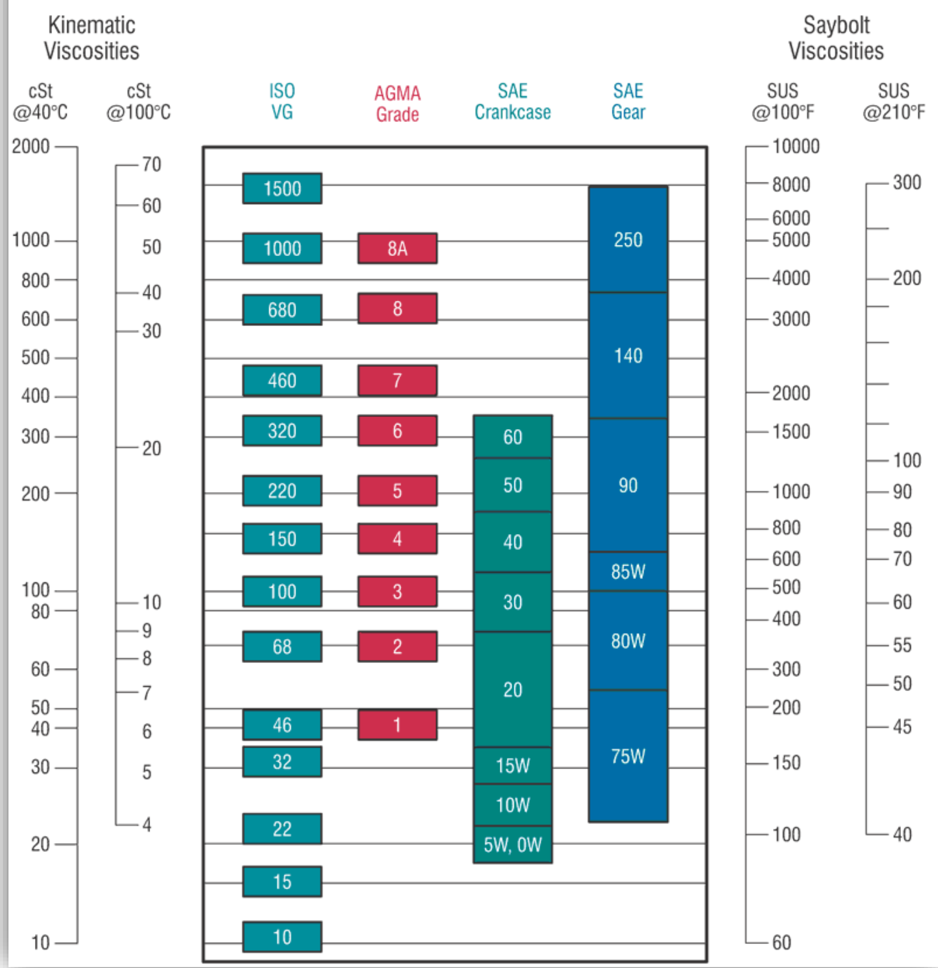
While it is true that some viscosity grades will be left out of the mix as companies move toward adopting the ISO designation, it is not necessary that the users of those products have to move away from them. Further, there is no intention to offer quality definition of lubricants with this scale. That a product has an ISO VG number associated with it has no bearing on its performance characteristics.

The ISO designation has been under development since 1975. The most recent release in 1992 (ISO 3448) contains 20 gradients. This covers nearly every type of application that the lubricant practitioner can expect to encounter. The lubricant manufacturing community has accepted the recommended ISO gradients and has devoted appreciable effort and energy to conform to the new grading approach with old and new products.

Table 1. ISO Viscosity Classification

ISO Viscosity Grade	Midpoint Kinematic Viscosity mm²/s at 40°C (104°F)	Kinematic Viscosity Limit mm²/s at 40°C (104°F) Minimum	Kinematic Viscosity Limit mm²/s at 40°C (104°F) Maximum
ISO VG 2	2.2	1.98	2.42
ISO VG 3	3.2	2.88	3.52
ISO VG 5	4.6	4.14	5.06
ISO VG 7	6.8	6.12	7.46
ISO VG 10	10	9.00	11.0
ISO VG 15	15	13.5	16.5
ISO VG 22	22	19.8	24.2
ISO VG 32	32	29.8	35.2
ISO VG 46	46	41.4	50.6
ISO VG 68	68	61.2	74.8
ISO VG 100	100	90.0	110
ISO VG 150	150	135	165
ISO VG 220	220	198	242
ISO VG 320	320	288	352
ISO VG 460	460	414	506
ISO VG 680	680	612	748
ISO VG 1000	1000	900	1100
ISO VG 1500	1500	1350	1650
ISO VG 2200	2200	1980	2420
ISO VG 3200	3200	2880	3520

Table 2. Comparative Viscosity Classifications



MASS FLOW RATE

Is defined as the mass of fluid which passes per unit time and is denoted by the symbol $\dot{m}$ and measured in kg/s.

The mass flow rate through a system or indeed a coil is a constant and is independent of the temperature or pressure, assuming there are no losses or additions along the flow path.

However, as described in *section Volumetric flow rate*, the volumetric flow rate will fluctuate.

FORCE

Force is any interaction that, when unopposed, will change the motion of an object. It is denoted by the symbol **F** and measured in **N**ewtons (*named after English physicist Isaac Newton*).

Force is defined as mass x acceleration

F = m x a *where,* *F = force, N (kg.m/s²)*
m = mass, kg
a = acceleration, m/s²

$$F = m \times a$$

where, F = force, N ($kg.m/s^2$); m = mass, kg; a = acceleration, m/s^2

1 Newton = 1 kg.m/s²

In our industry sector we do not usually encounter physical acceleration of objects and thus gravitational acceleration, **g** is the parameter used in most equations.

g ≈ 9.81 m/s²

Thus, 1 kg exerts a force of 1 x 9.81 = 9.81 N

WORK

Work, otherwise known as energy has units of Joules (*named after the English physicist James Prescott Joule*) and denoted by the symbol **W**.

Work is defined a Force x Distance

$$W = F \times d$$

where, W = work (energy), J (Nm); F = force, N; d = distance, m

1 Joule = 1 Nm

POWER, DUTY, CAPACITY

Power is defined as the rate of doing work and is the amount of energy transferred per unit time. Power is denoted by the symbol **Q** and is measured in Watts (*named after the English engineer James Watt*).

Thus Power is derived from work / time

$$Q = W / t$$

where, Q = power, Watt (J/s); W = work, J; t = time, s

In our industry we usually refer to thermal power as **duty** or **capacity** which is usually given in kiloWatts (kW)

1 kW = 1000 Watts

TONS OF REFRIGERATION

Global regions that have had a North American influence often refer to cooling load requirements (***Refrigeration effect - RE***) in Tons of Refrigeration (TR).

This term originates from the early days of the refrigeration industry during the 1880s when the generation of ice was a huge industry. To quantify the energy requirements (duty) associated with the generation of ice, 1 TR was defined as the heat transfer necessary to generate 1 *short ton* (2000 lb or 907 kg) of pure ice at 32°F (0°C) in 24 hours. This equates to 288,000 Btu of energy, which in terms of the power requirements equates to 12,000 Btu/hr

1 Ton of Refrigeration = 12,000 Btu/hr = 3.517 kW

*A much less common definition used in Europe is **1 tonne of refrigeration** which is defined as the heat removal required to freeze 1 metric ton (1000 kg) of water at 0°C in 24 hours. Based upon the heat of fusion of 334 kJ/kg, 1 tonne of refrigeration = 3.86 kW. Thus 1 tonne of refrigeration is approximately 10% larger than 1 ton of refrigeration (TR).*

CONDENSER TONS OF REFRIGERATION

Refrigeration systems are concerned with cooling products or cooling processes and thus the cold side (evaporator) capacity is of prime importance, hence the definition of tons of refrigeration associated with the cooling side (refrigeration effect, **RE**) and related to the freezing of a commonly available commodity i.e. water into ice.

A refrigeration system (*or indeed a heat pump system*) comprises 4 major components, the compressor, an evaporator, an expansion device and a condenser. The condenser is the device that dumps the unwanted energy, usually to atmosphere, to enable the thermodynamic Carnot cycle to function.

A condenser's total heat of rejection (THR) comprises the cooling load (RE) plus the heat of compression associated with the work done by the compressor. Depending upon whether the compressor is an 'open', semi-hermetic or fully hermetic machine, plus whether it is a reciprocating, screw, sliding vane, centrifugal etc. will dictate the magnitude of the heat of compression that is generated and required to be dissipated in addition to the RE capacity.

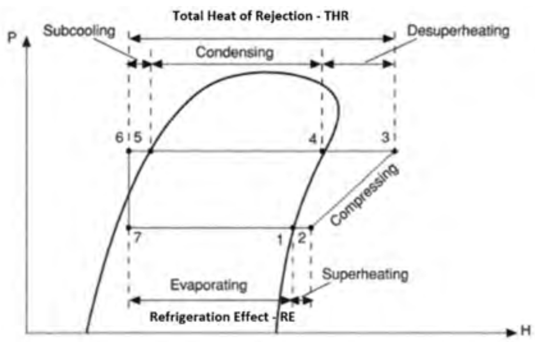
However, typically the condenser THR is 25% greater than the refrigeration effect (RE) and thus for simplicity, when selecting a condenser for an application where only the cooling load requirements (RE) are known, it is common practice to use 15,000 Btu/hr as the practical value for 1 ton of refrigeration, when applied to the condenser.

1 Condenser TR = 12,000 Btu/hr x 1.25 = 15,000 Btu/hr

However, in some parts of our industry sector 1 Condenser TR = 14,700 Btu/hr

CONDENSER TOTAL HEAT OF REJECTION

The total heat of rejection **THR** is a term used to describe the energy associated with the desuperheating, condensing and subcooling portion of the refrigeration system i.e. the high pressure, high temperature process of the refrigeration Carnot cycle.



The Total Heat of Rejection (3, 6) can be equated to Refrigeration Effect (7, 2) + compressor heat of compression (2, 3).

A condenser will usually be sized to dissipate the duty (kW) associated with either all or part of this energy process line.

Some of the heat dissipation may be through the uninsulated compressor hot gas discharge line to atmosphere (partially 3, 4) and then again further heat loss from the uninsulated condenser liquid line to the TEV (5, 6). The residual heat required for rejection should then be handled by the condenser (4, 5) and is referred to as the condenser total heat of rejection.

TOTAL DUTY

Total duty also often referred to as just duty, is denoted by the symbol $\mathbf{Q_T}$ and measured in Watts or kW. It represents the total heat transferred by a process and is define as …

$$Q_T = \dot{m} \times \Delta h$$

where, Q_T = duty, kW

$\dot{m}$ = mass flow rate, kg/s

Δh = enthalpy difference, kJ/kg

This fundamental equation can be applied to both the air side and fluid side of the heat transfer process and 'encapsulates' the total enthalpy change of the two process streams.

However, air enthalpy cannot be directly measured but can be derived from the measured dry bulb and/or wet bulb temperatures.

SENSIBLE DUTY

Sensible heat is a type of energy and relates to a change in temperature of a substance with no change in phase. In our industry this relates to single phase liquids and gases. It is denoted by the symbol $\mathbf{Q_S}$ and measured in Watts or kW.

$$Q_S = \dot{m} \times C_p \times \Delta t$$

where, Q_s = duty, kW

$\dot{m}$ = mass flow rate, kg/s

C_p = specific heat, kJ/kg/K

Δt = temperature difference, K

In the case of air, the temperature difference, Δt relates to the dry bulb temperatures.

Furthermore, for sensible heating and cooling processes …

$$\mathbf{Q_T = Q_S}$$

SENSIBLE HEAT RATIO

Is defined as the ratio between the sensible cooling capacity and the total cooling capacity …

SHR $= Q_S / Q_T$ which reduces to … **SHR** $= C_p \times \Delta t / \Delta h$

An SHR =1.0 signifies sensible cooling whilst an SHR <1.0 signifies a latent cooling process and thus the generation of condensate.

LATENT DUTY

Latent duty is the capacity associated with latent heat and is the thermal energy released or absorbed by a substance during a constant temperature process e.g. during boiling or melting where the terms are better known as latent heat of vapourisation or latent heat of fusion, respectively. It is often denoted by the symbol, **L** and is measured in kJ/kg.

The term was introduced around 1762 by British chemist *Joseph Black* and was derived from the Latin word for 'hidden', in the context that latent heat was 'hidden energy'.

Expressed in terms of the amount of energy in the form of heat (duty) …

$$Q_L = \dot{m} \times \Delta L$$

where, Q_L = *latent duty, kW*
$\dot{m}$ = *mass flow rate, kg/s*
ΔL = *specific latent heat, kJ/kg*

Alternatively, if the humidity ration (moisture content) in kg/kg is known for the inlet and outlet conditions, then …

$$Q_L = \dot{m} \times \Delta W \times 2500$$

where, ΔW = *moisture content difference, kg/kg*
2500 = latent heat for water, kJ/kg

As a consequence, cooling an air sample below its dew point will result in …

- Sensible duty, calculated from the dry bulb temperature difference
- Latent duty, calculated from the latent heat of vapourisation
- Total duty, $\mathbf{Q_T = Q_S + Q_L}$

EXERGY

In thermodynamics, the exergy of a system is the maximum useful work possible during a process that brings the system into equilibrium with a heat reservoir … usually the environment.

When the surroundings are the reservoir, exergy is the potential of a system to cause a change as it achieves equilibrium with its environment. Exergy is the energy that is available to be used. After the system and surroundings reach equilibrium, the exergy is zero. Determining exergy was also the first goal of thermodynamics.

The term exergy was coined in 1956 by *Zoran Rant* (1904–1972) by using the Greek 'ex' and 'ergon' meaning 'from work', but the concept was developed by *J. Willard Gibbs* in 1873.

PRESSURE

Pressure is defined as the normal force exerted per unit area of surface within the system and denoted by the symbol **P**. In SI units, pressure is denoted by Pascal (Pa) – *named after the French physicist Blaise Pascal*.

1 Pa = 1 N/m² … *which a very small engineering unit*.

ATMOSPHERIC PRESSURE

Standard atmospheric pressure is defined as 101.325 kPa or 1.01325 barA or 760 Torr (mmHg) and often referred to as 1 atm (atmosphere).

ABSOLUTE & GAUGE PRESSURE

Generally, in our industry, we refer to pressures related to atmospheric pressure (*see above*) and thus are effectively variable differential pressures. Such pressures are traditionally measured using analogue (dial) gauges, but nowadays digitally via pressure transducers.

If a gauge is calibrated in kPa (kiloPascals), then 'standard' atmospheric pressure would be indicated as 101.325 kPa and 0 kPa would represent a perfect vacuum. However, such a gauge would have a rather extensive scale to cover typical refrigeration applications !

More commonly the unit 'bar' has been adopted as the measurement of pressure and by definition, 1 bar = 100,000 Pa. This is rather close to standard atmospheric pressure; which is defined as 101,325 Pa; and equates to 1.01325 bar. So, one can see why atmospheric pressure is simplified and often referred to as 1 bar.

This concept means that a pressure gauge will show zero pressure at atmospheric pressure. So, when disconnected from a system, the needle will indicate zero. Thus, atmospheric pressure equates to zero gauge pressure (0 barg).

Any pressure greater than atmospheric pressure will show a positive (greater than zero) pressure, whilst a pressure below atmospheric pressure will show a negative value.

But the unit 'bar', although defined as 100,000 Pa, can be referred to in two different ways … either as an absolute value or referenced to atmospheric pressure. When referring to its absolute variant **barA**, then 0 barA is an absolute vacuum and atmospheric pressure is 1.01325 barA. But when referred to in its 'gauge' form **barg**, atmospheric pressure would strictly speaking be 0.01325 barg … although zero barg is more often used !!

Note : When using gauge pressures (barg), the related absolute pressure is the gauge pressure reading (barg) + atmospheric pressure, and atmospheric pressure is a 'variable'. We usually assume 1.01325 bar as atmospheric pressure, but at altitude or down a mine, atmospheric pressure can vary quite considerably.

*Although barg are the normal units to describe gauge pressure, some compressor manufacturers use the term **bar(e)** instead, where this in interpreted as 'effective' pressure.*

Therefore, if the units of 'bar' are not fully quantified there can be some confusion. So, what is 2 bar? Is it 2 barA or could it be 2 barg / bar(e), which is actually 3 barA and in some cases this can be rather significant.

To confuse matters even further, pressures below atmospheric pressure (*sometimes referred to as below gauge pressure or 'in a vacuum'*) can be measured using a variety of units, such as barA, (-ve) barg, inches or mercury (ins.Hg), millimetres of mercury (mm Hg), Torr or micron.

The unit Torr is named after an Italian scientist Torricelli and defined as the pressure required to raise mercury by 1 mm in a Fortin barometer. Standard atmospheric pressure equates to 760 Torr (760 mm Hg or 29.92 in Hg)

Although any system can operate at a pressure below atmospheric i.e. a negative gauge (barg) pressure, this is referred to as a partial vacuum. Deeper vacuums often relate to refrigeration systems, especially when preparing the system for their refrigerant charge. In such cases, a near complete vacuum is required to be 'pulled' to ensure that no moisture (water) is present in the system, prior to the introduction of the refrigerant.

VAPOUR PRESSURE

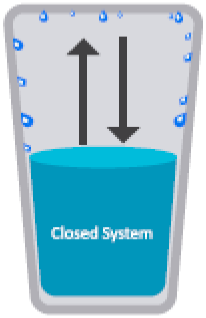
Water vapour is one of several gases that makes up air. For example, if the total pressure of a system such as air at sea level is 101.325 kPa, and that air is made up of Nitrogen, Oxygen, water vapour and other trace gases, each of those gases contributes to the total pressure of 101.325 kPa. The portion that is water vapor is called the partial pressure of water vapour.

The partial pressure of water vapour is a key metric found as a component in the formulas that define all other humidity parameters.

Vapour pressure above a liquid

Because molecules in a liquid are closer to one another than they are in a gas, intermolecular forces are stronger than in a gas. For a liquid to vapourise, the intermolecular forces have to be overcome by the kinetic energy of the molecules.

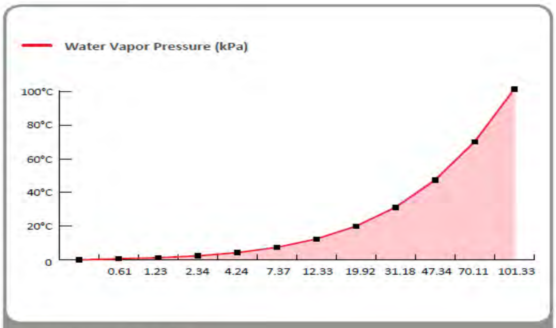
If a liquid is placed in a closed container, the particles entering the vapour phase cannot escape. In their random motion, particles collide with the liquid and are recaptured by intermolecular forces. Thus, two processes occur simultaneously: evaporation and condensation.



The rate of evaporation increases as temperature increases. This is because an increase in temperature corresponds to an increase in the kinetic energy of molecules. At the same time, the rate of condensation increases as the number of particles in the vapour phase increases: more molecules collide with the surface of the liquid. When these two processes become equal, the number of particles and, therefore, the pressure in the vapour phase, reaches equilibrium.

The value of the equilibrium vapour pressure depends upon the attractive forces between particles of the liquid and on the temperature of the liquid. Vapour pressure above a liquid, increases with increasing temperature.

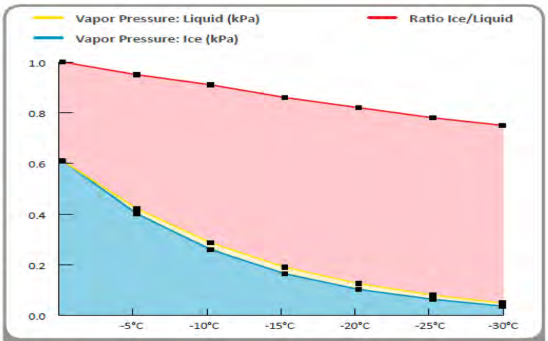
The vapour pressure of water increases rapidly with increasing temperature ..



Vapor Pressure of Water

Vapour pressure above ice

When water freezes, the molecules assume a structure that permits the maximum number of hydrogen-bonding interactions between molecules. Because this structure has large hexagonal holes, ice is more open and less dense than liquid water. As hydrogen bonding is stronger in ice than in liquid water, the intermolecular attraction forces are the strongest in ice. That’s why the vapour pressure above ice is less than the vapour pressure above liquid water.



Vapor Pressure Above Ice

AIR SIDE

Pressure in Pascals (Pa) is adequate to denote airside pressure drop.

FLUID SIDE

For the fluid or refrigerant side pressure it is more common to use kiloPascals, MegaPascals or bar.

1 kPa = 10^3 Pa or 1000 Pa

1 bar = 10^5 Pa or 100,000 Pa

1 MPa = 10^6 Pa or 1,000,000 Pa

In this industry sector pressures are usually measured with respect to atmospheric pressure rather than with respect to absolute vacuum and thus referred to as gauge pressure (barg).

$P_{abs} = P_{atm} + P_{gauge}$

So strictly speaking …

1 barg = (1 + 1.01325) barA = 2.01325 barA

However, usually atmospheric pressure is referred to as 0 barg = 1 barA, so …

1 barg = (1 + 1) barA = 2 barA

It is often assumed that pressures quoted in 'bar' refers to absolute pressure or barA. However, this is not always the case and thus it is recommended that all pressures are suffixed with either barg or barA to avoid confusion.

***All** pressures referred to in the Pressure Vessel Directive ~ **PED 2014/68/EU** relate to **gauge pressure** (barg).*

INTERNAL ENERGY

Is the property of the system comprising all forms of energy arising from the internal structure of the substance and has units of kJ/kg and is denotes by the symbol **U**.

Unless detailed thermodynamic or psychrometric analysis is conducted, internal energy is not usually encountered in everyday life.

ENTHALPY

Enthalpy is a thermodynamic property of the system with units of kJ/kg and denoted by the symbol **h** and defined as ..

h = u + P x V *where,* *h = enthalpy, kJ/kg*
u = internal energy, kJ/kg
P = pressure, Pa
V = specific volume, m^3/kg

ENTROPY

For completeness, Entropy denoted by the symbol **s** with units of kJ/kg/K is defined as the microscopic disorder of a system both describes and is extensively an equilibrium property.

In general this property is used whilst performing thermodynamic calculations but is not often encountered in heat exchanger analysis and design.

SATURATION CONDITIONS

The temperature and pressure state of a fluid whereupon a liquid is about to vapourise or a gas is about to liquify.

SATURATED LIQUID

A saturated liquid is a fluid whose dryness fraction is equal to zero i.e. x = 0

An example is liquid water at 100°C and atmospheric pressure.

SATURATED VAPOUR

A saturated vapour is a fluid with a quality of 100% or dryness fraction, x = 1.0

An example is water vapour at 100°C and atmospheric pressure, otherwise known as steam.

When a kettle boils, the visible cloud issuing from the spout is often thought to be steam, but in fact, it is not steam, but condensed water vapour slightly below 100°C. If you look closely at the end of the spout, before the visible water vapour cloud, there is small invisible region … this is the saturated steam region.

Steam is a colourless gas and thus invisible, which is one of the dangerous aspects of steam.

SATURATION TEMPERATURE

Is another term for the boiling point and is the temperature, at a corresponding pressure, at which a liquid changes phase and becomes a vapour/gas.

SATURATION PRESSURE

Is the pressure at which a vapour is in thermodynamic equilibrium with its condensed state.

LATENT HEAT

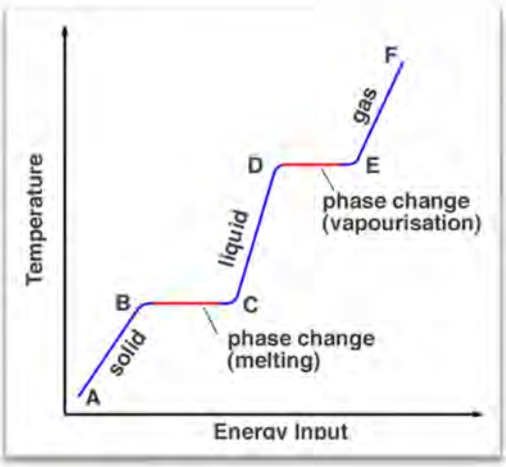
Is the thermal energy associated with a change of phase and is a constant temperature process and is measured in kJ/kg.

It is named after the Latin word for 'hidden' as it was seemingly deemed to be a mysterious thermodynamic property.

LATENT HEAT OF VAPOURISATION

Is the energy associated with changing the phase of a liquid into a vapour e.g. boiling a liquid.

For water, the latent heat of vapourisation is **2256 kJ/kg** at 100°C, whilst at 0°C it becomes 2500 kJ/kg and at 15°C, 2465 kJ/kg



LATENT HEAT OF FUSION

Is the energy associated with changing the phase of a liquid into a solid e.g. freezing water.

For water the latent heat of fusion is **334 kJ/kg**

Incidentally, besides the latent energy associated in turning liquid water into a solid … frost/ice, if the frost layer is further cooled below 0°C, then there is additional energy expended to sensibly cool the layer to the final air leaving temperature. The Specific Heat of frost is assumed to ***~2.1 kJ/kg***

DRYNESS FRACTION OR QUALITY

Both terminologies are denoted by the symbol **x** and defined as the ratio of the mass of pure vapour present in the total mass of the mixture.

Quality is usually given as a percentage e.g. 40%, whilst dryness fraction is given a number, 0.4

SUPERHEAT

Is when a vapour at a given pressure is heated beyond its saturation temperature.

So, at atmospheric pressure water boils at 100°C and if the generated water vapour (steam) is further heated resulting in a temperature of say 120°C, the vapour is said to be superheated by 20K.

Incidentally, the water vapour portion contained within moist ambient air exists as a superheated vapour in relation to its saturation temperature at its partial pressure.

The term superheat is usually used in the following contexts …

- The superheated hot gas discharge temperature leaving a compressor, which is often the same temperature that the refrigerant enters the condenser or a desuperheater, if fitted
- The superheated suction gas temperature exiting from a DX evaporator

DESUPERHEAT

Refers to the removal of the superheat to either lower its temperature or return the gas/vapour to its saturation temperature.

Often in industrial ammonia closed circuit condensers, a desuperheating dry coil is mounted above the ‘wet section’ of the evaporative condenser to reduce the high hot gas discharge temperatures exiting the compressors to more acceptable temperatures suitable for the ‘wet’ condenser.

Ammonia refrigeration systems condensing at typically 35°C can generate hot gas discharge temperatures in excess of 150°C. *As a guideline, reciprocating compressors might discharge at 120°C, whilst screw compressors discharge at perhaps 80°C.*

In relation to an evaporative condenser, water hitting the condenser tubes at temperatures much above 60°C, radically increase the rate at which ‘scale’ builds up on the tubes, fouling the primary surface and decreasing the condenser efficiency. Thus, using a desuperheater to reduce the hot gas to below 60°C can have significant advantages regarding the longevity of the product.

In related market sectors, gas-to-liquid plate-type heat exchangers can be used to generate hot water resulting from the desuperheating of the hot discharge gas.

SUBCOOLING

Subcooling relates to the undercooling of a liquid below its saturation temperature at a given pressure.

Water at 20°C is in fact a subcooled liquid (subcooled by 80K) on the basis that at standard atmospheric pressure the saturation temperature of water is 100°C.

Subcooling is often achieved within a condenser by either slightly over surfacing the condenser coil or designing the condenser with a discrete subcooling section. Alternatively, a separate coil or heat exchanger may be used to provide the desired subcooling.

Ensuring that a subcooled liquid enters a refrigeration system’s expansion device is advantageous as it both increases the enthalpy difference across the evaporator, increasing the potential refrigeration effect (cooling capacity) plus eradicates the possibility of ‘flash gas’ forming in the liquid line prior to the TEV, which would likely affect the stability of the system.

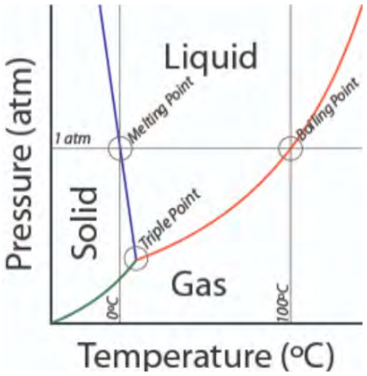
Usually a ‘sight glass’ is installed in the liquid line to ‘visually’ ensure that flash gas does not exist, and the liquid is at least saturated or marginally subcooled.

FLASH GAS

Flash gas is the term used when portions of a refrigerant liquid; close to its saturation temperature; spontaneously changes phase into a vapour as a result of a pressure drop, usually associated with frictional resistance in a pipe.

If flash gas is present in the liquid line between a condenser and the expansion device - *which is designed to handle 100% liquid, preferably subcooled* - then the liquid/vapour mixture will affect the controlling behaviour of the valve and perhaps cause the system to ‘hunt’ i.e. become unstable.

TRIPLE POINT



Is defined as the temperature at which all three phases i.e. solid, liquid, vapour coexist in equilibrium.

For water it is defined as 0.01°C at 611.2 Pa

CRITICAL POINT

As a substance approaches its critical temperature, the properties of its gas and liquid phases converge, resulting in only one phase at the critical point: a homogeneous supercritical fluid.

For water, the critical point occurs at approximately 374 °C and 22.064 MPa.

FLUID PHASES

Generally, a substance can exist as a solid, liquid or gas or combinations of these three states. However, in the HVAC/R industry sector we shall primarily consider fluids in their liquid and/or gaseous states.

SINGLE PHASE

Defined as a fluid whose phase does not change during a thermodynamic process i.e. it stays either a liquid, a gas or a solid. The single phase solid state is not generally of concern in our industry sector, except for ice slurry cooling applications, which is not a market sector that EAS is currently involved with.

TWO PHASE

A combination of two phases, for example during condensation or evaporation, where a liquid and a vapour coexist in equilibrium.

Alternatively, a combination of a solid and a liquid, such as ice slurry/binary ice applications utilising the latent heat of fusion.

ADIABATIC PROCESSES

Thermodynamically, and according to the first law of thermodynamics, an adiabatic process is one that does not involve heat or mass transfer with its surrounds.

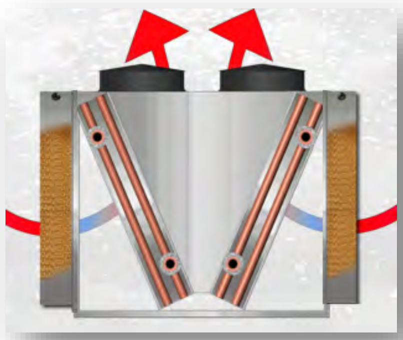
In the HVAC & R industry section, adiabatic processes are encountered in …

REFRIGERATION CYCLE

The refrigerant expansion process via the thermostatic/electronic expansion valve (TEV/EEV), which drops the pressure from the condensing pressure to the evaporating pressure, is deemed to be a reversible adiabatic process i.e. constant enthalpy.

However in reality, the process involves a slight increase in enthalpy as the liquid expands and changes phase into a mixture of liquid and flash gas, typically x = 0.15 to 0.2, but practically/analytically the outlet enthalpy is considered to equal the inlet enthalpy.

PRECOOLING PADS



Precooling (adiabatic) pads are fitted to air cooled condensers and dry coolers to depress the ambient air dry bulb temperature entering the coils to increase the operating temperature difference (TD) and thus the product's capacity.

The resulting reduction in the dry bulb temperature of the air exiting the pads - having passed through a 'wetted media' - results in an adiabatic process triggered by the absorption of moisture into the air stream causing an associated reduction in the dry bulb temperature.

Psychrometrically, the **evaporative cooling process** is a 'near' constant enthalpy process, which is also a 'near' wet bulb temperature process and results in a depression of the dry bulb temperature.

This evaporative cooling effect can be felt following taking a shower and stepping out of the cubicle. The lower temperature that is 'felt' results from the evaporation of the water from one's body into the surrounding air. This triggers the thermodynamic evaporative cooling effect, which reduces the air temperature, hence the 'cold' feeling.

SATURATION EFFICIENCY

Saturation efficiency is denoted by the symbol **η** and defined as the ratio of the reduction in the dry bulb temperature compared with the difference between the entering dry bulb and wet bulb temperatures.

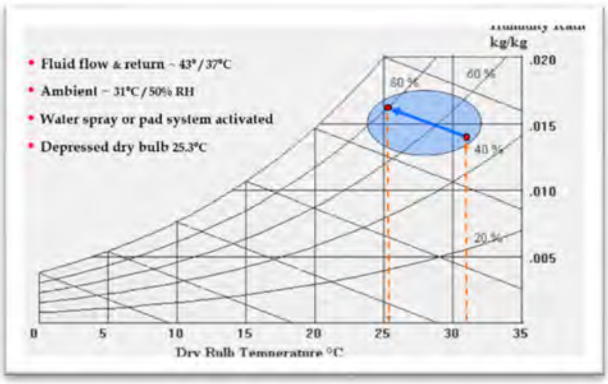
$$\eta = (t_{d1} - t_{d2}) / (t_{d1} - t_w)$$

where, η = *saturation efficiency*

t_{d1} = *inlet dry bulb, °C*

t_{d2} = *outlet dry bulb, °C*

t_w = *inlet wet bulb, °C*



The efficiency of precooling pads is governed by …

- The material that the pad is manufactured from
- The design and angle of the 'flutes'
- The thickness of the pad
- The air velocity through the pad

The lower the air velocity and the thicker the pad, then the higher the saturation efficiency attainable.

Typically, with a 6" (150 mm) thick pad, saturation efficiencies of 70 to 85% are achievable if the air velocity is around 2.0 m/s. However, a 50 mm thick pad is more likely to only yield an efficiency of 45-50%.

As an example, an air entering condition of 32°C db/40% RH equating to 21.6°C wb will provide a depressed dry bulb temperature of 24.2°C, assuming a pad saturation efficiency of 75%.

Considering a process with an air cooled condenser where the condensing temperature is 45°C, then without the pads, the operating TD will be (45 – 32) = 13K TD.

However, with the pads operational, the operating TD will increase to (32 - 24.2) = 20.8K, which is potentially a 60% increase in the condenser's total heat of rejection capability or alternatively, the need for a 60% less condenser surface area to meet the duty requirement.

Historically, saturation efficiencies of 80% had been used, however following tests around 2015, this was lowered to 70.2%, but later with the introduction of a new Pad profile, efficiencies of 86.5% were achievable. Although this figure is still used in CoilCalc, the consensus of 80% is perhaps a preferable figure !

The amount of water required to be fed to a pad system is calculated from the air mass flow rate and additional moisture content required to achieve the saturation efficiency of typically 86.5% as follows …

$$Z = \dot{V} \times \rho_{air} \times (w_2 - w_1) \times 60 \times \text{Excess Factor}$$

where, Z = *Recommended water flow rate, L/min*

$\dot{V}$ = *air volume, m³/s*

ρ_{air} = *air density, kg/m³*

w_1 = *moisture content @ inlet, kg/kg*

w_2 = *moisture content @ outlet, kg/kg*

2.5 = Excess Factor

The Excess Factor of 2.5 is to ensure that the pads are indeed wet enough to achieve an +80% saturation efficiency plus ensure that the pads are rinsed sufficiently to reduce scaling effects.

During 2025, EAS is expected to adopt the eco-Air Gen 5 water distribution system for their adiabatic Vee type dry cooler & condenser ranges. This system uses a higher pressure water spray feed onto the top of the pads. As a result, comparable saturation efficiencies are achievable with a significant reduction in water usage … typically -50%.

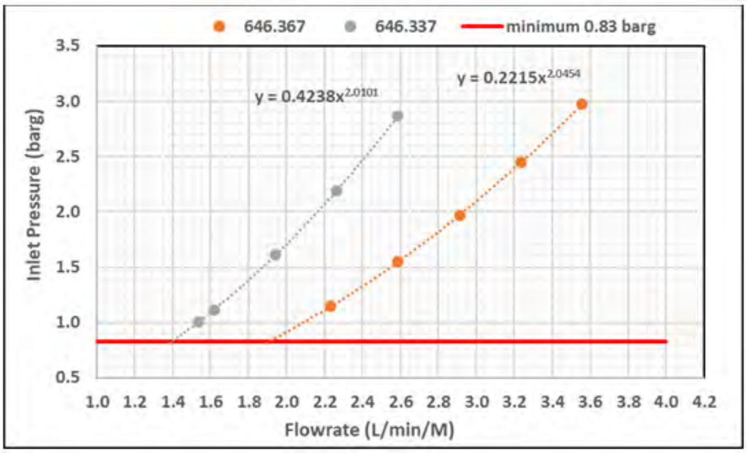
A further enhancement to this type of adiabatic system is the adoption of a recirculation tank and controls, which again reduces total water consumption compared with the previous 'total loss' water concept. This new feature has yet to be released or adopted within Spectrum or EAS's EASiPad technology.

The Gen 5 water distribution concept is based upon tests conducted in the U.S.A. and besides the theoretical moisture addition, an additional flow rate is based upon the 'plan area' of the Pads. The original data states 0.07 USGPM/ft² plan area rather than just an excess Factor of 2.5

As the Pads used are 6" (150 mm) deep, the above can be transposed into an additional water flow per meter length of Pad, suggesting ***0.435 l/min per meter****.*

Thus the recommended flow rate, $Z = \dot{V} \times \rho_{air} \times (w_2 - w_1) \times 60 + (0.435 \times \text{Pad Length})$

The water is introduced via interfering 'flat' format water spray nozzles spaced at 175 mm intervals and offered in two sizes which cover the expected water flow range. These nozzles are designed to operate above 0.83 barg and maximum of 3 barg, where the actual pressure; to ensure adequate distribution; is determined from the graph below or detailed equations.



An alternative method to achieve an evaporative cooling effect is to spray water into the incoming air stream prior to it entering the coil. The resultant depressed dry bulb temperature of the air hitting the coil widens the operating TD and the equipment is capable of an increased capacity.

Water can be spayed either directly onto the coil in sympathy with the air flow direction or in opposition to the air direction.

- Traditionally water is sprayed, via simple hollow cone ceramic nozzles, in sympathy with the air flow direction directly onto the fins.

  Of the two alternative approaches, spraying directly onto the coil can have thermal performance benefits, due to the additional energy associated with the water evaporating when in contact with the hot surface of the fins in addition to the depressed dry bulb temperature of the air resulting from the absorption of the water into the dry air stream.

  However, there is a greater propensity for fouling of the fins from the scaling associated with large amounts of water on the fins. This problem is exacerbated at fin surface temperatures above 60°C.

- The second option, favoured by EAS, is intended to function in a fashion where the generated 'spray mist', through which the air passes, is completely absorbed by the incoming ambient air before entering the coil. Thus, ideally, there will be no overspray or water hitting the coil surface.

  However, such idealistic concepts are unrealistic, and the amount of water sprayed into the air stream is calculated to exceed the theoretical amount needed. Thus, there is an 'overspray' situation where the coil surface is indeed seen to be wet.

  In view of the above, it is recommended to use a hydrophilic coated fin material at a minimum of 2.5 mm fin pitch to ensure that the water on the fins is dispersed effectively and does not cause water agglomeration issues.

In both scenarios, the saturation efficiency is considered to be conservatively, 75-80% and the required water spray flow rate calculated in a similar fashion as described above (*section* Calculation of Pad water flow rate) but with the **Excess Factor** = **1.75** instead of 2.5 … or the newer 2025 correlation.

It should be noted that as over-spray water does indeed come into contact with the finned surface, there is an additional 'cooling effect' as the water is evaporated from the warm surface. This 'cherry on the cake' load is not accounted for in the calculations and considered as a safety margin.

GAS LAWS

HISTORY

During the 17th and 18th centuries several scientists quantified the behaviour of gases and identified the interrelationship between the amount, volume, pressure and temperature.

BOYLE'S LAW

The Irish chemist *Robert Boyle* (1627-1691) conducted experiments that determined the quantitative relationship between the pressure and volume of a gas, resulting in …

$P \times V = \text{constant}$ alternatively, **$V \propto 1 / P$**

where, P is absolute pressure

Simply put … at constant temperature, the volume of a fixed amount of gas is inversely proportional to its pressure.

CHARLES'S LAW

The French chemist *Jacques Alexandre Charles* (1746-1823) identified the relationship between the volume of a gas and temperature, resulting in …

$V = \text{constant } T$ alternatively, **$V \propto T$**

where, T = temperature in Kelvin (°C + 273.15)

Simply put … at constant pressure, the volume of a fixed amount of gas is directly proportional to its absolute temperature.

GAY-LUSSAC'S LAW

The French chemist *Joseph Louis Gay-Lussac* (1778 – 1850) established that the pressure of a fixed mass and volume of gas was directional proportional to the absolute temperature.

$P / T = \text{constant}$ alternatively, **$P \propto T$**

where, T = temperature in Kelvin (°C + 273.15)

AVOGADRO'S HYPOTHESIS

The Italian chemist *Amedeo Avogadro* postulated that at the same temperature and pressure, equal volumes of gases contain the same number of molecules.

$V = \text{constant } (n)$ alternatively, **$V \propto n$ @ constant P & T**

Simply put, the volume of a gas is directly proportional to number of moles of gas present.

AVOGADRO CONSTANT

Historically this constant has slightly changed over time, but during 2017 new measurements led to a new definition, which took effect from 20th May 2019.

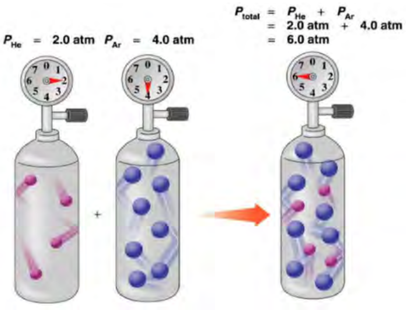
The Avogadro constant **N_A** = (*exactly*) $6.022\ 140\ 76 \times 10^{23}\ mol^{-1}$

DALTON'S LAW

Named after the English physicist *John Dalton* and relates to partial pressures of gases.

Dalton's law states that the pressure of a mixture of gases is the sum of the partial pressures of the individual components given by …

$P_{total} = P_1 + P_2 + P_3 \dots + P_n$



In the case of moist atmospheric air …

$P_{atm} = P_{dry\ air} + P_{H2O}$

where, P_{atm} *= atmospheric pressure, Pa*
$P_{dry\ air}$ *= dry air partial pressure, Pa*
P_{H2O} *= water vapour partial pressure, Pa*

AMAGAT'S LAW

Named after the French physicist *Émile Hilaire Amagat* and relates to partial volumes and states that …

The total volume of a gas mixture is equal to the sum of the partial volumes each gas would occupy if it existed alone at the temperature and pressure of the mixture, thus …

$V_{total} = V_1 + V_2 + V_3 \dots + V_n$

In the case of moist atmospheric air …

$V_T = V_{DA} + V_{WV}$

where, V_T *= total mixture volume*
V_{DA} *= dry air partial volume*
V_{WV} *= water vapour partial pressure, Pa*

Both Amagat's and Dalton's laws predict the properties of gas mixtures. Their predictions are the same for ideal gases. However, for real (non-ideal) gases, the results differ, especially for high temperature and/or pressure. Dalton's law of partial pressures assumes that the gases in the mixture are non-interacting (with each other) and each gas independently applies its own pressure, the sum of which is the total pressure. Amagat's law assumes that the volumes of the component gases … again at the same temperature and pressure … are additive; the interactions of the different gases are the same as the average interactions of the components. The interactions can be catered for in terms of a second virial coefficient, B(T), for the mixture.

PERFECT GAS EQUATIONS

PRACTICAL EQUATIONS

Following on from the above and reference to the Universal Gas constant, the Ideal Gas Laws were established by *Emile Clapeyron* in 1834 as an amalgam of Boyle's, Charles, Avogadro and Gay-Lussac's laws. This stated …

$PV = nRT$ *where, n = number of moles of gas*

For our purposes the equation can be simplified into …

$(P \times V/T)_1 = (P \times V/T)_2$

Now in terms of density …

$[P / (\rho \times T)]_1 = [P / (\rho \times T)]_2$

where, *Temperatures must be Kelvin where T = t°C + 273.15*
Pressure must be absolute pressure e.g. barA or (barg + 1.01325)
Density, ρ in kg/m³

EXAMPLE – COMPRESSED AIR

As an example, 2.5 Nm³/s is compressed to 5 barg and exits the compressor at 150°C. What is the volume and density of the compressed air leaving the compressor?

By definition, 2.5 Nm³/s *(see section Density, Normal air)* indicates an air temperature of 0°C and pressure of 0 barg and thus an air density of 1.292 kg/m³.

Using the rearranged perfect gas equation in terms of density, we get …

$\rho_2 = \rho_1 \times (P_2 / P_1) \times (T_1 / T_2)$

Therefore,

ρ_2 = 1.292 x [(5 + 1.01325) / (0 + 1.01325)] x [(0 + 273.15) / (150 + 273.15)]

= **4.9495 kg/m³**

Similarly, rearraigning in terms of volume, we get …

$V_2 = V_1 \times (P_1 / P_2) \times (T_2 / T_1)$

Therefore, transposing the volumes into volumetric flow rates …

$\dot{V}_2$ = 2.5 x (0 + 1.01325) / (5 + 1.01325) x (150 + 273.15) / (0 +273.15)

= **0.6526 m³/s**

Alternatively, having found the density, we know that the mass flow rate through a system is constant and thus the air entering mass flow is …

$\dot{m} = \dot{V}_1 \times \rho_1$

= 2.5 x 1.292 = 3.23 kg/s … *and this will be the air mass flow leaving the compressor*

Now rearranging, $\dot{V}_2 = \dot{m} / \rho_2$

$\dot{V}_2$ = 3.23 / 4.9495 = **0.6526 m³/s**

ORIGINS

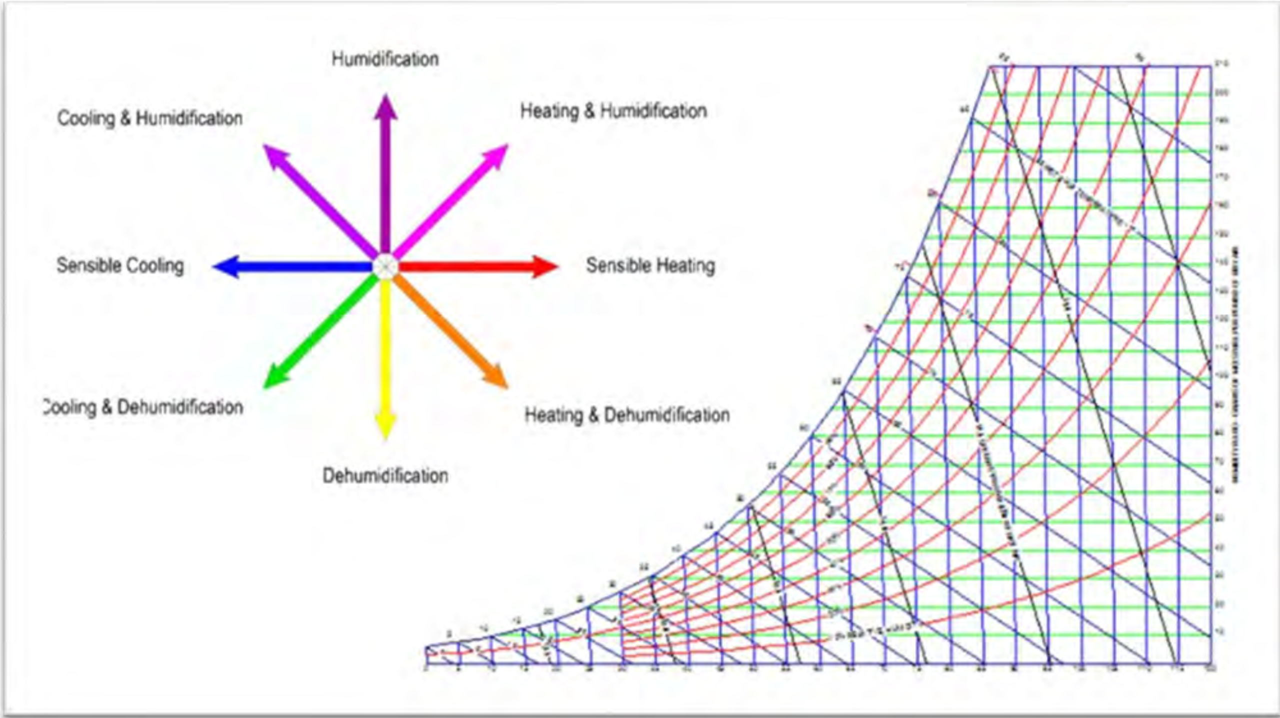
A German scientist, *Ernest Ferdinand August*, in 1825 conjured up the name for a wet bulb thermometer from the Latin words ***psychro*** meaning 'to make cold' and ***meter*** meaning 'to measure'.

Variations of this word now describe the science of the behaviour of the binary mixture of dry air and water vapour.

PSYCHROMETRIC CHART

The psychrometric chart is a plot of the major properties of air using the dry bulb temperature as the x-axis and the moisture content as the y axis. Thereafter, parameters such as relative humidity, specific volume, wet bulb temperature and enthalpy are superimposed. Consequently, the behaviour of an air heating, cooling, humidification etc. process can be graphically plotted and the resultant 'unknown' parameters easily determined.

This chart, developed by W.H. Carrier (1876 – 1950), is primarily used in countries that have had a North American influence.

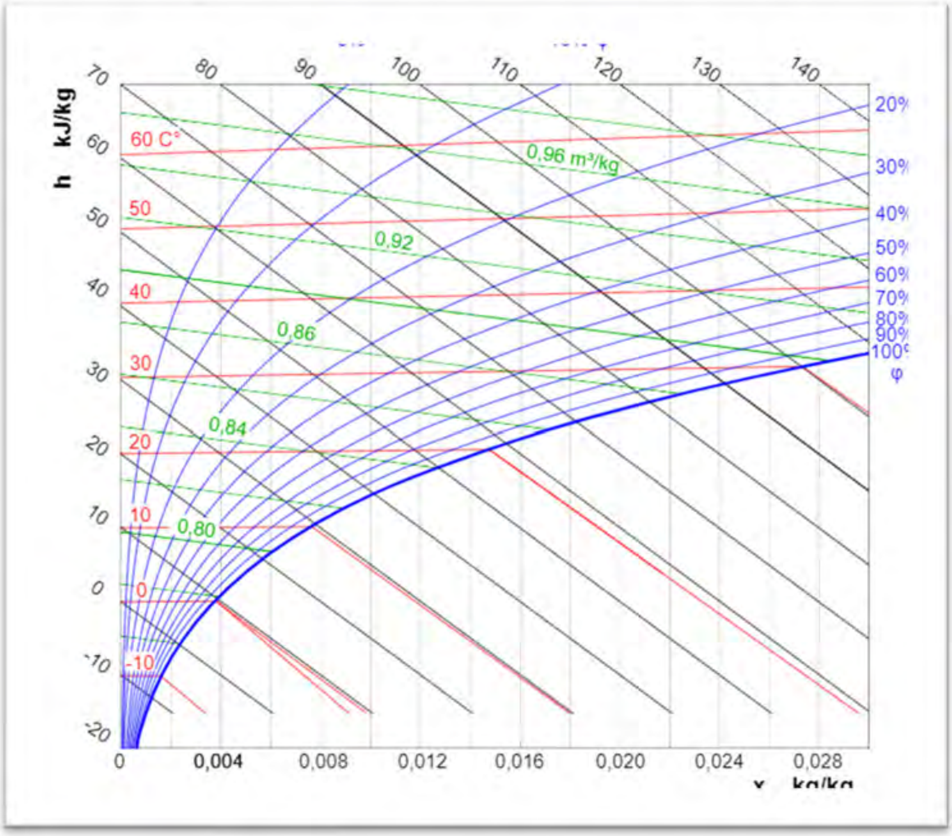


The above psychrometric chart also indicates the direction that the process takes when conditioning the air.

MÖLLIER CHART

This chart, developed by Richard Möllier (1863 -1935) has the same function as the psychrometric chart but is displayed with the temperature and moisture content axes transposed.

It is more common to use this format of air's behaviour in European communities.

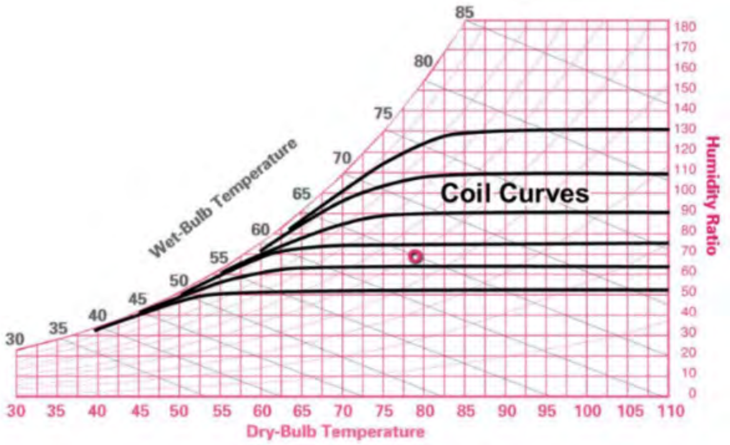


COIL COOLING CURVE

Typical coil cooling curves are shown on a psychrometric chart. However, these relate to latent cooling applications where the coil fin surface temperature is below the dew point of the entering air conditions.

Thus, the inlet fluid temperature inside the tubes would need to be >= 2K below the air entering dew point temperature.

Clearly, the lower the inlet fluid temperature, for a given air entry condition, the greater the ability to condense moisture from the air and thus dehumidify the air stream.



The complex cooling curve lines are mathematically generated from the locus of a series of virtual straight lines constructed on the psychrometric chart as the dry bulb temperature is reduced, and the moisture content reduces too whilst the outlet conditions approach the surface temperature of the fins, which in itself is governed by the fluid temperatures.

The shape of the resultant cooling curve is affected by the air inlet position on the chart and can result in sensible only cooling or varying degrees of latent cooling.

FROST FORMATION

If the coil extended (finned) surface temperature is below the dew point temperature of the entering air, then moisture will condense on the fins. If the surface temperature of the fins drops below 0°C then any condensate will freeze and initially turn into frost … and over time into ice … which involves additional energy in the form of the latent heat of fusion (334 kJ/kg).

Furthermore, if the air leaving temperature is well below 0°C, then this frost layer must be sensibly cooled to the air leaving temperature, where the frost specific heat is ~2.1 kJ/kg/K.

ATOMIC MASSES

Atomic No.	Element	Symbol	Atomic Mass
1	Hydrogen	H	1.0079
2	Helium	He	4.0026
3	Lithium	Li	6.9410
4	Beryllium	Be	9.0122
5	Boron	B	10.8110
6	Carbon	C	12.0107
7	Nitrogen	N	14.0067
8	Oxygen	O	15.9994
9	Fluorine	F	18.9984
10	Neon	Ne	20.1797
11	Sodium	Na	22.9897
12	Magnesium	Mg	24.3050
13	Aluminum	Al	26.9815
14	Silicon	Si	28.0855
15	Phosphorus	P	30.9738
16	Sulfur	S	32.0650
17	Chlorine	Cl	35.4530
19	Potassium	K	39.0983
18	Argon	Ar	39.9480
20	Calcium	Ca	40.0780
21	Scandium	Sc	44.9559
22	Titanium	Ti	47.8670
23	Vanadium	V	50.9415
24	Chromium	Cr	51.9961
25	Manganese	Mn	54.9380
26	Iron	Fe	55.8450
28	Nickel	Ni	58.6934
27	Cobalt	Co	58.9332
29	Copper	Cu	63.5460
30	Zinc	Zn	65.3900
31	Gallium	Ga	69.7230
32	Germanium	Ge	72.6400
33	Arsenic	As	74.9216
34	Selenium	Se	78.9600
35	Bromine	Br	79.9040
36	Krypton	Kr	83.8000
42	Molybdenum	Mo	95.9400
45	Rhodium	Rh	102.9055
46	Palladium	Pd	106.4200
47	Silver	Ag	107.8682
48	Cadmium	Cd	112.4110
50	Tin	Sn	118.7100
51	Antimony	Sb	121.7600
53	Iodine	I	126.9045
54	Xenon	Xe	131.2930
55	Cesium	Cs	132.9055
56	Barium	Ba	137.3270
74	Tungsten	W	183.8400
77	Iridium	Ir	192.2170
78	Platinum	Pt	195.0780
79	Gold	Au	196.9665
80	Mercury	Hg	200.5900
82	Lead	Pb	207.2000

The properties of a gas mixture depend upon the properties of the individual gases (often referred to as components or constituents) as well as on the amount of each gas in the mixture.

At temperatures below the critical temperature, the gas phase of a substance is frequently referred to as a vapor. The term vapor implies a gaseous state that is close to the saturation region of the substance, raising the possibility of condensation during a process.

During the processing of a gas–vapor mixture, the vapor may condense out of the mixture, forming a two-phase mixture. This may complicate the analysis considerably. Therefore, a gas–vapor mixture needs to be treated differently from an ordinary gas mixture.

MASS & MOLE FRACTIONS

To determine the properties of a mixture, we need to know the composition of the mixture as well as the properties of the individual components. There are two ways to describe the composition of a mixture: either by specifying the number of moles of each component, called molar analysis, or by specifying the mass of each component, called gravimetric analysis.

Consider a gas mixture composed of *k* components. The mass of the mixture m_m is the sum of the masses of the individual components, and the mole number of the mixture N_m is the sum of the mole numbers of the individual components. That is,

$$m_m = \sum_{i=1}^{k} m_i \quad \text{and} \quad N_m = \sum_{i=1}^{k} N_i$$

Subscript $_m$ & $_i$ refer to mixture and a single component of the mixture respectively.

The ratio of the mass of a component to the mass of the mixture is called the **mass fraction** *mf*, and the ratio of the mole number of a component to the mole number of the mixture is called the **mole fraction** *y* …

$$\mathrm{mf}_i = \frac{m_i}{m_m} \quad \text{and} \quad y_i = \frac{N_i}{N_m}$$

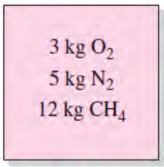


FIGURE 4

EXAMPLE 1 Mass and Mole Fractions of a Gas Mixture

Consider a gas mixture that consists of 3 kg of O_2, 5 kg of N_2, and 12 kg of CH_4, as shown in Fig. 4. Determine (*a*) the mass fraction of each component, (*b*) the mole fraction of each component, and (*c*) the average molar mass and gas constant of the mixture.

SOLUTION The schematic of the gas mixture is given in Fig. 4. We note that this is a gas mixture that consists of three gases of known masses.

Analysis (*a*) The total mass of the mixture is

$$m_m + m_{O_2} + m_{N_2} + m_{CH_4} = 3 + 5 + 12 = 20 \text{ kg}$$

Then the mass fraction of each component becomes

$$\mathrm{mf}_{O_2} = \frac{m_{O_2}}{m_m} = \frac{3 \text{ kg}}{20 \text{ kg}} = \mathbf{0.15}$$

$$\mathrm{mf}_{N_2} = \frac{m_{N_2}}{m_m} = \frac{5 \text{ kg}}{20 \text{ kg}} = \mathbf{0.25}$$

$$\mathrm{mf}_{CH_4} = \frac{m_{CH_4}}{m_m} = \frac{12 \text{ kg}}{20 \text{ kg}} = \mathbf{0.60}$$

(*b*) To find the mole fractions, we need to determine the mole numbers of each component first:

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{3 \text{ kg}}{32 \text{ kg/kmol}} = 0.094 \text{ kmol}$$
$$N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{5 \text{ kg}}{28 \text{ kg/kmol}} = 0.179 \text{ kmol}$$
$$N_{CH_4} = \frac{m_{CH_4}}{M_{CH_4}} = \frac{12 \text{ kg}}{16 \text{ kg/kmol}} = 0.750 \text{ kmol}$$

Thus,

$$N_m = N_{O_2} + N_{N_2} + N_{CH_4} = 0.094 + 0.179 + 0.750 = 1.023 \text{ kmol}$$

and

$$y_{O_2} = \frac{N_{O_2}}{N_m} = \frac{0.094 \text{ kmol}}{1.023 \text{ kmol}} = \mathbf{0.092}$$
$$y_{N_2} = \frac{N_{N_2}}{N_m} = \frac{0.179 \text{ kmol}}{1.023 \text{ kmol}} = \mathbf{0.175}$$
$$y_{CH_4} = \frac{N_{CH_4}}{N_m} = \frac{0.750 \text{ kmol}}{1.023 \text{ kmol}} = \mathbf{0.733}$$

(*c*) The average molar mass and gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{20 \text{ kg}}{1.023 \text{ kmol}} = \mathbf{19.6 \text{ kg/kmol}}$$

or

$$\begin{aligned} M_m &= \sum y_i M_i = y_{O_2} M_{O_2} + y_{N_2} M_{N_2} + y_{CH_4} M_{CH_4} \\ &= (0.092)(32) + (0.175)(28) + (0.733)(16) \\ &= 19.6 \text{ kg/kmol} \end{aligned}$$

Also,

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/(kmol} \cdot \text{K)}}{19.6 \text{ kg/kmol}} = \mathbf{0.424 \text{ kJ/(kg} \cdot \text{K)}}$$

IDEAL & REAL GASES

An ideal gas is defined as a gas whose molecules are spaced far enough apart so that the behaviour of a molecule is not influenced by the presence of other molecules … a situation encountered at low densities. Real gases approximate this behaviour closely when they are at a low pressure or high temperature relative to their critical-point values.

The P-*v*-T behaviour of an ideal gas is expressed by the simple relation P*v* = RT, known as the ideal-gas equation of state. The P-*v*-T behaviour of real gases is expressed by rather more complex equations of state or by P*v* = ZRT, where Z is the compressibility factor.

When two or more ideal gases are mixed, the behaviour of a molecule is not normally influenced by the presence of other similar or dissimilar molecules, and therefore a nonreacting mixture of ideal gases also behaves as an ideal gas.

Air, for example, is conveniently treated as an ideal gas in the range where nitrogen and oxygen behave as ideal gases. When a gas mixture consists of real (non-ideal) gases, however, the prediction of the P-*v*-T behaviour of the mixture becomes rather involved.

The prediction of the P-v-T behaviour of gas mixtures is usually based on two models …

- Dalton's law of additive pressures and
- Amagat's law of additive volumes

$$\text{Dalton's law:} \quad P_m = \sum_{i=1}^{k} P_i(T_m, V_m)$$

$$\text{Amagat's law:} \quad V_m = \sum_{i=1}^{k} V_i(T_m, P_m)$$

$$\left.\right\} \text{exact for ideal gases, approximate for real gases}$$

In these relations, P_i is called the **component pressure** and V_i is called the **component volume**.

V_i is the volume a component would occupy if it existed alone at T_m and P_m, not the actual volume occupied by the component in the mixture. (In a vessel that holds a gas mixture, each component fills the entire volume of the vessel. Therefore, the volume of each component is equal to the volume of the vessel.) Also, the ratio P_i/P_m is called the **pressure fraction** and the ratio V_i/V_m is called the **volume fraction** of component *i*.

Therefore,

$$\frac{P_i}{P_m} = \frac{V_i}{V_m} = \frac{N_i}{N_m} = y_i$$

and is strictly valid for ideal-gas mixtures since it is derived by assuming ideal-gas behaviour for the gas mixture and each of its components. The quantity $y_i P_m$ is called the **partial pressure** (identical to the component pressure for ideal gases), and the quantity $y_i V_m$ is called the **partial volume** (identical to the component volume for ideal gases).

Note that for an ideal-gas mixture, the mole fraction, the pressure fraction, and the volume fraction of a component are identical.

DRY & ATMOSPHERIC AIR

Air is a mixture of nitrogen, oxygen, and small amounts of some other gases. Air in the atmosphere normally contains some water vapor (or moisture) and is referred to as **atmospheric air**. By contrast, air that contains no water vapor is called **dry air**. It is often convenient to treat air as a mixture of water vapour and dry air since the composition of dry air remains relatively constant, but the amount of water vapour changes as a result of condensation and evaporation from oceans, lakes, rivers, showers, and even the human body.

It would be convenient to also treat the water vapour in the air as an ideal gas at the cost of a little accuracy and this is in fact feasible for typical HVAC applications up to 50°C.

At 50°C, the saturation pressure of water is 12.3 kPa and at pressures below this value, water vapour can be treated as an ideal gas with negligible error (<0.2%), even when it is a saturated vapour. Therefore, water vapour in air behaves as if it existed alone and obeys the ideal-gas relation Pv = RT. Then the atmospheric air can be treated as an ideal-gas mixture whose pressure is the sum of the partial pressure of dry air P_a and that of water vapour P_v …

$$P = P_a + P_v$$

The partial pressure of water vapour is usually referred to as the **vapour pressure**. It is the pressure water vapour would exert if it existed alone at the temperature and volume of atmospheric air.

SPECIFIC & RELATIVE HUMIDITY

The amount of water vapour in the air can be specified in various ways. Probably the most logical way is to specify it directly as the mass of water vapor present in a unit mass of dry air. This is called **absolute** or **specific humidity** (also called *humidity ratio*) and is denoted by ω …

$$\omega = \frac{m_v}{m_a} \qquad (\textit{kg water vapour/kg dry air})$$

The specific humidity can be defined as …

$$\omega = \frac{m_v}{m_a} = \frac{P_v V/(R_v T)}{P_a V/(R_a T)} = \frac{P_v / R_v}{P_a / R_a} = 0.622 \frac{P_v}{P_a}$$

… and if *P* is the total pressure or atmospheric pressure …

$$\omega = \frac{0.622\, P_v}{P - P_v}$$

Consider 1 kg of dry air, then by definition, dry air contains no water vapour, and thus its specific humidity is zero. When water vapour is added to this dry air the specific humidity will increase. As more vapour or moisture is added, the specific humidity will keep increasing until the air can hold no more moisture. At this point, the air is said to be saturated with moisture, and it is called **saturated air**.

Any further moisture introduced into saturated air will condense. The amount of water vapour in saturated air at a specified temperature and pressure can be determined from the above equation by replacing P_v with P_g, the saturation pressure of water at that temperature.

The ratio of amount of moisture the air holds (m_v) relative to the maximum the air can hold at the same temperature (m_g) is referred to as the **relative humidity** ϕ.

$$\phi = \frac{m_v}{m_g} = \frac{P_v V/(R_v T)}{P_g V/(R_v T)} = \frac{P_v}{P_g}$$

… where $P_g = P_{sat\ @\ T}$

Combining the above equations allows the relative humidity to be expressed as …

$$\phi = \frac{\omega P}{(0.622 + \omega) P_g} \quad \text{and} \quad \omega = \frac{0.622 \phi P_g}{P - \phi P_g}$$

EXAMPLE 3 The Amount of Water Vapor in Room Air

A 5-m × 5-m × 3-m room shown in Fig. 18 contains air at 25°C and 100 kPa at a relative humidity of 75 percent. Determine (*a*) the partial pressure of dry air, (*b*) the specific humidity, (*c*) the enthalpy per unit mass of the dry air, and (*d*) the masses of the dry air and water vapor in the room.

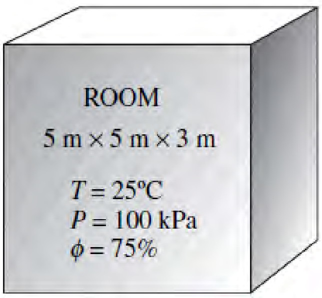


FIGURE 18

SOLUTION A sketch of the room is given in Fig. 18. Both the air and the vapor fill the entire room, and thus the volume of each gas is equal to the volume of the room.

Assumptions The dry air and the water vapor in the room are ideal gases.

Analysis (*a*) The partial pressure of dry air can be determined from

$$P_a = P - P_{v+}$$

where

$$P_v = \phi P_g = \phi P_{\text{sat @ 25°C}} = (0.75)(3.169\text{ kPa}) = 2.38\text{ kPa}$$

Thus,

$$P_a = (100 - 2.38)\text{ kPa} = \mathbf{97.62\ kPa}$$

(*b*) The specific humidity of air is

$$\omega = \frac{0.622P_v}{P - P_v} = \frac{(0.622)(2.38\text{ kPa})}{(100 - 2.38)\text{ kPa}} = \mathbf{0.0152\ kg\ H_2O/kg\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from Eq. 33, where h_g is taken from Table A–4:

$$\begin{aligned} h &= h_a + \omega h_v \cong C_pT + \omega h_g \\ &= (1.005\text{ kJ/kg}\cdot°\text{C})(25°\text{C}) + (0.0152)(2547.2\text{ kJ/kg}) \\ &= \mathbf{63.8\ kJ/kg\ dry\ air} \end{aligned}$$

The enthalpy of water vapor (2547.2 kJ/kg) could also be determined from the approximation given by Eq. 25:

$$h_{g\text{ @ 25°C}} \cong 2501.3 + 1.82(25) = 2546.8\text{ kJ/kg}$$

which is very close to the value obtained from Table A–4.

(*d*) Both the dry air and the water vapor fill the entire room completely. Therefore, the volume of each gas is equal to the volume of the room:

$$V_a = V_v = V_{\text{room}} = (5)(5)(3) = 75\text{ m}^3$$

The masses of the dry air and the water vapor are determined from the ideal-gas relation applied to each gas separately:

$$m_a = \frac{P_aV_a}{R_aT} = \frac{(97.62\text{ kPa})(75\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = \mathbf{85.61\ kg}$$

$$m_v = \frac{P_vV_v}{R_vT} = \frac{(2.38\text{ kPa})(75\text{ m}^3)}{(0.4615\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = \mathbf{1.3\ kg}$$

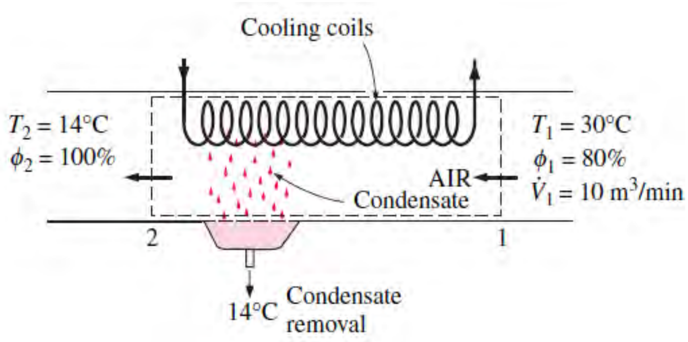
The mass of the water vapor in the air could also be determined from Eq. 27:

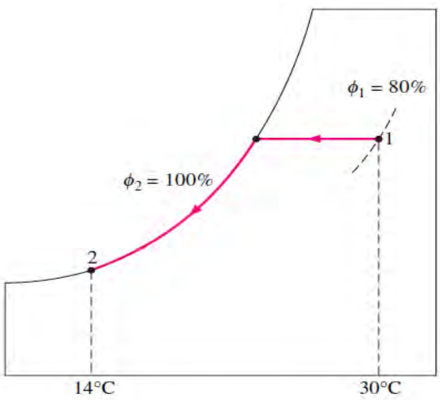
$$m_v = \omega m_a = (0.0152)(85.61\text{ kg}) = 1.3\text{ kg}$$

The specific humidity of air remains constant during a simple sensible cooling process, but its relative humidity increases. If the relative humidity reaches undesirably high levels, it may be necessary to remove some moisture from the air, that is, to dehumidify it. This requires cooling the air below its dew-point temperature.

Theoretically, the cooling process with dehumidifying is illustrated schematically and on the following psychrometric chart in conjunction with Example 8 where hot, moist air enters the cooling section at state 1. As it passes through the cooling coils, its temperature decreases and its relative humidity increases at constant specific humidity. If the cooling section is sufficiently long, air will reach its dew point (state 2, saturated air). Further cooling of air results in the condensation of part of the moisture in the air. Air remains saturated during the entire condensation process, which follows a line of 100 percent relative humidity until the final state (state 3) is reached. The water vapor that condenses out of the air during this process is removed from the cooling section via a drain pan.

The cool, saturated air at state 2 is usually routed directly to the room, where it mixes with the room air. In some cases, however, the air at state 2 may be at the right specific humidity but at a very low temperature. In such cases, air is passed through a heating section where its temperature is raised to a more comfortable level before it is routed to the room.





Note : The 'real' shape of the cooling process line for a tube & fin heat exchanger is a complex curve starting with sensible cooling and then approaching the outlet conditions …

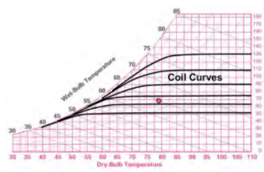


FIGURE 36

EXAMPLE 8 Cooling and Dehumidification of Air

Air enters a window air conditioner at 1 atm, 30°C, and 80 percent relative humidity at a rate of 10 m^3/min, and it leaves as saturated air at 14°C. Part of the moisture in the air that condenses during the process is also removed at 14°C. Determine the rates of heat and moisture removal from the air.

SOLUTION We take the *cooling section* to be the system. The schematic of the system and the psychrometric chart of the process are shown in Fig. 36.

We note that the amount of water vapor in the air decreases during the process ($\omega_2 < \omega_1$) due to dehumidification.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and the water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis Applying the mass and energy balances on the cooling and dehumidification section gives

Dry air mass balance: $\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a$

Water mass balance: $\dot{m}_{a_1}\omega_1 = \dot{m}_{a_2}\omega_2 + \dot{m}_w \rightarrow \dot{m}_w = \dot{m}_a(\omega_1 - \omega_2)$

Energy balance: $\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e \rightarrow$

$$\dot{Q}_{out} = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. Therefore, we can determine the properties of the air at both states from the psychrometric chart to be

$$h_1 = 85.4 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0216 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.889 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 39.3 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0100 \text{ kg H}_2\text{O/kg dry air}$$

Also,

$$h_w = h_{f\,@\,14°C} = 58.8 \text{ kJ/kg} \qquad \text{(Table A–4)}$$

Then

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{10 \text{ m}^3\text{/min}}{0.889 \text{ m}^3\text{/kg dry air}} = 11.25 \text{ kg/min}$$

$$\dot{m}_w = (11.25 \text{ kg/min})(0.0216 - 0.0100) = \mathbf{0.131\ kg/min}$$

$$\dot{Q}_{out} = (11.25 \text{ kg/min})[(85.4 - 39.3) \text{ kJ/kg}] - (0.131 \text{ kg/min})(58.8 \text{ kJ/kg})$$
$$= \mathbf{511\ kJ/min}$$

Therefore, this air-conditioning unit removes moisture and heat from the air at rates of 0.131 kg/min and 511 kJ/min, respectively.

Note : The above example calculates the 'useful' energy available at the outlet of the process … bearing in mind that the condensate has been drained away and is no longer a useful component and thus deducted from the final answer!

When we perform a 'coil calculation', we need to base the required capacity and thus surface requirements upon the energy to cool and dry air component and condense the moisture from the air stream. Therefore the energy associated with generation of the condensate is NOT deducted from the calculation.

Biot number (Bi)	$\frac{hL}{k_s}$	Ratio of the internal thermal resistance of a solid to the boundary layer thermal resistance.
Mass transfer Biot number (Bi_m)	$\frac{h_m L}{D_{AB}}$	Ratio of the internal species transfer resistance to the boundary layer species transfer resistance.
Bond number (Bo)	$\frac{g(\rho_l - \rho_v)L^2}{\sigma}$	Ratio of gravitational and surface tension forces.
Coefficient of friction (C_f)	$\frac{\tau_s}{\rho V^2/2}$	Dimensionless surface shear stress.
Eckert number (Ec)	$\frac{V^2}{c_p(T_s - T_\infty)}$	Kinetic energy of the flow relative to the boundary layer enthalpy difference.
Fourier number (Fo)	$\frac{\alpha t}{L^2}$	Ratio of the heat conduction rate to the rate of thermal energy storage in a solid. Dimensionless time.
Mass transfer Fourier number (Fo_m)	$\frac{D_{AB} t}{L^2}$	Ratio of the species diffusion rate to the rate of species storage. Dimensionless time.
Friction factor (f)	$\frac{\Delta p}{(L/D)(\rho u_m^2/2)}$	Dimensionless pressure drop for internal flow.
Grashof number (Gr_L)	$\frac{g\beta(T_s - T_\infty)L^3}{\nu^2}$	Measure of the ratio of buoyancy forces to viscous forces.
Colburn j factor (j_H)	$St\, Pr^{2/3}$	Dimensionless heat transfer coefficient.
Colburn j factor (j_m)	$St_m\, Sc^{2/3}$	Dimensionless mass transfer coefficient.
Jakob number (Ja)	$\frac{c_p(T_s - T_{sat})}{h_{fg}}$	Ratio of sensible to latent energy absorbed during liquid–vapor phase change.
Lewis number (Le)	$\frac{\alpha}{D_{AB}}$	Ratio of the thermal and mass diffusivities.
Nusselt number (Nu_L)	$\frac{hL}{k_f}$	Ratio of convection to pure conduction heat transfer.
Peclet number (Pe_L)	$\frac{VL}{\alpha} = Re_L\, Pr$	Ratio of advection to conduction heat transfer rates.
Prandtl number (Pr)	$\frac{c_p \mu}{k} = \frac{\nu}{\alpha}$	Ratio of the momentum and thermal diffusivities.
Reynolds number (Re_L)	$\frac{VL}{\nu}$	Ratio of the inertia and viscous forces.
Schmidt number (Sc)	$\frac{\nu}{D_{AB}}$	Ratio of the momentum and mass diffusivities.
Sherwood number (Sh_L)	$\frac{h_m L}{D_{AB}}$	Dimensionless concentration gradient at the surface.
Stanton number (St)	$\frac{h}{\rho V c_p} = \frac{Nu_L}{Re_L\, Pr}$	Modified Nusselt number.
Mass transfer Stanton number (St_m)	$\frac{h_m}{V} = \frac{Sh_L}{Re_L Sc}$	Modified Sherwood number.
Weber number (We)	$\frac{\rho V^2 L}{\sigma}$	Ratio of inertia to surface tension forces.

NEWTON'S LAW OF COOLING

States that the rate of heat loss of a body is directly proportional to the difference in the temperatures between the body and its surroundings, provided the temperature difference is small.

In essence, the convective heat transfer is proportional to the temperature difference.

Q = h Δt

where, q *= local heat flux density, W/m²*
h *= heat transfer coefficient, W/m²/K*
Δt *= temperature difference, K*

The convective heat transfer coefficient is governed by the physical properties of the fluid and its surroundings and derived experimentally.

Magnitudes of the coefficient depend upon whether the flow regime is laminar (low) or turbulent (higher) which is affected by the growth and thickness of the boundary layer, which is related to the fluid velocity, the surface roughness and fluid properties.

Typical values of the convective heat transfer	
Process	h $(\mathrm{W/m^2 \cdot K})$
Free convection	
Gases	2 - 20
Liquids	50 - 1000
Forced convection	
Gases	25 - 300
Liquids	100 - 40,000
Convection with phase change	
Boiling or condensation	2500 - 100,000

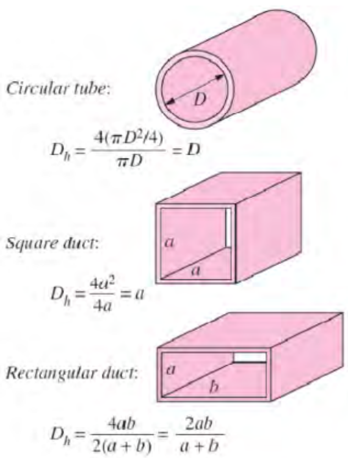
HYDRAULIC DIAMETER - D_H

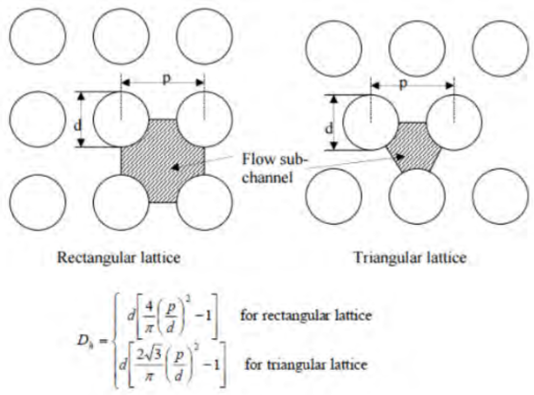
Generally, many thermodynamic and fluid dynamic correlations use a 'diameter' parameter. For a simple circular tube or pipe this is its internal or external diameter. When the channel is not a simple circular tube or pipe but perhaps non-circular, rectangular or triangular, then an 'equivalent diameter' is often used, also known as the hydraulic diameter and defined as …

D_H = 4 x A / P

where, D_H *= hydraulic or equivalent diameter, m*
A *= cross-sectional area, m²*
P *= wetted perimeter, m*

For a simple circular pipe of diameter D, $A = \pi D^2/4$ *and* $P = \pi D$*, resulting in* $D_H = D$





REYNOLDS NUMBER - RE

Reynolds number is a fluid mechanics dimensionless number that defines flow patterns and the onset of the laminar, transitional and turbulent flow regimes. Furthermore, but not applicable in our industry sector, it is used in scaling of fluid dynamic problems to determine dynamic similitude.

The concept was introduced by *Sir George Stokes* in 1851 but named by *Arnold Sommerfeld* in 1908 after *Osborne Reynolds* (1842-1912). It is denoted by the symbols **Re** and has no units.

Re = ρ v D / μ

where, ρ = density, kg/m3
v = velocity, m/s
D = diameter, m
μ = dynamic viscosity, Pa.s

The 3 flow regimes are traditionally defined as follows …

- Laminar - Re < 2100 (*or < 2300 in many American references*)
- Transitional - 2100 < Re > 7000 (*although some references suggest up to 4000*)
- Turbulent – Re > 7000 (*or 4000*)

The Reynolds number is used in both pressure drop calculations, where the Friction Factor is dictated by the flow regime and heat transfer coefficient calculations.

FRICTION FACTOR - Ƒ

The Friction Factor is denoted by the symbol **$ƒ_D$** and is dimensionless (*otherwise known as the Darcy-Weisbach friction factor*), is a parameter used in the calculation of heat transfer coefficient and pressure loss in a tube or pipe and is both roughness and Reynolds Number dependent.

As the characteristic is quite complex it is often represented on a Moody diagram (*named after L.F. Moody*), see below.

However, for laminar flow when Re < 2100 to 2300 …

$ƒ_D$ = 64 / Re

Above the laminar Reynolds number threshold, the friction factor calculation becomes rather more complex, but an accurate implicit correlation is the *Colebrook* equation which includes Ɛ, the roughness of the inside of the tube.

$$\frac{1}{\sqrt{f_D}} = -2.00 \log\left(2.51\frac{1}{\mathrm{Re}\sqrt{f_D}} + \frac{1}{3.7}\frac{\varepsilon}{D}\right)$$

However, this <u>must</u> be solved iteratively, so some simpler equations are …

$$f = 0.316 Re_D^{-1/4} \qquad Re_D \lesssim 2 \times 10^4$$
$$f = 0.184 Re_D^{-1/5} \qquad Re_D \gtrsim 2 \times 10^4$$

Alternatively, a single correlation that encompasses a large Reynolds number range has been developed by *Petukhov* and is of the form …

$$f = (0.790 \ln Re_D - 1.64)^{-2} \qquad 3000 \lesssim Re_D \lesssim 5 \times 10^6$$

However, Evapco's preferred explicit correlation, including the Ɛ, the roughness is by Fang at al. (2011) …

$$f = 1.613\, [\mathrm{Log_e}(0.234\, (Ɛ / D)^{1.1007} - 60.525 / Re^{1.1105} + 56.291 / Re^{1.0712}\,]^{-2}$$

where, Ɛ = Roughness, μm
1.5 μm : Copper
15 μm : SS304/316

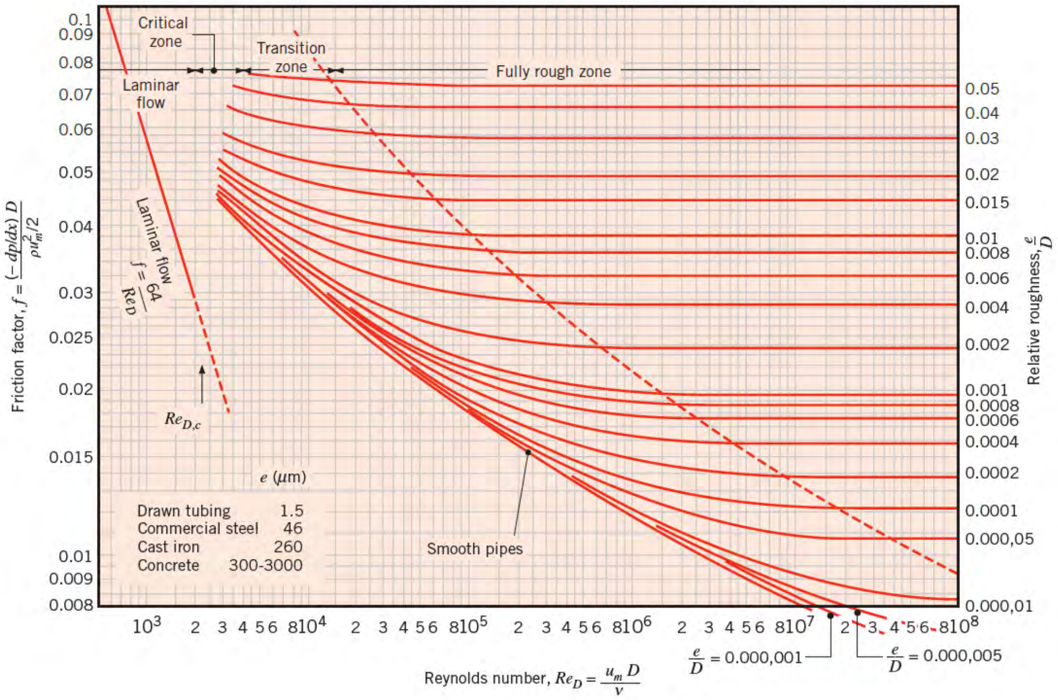
However, a simple to remember and adequate for approximation purposes for smooth tubes in the transitional/turbulent flow, where Re > 2100 …

$ƒ_D$ = 0.2 / $Re^{0.2}$

The following Moody diagram corresponds to the Colebrook equation, includes the relative roughness parameter, which will both increase the pressure drop and enhance the heat transfer coefficient if the internal tube surface is rough.

However, the tubes used in our products are considered smooth and thus the simplified equations are justified.

Moody Diagram



Note that the Reynolds number is shown in terms of the kinematic viscosity and not density & dynamic viscosity

***Beware** : There are **two** friction factor definitions commonly used in fluid dynamics, the Darcy-Weisbach friction factor, f_D as used above and the Fanning friction factor, f which is ¼ of the magnitude of f_D.*

Thus, the laminar flow equation becomes 16 / Re.

*Care should be taken when reading charts that use the term **Friction Factor**, but do not provide a definition.*

PRANDTL NUMBER - PR

The Prandtl number is denoted by the symbols **Pr** and named after *Ludwig Prandtl* (1875 – 1953) is dimensionless and expresses the ratio of momentum diffusivity to thermal diffusivity of a fluid and is defined as …

Pr = C_p µ / k

where, C_p *= specific heat, kJ/kg/K*
µ = dynamic viscosity, mPa.s
k = thermal conductivity, W/m/K

The equation is used when deriving heat transfer coefficients, but its magnitude can indicate the fluid's suitability for different purposes.

Typically for air, Pr ≈ 0.7, for water Pr ≈ 6.9 @ 20°C and for oils Pr may be in the range from 50-100, but because Pr is viscosity dependent, temperature can affect the magnitude considerably.

NUSSELT NUMBER - NU

The Nusselt number is denoted by the symbols **Nu** and named after *Wilhelm Nusselt* (1882 - 1957), is dimensionless and expresses the ratio of convective to conductive heat transfer across a fluid boundary and is defined as …

Nu = h D / k

where, h = convective heat transfer coefficient, W/m²/K
D = characteristic dimension, m
k = thermal conductivity, W/m/K

The characteristic dimension, D is dependent upon the correlation that the Nusselt number is applied to. It may be the internal or outer diameter of a tube/pipe, the hydraulic diameter or some other defined parameter, such as the centre-to-centre distance between tube rows in a coil.

STANTON NUMBER – ST

The Stanton number is a combination of the Nusselt, Reynolds and Prandtl numbers and is obviously dimensionless too.

St = Nu / (Re Pr)

which resolves to …

St = h / (ρ C_p V) ***which is a term used in the Colburn J factor, see below***

COLBURN MODULUS – J FACTOR

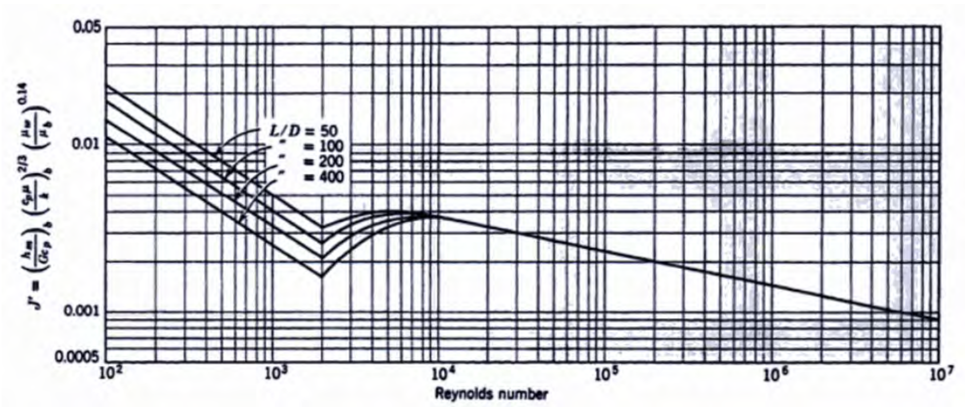
During the extensive fluid dynamics work that Reynolds conducted, he concluded that the heat transfer coefficient was proportional to the friction factor, which led to the following ***Reynolds analogy*** …

$$\frac{f}{8} = \frac{h}{\rho C_p V}$$

Colburn took this a stage further and quantified the empirical Colburn J factor by introducing the Prandtl number, which had previously been considered to be unity …

$$j_H = \frac{f}{8} = \frac{h}{\rho C_p V} \mathrm{Pr}^{2/3}$$

Further embellishments introduced the ratio of the viscosity at the wall temperature to the viscosity at the bulk temperature, albeit that this ratio is often considered as unity and when plotted against the Reynolds number yields the following Colburn Modulus plot …



Use of the above Colburn Modulus plot provides a quick and simple method to ascertain the $\mathbf{J_H}$ from the calculated Reynolds number, **Re** for the given fluid and then solving for ***h***, the internal heat transfer coefficient, by rearranging the above equation.

However, our coil and product selection software use the fundamental equations as described both above and in the following sections.

INTERNAL TUBE H_I COEFFICIENTS

Internal heat transfer coefficients in our scope of supply are generally confined to smooth circular tubes and can encompass both single and two phase flow.

Dry coolers generally cope with single phase fluids – liquids, vapours or gases, whilst air cooled condensers handle fluids that change phase such as refrigerants or on occasions, steam.

Single phase heat transfer, be it a liquid or vapour/gas is relatively straight forward to calculate and involves knowledge of the four thermo-physical properties of the fluid at both the operating pressure and mean fluid temperature, namely …

- Density, kg/m^3
- Specific heat, kJ/kg/K
- Thermal conductivity, $W/m^2/K$
- Dynamic viscosity, mPa.s

Conversely, two phase heat transfer coefficients involve rather more complex correlations and involves the knowledge of both the saturated vapour and saturated liquid properties, including enthalpies, surface tension and pressure temperature relationship at the operating pressure and condensing temperature.

Often for two phase analysis, the use of NIST RefProp thermodynamic data calculation tool is required to glean the necessary data.

Unlike the single phase algorithms, which apply for both heating and cooling scenarios, two phase evaporation (cooling) uses somewhat different correlations from those applicable to two phase condensation (heating).

SINGLE PHASE

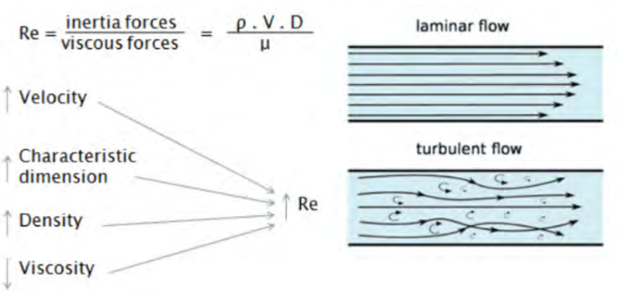
Refers to the behaviour of the 'fluid', where 'in English' the word 'fluid' can refer to a gas, a vapour or a liquid. Single phase behaviour indicates that the fluid remains in the same state throughout the entire heating or cooling process.

Typically, such fluids might be water, water/glycol mixes, oils, compressed air or other gases requiring cooling or heating.

In the past the Reynolds Number dependent flow behaviour was divided into 3 regimes … laminar, transitional & turbulent, however, as described below, the flow behaviour can now be handled by just two correlations.

LAMINAR FLOW

The evolution of the correlations and equations used to describe the heat transfer coefficient for single phase fluids is quite extensive and it is not the purpose of this guide to discuss the detailed history or derivation, but present the formulae currently used by the selection software.



Traditionally, the flow inside a tube or pipe is considered to be either laminar (also known as streamline or viscous) or turbulent, where the Reynolds number is the parameter used to identify the flow regime.

Typically, Re < 3000 to 4000 is considered to be laminar (*albeit that Re = 2100 … or 2300 in USA, is the traditional delimiter*), whilst above this figure is deemed to be turbulent. Clearly, there is the 'fuzzy' transitional regime, but let's not dwell on this.

Wilhelm Nusselt was instrumental in defining the Nusselt equation, which associates a variety of parameters to describe heat transfer in the form …

Nu ø (Re, Pr, L/D, μ_b/μ_w) *where, Nu = Nusselt number*

Re = Reynolds number

Pr = Prandtl number

L = *length, m*
D = *diameter, m*
μ_b = *bulk dynamic viscosity, mPa.s*
μ_w = *wall dynamic viscosity, mPa.s*

Work conducted by Sieder and Tate culminated in the classical equation for laminar flow …

$$Nu = 1.86\,\mathrm{Re}^{1/3}\,\mathrm{Pr}^{1/3}\left(\frac{D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

This expression has limitations as it implies that the Nusselt number approaches zero as the length increases and is thus only applicable to the thermal entrance region.

However, for long tubes – *generally the situation we encounter* - most of the heat transfer occurs in the thermally fully-developed region, where the Nusselt number is nearly a constant and independent of any of the above parameters.

When the wall temperature is uniform and the fluid velocity low, Nu approaches a constant value of 3.66

In 1999 *A.F. Mills* proposed a more generic equation that considered both the limiting conditions and long tube length anomaly …

$$Nu = \frac{hD}{k} = 3.66 + \frac{0.065\,\mathrm{Re\,Pr}\,\dfrac{D}{L}}{1+0.04\left(\mathrm{Re\,Pr}\,\dfrac{D}{L}\right)^{2/3}}$$

Interestingly, short tube heat exchangers can exhibit greater than predicted heat transfer coefficients, due to 'induced turbulence' from the 180° bends and insufficient tube length to generate fully developed laminar flow, creating a higher than calculated, pseudo-Reynolds number.

TRANSITIONAL FLOW

As previously mentioned, Laminar flow extends to Reynolds numbers of 2100 to 2300 whilst Turbulent flow begins, according to the Moody & Colburn Modulus diagrams mentioned above, around Reynolds numbers > 3000 or 4000 … however, other sources suggest >7000 and certainly, above 10000 the flow is fully developed.

The discontinuity at around Re = 2100 … 2300 poses chaotic behaviour when iterative calculations are being performed that finish up in this Transitional region.

A technique to resolve this issue is to apply an exponential 'smoothing' function to the transitional zone between Re = 2300 and 3000.

Furthermore, for Re < 1500, the magnitude of the Nusselt Number is affected by the Prandlt Number and if Pr <= 5, the Sieder-Tate correlation is used, whilst for Pr > 5, the Mills correlation is used. Finally, if the resulting Nusselt Number is < 3.66 then Nu = 3.66.

In the range 1500 < Re > 3000 where a smoothing function is implemented, the appropriate Laminar Nu_{Low} at Re = 1500 and the Gnielinski Turbulent Nu_{High} at Re = 3000 are used to arrive at the 'blended' Nusselt Number.

Smoothing function …

$W = 1 / (1 + e^{(-0.005 \times (Re - 2300))})$

$Nu = (1 - W) \times Nu_{Low} + W \times Nu_{High}$

TURBULENT FLOW

The classical expression for computing the local Nusselt number for fully turbulent flow in a smooth tube is attributed to *Dittus and Boelter* …

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{n}$$

When the fluid is being heated, n = 0.4 and when being cooled, n = 0.3

The above correlation is known to have limitations yielding errors as high as 25%, so in 1976 *Gnielinski* proposed a new equation which covers both the **transitional** and **turbulent** regimes with a Reynolds number validity of 3000 <= Re <= 5 x 10^6

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\,\mathrm{Pr}}{1+12.7\left(\mathrm{Pr}^{2/3}-1\right)\sqrt{f/8}}$$

This equation is the foundation for estimating the internal heat transfer coefficient for any single phase fluid, whose Reynolds number is >= 3000.

DESUPERHEATERS

In the case of refrigerant condensers that are fed with superheated gas from the compressor, the desuperheating of the refrigerant is usually completed during the first pass or couple of passes of the condenser circuit and considered part of the total heat of rejection of the condenser.

However the terminology 'desuperheater' is usually reserved for a dedicated coil/unit or a dedicated portion of a condenser designed to either partially cool down a single phase gas; which is above its saturated vapour temperature (*usually some sort of refrigerant*); or completely cool the superheated gas down to its saturation temperature, ready for further condensing, such as in the case of an Ammonia refrigeration system.

For example, the high pressure, hot superheated gas exiting from the compressor at say 120°C is required to be cooled to saturation i.e. 35°C if this is the condensing temperature.

This process is purely a single phase process and is handled in a similar fashion to a straightforward air heating water coil, where the water (*in this case a single phase liquid*) is cooled whilst the cooling air stream is heated. However, by comparison, the single phase cooling of a gas or vapour is a far less thermodynamically efficient process and the cooling of a single phase liquid.

As the internal tube fluid is a gas rather than a liquid, the traditional 2.0 m/sec liquid velocity restrictions can be breached and gas velocities of 10 to 30 m/sec can be considered … the density & viscosity of a gas permit such velocities whilst generated somewhat lower friction pressure drops.

If the gas being cooled is <u>not</u> condensed at some later stage in the process, then the product would not normally be referred to as a 'desuperheater', but an ordinary single phase cooler.

SUBCOOLERS

Essentially a subcooler is a normal air heating coil where ambient air is used to cool down a single phase liquid in the tubes. So cooling water in a coil is a 'subcooling' process. If the liquid water is below its boiling point, it is already a subcooled liquid !!

However, the term 'subcooler' is either reserved for the subcooling of steam after a condensing process or relates to the cooling down of the saturated refrigerant liquid exiting an air cooled condenser prior to being fed to an expansion device (TEV).

Unlike single phase heating and cooling processes, where the fluid remains in the same state during the process e.g. either a liquid or a gas, the two phase process involves a change of phase as heat is transferred to or from the fluid.

CONDENSATION

INTRODUCTION

In the case of a condensing process, the entering superheated hot gas is firstly de-superheated to saturation and whilst dissipating latent energy to the cooling air stream, changes phase from a gas to a liquid.

There have been many algorithms proposed to predict the internal condensing two phase heat transfer coefficient, however many of these are related to specific flow regimes and may or may not be ΔT (difference between the saturation and internal wall temperature) dependent. Furthermore, such algorithms are not necessarily suitable for both azeotropic and zeotropic refrigerants.

Cavellini et al. (*Condensation in horizontal smooth tubes : A new transfer model - 2006*) proposed a new and simplified methodology which employed only two equations, related respectively to ΔT -independent and to ΔT -dependent fluid flows.

INTERNAL HEAT TRANSFER COEFFICIENT METHODOLOGY

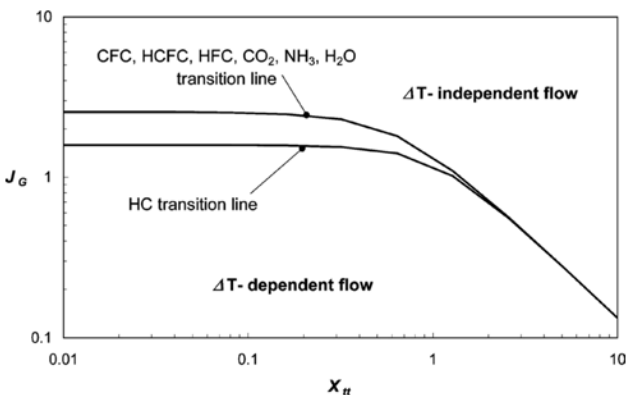
Experimental results identified deviations in the measured local heat transfer coefficient in the upper and lower part of the same cross-section of a tube during the condensation of some natural and CFC fluids. It transpired that there was a ΔT-dependence when the ratio between the two values exceeded unity.

A transition indicator was identified in terms of the dimensionless gas velocity (or modified Froude number) J_G as follows …

$$J_G = xG / [gD\rho_G(\rho_L - \rho_G)]^{0.5}$$

where, x = *Vapour quality*
G = *Mass velocity, kg/m²/s*
D = *Inside tube diameter, m*
ρ_G = *Vapour density, kg/m³*
ρ_L = *Liquid density, kg/m³*

The figure shows the transition between the ΔT -dependent and the ΔT -independent flow regime, Eq. 1 below.



A simple relation for the transition value of J_G (or J_{TG}) as a function of X_{tt} – Martinell two phase multiplier

$$J_{TG} = \{[7.5/(4.3X_{tt}^{1.111}+1)]^{-3}+C_T^{-3}\}^{-1/3} \quad (1)$$

where, C_T = **1.6** *for hydrocarbons*
C_T = **2.6** *for other refrigerants*
$X_{tt} = (\mu_L/\mu_G)^{0.1}(\rho_G/\rho_L)^{0.5}[(1-x)/x]^{0.9}$

For the ΔT-independent flow regime, a simple two-phase multiplier corrects the liquid phase heat transfer coefficient providing α_A as given in Eq. (2).

For the ΔT -dependent flow regime, the heat transfer coefficient α_D is given in Eq. (3) related to the heat transfer coefficient α_A and to a fully-stratified flow heat transfer coefficient (α_{STRAT} from Eq. 5). Thus, Eq. (3) includes a progressive transition from the wavy-stratified to the smooth stratified flow.

ΔT-independent flow regime ($J_G > J_{TG}$) ~ Annular & Slug flow coefficient:

$$\alpha_A = \alpha_{LO}[1+1.128x^{0.817}(\rho_L/\rho_G)^{0.3685}(\mu_L/\mu_G)^{0.2363} \times (1-\mu_G/\mu_L)^{2.144} Pr_L^{-0.100}] \quad (2)$$

ΔT-dependent flow regime ($J_G \le J_{TG}$) ~ Stratified-smooth & wavy flow & Slug flow coefficient:

$$\alpha_D = [\alpha A\,(J_{TG}/J_G)^{0.8} - \alpha_{STRAT}]\,(J_G/J_{TG}) + \alpha_{STRAT} \quad (3)$$

Liquid phase coefficient:

$$\alpha_{LO} = 0.023\,Re_{LO}^{0.8}\,Pr_L^{0.4}\,\lambda_L/D \quad (4)$$

Fully Stratified flow coefficient:

$$\alpha_{STRAT} = 0.725\{1+0.741[(1-x)/x]^{0.3321}\}^{-1} \times [\lambda_L^3 \rho_L(\rho_L-\rho_G)\,g\,h_{LG}/(\mu_L D\,\Delta T)]^{0.25} + (1-x^{0.087})\alpha_{LO} \quad (5)$$

where, $\Delta T = (T_s - T_w)$ (6)

$T_w = T_{AirIn} + [(A_i/A_o)\,\alpha_A / (\alpha_{Air} + (A_i/A_o)\,\alpha_A)] \times (T_s - T_{AirIn})$ (7)

T_s *= Saturation temperature, °C*

T_w *= Internal tube wall temperature, °C*

A_i *= Inside tube surface area, m²*

A_o *= Outside fin surface area, m²*

T_{AirIn} *= Air inlet temperature, °C*

α_{Air} *= Airside heat transfer coefficient, W/m²/K*

μ_L *= Liquid viscosity, Pa.s*

λ_L *= Liquid thermal conductivity, W/m/K*

h_L *= Latent heat, J/kg*

g *= Acceleration, 9.81 m/s²*

Re_{LO} *= Liquid Reynolds number,* $G\,D/\mu_L$

Pr_L *= Liquid Prandtl number,* $c_{pL}\,\mu_L/\lambda_L$

For zeotropic mixtures (*refrigerant blends with a temperature glide*), the impact upon the total heat transfer coefficient (8) resulting from the thermal resistance of the liquid film is approximated using the Silver-Bell-Ghaly modifier as follows …

$$1/\alpha_{Eff} = 1/\alpha_{A\,or\,D} + Z/\alpha_G \quad (8)$$

where, α_{Eff} *= Modified total heat internal heat transfer coefficient, W/m²/K*

$\alpha_G = 0.023\,Re_G^{0.8}\,Pr_G^{0.4}\,\lambda_G/D$, *W/m²/K {Gas phase}*

$Z = x\,c_{pG} \times Glide / dh$

$dh = [(c_{pL} + c_{pG})/2] \times Glide + (h_{DEW} - h_{BUB})$

$Glide = (T_{DEW} - T_{BUB})$, *K*

h_{DEW} *= Enthalpy at Dew Point, J/kg*

h_{BUB} *= Enthalpy at Bubble Point, J/kg*

TWO PHASE STRAIGHT TUBE PRESSURE DROP METHODOLOGY

The two phase straight tube pressure drop utilises the Bo Pierre (1964) correlation modified by - Choi, Kedzierski, Domanski, 1999

$\Delta P = G^2 \{f\ L \times (sv_{out} + sv_{in}) / D + (sv_{out} - sv_{in})\} / 1000$, kPa

where, L *= Tube length, meters*

sv_{in} *= Specific volume @ inlet, m3/kg*

sv_{out} *= Specific volume @ outlet, m3/kg*

$f = 0.00506\ Re_L^{-0.0951}\ K^{0.1554}$ *… Two phase friction factor*

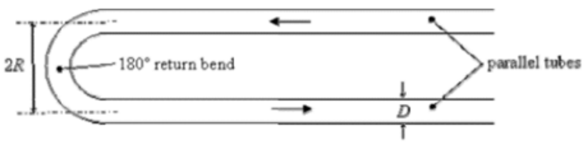
$K = \Delta x\ (h_{in} - h_{out}) / L /$ *… Bo Pierre Boiling No. (Two Phase No.)*

h_x *= Inlet/outlet enthalpy, J/kg*

Δx *= 1.0 for condensers*

RETURN BEND PRESSURE DROP METHODOLOGY

Calculation of the nominal return bend pressure drop for one 180° bend based upon the average two phase quality proposed by P.A. Domanski & C.J.L. Hermes ~ NIST 2006 providing an improved correlation to the methodology first propose by Muller-Steinhagen & Heck for straight tubes with a modifier derived from Geary & Chen's experimental data.



To calculate the nominal return bend pressure drop for one bend based upon the weighted average two phase quality …

$\Delta P_{rb} = C \times \Delta P_{str} / 1000$, kPa per bend

where, ΔP_{str} *= Straight tube pressure drop, kPa/m*

C *= Modifier applied to the straight tube press drop*

and

$C = 0.0065\ (G\ x_{av}\ D / \mu_G)^{0.54} \times (1 / x_{av} - 1)^{0.21} \times (\rho_L / \rho_G)^{0.34} \times (Z_{av} / D)^{-0.67}$ *.. Straight tube modifier*

$Z_{av} = [T_p + 2T_s + ((T_p/2)^2 + T_s^2)^{0.5}] / 3 / 1000$ *.. Average bend centreline diameter*

$x_{av} = 0.667$ *.. Weighted average quality*

where, T_p *= Tube pitch, mm*

T_s *= Tube stagger (row pitch), mm*

$\Delta Pstr = [\Delta P_L + 2\ x_{av}\ (\Delta P_V - \Delta P_L)] \times (1 - x_{av})^{0.333} + \Delta P_V\ x_{av}^3$

where liquid & vapour phase pressure drops are defined below …

$\Delta P_x = f_x\ G^2 / 2 / D / \rho_x$ *.. Darcy-Weissbach*

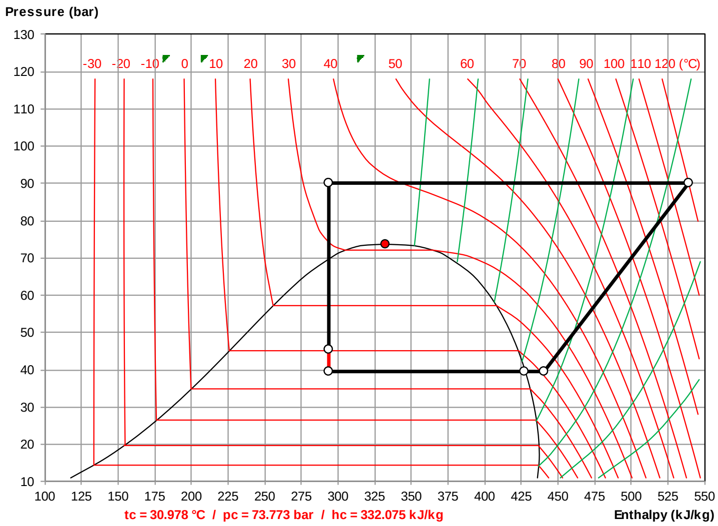
$f = 64 / Re_x$ *.. Paliwoda - Re<=1187*

$f = 0.3164\ Re_x^{-0.25}$ *.. Blasius - Re>1187*

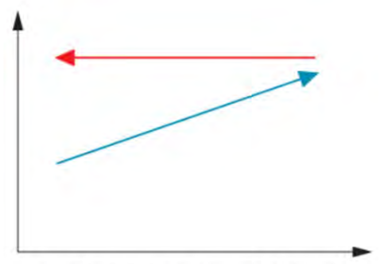
GAS COOLERS

A gas cooler operates in an area above the critical point where there is no distinction between the vapour and liquid phases and where the fluid is sometimes referred to as a 4th phase.

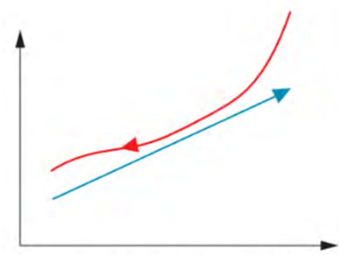
The fluid exhibits some unusual behaviour in this zone and thermophysical properties can change dramatically with small changes in temperature.



Furthermore, unlike a condensing process, which is essentially a constant temperature process, a gas cooler's temperature cooling profile is fairly radical requiring a step-wise calculation process when attempting to calculate the surface requirements for the gas cooler.



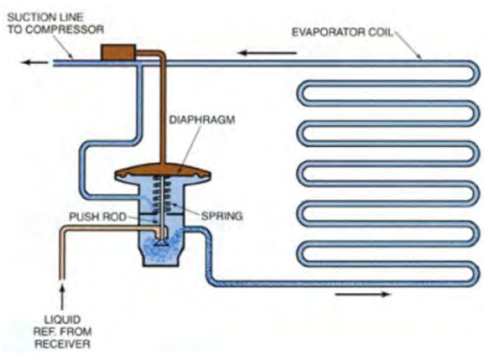
Heat rejection condensing refrigerant



Heat rejection - transcritical CO_2

EVAPORATION

DX EVAPORATORS



The term DX has two common definitions, either Direct Expansion or Dry Expansion and relates to a coil that is fed with refrigerant directly from a thermostatic or electronic expansion valve, via a distributor and distributor/capillary leads, if the coil has more than one circuit per section.

Such systems are usually confined to comfort air conditioning systems below 100 tons (350 kW) of cooling capacity.

A DX coil is controlled via a TEV/EEV where the refrigerant mass flow is modulated. The controlling parameter is the *degree of superheat* sensed at the outlet of the coil, usually on the suction header outlet connection tube.

The sensor indirectly measures the temperature of the vapour leaving the coil and compares it with the saturation temperature equating to the suction/evaporating pressure. The refrigerant mass flow rate is adjusted accordingly to ensure the appropriate superheat is maintained.

A simple refrigeration system using a DX coil relies upon the evaporator generating sufficient superheated vapour to protect the compressor by ensuring that the state of the low temperature refrigerant is indeed a 100% gas and does not entrain any unevaporated liquid slugs, which could destroy the compressor.

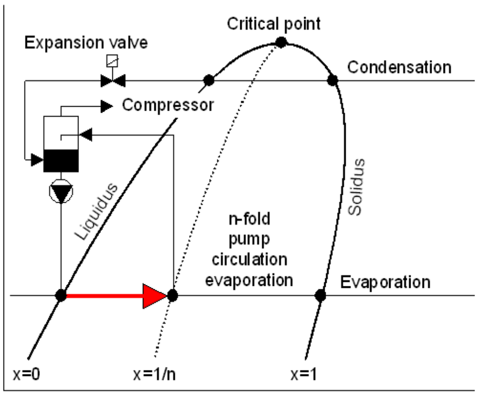
Thus, there is a need for this superheat and hence a portion of the coil surface is allocated for this purpose, at the detriment to the primary purpose of the coil, which is to provide the cooling load.

The two phase heat transfer during the evaporation process results in a high level heat transfer coefficient, but the superheating of the saturated vapour (*dry out zone*) is associated with a low level heat transfer coefficient, which under-utilises this portion of the coil surface.

For stable systems with modern day electronic expansion valves, this superheat setting can be adjusted to 2-3K, however in many 'real life' applications, especially when the older style thermostatic expansion values are fitted, 5K superheat or more may be required for stable operation.

The greater the superheat required, the less efficient the available surface becomes, which accounts for the typically - 15/20% lower capacity that a DX coil can deliver compared with the pump circulated variant described in the following section.

FLOODED/LIQUID OVERFEED/PUMP CIRCULATED EVAPORATORS



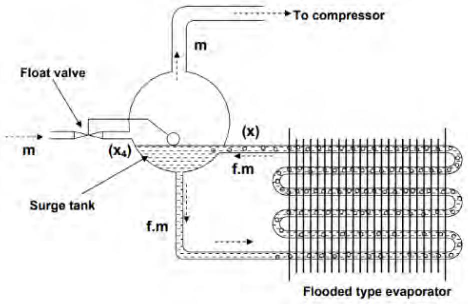
A flooded or overfed evaporator incorporates a vessel which is used as a separator (surge chamber) to allow the evaporated saturated vapour to be routed back to the compressor, whilst the unevaporated liquid can pass through the evaporator once again.

The driving force for the refrigerant flow is both the pressure differential between inlet and outlet of the separator and gravity. The refrigerant flow is regulated via a float valve or similar device which maintains the liquid level in the separator.

Such systems are often used in large air conditioning systems and industrial process refrigeration applications especially where centrifugal compressors or absorption systems are used.

A variation to the flooded system is the pump circulated system which typically overfeeds the liquid via a circulation pump, where the pump factor, **n** ranges between 1.5 to 5 times the theoretical evaporation rate.

Often for ammonia/NH_3/R717 systems, the **n** is more typically 2-3 and the circuitry is designed to achieve a liquid ammonia velocity in the tubes of <= 0.1 m/sec.



Unlike DX coils, which apart from the inlet distributor and capillary leads has vertical headers, flooded & pump circulated systems are fitted with horizontal headers with vertical circuitry.

There are two 'schools of thought' regarding whether such systems should be 'bottom fed' or 'top fed'. EAS favours 'bottom fed' systems, where the circulation rate is usually lower than the alternative method.

Basic coil concept

Pump circulated coils are a little different from the more common DX variety, primarily in the way the liquid inlet and suction connections/headers are positioned and do not utilise a liquid distributor, but only horizontal headers. Furthermore, the circuitry (tube interconnection) differs somewhat from the DX alternative.

A conventional DX system operates on the basis that the sub-cooled liquid leaving the condenser is fed to a TEV (thermal expansion valve) and following expansion enters the DX evaporator via a distributor and capillary leads as a binary mixture of typically 80% liquid and 20% vapour (flash gas) - dryness fraction or quality of 0.2.

As the name implies, the evaporator is designed to completely evaporate the refrigerant plus marginally superheat the suction gas, which is necessary for the TEV to control the refrigerant mass flow rate. Consequently, the required tube length in a circuit of a DX coil is perhaps using the latter 5-10% of its length to perform the superheating process, which when compared with the two phase evaporating process, is far less efficient process.

Hence DX coils, by nature are simple but operate inefficiently, which is where pump circulated (also known as overfeed) systems have a 'performance for size' advantage.

In the case of a pumped system, expansion is performed in a low pressure vessel where the liquid portion of the refrigerant lies in the bottom of the vessel, whilst the vapour portion is drawn off and induced back to the compressor for recompression.

100% saturated liquid from the bottom of the vessel is pumped into the bottom of the evaporator and begins to evaporate whilst it is forced through the coil circuitry in an upward fashion, finally to be collected in the horizontal suction header at the top of the coil and then returned to the low pressure vessel. The returned mixture of vapour and liquid again separate and the unevaporated liquid is pumped through the coil again, whilst the evaporated vapour returns to the compressor.

If the mass flow pumped through the coil was to equate to the …

Duty / Enthalpy Difference [vapour - liquid]

… the system would be referred to as a 'flooded' system and the pump or circulation rate, n = 1. If however the mass flow pumped from the liquid vessel to the coil and back is greater than 1.0 then the system is referred to as an 'overfeed' or pump circulated system with a 'pump rate' or 'circulation rate' of n > 1.

The pump rate, n is often specified by the client and is usually in the range of 1.5 to 5.

As a consequence of the overfeed rate of liquid through the circuitry of the evaporator, the refrigerant is not able to 'dry out' and become superheated (as is the case with a DX coil) and thus this increased internal 'wetted' surface of the inside of the tube, results in an increased internal heat transfer coefficient. Typically, pumped evaporators are upward of 15-20% more effective than a DX evaporator or alternatively require proportionately less surface.

It is generally accepted that for heat exchangers fitted with Ø15 mm or Ø5/8" diameter tubes the circuitry should be adjusted to achieve a refrigerant liquid velocity = 0.1 m/sec (+/- 10%). Too high a velocity will impact upon the refrigerant pressure/temperature drop and adversely affect the operating LMTD. Too low a velocity will result in a lower than preferred internal heat transfer coefficient and again be detrimental to the coil performance and surface (rows) required.

So if the pump rate is pre-specified then the number of circuits must be adjusted to achieve the 0.1 m/sec liquid velocity.

If the pump rate is not defined, then pump rate can be nominated, say 2.0 and the number of circuits is adjusted to reach the 0.1 m/sec liquid velocity.

However, this is where the 'challenge' begins !!

As previously mentioned, a pumped coil is fitted with horizontal headers, the liquid inlet located at the bottom of the coil and the suction outlet at the top and thus the circuit pattern is generally vertically upwards.

Upward fed pump circulated coils is EAS's preferred methodology, however there is 'another school of thought' that favours down fed circuitry, but this is not considered in this guideline.

Generally the number of circuits must be a factor of the number of rows deep. Thus, if we were to have an 8 row deep coil, we could have 4, 8 or 16 circuits resulting in what is often referred to as half, full or double serpentine circuitry. Each circuit should have the same number of tubes or passes … uneven circuitry is not recommended.

If the coil is required to have 'same end' connections, then each circuit needs an 'even' number of tubes per circuit i.e. 2, 4, 6 etc. and if 'opposite end' connections are needed then 1, 3, 5, 7 etc.

If more circuits are required because the pump rate has been specified and with 16 circuits the liquid velocity is still greater than 0.1 m/s, we can consider splitting the coil into 2 sections in the height and thus we can achieve 32 circuits as a maximum. Three sections in the height would allow for 48 circuits and so on.

However, one very important aspect to bear in mind at this juncture is that had the coil had a finned height of 840 mm (14 tubes high and total of 112 tubes), then if the coil was to be built as a 1 section coil, only 4 and 8 circuits would be feasible to achieve 'same end 'connections. 16 circuits would result in 7 tubes per circuit and thus give 'opposite end' connections and thus not suitable.

Similarly, splitting this coil into 2 sections in the height would give 7 tubes high per section … not good as a starter for achieving 'same end' connections. But interestingly in this case, one of the above circuit options is feasible and that is a total of 8 circuits giving 4 circuits in each section resulting in 14 tubes per circuit and thus 'same end' connections !!

If a scenario is reached where the rows deep and circuitry cannot achieve the desired liquid velocity, then in such a case there are a couple of options available ..

- Increase the rows deep to increase the circuit options and then widen the fin pitch so as not to over surface the coil
- Consult the customer and discuss the possibility of adjusting the pump rate to allow the 0.1 m/sec liquid velocity to be achieved

Referring to the above example, if 16 circuits were necessary to reach 0.1 m/sec, plus the coil was required to be split into 2 sections in the height, then this infers each section having 7 tubes in the height and with 8 circuits per section the result would be opposite end connections.

Thus a ‘work around’ to solve this problem would be make the upper section with 6 tubes in the height and the lower section with 8 tubes in the height and each section would have 8 circuits. Both sections would then have an even number of tubes per circuit i.e. 6 and 8 tubes per circuit respectively. Please note that this ‘work around’ is not a preferred method.

The above can be accommodated in a pumped system on the basis that the liquid/vapour mixture exiting from the two dissimilar sections would return to the same low pressure vessel and only liquid be pumped back to the coil sections. The different proportions of evaporated vapour would combine and return to the compressor.

Unlike a DX system where such a disparity would cause havoc to the performance of the system, pumped systems are much more forgiving.

COIL SPECIFIC EQUATIONS

To be able to design a coil to meet a particular capacity firstly involves ensuring that the desired thermal duty on the external finned surface side - typically air, equals the thermal duty on the internal tube side, typically water, but could be steam, a refrigerant or oil etc.

AIR SIDE

VOLUMETRIC FLOW RATE

Air side volumetric flow rate may be provided or derived in a variety of ways, however there are three coil related parameters which govern its calculation …

H	Coil finned height, mm
L	Coil finned length, mm
v	Coil face velocity, m/s

The (finned height x finned length) defines the face area of the coil and the (face area x velocity) defines the volumetric flow rate as follows …

$\dot{V} = (H / 1000) . (L / 1000) . v$ *where,* $\dot{V}$ *= Volume flow rate, m^3/s*

Clearly, if any three parameters are known then the fourth can be calculated to complete the volumetric and physical relationship of the coil aspects.

MASS FLOW RATE

The mass flow rate though a system, assuming no losses, is a constant, but the volume flow rate changes with temperature as the density is affected.

As the temperature rises the density reduces and thus the air volume increases … assuming the mass flow is constant. Therefore, at discrete points through a system, with no losses, the air volume will increase and decrease as the temperature changes. Consequently, the velocity throughout the system varies.

$\dot{m} = \dot{v} \times \rho$ *where,* $\dot{m}$ *= mass flow rate, kg/s*
$\dot{v}$ *= volume flow rate, m^3/s*
ρ *= density, kg/m^3*

DUTY OR CAPACITY

The equation applicable to both the air side (secondary) and internal tube fluid side (primary) involves the mass flow rate of the fluid under consideration and enthalpy difference, as follows …

$Q_T = \dot{m} \times \Delta h$

Alternatively this can be represented by …

$Q_S = \dot{m} \times C_p \times \Delta t$ *for single phase sensible heating/cooling processes*

Thus, for a sensible heating or sensible cooling process …

$Q_T = Q_S$

SENSIBLE HEAT RATIO

Is defined as the ratio between the sensible duty to the total duty and can either be represented as a percentage, such as 86% or as a factor e.g. 0.86

$SHR = Q_S / Q_T$

$= \dot{m} \times C_p \times \Delta t / (\dot{m} \times \Delta h)$

$= C_p \times \Delta t / \Delta h$ *where,* *SHR = sensible heat ratio*
Q_S *= sensible duty, kW*
Q_T *= total duty, kW*

$\dot{m}$ = *mass flow rate, kg/s*
C_p = *specific heat, kJ/kg/K*
Δh = *enthalpy difference, kJ/kg*
Δt = *temperature difference, K*

SHR = 1.0 : sensible cooling

SHR < 1.0 : latent cooling

When latent cooling is involved and moisture condenses, the total duty becomes the addition of the sensible load plus the latent load, *see section Latent duty*

$$Q_T = Q_S + Q_L$$

The use of the term SHR is far less common these days than before the advent of computers. Often a cooling applications process behaviour was indicated by the SHR. This parameter could easily be read from a psychrometric chart, having plotted the inlet and outlet conditions .. db, wb or %RH .. and transposed the 'slope' of the process to the SHR scale which provided the SHR value e.g. 0.83

This meant that if the total cooling capacity was known, then the sensible duty was easily calculated by …

$Q_S = Q_T \times SHR$ *which, from above, also provides the latent duty too.*

OPERATION BELOW ZERO

During a latent cooling process, if the surface temperature of the fins is below 0°C, any condensate will freeze and accumulate on the fins as a frost layer. Initially, this thin frost layer results in a 'roughened' surface which promotes an increased airside heat transfer coefficient, slightly improving performance. However, over time, the frost layer turns into ice, which effectively insulates the fin surface and detrimentally affects the heat exchanger performance, hence the need for defrosting.

The generation of frost involves the additional latent heat of fusion (334 kJ/kg) involved in changing the liquid into a solid. Furthermore, if the condensate that has become the frost layer is cooled below 0°C, then there is an additional sensible energy component related to the specific heat of frost equating to 2.1 kJ/kg/K.

Depending upon the process in question, temperatures, fin material and fin efficiency, it is feasible that a proportion of the finned surface is above 0°C and the condensate generated in this vicinity will stay a liquid and drain away. Therefore, the duty requirement associated with the frost generation and subsequent sensible cooling of the frost layer, may be somewhat less than the 'worst case scenario' that all the condensate freezes and is sensibly cooled to the air leaving temperature.

SURFACE

The next governing equation combines the overall heat transfer coefficient with the coil surface area and operating temperature difference to match the required capacity as follows …

$$Q = U \times A \times LMTD \times F$$

where, Q = *Duty, kW*
U = *Overall heat transfer coefficient, W/m²/K*
A = *Total external surface area, m²*
$LMTD$ = *Logarithmic mean temperature difference, K*
F = *LMTD correction factor*

The combination of the air side external heat transfer coefficient with the fluid side internal heat transfer coefficient in addition to the internal and external fouling factors, fin efficiency, internal to external surfaces and the tube-fin bond contact resistance provides the overall heat transfer coefficient, U.

The total extended finned surface area is defined as that 'as seen' by the air stream and relates to the finned height, finned length, fin pitch, fin thickness, tube diameter, tube pattern and rows deep.

The LMTD is calculated from the inlet and outlet temperatures on both the air side and fluid side.

From the above, depending upon which coil related parameters are provided, allows the 'unknown' to be calculated, which is generally either the number of rows deep that a coil requires to meet the capacity requirement or the capacity that a given coil size will deliver at a given set of conditions.

PRIMARY SURFACE

The heat transfer surface of a heat exchanger core relates to the surfaces in direct contact with fluids through which heat is transferred by conduction.

In the case of EAS heat exchangers, the tubes are considered as the primary surface, also known as the direct surface.

When calculating the external surface area of a heat exchanger, it is the external surface area of the tubes that constitutes the primary surface area and denoted by $\mathbf{A_p}$

SECONDARY SURFACE

To increase the heat transfer area, appendages known as fins may be intimately connected to the primary surface to provide extended, secondary or indirect surface and denoted by $\mathbf{A_s}$

The addition of fins increases the surface area in contact with the secondary fluid stream and reduces the thermal resistance on that side of the process and thereby increases the net heat transfer from/to the surface for the same temperature difference.

UNIT SURFACE AREA

By definition, is the addition of the outside area of the primary surface i.e. the tubes and the secondary (extended) surface areas of the heat exchanger and denoted by $\mathbf{A_o}$

However, in our industry and with our type of construction involving the use of plate fins with extruded collars; *which creates the appropriate fin pitch and thus encapsulate the primary tube surface area*; is related to the surface area in contact with the air stream per square meter of face area per row deep, for a given fin pitch ~ *m²/m²/row*. This sub parameter is often referred to as the **A factor**

For geometries used for low temperature applications, electric heater elements may be inserted into the heat exchanger fin block and thus heater holes are provided for within the fin geometry.

Eurovent provides a formula to calculate the total surface area per unit area per row, supposedly taking into account all of these parameters …

```
Function CalcEuroventExternalArea(TubeOD, TubePitch, RowPitch, FinPitch, FinThk, NumHeaterHoles, HeaterHoleOD)
    ' Surface area per unit face area per row ~ m²/m²/row
    π = 3.14159
    Ratio = 1  ' To allow for wavy, corrugated etc. surfaces which have more surface area than the projected area
    NumHeaterHoles = 0    ' No heater holes
    HeaterHoleOD = 0      ' Diameter not defined
    SA = (2 / FinPitch / 1000) * (Ratio * TubePitch * RowPitch - π / 4 * ((TubeOD + 2 * FinThk)² +
            NumHeaterHoles * HeaterHoleOD²)) + π * (TubeOD + 2 * FinThk) / 1000

    CalcEuroventExternalArea = 1000 / TubePitch * SA

End Function
```

Note : EAS uses a modification of this formula to more accurately account for the fin thickness associated with the extruded collars. Furthermore, although EAS has a fin die that provides for Ø6 mm heater holes, these are never used and thus the following formula ignores the heater holes.

```
Function CalcExternalArea(Ratio, TubeOD, TubePitch, RowPitch, FinPitch, FinThk)
    ' Surface area per unit face area per row ~ m²/m²/row
    π = 3.14159
    CalcExternalArea = (2 * 1000 / FinPitch * (Ratio * TubePitch / 1000 * RowPitch / 1000 - π * (TubeOD
                        / 1000 / 2 + FinThk / 1000)²) + 2 * π * (TubeOD / 1000 / 2 + FinThk / 1000) * (1 –
                        FinThk / FinPitch)) / TubePitch

End Function
```

where,	*TubeOD*	*= Expanded tube outside diameter, mm*
	TubePitch	*= Tube spacing normal to air flow, mm*
	RowPitch	*= Tube spacing in direction of air flow, mm*
	FinPitch	*= Fin pitch, mm*
	FinThk	*= Fin thickness, mm*
	Ratio	*= Actual to projected fin area ratio*

Incidentally, some literature relates the surface areas to a 1 metre length of tube, which gives the same answer when the total tube length of the coil is used.

TOTAL SURFACE AREA

Denoted by $\mathbf{A_T}$ for a given coil with a given face area, number of rows and fin pitch …

Total Surface Area = Unit Surface Area x Face Area x Rows Deep

$A_T = A_o$ x FA x Rows

This total surface area is the value used in the heat transfer equations which combine the area with the overall heat transfer coefficient and operating log mean temperature difference in the following equation …

Q = U x A_T x LMTD

INTERNAL SURFACE AREA

Denoted by $\mathbf{A_i}$ and refers to the internal primary surface i.e. the tube internal surface area for a given coil of a given face area and number of rows ~ *m^2/m^2/row*

As mentioned above, some literature will relate this parameter to 1 metre of a single tube. But the result is the same when the complete coil is considered.

```
Function CalcInternalArea(TubeID, TubePitch)
    ' Surface area per unit face area per row ~ m²/m²/row
    π = 3.14159
    CalcInternalArea = π * TubeID / TubePitch
End Function
```

Therefore,

A_i = A x FA x Rows

where,	*TubeID*	*= Expanded tube inside diameter, mm*
	TubePitch	*= Tube spacing normal to air flow, mm*
	A	*= Calculated internal surface area, m^2*
	FA	*= Face area, m^2*
	Rows	*= Rows deep*

R FACTOR

Denoted by **R** is defined as the ratio between the external, A_o and internal surface, A_i areas per unit face area per row for a given fin pitch. Thus for a given coil face size and a single row coil, this ratio increases as the fin pitch reduces.

$R = A_o / A_i$

This ratio has become redundant these days with the availability of computers, but when calculations were performed by hand and both slide rules and log tables were commonly used, tabulated values for the R factor against the fin pitch for different geometries, eased the calculation procedure.

So typically, the simplified equation for the overall heat transfer coefficient, U is …

$$1 / U = R / h_i + 1 / h_o$$

… which was quicker to use compared with …

$$1 / U = A_o / A_i / h_i + 1 / h_o$$

The above assumes that the effects of the tube wall thickness and tube thermal conductivity are negligible and ignores both the internal and external fouling factors.

AIR-TO-AIR ENERGY RECOVERY DEVICES

- **Plate Heat Exchangers (Crossflow or Counterflow)**
  - No moving parts, air streams separated
  - Sensible (temperature only) or enthalpy (temperature + moisture) exchange

- **Heat Pipes**
  - Refrigerant-based passive devices
  - Transfer heat from exhaust to supply air via phase change

- **Rotary Heat Wheels (Thermal Wheels/Enthalpy Wheels)**
  - Slowly rotating media transfers sensible and latent heat
  - High effectiveness, but some cross-contamination risk

RUN-AROUND COIL SYSTEMS

- Water/glycol loop with coils in both exhaust and supply air streams
- Suitable when ducts are separated or cross-contamination is unacceptable
- Lower efficiency than wheels but highly flexible

REFRIGERANT-BASED RECLAIM SYSTEMS

- Recover heat from refrigeration/compressor systems
- Typical in supermarkets, chilled-water plants, and VRF systems
- Heat used for domestic hot water, hydronic heating, or reheat in HVAC.

HEAT RECOVERY CHILLERS / HEAT PUMP SYSTEMS

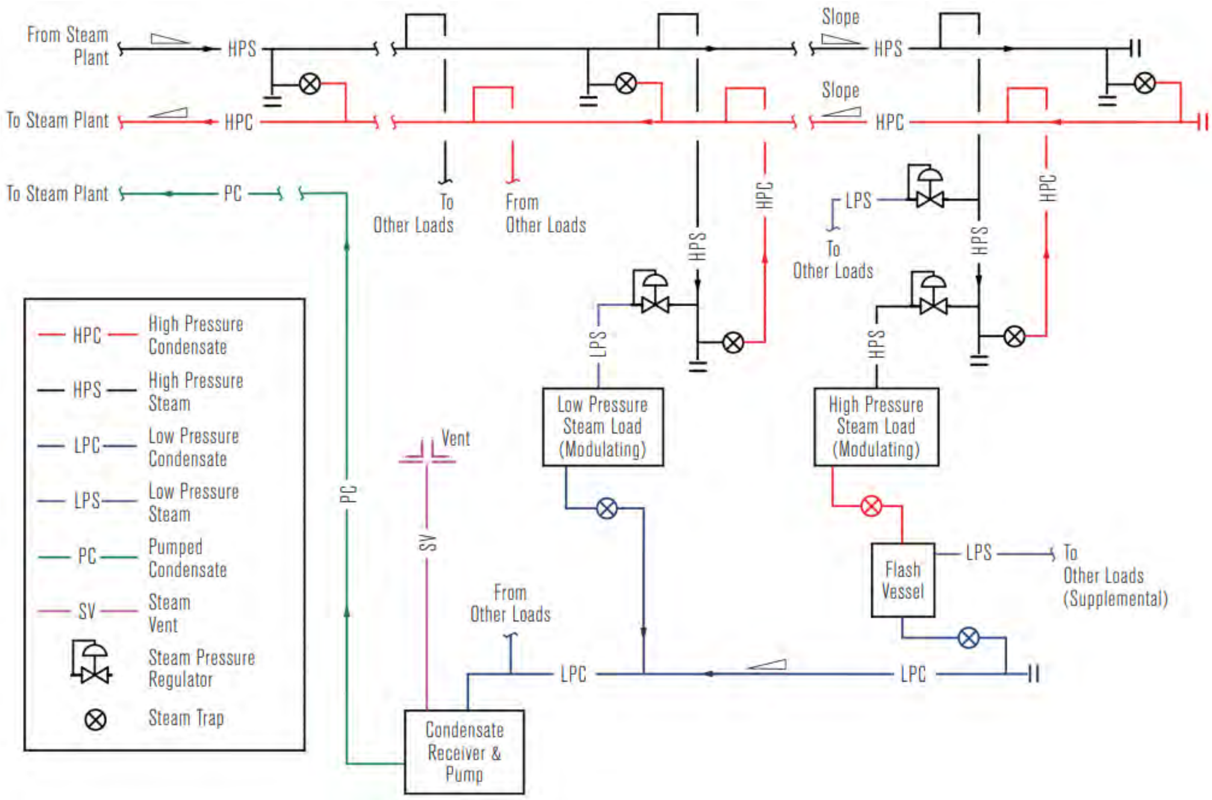
- Reversible chillers or dedicated heat recovery chillers
- Provide simultaneous cooling and heating (e.g., chilled water + hot water)
- High efficiency in buildings with overlapping heating and cooling demands

GENERAL

Extracts from an ASHRAE Journal : June 2023

To ensure the safety and reliability of steam systems it is imperative that the proper routing and sizing of this piping is conducted to avoid catastrophic failure, causing harm to people and damaging property. In less extreme scenarios, piping may be noisy, fail prematurely or not function as intended.

SIMPLIFIED STEAM DISTRIBUTION SYSTEM



The simplified steam distribution system shown above comprises steam supply, liquid condensate recovery, two-phase condensate recovery and vent piping. Supply piping delivers steam from a boiler plant or other source to various high-pressure (100 to 1000 kPa) use points and pressure reducing stations. Condensate recovery piping returns steam condensate to the plant in liquid or two-phase form. Steam vents ensure atmospheric condensate recovery vessels do not become pressurized, while aiding in the prompt removal of condensate from equipment and drip legs.

FUNDAMENTAL FLUID PRESSURE DROP EQUATIONS

Dynamic pressure losses for fluids flowing in closed circular conduits are characterized by the Darcy Weisbach equation, which is valid for incompressible, fully developed, single-phase and Newtonian fluids. Friction factor (ƒ) and Reynolds number (Re) are dimensionless variables in the Darcy-Weisbach equation that require intermediate calculations. The implicit Colebrook equation is used to precisely determine friction factor; close approximations can also be achieved using explicit methods e.g. Swamee-Jain. Reynolds number is the ratio of inertial to viscous forces; its value indicates the fluid's flow regime (laminar, transitional or turbulent).

EQUATION	CONSTANTS AND VARIABLES
Darcy-Weisbach $\Delta P = \dfrac{C_1 f L V^2}{d\upsilon}$ (1)	ΔP = pressure drop, psi or Pa C_1 = 3.597×10^{-7} (I-P); 500 (SI), dimensionless f = friction factor, dimensionless L = pipe length, ft or m V = steam velocity, fpm or m/s d = pipe inside diameter, in. or mm υ = specific volume, ft^3/lb$_m$ or m^3/kg
Swamee-Jain $f = \dfrac{0.25}{\left[\log_{10}\left(\dfrac{\epsilon/d}{3.7} + \dfrac{5.74}{Re^{0.9}}\right)\right]^2}$ (2)	ϵ = absolute roughness; 0.00216 in. (0.0549 mm) for carbon steel, 0.000072 in. (0.00183 mm) for copper and stainless steel* Re = Reynolds number, dimensionless
Reynolds Number $Re = \dfrac{dV}{C_2 \nu}$ (3)	C_2 = 720 (I-P); 1,000 (SI), dimensionless ν = kinematic viscosity, ft^2/sec or m^2/sec

Industrial steam plants typically generate and distribute saturated steam at high pressure. Elevated steam pressures are often preferred for large piping networks since the same mass flow can be transported with smaller pipe sizes due to higher steam density at increased pressures. Higher steam pressures also result in increased fluid temperatures, which is often required for process and sterilization applications. Steam pressure reducing stations are employed at various use points to serve equipment with maximum allowable working pressures (MAWPs) below the highest anticipated upstream supply pressure. Low-pressure (less than 100 kPa) steam offers the benefit of higher latent energy content for loads not requiring elevated temperatures.

Steam supply pipe sizing is a function of mass flow rate, velocity, initial pressure and total allowable system pressure drop. Mass flow rate is determined by equipment requirements plus incidental losses (e.g, condensed steam in piping, release through vents). Velocity limitations are established to prevent pipe erosion and water carryover without creating excessive noise. Initial pressure is established by the steam pressure required at use points and total distribution system pressure losses. Total allowable steam supply piping pressure losses are most commonly in the range of 20% to 30% of generation pressure at the source.

Steam supply piping is sloped a minimum of 2 mm/m in the direction of flow to promote drainage of condensate to collection points. Short branch runouts of 3 m or less may slope opposite the direction of flow (back toward the main) at a minimum slope of 4 mm/m.

General guidelines for steam pipe sizing are provided in the Table; pipes should be sized so that the maximum velocity and pressure drop values indicated are not exceeded. For a given steam supply pressure condition, pressure drop limitations will determine pipe size at lower flow rates, whereas velocity limitations will determine pipe size at higher flow rates. A suggested procedure for sizing steam supply piping by formula is as follows:

OPERATING PRESSURE PSIG	MAX. VELOCITY, FPM	MAX. PRESSURE DROP PER 100 FT[1]	TOTAL SYSTEM PRESSURE DROP[1]
0	4,000*† 6,000‡	0.5 oz/in^2	1.0 oz/in^2
5		0.25 psi	1.5 psi
10		0.5 psi	3 psi
15		1 psi	4 psi
30	4,500* 6,000† 8,000‡ 10,000§	2 psi	5-10 psi
50		2-5 psi	10-15 psi
100		2-5 psi	15-25 psi
150		2-10 psi	25-30 psi

Pressure drop per unit length values may be adjusted to maintain total allowable system pressure drop.
* Steam headers
† Runouts to equipment and downstream of pressure regulators
‡ Mains and branches (heating or noise sensitive)
§ Mains and branches (process and not noise sensitive)

1. Determine required pipe diameter using …

$$d = C_3 * \sqrt{\frac{\dot{m}v}{\pi V}}$$

where,

d = pipe inside diameter, mm
C_3 = 33.333
$\dot{m}$ = mass flow rate, kg/h
υ = specific volume, m^3/kg
V = steam velocity, m/s

2. Select the closest standard pipe size with at least the internal pipe diameter calculated in step 1.

3. Determine the resulting steam velocity using the selected inside pipe diameter from step 2 and the following equation …

$$V = \frac{\dot{m}v}{\pi\left(\frac{d}{C_3}\right)^2}$$

4. Calculate pressure drop per unit length using the Darcy-Weisbach equation (Equation 1 in Table above). Select a larger pipe size and repeat steps 3 and 4 if pressure drop per unit length exceeds the design value.

Total system pressure drop must be determined after the steam piping network has been sized, accounting for valve, fitting, and piping frictional losses. If total pressure losses exceed the maximum allowable value, pipe sizing and routing should be optimized to reduce pressure losses to within an acceptable range.

STEAM CONDENSATE RECOVERY AND VENT PIPE SIZING

Steam condensate recovery systems return hot, previously deaerated and chemically treated condensate to the steam plant. If condensate is not returned, it must be replaced with feedwater requiring additional deaeration, chemical treatment and thermal energy to produce steam. Operating costs associated with these treatments are substantial and should be minimized whenever possible. Hence, every reasonable effort should be made to recover condensate when it is not contaminated or consumed in the process.

Industrial steam condensate recovery systems have the

following pipe sizing cases:

- Piping from equipment drains to steam trap inlets
- Steam trap outlet piping
- Dry-closed condensate return piping
- Pumped condensate return piping
- Steam vent piping

Liquid-filled condensate returns are usually separated from those containing two-phase fluid to prevent excessive noise and piping damage due to water hammer.

Dry-closed returns are employed when adequate back pressure is continuously available to convey condensate from the steam load to the plant. Pumped condensate systems rely on electric or pressure motive pumps to overcome system pressure losses when adequate back pressure is not continuously available. Steam vents in condensate recovery systems release incidental flash steam discharged to atmospheric vessels, ensuring the vessels do not become pressurized. General guidelines for steam condensate recovery and vent pipe sizing are summarized in Table below …

PIPE SERVICE	SIZING CASE	MAX. VELOCITY	MAX. PRESSURE DROP
Steam trap discharge & dry-closed returns (falling)	Flash steam	4,000 fpm	0.125 to 2 psig/100 ft*
Steam trap discharge & dry-closed returns (rising)	Flash steam	3,000 fpm	0.125 psi/100 ft
Atmospheric vent	Flash steam	3,000 fpm	0.125 psi/100 ft
Pumped condensate	Liquid	7.0 fps	3.0 ft wg/100 ft
Equipment drains	Liquid	3.0 fps	1.0 ft wg/100 ft

Oxygen (O_2) and carbon dioxide (CO_2) are constituents of makeup water that cause corrosion in steam systems, and these gases can travel along with steam through the distribution system. When steam condenses, CO_2 dissolves in hot condensate, forming corrosive carbonic acid (H_2CO_3). Although proper treatment abates corrosion, it can be difficult to eliminate completely. Therefore, Schedule 80 carbon steel is commonly used for steam condensate return piping to allow for a reasonable degree of corrosion over time and account for reduced pipe material at threaded components.

Stainless steel piping may also be used to provide improved corrosion resistance. Internal pipe dimensions corresponding to the selected pipe wall thickness should be used in pipe sizing calculations.

PIPING FROM EQUIPMENT DRAINS TO STEAM TRAP INLETS

Piping from equipment drains to steam trap inlets should drain by gravity, be of minimal length, and not impose significant pressure drop or contain flash steam. In most cases, the proper drain line size is the larger of equipment connection size and steam trap inlet size. The latter will often govern due to safety factors and warm-up loads considered when sizing steam traps. Equipment drain piping should slope a minimum of 8.3 mm/m in the direction of flow to ensure efficient removal of condensate from equipment.

Excessive drain line lengths from equipment should be avoided whenever possible. Should the length exceed 9 m, the pipe size determined from the criteria above should be verified not to exceed a velocity of 0.9 m/s and a pressure drop of 100 Pa/m. Equipment drain piping should be increased in size as required to comply with these parameters.

STEAM TRAP OUTLET PIPING

Steam trap outlet piping contains two-phase fluid flow: flash steam produced by pressure drop across the trap and liquid condensate. Percentage of flash steam is calculated according to the following equation …

$$\% \text{ Flash steam} = \frac{100\left(h_{f1} - h_{f2}\right)}{h_{fg2}}$$

where,

h_{f1} = enthalpy of liquid at higher pressure, kJ/kg
h_{f2} = enthalpy of liquid at lower pressure, kJ/kg
h_{fg2} = latent heat of vaporization at lower pressure, kJ/kg

Although flash steam represents only a small portion of mass flow, it accounts for 96.2% to 99.7% of internal pipe volume for two-phase piping systems with upstream operating pressures between 35 and 1000 kPa. Consequently, flash steam flow rate is commonly used as the basis of sizing for steam trap outlet piping. This empirically proven method does not quantify the nominal volume of liquid occupying the pipe, or differences in velocity between steam and condensate; sufficiently low velocity and pressure drop limits ensure that this simplified approach yields acceptable results. A maximum flash steam velocity of 20 m/s is suggested for falling lines. Steam trap outlet piping requiring condensate to be lifted to a higher elevation is sized for a flash steam velocity not exceeding 3,000 fpm (15 m/s) and a pressure drop not exceeding 28 Pa/m to prevent water hammer. Steam trap outlet piping is sloped at a minimum of 10.4 mm/m in the direction of flow. This piping should never be smaller than the steam trap outlet connection size.

DRY-CLOSED CONDENSATE RETURN PIPING

Condensate returns conveying two-phase fluid from multiple steam trap outlets are used when adequate back pressure is continuously available. Flash steam at steam trap outlets serves as the motive force to convey condensate through the piping network. Alternatively, this condensate may discharge to a flash steam recovery vessel, where thermal energy is captured and used to supplement steam service to low-pressure loads.

Common dry-closed condensate return piping is non-vented and has a continuous pressure difference between the point where condensate enters the line and the point where it exits. These piping networks are commonly referred to as high, medium, and low-pressure condensate lines, in reference to the steam operating pressure upstream of the trap. However, these naming conventions are not indicative of actual condensate line operating pressure. Properly sized condensate piping will be subjected to normal operating pressures much lower than that of the associated steam supply.

Dry-closed return pipe sizing follows the same general principle as steam trap outlet pipe sizing, in that flash steam flow rate is typically used as the design case. Flash steam velocity in falling lines should not exceed 4,000 fpm (20 m/s), and piping should slope at a minimum of 4 mm/m in the direction of flow. Common return lines requiring condensate to be lifted to a higher elevation are sized for the combined flash steam flow rate, with a maximum velocity of 15 m/s and a maximum pressure drop of 28 Pa/m.

The operating pressure of dry-closed returns is highest at steam trap outlets, where it is approximately equal to local supply pressure less the steam trap pressure drop. Total equivalent length of return piping and the amount of back pressure at the point of exit must be considered when determining the design total system pressure drop. Typical pressure drops per unit length range from 30 to 450 Pa/m, with higher values corresponding to higher steam supply pressures.

Occasionally, dedicated common return piping is provided for each steam supply pressure level in a system. However, segregation of these piping systems is not always required. Steam traps discharging from different upstream pressures may be combined if …

- each steam trap is selected to provide a similar outlet pressure
- steam supply pressure is sufficiently greater than condensate return main pressure
- the common return is sized to convey the combined rate of flash steam from connected steam traps

PUMPED CONDENSATE RETURN PIPING

Pumps are used to transport liquid condensate back to the plant when adequate back pressure is not continuously available. Pumped condensate return piping is filled with liquid (flooded). Fluid velocities ranging from 0.9 to 2.1 m/s and pressure drops between 100 and 294 Pa/m are used for sizing. Condensate pump flow rates can be two to five times greater than the incoming condensate load; therefore, it is essential to size pumped condensate piping using rated pump flow, not incoming condensate load.

Common return mains receiving only liquid condensate may be vented and drained by gravity after being pumped to a high point. In such cases, piping is sized for open channel flow in accordance with ASHRAE Handbook—Fundamentals. However, additional lift and back pressure are frequently required in large piping networks, often precluding the use of this method in industrial steam systems.

STEAM VENT PIPING

Atmospheric steam vents are provided for condensate receivers, flash tanks, surge tanks and boiler feed tanks to prevent an accumulation of pressure by relieving incidental flash steam. Properly sized vents will limit flash steam velocity and pressure drop to 15 m/s and 30 Pa/m, respectively. Vent piping should be no smaller than the associated vessel's vent connection size. Horizontal vent sections are sloped opposite the direction of flow (back toward the vessel) at a minimum of 4 mm/m. Multiple steam vents may be combined into a common vent, provided the common vent internal area is at least equal to the sum of internal areas of the connecting vents. Atmospheric steam vent sizing guidelines do not apply to safety relief valve vent piping, which is subject to requirements outside the scope of this article.

COMMON STEAM SYSTEM PIPING PITFALLS

Proper steam system pipe sizing is effective only when implemented along with good routing, design, and installation practices. Awareness and prevention of the following common steam system piping pitfalls will help ensure that steam systems operate as intended:

- Attempting to lift condensate from modulating steam loads. During periods of minimal load, available pressure at the steam trap outlet is inadequate to lift condensate due to control valve throttling. These loads must be served by a pump trap or drain by gravity to a condensate receiver.
- Two-phase condensate discharging to a flooded return line. This practice should be avoided whenever possible.6 If it is necessary, steam traps should have a continuous discharge, and a trap diffuser or sparge tube should be provided in the steam trap outlet piping to prevent water hammer.
- Concentric reducing fittings installed in sloped steam and condensate piping. Eccentric (flat on bottom) reducing fittings should be provided for sloped steam and condensate piping to ensure uninhibited flow of condensate and to prevent water hammer.
- Steam supply strainers installed with wye in downward position. Install strainers with the wye in the horizontal position to prevent collection of condensate or provide a steam trap at the strainer blowdown connection.
- Air vents and vacuum breakers not provided. These air control devices must be placed at strategic locations within the piping network to ensure proper system operation. Air and other non-condensable gases must be released from piping to allow steam and condensate to move through the system. Sub-atmospheric pressures caused by a dramatic reduction in fluid volume during a phase change from steam to condensate will inhibit the flow of condensate.
- Operational diversity and future capacity need not considered in pipe sizing. Diversity factors may be appropriate if all steam-using equipment is not in operation concurrently (e.g., production and cleaning). Production schedules and equipment operating load profiles will assist in determining peak coincident steam and condensate loads for common piping. Provisions for additional capacity should also be considered if the process is expected to expand in the future.

STEAM COILS

BACKGROUND

Steam is an efficient process for providing heating and has been in use for many years but has always been hounded by heat exchanger longevity issues as the industry has strived to provide lighter weight solutions as the market has become more competitive.

As dry saturated steam condenses its volume collapses dramatically and water is formed. Thus steam is induced (not forced) into a coil by the suction caused when it collapses into condensed water.

The primary issue with steam coils is one of regulation to balance varying heat load demands. It is important to realise that when steam condenses it releases latent heat fairly consistently regardless of pressure.

When the steam flow is controlled to reduce heat output its condensing temperature falls rapidly, to balance the reduced pressure, which may be below the system condensate return pressure. Therefore, condensed water ceases to be drained from the coil thus flooding the lower portion of the coil, resulting in sub-cooling of the condensate.

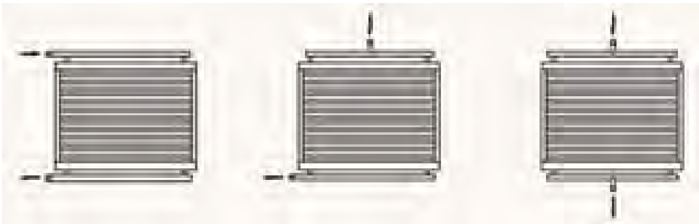
As the automatic regulating valve adjusts, the condensate level may rise and fall, which may induce both *water hammer* and *thermal shock*.

STEAM COIL ALTERNATIVES

VERTICAL TUBE STEAM COIL

Traditionally and by far the most stable of the steam coil alternatives, is the vertical tube arrangement. This variant ensures complete condensate drainage; providing an adequate steam trap is employed; and is EAS's preferred solution.

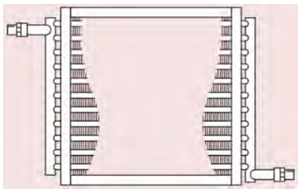
The major concern with this arrangement is the restriction upon the available coil height (*actually the finned length in this orientation*), due to the space required to accommodate the steam inlet and condensate outlet headers. However, this can be mitigated by feeding the steam into the end of the inlet manifold and also collecting the condensate horizontally as shown below ..



HORIZONTAL TUBE STEAM COIL

Horizontal tube steam coils can be problematic if condensate retention is an issue. The associated water hammer consequences can quickly cause the coil to fail.

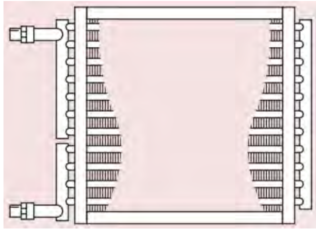
The figure to the right shows a horizontal tube coil concept, however, this is not a configuration that EAS deems desirable because of the potential problems associated with the entrainment of condensate 'slugs' by the high velocity steam flow, which during frequent 'on/off' operation or cold start-up conditions can adversely affect the coil.



To eliminate this problem it is preferred to incline the tubes, typically 4%, toward the condensate outlet manifold. The picture below shows horizontal air flow inclined tube coil incorporated within the scope of a rectangular duct mounted casework construction.

If the air flow is vertically up or down, then the following sketch indicates the necessary tube slope to ensure full condensate drainage.

The above only provides for opposite end connections and when this is undesirable an alternative horizontal tube coil arrangement can be offered, but again is not EAS's preferred alternative.



Although the arrangement to the right could accommodate hairpin or return bends, EAS prefers to offer a 'transfer header' in place of the bends to ensure durability and longevity.

This arrangement must be installed horizontally and may also be susceptible to water hammer or water 'slug' related issues if the condensate is not adequately removed.

CHAMBER STYLE STEAM COIL

A non-preferred EAS coil variant of the steam coil construction.

Otherwise known as a *tube plate & header box* construction, where each tube is welded into the 6 mm thick tube plate and then a header box/chamber is fully welded into position.

This less common alternative has the following limitations ..

- Maximum pressure **6 barg** (*7 barA*) – 165°C
- Only all **SS304** or **316** construction
- Maximum **6** rows deep
- Variants …
    - 4% inclined tubes – opposite end connections
    - Horizontal tubes – 2 pass arrangement with same end connections
    - Vertical tubes – opposite end connections

Note

These types of coils can only be considered as 'vessels', and as EAS is not currently able to provide the supporting strength calculations for a category version of such a design, they **must** fall into Article 4/3 and thus **PS x V_T <= 50 bar.L**

Consequently, if PS = 6 barg, the maximum volume, **V_T** of this type of coil **must** be **<= 8.33 Litres**

Outside of this 50 bar.L constraint would result in a Category I / Module D or higher coil … which EAS is not currently approved to manufacture.

STEAM COIL DESIGN GUIDELINES

The purpose of this section is to define the design and manufacturing constraints that Evapco Europe A/S imposes when considering the provision of steam heating coils to ensure coil longevity.

For low pressure steam applications <= 3.0 barg (143°C), thick wall copper tubes can be used, however beyond this temperature/pressure, EAS reverts to stainless steel tubes and headers. However, the 0.6 mm wall thickness is rather thin and because most steam applications use an On/Off mode of control, the heat exchangers are subject to cyclic thermal shocks/loads and can be subject to leaks where cracks appear in the HAZ (heat affected zone) in the vicinity of tube to header welded joints.

A more durable wall thickness would be 0.9 or 1.0 mm wall thickness, however at the time of writing, this thick wall is not a standard material option.

The following describes the fundamental limitations for all applications involving steam as the primary heat transfer fluid.

- Primarily, steam coils shall have vertical tubes with horizontal inlet and outlet headers
- Steam inlet quality shall only be '**dry saturated**' steam. Superheated steam applications shall **NOT** be considered

- Condensate outlet – **saturated liquid** – no sub-cooling shall be considered.
  If sub-cooling is required, then this part of the process should be handled either by a separate sub-cooler or alternatively a horizontal tube single phase cooling coil, which could be mounted on the air entry side of the vertical tube steam coil
- Steam coils are assumed to be fitted with adequate steam traps (*supplied by others*) and provided with the correct piping installation to avoid condensate retention and potential water hammer issues resulting in – worst case – coil failure
- Only Ø15 mm tube geometries shall be offered as steam coils : A & C-Fin
- Maximum tube length – **3000 mm**
- Maximum header length per section – **1500 mm** (*25 tubes per row per section for A-fin & 38 tubes per row for C-fin*). *Invoked to minimise thermal stresses on adjacent header legs*
  Proceed with caution if 1500 mm exceeded
- Maximum rows per header …
  - Copper tubes – **2 rows**
  - SS304/316 tubes – **2 rows**
- Maximum pressure limitations
  - **Copper tubes** (0.75 mm) – **3 barg** (*4 barA*) – 143.6°C fitted with …
    - Standard Copper headers
    - K65 (CuFe2P) headers. *Use 38% Ag (silver) content brazing alloy*
  - **SS304/316 tubes** (0.6 mm) – **7 barg** (*8 barA*) – 170.4°C fitted with …
    - SS304/316 headers
    - *Special cases : Proceed with caution for steam pressures above 7 barg. See discussion below*
- Design guidelines when alternative steam pressures given …
  - <u>Always</u> base the 'surface requirements' upon the lowest pressure specified
- Aluminium fin material thickness suggestions …
  - Up to 1 barg (*2 barA – 120°C*) use 0.12 mm Aluminium fins
  - From 1 barg (*2 barA*) to 3.5 barg (*4.5 barA – 150°C*) use 0.18 mm Aluminium fins
  - Above >3.5 barg (*>4.5 barA - >150°C*) use 0.25 mm AlMg fins
- When air leaving temperature and steam condensing temperature difference is <10K conduct calculation with a surface reserve of 10-15% (*SF = 1.1 to 1.15*)
- Connections sizes …
  - Steam inlet connection size based upon 25 – 40 m/sec
  - Minimum 1" (DN25)
  - Governing connection size based upon the **steam inlet** conditions
  - Condensate outlet connections size traditionally **one size smaller**, unless inlet is 1" (DN25) and then condensate size will also be 1" (DN25)
- Minimum distance between the tube plate and inside of the header i.e. header stub length is 90 mm for <u>both</u> **copper** and **stainless** steam coils

DISCUSSION – STEAM PRODUCTS

Although the above provides guidelines, the types of steam coil/product enquiry we can often receive 'fall into the cracks', and it can be confusing whether we should accept the order or decline to quote.

Historically, EAS have manufactured process heating steam coils for pressures up to 12 barg, but also problems have been experienced with the occasional leak … after the fact, identified as stress related cracking.

Clearly, such failures are both costly and affect credibility and so in around 2015, it was decided to limit the design envelope for steam coil applications. Exceeding 7 barg was considered as an exception and any jobs warranting such pressures should be carefully considered. Similarly, if the tube length exceeds 3000 mm, caution is advised.

In the past, less concern was expressed when applications called for air cooled condenser related products using dry saturated steam. Such products could potentially be up to 12 metres in length and would need to either have their coil block mounted on an incline or the whole product might be mounted on a slope by using longer legs at one end of the product.

Obviously, such long units are subject to thermal expansion & contraction effects, especially if the process is cyclic and problems can be compounded if ambient temperatures are low too.

Such products would only be offered as a single pass, opposite end connection product with the inlet headers canted upward and the condensate outlet canted downwards to ensure that no residual liquid remains in the tubes of the coil.

Residual liquid slugs lying in the tubes, whilst the product is in stand-by mode or off cycle, can be propelled at high speed down the tube when the steam cycle become active again. Besides damaging bent header legs, extreme water retention can cause water hammer effects, which again can damage and over-stress the coil, leading to stress fractures and leaks.

To eliminate residual water, placing the coil on a purposeful slope will ensure that the condensate will drain to the canted downward condensate collection header and then leave the condenser via the steam trap.

However, mounting the coil; inside the product; on a slope is an engineering challenge and it is far simpler to place the whole product in an incline. But such a technique is not always accepted by the Customer or may create issues on site.

INSTALLATION

Steam coils **must** be installed in a manner that promotes condensate drainage. This will aid in preventing destructive water hammer, freezing and the build-up of corrosion within the coil.

Coils **must** not to be used with the tubes inclined upwards.

Where coils are incorporated into ducting, it is important that they are properly installed.

PIPING

- Coils should be piped according to any relevant local codes of practice
- Where threaded connections are supplied, the only approved method of jointing method is by use of ‘Boss white and hemp’. The thread fitted to the coil is to be supported at all times whilst making joints. All external piping is to be supported independently from the coil
- The use of flexible connections is recommended, where applicable
- Trap each coil section separately. Locate the trap a minimum of 350 mm below the condensate connection of the coil if the method of control is On/Off (non- modulating)
- Locate the trap a minimum of 450 mm below the condensate connection of the coil if used with a modulating control valve
- Use only continuous draining styles of traps such as inverted bucket or float and thermostatic type. Only use a float and thermostatic trap on coils with a steam supply from a modulating valve
- Do not allow the steam pressure to fall below 0.35 barg
- Never oversize control valves; *bigger is not better*
- Fit a vacuum breaker to the steam supply line. Also install a vacuum breaker on the downstream side of the coil when steam pressure is to be modulated
- When the condensate must be lifted into overhead or pressurized return mains, a vented condensate receiver must be installed with a correctly sized steam pressure pump to return the condensate.

EXAMPLES

SCENARIO 1

A steam coil designed to operate at full load at 4 barg with a mean air temperature of 20°C has an operating TD of (152°C - 20°C) = 132 K

To operate the coil at half load the TD must be halved to 66 K.

Thus, the condensing temperature will be (20 + 66) = 86°C equating to a pressure of ≈ -0.4 barg (0.6 barA), which is below atmospheric pressure. Consequently, the coil will flood to approximately 40% of its volume to reduce the active surface area to accomplish the desired load.

To avoid such problems, particularly where varying loads are to be controlled accurately, it is suggested that a hot water circulation system be adopted, generated by a local calorifier and pump set. This is economic, particularly when a number of coils are to be installed in one area.

If this is not convenient or practicable then it is recommended to use *Face and Bypass dampers* to modulate the air volume passing through the coil. This control methodology ensures that the entire surface of the coil is working, operating with a positive pressure and allowing for condensate removal at all times.

Additionally, no steam control valve will be necessary since only that steam quantity required to perform the demand duty will be used. The face damper servo-motor controls the output.

SCENARIO 2

Calculate the steam pressure drop in a large diameter pipe (DN250) 153 meters long supplied with 15 ton/hr dry saturated steam supplied at 3.5 barg equating to a condensing temperature of 147.0°C.

Solution :

Steam density, **ρ** at this condition is 2.4 kg/m^3 and the viscosity, **μ** is 0.0139 mPa.s (0.0000139 Pa.s)

The steam mass flow rate of 15 ton/hr equates to 15000 kg/h and thus 4.16 kg/s

The volumetric flow rate is thus 4.16 / 2.4 = 1.733 m^3/s

Internal diameter of a 10" (DN250) pipe = 254.5 mm or 0.2545 metres.

Hence, the velocity is the pipe is the Volume flow / flow Area ..

$V = 1.733 / (\pi \times 0.2545^2 / 4) = 34.07$ m/s *so this looks good and is in the range of 25 to 40 m/s as recommended.*

The Reynolds Number, $Re = \rho\ V\ D / \mu$

$Re = 2.4 \times 34.07 \times 0.2545 / 0.0000139 = 1496985$

Next, the friction factor, $f = 0.2 / Re^{0.2}$

Therefore, f = 0.01164

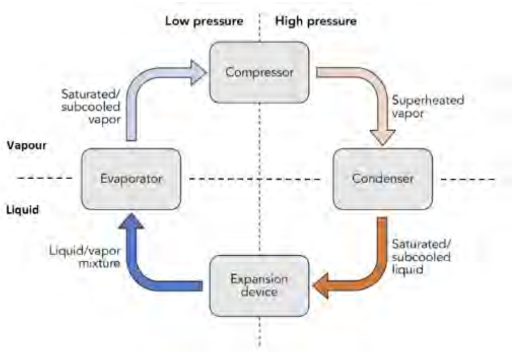
Now we can apply these parameters to the general equation …

$\Delta P = f\ L\ \rho\ V^2 / (2\ D)$

$\Delta P = 0.01164 \times 153 \times 2.4 \times 34.07^2 / (2 \times 0.2545) = 9747$ Pa (0.0975 bar)

This pressure drop is acceptably low and will thus not adversely affect the condensing temperature.

CARNOT CYCLE

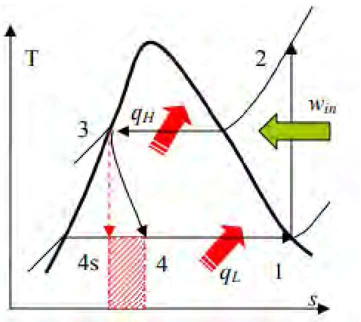


The origins of today's refrigeration industry stem from the theoretical Carnot cycle proposed by French physicist *Sadi Carnot* in 1824, which he referred to as a 'heat engine' and was thermodynamically a reversible process.

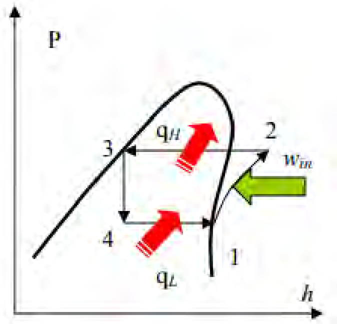
Reversing the Carnot cycle was the key to the modern day refrigeration and heat pump systems.

As a theoretical model it involved two isentropic processes and two isothermal (reversible adiabatic) processes as shown on the T-S diagram, which clearly is not realistic.

However, to make the concept function in the 'real world', with the available compressors and expansion devices, the T-S diagram transforms into the following …



… and when transposed onto the more familiar p-h diagram utilised by our industry sector, becomes …



BASIC SYSTEM

There is often confusion or mystery associated with refrigeration systems and heat pumps. They are often thought of as different systems, but are in fact, identical. Their definitions depend upon whether the focus is upon cooling (refrigeration system) or heating (heat pump).

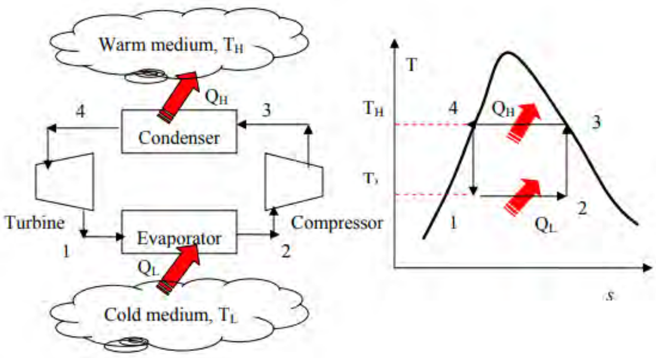
These systems comprise 4 major components …

- Compressor
- Condenser
- Expansion device
- Evaporator

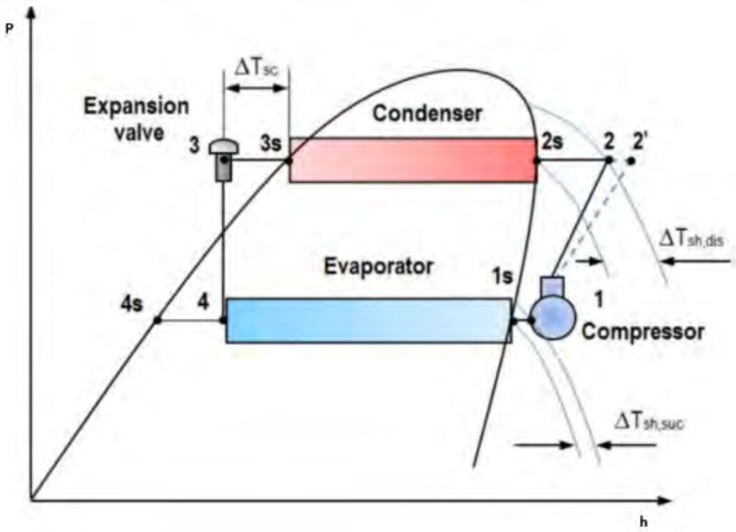
and the schematic simplifies the processes involved.

Essentially the cycle can be broken down into 4 quadrants. Starting from the compressor discharge and progressing clockwise …

- High pressure, high temperature gas
- High pressure, warm liquid
- Low pressure, cold liquid
- Low pressure, cold gas



If we overlay the 4 major components onto the familiar p-h chart, we get …

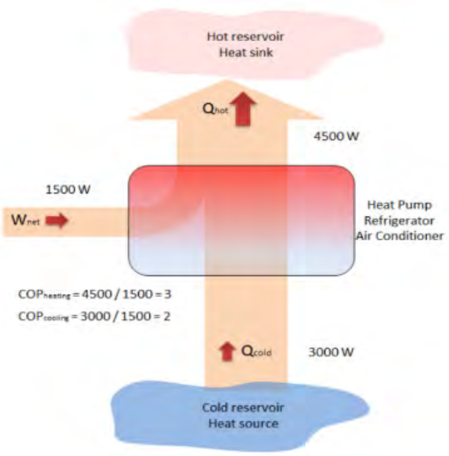


If the useful energy utilised is from the evaporator, then we call this a refrigeration system, where this energy is referred to as the **Refrigeration Effect** (**RE**).

If the useful energy utilised is from the condenser, then we call this a heat pump, where the potential useful energy, known as the **Total Heat of Rejection** (**THR**) is defined by process 2' to 3.

HEAT PUMP

When considering a heat pump, COP (coefficient of performance) is a big consideration and this simplistically relates to the ratio of the energy expended (what you pay for) compared with what you get out of the system, essentially the THR … *if all the energy from the high temperature, high pressure process is utilised*.



So in the example, we pay for 1500W of electrical energy to drive the compressor and get 4500W of useful heat out of the condenser, resulting in a COP = 3.0

COPs up to 5 or 6 can be achieved if the cold reservoir temperature is stable and the delivery conditions fixed.

Ground source compared with air source heat pumps tend to have higher COPs especially in winter conditions where heat is needed and in the case of air source variant, the heat source (outside air) drops in temperature resulting in lower energy availability, lower evaporating temperatures and consequential reduction in compressor performance and hence COP.

WEIRD SCIENCE

The following is a precis of an article published in the Feb. 2024 edition of ASHRAE Journal and written by Andy Pearson. It helps to clarify how fluids behave inside a closed refrigeration system.

We tend to develop a misunderstanding of how fluids behave inside a closed system from observing the common occurrence of water boiling in a kettle. However, we ignore the fact that the gas space above the boiling liquid is filled with air, whilst in a refrigeration system the space is filled with the same substance as the liquid, but just in a different state. Consequently, the pressure in the refrigerator rises and falls with temperature … *following a fixed relationship* … so when both the liquid and gas are present in the same space, the liquid is always at its boiling point.

Any reduction in the pressure will cause more of the liquid to turn into gas; and any increase in pressure will turn the gas into liquid. This behaviour causes several strange things to happen in refrigeration systems …

- The weight of the liquid creates a slight pressure gradient with depth, so the liquid at the surface of the pool is ready to boil but deeper in the pool it is slightly subcooled and so boiling is easiest at the top. If a large tank has a float valve controller to add more liquid when the level drops then the first thing that happens when the float valve opens is that the level drops even further because the small rise in pressure will condense some of the gas bubbles that are suspended in the liquid. This can cause control instability through a positive feedback mechanism, where opening the valve to add liquid reduces the volume of the liquid that is already there. Conversely, when liquid is being drawn out of the bottom of a tank, any pressure drop; for example at the inlet of a pump; can cause the liquid to boil, creating bubbles that can go great harm when they subsequently collapse again in a phenomenon known as ‘cavitation’.

- The second difference is that boiling water is not a good model of our intuition because the bonds between molecules in liquid water are so high requiring a lot of energy to turn the liquid into a gas. The high bond strength is because the H_2O molecule is kinked … like a broken stick … where the middle od the molecule (the oxygen atom) has a slight -ve charge and the two end hydrogen atoms are slightly +ve charged.

  In refrigeration jargon we say that the latent heat of vapourisation of water is high (~2260kJ/kg). Ammonia comes second for a similar reason … the NH_3 molecule is polar so the bonds are strong too, requiring ~1167 kJ/kg to evaporate it @ 25°C.

  Boiling a non-polar molecule such as R290 (Propane) requires ~335 kJ/kg or R134a requiring ~177 kJ/kg at the same temperature is much easier.

Many of the operational problems in refrigeration systems relate to gas turning into a liquid when it should not or filing to do so when required. Thus this is one area where ‘intuition’ based upon observation can be misleading.

BACKGROUND

A refrigerant can potentially be any fluid that exhibits a gas to liquid phase change, however, making practical use of this characteristic is pressure related, thus certain fluids are less suitable than others for refrigeration applications.

Depending upon the evaporating temperature required, the refrigerants that were traditionally chosen had saturation pressures greater than atmospheric pressure, so that in the event of a rupture/leak, the refrigerant would leak from the system rather than ambient air being drawn into the system.

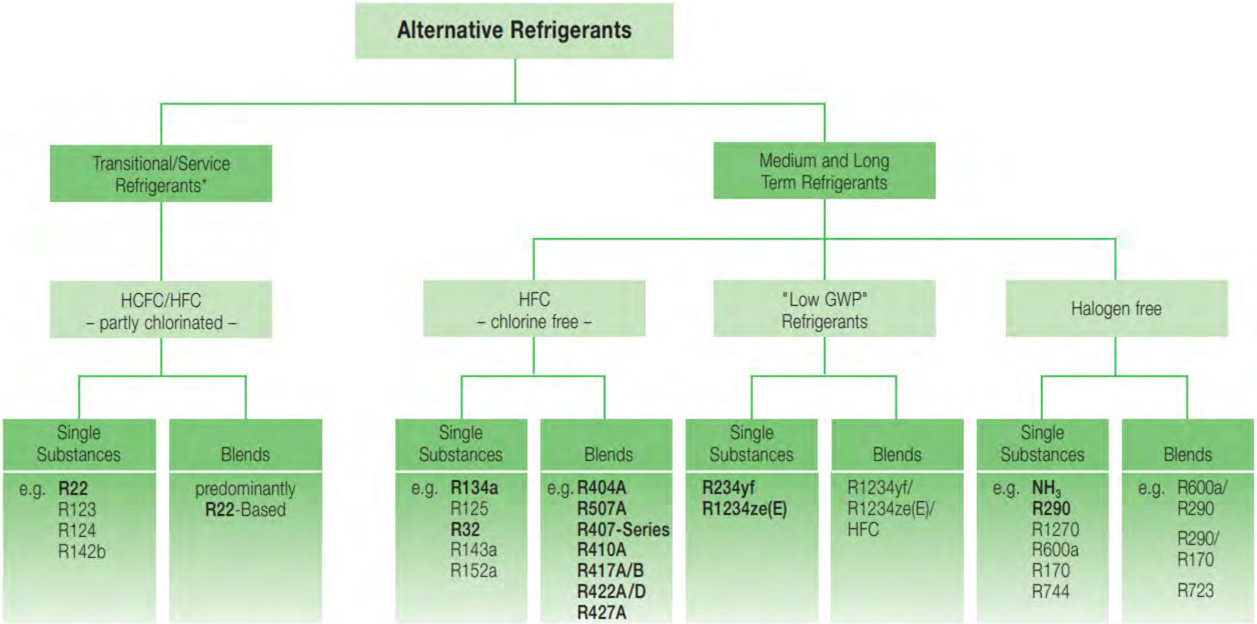
The ingress of air into a refrigeration system is highly detrimental to its performance. Clearly compressors and systems operating below atmospheric pressure (operating in a vacuum) have a greater susceptibility to these adverse effects.

Historically, refrigerants such as ammonia, methyl chloride and sulphur dioxide were commonly used however, with the advent of improved synthesis methods in the 1950s, chlorofluorocarbons (CFCs) became popular establishing R12, 22 & 502 as the leading contenders.

The environmental impact of CFCs was identified during the 1980s and in 1997, the Kyoto Protocol led the way to the phasing out these harmful CFCs, HCFCs & HFCs.

In addition to the introduction of a whole raft of 'environmentally friendly' refrigerants, both Ammonia and Carbon Dioxide (so called natural refrigerants) have fallen back into favour, along with Propane also becoming increasingly popular.

The common trend to invent new refrigerants that have a low global warming potential (GWP) and do not deplete the ozone layer has spawned many new refrigerants. When these refrigerants become established, they are allocated an ASHRAE R number … *originally conceived by DuPont for their range of Freons*. This list is growing monthly.



There is some rationale behind the ASHRAE R number designation, which has some foundation relating to the molecular structure of the compound, but briefly …

- R-10 series : single carbon molecule 'Methane series' e.g. R12, 22, 23, 32, 50
- R-100 series : two carbon molecules 'Ethane series' e.g. R123, 134a, 152a, 170
- R-200 series : three carbon molecule ' Propane series' e.g. R227ea, 245fa, 290
- R-400 series : comprises binary or tertiary zeotropic blends of refrigerants which <u>may</u> exhibit a discernible temperature glide e.g. R404A, 407C, 410A, 449A, 454A
- R-500 series : comprises binary azeotropic blends of refrigerants which <u>do not</u> exhibit any temperature glide e.g. R502, 507A
- R-600 series : organic compounds e.g. R600, 600a
- R-700 series : inorganic compounds e.g. R717, 728, 744
- R-1000 series : unsaturated organic compounds e.g. R1150, 1234yf, 1234ze(E)

Isomers (molecules with the same chemical formula as another molecule but with a different chemical structure) are identified with a ‘lower case’ letter after the ‘R number’ e.g. R123a.

Refrigerant blends having the same pure components but with different compositions are identified with an ‘upper case’ letter after the ‘R number’ e.g. R401A or R410A.

Concerning the above categories of refrigerants, it is the R-400 series that tend to exhibit temperature glide and in the case of condensers, this can impact upon the operating temperatures, approach temperature and selection.

HAZARD CLASSIFICATION – SAFETY GROUPS

ANSI/ASHRAE Standard 34 & ISO 817 assigns an identifying reference letter and number to each refrigerant to classify it according to the hazard involved in use.

The capital letter designates a toxicity class based upon allowable exposure and the number denotes the flammability.

REFRIGERANT TOXICITY

There are two classes for toxicity: lower toxicity (Class A) and higher toxicity (Class B).

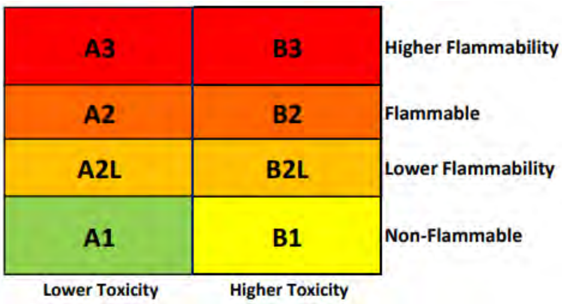
Class A refrigerants are refrigerants for which toxicity has not been identified at concentrations less than or equal to 400 parts per million (ppm) by volume, based on data used to determine threshold limit values (TLV)-time weighted average (TWA) or consistent indices.

Class B refrigerants are refrigerants for which there is evidence of toxicity at concentrations below 400 ppm by volume, based on data used to determine TLV-TWA or consistent indices.

REFRIGERANT FLAMMABILITY

There are four classes of flammability: 1, 2L, 2 or 3 …

- Class 1 is for refrigerants that, when tested, show no flame propagation at 60°C and 101.3 kPa
- Class 2 is for refrigerants that, when tested, exhibit flame propagation at 60°C and 101.3 kPa, have a heat of combustion less than 19,000 kJ/kg and have a lower flammability limit (LFL) greater than 0.10 kg/m^3. *Class 2 is divided up into a subcategory of low versus high. In fact, refrigerants are designated in the 2L subclass if they have a maximum burning velocity of 10 cm/s or lower when tested at 23.0°C and 101.3 kPa. The purpose of the 2L subclass is to reflect the lower flammability properties of the new low-GWP refrigerant options on the rise, such as hydrofluoro-Olefins (HFOs), like R-1234yf and R-1234ze*
- Class 3 is for refrigerants that, when tested, exhibit flame propagation at 60°C and 101.3 kPa and that either have a heat of combustion of 19,000 kJ/kg or greater or a LFL of 0.10 kg/m^3 or lower



Consequently, refrigerants in Class 3 are highly flammable; refrigerants in Class 2 are considered less flammable; and those in Class 2L are mildly flammable.

These classifications are used in the guidelines for determining how much refrigerant can be used in an occupied space. For example, in the case of ammonia, which is both toxic and flammable at the right concentration, the allowed concentration in an occupied space for Ammonia is 320 ppm per ASHRAE Standard 34. (By contrast, R-32 is 36,000 ppm; it has a low toxicity but it is flammable.)

Standards (e.g., ASHRAE 15 & EN 378) and guidelines use this number to determine what size charge can be permitted in a particular facility such that, if all the charge was to leak into an occupied space, the concentration limit would not be exceeded.

Some examples …

Refrigerant type	Boiling temperature [°C] ①	Temperature glide [K] ②	Critical temperature [°C] ①	Cond. temp. at 26 bar (abs) [°C] ①	Refr. capacity [%] ③	Discharge gas temp. [K] ③
HCFC-Refrigerants						
R22	-41	0	96	63	80 (L)④	+35④
R124	-11	0	122	105	⑤	⑤
R142b	-10	0	137	110		
HFC Single-component Refrigerants						
R134a	-26	0	101	80	97 (M)	-8
R152a	-24	0	113	85	N/A	N/A
R125	-48	0	66	51	N/A	N/A
R143a	-48	0	73	56	N/A	N/A
R32	-52	0	78	42	N/A	N/A
R227ea	-16	0	102	96	⑤	⑤
R236fa	-1	0	>120	117		
R23	-82	0	26	1	⑤	⑤
HFC Blends						
R404A	-47	0.7	73	55	105 (M)	-34
R507A	-47	0	71	54	107 (M)	-34
R407A	-46	6.6	83	56	98 (M)	-19
R407F	-46	6.4	83	57	104 (M)	-11
R422A	-49	2.5	72	56	100 (M)	-39
R437A	-33	3.6	95	75	108 (M)	-7
R407C	-44	7.4	87	58	100 (H)	-8
R417A	-39	5.6	87	68	97 (H)	-25
R417B	-45	3.4	75	58	95 (M)	-37
R422D	-45	4.5	81	62	90 (M)	-36
R427A	-43	7.1	87	64	90 (M)	-20
R438A	-42	6.6	80	63	88 (M)	-27
R410A	-51	<0.2	72	43	140 (H)	-4
ISCEON MO89	-55	4.0	70	50	⑤	⑤
R508A	-86	0	13	-3	⑤	⑤
R508B	-88	0	14	-3		
HFO and HFO/HFC Blends – further blends and data see page 25						
R1234yf	-30	0	95	82	98 (M)	-14
R1234ze(E)	-19	0	110	92	⑤	⑤
R513A (XP10)	-29	0	97	78	102 (M)	-7
R450A (N-13)	-24	0.6	105	86	88 (M)	-6
R448A (N-40)	-46	6.2	83	58	96 (M)	+12
R449A (XP40)	-46	4.5	82	58	96 (M)	+12
Halogen free Refrigerants						
R717	-33	0	133	60	100 (M)	+60
R723 ⑦	-37	0	131	58	105 (M)	+35
R600a	-12	0	135	114	N/A	N/A
R290	-42	0	97	70	89 (M)	-25
R1270	-48	0	92	61	112 (M)	-20
R170	-89	0	32	3	⑤	⑤
R744	-57⑥	0	31	-11	⑤	⑤

① Rounded values

② Total glide from bubble to dew line – based on 1 bar (abs.) pressure. Real glide dependent on operating conditions. Approx. values in evaporator: H/M 70%; L 60% of total glide

③ Reference refrigerant for these values is stated in Fig. 33 under the nomination "Substitute for" (column 3)
Letter within brackets indicates operating conditions
H High temp (+5/50°C)
M Medium temp (-10/45°C)
L Low temp (-35/40°C)

④ Valid for single stage compressors

⑤ Data on request (operating conditions must be given)

⑥ Triple point at 5.27 bar

Stated performance data are average values based on calorimeter tests.

Actual HFC Refrigerants	Alternatives		Components / Mixture components "Low GWP" alternatives							
	Safety Group ↓	GWP ④ ↓	R1234yf A2L GWP 4	R1234ze(E) A2L 7	R32 A2L 675	R152a A2 124	R134a A1 1430	R125 A1 3500	CO_2 ② A1 1	R290 ② A3 3
R134a GWP 1430	A1	~ 600	✔	✔	✔		✔			
	A2L	< 150	✔	✔	✔	✔	✔			
	A2L	< 10	✔	✔						
R404A/R507A GWP 3922/3985	A1	< 2500①	✔		✔			✔		✔
	A1	~ 1400	✔	✔	✔		✔	✔		
	A2L	< 250	✔		✔	✔				✔
	A2L③	< 150	✔		✔				✔	
	A2	< 150	✔		✔					✔
R22/R407C GWP 1810/1774	A1	900..1400	✔	✔	✔		✔	✔		
	A2L	< 250	✔		✔	✔				
	A2L③	< 150	✔		✔					
	A2	< 150	✔		✔					✔
R410A GWP 2088	A2L	< 750			✔					
	A2L	~ 400..750	✔	✔	✔			✔	✔	

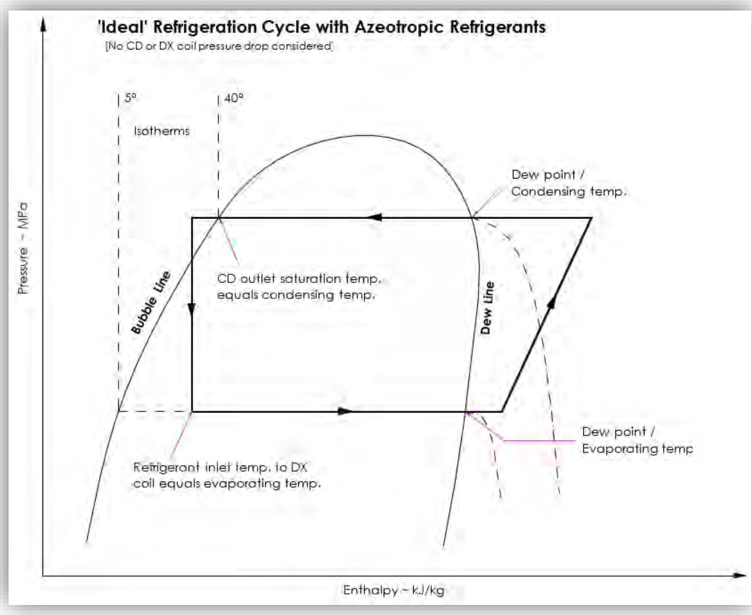
① Refrigerating capacity, mass flow, discharge gas temperature similar to R404A
② Only low percentage – due to temperature glide (CO_2) and flammability (R290)
③ R32/HFO blends show lower refrigerating capacity than reference refrigerant, the addtion of CO_2 leads to high temperature glide
④ Approx. values according to IPCC IV

Current HFC refrigerant	"Low GWP" Alternatives for HFC refrigerants ⑤					
	ASHRAE Number	Trade Name		Composition (with blends)	GWP ⑤ AR4 (AR5)	Safety Group
R134a GWP 1430 ①	**R450A**	Solstice® N-13	Honeywell	R1234ze(E)/134a	604 (547)	A1
	R513A	Opteon® XP10	Chemours	R1234yf/134a	631 (573)	A1
	R513B	–	Daikin Chemical	R1234yf/134a	596 (540)	A1
	R456A	AC5X ③	Mexichem	R32/1234ze(E)/134a	687 (627)	A1
	R1234yf	various		–	4 (< 1)	A2L
	R1234ze(E) ②	various		–	7 (< 1)	A2L
	R444A	AC5 ③	Mexichem	R32/152a/1234ze(E)	92 (89)	A2L
	–	ARM-42 ④	Arkema	R1234yf/152a/134a	142 (131)	A2L
R404A/R507A GWP 3922/3985 (**R22/R407C**)	**R448A**	Solstice® N-40	Honeywell	R32/125/1234yf/1234ze(E)/134a	1387 (1273)	A1
	R449A	Opteon® XP40	Chemours	R32/125/1234yf/134a	1397 (1282)	A1
	R449B ④	–	Arkema	R32/125/1234yf/134a	1412 (1296)	A1
	R460B	LTR4X ③	Mexichem	R32/125/1234ze(E)/134a	1352 (1242)	A1
	R452A	Opteon® XP44	Chemours	R32/125/1234yf	2140 (1945)	A1
	R452C ④	–	Arkema	R32/125/1234yf	2220 (2019)	A1
	R460A	LTR10 ③	Mexichem	R32/125/1234ze(E)/134a	2103 (1911)	A1
	R454A	Opteon® XL40	Chemours Daikin Chemical	R32/1234yf	239 (238)	A2L
	R454C ②	Opteon® XL20	Chemours	R32/1234yf	148 (146)	A2L
	R455A	Solstice® L-40X	Honeywell	R32/1234yf/CO_2	148 (146)	A2L
	–	ARM-20b ④	Arkema	R32/1234yf/152a	251 (251)	A2L
	R457A ②	ARM-20a ④	Arkema	R32/1234yf/152a	139 (139)	A2L
	R459B ②	LTR11 ③	Mexichem	R32/1234yf/1234ze(E)	144 (143)	A2L
R22/R407C GWP 1810/1774	**–**	Solstice® N-20	Honeywell	R32/125/1234yf/1234ze(E)/134a	975 (891)	A1
	R444B	Solstice® L 20	Honeywell	R32/152a/1234ze(E)	295 (295)	A2L
R410A GWP 2088	**R32**	various		–	675 (677)	A2L
	R447B	Solstice® L-41z	Honeywell	R32/125/1234ze(E)	740 (714)	A2L
	R452B	Opteon® XL55	Chemours	R32/125/1234yf	698 (676)	A2L
	R454B	Opteon® XL41	Chemours	R32/1234yf	466 (467)	A2L
	R459A	ARM-71 ④	Arkema	R32/1234yf/1234ze(E)	460 (461)	A2L

① The relatively low GWP allows the use of R134a also longer term
② Lower refrigerating capacity than reference refrigerant
③ Development product
④ Availability 2017 .. 2020
⑤ AR4: according to IPCC IV // AR5: according to IPCC V – time horizon 100 years

IDEAL AZEOTROPE P-H DIAGRAM

In preparation for the following sections concerning azeotropic and zeotropic mixtures, the ideal refrigeration cycle (*no component pressure drop allowance*) is presented for comparison purposes.



AZEOTROPIC MIXTURE

An azeotrope is a mixture of liquids that maintains its composition at its boiling point resulting in a constant boiling point.

The term was coined by *John Wade* and *Richard William Merriman* in 1911from the Greek for 'boil', 'turning' & 'no'.

Some common examples of azeotropes are R22, R134a, R290, R717 & R744 all of which evaporate or condense at a constant temperature. Thus, if pressure drop in the condenser or evaporator is ignored, will conform to the above 'ideal' refrigeration cycle.

But every component in a refrigeration system creates a pressure drop of some degree and this pressure drop equates to a refrigerant temperature drop. Therefore, when a 'real' refrigeration cycle is plotted on a p-h chart, the otherwise horizontal condensing and evaporating pressure process lines are slightly inclined downwards in sympathy with the refrigerant flow direction.

CONDENSING

The consequence of this is that if the condensing pressure equates, for example, to a condensing temperature of 40°C (*the inlet saturation temperature at the saturated vapour/dew line*), then the outlet temperature will not be 40°C but slightly lower equating to the refrigerant temperature drop associated with the pressure loss in the condenser. So perhaps 38°C or maybe 35°C as shown in the example below.

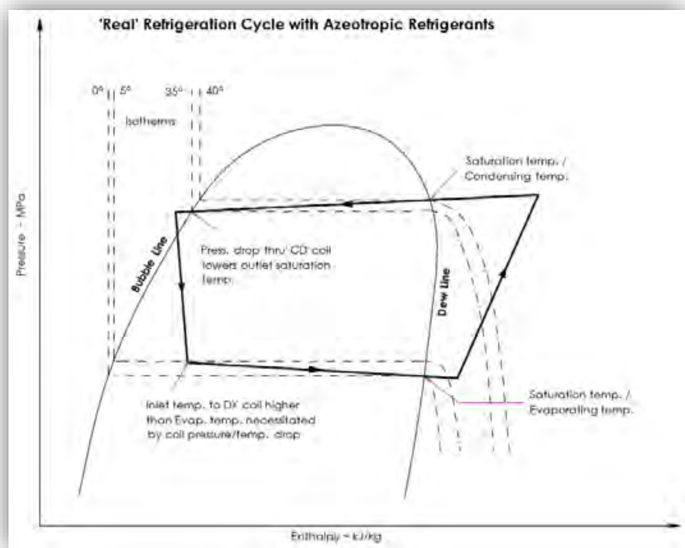
Traditionally, a 2K temperature/pressure loss is considered when analysing refrigeration systems, before the actual pressure loss is known/calculated. However, providing this relationship is known and allowed for in the calculation of the LMTD, then the advantages of a higher circuit loading (circuit mass flow rate) and resulting higher internal heat transfer coefficient may offset the detrimental impact that the pressure/temperature loss has upon the LMTD. Thus, the nett effect may be positive.

EVAPORATING

In the case of an evaporator, the definition of the evaporating temperature is defined as the saturated vapour temperature at the evaporating pressure i.e. the outlet conditions of the evaporator.

Clearly, an evaporator exhibits a pressure loss too and as above, the associated refrigerant temperature drop suggests that the refrigerant inlet temperature must be higher than the evaporating temperature, so as the refrigerant evaporates, it drops in temperature, eventually reaching the evaporating temperature towards the outlet of the evaporator, prior to any superheating of the saturated gas that may be required.

Both the condensing and evaporating behaviour is shown in the following figure.



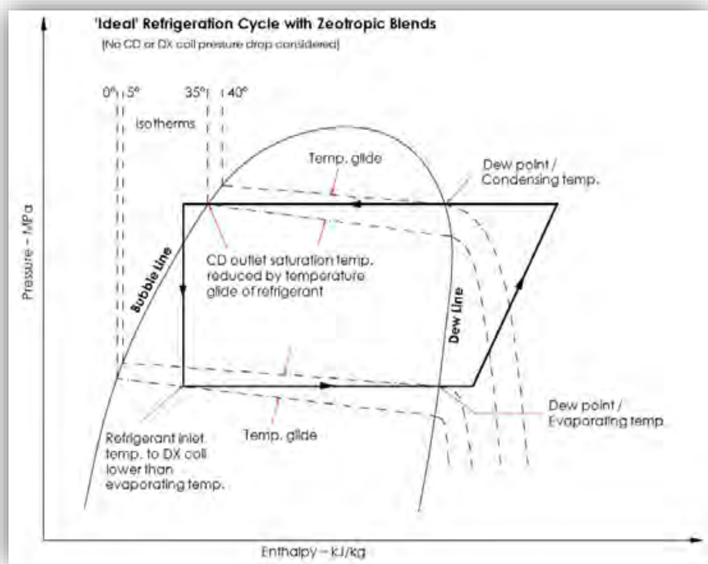
ZEOTROPIC MIXTURE

A zeotropic mixture (*or non-azeotropic mixture*) is a combination of components each having different boiling points.

These individual substances within the mixture do not evaporate or condense at the same temperatures and thus, such a fluid exhibits a 'temperature glide', which can typically be between 4 - 7K.

Some common examples of zeotropic mixtures are R404A, 407C, 410A all of which exhibit temperature glide, where R407C is typically around 5K.

When shown on a p-h chart the constant temperature lines inside the '*vapour dome*' are not parallel and in sympathy with the pressure lines (*as an azeotrope*) but slope downwards from left to right.



CONDENSING

Thus, in the case of a condensing process, the inlet condensing temperature of say 40°C equates to say 35°C at the outlet saturated liquid condition, as a result of the 5K glide … and this is without the condenser coil pressure drop superimposed.

However, when the glide is compounded with the refrigerant temperature drop associated with the pressure drop in the condenser, the effects are worsened … see figure below.

In this instance, the temperature drop associated with the pressure drop has an accumulative effect of an additional 3K and thus the 'real' saturated liquid temperature at the condenser outlet is 32°C and not 35°C (*glide only*) or indeed 40°C (*ideal*). The nett effect is an impairment to the condenser operating LMTD and thus performance.

SUBCOOLING

This characteristic of zeotropic mixtures also has an impact upon the interpretation of/calculation of the subcooling that may be required.

Traditionally, in the days of R12, 22 & 502, condensers were designed for pressure drops equating to perhaps 1K temperature drop and being azeotropes, the evaporating process was at a constant temperature. So with a condensing temperature of 40°C a requirement of 5K subcooling suggested a subcooled liquid refrigerant temperature of 40 - 1 - 5 = 34°C.

However, today's zeotropic mixtures, such as R407C with a 5K glide would result in … 40 - 5 - 1 - 5 = 29°C.

Clearly, with an operating TD of 12K and thus an air temperature cooling this condenser of 28°C, an approach temperature of 34 – 28 = 6K is reasonable for the azeotropic condenser, but an approach of 29 – 28°C = 1K for the zeotropic mixture condenser, is an issue and perhaps an impractical solution.

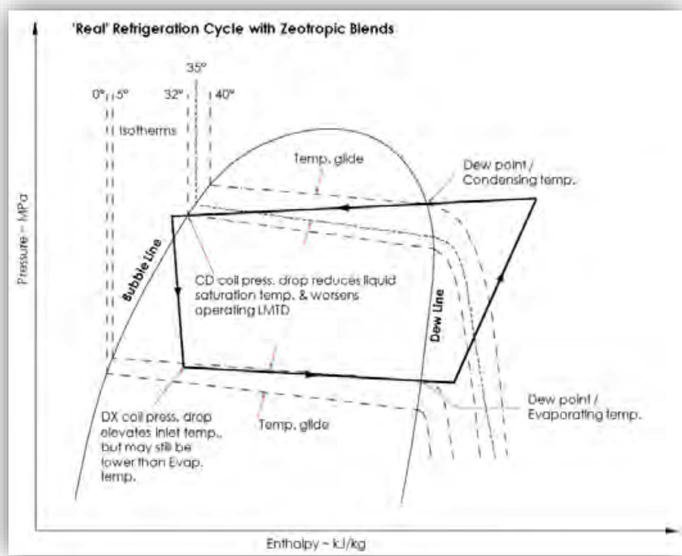
EVAPORATING

In a similar fashion, the evaporating process - *assuming no pressure loss* – results in the refrigerant inlet liquid/flash gas mixture (x ≈ 0.15) does not evaporate at a constant temperature, but increases in temperature; due to the glide; as it evaporates at constant pressure.

So, in the example above, the refrigerant enters at 0°C and whilst boiling, increases in temperature reaching +5°C evaporating temperature prior to any superheat requirements, before exiting the evaporator.

This unusual behaviour benefits the evaporator by widening the approach temperature and thus operating LMTD and hence the performance.

However, the pressure loss/temperature penalty moderates the temperature glide implications and if the pressure drop is significant, can destroy this advantage and result in a more or less constant evaporating temperature.



CIRCUITRY PREFERENCES

Circuitry is always a very subjective issue as there are number of ‘schools of thought’ regarding this topic.

SINGLE PHASE FLUIDS

For a single phase fluid, ensuring a cross counter flow circuitry arrangement is necessary to emulate the assumed ‘counter flow LMTD’ used in the thermal calculations. Furthermore, it is often desirable to ensure that the circuit flow path proceeds ‘uphill’ to force air out of the system during the filling operation and assisting the evacuation of the fluid when draining the system … bearing in mind that natural ‘free draining’ is nigh on impossible with long tube lengths due to surface tension retention implications!!

TRADITIONAL REFRIGERANTS

Back in the days of R12, 22, 502 and Ammonia (NH_3, R717), all these refrigerants were Azeotropes and do not exhibit temperature glide. So, the ‘rules of thumb’ in those days were that above -20°C evaporating temperatures, all circuitry should generally feed ‘uphill’ towards the outlet (suction) of the circuit, which is obviously positioned on the ‘air inlet’ face of the coil. *This is to ensure that the widest possible TD is available to provide the design superheat of the refrigerant.*

Furthermore, the ‘uphill’ path also ensures that the entering liquid refrigerant (*actually around 85% liquid and 15% flash gas*) is forced to remain in the coil circuit to fully evaporate and finally become superheated to the correct temperature.

When the circuit path is ‘downhill’ then there can be a propensity for the liquid refrigerant; under gravity; to reach the suction header without reaching the saturation temperature or the desired superheated temperature. Therefore, when ‘slugs’ of cold unevaporated refrigerant liquid enter the suction header and reach the vicinity of the TEV temperature sensor, they tend to cause the TEV to close and thus reduce the refrigerant mass flow rate, which detrimentally affects the coil performance and usually results in unstable behaviour.

At evaporating temperatures below -20°C, oil return to the compressor can become a major concern and thus the circuitry pattern for such applications tended to be ‘downhill’.

EAS does not encounter too many low temperature DX applications these days, and besides, these are usually pump circulated systems using Ammonia.

Generally using and uphill philosophy for the circuitry pattern is advantageous. In particular, the necessity to ensure that the last couple of passes of the circuitry are uphill for <u>all</u> circuits … and these last couple of passes are positioned in the first row of the coil … the air entering face.

It is important that all circuits and their associated pattern are similar, to ensure that the evaporation behaviour and superheating in each circuit is similar, which is coined by the phrase … ‘a DX coil’s performance is only as good as the worst performing circuit’. This is justified by the fact that all circuits dump their, hopefully superheated refrigerant, into a common suction header, which is where the TEV sensor is located. So, if any one circuit is delivering unevaporated slugs of liquid into the header, then the whole coil will be affected.

The above premise also assumes that both the air inlet temperature and air velocity profile is uniform across the whole coil face. However in reality, this is rarely true because the lower air velocity at the top and bottom of the coil (*and indeed the two sides of the coil*) reduces the ability to superheat the refrigerant to the same level as the bulk of the circuits in the rest of the coil. This is usually why the ‘real’ operating superheat for many DX coils is somewhat greater than the suggested ‘low’ operating superheat that the TEV manufacturers suggest!!

Providing the operating TD of the evaporator is reasonable, the refrigerant temperature penalty of the evaporating refrigerant resulting in the inlet temperature needing to be higher than the evaporating temperature, will not be too badly affected.

However in more limiting cases, the pinch of the TD between the air leaving temperature and elevated refrigerant inlet temperature, may result in performance issues … especially at low evaporating temperatures where the temperature penalty is somewhat greater.

In such cases, adopting a special circuit pattern where the initial part of the circuit causes the refrigerant to flow is in sympathy with the air direction, will optimise the air/refrigerant temperature profile to maximise the LMTD, but ensuring that the last few passes of the circuit are focused on the air entering face to maximise the suction gas superheat.

REFRIGERANTS WITH GLIDE

Zeotropic refrigerant blends such as R407C, exhibit a 5K glide and for normally cross counter flow circuited evaporator coils, this will advantage the LMTD and the associated temperature differences on either face of the coil. So rather than exhibiting the ‘pinch TD’ at the air off side of the coil, the ‘glide’ will effectively widen this TD, thus improving the LMTD … ***Remember*** that this is not the case with an azeotropic refrigerant.

However, for condensers, the temperature glide is in sympathy with refrigerant temperature drop associated with the pressure drop in the circuit and compounds the total temperature penalty, worsening the LMTD and for limiting cases where the operating TD is small, detrimentally affects the coil performance.

CARBON DIOXIDE - CO2 - R744

BACKGROUND

CO_2 is an unusual fluid and has recently been resurrected as a popular refrigerant used in refrigeration and heat pump applications.

CO_2 may not be the most efficient refrigerant but is considered safe, non-toxic, non-flammable, does not deplete the ozone layer and has a GWP of only 1.0, whilst many other popular 'natural' refrigerants are either or both toxic (ammonia), flammable (propane), have a somewhat higher GWP and/or may not be 'kind' to the environment.

Although CO_2 has several benefits it only has a critical temperature of 30.98°C and thus in many 'normal' global ambient temperatures, cannot operate like a convention refrigeration system i.e. with a condenser as the high pressure and temperature energy emitter, and must operate in the supercritical region with the condenser replaced by a gas cooler.

In addition to CO_2's low critical temperature resulting in the need for the gas cooling process, CO_2 exhibits a particularly high pressure e.g. the critical pressure is 72.8 barg.

Consequently, typical gas cooler operating conditions … in say an ambient of 32°C … will often be 90 to 120 barg with perhaps an inlet temperature of 120 to 125°C, where the gas is cooler down to perhaps 35°C

Such operating pressures involve more specialised and expensive components such as compressors, valves and heavier duty pipe work. However, due the CO_2's density at these conditions, the volumetric flow rate is lower and smaller diameter pipe work can be used compared with other refrigerant systems. This helps to offset the economic implications associated with CO_2 systems.

When a CO_2 system is designed with a gas cooler; hence operating as a transcritical system; high COPs can be achieved and the high compressor discharge temperatures are well suited to producing high temperature water for both domestic and district heating purposes.

CO_2 systems can operate supercritically (transcritical process), subcritically (as a conventional refrigeration system) and on the evaporator side, can operate as a DX, flooded or as a pumped system. Additionally, low pressure, low temperature CO_2 liquid can be pumped around, for example supermarket display cabinets, in the form of a secondary circulating system.

So for low temperature applications, such as evaporators or subcritical condensers as part of a cascade system, conventional materials of construction can be adopted, providing PS & TS are sufficiently low. However, at elevated conditions, either thick wall copper, K65 or stainless steel materials are required.

If the pressure equipment (evaporator, condenser or gas cooler) falls outside PED Art 4/3 and is thus PED Cat I to IV, EAS is not currently approved to manufacture copper or K65 coils of this type, plus if the PS is greater than 41 barg and the tube/headers are stainless steel, then again EAS is not approved to manufacture such equipment.

Currently, this means that we need to manipulate the design/construction to fall into PED S.E.P. (Art 4/3), which means that in-line with PED Guideline B-04, where the refrigeration/ heat pump application can be deemed to be a Piping System, we need to keep the 'largest' header of any discrete coil section, to <=DN32.

Unfortunately, this often results in splitting the coil into discrete sections resulting in multiple sets of connections, which the Client may not prefer!

Until EAS acquires approval for Category copper or K65 coils and/or extends its stainless steel pressure rating above 41 barg, we are unfortunately limited in what options we can offer.

For the future, it is expected that we will have to increase both the stainless steel and copper/K65 tube and header wall thicknesses to handle the somewhat higher pressures that are exhibited with CO_2 applications. This will have production related implications e.g. WPS & WPQR for the thicker materials and perhaps the need for 'safer' test pressure facilities when we are considering test pressure in the region of 172 barg !!

Note : During 2022 Evapco Inc. launched their Gas Cooler variant of the eco-Air product line where they use 1.05 mm wall thickness 5/8" SST tube and either ASME Sch 80 or Sch 160 header pipe to comply with their nominated ASME B31.5 heat exchanger design code.

EAS MATERIAL/COMPONENT GUIDELINES

Owing to the high gas cooler, condensing and evaporating pressures associated with CO_2 plus the current PED approval for stainless steel coils, EAS is not currently able to manufacture equipment for all operating conditions.

The following attempts to provide the scope and limitations etc. for CO_2 Art 4/3 and Category coils used in the following applications …

- Subcritical condensers
- Supercritical gas coolers
- DX evaporators

PED ARTICLE 4/3 APPLICATIONS

- HVAC/R applications invoking PED Guideline B-04, where we can consider the pressure equipment (coil) as …
  - '**Piping**' rather than a Vessel
  - Where the Vessel aspects (Cat_V) of the largest header <= to the Piping aspects (Cat_P) of the pressure equipment
- Maximum inlet or outlet header/connection size **<=DN32** (1¼") or if not DN rated material, I.D. <= 32 mm
- Thick wall copper headers, Tees & domed endcaps : <= 50 barg
- K65 high tensile copper headers, Tees & domed endcaps : <= 120 barg
- Z-fin using 3/8" x 0.5 mm copper tube : <= 120 barg (Test pressure : 172 barg)
- A-fin & C-fin using Ø15 mm x 0.75 mm copper tube : <= 80 barg (Test pressure : 115 barg)

PED CATEGORY I – IV/MOD B+D APPLICATIONS

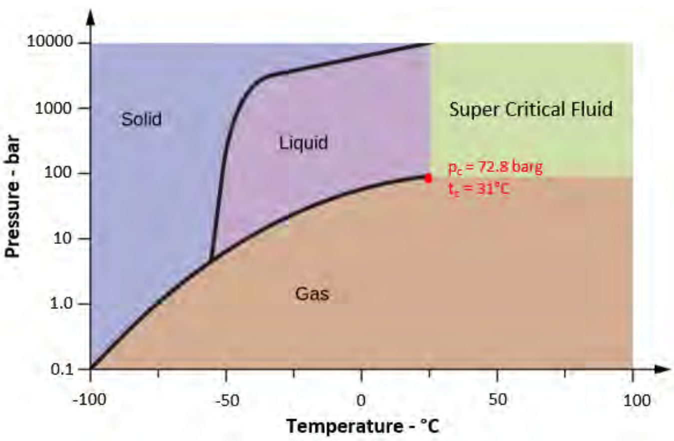
- Not approved for copper or K65 tube/header material
- Stainless 304/316 Ø15 mm tube & header material …
  - Maximum PS = 41 barg
  - Maximum TS = 300°C

    Use PED Strength Tool to validate tube and header wall thickness suitability
- 41 barg saturation pressure for CO_2 = **+7.23°C** saturation temperature
- Cannot manufacture pressure equipment with stainless tubes and/or headers with either evaporating or condensing temperatures >+7.23°C

CO2 BEHAVIOUR

Dictated by both temperature and pressure, CO_2 can exist as a solid (dry ice), a liquid, a gas and when operating above the critical point, a super critical fluid.

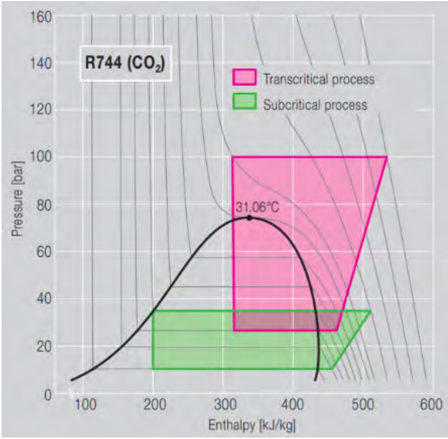
Compared with traditional and currently used refrigerants, it has some distinct advantages, however, almost monthly new blends are being concocted which exhibit excellent thermal properties, low GWP and fall into low safety groups.

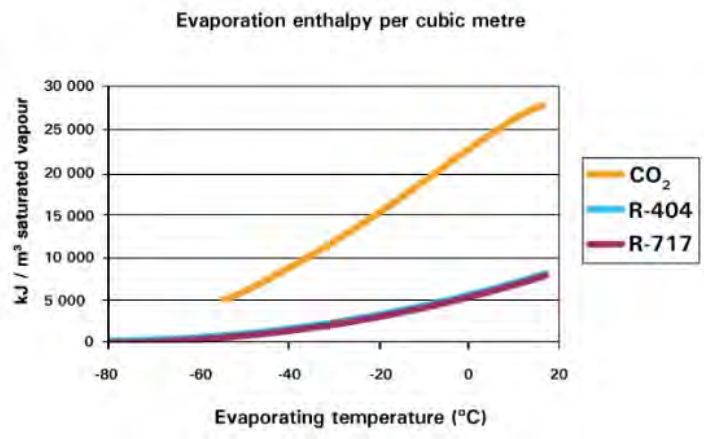


Although Ammonia exhibits a somewhat greater latent heat compared with all other refrigerants, the superior volumetric efficiency of CO_2 is much higher than other refrigerants, resulting in a smaller compressor displacement and smaller interconnecting pipe diameters.

			Critical							Properties at 0°C								
			Temp.	Pressure	Triple Point	T_{SAT} @ 0 barg	P_{SAT} @ 20°C	GWP	Safety	Press	Density - kg/m³	Spec. Heat - kJ/kg/K		Thermal Cond. - W/m/K		Viscosity - mPa.s		Latent Heat
	Refrigerant		°C	barg	°C	°C	barg	100 years	Group	barg	Vapour	Liquid	Vapour	Liquid	Vapour	Liquid	Vapour	kJ/kg
	R22		96.1	48.9	-157.4	-40.8	8.1	1790	A1	3.90	21.20	1.17	0.74	0.095	0.0094	0.216	0.0110	205
	R134a		101.1	39.6	-103.3	-26.1	4.7	1430	A1	1.91	14.40	1.34	0.89	0.092	0.0115	0.267	0.0107	198
	R404A		72.1	36.3	-	-46.2	9.9	3922	A1	5.10	30.40	1.39	1.00	0.074	0.0133	0.181	0.0111	165
	R407C		86.1	45.3	-	-43.6	9.3	1774	A1	4.60	22.70	1.42	1.01	0.096	0.0121	0.211	0.0116	217
	R410A		71.3	47.9	-	-51.4	13.4	2229	A2L	6.99	30.57	1.51	1.13	0.101	0.0123	0.163	0.0122	221
Natural	R290	Propane	96.7	41.5	-187.6	-42.1	7.4	3.3	A3	3.73	10.35	2.49	1.74	0.106	0.0157	0.126	0.0074	374
	R717	Ammonia	132.4	112.6	-77.6	-33.3	7.6	0	B2L	3.28	3.45	4.61	2.69	0.522	0.0236	0.165	0.0091	1262
	R744	CO_2	31.0	72.8	-56.6	-78.5	56.2	1.0	A1	33.84	97.60	2.54	1.86	0.109	0.0204	0.100	0.0150	231

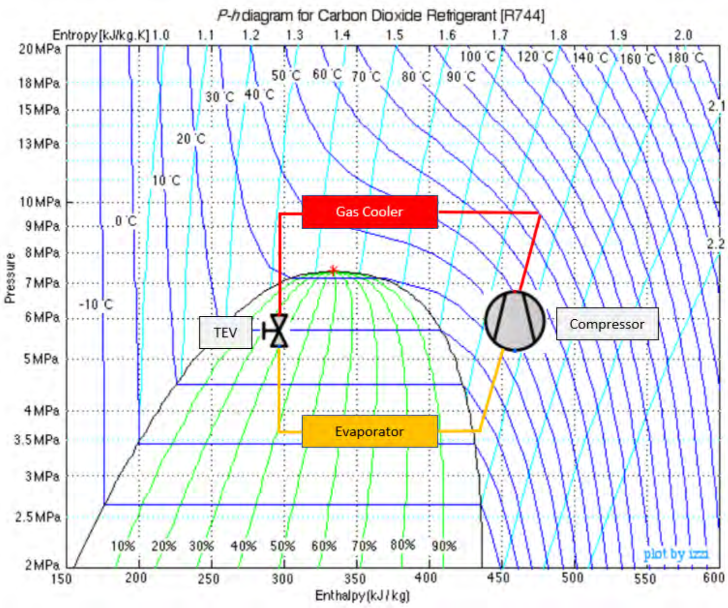
CO_2 applications can be designed to be conventional subcritical systems or transcritical when the high side pressure exceeds the critical point and thus a gas cooler replaces the condenser.



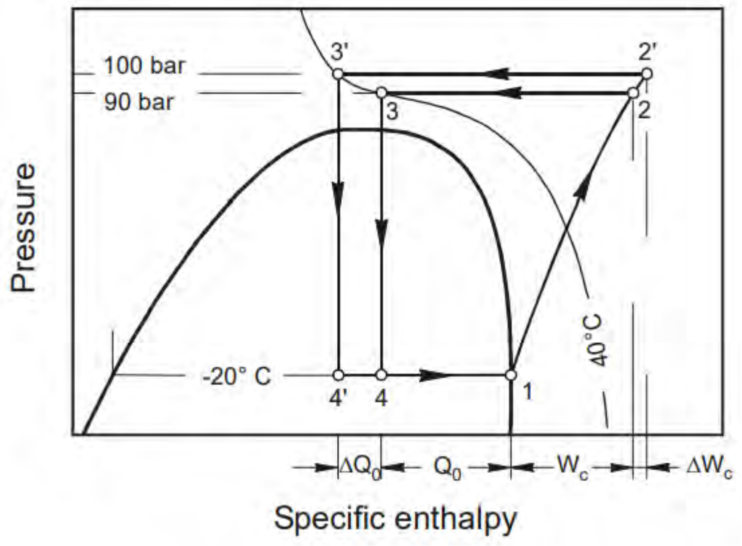


SINGLE STAGE TRANSCRITICAL SYSTEM

The following is an example of a typical simple single stage transcritical CO_2 system superimposed on a p-h chart …

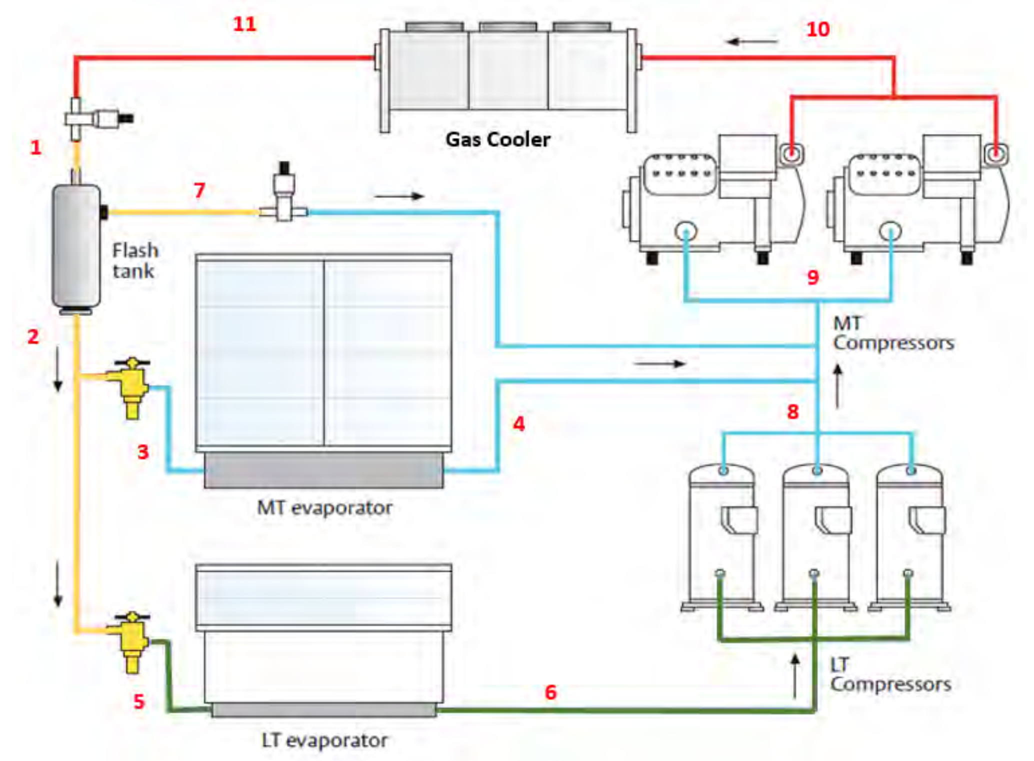


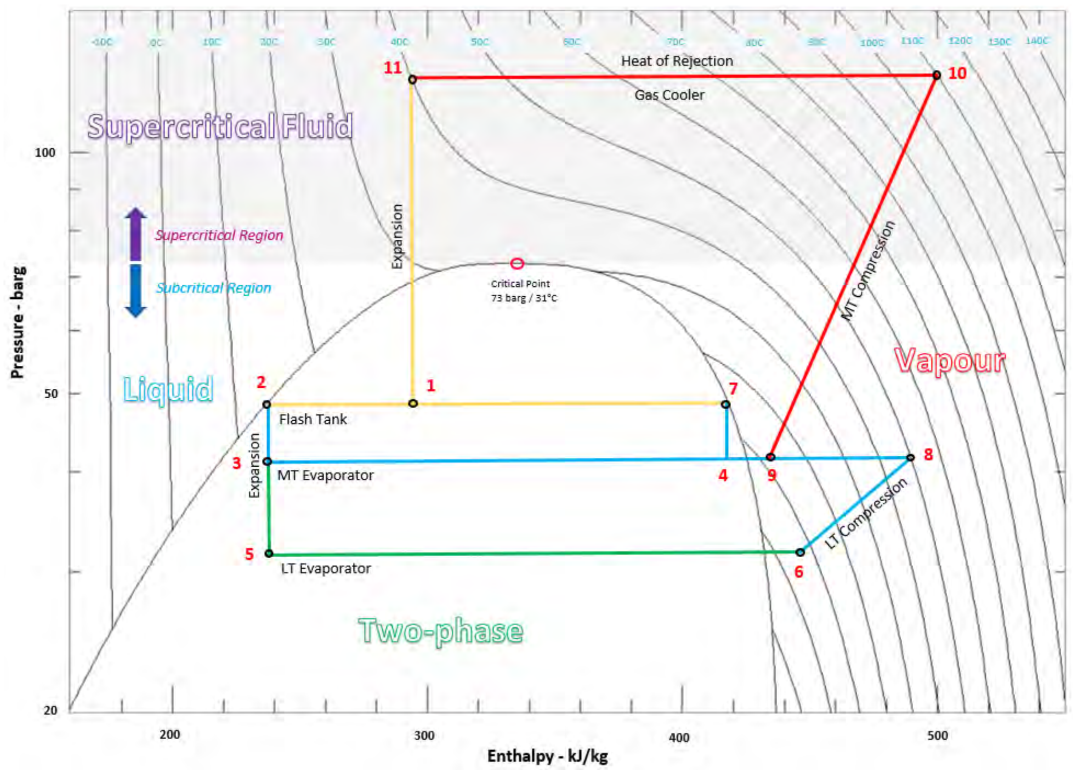
The unusual supercritical (transcritical) behaviour exhibited by CO_2 enables the Refrigeration Effect @ -20°C evaporating temperature (*enthalpy process 4 to 1*) to be increased by simply elevating the gas cooler operating pressure from 90 to 100 bar, whilst maintaining the same gas cooler outlet temperature of 40°C shown in the figure below …



Such behaviour is not reflected in subcritical systems, where the result of elevating the condensing temperature would likely reduce the available refrigeration effect (capacity) and unless additional subcooling provided, will result in an increased condenser outlet temperature.

The following represents a two stage low pressure (subcritical) and high pressure (transcritical) system where the component diagram is also represented on the following p-h chart indicating the CO_2 state at each point of the system.





CURRENT POSITION - 2023

The Montreal Protocol regulations on gases that deplete the earth's ozone layer have led to the phaseout of chloro-fluorocarbons (CFCs) as refrigerants in industrialized countries. Moreover, hydrochloro-fluorocarbons (HCFCs) are only an interim solution in industrialized countries until the year 2020.

Another environmental concern regarding these refrigerants is their behaviour as greenhouse gases in the atmosphere. This also applies to the newly developed hydrofluorocarbons (HFCs). For this reason, these new HFC refrigerants are grouped with five other gases covered by the Kyoto Protocol on greenhouse gases. This situation has led to increased use of the "old" refrigerants — ammonia and hydrocarbons. Although both are environmentally benign, they can exhibit a certain degree of local danger due to their flammability and/or toxicity. Therefore, carbon dioxide (CO_2), an "old" refrigerant used in industrial and marine refrigeration, was proposed by the late Prof. Gustav Lorentzen in 1990 to be used as an alternative refrigerant, mainly because of its non-flammability.

OZONE DEPLETION POTENTIAL

As opposed to CFCs and HCFCs, ammonia, hydrocarbons, and CO_2 all have an Ozone Depletion Potential (ODP) of zero and a negligible Global Warming Potential (GWP). As for HFCs, their ODP is zero, however their GWP ranges from a few hundred in the case of the flammable HFC-32 to several thousand in the case of the flammable HFC-143a and the non-flammable R-125. With respect to the safety of "old" refrigerants, only CO_2 can compete with the non-flammable HFCs.

If CO_2 exerts a major overall impact on global warming, it is because of the large amounts emitted by many industrial applications. However, contrary to HFCs, its GWP is negligible when applied as a refrigerant. Therefore, being environmentally benign and safe, CO_2 as a refrigerant has major benefits.

CARBON DIOXIDE CYCLE

Owing to its thermodynamic properties, CO_2 differs from the other refrigerants mentioned. Its vapor pressure is much higher, with a critical temperature around 31°C. Because the heat discharge into the ambient atmosphere above this temperature is impossible through condensation, as happens in the normal vapor compression cycle, such systems need to adopt a gas cooler.

CO_2 can only be used in the classic and very efficient refrigeration cycle when heat discharge temperatures are lower than the critical temperature (e.g., when used in the lower stage of a cascade system, with another refrigerant being used in the higher stage).

For heat rejection at supercritical pressure, only gas cooling, not condensation, is possible. This leads to the cycle known as the transcritical cycle. It was proposed by Lorentzen and his co-workers for automotive air conditioning and heat pump systems.

This transcritical cycle is not new. It has been well known since the last century as the Linde-Hampson process for air liquefaction, based on the Joule-Thomson effect. In this context, it exhibits a certain lack of efficiency.

APPLICATIONS OF CARBON DIOXIDE

Refrigerant and CO_2 emissions from energy supply to refrigerating systems both contribute to greenhouse gas emissions expressed by using Total Equivalent Warming Impact (TEWI). Therefore, refrigeration systems with a high degree of emission are preferred application areas for CO_2 as an alternative refrigerant, as long as the energy efficiency, defined as Coefficient of Performance (COP), can be kept at the same level.

In the 1991 Technical Options Report of UNEP, automotive air conditioning was identified as the application with the largest refrigerant consumption worldwide and the highest direct effect on TEWI, expressed as a percentage. Therefore, Lorentzen and his co-workers first drew attention to the application for CO_2 as a refrigerant. Out of necessity, they employed the trans-critical cycle because of higher outside air and heat discharge temperatures when running mobile air conditioning systems. But the entire transport sector can be a main application for CO_2 as a refrigerant.

Commercial refrigeration, including systems used in supermarkets, also has a rather large impact on TEWI due to the long refrigerant lines and the large refrigerant charges. Cascade systems with CO_2 as the low-temperature refrigerant in a classic vapor compression cycle, or CO_2 as secondary refrigerant, are possibilities enabling reduction of greenhouse gas emissions of refrigerants without the disadvantage of higher energy consumption.

The third largest quantity of refrigerant emission per system apply to unit air conditioning and heat pump systems. In the heat-pump application, unit systems and chillers offer good perspectives for CO_2 as a refrigerant, thanks to use of the transcritical cycle. The heat rejected on the high-temperature side is used for space heating or hot water production.

Since the transcritical cycle also shows a temperature glide in the gas cooler, the temperature profiles of the refrigerant and the secondary fluid can be advantageously adapted to minimize heat transfer loss and hence improve energy efficiency. Good results can be achieved only with similar and rather large temperature intervals on both sides, so the preferred application should be hot air or water production.

ADVANTAGES OF CARBON DIOXIDE

In the transcritical cycle, gas cooler pressure and temperature are not linked as in the sub-critical two-phase region.

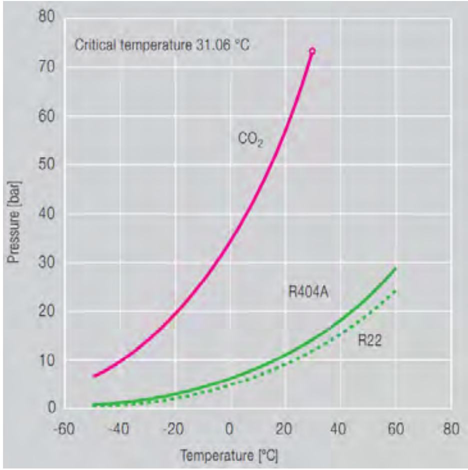
Since the high-side pressure greatly affects - via the pressure ratio - compressor work and efficiency, high temperatures can be achieved with reasonable compressor power. Therefore, the application of CO_2 in heat pumps (e.g., for hot water at 90°C) can be an excellent choice.

The high vapor pressure leads not only to a low-pressure ratio with the advantage of high compressor efficiency, but also to high heat transfer coefficients and low relative pressure losses. Thus, despite the lack of efficiency of the theoretical transcritical cycle, the CO_2 supercritical refrigeration cycle can still compete with the vapor compression cycle using other refrigerants.

At the same time, CO_2 has a low compression pressure ratio (20 to 50% less than HFCs and ammonia), which improves volumetric efficiency. With evaporation temperatures in the range of -55°C to 0°C, the volumetric performance of CO_2 is for example, four to twelve times better than that of ammonia, which allows compressors with smaller swept volumes to be used.

DRAWBACKS OF CARBON DIOXIDE

The main drawback of carbon dioxide as a refrigerant is its inherent high working pressure. This pressure is much higher than that of the other natural and synthetic refrigerants mentioned.



On one hand, this means that for CO_2 cycles, components must be redesigned. Since CO_2 offers a much higher volumetric capacity, the problem of the higher working pressure can be overcome by optimal design involving smaller, stronger components.

During standstill, the ambient temperature can reach and exceed the critical temperature and the pressure can exceed the critical pressure. Hence systems are typically designed to withstand pressures up to 90 bar, or sometimes even equipped with a small standstill condensing unit to keep pressures low.

The triple point and critical point of CO_2 are very close to the working range. The critical point may be reached during normal system operation. During system service, the triple point may be reached, as indicated by the formation of dry ice when liquid containing parts of the systems are exposed to atmospheric pressure. Special procedures are necessary to prevent the formation of dry ice during service venting.

Carbon dioxide can change from a gas directly to a solid (triple point) at 4.2 bar therefore during commissioning care is required when charging the system with refrigerant. Also the design of PRV vent lines must allow for the possible freezing of carbon dioxide as it approaches atmospheric pressure.

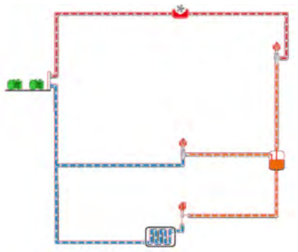
CO_2 is very sensitive to water contamination and can form unusual compounds if there is a leak or water ingress. It is also important to ensure that no moisture is allowed into the system when charging with refrigerant or oil.

Furthermore, CO_2 is odourless, heavier than air and is an asphyxiant. The practical limit of CO_2 is lower than hydrofluorocarbons (HFCs) because of its potential for high toxicity … *HFCs are non-toxic.*

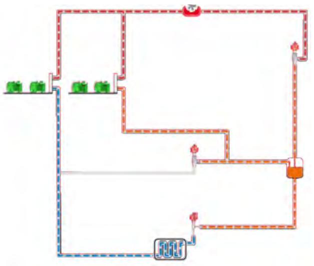
Practical limit of CO_2 is 0.1 kg/m^3 (56,000 ppm), whilst the limit for R404A: 0.48 kg/m^3 (120,000 ppm)

ALTERNATIVE CO2 SYSTEMS

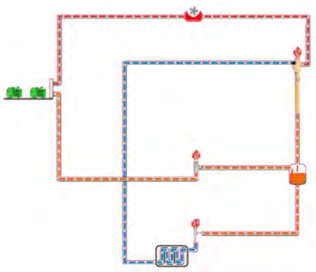
- Traditional single valve direct expansion cycle: in this configuration, the expansion valve needs to keep both superheat and gas cooler pressure under control
    - Under transcritical conditions, for a given gas cooler outlet temperature, an optimal working pressure can be determined that gives the best possible COP. Advanced controllers are required to keep both gas cooler pressure and superheat under control
    - Usually adiabatic systems and/or mechanical subcooling are adopted to improve the system efficiency, but the result is generally a significant performance gap against HFC/HFO applications
- Single valve direct expansion cycle with COP optimisation: in this configuration, the system provides a good compromise between optimal gas cooler pressure and superheat control
- Three valve configuration with intermediate liquid receiver: by decoupling the high pressure side from the low pressure side, gas cooler pressure and superheat can be optimised separately, each with its own valve. A third valve is necessary to control the pressure inside the liquid receiver. This system is perfectly suitable for also working in subcritical conditions: in this case, the flash gas valve will work practically closed and the high pressure valve will basically control subcooling at the condenser



- Parallel compression: flash gas is refrigerant that does not take part in the cooling process. In order to further improve system efficiency, especially in warmer climates where the amount of flash gas may be considerable, parallel compressors can be utilised replacing the flash gas valve when its relative opening is quite high. Parallel compressors reduce the flow rate handled by the main compressors and work in much more efficient conditions, as they have higher suction pressures (receiver pressure rather than MT evaporator pressure)



- Ejector systems: used to improve the efficiency of CO_2 transcritical cycle. Ejectors are devices engineered to entrain a low pressure fluid (suction) using a high pressure fluid (motive). This is possible without any external power consumption (electrical/mechanical) but is achieved simply using the physical principle of the "Venturi" effect. In ejector systems, ejectors replace the HPV valve and keeping the gas cooler outlet pressure in optimal conditions (max COP). MT compressors are used to keep the pressure inside the receiver under control, while the work to lift the pressure from the MT evaporator to receiver value is achieved by the ejector

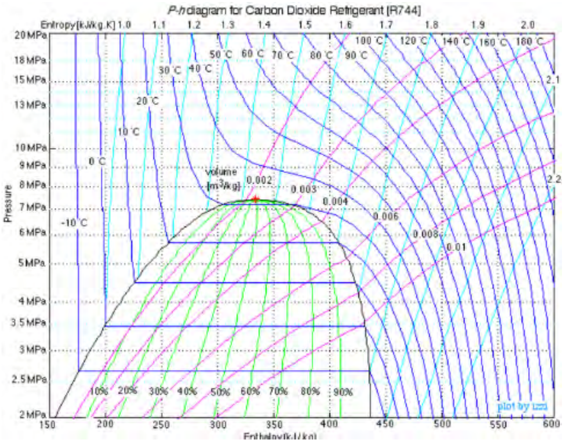


  There are considerable gains in terms of efficiency which depend upon three main factors …
    - The lift effect is obtained without any external power
    - MT compressors can operate with a lower DP because their suction pressure is now the receiver pressure value and no longer the MT unit pressure
    - As MT suction is no longer the MT evaporator pressure, the MT unit superheat can be reduced, thus increasing the evaporator working pressure and consequently overall system efficiency

Gas coolers perform a similar role to condensers i.e. reject the heat from the high pressure side of the refrigeration cycle, but because this process takes place above the Critical Point, the behaviour of the carbon dioxide exhibits some unusual characteristics.

BEHAVIOUR NEAR THE CRITICAL POINT

The critical point (*see Critical point*) is the condition at which vapour, and liquid coexist; in other words, it is the vertex of the saturation curve, where there is a change of phase from vapour to liquid and vice versa.



The critical point is characterised by a temperature & pressure i.e. critical temperature (t_c) and critical pressure (P_c). This definition is generic and all fluids have a critical point. For CO_2 the critical point is defined as t_c = 30.98°C (~31°C) and p_c = 73.77 barA (~73 barg).

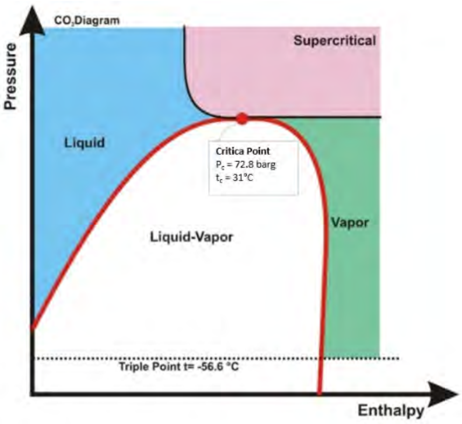
Note : Although the refrigeration industry tends to refer to pressures in barg, much of the published data for CO_2 refers to absolute pressures such as bar or barA, kPa or MPa.

Clearly, this low critical temperature, which is close to typical summer ambient temperatures in a European climate, implies the inability to reject heat from a refrigerating cycle using a conventional condenser.

Obviously, at typical condensing temperatures of 35°C to 40°C, the CO_2 is above the critical point and is referred to as Supercritical and thus already exists as a gas and cannot be condensed, only cooled.

During the expansion process the carbon dioxide will drop in pressure to something below the critical point and this mode of operation that straddles the critical point is known as a ‘transcritical cycle’.

The figure shows the thermodynamic states of carbon dioxide (CO_2 - R744), plotted on the p-h diagram. The saturation curve is plotted in red, under this curve - the *vapour dome* - liquid and vapour coexist; on the left side (blue region) of the curve, only liquid is present and on the right side (green region), only vapour exists. Above the critical point, there is a region characterised by a thermodynamic state named Supercritical (pink region) where the fluid is in vapour/fluid hybrid phase.



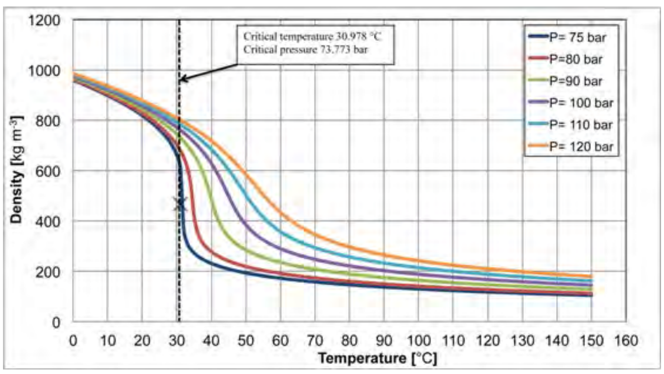
During the operation of a CO_2 transcritical cycle, operation is close to the critical zone where the thermophysical properties of the fluid exhibit sudden changes. This characteristic can greatly affect the design of a suitable heat exchanger (*Gas cooler*), where the supercritical CO_2 from the compressor is cooled down before it reaches the expansion device.

The thermophysical properties that affect the heat transfer characteristics of a refrigerant are .. the specific heat capacity, thermal conductivity, dynamic viscosity and the density.

The first three properties relate to the Prandtl number, a dimensionless parameter, which is used in the correlations for the calculation of the heat transfer coefficient.

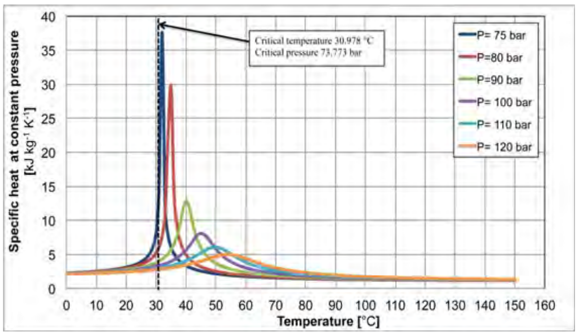
The density exhibits some unusual characteristics (see below). At the critical point, the liquid and vapour densities have the same value of 467.6 kg/m^3 and the variation of the density close to the critical point is rather radical.

The figure shows the density as a function of the temperature with increasing pressure from 75 barA (close to the critical pressure) through to 120 barA.

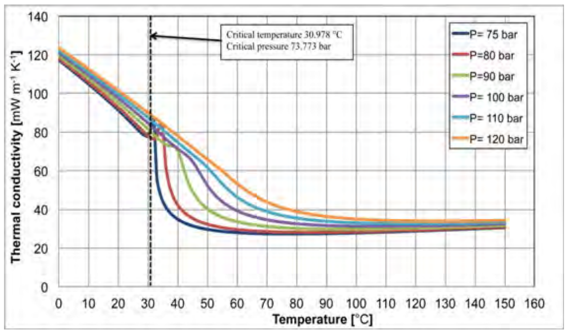


However, the density drops from approx. 700 kg/m^3 to 300 kg/m^3 with only an 8K rise in temperature i.e. at 40°C. Furthermore, as the pressure increases, the density variation lowers and for pressures greater than 100 barA, it becomes more consistent.

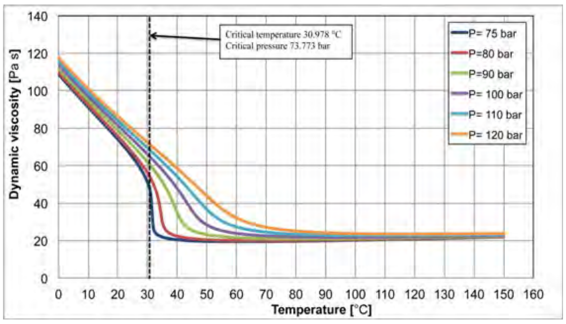
The following figures show the specific heat, thermal conductivity & viscosity behaviour with temperature …



Specific heat at constant pressure



Thermal conductivity



Dynamic viscosity

The above figure showing the specific heat at constant pressure presents several peaks at different temperatures as a function of the pressure. The peak temperature is called 'pseudo-critical temperature'. As the pressure increases the specific heat decreases and its peak temperature increases. Furthermore, as the temperature increases, the specific heat at constant pressure reduces until about 80°C, thereafter it is only marginally affected by the pressure.

This high gradient of the specific heat must be considered during a Gas cooler design because the heat flow rate Q is proportional to the CO_2 mass flow rate, the specific heat and to the temperature difference as given by the fundamental equation … $\mathbf{Q = \dot{m} \times C_p \times \Delta t}$

Considering a Gas cooler where the carbon dioxide is cooled down from say 120°C to 25°C at constant pressure of P = 80 barA (red line in above figures).

The specific heat at constant pressure can be considered constant whilst cooling down to around 100 °C. In this region the CO_2 rejects the same amount of heat when its temperature decreases by 1K, but after that, the C_p starts to rise, slowly at first, then sharply.

At 100°C the C_p = 1.33 kJ/kg/K but at 50°C, C_p = 2.51 kJ/kg/K and it reaches a maximum value of 29.59 kJ/kg/K at 35°C. Thereafter it decreases sharply down to a level of C_p = 2.97 kJ/kg/K at 20°C.

Thus, in this cooling region the heat flux changes dramatically for each 1K temperature variation. Firstly, it rises sharply while the gas is cooled from 100°C to 35°C and then drops dramatically as the temperature reduces still further to 20°C. This unusual behaviour must be taken into account during the Gas cooler design.

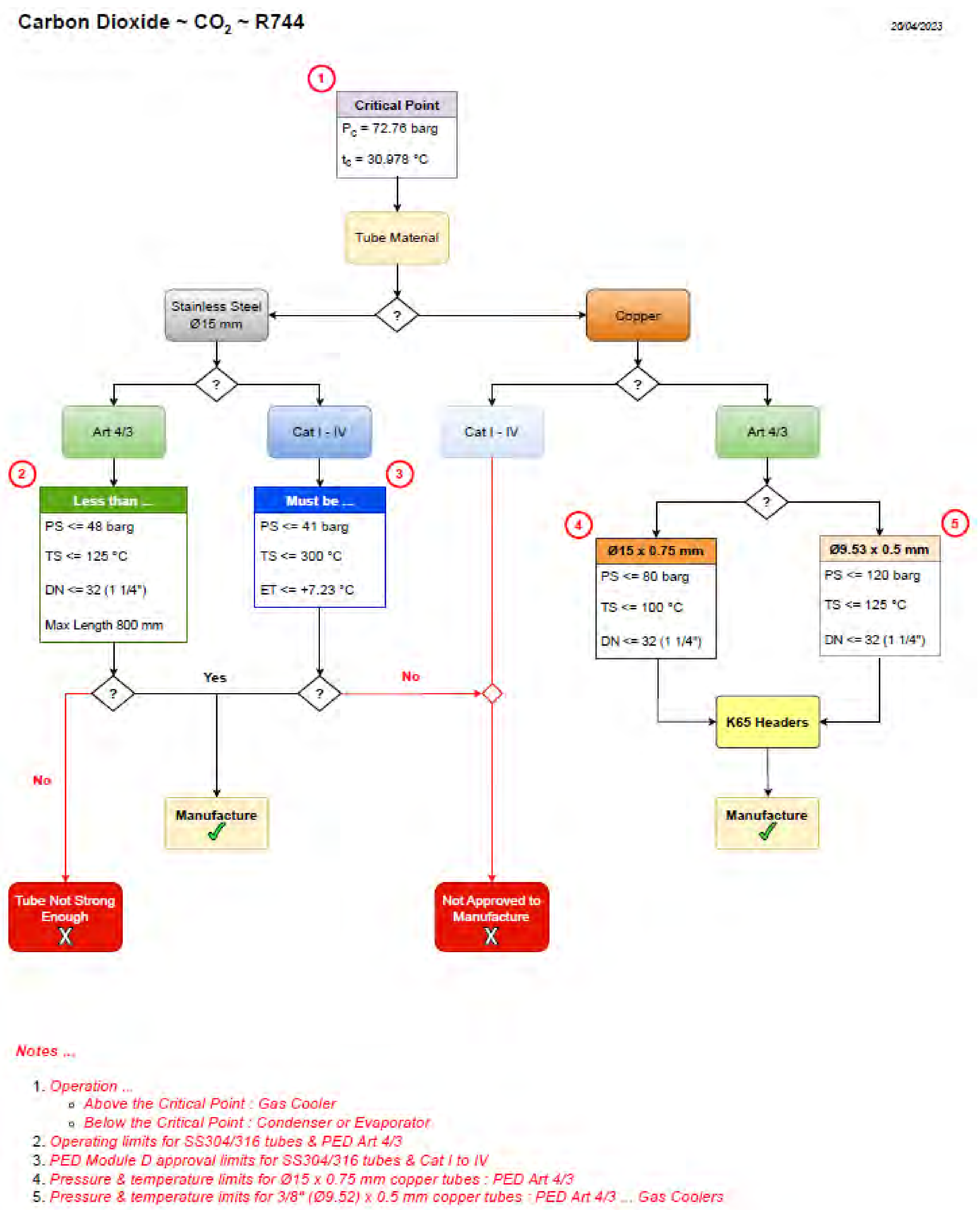
The thermal conductivity diagram above also shows some peaks, which are noticeable for pressures below 100 barA, but with a less significant intensity than those exhibited by the specific heat properties. As the pressure increases, the thermal conductivity variations become smaller, and the effects of the critical region completely disappear.

In the last diagram, the dynamic viscosity is plotted as a function of the temperature and pressure. In this case, there are no peaks but for pressures close to the critical point there are sudden variations in the dynamic viscosity showing a similar behaviour as the density properties.

From the above it is clear that it is not possible to consider the thermophysical properties close to the critical point as constant. The sudden variations resulting in peak values can lead to unmanageable behaviour during the transcritical heat exchanger design.

Heat exchangers i.e. gas coolers designed for super-critical operation must have their selections checked for appropriate functionality, especially if they are expected to operate sub-critically i.e. as a conventional condenser, often during night-time load conditions !

MANUFACTURING DECISION TREE



COIL CALCULATION PHILOSOPHY

BASIC EQUATIONS

The basic equations governing the design of all heat exchangers are rather straight forward and related to the evaluation of unknown information derived from given information. This infers that providing there is only one unknown parameter in an equation it can be calculated from the known data.

There are times where substitution of equations within equations is required to obtain the unknowns but in view of the simple nature of the relationships, this is straightforward. Once all the individual items of data are known the surface requirements of a coil can be determined.

This philosophy can be inverted to ascertain the performance of a given coil. However this calculation involves performing an iteration based upon a trial and error convergence methodology.

The only difference between fluid, refrigerant and steam coils in this analysis is the magnitude and calculation of the internal heat transfer coefficient.

In the case of a single phase fluid, if the thermophysical properties of the fluid are known at the mean fluid temperature, the application of the Nusselt correlation will give the heat transfer coefficient.

Refrigerant heat transfer coefficients are evaluated from established formulae that relates the heat transfer coefficient to the refrigerant mass flow rate and quality.

In the case of steam condensing coils, again industry accepted empirical data is used which for 5/8” tubes is assumed to be **8500** W/m2/K.

However, where steam pressures exceed 3 barg and the coil air leaving temperature begins to approach the steam temperature, correction factors are applied to derate the condensing performance.

Apart from the above, most of the following relationships apply to all coil types …

AIR MASS FLOW RATE

AirMassFlow = **AirVolume** x **AirDensity**

AIRSIDE SENSIBLE DUTY

$\mathbf{Q_s}$ = **AirMassFlow** x **AirSpecHeat** x *ABS*(**AirInletDB** – **AirOutletDB**)

AIRSIDE TOTAL DUTY

$\mathbf{Q_t}$ = **AirMassFlow** x *ABS*(**AirInletEnthalpy** – **AirOutletEnthalpy**)

FLUID MASS FLOW RATE

FluidMassFlow = **FluidVolFlowRate** x **FluidDensity**

FLUID-SIDE TOTAL DUTY

$\mathbf{Q_w}$ = **FluidMassFlow** x **FluidSpecHeat** x *ABS*(**FluidInlet** – **FluidOutlet**)

If the coil is to operate in thermal equilibrium, then $Q_w = Q_t$, i.e. the airside duty must equal the fluid-side duty. If this is not the case, then the coil is in a transient state and an accurate calculation cannot be performed.

However, we always consider the coil to be in equilibrium and thus from the above equations, if certain data in not given it can be calculated by manipulating the equations to solve for the unknowns.

REFRIGERANT TOTAL DUTY

$\mathbf{Q_r}$ = **RefrigMassFlow** x *ABS*(**RefrigInletEnthalpy** - **RefrigOutletEnthalpy**)

Currently the design software does not allow either the refrigerant mass flow rate or enthalpies to be entered to calculate the DX evaporator or CD condenser duty. However, the duty is either entered or calculated from the airside conditions in the form of Q_s or Q_t.

In the case of a DX coil, the system's condensing temperature and operating sub-cooling determines the liquid temperature entering the TEV, which in turn, dictates the entry conditions for the evaporating process. Finally, the evaporating temperature and operating superheat should be specified.

Alternatively, for a CD coil, the hot gas inlet temperature and condensing temperature plus the desired subcooling should be defined.

These additional parameters are needed to ascertain the operating pressures and temperatures to allow the inlet and outlet enthalpies to be evaluated. Once the enthalpy difference is known, the refrigerant mass flow rate can be evaluated, which allows the refrigerant mass velocity to be calculated. Following certain other estimations relating to the anticipated refrigerant flow regime, the internal heat transfer coefficient can be derived.

Since the demise of the 'old' traditional R12, R22 & R502, there have been a raft of replacement refrigerants designed and offered to the marketplace. These new refrigerants: which tend to be binary or tertiary blends of azeotropic refrigerants; can exhibit a phenomenon known as 'temperature glide' which for some blends, can exceed 5K.

On first sight temperature glide could be thought to be a detrimental characteristic, but if the coil circuitry is designed with this in mind, then this characteristic can be a benefit.

SENSIBLE HEAT RATIO (SHR)

$$\mathbf{SHR} = \mathbf{Q_s} / \mathbf{Q_t}$$

This characteristic defines the operating process of the coil and can be plotted on a psychrometric chart to show the nature of the cooling process. For all heating coil applications, SHR =1.0, but if cooling is performed then the SHR can be < 1.0

This characteristic defines how 'wet' the coil is operating as defined by the 'sensible to total' heat transfer relationship. This parameter directly affects the thermal heat transfer properties of the 'wet' extended surface of the coil. The lower the SHR the more the latent cooling affects the coil surface requirements.

Theoretically, this keeps on improving thus requiring less and less surface, but in practice thermal / latent saturation is reached which does not allow further improvement in performance.

To control this, a 'cut-off' is invoked if the SHR < 0.45. It is assumed that no further enhancement is exhibited when SHR < 0.45 and the enhancement factor remains constant and equals the value at SHR = 0.45

TOTAL SURFACE AREA

$$\mathbf{A_t} = \mathbf{A_o} \times \mathbf{FaceArea} \times \mathbf{RowsDeep}$$

A_t is defined as the total surface area of the coil that is in contact with the air stream. Traditionally, this is often referred to as the sum of the primary and secondary surface areas.

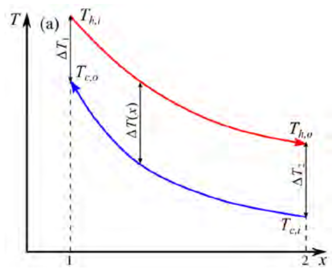
However, when plate fin heat exchangers are considered that utilise full collar fin forms, there is no primary (tube) surface directly in contact with the air and thus the 'real' secondary surface becomes the total surface area.

LOG MEAN TEMPERATURE DIFFERENCE - LMTD

Tube and fin heat exchangers exhibit a combination of both counter flow and cross flow, however the traditional method for calculation involves the use of the LMTD, *logarithmic mean temperature difference*.

Simplistic theory using the MTD, *mean temperature difference* would be reasonable if the tube side and air side temperature profiles were linear. In such a case, the MTD is the TD between the average fluid temperatures and average air temperatures.

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$



As the temperature profiles are not linear, using the LMTD is a better approximation to 'reality' by imposing an exponential relationship upon the temperature differences at both ends of the temperature profile.

However, narrow coils such as 1 or perhaps 2 row deep, cannot be circuitry in a generally counter flow fashion and in such cases, using MTD in place of the LMTD is a more realistic solution.

LMTD CORRECTION FACTOR

Traditional heat exchanger design manuals provide multiple charts related to dimensionless parameters to provide a correction factor to reduce the theoretical LMTD to account for the number of passes on the tube side, resulting in more surface area necessary to meet the thermal load.

EAS uses the following correlation to modify the LMTD …

$\mathbf{F = 0.9^{[1/x]}}$ *where **x** is the number of passes or tubes/circuit*

This translates into a factor of 0.9 when there is only 1 tube in a circuit and thus a 'close to' pure cross flow arrangement. The more the tubes per circuit of passes, the closer the factor approaches 1.0

EXTERNAL HEAT TRANSFER H_O COEFFICIENT

The external heat transfer coefficient refers to the 'airside' dry coefficient and is affected by parameters such as the tube & fin geometry, fin efficiency, air velocity through the tube bundle and temperature.

During 2016 a new method to derive a generic h_o algorithm was deemed necessary and it was decided to use an adaption of a German VDI methodology for staggered banks of continuous finned tubes, which derived the airside Nusselt Number from the Hydraulic diameter, which in turn was calculated in a novel fashion.

Traditional Fin Efficiency (Schmidt) and Surface Effectiveness equations were adopted, however, the surface associated with a single tube within a staggered tube bundle was approximated, again using Schmidt's hexagonal fin area approximation for staggered banks of tubes.

Provision in the equations allows for the difference between the 'projected' fin surface area compared with the 'true' surface area when the extended surface invokes a waffle or wavy profile. This impact upon the net results is only marginal and to begin with, this parameter was set to 1.0

Traditional air pressure correlations suggest a 'square law' relationship and AMCA wind tunnel test have confirmed that for EAS's C-fin (P40), the relationship more closely follows an index = 1.63 rather than 2.0. The correlations were adjusted accordingly providing a closer relationship with the test measurements.

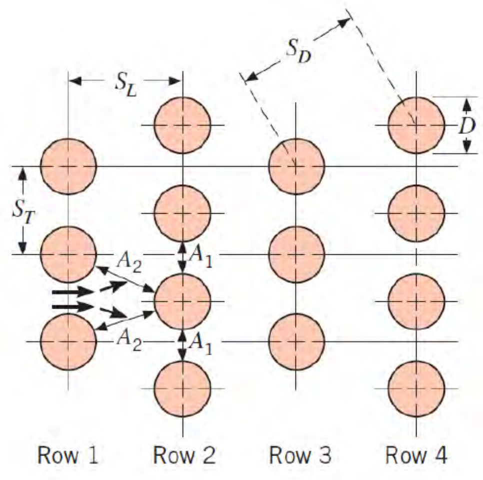
C-fin (P40) : 40 x 34.64 - Ø15 mm

The figure to the right shows a typical staggered tube pattern indicating the minimum flow area (A_1) and thus the location of the highest interstitial velocity through the tube bundle.

Although A_2 is physically the same as A_1, for this geometry, A_1 is subject to the full portion of the air volume, whilst A_2 is only subject to 50% of the air volume as the air stream divides to pass over the tubes in the subsequent tube bank.

In this figure, D denotes a diameter and for the purpose of this algorithm, is the outside diameter of the fin collar, which is the expanded tube outside diameter plus 2 x fin thickness … assuming there has been no thinning of the fin material during the expansion process.

S_T, S_L relate to variables TubePitch and TubeStagger and S_D (DiagonalPitch) is a temporary variable used within the Fin Efficiency function.



Staggered

Geometry related data

Variable	Units	Description
Pattern	-	1 : Staggered, 2 : In-line
TubePitch	mm	Tube pitch normal to air flow
TubeStagger	mm	Tube row pitch in direction of airflow
Ratio	-	True/Projected fin area Currently : 1.0
ExpTubeOD	mm	Expanded tube OD
TubeWall	mm	Tube wall thickness

Thermophysical data

Variable	Units	Description
TubeThermCond	W/m/K	Tube material thermal conductivity Alum : 205 Copper : 385 SS304/316 : 16.2
FinThermCond	W/m/K	Fin material thermal conductivity Alum : 205 Copper : 385 SS304/316 : 16.2
AirDensity	kg/m^3	Air density
AirSpecHeat	J/kg/K	Air specific heat
AirThermCond	W/m/K	Air thermal conductivity
AirViscosity	mPa.s	Air dynamic viscosity
Prandtl Num	-	AirSpecHeat x AirViscosity / 1000 / AirThermCond
ReynoldsNum	-	AirDensity x InterstitialVel x HydraulicDiameter / AirViscosity x 1000

Calculated data

Variable	Units	Description
TubeID	mm	TubeOD - 2 x WallThk
TubesHigh	-	Number of tubes in the Finned Height
FaceArea	m2	Face height x length
FaceVel	m/s	Frontal velocity
Ai	m2/m2/row	Internal tube surface area
Ap	m2/m2/row	Primary fin surface area
As	m2/m2/row	Secondary fin surface area
Ao	m2/m2/row	Total fin surface area as seen by air stream (Ap + As)
Am	m2/m2/row	Mean tube wall surface area
TubeWallRes	m2.K/W	Tube material wall resistance
IntFouling	m2.K/W	Internal fouling factor
ExtFouling	m2.K/W	External fouling factor
UValue	W/m2/K	Overall heat transfer coefficient
BasicAirHo	W/m2/K	Basic airside Ho value
FinEffy	-	Fin efficiency
SurfaceEffy	-	Surface Efficiency
HoValue	W/m2/K	Airside Ho value
BasicDryAirPD	Pa	Airside pressure drop

Adjustable coefficients

Variable	Units	Description
NusseltCoeff	-	Adjustment coefficient for Nusselt Number Currently : 0.29 *Was : 0.31*
FrictionFactorCoeff	-	Adjustment coefficient for airside friction factor Currently : 21.0 *Was : 10.5*
AirPressDropIndex	-	Adjustment index for air pressure drop Currently : 1.63 *Was : 2.0*

OBJECT PASCAL PROCEDURES

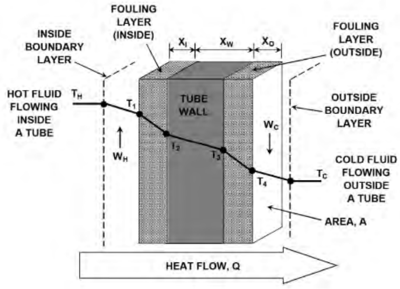
```
procedure SetTubeID(const ExpTubeOD, TubeWall: double);
begin
 TubeID := ExpTubeOD - 2 * TubeWall;  // mm
end;

procedure SetTubesHigh(const FinHeight, TubePitch: double);
begin
 TubesHigh := int(FinHeight / TubePitch + 0.01);
end;

procedure SetFaceArea(const FinHeight, FinLength: double);
begin
 FaceArea := FinHeight * FinLength * 1E-6; // m2
end;

procedure SetFaceVelocity(const AirVol, FaceArea: double);
begin
 FaceVelocity := AirVol / FaceArea; // m/sec
end;

procedure SetAi(const TubeID, TubePitch: double);
begin
 // Internal tube surface area per m2 face area per row
 Ai := Pi * TubeID / TubePitch; // m2/m2/row
end;

procedure SetAp(const ExpTubeOD, TubePitch, FinPitch, FinThk: double);
begin
 // Primary surface area per m2 face area per row
 Ap := Pi * (ExpTubeOD + 2 * FinThk) * (1 - FinThk / FinPitch) / TubePitch; // m2/m2/row
end;

procedure SetAs(const Ratio, ExpTubeOD, TubePitch, TubeStagger, TubePitch, FinPitch, FinThk: double);
begin
 // Secondary fin surface area per m2 face area per row
 As := 2 * (Ratio * TubePitch * TubeStagger - Pi /4 * Sqr(ExpTubeOD + 2 * FinThk)) / FinPitch / TubePitch;  // m2
end;

procedure SetAo(const Ap, As: double);
begin
 // Total fin surface area (as seen by air stream) per m2 face area per row
 Ao := Ap + As;  // m2/m2/row
end;

procedure SetAm(const ExpTubeOD, TubeID, TubePitch: double);
begin
```

```
  // Mean tube surface area per m2 face area per row
  Am := Pi * (ExpTubeOD + TubeID) / 2 / TubePitch;  // m2/m2/row
end;

procedure SetTubeWallRes(const TubeWall, TubeThermCond, Ao, Am: double);
begin
  // Tube material wall resistance
  TubeWallRes := TubeWall / 1000 / TubeThermCond * Ao / Am;  // m2.K/W
end;

procedure SetUValue(const Ao, Ai, HiValue, HoValue, TubeWallRes, IntFouling, ExtFouling: double);
begin
  // Overall heat transfer coefficient
  UValue := 1 / ((Ao / Ai * (1 / HiValue + IntFouling)) + 1 / HoValue + TubeWallRes + ExtFouling) ;  // W/m2/K
end;

procedure SetBasicAirHo(const Pattern: integer;
                              Ratio, Ao, AirVol, FaceAea, ExpTubeOD, TubePitch, TubeStagger, Rows, FinLength, FinPitch,
                              FinThk, AirDensity, AirSpecHeat, AirThermCond, AirViscosity : double);
var
  Phi, AirsideArea, EffectiveVolume, HydraulicDiameter, ReynoldsNum, InterstitialVel,
  NusseltNum, PrandtlNum: double;
begin
  AirsideArea := Ao * FinHeight * FinLength * Rows *1E-6;
  EffectiveVolume := TubesHigh * TubePitch * TubeStagger * Rows * FinLength * 1E-9;
  Phi := 1 - FinThk / FinPitch - Pi * sqr(ExpTubeOD + 2 * FinThk) / 4 * (FinPitch - FinThk) /
         (TubePitch * TubeStagger * FinPitch);
  HydraulicDiameter := 4 * EffectiveVolume * Phi / AirsideArea;
  InterstitialVel := AirVol / Phi / FaceArea;
  ReynoldsNum := InterstitialVel * HydraulicDiameter * AirDensity / AirViscosity * 1000;
  PrandtlNum := AirSpecHeat *  AirViscosity / AirThermCond / 1000;

  if Pattern = 1 then
    NusseltNum := NusseltCoeff * Power(ReynoldsNum, 0.625) * Power(PrandtlNum, 0.333) *
                  Power((HydraulicDiameter / TubeStagger * 1000), 0.333)
  else
    NusseltNum := 0.21 * Power(ReynoldsNum, 0.625) * Power(PrandtlNum, 0.333) *
                  Power((HydraulicDiameter / TubeStagger * 1000), 0.333);

  BasicAirHo := NusseltNum * AirThermCond / HydraulicDiameter;
end;

procedure SetBasicDryAirPD(const Pattern: integer;
                              Ratio, Ao, AirVol, FaceArea, ExpTubeOD, TubePitch, TubeStagger, Rows,
                           FinLength, FinPitch, FinThk, AirDensity, AirViscosity: double);
var
  Phi, AirsideArea, EffectiveVolume, HydraulicDiameter, ReynoldsNum, InterstitialVel, FrictionFactor: double;
begin
  AirsideArea := Ao * FinHeight * FinLength * Rows *1E-6;
  EffectiveVolume := TubesHigh * TubePitch * TubeStagger * Rows * FinLength * 1E-9;
  Phi := 1 - FinThk / FinPitch - Pi * sqr(ExpTubeOD + 2 * FinThk) / 4 * (FinPitch - FinThk) /
          (TubePitch * TubeStagger * FinPitch);
  HydraulicDiameter := 4 * EffectiveVolume * Phi / AirsideArea;
  InterstitialVel := AirVol / Phi / FaceArea;
  ReynoldsNum := InterstitialVel * HydraulicDiameter * AirDensity / AirViscosity * 1000;

  if Pattern = 1 then
    FrictionFactor := FrictionFactorCoeff * Power(ReynoldsNum, -0.333) *
                      Power((HydraulicDiameter / TubeStagger * 1000), 0.6)
  else
    FrictionFactor := 6 * Power(ReynoldsNum, -0.333) * Power((HydraulicDiameter / TubeStagger * 1000), 0.6);
```

```
  BasicDryAirPD := Frictionfactor * TubeStagger / 1000 * Rows / HydraulicDiameter * AirDensity / 2 *
                   Power(InterstitialVel, AirPressDropIndex);
end;

procedure SetFinEffy(const Pattern: integer; ExpTubeOD, TubePitch, TubeStagger, FinThk,
                         FinThermCond: double);
Var DiagonalPitch, Phi, Beta, X: double;
begin
  DiagonalPitch := sqrt(sqr(TubePitch / 2) + sqr(TubeStagger));

  if Pattern = 1 then
  begin
    if TubeStagger < (TubePitch / 2) then
      X := 2 * TubeStagger
    else
      X := TubePitch;
  end;

   Phi := 1.27 * X / (ExpTubeOD + 2 * FinThk) * sqrt(DiagonalPitch / TubePitch - 0.3)
  else
  begin
   if TubeStagger < TubePitch then
    Phi := 1.28 * TubeStagger / (ExpTubeOD + 2 * FinThk) * sqrt(TubePitch / TubeStagger - 0.2)
   else
    Phi := 1.28 * TubePitch / (ExpTubeOD + 2 * FinThk) * sqrt(TubeStagger / TubePitch - 0.2);
  end;

  Beta := (ExpTubeOD + 2 * FinThk) / 1000 / 2 * (Phi -1) * (1 + 0.35 * Ln(Phi));
  X := sqrt(2 * FBasicAirHo / FinThk * 1000 / FinThermCond) * Beta;
  FinEffy := Tanh(X) / X;
end;

procedure SetSurfaceEffy(const SecondaryArea, TotalArea, FinEffy: double);
begin
  SurfaceEffy := 1 – As / Ao * (1 – FinEffy);
end;

procedure SetHoValue(const BasicAirHo, SurfaceEffy: double);
begin
  HoValue := BasicAirHo * SurfaceEffy;
end;
```

FOULING FACTORS

The fouling factors to be used in the design of heat exchangers are normally specified by the client based on their experience of running the plant or process. If uncontrolled, levels of fouling can negate any benefits produced by careful heat exchanger design.



The fouling factor represents the theoretical resistance to heat flow due to a build-up of a layer of dirt or other fouling substance on the tube and extended surfaces of the heat exchanger, but they are often overstated by the end user in an attempt to minimise the frequency of cleaning. If the wrong fouling factor is used, cleaning may actually be required more frequently.

Fouling mechanisms vary with the application but can be broadly classified into four common and readily identifiable types.

TYPES OF FOULING

- **Chemical fouling** : When chemical changes within the fluid cause a fouling layer to be deposited onto the tube surface. A common example of this phenomenon is scaling in a kettle or boiler caused by “hardness” minerals depositing onto the heating elements as the solubility of the salts reduce with increasing temperature. This is outside the control of the heat exchanger designer but can be minimised by careful control of the tube wall temperature in contact with the fluid. When this type of fouling occurs it must be removed by either chemical treatment or mechanical descaling processes (wire brushes or even drills to remove the scale or sometimes high-pressure water jets).
- **Biological fouling :** Caused by the growth of organisms within the fluid which deposit out onto the surfaces of the heat exchanger. Again this is outside the direct control of the heat exchanger designer, but it can be influenced by the choice of materials as some, notably the non-ferrous brasses, are poisonous to some organisms. When this type of fouling occurs it is normally removed by either chemical treatment or mechanical brushing processes.
- **Deposition fouling** : When particles contained within the fluid settle out onto the surface when the fluid velocity falls below a critical level. To a large extent this is within the control of the heat exchanger designer, as the critical velocity for any fluid/particle combination can be calculated to allow a design to be developed with minimum velocity levels higher than the critical level. Mounting the heat exchanger vertically can also minimise the effect as gravity would tend to pull the particles out of the heat exchanger away from the heat transfer surface even at low velocity levels. When this type of fouling occurs it is normally removed by mechanical brushing processes.
- **Corrosion fouling** : When a layer of corrosion products build up on the surfaces of the tube forming an extra layer of, usually, high thermal resistance material. By careful choice of materials of construction the effects can be minimised as a wide range of corrosion resistant materials based on stainless steel and other nickel-based alloys are now available to the heat exchanger manufacturer.

TYPICAL FOULING FACTORS

Fouling factors, sometimes known as fouling resistances, are usually presented in the units *$m^2.K/W$*, which is the reciprocal of the units for heat transfer coefficients and added into the calculation of the overall heat transfer coefficient described below. Some examples of typical fouling factors are …

Fluid	*$m^2.K/W$*
Seawater	0.00009
Boiler feed water	0.0002
Glycol solutions	0.00035
Gasoline	0.00035
Heavy fuel oil	0.0009
Hydraulic oil	0.00018
Refrigerant liquids	0.00018

INTERNAL FOULING FACTOR

When the internal surface of a tube collects debris or deposits such as scale etc., this insulating layer reduces the ability for heat to transfer from the internally flow fluid through the tube wall and into the fins and thus onward into the airstream. The net result is a reduction in thermal performance.

This form of Internal fouling is denoted by, $\mathbf{f_i}$ [$m^2.K/W$]

EXTERNAL FOULING FACTOR

In much the same fashion as Internal fouling, corrosion, debris etc. build-up on the finned surface of a coil will impede heat transfer and impact upon the thermal performance.

This form of External fouling is denoted by, $\mathbf{f_o}$ [$m^2.K/W$]

U-VALUE : OVERALL HEAT TRANSFER COEFFICIENT

The overall heat transfer coefficient is the reciprocal summation of the thermal coefficient/barriers encountered when heat travels from the fluid inside the tubes to the air in contact with the extended finned surface or vice versa.

These thermal barriers comprise …

- Airside external heat transfer coefficient, $\mathbf{h_o}$ [$W/m^2/K$]
- Fluid side internal heat transfer coefficient, $\mathbf{h_i}$ [$W/m^2/K$]
- Tube wall resistance [$m^2.K/W$]
- Internal fouling, $\mathbf{f_i}$ [$m^2.K/W$]
- External fouling, $\mathbf{f_o}$ [$m^2.K/W$]

Furthermore, as the thermal capacity equation involves the overall surface area, $\mathbf{A_o}$ then these coefficients are modified accordingly with respect to their applicable surface areas to ensure the calculation continuity.

Typically, **U** is presented in this form …

$$1 / \mathbf{U} = (\mathbf{A_o} / \mathbf{A_i}\ [1 / \mathbf{h_i} + \mathbf{f_i}]) + (1 / \mathbf{h_o}) + (\mathbf{t} / \mathbf{k_t} \,.\, \mathbf{A_o} / \mathbf{A_m}) + \mathbf{f_o}$$

where, t *= tube wall thickness, mm*
k_t *= tube material thermal conductivity,* $W/m^2/K$
A_m *= area based upon the mean tube diameter,* m^2

U-VALUE ENHANCEMENT FACTOR

$$\mathbf{HoFactor} = (\ \mathbf{1} / \mathbf{SHR}\ )^{\mathbf{Index}}$$

This function uses HoFactor defined above which in-turn is related to the SHR and an Index that is characterized by the tube and fin geometry and held in the database.

This correction factor is applied to the overall heat transfer coefficient (UValue), but in reality, only affects the external heat transfer coefficient.

The magnitude of the Index is typically 0.7

The reason for misuse of this parameter is historical. During the mid-1980's the original versions of these programs were developed to provide compatible results to earlier design programs sourced from Italy. To 'bastardise' the program results to match the Italian results, especially for wet coils, resulted in the application of the 'wetness enhancement' to the overall heat transfer coefficient rather than just the external coefficient.

At the time this 'fix' seemed to produce acceptable results and has never been changed since.

COIL SURFACE TOTAL DUTY

$$\mathbf{Q_t} = \mathbf{UValue} \times \mathbf{HoFactor} \times \mathbf{A_t} \times \mathbf{LMTD} \times \mathbf{F}$$

Regarding the LMTD, it is assumed that all coils operate and are thus circuited for counter flow operation … *actually, cross-counter flow*. In practice, this cannot always be achieved. Furthermore, there are few occasions where parallel flow is the only option.

Similarly, there are refrigerant evaporator applications where cross-parallel flow provides benefits and allows the refrigerant temperature drop; due to the pressure drop in the circuit; to maximise the LMTD.

Coils with only 1, 2 or perhaps 3 rows and multiple tubes per circuit (passes) are unable to achieve a counter flow circuit pattern and are thus really cross flow heat exchangers. Such coils should not use a true LMTD but MTD (mean TD) based upon the mid-point fluid temperature.

Similarly, a 4 row deep coil; that in certain circumstances would be considered as a good contender for pseudo-counter flow; does not fulfil the counter flow criteria if it were designed with a large number of tubes per circuit.

MAXIMUM PERFORMANCE CALCULATIONS

This functionality enables the performance of a known coil to be determined when applied with specific entering conditions.

The full physical characteristics of the coil need to be specified or assumed and in the case of the airside data the air volume and inlet temperature must be known.

Regarding the fluid-side, the fluid inlet temperature and either the fluid outlet temperature or the fluid flow rate must be specified.

The calculation logic performs iterations based upon an assumed starting performance and from this figure calculates the air leaving and fluid outlet or flow rate, depending upon the entered data. These initial calculations allow the LMTD and internal heat transfer coefficients to be determined which allows an expected duty calculation to be performed.

The calculated duty is compared with the starting value and if this deviates by more than 0.05%, the duties are binary split to arrive at a new starting condition. This process is continued until convergence is achieved within the specified limits.

During the iteration process, tests are conducted to ensure thermodynamic feasibility and checking for 'thermodynamic saturation', which can be reached when the surface available exceeds the duty achievable.

The final duty arrived at for this given coil should equal the surface calculated i.e. the whole number of rows indicated, when the normal surface / rows calculation is performed as previously described.

The exception to this is when 'thermal saturation' is reached and then the surface (rows) entered may greatly exceed the actual surface necessary. This scenario can often arise when the fluid inlet temperature and flow rate are specified and thus the fluid outlet temperature 'floats' as the iterated duty varies.

Clearly, the calculated fluid outlet can only approach the associated air inlet temperature to within, say 2.5K, and when this condition is reached, no matter how many rows or surface is available, the coil cannot generate a larger capacity.

CHECKS & LIMITATIONS

During the calculation procedure of either of the above methodologies, certain checks need to be performed to ensure that thermodynamic continuity is achieved as well as not exceeding physical or practical limitations which could lead to the coil under-performing or indeed, not working at all.

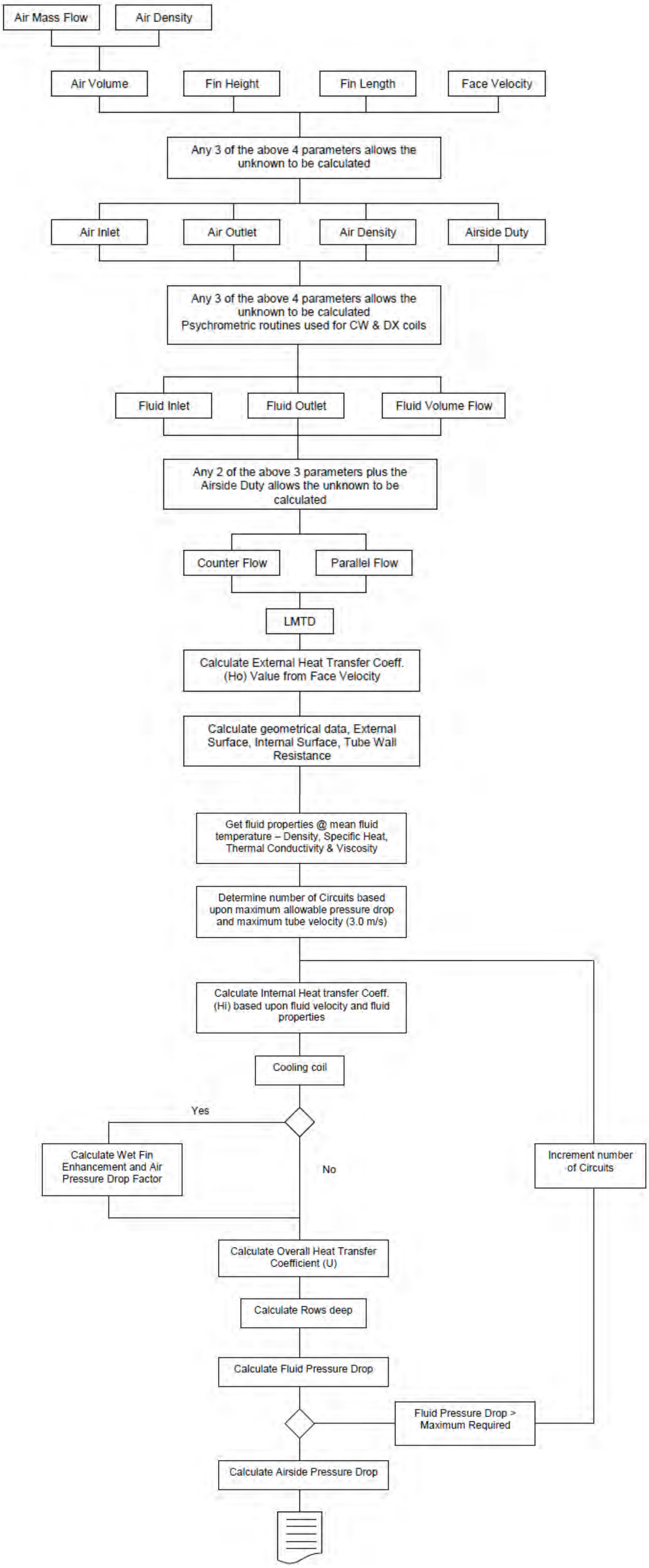
Below are details of some of the more important considerations which must be considered.

- The maximum tube length allowable associated with the expansion method and tube material
- The maximum coil height and ability to manufacture as one section or the need to split into vertically stacked sections
- The maximum rows deep per casework section and relationship to casework construction and sheet metal thickness
- Inclusion of coil divider plates when limiting finned length is exceeded
- Suitability of tube, fin and casework material for proposed application
- Material strength suitability at operating temperatures
    - Aluminium fins < 150°C
    - Copper fins < 200°C
    - Titanium tubes <150 °C
- Face velocity limits
    - Greater than 0.5 m/s
    - Less than 5.0 m/s
- Moist air temperature limits – maximum of 90°C saturated
- Psychrometric routine suitability for pressures other than atmospheric pressure

- Validity check of air and fluid temperatures in relation to application and parallel or counter flow circuitry design.
  The following relates to counter flow only …
    - Heating – air entering must be at a lower temperature than the air leaving
    - Heating – fluid inlet must be above air leaving and fluid outlet must be above air entering temperature
    - Cooling – air entering must be above the air leaving temperature
    - Cooling – fluid inlet must be below the air leaving and fluid outlet must be below the air entering temperature
    - Cooling – the air leaving dew point must be less than or equal to the air entering dew point
- None of the entered or calculated temperatures should breach the design approach temperature difference – currently 2K
- Liquid tube velocities for copper should not exceed 2.0 m/s unless stainless or titanium is used or excessive erosion (short life) is accepted
- Gas tube velocities should be limited to 50 m/s and preferably 30 m/s
- Minimum fluid velocities cannot be defined because this lower limit is governed by the fluid Reynolds Number
- Laminar flow (Re <2100) should be avoided
- Maximum fluid pressure drops are usually customer specified and the circuitry logic will adjust the circuits accordingly to achieve the pressure drop constraints
- Refrigerant pressure drops are considered within the DX & CD circuitry logic as they are directly related to the refrigerant temperature drop which is the controlling aspect of refrigerant coil design
- Circuitry and in particular the number of tubes in a circuit when compared with the rows deep have both thermal design, drainage and manufacturing implications
- Fin pitches relating to fin geometry, tube and fin materials and application must be considered
- Surface margins ought to be used when there is any uncertainty regarding the selection or application
    - Margins < 1.0 should be used with extreme care
- Fouling factors, both internal and external ought to be applied if known
- Awareness that the predicted performance is for a 'clean new' coil and thus in reality the coil is likely to underperform very quickly after installation as it starts to corrode or becomes dirty
- Coil 'fall-off' in performance under low temperature applications where frost / ice build-up is a problem should be understood and surface margins applied to maintain acceptable operation between defrosts

If appropriate consideration of the above checks and limitations are invoked within both the data entry and calculation functionality of the program, then the resulting coil should perform as predicted.

Clearly, there are instances when the above guidelines can or need to be broken, but these deviations from the preferred approach should only be invoke with full knowledge of the possible implications.


Air Mass Flow
Air Density
Air Volume
Fin Height
Fin Length
Face Velocity
Any 3 of the above 4 parameters allows the unknown to be calculated
Air Inlet
Air Outlet
Air Density
Airside Duty
Any 3 of the above 4 parameters allows the unknown to be calculated
Psychrometric routines used for CW & DX coils
Fluid Inlet
Fluid Outlet
Fluid Volume Flow
Any 2 of the above 3 parameters plus the Airside Duty allows the unknown to be calculated
Counter Flow
Parallel Flow
LMTD
Calculate External Heat Transfer Coeff. (Ho) Value from Face Velocity
Calculate geometrical data, External Surface, Internal Surface, Tube Wall Resistance
Get fluid properties @ mean fluid temperature – Density, Specific Heat, Thermal Conductivity & Viscosity
Determine number of Circuits based upon maximum allowable pressure drop and maximum tube velocity (3.0 m/s)
Calculate Internal Heat transfer Coeff. (Hi) based upon fluid velocity and fluid properties
Cooling coil
Yes
Calculate Wet Fin Enhancement and Air Pressure Drop Factor
No
Increment number of Circuits
Calculate Overall Heat Transfer Coefficient (U)
Calculate Rows deep
Calculate Fluid Pressure Drop
Fluid Pressure Drop > Maximum Required
Calculate Airside Pressure Drop

HEAT EXCHANGER WATER QUALITY GUIDE

The chemistry of the media flowing inside the tubes of a coil (heat exchanger) is critical to ensure operational longevity without undue corrosion implications.

In the case of water, municipal drinking water (city water) that is pollution free, bacteriologically safe and has a neutral pH is acceptable for all our standard tube materials, including copper.

Natural water sources such as wells, rivers or ponds must be free from pollutants and treated to reduce contaminants to the same level as municipal drinking water.

Softened, distilled or demineralised water is not generally suitable for copper tube applications because most of the minerals have been removed and there is usually a higher level of carbon dioxide and oxygen present in the water. These higher levels of carbon dioxide and oxygen can inhibit the formation of the protective copper oxide layer and any corrosion which could take place will be accelerated, thus reducing the longevity of the heat exchanger.

However, demineralised water can be used safely with both aluminium and stainless steel 304 or 316 tubes.

Generally, water circulating through copper tube coils is part of a closed circuit system where continual re- aeration of the water is minimal. Therefore any corrosion that takes place from any contaminants in the water will reduce over time providing there is no continual ingress of oxygen into the circulating water.

Different types of contaminants in the water can react in combination and create corrosion rates many times higher than individual contaminants acting alone. The following table gives guidelines as to the acceptable water chemistry levels. Should these be exceeded then there is the possibility that accelerated corrosion may take place.

Water bourne compounds	Allowable quantity - parts per million
Ammonia	None
Bacteria	Bacteriologically safe
Calcium	< 800 ppm
Chlorides	< 5 ppm
Conductivity	< 500 µS/cm
Dissolved solids	50 - 150 ppm
Iron	3 ppm
Nitrates	< 10 ppm
Nitrogen compounds	None
Oxidising salts or acids	None
pH level	6 - 8.5
Silica as SiO_2	< 150 ppm
Sulphides	< 1 ppm
Sulphur Dioxide	< 50 ppm

CORROSION

TYPES OF CORROSION

Corrosion is generally an electrochemical process where metals react with the surrounding environment and the resulting metal degradation leads to a loss of material or material properties such as mechanical strength.

There are a variety of corrosion mechanisms that are encountered, both externally to the fin material or internally to the tube material or at the transition between dissimilar materials. These are classified as follows …

UNIFORM CORROSION

Occurs when the passive layer, which is usually an oxide layer, is destroyed over large areas of the parent material. This is usually associated with an attack by acid solutions or in the case of stainless steels, hot alkaline or chloride salt solutions.

PITTING CORROSION

Pitting corrosion is localised attack and occurs in materials that have a protective film such as an anti-corrosion product or when a coating breaks down. The exposed metal gives up electrons easily and the reaction initiates tiny pits with localised chemistry supporting rapid further attack.

Such corrosion is associated with stagnant fluids or areas with low fluid velocities.

CREVICE CORROSION

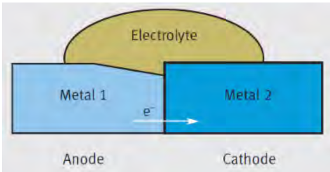
As the name suggests, this form of corrosion occurs in crevices or confined spaces. In narrow crevices capillary action forces liquid into the crevice and the natural chemical reactions that occur consume oxygen and the nature of the crevice prevents reoxygenation of the liquid.

This inequality in the solution inside and outside of the crevice cause hydrolysis of small amounts of dissolved metal ions causing a reduction in the pH level, further promoting corrosion.

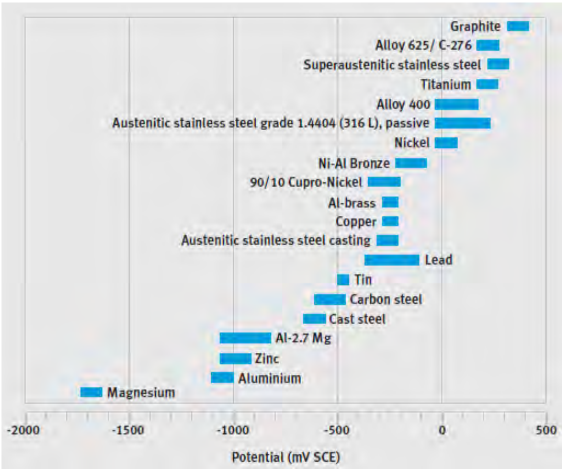
Such corrosion is associated with gaskets, bolts and lap joints.

GALVANIC CORROSION

This can occur when two different metals (with different electro-potentials) are placed in intimate contact with one another, in the presence of an electrolyte and results from the greater willingness of one to give up electrons than the other.



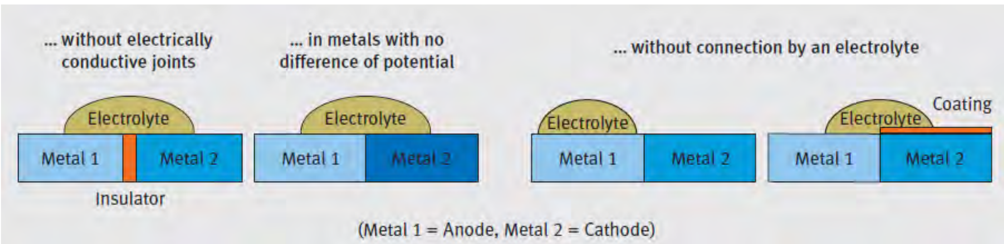
Generally, the less noble (reactive) material becomes the anode and is more severely attacked, whilst the more noble (passive) metal, the cathode, is essentially protected from corrosion. Often the anode develops deep pits and grooves on the surface.



Contrary to widespread belief, the difference of potential in an electrochemical cell alone is not a good indicator of the actual risk of galvanic corrosion. It only indicates whether such a risk should be considered. The decisive factor is not the difference of potential observed under standardised experimental conditions but rather the actual difference of potential under 'real' operating conditions.

The table to the left positions the potential of various metals in sea water @ 10°C.

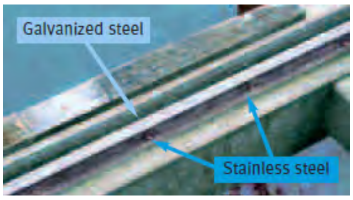
However, there are conditions when corrosion cannot occur …



The kinetics of electrode reactions depend upon the metal. For example, titanium reduces dissolves oxygen much less readily than copper, explaining why carbon steel corrodes more quickly in contact with copper than with titanium, although the latter has a higher potential than Copper.

Furthermore, the formation of corrosion layers can significantly change the potential of a material and become an obstacle to the anodic and/or cathodic partial reaction.

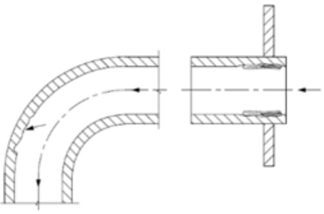
Thus stainless steel fasteners connecting much larger galvanised steel components do not normally cause corrosion …



Interestingly, stainless steel weld-neck flanges TIG welded to copper connection tubes do not exhibit galvanic corrosion because the 'intimate contact' is at a metal granular/molecular level not involving any dielectric, which is needed to promote this type of corrosion.

EROSION CORROSION

The mechanical effect of the velocity of a fluid combined with the corrosive action of the fluid causes accelerated loss of metal. The initial stage involves the mechanical removal of a metal's protective, often an oxide film, and then corrosion of the bare metal by a flowing fluid. The process is cyclic until perforation of the component occurs.



Erosion-corrosion is usually found at high flow rates around tube disturbances e.g. return bends or tube inlet ends.

The viscosity of the fluid also affects the onset of erosion corrosion. Typically for copper tubes 2.0 m/sec is deemed to be the maximum water velocity in a Ø15 mm tube, however this figure can be exceeded for chilled water applications or fluids with high viscosity, but for elevated temperatures close to 100°C, a velocity closer to 1.0 m/sec is recommended.

Cavitation-corrosion is a special form of erosion-corrosion and is caused by water bubbles produced by a high velocity flow, which then collapse causing pits on the metal surface.

STRESS CORROSION CRACKING

The combined action of a static tensile stress and corrosion which forms cracks and eventually catastrophic failure of the component. This is specific to a metal material paired with a specific environment.

INTERGRANULAR CORROSION

This is preferential attack of the boundaries of the grains that form the metal. It is caused by the physical and chemical differences between the centres and edges of the grains.

If coils are subject to hostile environments various material alternatives are available …

STAINLESS STEEL DATA

Both SS304 & 316 are available in a low carbon content variety i.e. SS304L (1.4307) or 316L (1.4404), which improves weldability, however, the material strength and corrosion properties are very similar. SS316 has a marginally higher density and is also a slightly harder alloy, otherwise many of the material and thermal properties of the two alloys are similar.

The composition of SS304 & 316 is very similar, both are austenitic steel alloys however, SS304 contains more chromium (18% Cr) than SS316 (16% Cr), whilst SS316 contains more nickel (10% Ni) than SS304 (8% Ni). Furthermore, SS316 contains 2% molybdenum, improving its resistance to corrosion.

STAINLESS ALLOY COMPOSITION

	Chemical Composition									
Grade	**C**	**Mn**	**Si**	**P**	**S**	**Cr**	**Mo**	**Ni**	**Density kg/m³**	**Hardness Rockwell B**
304	0.07	2	0.75	0.045	0.03	18	-	8	7900	92
304L	0.03									
316	0.08	2	0.75	0.045	0.03	16	2	10	8000	95
316L	0.03									

CORROSION RESISTANCE

SS304 is generally regarded as a general purpose stainless steel with good corrosion resistance properties, but SS316 with its additional 2% molybdenum content, is generally referred to as 'acid proof' and better suited to marine related applications.

Although the above is generally correct, there are some acids/chemicals that attack SS316 in preference to SS304 (see list below). Furthermore, although SS316 is fine for sheet metal fabrication for marine or offshore applications, it is unsuitable as the tube material for processes using sea water. In such cases **Titanium** is the preferred tube material.

Both stainless grades are subject to pitting and crevice corrosion in warm chloride environments and are subject to stress corrosion cracking above 60°C. Otherwise, for harsh environment applications, SS316 is always the stainless steel alloy of choice.

It appears that AISI 304 is not necessarily regarded as C4 compliant, approval from the Client is necessary to ensure suitability and acceptability.

Although AISI 316 is considered compliant for C5 applications, Client approval should be requested if CX (earlier C5-M [Marine]) is specified.

CHEMICALS THAT ATTACK SS316 BUT NOT SS304

In the case of an application using the following chemicals/fluids, written acceptance should be provided by the Client acknowledging that a 100% stainless SS304 (tube side) heat exchanger is to be provided due to the unsuitability of the use of SS316.

Do not use SS316 tube material with the following fluids, only SS304 may be suitable …

- NHO_3 - Nitric acid
- C_6H_5OH - Carbolic acid
- NaOH - Sodium hydroxide – 80%
- $CaCl_2$ - Calcium chloride
- SCl_2 - Sulphur chloride
- $NiCl_2$ - Nickel chloride

Aluminium is the most widely used fin material for tube and fin heat exchangers. However, it may be unsuitable for operating temperatures more than 150°C and can be susceptible to severe corrosion where the environment is both moist and contains an acidic component or is indeed salty, such as a coastal application.

Alternative Aluminium based alternatives that provide improved protection and longevity are …

ALUMINIUM MAGNESIUM ALLOY - ALMG

This fin material option is an aluminium alloy with a nominally 1% magnesium content, also referred to as EN AW 5005, often referred to as AlMg1. The 5000 series of alloys have superior strength properties compared with pure aluminium plus improved strain hardening ability and are also classified as non-heat treatable and thus often used for applications exceeding 150°C, which is the notional limit for pure aluminium.

Unlike the coated aluminium alternatives detailed below, the entire fin material is AlMg and thus does not exhibit an exposed ‘base’ material cut edge, which may be subject to attack in aggressive environments.

Traditionally, aluminium alloy AlMg3 was considered the de-facto fin material for marine/coastal locations, however more recently the lower magnesium content alloy (1%) has proved to be as corrosion resistant plus exhibiting better ductility and thus improved fin profile quality. Additionally, the higher thermal conductivity of 5005 compared with AlMg3 is also a benefit.

EN-Name	AW-1050A	AW-1350A	AW-2007	AW-2011	AW-2017A	AW-2024	AW-5005A	AW-5083	AW-5754	AW-6012	AW-6026	AW-6060	AW-6063	AW-6082	AW-7020	AW-7022	AW-7075
EN- Alloy	Al99,5	EAl99,5A	AlCu4Pb MgMn	AlCu6BiPb	AlCu4SiA	AlCu4 Mg1	AlMg1	AlMg4,5 Mn0,7	AlMg3	AlMgSiPb	AlMgSi MnBi	AlMgSi	AlMg0,7Si	AlSi1 MgMn	AlZn4,5Mg1	AlZn5 Mg3Cu	AlZn5,5 MgCu
DIN Material- No.	3.0255	3.0257	3.1645	3.1655	3.1325	3.1355	3.3315	3.3547	3.3535	3.0615	3.4345	3.3206	3.3206	3.2315	3.4335	3.4345	3.4365
spez. desity	2,7	2,7	2,85	2,83	2,8	2,77	2,69	2,66	2,66	2,75	2,75	2,7	2,7	2,71	2,77	2,78	2,75
Corrosion resistance																	
Weather	2	2	5	4	4	5	1	1	1	2	5	1	2	1	3	4	4 bis 5
Sea water	3	3	5	5	5	5	2	1	2	3	5	2	2	2	4	5	4 bis 5
Maritime- / Offshore applications							o										

Generally, the 5000 group of AlMg alloys have the best corrosion resistance of all alloys. However, above 4% Mg, the handling can have a marked influence on long-term corrosion behaviour. Where temperatures exceed 60°C and they are highly strain hardened, they may be susceptible to stress or exfoliation corrosion. The reason for this is that when magnesium is present in quantities >3-4%, there is a tendency for the AlMg beta-phase to precipitate in the grain boundaries. This leads to inter-granular and stress corrosion in aggressive environments. Marine codes place limits on Mg at 5.5% and temper at H116, a low degree of strain hardening.

Clearly, as the heat exchanger ages, the surface of the fin material will marginally oxidise and become less smooth, which does not lend itself to applications where the finned surface must cope with condensation or indeed adiabatic cooling spray water. For such systems the following AlEP or AlHy coated fin material is recommended.

Nevertheless, for dry cooler or air-cooled condenser applications … not fitted with an adiabatic spray option … where the atmosphere is maritime/coastal i.e. salt laden, then AlMg’s resistance to such environments makes it the preferred fin material option.

HYDROPHOBIC EPOXY COATING - ALEP

Is an Epoxy coating process for protecting aluminium surfaces from salty or acidic environments. AlEP is one pre-coated aluminium fin material option, which is available with the base aluminium material of thickness of 0.12 or 0.2 mm plus a 4 µm epoxy coating.

The pre-coated epoxy aluminium fin material is the most economical and effective method particularly for protecting metal surfaces exposed to the corrosive influence of the humid and salty air in regions with maritime climates.

The surface coating promotes ‘drop-wise’ water formation, which if the coil orientation is correct, will allow the water to drain from the surface. This type of coating in often referred to as exhibiting hydrophobic (*water fearing*) properties.

Tests conducted in accordance with ASTM B117(5% NaCI, 35°C) indicate that the resistance of Epoxy coated Aluminium samples to corrosion to a salty environment have a 4-7 times better life expectancy when compared with pure non-coated Aluminium. Furthermore, the resistance of the Epoxy coating to diluted acids (2% HCI, 2% H2S04) is also acceptably high.

Although the base material of this option is pure aluminium; which has a higher thermal conductivity than the above AlMg; the epoxy coating marginally insulates the surface and thus the net thermal conductivity is lower than pure aluminium but slightly better than AlMg. So, it must be remembered that there is a slight penalty for improved corrosion resistance and water dispersal.

HYDROPHILIC COATING - ALHY

The second pre-coated aluminium option is Hydrophilic coatings which can be effective in environments of excessive condensation (cooling coil applications) or indeed adiabatic spray applications and help protect the heat exchanger against the corrosive effect of contaminated water. Often, when water collects in large droplets on an uncoated finned surface, localised corrosion results.

Furthermore, the accumulated water droplets increase the air resistance and besides having a detrimental effect upon the capacity of the exchanger, also create water agglomeration (a 'water logging' effect), which can often lead to water carry-over issues.

To reduce the agglomeration of water droplets on the surface, Hydrophilic coated fin material can be used. This 'water loving' coating exhibits low surface tension characteristics thus promoting the generation of 'film wise' water layer allowing the water to flow more freely from the surface. Generally, higher face velocities can be accommodated before water carry-over occurs and thus moisture eliminators are required.

In 'wet' situations where the corrosive effect of liquid water is a more important factor than that of acid and salt, the removal of the water as soon as possible is desirable and AlHy fin material provides a solution.

Similar to AlEP fin material, the hydrophilic coating slightly impedes the thermal performance, but benefits are gained with longevity and water dispersal.

Generic salt spray tests indicate that Hydrophilic coated aluminium surfaces can last 3 times longer than untreated aluminium surfaces (500 hours for AlHy compared with 150 hours for aluminium) when exposed to a 5% NaCl spray. Furthermore, additional tests confirm the suitability of the surface coating when exposed to temperatures of 200°C for 5 minutes.

It is recommended that AlHy fin material with a minimum fin pitch of 2.5 mm should be used as a minimum requirement for 'peak load' adiabatic spray applications.

COPPER TINNED – CUSN

Pre-tinned copper fin strip is more durable than pure copper fin material and is often offered for off-shore applications with harsh environments. Such coils would usually be manufactured with either SS316 for non-seawater applications e.g. refrigerant or closed circuit water/glycol circuits or Titanium tubes when seawater is used inside the tubes.

CuSn fin strip is manufactured by passing the copper fin strip through an electro-tinning bath as the final stage of manufacture, which deposits a 4-6 µm layer of tin (Sn) on both sides of the fin strip

When the fin strip passes through the fin press and the tube holes are punched, the collars extruded and the strip slit into the appropriate rows deep and fin strip length, the cutting/shearing process tends to seal the exposed copper with a tin coating ensuring that the pure copper base material is not significantly exposed to atmosphere. Any base material still exposed will oxidise forming copper oxide, which helps protect the pure copper.

Although this fin material option is deemed to be very durable against corrosion, excessively corrosive atmospheres can still attack any exposed copper 'edges', which may have already oxidised and 'eat away' the base copper material between the passive tin coating.

In such circumstances, stainless or titanium fins are the only solution. However, the lousy thermal conductivity of stainless or titanium results in a coil with perhaps 2-3 times the number of rows deep.

COPPER TINNED AFTER MANUFACTURE – CUET

A once popular 'all copper coil' treatment, is the manufacture of a coil from copper tubes and copper fins and perhaps copper or brass casework. After the assembly and expansion plus the brazing on of the headers etc. and final pressure testing of the coil, it is placed in an electro-tinning bath and the whole coil is covered with a tin deposit.

This manufacturing process overcomes the one weakness of a coil manufactured using (pre-tinned) CuSn tubes and CuSn fins, which is the 'cut edge'. These cut edges expose the parent underlying material … copper … to the hostile/aggressive environment and thus corrosion. In worst case scenarios, the parent copper fin material is 'eaten' away leaving only the thin inert tin protection.

The following E-coat treatment is a 'modern day' alternative to the above process and allows any material combination to be coated and protected.

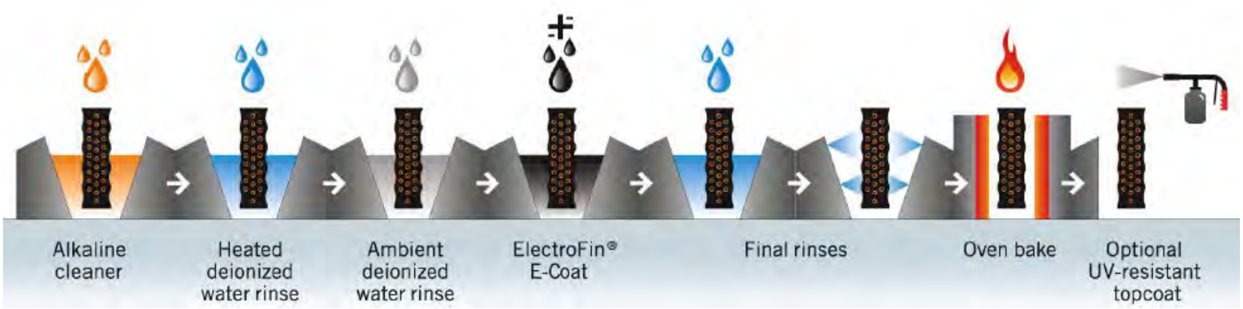
ELECTROFIN COATING

The problem with using corrosive resistant fin material such as stainless steel is the 2 to 3 times increase in the surface area (rows deep) required to meet the thermal performance, which can create product size implications besides a more expensive solution.

One solution is to use standard materials Cu/Al or Cu/Cu and then treat the whole coil in a durable C5/CX compliant coating.

One such treatment is ElectroFin E-coat, which is a cathodic epoxy dipped coating that penetrates and consistently covers the whole surface area including the edges of the fins and any gaps between adjacent fin collars.

The process offers 15 year longevity ISO 12944/C5M and is popular for offshore platform applications.



ALTERNATIVE TUBE MATERIALS

Copper and copper alloys are widely used in many environments and applications because of their excellent corrosion resistance, which is coupled with combinations of other desirable properties, such as superior electrical and thermal conductivity, ease of fabricating and joining, wide range of attainable mechanical properties, and resistance to biofouling.

COPPER

Copper corrodes at negligible rates in unpolluted air, water, and deaerated nonoxidizing acids. However, copper is susceptible to more rapid attack in oxidizing acids, oxidizing heavy-metal salts, sulphur, ammonia (NH_3) and some sulphur and NH_3 compounds.

Resistance to acid solution depends mainly on the severity of oxidizing conditions in the solution. Reaction of copper with sulphur and sulphides to form copper sulphide (CuS or Cu_2S) usually precludes the use of copper and copper alloys in environments known to contain certain sulphur species.

In aqueous environments at ambient temperatures, the corrosion product predominantly responsible for protection is cuprous oxide (Cu_2O). This Cu_2O film is adherent and follows parabolic growth kinetics.

For the corrosion reaction to proceed, copper ions and electrons must migrate through the Cu_2O film. Consequently, reducing the ionic or electronic conductivity of the film by doping with cations should improves corrosion resistance. In practice, alloying additions of aluminium, zinc, tin, iron, and nickel are used to dope the corrosion product films, and they generally reduce corrosion rates significantly.

Cupronickel (CuNi) is an example of such an alloy and is often used for seawater applications, albeit that EAS prefers to use titanium, which has an even better longevity.

STAINLESS STEELS

Generally, stainless grade 316L (1.4401); often referred to as 'acid proof' or 'marine grade' quality stainless steel; as superior to grade 304L (1.4301 also known as 18/8 stainless) and thus often use it as an alternative to SS304L even when SS304L is requested by the Customer. In fact, it is standard EAS practice to use SS316L return bends, end caps, vents & drains and ancillary fittings along with SS304L tubes and header material.

However, 99.9% of the time this mixture of the stainless steel grades is not an issue, but there are a few (0.1%) occasions, when this is not true. There are a few chemicals/fluids which can adversely react with or corrode SS316L tube material, whilst there would be no reaction with SS304L. *(see above)*

TITANIUM

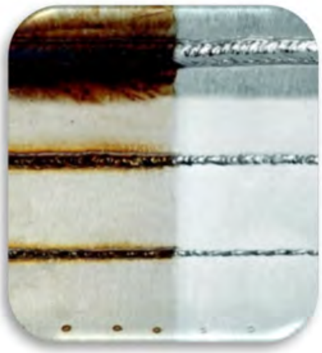
Titanium alloys were originally developed in the early 1950s for aerospace applications where their high strength-to-density ratios were especially attractive.

Although titanium alloys are still vital to the aerospace industry for these properties, recognition of the excellent resistance of titanium to many highly corrosive environments, particularly oxidizing and chloride-containing process streams, has led to widespread non-aerospace (industrial) applications.

Stemming from decreasing cost and increasing availability of mill and fabricated products, titanium and its alloys have become fairly standard engineering materials for a host of common industrial applications.

One such industry sector is 'off-shore' where raw seawater is used directly inside of the tubes as the primary coolant. Owing to the harsh environment, titanium is ideal as both a rugged and inert material able to withstand this erosive and corrosive nature of the application.

POST WELD TREATMENT



Welding of Austenitic Stainless Steel causes both discolouration (heat tint) and molecular modification of the HAZ (heat affected zone). If left untreated, accelerated corrosion may result.

Steels containing more than 12% Chromium form an invisible, inert or passive, self-repairing chromium oxide film on their surface. It is this passive layer that gives stainless steels their corrosion resistance and is a key characteristic of stainless steel and without it, the functionality of a stainless steel can be compromised.

To be in a passive state, a thin layer of chromium oxide must form on the surface of the metal. This can be achieved through the process of pickling and/or passivation.

Chemical pickling is usually achieved using strong acids; usually Nitric & Hydrofluoric; which dissolves both the heat tinted and unaffected stainless steel surface to leave a dull grey, matt finish that passivates spontaneously in the presence of air, forming a uniform layer of chromium oxide.

The pickling bath chemical contains 25-50% Nitric Acid plus 12.5-25% Hydrofluoric Acid, whilst the pickling spray contains 12.5-25% Nitric Acid plus 2.5-5% Hydrofluoric Acid

Chemical passivation is rarely needed to improve corrosion resistance if the stainless steel has been properly pickled. However, if the part/component has become contaminated following the pickling process, then chemical passivation can be used to 'polish' the metal surface and regenerate the passive chromium oxide layer.

Technically speaking, stainless steel passivation uni-potentialises the stainless steel with the oxygen absorbed by the metal surface, creating a monomolecular oxide film.

If a stainless steel surface is scratched, then more chromium is exposed which reacts with oxygen allowing the passive layer to reform. However, if a particle of carbon steel is embedded in the scratch, then the passive layer cannot reform and corrosion will occur when the metal is wetted or exposed to a corrosive environment, giving the appearance that the steel has rusted.

EFFECTIVENESS

Stainless steels are normally in a natural passive state. However, whenever the steel is manipulated or fabricated in any way, for example welded, the natural chromium oxide film on the metal surface is tarnished. Therefore, the corrosion resistance of the stainless steel is diminished. To restore the chromium oxide film on the surface of the metal, passivation is required.

LIMITATIONS

During the welding process, a metal loses its 'free iron' from the alloy and the overall structure of the metal surface is transformed. In the heat affected zone of the weld, the chrome to iron ratio is significantly decreased. The iron that is free flowing on the metal surface of the stainless steel can facilitate corrosion/roughing, hence the need for passivation.

Weld quality determines the success of the passivation process which will generally eliminate most of problems that are related to welding. However, the effectiveness of the passivation process relies heavily upon the quality of the weld created. If intense discoloration takes place during the welding process, then passivation will have little effect.

Furthermore, passivation is quite ineffective against the ill effects of 'sugared' welds.

In such cases, 'pickling' of the welded area may be required to remove scale and impurities followed by thorough rinsing followed by passivation to ensure development of the protective chromium oxide layer.

ACID PICKLING & PASSIVATION

Evapco Europe A/S prides itself upon manufacturing high quality products and has chosen; as standard; to use a post-weld cleaning process to maximise TIG welded joint corrosion resistance.

The generic term for this process is passivation and depending upon the extent of the surface oxidisation can be achieved by chemical pickling … an aggressive acid treatment, or chemical passivation … a less aggressive acid treatment.

Evapco's stainless steel TIG welding fabrication area, raw materials and processes are inherently 'clean' enabling passivation via acid pickling to be performed to provide the required level of passivation to achieve the improved TIG welded joint corrosion resistance. Secondary chemical passivation is thus unnecessary.

TIG welded header sub-assemblies are placed in an acid pickling bath, whilst tube-to-header leg & tube-to-return bend joints are externally treated with an acid pickling spray. Therefore, header assemblies are both internally and externally acid pickled.

Clearly, all treated parts/assemblies are thoroughly washed to remove all traces of acid.

Owing to the use of an inert 'backing gas' during the TIG welding process of return bends-to-tube joints and header leg-to-tube joints, internal weld oxidation is almost entirely avoided and thus to internally passivate these joints has never been deemed necessary or indeed, ever created cause for concern.

Furthermore, the internal fluid circulating through the tubes and header assembles is clearly non-aggressive towards the stainless material and additionally, the process systems are not usually aeriated, which eliminates ongoing oxidation of any HAZ/heat tinted areas.

Evapco's experience of corrosion/attack of welded joints is almost exclusively an externally initiated process. Any internal corrosion would likely be in the form of erosion corrosion resulting from excessive tube velocities when the heat exchangers are used outside of their design envelope.

ELECTRO-PASSIVATION

Although Evapco Europe 'passivates' its stainless steel TIG welded joints using an acid pickling process, in the event of a weld failure such as a 'pin hole' leak discovered during the pressure testing operation, the repair is passivated using an electro-passivation process.

Electro-passivation utilises a fluid containing 50-60% Phosphoric acid and an electrically connected (DC supply) application brush, which when a current is applied, dissolves the chromium depleted layer and surrounding stainless steel to expose a fresh layer allowing the spontaneous formation of the passive chromium oxide layer.

This technique is a more controllable procedure and can be applied to a single welded joint rather than affect a somewhat wider area. It is also claimed to combine the smoothing and cleaning benefits of pickling with the corrosion boosting benefits of passivation.

BACKGROUND

Corrosion classes were covered by DIN/ISO/EN12944 "Coating materials - Corrosion protection of steel structures by coating systems" standardizes corrosion protection by organic wet coating systems on "black" and/or hot-dip galvanized steel (duplex systems).

The standard, which dates mainly from 1998, has been completely updated and supplemented by the new Part 9, which regulates coating systems for offshore structures. This part replaces the withdrawn standard ISO 20340: 2009 and defines performance requirements for coating systems as well as test methods and evaluation criteria for coating systems for offshore and related structures.

For the first time, the coating standard DIN/ISO/EN12944 takes into account the durability of hot-dip galvanized steel to some extent and enables coatings on hot-dip galvanized steel to be thinned by a third, allowing this to be done even in highly corrosive offshore applications. Part 9 defines the minimum requirements for coating systems and their initial performance.

Comparing the requirements for the new corrosion category CX (**C**orrosivity: e**X**treme), which includes offshore areas with high salt exposure such as oil platforms and wind farms, but also industrial areas with extreme humidity and aggressive atmosphere, it is evident that coatings on hot-dip galvanised steel can be thinner and can be applied with fewer layers. The minimum number of layers is indicated as 2 and the minimum nominal layer thickness on the metallic coatings as ≥ 200 µm.

TESTING METHODS

ASTM B-117 tests coatings against a continuous salt spray. Since the procedure was first introduced in 1939, the test has been conducted for increasingly longer and longer exposure periods. Some tests now run for 15,000 – 20,000 hours, which is the equivalent of 20+ months of continuous exposure to the exact same salt spray environment. Although it might seem obvious that no real-world environment remains perfectly consistent for nearly two years, in 1995 The Society for Protective Coatings (SSPC) studied ASTM B-117 and published a paper that concluded that the ASTM B-117 test was an “*unreliable means for predicting coating behaviour*”.

ISO 12944-9 (*formerly known as ISO 20340*) uses a cyclic test where the test subject is exposed to 25 one-week cycles of common extreme coastal environment conditions. Each cycle consists of 72 hours of UV exposure, 72 hours of salt spray, and 24 hours of freezing. This cyclic method of testing correlates far better to the real-world environment HVAC coils will experience while in service. In addition, ISO/EN12944-9 has combined the C5-I (very highly corrosive, industrial) and C5-M (very highly corrosive, marine) environments into a new environmental corrosivity category – CX (extreme). The new CX Extreme environment provides a standard for atmospheric conditions that result in greater mass loss per unit surface/thickness loss after first year of exposure than the previous C5 environments.

WHAT HAS CHANGED IN THE 2018 STANDARD

The old C5-I and C5-M categories have been replaced with C5 for harsh onshore categories and by CX for offshore categories. CX is taken care of in a new Part 9.

There is also the addition of a fourth immersion category, IM4, which covers immersed structures in sea or brackish water which are protected by cathodic protection.

Category	Corrosivity	12944:1998	12944:2018
C1	Very Low	Heated buildings	Dry or cold with very low pollution
C2	Low	Low levels of pollution	Temperate low pollution
C3	Medium	Urban and industrial atmospheres, moderate pollution or low salinity	Temperate, medium pollution, tropical low pollution
C4	High	Industrial areas or coastal areas with moderate salinity	Temperate with high pollution, tropical with moderate pollution
C5-I	Very High	Industrial, high humidity, aggressive atmosphere	N/A
C5-M	Very High	Coastal and offshore areas with high salinity	N/A
C5	Very High	N/A	Temperate and subtropical with very high pollution and/or significant chloride effects
CX	Extreme	N/A	Extreme industrial areas, offshore areas, salt spray
IM1	Fresh water	River installations and hydro plants	River installations and hydro plants
IM2	Sea or brackish water	Harbour areas with structures and offshore structures	Immersed structures without cathodic protection
IM3	Soil	Buried structures	Buried structures
IM4	Sea or brackish water with cathodic protection	N/A	Immersed structures with cathodic protection

Durability Category	12944:1998	12944:2018
Low (L)	2-5 years	Up to 7 years
Medium (M)	5-15 years	7-15 years
High (H)	More than 15 years	15-25 years
Very High (VH)	-	More than 25 years

Up to C4 high, the test methods have not changed, however, for C4 very high, C5 high and C5 very high, cyclic testing has been introduced to replicate in-field conditions.

<table>
<tr><th>Category</th><th>Low (<7 yrs)</th><th>Med (7-15 yrs)</th><th>High (15-25 yrs)</th><th>Very High (25+ yrs)</th></tr>
<tr><td>C2</td><td colspan="3" rowspan="3">Non-cyclic Testing
Durations as 1998(E) revision
ISO 6270 / ISO 9227</td><td rowspan="2">Non-cyclic Testing:
Linear Durations TBC
ISO 6270 / ISO 9227</td></tr>
<tr><td>C3</td></tr>
<tr><td>C4</td><td>As C5 High. Phased cyclic testing: 10 cycles</td></tr>
<tr><td>C5</td><td colspan="2">Non-cyclic Testing Durations as 1998 (E)</td><td>Phased introduction of ISO 12944-9
Cyclic Testing: 10 cycles
Non-cyclic Testing valid for 5 years</td><td>Immediate introduction of
Cyclic Testing: 16 Cycles
ISO 12944-9</td></tr>
</table>

The introduction of Part 9 to the ISO12944 standard introduces the old ISO 20340 standard into ISO/EN 12944.

Part 9 mandates the use of cyclic testing for offshore structures. In previous editions of the standard offshore structures were referred to as C5-M however a new environmental category, CX, has now been introduced for all offshore structures.

All offshore systems must continue to go through 4,200 hours of cycling testing, which equates to 25 weeks.

Part 9 sets both the minimum number of coats and minimum film thickness per system, with some changes from the previous standard's requirements for C5-M. The table below outlines the requirements for steel substrates.

Category	CX		Splash & tidal zones			IM4	
Primer coat	Zinc (R)	Other primers	Zinc (R)	Other primers		Other primers	
NDFT (µm)	≥40	≥60	≥40	≥60	≥200	-	≥150
MNOC	3	3	3	3	2	1	2
NDFT of system (µm)	≥280	≥350	≥450	≥450	≥600	≥800	≥350

One of the main changes in ISO 12944 Part 9 from ISO 20340 is in the performance criteria on corrosion creep – this now states that coating systems for high impact areas shall be less than or equal to 8.0 mm and all other CX applications less than or equal to 3.0mm. Sea water immersion now states 6.0 mm pass criteria and there are slight changes to adhesion as well.

TUBE, FIN & CASEWORK MATERIAL CLASSES

It can be rather subjective as to the corrosion class that a particular material may fall into. Although the general environment and atmosphere may not be considered aggressive, certain airborne chemicals may be harmful to the parent metal.

Often Customers request a particular corrosion class but only relate this to the casework material … because they want an efficient heat exchanger tube bundle and do not realise that a C5 requirement for the casework and thus AISI 316, also means AISI tubes and fins … necessitating a significantly lower performing and somewhat more costly coil.

However, this conundrum can be solved using a 'good' thermal conductivity coil (Cu/Al), typically ElectroFin coated and mounted in AISI 316 casework.

The following is a list of common tube, fin & casework materials appropriately classified …

Tube materials …

- Copper : C4
- Aluminium : C3
- AISI 304 : C3/C4
- AISI 316 : C4/C5/CX
- Titanium : C5/CX

Fin materials …

- Aluminium : C3
- AlMg : C4
- AlEP : C3/C4
- AlHy : C3/C4
- Copper : C4
- CuSn : C4/C5
- AISI 304 : C3/C4
- AISI 316 : C4/C5/CX

Casework materials …

- Galvanised steel : C3
- AluZk 185 (25µm) : C3/C4
- $AlMg_3$: C4
- Copper : C4
- AISI 304 : C3/C4
- AISI 316 : C4/C5/CX

SAFE WORKING PRESSSURE & BURST PRESSURE

INTRODUCTION

Burst pressure is literally the pressure that a pressure vessel like pipe or tube can handle before rupturing or “bursting”. Environmental conditions and other factors can play a part in how a pipe will fail. Designing of a pressure vessel and selecting the material to be used can be a complex process. There are formulas that can be used to help estimate pipe or tubing’s burst pressure. The following should be used as a reference only.

The two most commonly used formulas are Barlow’s and Lamè equations. Burst pressure is a function of the material’s strength and pressure vessel dimensions (i.e. wall thickness, inside and outside diameter). Temperature can also affect the burst pressure since variations in temperature can have a direct effect on the material strength. The following discussion will take only room temperature conditions into consideration and this information should only be used a guideline or reference. A safety factor must be taken into consideration when calculating working pressure.

$$P = \frac{2 * WT * S}{\mathrm{OD}}$$

Barlow’s Equation is used for calculating the appropriate internal pressure. This can be either Burst Pressure or Yield Pressure depending on what material property is used in the equation. Used for determining the onset of plastic or permanent deformation when using the yield strength, whilst using the ultimate tensile strength identifies the bursting pressure or the point at which the material ultimately fails. Generally used for thin wall tubes when $t/D_o < \sim 0.1$

Lamè Formula – is a more general equation for both thin & thick wall tubes & cylinders. This formula was simplified from the complete Lamè solution.

$$P = S * \frac{(OD^2 - \mathrm{ID}^2)}{(\mathrm{OD}^2 + \mathrm{ID}^2)}$$

- P=internal pressure(i.e. burst pressure)
- WT= wall thickness
- OD=outside diameter
- ID=inside diameter
- S=stress(ultimate tensile strength or yield strength)

The above equation can be transformed to utilise the radii rather than the diameters, as implemented below.

YIELD STRESS - 0.2% OR 1.0% PROOF STRESS

Yield stress is the stress at which a material starts to plastically deform resulting in permanent deformation.

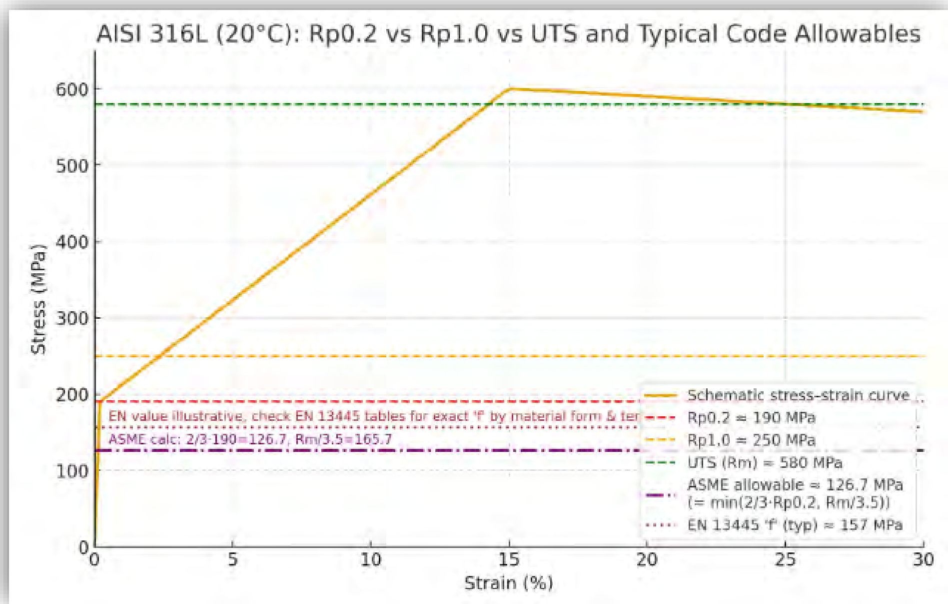
Materials such as stainless steels (e.g. 316L), aluminium and many non-ferrous alloys, the stress–strain curve does not exhibit a distinct ‘yield point’. Instead, it shows a gradual transition from elastic to plastic behaviour.

Thus, to define a pseudo-yield point a proof stress is used, which is defined as the stress that causes a specified small permanent (plastic) strain and often denoted by $Rp_{0.2}$ and $Rp_{1.0}$.

Definitions

- 0.2% proof stress - $Rp_{0.2}$
  - The stress required to produce a permanent plastic strain of 0.2% i.e. strain = 0.002 in engineering strain terms
  - Determined by projecting a line parallel to the elastic (Young’s modulus) portion of the stress-strain curve, but offset by 0.2% strain, and observing where it intersects the curve
  - This is commonly referred to as ‘yield strength’ for stainless steels, aluminium alloys and copper
- 1.0% proof stress - $Rp_{1.0}$
  - Same definition, but with a permanent strain of 1.0% (strain = 0.01)
  - This gives a higher numerical value since allowing more permanent deformation before calling it ‘yield’
  - Less conservative but often used where ductility is high and small plastic strains are acceptable

The following chart provides an example of the various terminologies …



- $Rp_{0.2}$ ≈ 190 MPa (red dashed)
- $Rp_{1.0}$ ≈ 250 MPa (orange dashed)
- R_m (UTS) ≈ 580 MPa (green dashed)
- ASME allowable (illustrative) ≈ 126.7 MPa (purple dash-dot)
  This was calculated as the minimum of {2/3·$Rp_{0.2}$ = 2/3·190 = 126.7 MPa and R_m/3.5 = 580/3.5 = 165.7 MPa} - ASME Section II, Part D tables give the formal allowable stress by material form and temperature; the code effectively enforces a safety factor via those limits
- EN 13445 illustrative *f* ≈ 157 MPa (brown dotted)
  EN uses a design stress; often called *f*; taken from EN tables and calculation rules. Values differ by material and temperature - the value used is a representative figure seen in EN/industry comparisons and literature

EXAMPLES

Calculate the burst pressures using both Barlow (thin-wall) and Lamè (thick-wall) formulae for a AISI 316 tube with Ø15.0 mm (outer dia.) and wall t = 0.6 mm at 20-50°C.

The following results are based on the yield strength and the ultimate tensile strength (R_m) detailed in EN 10217-7

Assumptions

- Ø15.0 mm = outer diameter (D_o)
- Inner radius ($r_i = r_o$ - t) … ($r_o = D_o$ / 2 = 7.5) mm → (r_i = 6.9) mm
- Material properties (EN 10217-7) …
  - o Yield strength $Rp_{0.2}$ (S_y) = 182 MPa
  - o R_m (S_u) = 590 MPa
- Thin-wall condition: (t / D_o = 0.6 / 15 = 0.04) … *so thin-wall approximations are reasonable*

Formulas used

Barlow - thin wall …

$$P_{barlow} = \frac{2S\,t}{D}$$

Lamè - general thick wall …

Bursts when hoop stress at inner surface equals (S):

$$P_{Lame} = S\,\frac{r_o^2 - r_i^2}{r_i^2 + r_o^2}$$

Units: using mm and MPa gives pressure in MPa

Results

Using Yield, $Rp_{0.2}$ - 182 MPa

- Barlow : P = 2 x 182 x 0.6 / 15 = 14.5 MPa = 145 barg
- Lamè : P = 15.1 MPa = 151 barg

Using UTS, R_m - 590 MPa - *theoretical burst (material rupture)*

- Barlow : P = 2 x 590 x 0.6 / 15 = 47.2 MPa = 472 barg
- Lamè : P = 49.1 MPa = 491 barg

Note: Lamè gives slightly higher values (exact solution) because (t / D) is small, Barlow (thin wall) is a very good approximation here.

Practical guidance

- Do not use R_m (UTS) burst values for design, which are theoretical failure pressures.
  Use an appropriate safety factor or the allowable stress from the applicable code (ASME, EN, etc.).
    - For example, with a safety factor 4 on the R_m, UTS-based Lamè burst pressure …
      Allowable working burst ≈ 49.1 / 4 ≈ 12.2 MPa ≈ 122 barg
    - Alternatively pressure-vessel/tubing codes instead limit to a fraction of yield (e.g. 0.6 x $Rp_{0.2}$) …
      Allowable working yield ≈ 0.6 x 15.1 ≈ 9.06 MPa ≈ 90.6 barg

Example 2

Find the allowable internal pressure using EN 13445 rules and AISI 316L (EN 1.4404) @ 300°C, D_o = 15.0 mm, wall t = 0.6 mm, welded tube with joint efficiency E = 0.70. using the EN-13445 method for the nominal design stress (f) and then evaluated both thin-wall (Barlow) and thick-wall (Lamè) forms using the weld efficiency as an effective thickness.

EN-based assumptions

- Material mechanical data for 1.4404 (316L) …
  $Rp_{0.2}$ at 300 °C = 118 MPa and R_m at 20-50 °C = 590 MPa
- EN 13445 nominal design stress f for pressure parts is taken as the minimum of the temperature proof strength divided by 1.5 and the room-temperature tensile divided by 2.4, i.e.

$$f = \min\left(\frac{Rp_{0.2,T}}{1.5}, \frac{R_{m,20}}{2.4}\right)$$

- Joint efficiency, E = 0.70 as the tube has a longitudinal welded seam

Calculations

- $Rp_{0.2}$ at 300 °C / 1.5 = 118 / 1.5 =78.6 MPa
- R_m at 20-50 °C / 2.4 = 590 / 2.4 = 245.9 MPa
- Therefore, f = 78.6 MPa
- Effective wall thickness, $t_{eff} = t \times E$ = 0.6 x 0.7 = 0.42 mm
- r_o = 15 / 2 = 7.5 mm
- r_i = 7.5 – 0.42 = 7.08 mm

Formulas used:

- $P_{Barlow} = \frac{2\, f\, t_{eff}}{D_o}$ = 4.4 MPa = 44 barg
- $P_{Lamè} = f\, \frac{r_o^2 - r_i^2}{r_o^2 + r_i^2}$ = 4.5 MPa = 45 barg

Therefore, at 300°C the conservative effective maximum pressure (per the code) that a Ø15 x 0.6 mm tube should be exposed to is 44 barg before it exhibits plastic deformation (yielding).

This compares with 145 barg for the same tube at 20-50°C with no allowances. However, using a safety factor of 4.0, suggests 491 / 4 = 122 barg or possibly 151 x 0.6 = 90.6 barg

INTRODUCTION

Extracts from 'Evaporator Frosting in Refrigeration Appliances: Fundamentals & Applications' by Christian J. L. Hermes

Frost is likely to build up whenever moist air flows over a chilled surface whose temperature is sub-zero and below the local dewpoint of the air stream. Therefore, low temperature liquid coolers and refrigerant evaporators are prone to frosting.

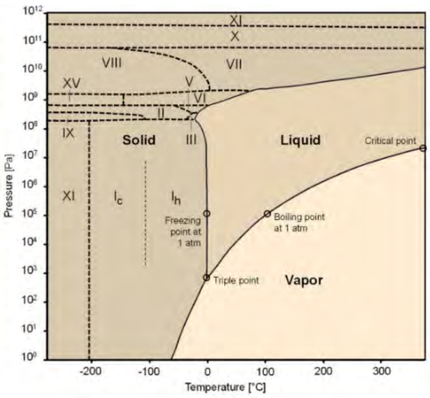
To mitigate evaporator frosting issues, periodic defrosting operations are carried out using a variety of systems.

In most applications, the defrost system employs electric heaters for melting the ice, albeit alternative strategies such as hot-gas and reverse cycles and fan delays can also be adopted.

To achieve defrost, the appliance must be switched off for a certain period, while the defrosting process takes place not only consuming energy but also increasing the cabinet thermal loads which requires removal by the cooling system upon start-up, thus further increasing the energy penalty. For instance, in the case of electric defrost, just a small part (*roughly 20%*) of the power supplied to the heater elements is actually used to melt the ice.

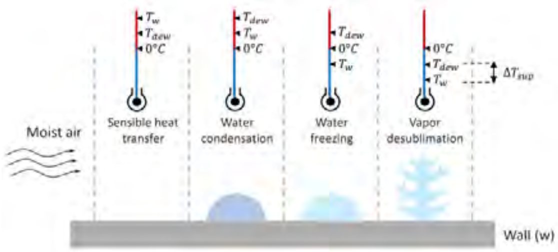
FROST FUNDAMENTALS

Ice is water in its solid-state and depending upon the thermodynamic condition (temperature and pressure), fifteen different forms of ice can be observed as shown.

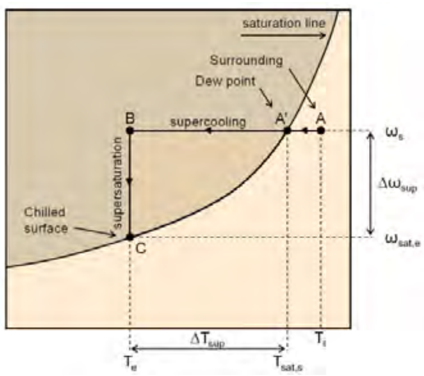


On the other hand, frost is a porous medium comprised of ice crystals and moist air formed due to a very particular set of surface and psychrometric conditions.

When moist air flows over a chilled surface, either condensation or freezing may occur. If the surface temperature is below the dewpoint of the moist air and above the water freezing point, moisture condensation may occur. However, if the surface temperature is below the dewpoint and below the freezing point, the water vapor may condense and freeze.



Nevertheless, if both the surface temperature and the dewpoint are below the freezing point, vapor desublimation into solid may take place. The latter is the most likely condition for the accretion of a frost layer, being attained at the solid-vapor boundary below the triple point.



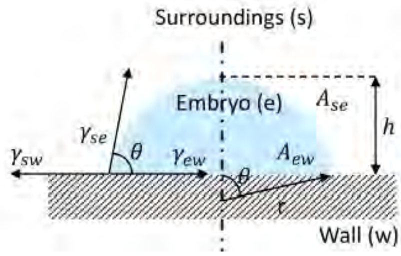
The phase diagram depicts only the lowest-energy state boundaries between solid, liquid, and vapor phases, notwithstanding the non-equilibrium in each phase transition, which can be represented by the band of metastability, shown in the figure to the right. In this case, the nucleation process is only triggered when the embryo is cooled down past a certain energy barrier, defining a fixed amount of supercooling required for the onset of the phase change. In the psychrometric chart illustrated, the existence of a metastable state characterized by a certain degree of supercooling in a vapor supersaturation that corresponds to the absolute humidity potential that drives the mass transfer process referred to as the degree of supersaturation degree. The nucleation can be represented by three distinct psychrometric processes, two of them occurring simultaneously: first, water vapor is cooled down in the boundary layer (A->A'), then heat and mass transfer take place, cooling down the water vapor in air (A'->B) and dehumidifying the air stream as the phase change goes on (B->C). The overall amount of energy to be removed is a combination of sensible heat of the moist air and latent heat of desublimation.

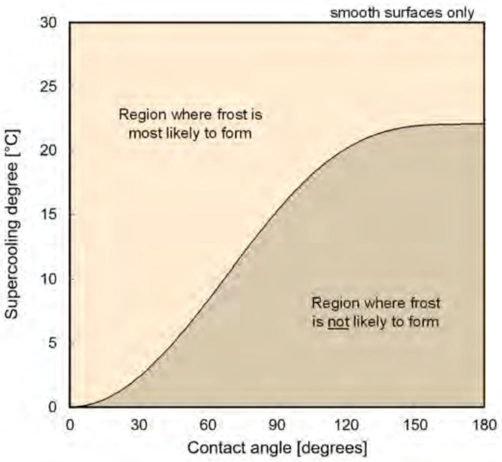
In order to quantify the amount of supercooling/supersaturation required for the onset of phase transition, classic nucleation theory should be considered.

CRYSTAL NUCLEATION

Nucleation is initiated when the amount of energy removed from a cluster of water molecules i.e. embryo, is high enough to surpass a certain energy barrier related to the supercooled/supersaturated metastable states shown in the above diagram.

Nucleation is called homogeneous when the phase change occurs with no contact with solid boundaries, or heterogeneous when there is an interface between the vapor and a substrate (a typical condition of refrigerator evaporators), as shown to the right. A hemispherical shape of radius r is considered as the cluster tends towards a state of minimum energy, seeking the smallest surface area for a given volume.





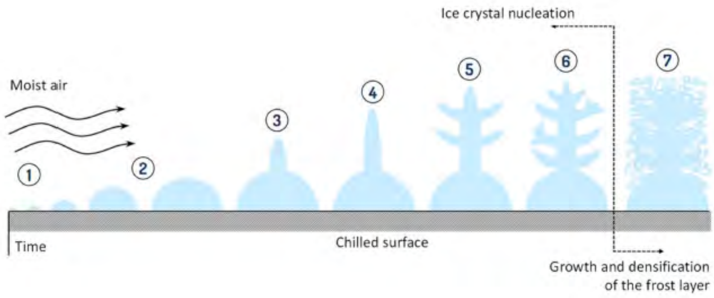
Due to the stochastic nature of the process, the frosting domain is divided into two regions, with the upper one indicating the conditions when frost is most likely to occur and the lower region indicating the opposite. It can be observed that the diagram shows the energy barrier for frosting on super-hydrophilic surfaces (ϑ -> 0°) is quite low, requiring virtually no supercooling for the onset of nucleation.

On the other hand, the highest supercooling is observed for super-hydrophobic surfaces (ϑ -> 180°), although a plateau around a supercooling of ~22 K is observed for (ϑ>135°).

This high initial energy barrier can **lead to a delay in the onset of frosting on superhydrophobic surfaces by as much as one hour**. In practical terms, this barrier to frosting is even more attenuated in the case of non-smooth surfaces typical of refrigeration applications, as the surface roughness is likely to favour the onset of nucleation due to the reduction on the interface area, i.e., less energy required to form a new interface.

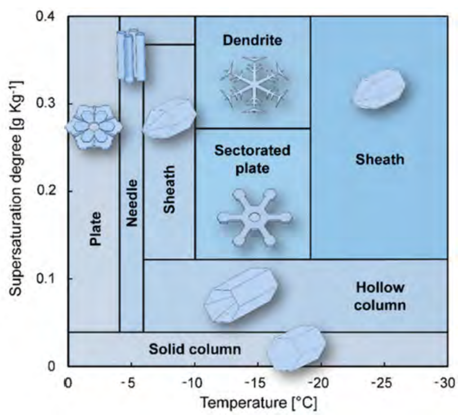
FROST BUILD-UP

The frost formation over a flat surface can be divided into several steps, as shown below.



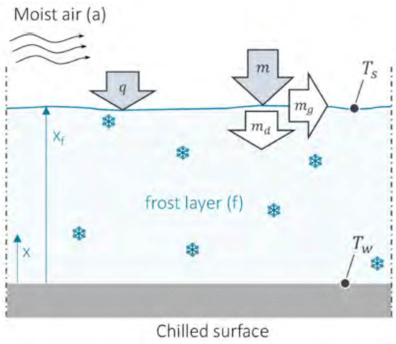
After the nucleation (1), the embryo starts growing (2). During this stage, the surface wettability can influence the size and shape of the growing embryo. Its surface area and temperature increase as it grows, requiring a higher amount of energy withdrawal to keep growing. When such amount becomes higher than the amount required for new nucleation, the primary embryo stops growing, and a secondary one emerges at a new site on the surface of the original embryo (3). Once again, the new embryo grows (4) so that successive embryo nucleation and crystal growth cycles (5, 6) go on until a mature layer of frost is formed (7). Stages (3) to (6) are called the early crystal growth period and are characterized by the morphology of the ice crystals.

It has been observed that ice crystals grow in distinct shapes (morphologies) depending on the psychrometric conditions, especially on the temperature and supersaturation of the air stream. This led to frost morphology maps which shows the crystal shapes that grow as a function of surface temperature (that sets whether ice crystals will grow into plates or columns) and water vapor supersaturation (which determines the complexity of the structures).

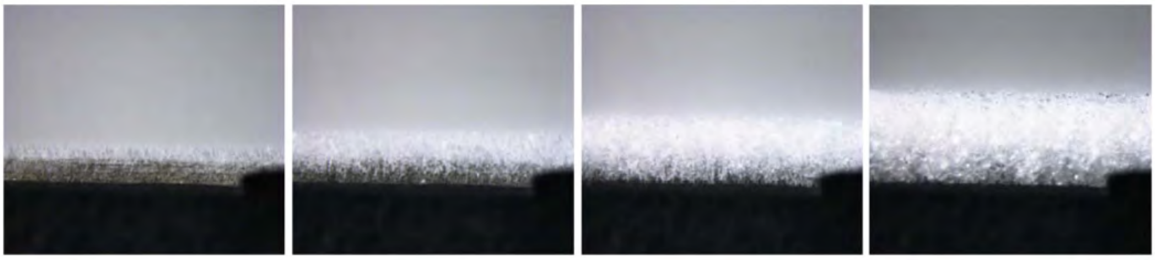


According to the theory, the variations observed in the growth patterns of ice crystals may be explained by a quasi-liquid layer on the surface of the crystals. Such a layer has properties like water in the liquid phase, as it contains atoms with greater mobility than those inside the crystal. The theory considers a strong effect of the temperature of the surroundings on the properties and thickness of the quasi-liquid layer on the basal and prism facets of the crystal, favouring the growth of the crystal in preferential directions, according to the psychrometric conditions of the environment. Later, the growth rate of the different faces of ice crystals was measured experimentally and showed good agreement with the quasi-liquid layer theory.

As shown in the frost build-up diagram on the previous page, it is observed that from stage (7) on, a mature layer of frost keeps growing and densifying due to simultaneous heat and mass transfer phenomena on and within the porous medium. This is illustrated in the following diagrams.

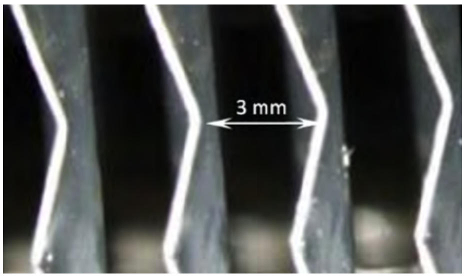


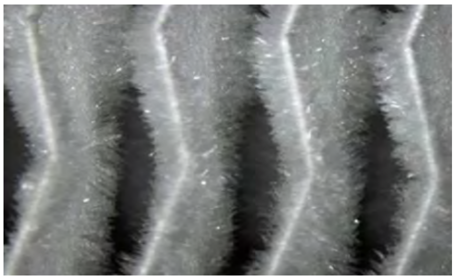
In general, the thickness of a growing frost layer follows the square root of time rule, as shown in the photos below of frost growth, due to the diffusive nature of the heat and mass transport inside the frost layer.



Frost formation over time : air velocity of 0.7 m/sec, air temperature 16°C/50% RH and plate temperature of -18°C

When related to a real coil …





EVAPORATOR FROSTING

Recent investigations on the performance of evaporators subjected to frosting conditions have focused on advancing first principles simulation models aiming not only at the frost accretion phenomena but also new features such as multidimensional frost distribution over the coil, fan-evaporator aerodynamic coupling, innovative frost mitigation and restraining techniques, and performance enhancement techniques such as vortex generation.

The thermal-hydraulic performance depletion of heat exchangers under frosting conditions was firstly reported by Stoecker, who investigated the heat transfer degradation and airside pressure drop augmentation in industrial evaporators. Remarkably, most works were aimed at the frost accumulation on evaporators for medium and large-capacity refrigeration systems. The majority investigated evaporator frosting with the aid of a wind-tunnel calorimeter, where the inlet psychrometric conditions and the air flow rate are held constant during the test. Therefore, the non-linear coupling between the rising pressure drop observed in frosted evaporators and the fan-supplied air flow rate, which is a key factor for the cooling capacity reduction, was not captured. Only a few studies tackle this issue. They designed and constructed a closed-loop wind tunnel facility for evaporators for light commercial applications. The air flow rate was controlled by a computer-driven variable-speed centrifugal fan that could emulate any desirable fan characteristic curve from a real application.

Additionally, in most applications, the evaporators are subjected to non-uniform frost formation from one row to another. Thicker frost layers tend to form in the first rows, where both the mass transfer coefficients and the humidity gradients are much higher, whereas thinner frost layers build up in the last rows.

Evaporators used for commercial applications, typically exhibit uniform psychrometric and air velocity profiles at the face area yield a homogeneous frost distribution at each transversal plane. Thus, despite changing along the airflow direction, the frost mass is likely to be evenly distributed at each tube row.

OPTIMAL DEFROSTING

In modern refrigeration systems, air-supplied evaporators are designed to be robust to frost blockage of the air passageway either by choosing a proper defrost strategy or managing an optimal defrost cycle (i.e., the time between defrost operations).

Generally, defrosting methods are defined as passive or active …

- Passive methods use surface characteristics to delay or minimize the frost accretion over the evaporator surface
- Active methods require some additional power input, such as …
  - Electric resistive heaters embedded in the evaporator
  - Hot gas bypass
  - Reverse cycles
  - Ultrasonic vibration
  - Off-cycle defrosting

Hot gas defrost cycle - diverts refrigerant from the compressor discharge directly into the evaporator. This technique is often referred to as “hot gas bypass” because the superheated refrigerant from the compressor discharge bypasses the condenser and the expansion device through a solenoid valve directly to the evaporator inlet.

The heat exchanger coil is warmed up from the inside and the defrost efficiency of this technology is high when compared to an ordinary electric heater. However, the net impact upon the refrigeration system performance, including not only thermal load increase but also refrigerant mass migration, cannot be neglected.

Reverse cycle - usually adopted in air source heat pump units, where a four-way valve reverses the refrigeration cycle so that the evaporator becomes the condenser and the enthalpy that would have been used to heat the indoor environment is used to melt the frost. Similarly, this method also returns a defrost efficiency higher than that of electric heaters.

One of the disadvantages of such an approach, besides requiring extra valves, is that the thermal comfort of the indoor space may be adversely affected.

Electric resistive heaters - probably the most common defrosting technique due to its low cost, fast response, easier control, and flexibility in terms of positioning, power magnitude, and power distribution.

Heater elements provide both a radiative-convective-dominant process, where working temperatures may exceed 300°C. Localised meltwater can often boil creating plumes of condensed water vapour which fill the cooling space, which then needs to be removed upon start-up.

COATING TREATMENT ON FIN SURFACE

Frost growth rate and density highly depend on the surface characteristics. Parameters such as liquid-solid contact angle is perhaps mostly widely used to describe the surface morphology. Several researchers have investigated the impact of surface type and associated frost growth rate and the general conclusions follow …

- It has been observed that frost growth on cold surfaces exposed to warm humid air streams can be reduced significantly by the presence of cross linked hydrophilic polymeric coatings. The frost thickness was decreased in the range of 10–30% when compared to using an uncoated metallic surface
- Investigations into both frosting and defrosting processes on hydrophilic and hydrophobic surfaces, show that hydrophilic coating was preferable for operation under frosting conditions
- Based upon the distribution of ice crystals and time of frost appearance on a normal copper surface, a hydrophobic coating (car wax coating) surface and a hygroscopic coating (glycerol coating) surface. The hygroscopic coating performed better than the hydrophobic coating; however, based upon parameters such as coating thickness, thermal resistance and expansion defect of the hygroscopic coating, the hydrophobic coating was found to be superior to the hygroscopic coating
- Another study on hydrophilic and hydrophobic treated heat exchangers concluded that during frosting, a relatively higher frost density formed on a hydrophilic surface, and the water draining rate during defrosting was

higher. On the other hand, for a hydrophobic surface the frost density was lower and the draining water rate during the frost melting process was increased mainly due to large chunks of incompletely melted frost. The conclusion was that the hydrophilic treatment influences the behaviour of frosting while the hydrophobic treatment becomes more important during defrosting.

- Hydrophobic treated fins appear to delay the onset of frost formation by at least 15 min. The thickness and the mass of the deposited frost layer was reduced by at least 40% compared with that on the uncoated copper surface. It is important to note that the deployment of a polymer layer introduced a thermal resistance due to its lower thermal conductivity.
- Frost layer on the hydrophobic surface was ‘‘thicker and fluffier” resulting in a less dense frost than the frost on the baseline surface.

The conclusions are not exactly definitive, but there is a strong indication that a hydrophobic coated fin may delay the onset of frosting, however the ‘water shedding’ behaviour of a hydrophilic coated fins may be preferable under certain circumstances.

ADDITIONAL DUTY ASSOCIATED WITH FROSTING

The formation of frost is associated with an additional thermal load related to the ‘latent heat of fusion’ ... the change of phase from a liquid to a solid. Any further cooling of the air stream also involves the sensible cooling of the frost layer.

Thus, the ‘total’ energy for the process of cooling a moist airstream to below zero involves both the sensible & latent cooling of the airstream, and once the frost layer is formed, the (relatively) small additional load related to the formation of the frost and thereafter the sensible cooling of the frost layer.

This later energy requirement is often ignored and besides, it is associated with the ‘fin surface temperature’, can be difficult to assess.

SUMMARY

Standard coil production techniques and the tolerances involved in the manufacture of the sheet metal casework and mechanical expansion of the tubes will not result in an airtight product. Indeed, a moderately negative or positively pressurised ducted system may induce or lose more air than is desirable, if the leakage can be quantified … and here lies the issue.

Currently, there is no recognised ISO or EU Standard that defines how air leakage for heat exchangers (coils) should be defined. Consequently, the 'Industry' tends to misapply Standards such as EN 15727 and define Class A, B or C or perhaps air/gas tight requirements.

Clearly the latter definition is accomplished via a fully welded casework construction and by inference, is indeed airtight. However, Classes A, B & C define a leakage rate based upon the operating pressure and a nominated 'dimensional parameter'. For ducting and AHU systems, this is quantifiable, but for coils no such parameter is defined.

In an endeavour to clarify matters, tests were conducted to measure air leakage rates for a variety of coil geometries and casework constructions/options and can now predict the leakage (in litres/second) for a given coil at a defined pressure.

DEFINITIONS & STANDARDS

EN 15727 refers to 'air-tightness classes', which in turn references EN 13779, EN 1507 (*Rectangular Ducts*), EN 12237 (*Circular Ducts*) & EN 1751 where the maximum leakage, f for each class is given by ..

- Class A : $f = 0.027 \cdot p^{0.65}$
- Class B : $f = 0.009 \cdot p^{0.65}$
- Class C : $f = 0.003 \cdot p^{0.65}$
- etc.

where, f *= air leakage, litre/sec/m²*
p = static pressure, Pa

The problem here is the definition of the surface area (m^2) for a coil. EAS's scope of supply does not involve any potential leakage issues associated with the casework/ductwork flanges … this is the responsibility of the Client when he connects the coil to the matching equipment.

However, air can escape from both the casework joints (if not welded or sealed) and the small clearance gap between the tubes and holes in the supporting tube-plates. *The gap between the tube and tube-plate is a design feature to allow the 'tube bundle' to move without distorting the casework following thermal expansion of the heat exchanger.*

If the coil is mounted inside a chamber or enclosure, such as an air handling unit, fan coil unit, condensing unit etc., then these small gaps will not result in any undue air leakage/bypass, because of the similarity in pressure 'seen by' the finned tube bundle and return bends & headers.

However, for coils exposed to negative of positive pressures in 'duct mounted' arrangements, air leakage may be a concern and on occasions, a significant issue.

If air leakage is a concern, then the coil must be specified to include one of the EAS's leak reduction options; once the Client has defined what level of air leakage, he is able to accept … and here lies another issue.

The Client often does not know what level of air leakage is acceptable! Often 'ductwork' Class A or B air leakage is specified, because this is the requirement for the rest of the HVAC system, but it cannot be directly applied to a coil.

REDUCED AIR LEAKAGE OPTIONS

EAS offer three different air tightness classes …

- **EAS Class A** – Moderate leakage - silicone sealing between the end fins and tube-plates both on the air entry and air leaving sides of the coil

- **EAS Class B** – Reduced leakage - cladded coil comprising internally silicone sealed pop-riveted boxes on both the header and return bend ends of the coil
- **EAS Class C** – Air/gas tight - Fully welded construction

RE-EVALUATION OF THE AIR LEAKAGE PARAMETERS

As defined above, the leakage attributed to a 'Class' for ventilation systems is based upon a 'surface area', which for ducting and other HVAC equipment, is definable. However, in the case of a coil, besides the casework joints (*not the duct work flanges*), leakage occurs around the tubes and the coil may comprise anything between 4 to 500 tubes … depending upon the face size of the coil and the rows deep. Thus, leakage is a 'fact of life', if not addressed.

Historically for coils, an 'equivalent surface area' had been conjured up, enabling the air leakage in litres/sec to be established when applied to the above Standards.

Equivalent surface area (m^2) = 10 x Finned Height x Finned Width x Rows Deep

… where, height & width are in meters

As an example, consider a coil 1000 mm high x 1000 mm wide and 4 rows deep operating at an internal duct pressure of 400 Pa. If **Class A** is the defined EN 15727 air tightness class required, then the allowable leakage would be ..

$f = 0.027 \times 400^{0.65} \times [10 \times 1 \times 1 \times 4] = 53.06$ litre/sec *… perhaps unrealistic !!*

Furthermore, if this same coil had a finned length of 2000 mm then the leakage would increase to 106.1 l/sec … again crazy … *why should the finned length of a coil affect the air leakage?*

The redefined method of predicting air leakage for a given coil based upon the 'number of holes', i.e. 2 x tubes high x rows deep and the leakage per hole (*Lph*), which is tube diameter dependent.

STANDARD COIL LEAKAGE

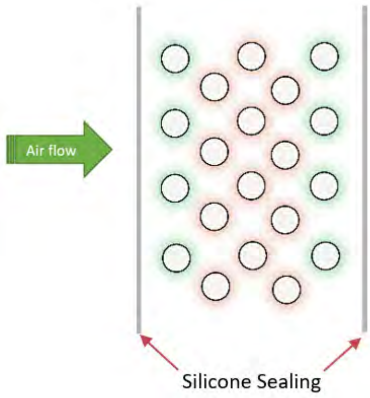
Air leakage is dictated by the tube diameter and the following algorithms …

- Ø9.52 mm tube – Z Fin
  - ***lph = 0.001 x $p^{0.78}$***
- Ø15 mm tube – A, B & C Fin
  - ***lph = 0.01 x $p^{0.55}$*** *where, lph = Air leakage per hole, litre/sec*
    p = Static pressure, Pa

Thus, the total air leakage from a given coil is …

Leakage = 2 x T x R x lph *where, Leakage, litre/sec*
T = Tubes high
R = Rows deep

EAS CLASS A – MODERATE LEAKAGE



This moderate level of tightness is accomplished via silicone sealing of the two end fins to the tube-plates on both the air entry and air leaving sides of the coil. This significantly reduces the air leakage originating from the small clearance gap between the tubes and holes in the supporting tube-plates in the first and last rows. However, as the top & bottom edges of the fins are not able to be sealed, the air leakage through the remainder of the rows is not affected.

Tests suggested that this method reduces the leakage by 70% in these outer two rows.

The algorithms for EAS Class A leakage are …

- 1 row deep
  - **Leakage = 0.6 x T x R x lph**
- 2 or more rows deep
  - **Leakage = 2 x T x lph x [R – 1.4]**

EAS CLASS B – REDUCED LEAKAGE

An improvement upon the above tightness class, which further reduces the leakage is accomplished by pop riveting internally sealed sheet metal cover boxes to both the header and return bend ends of the coil. This method avoids the need to silicone seal the end fins, (*which can prove an advantage if silicone sealant is not permitted by the specification*) and has the effect of partially containing the air escaping through the tube hole clearances.

This option is by no means air/gas tight, but an improvement upon EAS Class A.

The algorithm for EAS Class B leakage is …

Leakage = 0.25 x EAS Class A

EAS CLASS C – AIR/GAS TIGHT

Air/gas tight coil requirement involves a fully welded version of EAS Class B i.e. standard coil fitted with cover boxes, which dictates that the sheet metal should be manufactured from stainless 304/316 or Aluminium.

Depending upon the combination of the internal duct pressure and size of the unsupported surfaces, may necessitate the addition of stitch welded reinforcement ribs to avoid distortion of the casework.

Clearly, EAS Class C has **zero** leakage from EAS's scope of supply, however, in-situ leakage made result from incorrectly/insufficiently sealed duct-work/unit flanged joints.

THE BACKGROUND TO THE ECODESIGN DIRECTIVE

Under the Kyoto Protocol, the European Union has committed itself to reducing its CO_2 emissions by 20% by 2020.

The EuP Directive 2005/32/EC (***E**nergy-**u**sing **P**roducts Directive*) was adopted in 2005 to achieve this goal. This was renamed the ErP Directive 2009/125/EC (***E**nergy-**r**elated **P**roducts Directive*) in 2009, also known as the "Ecodesign Directive".

For energy-using products, it provides the general framework for their environmentally compatible design, i.e. for assessing savings potential, defining minimum energy efficiency requirements and considering resource efficiency over the entire product lifecycle. As there are a myriad of energy-using products, product groups (Lots) have been created for each of which implementing regulations were then drawn up. For example, Lot 6 groups together ventilation units, Lot 11 fans and Lot 21 air heaters/coolers and fan coils. These relate to the implementing regulations EU 2014/1253 (Lot 6), EU 2011/327 (Lot 11) and EU 2016/2281(Lot 21).

DIFFERENCE BETWEEN A DIRECTIVE AND A REGULATION

An EU directive is either transposed by the member states into national law for its implementation or it becomes effective via an EU regulation, which then becomes directly valid in all member states. This procedure was chosen for the requirements of the Ecodesign Directive (ErP Directive 2009/125/EC) for electric motors, fans, as well as HVAC systems and their energy-relevant components.

REGULATION EU 2009/640 FOR ELECTRIC MOTORS

This ErP implementing regulation came into force in 2011 and prescribes the efficiencies of IEC standard motors.

These values were increased in 2015 and then again in 2017 according to a specified timetable. Since the beginning of 2017, all motors with a rated output power of between 0.75 kW and 375 kW must <u>either</u> reach efficiency level **IE3** or correspond to efficiency level **IE2 and additionally have speed control**.

There is speculation that more stringent target efficiencies may be imposed in 2020 … we shall see !

The directive defines a 'motor' as an *'electric single speed, three-phase 50 Hz or 50/60 Hz, squirrel cage induction motor that has 2, 4 or 6-poles, a rated voltage of up to 1000 V, a rated output between 0.75 kW and 375 kW, rated on the basis of continuous operation*'.

REGULATION EU 2011/327 FOR FANS

This regulation applies to fans with induction motors with an electrical input power between 125 W and 500 kW.

It came into force in 2011 and prescribes minimum target efficiency requirements (corresponding to system efficiency) in two steps.

- Stage 1 - became effective at the beginning of 2013
- Stage 2 - which has been in force since the beginning of 2015 where these values were increased again
- Stage 3 - From 1. January 2017 electrical motors with a rated output of 0,75 to 375 kW shall not be less efficient than the IE3 level, or shall meet the IE2 level whilst used with a variable speed drive

The system efficiency of a fan unit represents the product of the efficiencies of the fan, motor, drive (V-belt, flat belt or direct) and speed control and is specified by the manufacturer.

The system efficiency must be at least equal to or greater than the target efficiency. In Annex I 2, Table 2 of the Directive provides equations for the target efficiencies $\boldsymbol{\eta}_{target}$ are given for each fan type, which can then be calculated depending on the electrical input power and specified efficiency levels.

In view of the above motor Regulation 2009/640, in our case all fans driven by 2, 4 & 6 pole AC motors (nominally 2800 to 900 rpm / 50 Hz) fall under the scope of the ErP Directive.

Exceptions

- Fan/motors running slower than 6 pole speed i.e. 8, 10, 12 & 16 pole motors (e.g. <750 rpm @ 50 Hz or < 850 rpm @ 60 Hz)
- ATEX applications
- AOM > 100°C
- AOM < -40°C
- Supply > 1000 VAC

Furthermore, as a consequence of the definition of a motor given above, all 'EC external rotor motor driven fans' are outside of the scope of the ErP, but in terms of their operating efficiency, well exceed the minimum 2015 efficiency targets and thus will be badged as 'ErP ready' by fan manufacturers.

EXTRACT FROM DIRECTIVE 2009/125/EC

The table below details the 'current in force' minimum energy efficiency requirements for axial fans.

Table 2

Second tier minimum energy efficiency requirements for fans from 1 January 2015

Fan types	Measurement category (A-D)	Efficiency category (static or total)	Power range P in kW	Target energy efficiency	Efficiency grade (N)
Axial fan	A, C	static	$0{,}125 \leq P \leq 10$	$\eta_{target} = 2{,}74 \cdot \ln(P) - 6{,}33 + N$	40
			$10 < P \leq 500$	$\eta_{target} = 0{,}78 \cdot \ln(P) - 1{,}88 + N$	
	B, D	total	$0{,}125 \leq P \leq 10$	$\eta_{target} = 2{,}74 \cdot \ln(P) - 6{,}33 + N$	58
			$10 < P \leq 500$	$\eta_{target} = 0{,}78 \cdot \ln(P) - 1{,}88 + N$	

In EAS's case ***Measurement category A*** *applies and is defined as 'an arrangement where the fan is measured with free inlet and outlet conditions'.*

BACKGROUND

Fans are used as the means to move air through a coil or a product and can be either axial, radial, centrifugal or mixed flow.

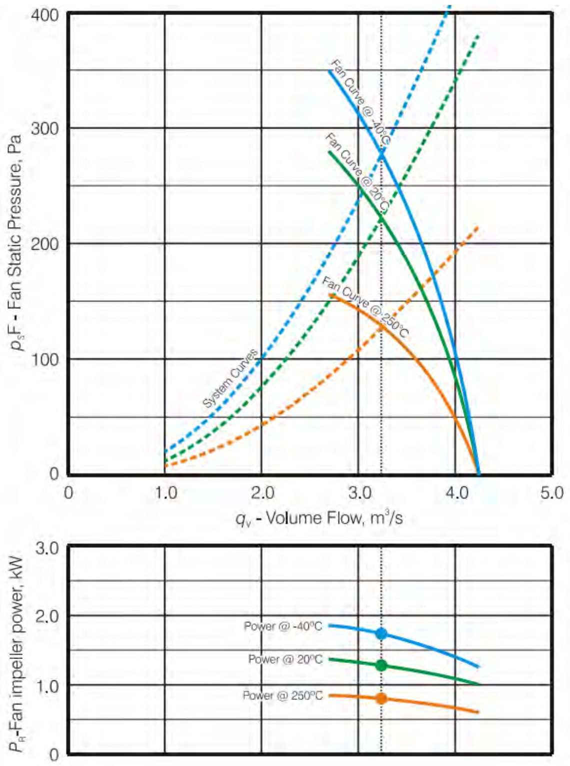
In equipment such as air handling units, centrifugal and mixed flow fans are often used, whereas products such as cooling towers and closed circuit coolers/condensers, tend to use large diameter axial or centrifugal fans. However, there is trend towards using multiple smaller diameter fans.

Historically, dry coolers and air cooled condensers used large diameter belt driven or gearbox driven axial fans. If the hot air was not being dumped to atmosphere, then belt driven centrifugal fans were used to duct the air to the required location.

Around 40 years ago saw the increased popularity of smaller direct driven axial impellers where variable voltage controllers provided a convenient method of fan speed control and thus capacity control.

More recently, with more competitively priced frequency inverters and the development of EC motor technology, compact fan/motor ranges have evolved, which simplifies the construction of dry coolers and air cooled condensers.

BASIC THEORY



The performance of a fan changes with the density of the gas being handled. Usually fan characteristics are published at 'standard air' conditions where the air density is typically 1.2 kg/m^3 and rated at 20°C & 101.325 kPa.

At any other condition, the fan's absorbed power requirement and its ability to develop pressure will vary.

Fundamentally, a fan is a '*constant volume machine*' and for a given speed will deliver the same air volume irrespective of the air temperature. So whether the temperature is -40°C or +500°C, at a given speed, it will deliver the same air volume range, but against difference static pressures.

As an example, the **green** curves show the performance of an axial fan with a duty of 3.25 m^3/s @ 220 Pa and 'standard air' @ 20°C, where the absorbed power indicates 1.35 kW.

The **blue** and **orange** curves indicate the behaviour at temperatures of –40ºC and 250ºC.

At 250°C the density of air is approximately half that at 20°C and as a consequence, the fan will only be able to create half the static pressure (123 Pa) and velocity pressures. Furthermore, the delivered air mass flow rate (*volume x density*) at 250°C will be halved too, thus the power required to move this lower mass of air will be halved too, around 0.75 kW.

The reduction in static pressure is proportional to the reduction in absorbed power and consequently the overall fan efficiency will remain unchanged.

Besides temperature affecting air density, altitude (*barometric pressure*) has the same impact and affects fan behaviour too. So for a given speed the fan will deliver the same air volume, as described above, but at a reduced static pressure. So if the static resistance of the system at altitude does not change in sympathy with the reduction in air density, the result may be an under-performing fan.

*Finally, moist air ... %RH > 0 ... has a **lower density** than dry air at the same temperature & pressure. This may, at first sight, seem odd because 'everyone knows that water is heavier than air, so if a moist air mixture contains water, then surely it must be heavier !!'*

But moist air does not contain 'liquid' water, but 'water vapour' ... water in its gaseous state ... which at its low partial pressure, exists as ***superheated steam****. The gaseous state of water i.e. steam, has a lower density that liquid water and indeed dry air. Thus, a moist air mixture has a nett lower density than dry air. So from the fan's perspective, the pressure generating ability is lower with moist air than with dry air.*

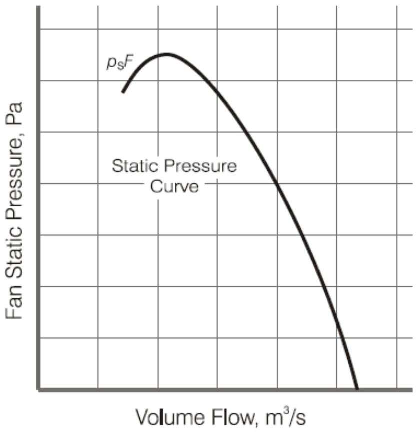
Although a fan's behaviour will change because of changes in the air density at varying temperatures and pressures, the static resistance of a component such as a coil will also change in perhaps a similar fashion. Thus it is normal practice to derive the operating air volume of the fan and coil etc. at standard temperature and pressure and thus air density (1.2 kg/m^3).

However, having arrived at the air volume for the system and remembering that for a given speed the air volume component of the fan characteristic is independent of temperature or pressure, the 'real' air density at the fan (*corrected for temperature and pressure*) must be used in the thermal calculations to ensure that the correct 'air mass flow rate' is considered.

Remember … assuming no losses or leakages in a system, the mass flow rate is constant, but the volumetric flow rate will vary if the temperature changes. So the mass flow rate in a system is governed by the air volume that the fan delivers multiplied by the air density at the temperature at the fan.

TERMINOLOGY

STATIC PRESSURE



Static Pressure is the difference between the pressure at a given point and atmospheric pressure.

An example of Static Pressure is the pressure that inflates a balloon; it acts in all directions. However, in a system, Static Pressure is a measure of the resistance to the flow of air caused by the coil and enclosure and it acts in all directions.

For air to pass through a coil or product, requires an amount of energy, or Static Pressure, to overcome the frictional losses. The magnitude of the energy requirement is dictated by the air volume and size, which in turn governs the air speed and resultant resistance. The higher the air volume required the greater the amount of Static Pressure required.

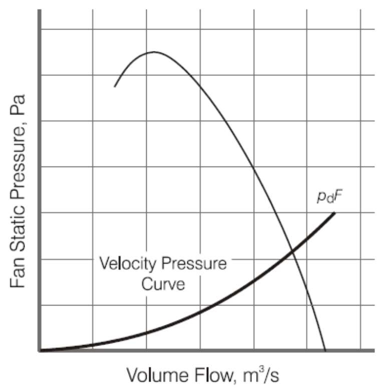
Static Pressure can have a positive or negative value. A negative value of the pressure would be found at the coil inlet or upstream side of the fan if the fan is inducing (sucking) the air through the coil/system.

The value of the Static Pressure at the fan inlet is the measure of the energy required to overcome the system resistance and would be a positive value on the discharge or downstream side of the fan, if the fan has enough energy, or Static Pressure, to blow the air through the coil/system.

The value of Fan Static Pressure is denoted by **P_sF** and measured in Pascals, Pa.

Static Pressure is the more common and simpler parameter used to select fans; however, the Total Pressure can be argued to be the 'more correct' parameter, albeit that total pressure v's volume fan characteristics are less commonly published. Nevertheless, using Static Pressure provides a small factor of safety in the selection.

VELOCITY PRESSURE



Air naturally flows from a region of high pressure to one of lower pressure and its velocity depends upon the resistance encountered.

In a coil/system, the Velocity Pressure, also known as the Dynamic Pressure, is calculated from the air volume and the cross-sectional area (face area) of the coil and can be plotted on the fan curve as shown.

Velocity Pressure is very important, as its value is used to determine the pressure losses throughout the system.

The value of Velocity Pressure is denoted by **P_dF** and measured in Pascals, Pa.

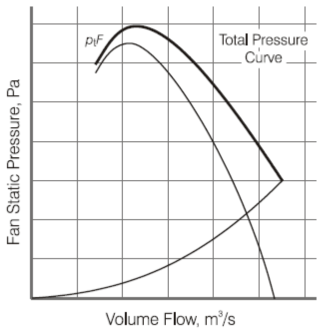
However, the Fan Velocity Pressure is determined using the same formula but is based upon the velocity at the **fan discharge** as opposed to somewhere else in the system where the velocity may vary due to changes in the cross-sectional area.

Velocity Pressure is calculated as follows …

$P_d = 0.5 \times \rho \times v^2$ *where,* *P_d = velocity pressure, Pa*
ρ = air density, 1.2 kg/m3 (Standard air density at 20°C & 101.325 kPa atmospheric pressure)
v = air velocity, m/s (based upon the applicable area)

TOTAL PRESSURE

The Static Pressure developed by a fan is the pressure it can create in order to move air against a resistance, but all air movement involves some degree of Velocity Pressure and Static Pressure according to the resistance of the system.

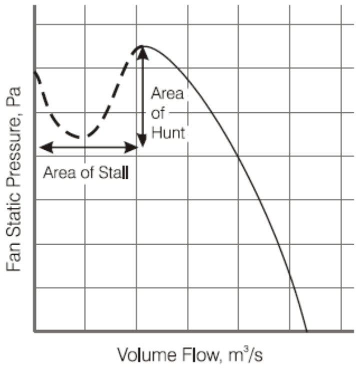


The Total Pressure developed by a fan is the sum of the Static Pressure and Velocity Pressure.

$P_TF = P_sF + P_dF$ *where,* *P_TF = total pressure, Pa*
P_sF = static pressure, Pa
P_dF = velocity pressure, Pa

STALL

Stall occurs when the flow of air over an impeller blade breaks down and turbulence is generated, causing the blades to lose their ability to maintain lift (pressure) and thus deliver the required air volume.



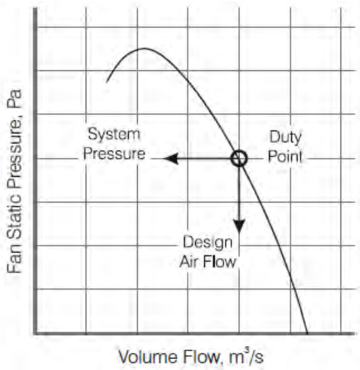
Axial fans with medium to high pitch angles are particularly prone to stall.

The area of Stall on a fan curve is anywhere to the left of the Peak Pressure point and is one reason why fans should never be selected to operate near this point.

When in Stall, a fan, apart from not performing as required, generates more noise, usually with a characteristic rumble. Occasionally there can be a tendency for the performance from the fan to 'run' up and down the curve, creating a condition called 'hunting', which can acoustically be undesirable and may cause physical damage to the impeller.

SYSTEM PRESSURE

System Pressure, also known as System Resistance, is the sum of all the pressure losses created by the system components, both before and after the fan such as coils, pre-cooling pads, discharge ducting, attenuators etc.

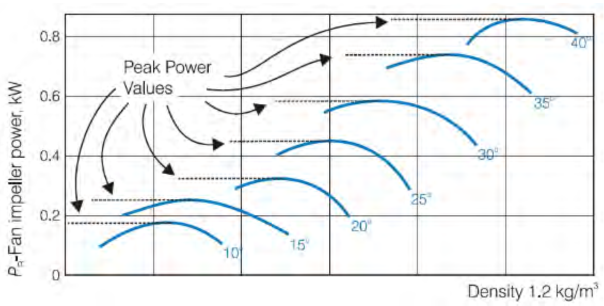


System Pressure is expressed in Pa, but always in conjunction with the airflow used to determine the pressure losses through the system.

The System Pressure is actually a Total Pressure, but in practice the value is taken as the Fan Static Pressure. The difference between the Fan Static Pressure and the System Pressure is the difference in the velocity pressures at the fan discharge and at the end of the system. Because the velocity at the discharge of a system is generally quite low, the difference provides a small safety margin.

PEAK POWER

Peak Power is the term used to describe the maximum absorbed power a fan of a given diameter, will require under a given condition.



In most instances, the fan speed is fixed for the assessment, but, in the case of axial fans, the blade pitch angle will have an impact.

Peak Power is expressed in Watts (W) or Kilowatts (kW). The diagram shows the absorbed power curves for an axial flow fan. Each curve is for a specific pitch angle of the fan blade and the highest absorbed power value for each curve is what is termed the Peak Power.

Depending upon the impeller design and shape, the power curve from free air to peak pressure can be fairly flat; the variation may be as low as 10-20%. These fans are commonly referred to as 'non-overloading' as, the motor can be selected to accommodate the power requirements at all pressures at the selected fan speed.

Non-overloading fan types include most axial flow, backward-curved centrifugal and mixed-flow fans.

Forward-curved fans have a steep power curve with the maximum power being absorbed at the free air condition, i.e. when the fan is not working against any resistance. Such fans are considered as overloading fans and the motor is generally selected to accommodate the duty power requirements plus a safety margin.

FAN EFFICIENCY

Fan Total Efficiency is based upon the total pressure and Static Efficiency is based on the static pressure, however, only the Total Efficiency is the true efficiency.

If a fan was delivering air against zero Static Pressure, its Static efficiency would be zero. This is obviously incorrect as the fan is moving air and must therefore have some magnitude of efficiency. Nevertheless, the term is used.

The efficiency is in effect, a measure of the quality of the design of the impeller and the fan casing. It is based upon the airflow being handled, the pressure being developed, and the amount of energy required to do that work. It is worth noting that the efficiency value quoted does not generally take into account the efficiency of the motor driving the fan, yet this can have a significant impact upon the energy consumed, particularly with small motors which can be very inefficient.

This is where the ErP is beginning to have an impact.

Fan Total Efficiency is defined as follows …

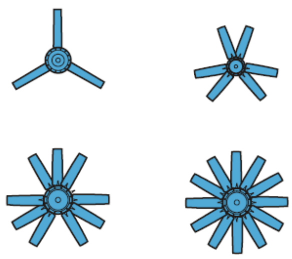
$$\text{Fan Total efficiency, \%} = \frac{q_v \times p_t F}{10 \times P_R}$$

where, *q_v = volume flow rate, m³/sec*
$P_T F$ = fan total pressure, Pa
P_R = power absorbed by the fan, kW

To calculate Static Efficiency simply change the Fan Total Pressure, $P_T F$ value for the fan static pressure, $P_s F$ value.

FRACTIONAL SOLIDITY

The term Fractional Solidity refers to the number of blades which are fitted to a given impeller hub, which can be varied to suit the needs of the application.

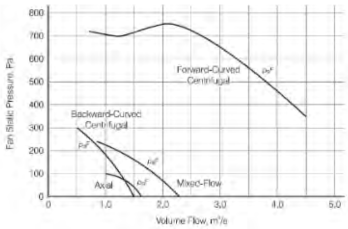


With fewer blades the peak pressure development is reduced, and the peak efficiency point moves to a lower pressure. The characteristics of the noise levels generally alter as well.

By fitting fewer blades there is a saving in cost and on occasions, a motor with a lower kW output can be fitted as the lower solidity impeller may be a more efficient selection for the particular duty than an impeller with a greater number of blades.

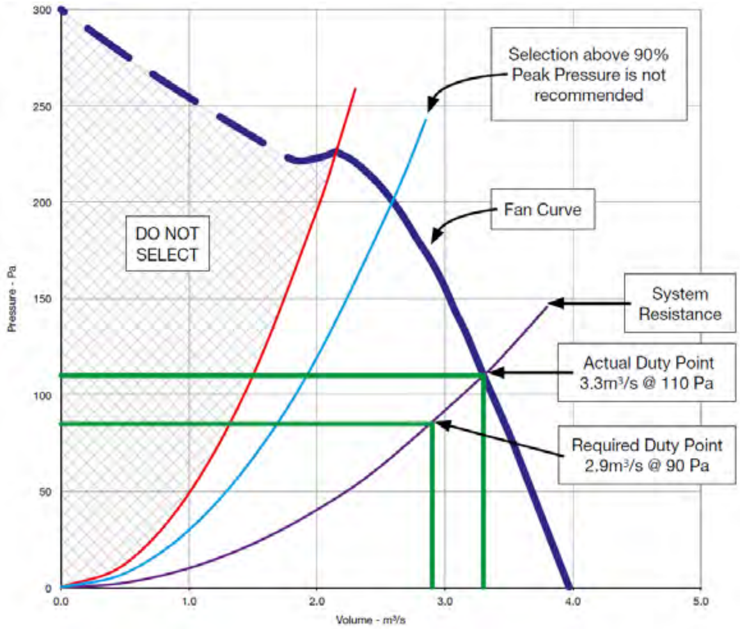
COMPARISON OF ALTERNATIVE FAN PERFORMANCES

Different types of fans exhibit slightly different characteristics and are able to cope with different ranges of air volume and static pressures, as shown.



SELECTING A FAN FROM A PERFORMANCE CURVE

Ensure that the operating Duty Point; which is the intersection of the fan curve with the system curve; is below the stall point … <u>do not</u> operate to the left of the red line.



REVERSAL OF AIRFLOW

If the rotation of an axial flow fan impeller is reversed, the direction of the airflow will also be reversed, however, the blades of the impeller will be operating tail first and have the camber, or aerofoil shape, in the wrong direction. As a consequence, the airflow will be reduced by approximately 30%.

In addition, the pressure development capability will be reduced by approximately 55% and the power absorbed by 25%.

AXIAL FANS WITH GUIDE VANES

Guide Vanes can be fitted either on the upstream or downstream side of the axial flow fan and effectively change the shape of the performance curve.

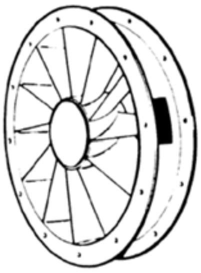
Upstream Guide Vanes

Upstream Guide Vanes are fitted to the air inlet side of the fan and cause the air to swirl in a manner that assists the fan by introducing the air to the fan blade at an optimum angle. This action enables the fan to generate substantially more pressure for a given airflow. To be effective, they have to be very close to the fan blade and this can result in a dramatic increase in noise levels. An increase in pressure development from 20 to 60% is achievable with a corresponding increase in power absorbed.

A well-designed unit should ensure the air leaves the impeller in an axial direction, i.e. there is no swirl.

Although the potential increase in pressure development is very attractive, the noise level increase generally means they are avoided. They can also be more expensive than going for a one size larger fan.

Downstream Guide Vanes



Downstream Guide Vanes are more commonly used than Upstream Guide Vanes. They improve the pressure development by 10 to 25% with 20% being the more usual improvement. There is no increase in the power absorbed by the fan as the guide vanes do their work downstream of the impeller, therefore there is a corresponding increase in fan efficiency. However, downstream guide vanes, to be really effective, require a considerable swirl component imparted by the impeller to enable the guide vanes to work effectively. As a result, guide vanes should not be used for pitch angle selections below 20º.

The noise level increase is generally only 1-2 dB above the impeller only selection. In addition to improving the pressure development and efficiency of the fan, the guide vanes remove the swirl component generated by the axial impeller on the air; this is particularly important when the fan is discharging into a long high velocity duct. Swirl tends to persist for a considerable distance in such situations thereby generating substantially higher pressure losses through the system. As with upstream guide vanes, a fan size larger can sometimes meet the duty at a lower cost.

FAN LAWS

The Fan Laws enable the prediction of the performance a fan under different conditions and as a result, the Laws are basically a ratio of known data to the data at another condition.

The main Laws are centred upon the airflow, pressure development and power absorbed, where 4 parameters affect these values as follows …

- Fan speed
- Impeller diameter
- Air temperature
- Barometric pressure

The Fan Laws can only be used to their full extent under special circumstances. For example, to determine the performance of a fan of a different diameter the new fan must be an exact scale model of the original and thus be 'Geometrically Similar" or part of a 'Homologous Series'.

However, in our case, only the speed, air temperature or barometric pressure vary, so the Laws can be applied fully and the 'diameter' element ignored from the equations.

The impact of the air temperature affects the air density (*in line with the **Gas Laws***) with respect to its absolute temperature, K (°C + 273.15), which in turn affects the air mass flow handled by the fan.

VOLUME FLOW

The volumetric flow rate is directly proportional to the speed and the diameter cubed …

$\dot{V} \propto n \times d^3$ *where,* $\dot{V}$ *= volume flow rate, m³/s*
n = fan speed, rpm
d = impeller diameter, mm

This transposes into a formula as follows …

$$\dot{V}_2 = \dot{V}_1 \times (n_2 / n_1) \times (d_2 / d_1)^3$$

However, practically because the diameter is fixed, rationalises to the familiar form …

$$\dot{V}_2 = \dot{V}_1 \times (n_2 / n_1)$$

PRESSURE

The pressure is directly proportional to the density, speed squared and the diameter cubed …

$P \propto \rho \times n^2 \times d^2$ *where,* *P = pressure, Pa*
ρ = density, kg/m³
n = fan speed, rpm
d = impeller diameter, mm

This transposes into a formula as follows …

$$P_2 = P_1 \times (\rho_2 / \rho_1) \times (n_2 / n_1)^2 \times (d_2 / d_1)^2$$

Because the air density is proportional to the barometric pressure and inversely proportional to the absolute temperature, then the densities in the above equation can be replaced with the temperature in Kelvin, plus as the diameter is fixed, the formula becomes …

$$P_2 = P_1 \times (T_1 / T_2) \times (B_2 / B_1) \times (n_2 / n_1)^2 \times (d_2 / d_1)^2$$

where, T_1 = *absolute temperature,* t_1*(°C)+ 273.15*
T_2 = *absolute temperature,* t_2*(°C)+ 273.15*
B = *barometric pressure, kPa or bar*

However, assuming no change in altitude/barometric pressure or air temperature and thus density, the equation further rationalises to the more familiar form …

$$P_2 = P_1 \times (n_2 / n_1)^2$$

ABSORBED POWER

The absorbed power is directly proportional to the density, speed cubed and the diameter to the power 5 …

$$kW \propto \rho \times n^3 \times d^5$$

where, kW = *absorbed power, kW*
ρ = *density, kg/m*^{3}
n = *fan speed, rpm*
d = *impeller diameter, mm*

This transposes into a formula as follows …

$$kW_2 = kW_1 \times (\rho_2 / \rho_1) \times (n_2 / n_1)^3 \times (d_2 / d_1)^5$$

Because the air density is proportional to the barometric pressure and inversely proportional to the absolute temperature, then the densities in the above equation can be replaced with the temperature in Kelvin, plus as the diameter is fixed, the formula becomes …

$$kW_2 = kW_1 \times (T_1 / T_2) \times (B_2 / B_1) \times (n_2 / n_1)^3 \times (d_2 / d_1)^5$$

where, T_1 = *absolute temperature,* t_1*(°C)+ 273.15*
T_2 = *absolute temperature,* t_2*(°C)+ 273.15*
B = *barometric pressure, kPa or bar*

However, assuming no change in altitude/barometric pressure or air temperature and thus density, the equation further rationalises to the familiar form …

$$kW_2 = kW_1 \times (n_2 / n_1)^3$$

FAN EFFICIENCY

The fan efficiency can be described in terms of the Total or Static efficiency, albeit that the Total Efficiency is the correct value as it caters for the velocity pressure component.

$$P_d = 0.5 \times \rho \times v^2$$

where, P_d = *velocity pressure, Pa*
ρ = *air density, kg/m3*
v = *air velocity, m/s*

$$P_TF = P_sF + P_dF$$

where, P_TF = *total pressure, Pa*
P_sF = *static pressure, Pa*
P_dF = *velocity pressure, Pa*

Thus the Fan Total Efficiency is given by …

$$\zeta = \dot{v} \times P_TF / (10 \times kW)$$

where, ζ = *fan efficiency, %*
$\dot{v}$ = *volume flow rate, m*^{3}*/sec*
P_TF = *fan total pressure, Pa*
kW = *power absorbed by the fan, kW*

and the Fan Static Efficiency is given by …

$$\zeta = \dot{v} \times P_sF / (10 \times kW)$$

where, ζ = fan efficiency, %
$\dot{v}$ = volume flow rate, m^3/sec
P_sF = fan static pressure, Pa
kW = power absorbed by the fan, kW

SOUND POWER LEVEL

The sound power level is usually calculated from physical sound pressure measurements conducted in the wind tunnel where the volume/pressure characteristic is measured.

The following, however, is a correlation which indicates the sound power level of a fan at a specific operating condition.

Beraneck equation … an estimate of base sound power level for an axial flow impeller …

$$L_wA = K + 10\, Log_{10}(\dot{V}) + 20\, Log_{10}(P_TF)$$

where, $\dot{v}$ = volume flow rate, m^3/sec
P_TF = fan total pressure, Pa
K = 40 for Aerofoil axial impellers
38 for broad chord/paddle impellers
37 for Sickle impellers

Another, generalised correlation from API Guidelines is …

$$L_wA = 56 + 30\, Log_{10}(U \times 197 / 1000) + 10\, Log_{10}(kW / 0.746)$$

where, L_wA = sound power level, dBA
U = impeller tip speed, m/s
kW = fan absorbed power, kW

If the sound power level of a fan is known at a given condition, then the sound power level at a different operating point can be predicted using the following …

$$L_wA_2 = L_wA_1 + 70\, Log_{10}(d_2/d_1) + 55\, Log_{10}(n_2/n_1) + 20\, Log_{10}(P_2/ P_1)$$

where, L_wA = sound power level, dBA
d = fan diameter, mm
n = fan speed, rpm
P = fan total pressure, Pa

However, in our case the fan diameter is not usually a variable and increases in system resistance i.e. total pressure, are not usually a consideration either … only the change in speed, applicable to products with frequency inverter speed control or fitted with EC fans.

So the above equation simplifies to …

$$L_wA_2 = L_wA_1 + 55\, Log_{10}(n_2/n_1)$$

Although the above equation uses a constant of **55**, this actually changes for different impeller and product configurations and can be as low as **45** on occasions. Field tests would confirm the magnitude for a given product and fan permutation.

COMPACT FANS

Compact fans generally refer to a ‘one piece’ component comprising impeller, motor, fan shroud and motor support/grille arrangements, which are usually purchased with a square ‘wall plate’ for simple attachment to our products via 4 off corner bolts.

Such fan/motor assemblies usually comprise **external rotor motors** and are offered with single or 3 phase AC or EC motors.

Clearly the EC motor variants provide a simple method of speed control either via a 0-10V signal or indeed digitally, via Modbus.

Unlike AC IEC squirrel cage induction motors, this concept of motor involves the conventional ‘rotor’ becoming the stationary part of the motor and is usually bolted to the motor support bracket, which often acts as the fan grille/guard too. The ‘stator’ becomes the rotating part of the motor to which the impeller blades are attached.

The ‘inside out’ arrangement results in a very compact design and is often considered as a disposable component i.e. if it fails it is usually replaced in total rather than replacing the faulty part.

Compact fans can either be fitted with AC motors or alternatively EC motors, which in turn, can be either 3 phase or single phase. The EC variant having the added bonus that they are suitable for 50 or 60 Hz electrical supplies.

EAS generally favours 3 phase motors, albeit if required, single phase alternatives can be supplied for special projects.

Currently, three compact fan manufacturers are used, namely EBM PAPST, Ziehl-Abegg and Solar Palau albeit that there is a wealth of other manufacturers of similar types of fans.

Compact fan ranges usually conform to the ISO range of sizes of which Ø500, Ø630, Ø710, Ø800, Ø910 are typically used. Both smaller and larger ISO diameters are available, but these are used for special projects.

Somewhat larger diameters up to Ø1600 mm fitted with up to12 kW motors are available, albeit that currently they tend to be rather expensive when compared with the alternative of using a WingFan impeller with an IEC motor combination.

When reading data from a published compact fan performance curve, it is important read the ‘small print’ with respect to how the data is presented. For example, the volume v’s pressure characteristic is often shown for the fan/motor combination **without** the fan guard fitted.

Clearly, in our case, the guard will almost always be fitted and thus its static resistance will slightly reduce the fan characteristic. The fan manufacturer usually provides a simple equation to apply this deration to the curve.

Furthermore, the fan guard creates a disturbance in the discharge air flow and besides the pressure drop implications, also increases the noise level of the fan compared with the fan without the guard. Depending upon the fan diameter, this increase can be up to +3dBs.

*One important consideration is that published fan curves; besides not showing the pressure drop implications of the guard; also **only** present the **suction side** sound power level, usually presented as **L_wA_5**. Although this data may indeed be correct from the manufacturer’s perspective, the data needs to be adjusted to enable it to be used for any product related sound power or pressure predictions.*

During the testing of a fan, it is usually mounted in some sort of fan plate/cowl located in the dividing wall separating the suction and discharge sides of the wind tunnel/sound chamber. Usually both the suction and discharge side sound power levels are derived (*from sound pressure level measurements*), however, only the suction side data is usually presented.

Clearly, for a ‘wall plate’ mounted fan application, the suction side data is perhaps acceptable, but in the case of a dry cooler or condenser, sound energy is both emanating from the discharge side of the fan as well as from the suction side of the fan.

Thus, the overall sound power level of the fan that must be considered in any calculations, which is the logarithmic addition of the suction and discharge sound power levels. Typically these two sound power levels are very similar and thus it is usual to consider the following …

$$L_wA_{Total} = L_wA_5 + 3$$

where, L_wA_{Total} *= Overall sound power level, dBA*
L_wA_5 *= Suction side power level, dBA*

Although the above accounts for the suction and discharge components of the fan sound power, the fan guard will increase this value by typically …

- <=Ø630 mm +3 dB
- Ø710 to Ø910 mm +2 dB
- >= Ø1000 mm +1 dB

In addition to this theoretical addition of the impacting sound power aspects, the biggest ‘unknown’ is the ***unit related regenerative*** sound component, which is related to the product design, stiffness of the construction and directivity implications of the product.

Experience and confirmation backed up by sound measurements has shown that this ‘unknown’ can be as high as +5dBs on top of all the above factors.

In conclusion, if a purely theoretical prediction of the sound level of a product is conducted based upon only published manufacture’s data, then for a Ø910 mm fan the following is suggested …

$$\mathbf{L_wA_{Total}} = L_wA_5 + 3 + 2 + 5$$

$$\mathbf{= L_wA_5 + 10\ dBA}$$

Compact AC fans are usually fitted with motors that are suitable for either Star or Delta connections and thus able to run at two discrete speeds along with their associated volume characteristic. However, unlike the EC alternatives, not all fan/motor permutations are suitable for 50 or 60 Hz operation.

When a 50 Hz AC motor is supplies with 60 Hz supply, the motor will operate at typically 20% higher speed. For a ‘stand-alone’ motor, this is not an issue, but as compact fans incorporate an integral impeller, an increase of 20% in the speed results in a 72.8% increase in the impeller’s absorbed power. However, even though at 60 Hz the motor produces around 20% more motor power, it is not able to cope with a 72% increase and would probably become vastly overloaded and eventually burn-out. However, there are a few compact fan options that are suitable of the dual frequency option, but these are the exception !

Another important consideration of **compact AC fans** is that the name-plate data refers to the worst case operating condition, which for an axial fan is at the highest static pressure prior to the ‘stall point’. So the published speed (rpm), consumed power (P1), full load current (FLC) etc. relate to this worst case condition.

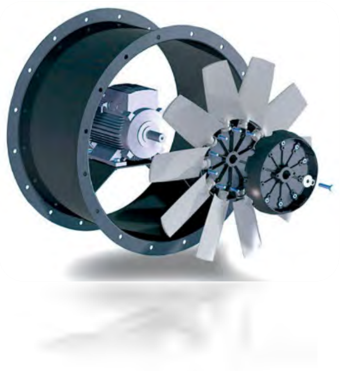
When an axial AC compact fan is operating below its maximum static pressure, the fan speed (rpm) will increase and both P1 and the FLC will reduce.

Thus, quoting the nameplate data should be ‘on the safe side’ and cover most operating conditions.

Unlike AC fans, a given **EC fan** is suitable for a range of supply voltages and frequencies e.g. 3 x 380 – 480V : 50/60 Hz (-10%/+15%) and any combination of voltage or frequency in this range will result in the fan running at the same speed, if provided with a 10V control signal or instructed digitally to run at full speed.

Furthermore, it is worth bearing in mind that as an EC fan/motor is able to maintain its speed as the static pressure is increased, the published volume v’s pressure characteristic is at a constant speed, unlike the AC alternatives which publish a characteristic where the speed increases as the air volume increases, and the static pressure reduces.

INDUSTRIAL FANS



The differentiator between Compact and Industrial fans is that when we purchase a Compact fan it is the responsibility of the manufacturer to ensure that the assembly is ErP compliant.

In the case of the Industrial fan option, where we purchase an impeller and IEC or NEMA motor and assemble the items to form a fan/motor combination, then EAS is considered a ‘Fan Manufacturer’ and thus is responsible for ensuring that the fan/motor assembly is ErP compliant.

The Industrial fan/motor option allows for greater flexibility when selecting a fan to perform a certain job. For example ATEX environments or when unusual supply voltages or larger than normal fluctuations in supply voltages and frequencies, which would otherwise adversely affect ‘standard’ compact fan solutions.

Furthermore, high and low temperature applications can call for special impeller materials, low/high temperature grease, anti-condensation heaters, all of which cannot always be offered with the compact fan ranges.

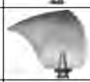
WINGFAN VERSUS MULTIWING

BACKGROUND

WingFan and MultiWing (*formerly F.S. Andersen*) had a working cooperation between 1960 and 2000, where a range of impellers were jointly developed at the R&D facility at F.S. Andersen. Following the termination of the agreement in 2000, both companies continued to produce both the ‘legacy’ impellers and proceeded to develop their own blade profiles and design features.

EAS worked extensively with MultiWing until 2010, thereafter WingFan was chosen as the preferred supplier. Since that date EAS & WingFan have had a close working relationship.

MULTI-WING		Schaufeltyp Blade type	WingFan		Schaufeltyp Blade type
H-Baureihe / H-Hubs					
2H		Tragflügel Airfoil	**P2H**		Tragflügel Airfoil
3H		Tragflügel Airfoil	**P3H**		Tragflügel Airfoil
1H		Sichel Sickle	**S1H**		Sichel Sickle
4H		Sichel Sickle	**S14H**		Sichel Sickle
6H		Kreisbogen Circular arc	-		

Z-Naben / Z-Hubs					
3Z		Tragflügel Airfoil	-		
4Z		Tragflügel Airfoil	**P4Z**		Tragflügel Airfoil
5Z		Tragflügel Airfoil	**P5Z**		Tragflügel Airfoil
-			**P6Z**		Tragflügel Airfoil
1Z		Sichel Sickle	**S2Z**		Sichel Sickle
2Z		Sichel Sickle	**S4Z**		Sichel Sickle
7Z		Sichel Sickle	-		
6Z		Kreisbogen Circular arc	**K6Z**		Kreisbogen Circular arc

MULTI-WING		Schaufeltyp Blade type	WingFan		Schaufeltyp Blade type
			K7Z		Kreisbogen Circular arc
Z-Rev		Reversibel Reversible	**R4Z**		Reversibel Reversible
TR7Z		Reversibel Reversible	-		
TR8Z		Reversibel Reversible	-		

W-Naben / W-Hubs					
5W		Tragflügel Airfoil	-		
6W		Tragflügel Airfoil	P8Y		Tragflügel Airfoil
-			P8T		Tragflügel Airfoil
7W		Tragflügel Airfoil	P7T		Tragflügel Airfoil
9W		Tragflügel Airfoil	P9T		Tragflügel Airfoil
-			S45Y		Sichel Sickle
1W		Sichel Sickle	S5Y		Sichel Sickle
2W		Sichel Sickle	S6Y		Sichel Sickle

MULTI-WING		Schaufeltyp Blade type	WingFan		Schaufeltyp Blade type
3W		Sichel Sickle	-		
8W		Kreisbogen Circular arc	U8Z		Kreisbogen Circular arc
TR11W		Reversibel Reversible	-		

BLADE MATERIAL OPTIONS

Symbol	Material description	Temperature Range	Characteristic	available	Application Suitability
PA	Glass fiber reinforced polyamid(nylon 6, black)	-40°C to +110°C	Heat ageing stabilized	P4Z, P5Z, P6Z	Standard Duty
PAG	Glass fiber reinforced polyamid(nylon 6, beige)	-40°C to +110°C	Heat ageing stabilized	P4Z, P5Z, P6Z	Heavy Duty
PAGST	Glass fiber reinforced polyamid(nylon 6, black)	-40°C to +110°C	Extremely vibration resistant, high impact strength	P4Z, P5Z	where extreme vibrations may occur
PACAS ***	Carbon fiber reinforced polyamid(nylon 6, black)	-35°C to +100°C	Electrically conductive, flame-retardant	P4Z, P5Z, P6Z	Duty where explosions may occur
ALU	Die cast aluminum	-40°C to +150°C	Greater temperature range	P4Z, P5Z	High temperature

* P4Z available with 20° to 45° pitch angles, P5Z and P6Z available with 25° to 50° pitch angles.
P4Z aluminium blades currently available for clockwise and counter clockwise rotation
P5Z aluminium blades currently available only for clockwise rotation

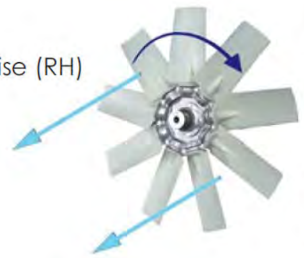
** For heavily corrosive atmospheres, the aluminum hubs can be supplied with a protective coating and stainless steel bolts and nuts.

*** For European compliance with regulations ATEX 100 and VDMA 24169, hubs are available with three layers of conductive paint.

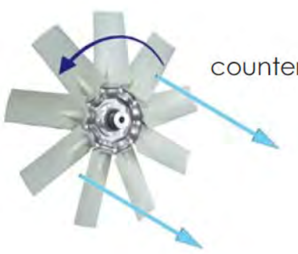
ROTATION CONVENTION

Direction of Rotation

clockwise (RH)



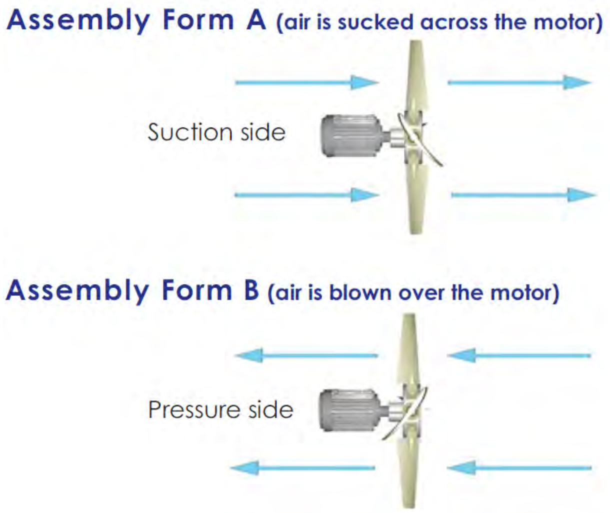
To determine the direction of rotation, the air must blow into the face of the observer. If the rotation is clockwise, then the direction of rotation is right handed – if counter clockwise, then left handed.



counter clockwise (LH)

AIR DIRECTION

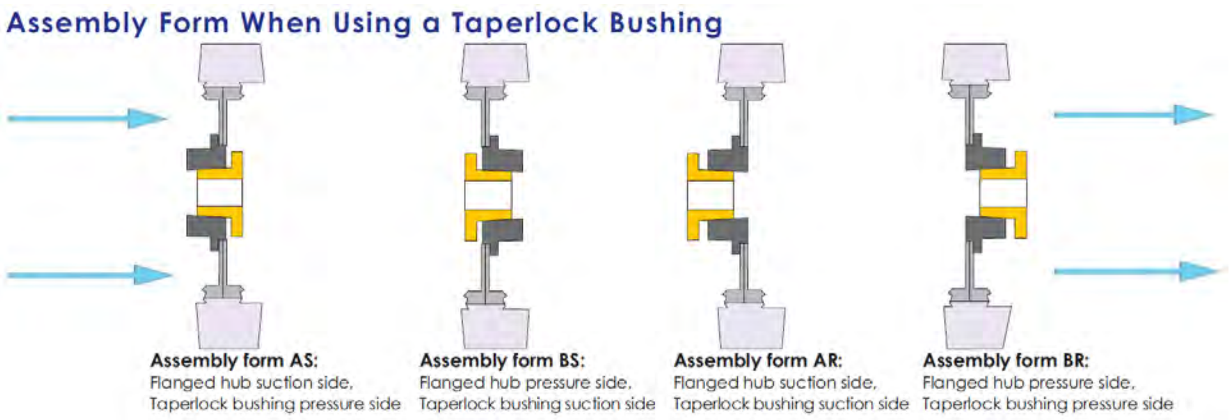
Note : Assembly form A is the standard induced draft arrangement used for EAS's FSW & VFW ranges



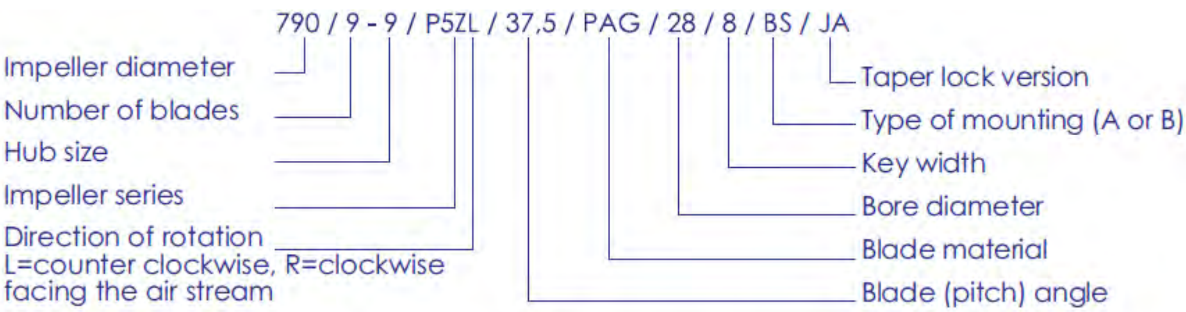
The assembly form is an indication of how the impeller should be fitted to the motor shaft. If the air is sucked across the motor (the drive motor is upstream of the impeller), this is described as "Assembly form A".

If the drive motor is on the pressure side of the impeller (the drive motor is downstream of the impeller), then we have "Assembly form B".

It is important to specify form A or form B to ensure that the impeller is assembled for correct airflow direction.



IMPELLER DESIGNATION



*For flange mounted impellers, the following information is required:

- Spigot hole diameter
- The number and size of bolt holes including bolt circle dimension (BCD).
- Additional information may be found at the website www.wingfan.com

BALANCING AND BORE TOLERANCE

All impellers < Ø1500 mm diameter are balanced to G6.3 (ISO 1940), whilst Ø1500 mm impellers are balanced to G6.2

Bore hole tolerance is now F7 (agreed 2011) … earlier was H7 and then G7, but both tolerances caused assembly issues with the Ø42 mm bosses for the Ø1500 mm impellers.

MOTOR SHAFT DETAILS

Frame Size	Shaft - mm		Tolerance	Woodruff Key
	Ø	Length		
80	19	40	J6	6
90	24	50		8
100	28	60		8
112	28	60		8
132	38	80	K6	10
160	42	110		12
180	48	110		14

ATEX IMPELLERS

Impellers used for hazardous/explosive applications (ATEX) are covered by EN 1127-1 in the Eurozone and by the Air Movement and Conditioning Association (AMCA) Standard 99-0401 Classification of Spark Resistant Construction, in America.

Such equipment must not generate electrostatic charges that could discharge and create a spark, which would likely be hazardous.

AMCA Standard 99-0401 has three main classifications for "spark resistant" fans and blowers …

Type A

Type A fan construction requirements specify that all materials that are in contact with the air stream be constructed of spark resistant nonferrous material such as aluminium or brass. These fans offer the highest degree of spark resistance. In addition to requiring that the fan components be comprised of non-ferrous material, they must also be assembled in a manner such as to reduce the possibility of contact between any stationary and rotating component.

Type B

Type B fans require a non-ferrous wheel and a rubbing ring around the hole where the fan or motor shaft enters the fan housing. These fans provide a medium level of spark resistance, requiring that the fan components be assembled in a way that reduces possible contact between any stationary and rotating component.

Type C

Type C fans offer entry level spark resistant construction and require that the fan components be designed to reduce the possibility of contact between any stationary and rotating component. Construction requires a nonferrous plate on both sides of the inside of the fan or blower housing.

The Eurozone ATEX requirements are somewhat similar to the AMCA guidelines and in the case of ATEX compliant impellers provided by WingFan, they offer a PACAS blade material comprising Carbon Fibre Reinforced Polyamid (Nylon), which is both electrically conductive and flame retardant and suitable where explosions are a risk.

Complete ATEX compliant fans from 3rd party suppliers are supplied with full ATEX compliant documentation.

Electrical supplies around the world vary quite considerably, both in terms of voltage and frequency.

Direct current (DC) is not really a concern as almost all applications in our industry sector use either single phase or 3 phase AC supplies.

High voltage DC supplies are often encountered in the generation of hydro-electric power or when the power needs to be transmitted over long distances, then DC is more efficient than high voltage AC supplies. Thereafter, transformation stations convert the incoming power, at say 600,000V DC, down to line voltage and convert from DC to AC.

FREQUENCY

Generally, the supply frequency is either 50 Hz or 60 Hz, however the voltage can be either single or three phase and can vary in magnitude.

VOLTAGE

Typically, single phase supplies range from 110 to 240V ~ 50 or 60 Hz depending upon the country. Similarly, 3 phase supplies range from 220 to 690V ~ 50 or 60 Hz.

Most of the equipment supplied in our industry sector is exclusively 3-phase but can be 50 or 60 Hz, depending upon where in the world the equipment is destined.

Some common electrical supplies are …

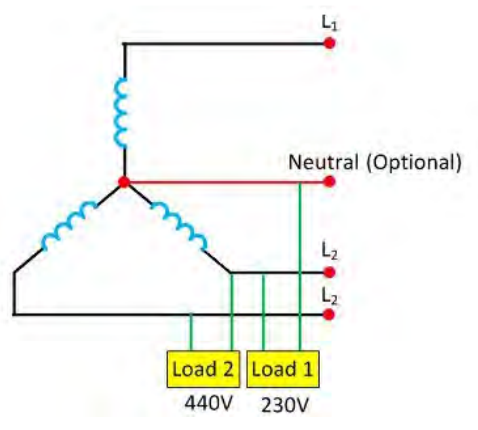
- 3 x 230V / 50 Hz … Norway
- 3 x 400V / 50 Hz … Europe
- 3 x 415V / 50 Hz … UK
- 3 x 480V / 60 Hz … USA
- 3 x 575V / 60 Hz … Canada … *some locations*

On occasions higher voltages are encountered, usually in the power generation/transformation sector such as 3 x 690V/50 Hz, 3 x 720V/50 Hz & 3 x 830V/60 Hz

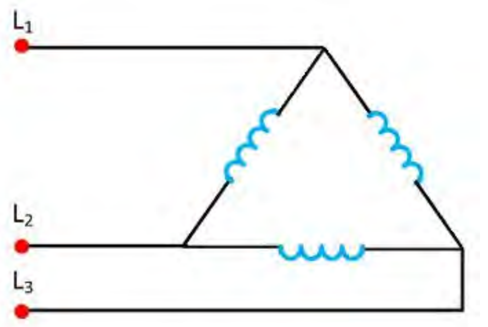
ELECTRICAL CONNECTION

Three-phase systems can be connected in two ways, either in …

Star



Delta



STAR CONNECTION

The star connection requires a minimum of three wires used for the three phase conductors and an optional forth core for the neutral conductor. The neutral point passes the unbalanced current to the earth resulting in a balanced supply.

In the case of a 400V electrical 3 phase supply, star connected three phase systems provide two different voltages, i.e., the 230 V and 400V. The voltage between any phase and neutral is 230V, whilst the voltage between any two phases is equal to the 400V.

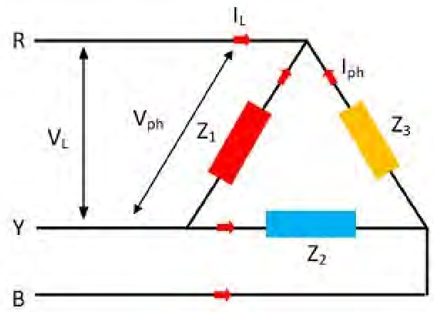
DELTA CONNECTION

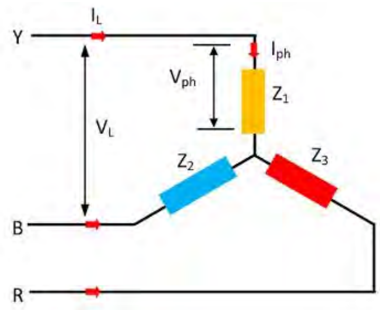
Delta connection uses only three wires and no neutral point. In this case the line voltage of the delta connection is equal to the phase voltage.

POWER FACTOR

The power factor (**pf**) is defined as the phase angle between voltage and current (**Cos ϕ**)

The three phase load may be balanced or unbalanced. If the three loads (impedances) Z_1, Z_2 and Z_3 have the same magnitude and phase angle then the three phase load is said to be a balanced load. Under a balanced condition, all the phases and the line voltages are equal in magnitude.





In electrical engineering, the power factor of an AC electrical power system is defined as the ratio of the **Real** power absorbed by the load to the **Apparent** power flowing in the circuit and is a dimensionless number in the closed interval of −1 to +1.

A power factor (pf) of less than one indicates the voltage and current are not in phase, reducing the instantaneous product of the two.

Real power is the instantaneous product of voltage and current and represents the capacity of the electricity for performing work.

Apparent power is the average product of current and voltage. The apparent power may be greater than the real power due to energy stored in the load and returned to the source or due to a non-linear load that distorts the wave shape of the current drawn from the source. A negative power factor occurs when the device (which is normally the load) generates power, which then flows back towards the source.

In an electric power system, a load with a low power factor draws more current than a load with a high power factor for the same amount of useful power transferred. The higher current increases the energy lost in the distribution system and requires larger wires and other equipment. Consequently the costs of larger equipment and wasted energy, electrical utilities providers will usually charge a higher cost to industrial or commercial customers where there is a low power factor.

Power-factor correction increases the power factor of a load, improving efficiency for the distribution system to which it is attached … *and thus the electricity cost*. Linear loads with a low power factor (such as induction motors) can be corrected with a passive network of capacitors or inductors. Non-linear loads, such as rectifiers, distort the current drawn from the system. In such cases, active or passive power factor correction may be used to counteract the distortion and raise the power factor. The devices for correction of the power factor may be at a central substation, spread out over a distribution system or built into power-consuming equipment.

***As an example** …*

An industrial plant runs at 100 kW (Working Power) and the Apparent Power meter records 125 kVA. To find the pf, divide 100 kW by 125 kVA to yield a pf = 80%. This means that only 80% of the incoming current does useful work and 20% is wasted through heating up the conductors. Because the electrical sub-station must supply both the kW and kVA needs of all customers, the higher the pf becomes, the more efficient the distribution system becomes.

Improving the pf can maximize current-carrying capacity, improve voltage to equipment, reduce power losses and lower electric bills.

Power factor (pf) or true pf (tpf) and Cos ϕ are the same in a linear network. But in a non-linear network when the harmonic components are summated, the term true power factor(tpf) has a different interpretation.

Simply true pf (tpf) = Displacement power factor (Cos ϕ) x Distortion power factor.

When a system is clean with no harmonic component the distortion pf value would be ideal. Thus making your true pf equal to Cos ϕ.

However, in non-linear network the harmonic distortion is summated with the displacement pf resulting in a reduction in the true pf.

MOTOR PROTECTION

Electrical installations involving electrical motors obviously necessitate the inclusion of line fuses or MCBs to protect the wiring, but when AC motors are involved, some form of motor protection is required to guard against thermal overload and avoid motor breakdown.

Protection can be achieved by fitting current overloads or two alternative thermal devices …

CURRENT OVERLOAD RELAYS

When the motor draws excess current, it is referred to as overloaded. This may cause overheating of the motor and damage the windings of the motor. Thus, it is important to protect the motor and motor branch circuit components from overload conditions.

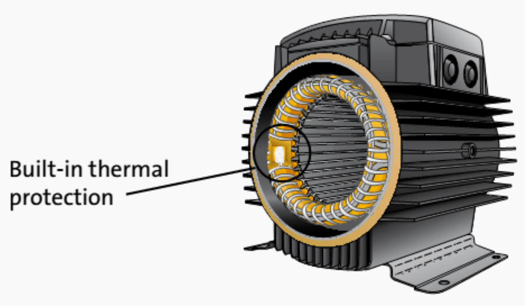
Overload relays are part of the motor starter (assembly of contactor plus overload relay). They protect the motor by monitoring the current flowing in the circuit. If the current rises above a preset limit over a certain period, then the overload relay will trip, operating an auxiliary contact which interrupts the motor control circuit and de-energising the contactor and removes the power to the motor.

Overload relays can be reset manually and/or will reset automatically after a preset period. Thereafter, the motor can be restarted.

PTO – KLIXON – THERMAL OVERLOAD

A PTO or thermal protector is in effect a thermostat utilising a snap action, bi-metallic, disc type switch to open or to close the circuit when it reaches a certain temperature. Thermal protectors are also referred to as Klixons, (*trade name from Texas Instruments*).

The sensors are hermetically sealed and pre-calibrated by the manufacturer and cannot be adjusted.



The PTO is embedded into the motor windings and usually connected to an external circuit breaker which will isolate the motor if it overheats.

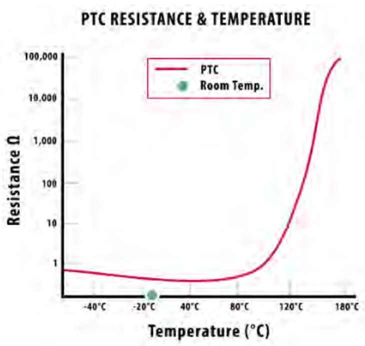
The degree of protection that an internal protection device provides is classified in the IEC 60034-11 standard.

PTC - THERMISTOR

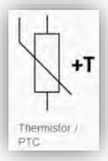
The second type of internal protection is the thermistor or **P**ositive **T**emperature **C**oefficient sensor (PTC).

Thermistors are built into the motor windings and protect the motor against locked-rotor conditions, continuous overload, or high ambient temperature. Often for 3-phase AC motors, 3 PTC sensors are connected in series, each monitoring one of the three phases.

Thermal protection is achieved by monitoring the temperature of the motor windings and if the temperature exceeds the rated trip temperature, the sensor undergoes a rapid change in resistance relative to the change in temperature.



As a result of this change, the control panel relay de-energises the control coil of the motor contactor and the motor goes off-line. As the motor cools and an acceptable motor winding temperature is restored, the sensor resistance decreases to the reset level.



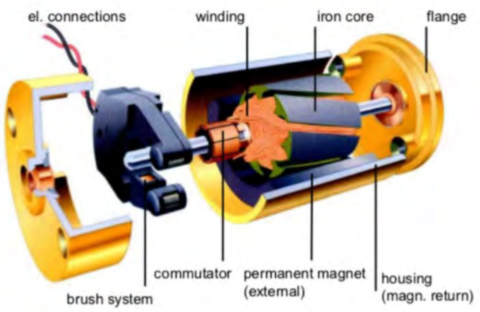
At this point, the module resets itself automatically, unless it was set-up for manual reset.

The non-linear characteristic of PTC thermistors is advantageous. In the critical range, a small temperature difference equates to a large resistance change which can be measured and monitored by a VFD or controller.

Traditionally there are two types of electric motors, depending upon the electrical supply available i.e. AC and DC, however a third alternative has appeared in recent years, namely the EC motor, which is powered by an AC supply but internally functions as a DC motor.

DC MOTORS

DC (direct current) electrical supplies can be originated from either a battery or DC generator.



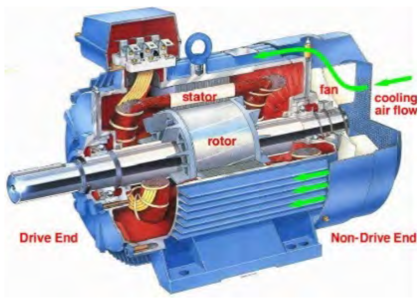
DC motors rely on carbon brushes and a commutation ring to switch the direction of the current and magnetic field polarity in a rotating armature. This interaction between the internal rotor and fixed permanent magnets induces the rotation of the motor.

Accordingly, DC motors are limited by the longevity of their brush system. The lifespan can range from as little as 100 hours, when under extreme loads, to 15,000 hours under favourable operating conditions. The high speed of rotation is only limited by commutation; thus DC motors can operate up to 10,000 rotations per minute.

DC motors have a high efficiency rate but suffer from specific losses associated with the initial resistance in the winding, brush friction and eddy-current losses.

AC INDUCTION MOTORS

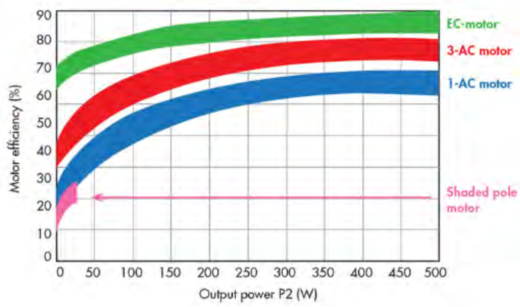
The most common electrical supplies available are AC (alternating current – sinusoidal wave form) where AC squirrel-cage induction motors or asynchronous motors are generally used.



An induction motor or asynchronous motor is an AC electric motor in which the electric current in the rotor needed to produce torque is obtained by electromagnetic induction from the magnetic field of the stator winding. An induction motor can therefore be made without electrical connections to the rotor.

AC induction motors use a series of coils powered and controlled by the AC input voltage. The stator magnetic field is created from the input voltage and the rotor magnetic field is induced by the stator field.

Some induction motors contain rotors made up of windings rather than a squirrel cage. The point of this wound-rotor configuration is to provide a means of reducing the rotor current as the motor first begins to spin. This is generally accomplished by connecting each rotor winding to a resistor in series. The windings receive current through a slip-ring arrangement. Once the rotor reaches final speed, the rotor poles get switched to a short circuit, thus becoming electrically the same as a squirrel cage rotor.



If the rotor exactly followed the moving stator pole, there would be no change in magnetic-field strength. Thus, the rotor field always lags behind the stator field by some amount, so it rotates at a speed that is somewhat slower than that of the stator. The difference between the two is called the slip.

The amount of slip can vary. It depends principally upon the load the motor drives, but also is affected by the resistance of the rotor circuit and the strength of the field that the stator flux induces.

AC motors are designed to operate at a specific point on a performance curve. This curve coincides with the peak efficiency of the motor. Outside of this point, the efficiency of the motor drops significantly because of the additional energy consumed to create the induced magnetic field in the rotor. Consequently, AC motors are typically 30% less efficient than DC motors, which use permanent magnets rather than copper windings.

AC motors were traditionally speed controlled with an optional variable voltage controller, which adjusts the incoming voltage, altering the sinusoidal waveform and reducing the input energy. Variable voltage controllers are prone to earth leakage issues, causing reduced bearing and motor lifetime issues. Furthermore, some motors exhibit excessive EMF generated resonance causing increases noise levels.

Alternatively, frequency inverters can be used, which although more efficient than variable voltage controllers, have their own motor related problems.

AC SYNCHRONOUS MOTORS

Another type of AC motor is a synchronous motor that can operate in sympathy with the supply frequency. The magnetic field is generated by a current delivered through slip rings or a permanent magnet. They can run faster than induction motors as they do not suffer from the ‘slip’ associated with asynchronous motors.

There are basically two types of synchronous motors:

- Self-excited - using principles similar to those of induction motors
- Directly excited - usually with permanent magnets, but not always

The self-excited synchronous motor, also called a switched-reluctance motor, contains a rotor cast of steel that includes notches or teeth, dubbed salient poles. It is the notches that let the rotor lock in and run at the same speed as the rotating magnetic field.

To move the rotor from one position to the next, circuitry must sequentially switch power to consecutive stator windings/phases in a manner analogous to that of a stepping motor. The directly excited synchronous motor may be called by various names such as ECPM (electronically commutated permanent magnet), BLDC (brushless dc) or just a brushless permanent-magnet motor. This design uses a rotor that contains permanent magnets. The magnets may be mounted on the rotor surface or be inserted within the rotor assembly (in which case the motor is called an interior permanent-magnet motor).

The permanent magnets are the salient poles of this design and prevent slip. A microprocessor controls sequential switching of power on the stator windings at the proper time using solid-state switches, minimizing torque ripple. The principle of operation of all these synchronous-motor types is basically the same. Power is applied to coils wound on stator teeth that cause a substantial amount of magnetic flux to cross the air gap between the rotor and stator. The flux flows perpendicular to the air gap. If a salient pole of the rotor is aligned perfectly with the stator tooth, there is no torque produced. If the rotor tooth is at some angle to the stator tooth, at least some of the flux crosses the gap at an angle that is not perpendicular to the tooth surfaces. The result is a torque on the rotor. Thus, switching power to stator windings at the right time causes a flux pattern that results in either clockwise or anticlockwise rotation.

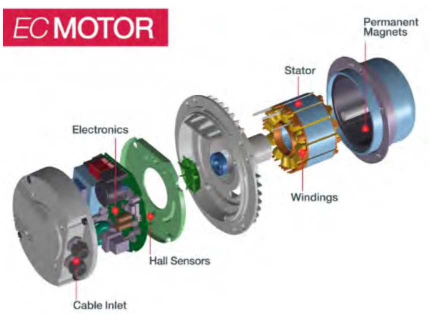
One other type of synchronous motor is called a switched reluctance (**SR**) motor. Its rotor consists of stacked steel laminations with a series of teeth. The teeth are magnetically permeable, and the areas surrounding them are weakly permeable by virtue of slots cut into them. Thus, the rotor needs no windings, rare-earth materials, or magnets.

Unlike induction motors, there are no rotor bars and consequently no torque-producing current flow in the rotor. The absence of any form of conductor on the SR rotor means that overall rotor losses are considerably lower than in other motors incorporating rotors carrying conductors. Torque produced by the SR motor is controlled by adjusting the magnitude of current in the stator electromagnets. Speed is then controlled by modulating the torque (via winding current). The technique is analogous to the same way speed is controlled via armature current in a traditional brush-dc motor.

An SR motor produces torque proportional to the amount of current put into its windings. Torque production is unaffected by motor speed. This is unlike AC-induction motors where, at high rotational speeds in the field-weakening region, rotor current increasingly lags behind the rotating field as motor rpm rises.

EC MOTORS

EC motors are brushless DC motors controlled by integrated electronics. The rotor contains permanent magnets, and the stator has a set of fixed windings. The traditional mechanical commutation is performed by the electronic circuitry, which supplies the correct armature current and switches the phases in the fixed windings to keep the motor turning. Since the motor’s speed is controlled by external electronics, EC motors do not have a limited synchronous speed.



EC motors have several benefits … they have no brushes; they do not spark or have the associated shorter life due to brushes fitted to conventional DC motors. Other advantages … they do not waste power because the electronics control the stator; they provide better performance and controllability and they run cooler than induction motors. In terms of size, the physically smaller motors can achieve the same output as traditional DC or AC motors. Furthermore, when designed as an external rotor motor rather than a shaft motor, significant space savings are possible.

Furthermore, the power distribution is much cleaner as a result of the integral intelligent electronics that ensure that the motor is less susceptible to voltage or frequency fluctuations. Typically, a 3 phase EC motor is suitable for an electrical supply in the range : 3 x 380 - 480 V (-10% / +15%) – 50/60 Hz . Within this range, the EC motor will operate at the same full speed and deliver the same power.

EC motors have variable speed control capability as a standard option. The commutation circuitry accepts inputs with pulse-width modulation of 4 - 20 mA and 0 - 10 V plus a digital Modbus input. These options control the speed in a range of 10% to 100%.

Monitoring of EC motors is simple with the integrated circuit and can be easily accessed by the designer to provide feedback. Lastly, EC motors provide a soft start, reduced noise and lower motor temperature.

SOFT START

Soft start is the terminology used for a variety of motor start-up systems to reduce the starting load, torque and current of a motor when activated.

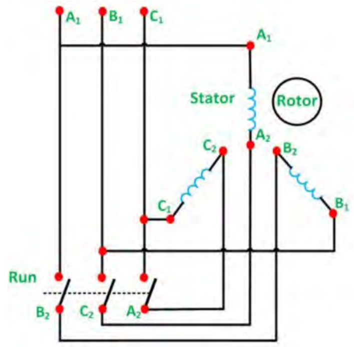
Star/Delta & Dahlander are two such systems, whilst other are …

- Variable voltage
- Frequency inverter
- Fluid coupling
- Clutches
- etc.

STAR/DELTA START

Motors > 5.5 kW generally draw an excessive starting current (surge) if started DOL and thus a mechanism of switching the electrical supply to the motor windings, via contactors, between Star and Delta, ensures a lower starting torque and thus a lower starting current.

Once the motor is running at its 'lower than full speed'; whilst the windings are connected in Star and thus subjected to a lower voltage potential; the Star/Delta changeover contactors are activated to connect the motor windings in Delta, which subjects them to the full supply voltage and enables the motor to run at its full load speed.



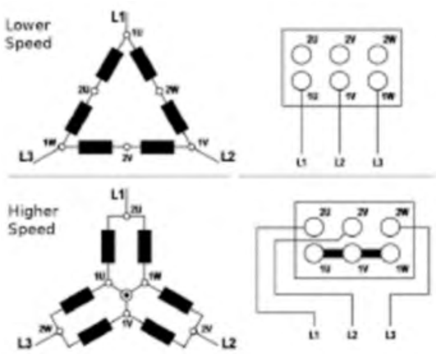
DAHLANDER

A Dahlander motor (*also referred to as a pole changing motor, dual or two speed motor*) is a type of multispeed induction motor where the speed is varied by altering the number of poles, which is achieved by changing the winding wiring connections inside the motor.

The motor may be designed for fixed or variable torque depending upon the stator winding.

This motor concept is named after a Swedish engineer *Robert Dahlander* (1870–1935).

These days, this speed control methodology is less common, being replace by frequency inverters or EC motors.



CONSUMED POWER – P1

The ‘consumed power’ denoted by **P1** is the electrical power in kW supplied to the motor, which is ultimately translated into usable power at the motor shaft.

P1 is what the ‘consumer pays’ for and is often used in life cycle costing analysis (LCC).

The power consumption P1 can be calculated by the formulas shown below depending upon whether the motor is a single-phase or a three-phase.

AC single-phase motor, e.g. 1 x 230 V

$$P1 = U \times I \times \cos \phi$$

AC three-phase motor, e.g. 3 x 400 V

$$P1 = \sqrt{3} \times U \times I \times \cos \phi$$

SHAFT POWER – P2

This is defined as the power available at the shaft of the motor and denoted **P2** and is provided in kW for IEC motors and HP (Horsepower) for NEMA motors.

The inefficiencies of the motor, both electromagnetic and mechanical, dictate what power is available at the shaft and is clearly less than the electrical power supplied to the motor.

The efficiency of smaller kW motors is generally somewhat less than for larger kW motors.

FLC – FULL LOAD CURRENT

The running current, otherwise known as the full load current, is denoted by **FLC** and measured in Amps and is the current drawn by the motor when operating at its rated shaft power.

LRC – LOCKED ROTOR CURRENT

The locked rotor current is denoted by **LRC** and is sometimes referred to as the ‘starting current’ and measured in Amps. On occasions the LCR is indicated as a percentage of the FLC e.g. 250% indicating that the LCR Amps is 2.5 times greater than the running Amps.

The starting current of an AC motor is often significantly greater than the FLC and results from the increase in power necessary to overcome the inertia of the rotor when starting the motor from a standstill position.

This starting current can be dramatically affected if the motor drives an impeller and in particular, the air temperature is low. As an example, the air density @ -40°C is 25% higher than @ +20°C and thus the absorbed power of the impeller and hence motor is increased by 25% too. Hence the sizing of the motor current overloads is a serious consideration.

If the LRC is not known, but only the FLC, then a guide is **LRC = FLC x 6**

DOL – DIRECT ON-LINE

Generally, AC motors <= 5.5 kW are started DOL, where the electrical supply is directly applied to the motor, typically via a motor starter/contactor.

AIR OVER MOTOR(AOM) TEMPERATURE

Motors are designed to withstand certain internal winding and bearing temperatures which are then related to an acceptable ‘air over motor’ (AOM) temperature. If this temperature is not exceeded, the longevity of the motor should meet the design life of the motor.

However, if the motor is subjected to prolonged periods of elevated temperatures, then either the viscous grease in the bearings will liquify and run out of the bearing and may result in bearing seizure or the insulation coating/lacquer of the windings may be damaged and again result in motor failure.

AC induction motors typically have AOMs up to 70°C and with Class H insulation, 80°C. However, compact fan external rotor motors and EC motors are more typically between 50 and 65°C. Thus it is important to consider this design parameter when considering higher temperature industrial applications.

BEARING BRINELLING

If a motor's shaft is subject to high impact loads or excessive loads, the balls or rollers in the bearings may come into intimate contact with the raceway and galling or spalling of the surfaces may result, which eventually leads to bearing failure.

By definition, Brinelling is the permanent indentation of a hard surface. It is named after the Brinell scale of hardness, in which a small ball is pushed against a hard surface at a pre-set level of force, and the depth and diameter of the mark indicates the Brinell hardness of the surface.

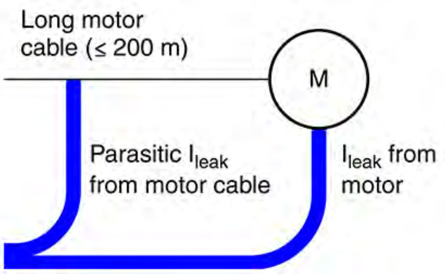
EARTH LEAKAGE

All electric motors operate with inherent losses from a number of different areas. Some are inherently mechanical (*consider friction losses as heat*) while others are electrical, from current in conductors, also known as I^2R losses, to losses due to hysteresis and eddy currents in the iron components of a motor.

One type of loss that is especially pernicious is leakage current from the motor to ground. A small amount of current can flow from the motor to ground in all electrical equipment, usually as a result of some type of fault in the equipment or cabling. The amount for the most part is small, perhaps in the order of milliamps, but it is present nonetheless and poses a safety risk to human operators.

Several sources of leakage current are shown, including from the motor itself as well as from the motor cable.

Leakage currents can flow through the insulation surrounding conductors. Electrically speaking, insulators have both resistance and capacitance, and leakage current can flow through both. Old or damaged insulation means a lower resistance and hence more potential current flow. Also, the longer the conductor the more capacitance, and therefore more leakage current flow. Typical allowable leakage currents are in the range of a 1-2 mA or sub-milliamps.



Several sources of leakage current are shown here including from the motor itself as well as from the motor cable. (Image via TDK)

In worst case scenarios, earth leakage can affect the bearings of the motor. In fact, microscopic sparks between the balls and its raceway can cause pitting and eventually the bearing may fail.

CONDENSATION

A well designed motor will not allow the ingress of water, either through its drain hole (*if necessary and fitted*) or bearing seals, however, motors are often deemed to have failed because of water entering the motor.

An AC induction motor will run with water inside the housing between the stator and rotor, and providing no water enters the terminal box, will continue to operate.

The problem arises when the end shield mounted bearings are adversely affected by the water, corrode and then fail.

Evidence of water in a motor, for example as a visible 'tide mark' of corrosion inside the housing, will generally accumulate from condensation.

Unless a motor is hermetically sealed, migration of moisture laden air from the surroundings to the inside cavities of the motor will always take place. If the temperature of the metal parts in contact with this moist air, drops below the dew point of the air, then condensation of the water vapour will take place. If this condensate cannot drain away, it will accumulate.

Typically, all outdoor located products will subject their motors to fluctuations in temperature and internal condensation will follow. Thus such motors are usually fitted with drain holes positioned at the lowest point in the motor with respect to the motor's orientation, which may be shaft up, shaft down or shaft horizontal, to allow for drainage.

Motors in our market sector often fail from bearing issues rather than electrical related issues.

If the motor drain hole is positioned incorrectly, condensation (water) build-up inside the motor eventually immerses a portion of the 'sealed for life' ball or roller bearings. Bearings are not designed to run under water and over time the grease is degraded by the water and the lubrication ability is diminished, causing the bearing to become noisy and ultimately fail.

ANTI-CONDENSATION HEATERS (ACH)

Anti-condensation heaters are usually small single phase (1 x 230V) heaters embedded into the windings of a 3 phase AC motor.

If the required motor IP Class does not allow for any drain holes e.g. ATEX motors, then anti-condensation heaters (ACH) may be required to ensure that the motor is kept at a temperature above the dew point of the air.

ACHs may also be required if equipment is stored for extended periods prior to installation and/or operation or the equipment is subject to extended period of idle time, in an environment subject to high relative humidities where temperature may drop below the dew point. In such cases, these ACHs must be supplied with power during the full term of storage or off-cycle periods.

ANTI-WINDMILLING BRAKE

When the fans fitted to a product are not running, prevailing winds may blow in opposition to the air direction leaving the product.

In certain circumstances and orientation of the product, may result in the fans running backwards, often at a greater rpm than the operating speed. Although this, in itself, is not a problem, when it is time to run the fan(s) for normal operation, the AC motor contactor current overload may well trip, isolating the motor and thus thermal capacity of the product cannot be achieved.

This is one of the reasons why product orientation (horizontal or vertical) with respect to the prevailing wind direction can be important, especially when the motor speed is low e.g. 8, 12 or 16 pole motors.

Most 'small' AC motors are started DOL (direct online), so if the motor is running in the wrong direction, when the power is supplied there will be an excessively high starting current, which may prevent the motor from starting at all.

One way to eliminate this problem is to fit the motor with an anti-windmilling brake, that prevents the motor from running backwards … problem solved !

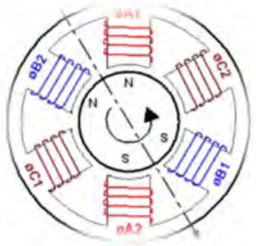
EC motors are not so susceptible to windmilling issues because their onboard electronic intelligence allows for the sensing of reverse rotation and the gradual application of power to slow down and stop (*electrically brake*) the reverse direction and 'soft start' the run up to full speed in the correct direction.

IEC and NEMA motors reference two standards for asynchronous squirrel cage induction motors, European and North American respectively.

An AC motor designed for a 50 Hz supply will run 20% faster if the frequency is increased to 60 Hz. As a consequence, the output power (P_2) will increase along with the required running current (Amps).

However, when an impeller is fitted to a motor shaft the above speed implications have dramatic consequences upon the absorbed power of the impeller and may result in the motor being incapable of meeting the significantly increased power requirements. *See section Fan technology*

MOTOR POLES



The term 'Poles' relates to the number of North-South electromagnetic pairs that are generated in the motor windings. A 3-phase motor has 3 sets of windings per 'pole'. Thus a 2 pole motor has 6 windings and a 4 pole, 12 windings etc.

The speed (rpm) of motors is often referred to in terms of their number of poles. The number of poles indicates the nominal rotational speed which a motor can achieve and can be calculated in the following section.

SYNCHRONOUS SPEED

The synchronous speed of an AC motor, denoted by $\mathbf{n_s}$ is the rotation rate of the stator's magnetic field and is the theoretical speed of a motor if there were no electromagnetic losses and is defined by …

$$\mathbf{n_s = 2 \times \mathit{f} / P}$$

where, n_s *= synchronous speed, revs/sec*
f *= frequency of the power supply, Hz*
P *= is the number of magnetic poles*

Transposed into the more usual form where $\mathbf{n_s}$ is defined as rpm, the above equation becomes …

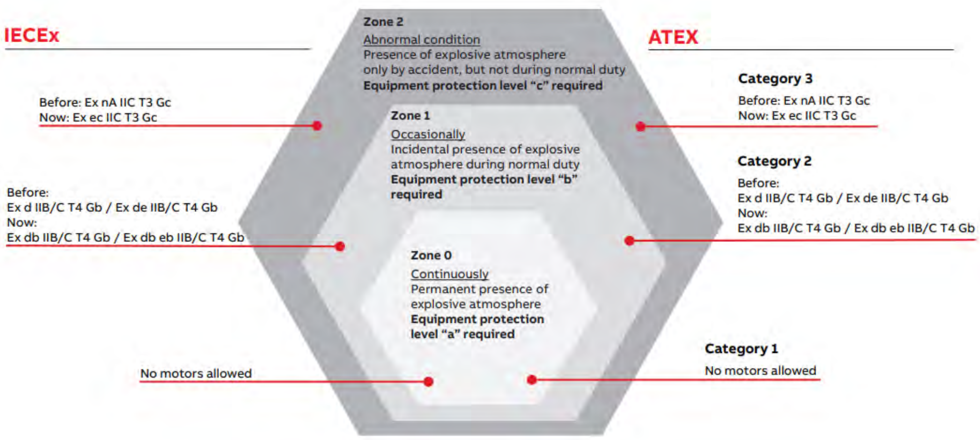
$$\mathbf{n_s = 120 \times \mathit{f} / P}$$

Thus, a 4 pole motor's synchronous speed operating with a supply frequency of 50 Hz is 1500 rpm and if the frequency is 60 Hz, will be1800 rpm.

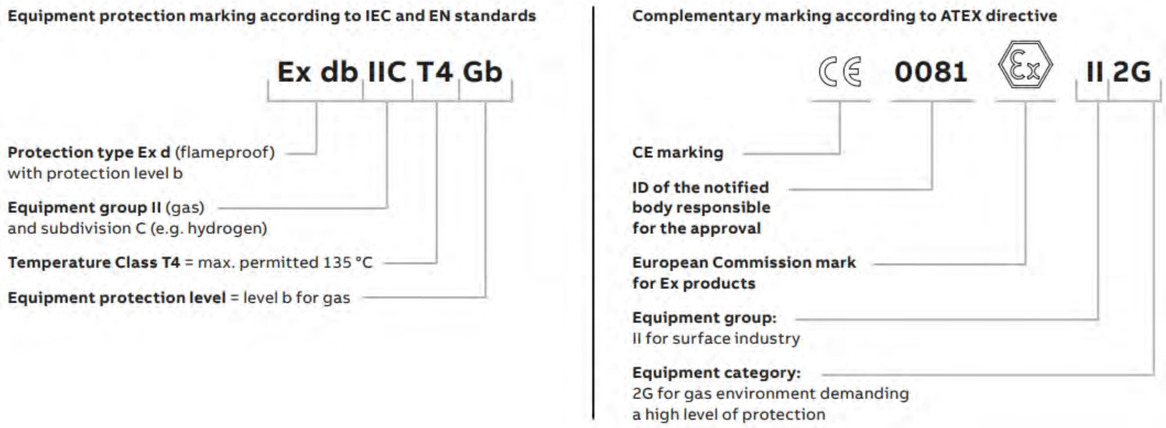
However in reality, electromagnetic inefficiencies result in 'slip' and thus small power (fractional) motors often run at around -5/10% lower speed, whilst larger power motors run at perhaps -2/5% lower than synchronous speeds.

Globally there are a variety of national regulative bodies and certification systems with different requirements that govern equipment used in dangerous or explosive atmospheres. Two such systems that EAS encounter are …

- IECEx : **I**nternational **E**lectrotechnical **C**ommission for **Ex**plosive Atmospheres
- ATEX : **At**mospheres **Ex**plosibles



Example of a flameproof motor according to IECEx and ATEX …



ATEX DIRECTIVE

As from July 2003 the EU introduced legislation to protect employees from explosion risk in areas with explosive atmospheres.

Explosive atmospheres are work areas that contain flammable gases, mists or vapours or combustible dusts. All it needs is an ignition source to cause an explosion.

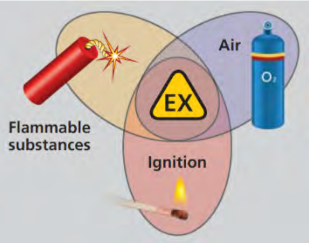
The latest **ATEX directive 2014/34/EU** (*also referred to as ATEX 114*) was mandatory for manufacturers as from 20th April 2016.

ATEX is primarily for use within the European Union and becomes law when adopted by any member state. However, it is not universally recognised internationally.

Potentially explosive environments are classified into 3 zones in accordance with the ATEX Directive and are …

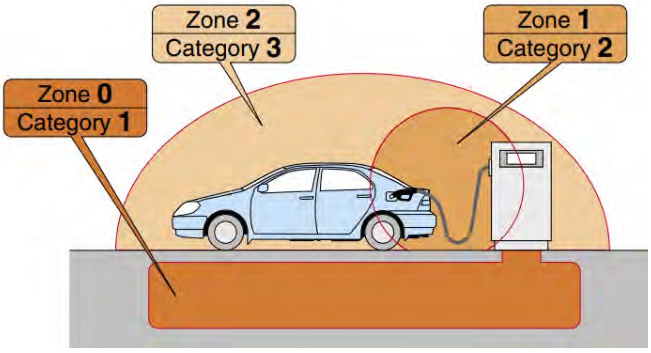
- 0, 1, 2 for gas explosive atmospheres
- 20, 21, 22 for dust explosive atmospheres

For explosions to occur the following three elements need to occur simultaneously …



- Flammable and combustible substances e.g. fuel or combustible dust
- Comburents e.g. Oxygen, methane etc.
- Ignition source e.g. open flames, lightning strikes, mechanically generated impact/friction sparks, electrostatic discharge, radiation, adiabatic compression

In simple visual terms …



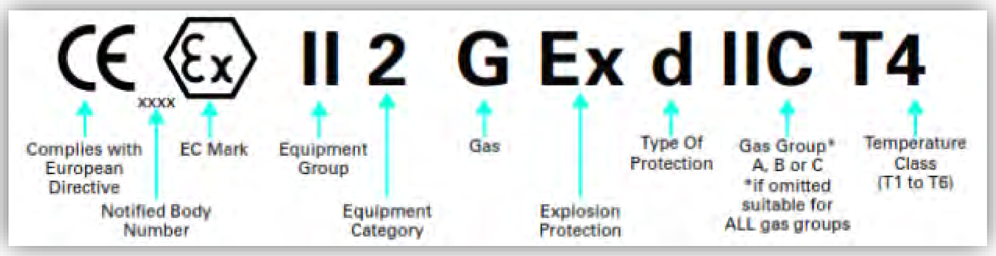
The ATEX directive defines categories of equipment and protective systems, which can be used in the corresponding zones as per the following table …

Zone		Equipment category	Presence of the explosive atmosphere
Gas	Dust		
0	20	1	Continuously or for long periods >1000 hours/year
1	21	2	Occasionally 10~1000 hours/year
2	22	3	Rarely or for short periods <10 hours/year

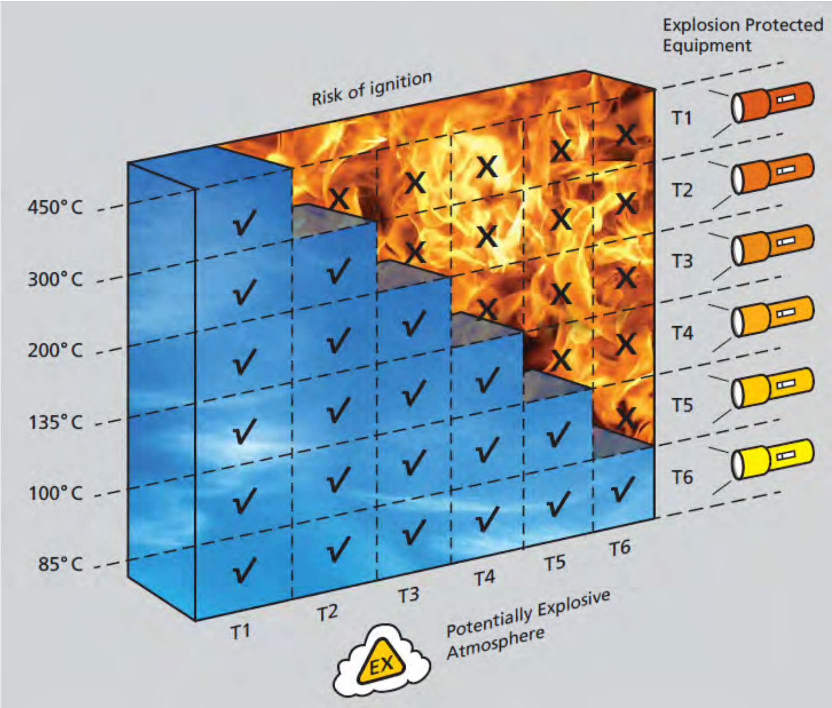
The most common Zone, hence Category, that our type of equipment is exposed to is Zone 2 and the ATEX Directive defines Zone 2 as an area in which an explosive atmosphere (a mixture of air and flammable gases, steam or mist) is not to be expected under normal circumstances or just for a short period of time. Just one brief hazardous situation per year is enough to rate an area Zone 2 and thus such devices provide the required ruggedness and drop protection, which ensures safety of use during normal operations without technical failures such as a short circuit.

However, an area must be classified Zone 1 if an explosive atmosphere can develop occasionally during normal operations. In that case only Zone 1 certified products are permissible to be used, which ensures necessary safety in case of failure.

The marking for Gas Hazards conforms to the following format …



Hot surfaces can ignite an explosive atmosphere and thus to avoid high surface temperatures on enclosures, ATEX defines a range of Temperature Classes ..



Clearly for motors, keeping the surface temperature below the ignition temperature is important and a motor conforming to class T6 (<= 85°C) would invariably be a rather expensive motor.

Typically, in our industry section, most applications fall under T3 (<= 200°C) and result in more affordable solutions.

<table>
<tr><th></th><th colspan="5">An example of the markings : CE 1354 Ex II 2 G Ex db IIC T4 Gb</th></tr>
<tr><td>CE 1354</td><td colspan="5">The designation number of Notified Body (NB) is added if it is involved in the conformity assessment Proces, NB number TI a.s. 1354</td></tr>
<tr><td>Ex</td><td colspan="5">The specific marking of explosion protection according to ATEX Directive 2014/34/EU</td></tr>
<tr><td rowspan="3">II</td><td colspan="5">Marking of the equipment according to Directive ATEX 2014/34/EU</td></tr>
<tr><td colspan="5">(I) -intended for use in underground parts of mines as well as those parts of surface installations of such mines, which are endangered by firedamp and/or combustible dust</td></tr>
<tr><td colspan="5">(II) -intended for use in areas in which explosive atmospheres caused by mixtures of air and gases, vapours or mists or air/dust mixtures</td></tr>
<tr><td rowspan="7">2</td><td colspan="5">Designation of categories of equipment for the Group</td></tr>
<tr><td>Equipment Group</td><td>Equipment Category</td><td>Zone of Use</td><td colspan="2">Environment</td></tr>
<tr><td>I</td><td>M1</td><td rowspan="2">N/A</td><td colspan="2" rowspan="2">Methane & Coal Dust</td></tr>
<tr><td>I</td><td>M2</td></tr>
<tr><td>II</td><td>1</td><td>0/20</td><td colspan="2" rowspan="3">Gas, Vapour, Mists & Dusts</td></tr>
<tr><td>II</td><td>2</td><td>1/21</td></tr>
<tr><td>II</td><td>3</td><td>2/22</td></tr>
<tr><td rowspan="3">G</td><td colspan="5">Only for device group II indicates the letter G (for gas) or D (dust)</td></tr>
<tr><td colspan="5">G – for equipment designed for explosive atmospheres caused by gases, vapours or mists</td></tr>
<tr><td colspan="5">(D) – for equipment designed for explosive atmospheres caused by dust</td></tr>
<tr><td>Ex</td><td colspan="5">Ex symbol means that the device corresponds to one or more of the type of protection, which are subject to specific standards of the series EN 60079 and/or EN 80079-36</td></tr>
<tr><td rowspan="7">db</td><td rowspan="7">Symbol used according to the kind of protection of the specific standards or standards under which the device is made and approved, in this case, a "hard Cap" with a level of protection Gb according to EN- 60079-1</td><td>Type of Protection</td><td>Symbol</td><td>IEC/EN Standard</td><td>Basic Concept of Protection</td></tr>
<tr><td>Flameproof</td><td>d</td><td>60079-1</td><td>Contains Explosion, Prevents Explosion</td></tr>
<tr><td>Increased Safety</td><td>e</td><td>60079-7</td><td>No arcs, sparks or hot surfaces</td></tr>
<tr><td>Intrinsic Safety</td><td>i</td><td>60079-11</td><td>Limits the energy of the spark and surface temperature</td></tr>
<tr><td>Encapsulation</td><td>m</td><td>60079-18</td><td>Keeps Combustible dust out and avoids hot surface</td></tr>
<tr><td>Protection by Enclosure</td><td>t</td><td>60079-31</td><td>Keeps Combustible dust out and avoids hot surface</td></tr>
<tr><td></td><td></td><td></td><td></td></tr>
<tr><td rowspan="8">IIC</td><td colspan="5">The symbol of the sub-groups of gases or dusts to which your device is suitable</td></tr>
<tr><td colspan="5">(I) for gas in the mines – Coal mining and/or combustible dusts</td></tr>
<tr><td colspan="5">IIA – intended to subsets of the explosive gas atmosphere, which is a typical gas propane</td></tr>
<tr><td colspan="5">IIB – intended to subsets of the explosive gas atmosphere, which is a typical gas ethylene is appropriate and where required by the IIA</td></tr>
<tr><td colspan="5">IIC – intended to subsets of the explosive gas atmosphere, where a typical gas is hydrogen, it is appropriate and where required by the IIA or IIB</td></tr>
<tr><td colspan="5">IIIA- designed into sub-groups of explosive dust atmosphere consist of combustible dusts</td></tr>
<tr><td colspan="5">IIIB- designed into sub-groups of explosive dusts atmosphere consist of Non – Conductive dust</td></tr>
<tr><td colspan="5">IIIC – intended to subsets of the explosive dust atmosphere consisting conductive dust, atmosphere is appropriate where IIIA or IIIB is required.</td></tr>
<tr><td rowspan="9">T4</td><td colspan="5">The symbol for the temperature class for explosive gaseous atmospheres</td></tr>
<tr><td>Temperature class</td><td>The maximum surface temperature in ° C</td><td colspan="3" rowspan="7">- If the temperature is between two classes, for example, may be marked "T1 or 350 ° C", or "350 ° C (T1)".
- If it is higher than 450 ° C must be marked only the higher, such as 600 ° C
- Equipment ambient Temperature: -20°C to +40°C unless otherwise stated on the equipment</td></tr>
<tr><td>T1</td><td>450</td></tr>
<tr><td>T2</td><td>300</td></tr>
<tr><td>T3</td><td>200</td></tr>
<tr><td>T4</td><td>135</td></tr>
<tr><td>T5</td><td>100</td></tr>
<tr><td>T6</td><td>85</td></tr>
<tr><td colspan="5">For explosive dust atmospheres shall indicate the maximum surface of temperature of, for example, "T 85 ° C"</td></tr>
<tr><td rowspan="9">Gb</td><td colspan="5">The level of protection of the equipment</td></tr>
<tr><td>Ma</td><td colspan="4">facilities to be installed in the mine with "very high" levels of protection, guarantees sufficient protection that it is unlikely to become a source of ignition under normal operation, while the anticipated functional failures or malfunctions even if If it remains under exceptional tension in gas explosion</td></tr>
<tr><td>Mb</td><td colspan="4">facilities to be installed in the mine with the "high" levels of protection, guarantees sufficient that it is unlikely to become a source of ignition under normal operation or during foreseeable malfunctions.</td></tr>
<tr><td>Ga</td><td colspan="4">device to be installed in the explosive gaseous atmospheres with "very high" levels of protection, is not a source of ignition under normal operation, while the implied fault, or during exceptional functional disorders.</td></tr>
<tr><td>Gb</td><td colspan="4">device to be installed in the explosive gaseous atmospheres with a "high" level of protection, is not a source of ignition in the normal conditions of use or during foreseeable malfunctions.</td></tr>
<tr><td>Gc</td><td colspan="4">device to be installed in the explosive gaseous atmospheres with the "enhanced" level of protection, is not a source of ignition under normal operation.</td></tr>
<tr><td>Da</td><td colspan="4">device to be installed in the explosive dust atmospheres with "very high" levels of protection, is not a source of ignition under normal operation, while the anticipated functional failures or malfunctions during the exceptional condition.</td></tr>
<tr><td>Db</td><td colspan="4">device to be installed in the explosive dust atmospheres with a "high" level of protection, is not a source of ignition under normal operation or during foreseeable malfunctions</td></tr>
<tr><td>Dc</td><td colspan="4">device to be installed in the explosive dust atmospheres with the "enhanced" level of protection, is not a source of ignition under normal operation</td></tr>
</table>

This specification calls for special motors that meet the ATEX demands and are generally somewhat more expensive than standard IEC induction motors and on occasions, subject to long lead times.

Some compact fan manufacturers offer limited ATEX compliant fan/motors, both in AC and EC variants, but it is important to check with the supplier regarding the price and delivery.

Furthermore, if an IEC motor is used and EAS selects the impeller, then this component must also meet the ATEX requirements.

ATEX compliant impellers usually use anti-static blade materials such as carbon fibre impregnated polyamide or polypropylene. Furthermore, on occasions, the fan shroud/cowl may be required not to generate static electricity, which in a worst case scenario, may create a spark and thus an explosion.

Flameproof

Protection type Ex d

Protection type Ex d requirement options for use with an AC drive

- The motor has been tested together with the drive for the duty intended and with the protective device provided.
- Or, use direct temperature protection with embedded temperature sensors and with a sufficient margin to protect the bearings or the rotor. The actions of the protective devices used must cause the motor to be disconnected.

1 Joints with long spigots preventing flames escaping to the outside

2 Flame paths between shaft and inner bearing covers

3 Motor housing developed to withstand an internal explosion

Only external surface temperature needs to be considered for the Ex temperature class.

Increased safety “ec”

Protection type Ex ec

Protection type Ex ec requirement options for use with an AC drive

- The motor is tested with the drive or a comparable drive.
- Or, the motor’s temperature class is determined by calculation.

1 No hot surfaces in rated conditions

2 No sparking during normal running or starting

Surface temperature of any part (inside or outside) must not exceed the Ex temperature class limit.

CONTROL METHODOLOGY

BACKGROUND

When EAS supplies a dry cooler or air cooled condenser, it can be supplied ‘naked’ without any controls or switch gear or alternatively, supplied with a variety of wiring and electrical options plus controllers to fulfil the Client’s requirements.

FLYING LEADS

On occasion, EAS’s product scope of supply is only the dry cooler/condenser with wired motors but the Client requires the wiring to be long enough (flying leads) to connect to their; possibly remotely located; common terminal box or electrical control panel.

SCREENED CABLING – EMC COMPLIANT

Where ‘electrical noise’ issues might be a problem, especially when frequency inverters are implemented, which are not fitted with a sinus filter, EMC compliancy can be achieved by using screen cables.

Note : All electrical gear associated with the motors and inverters should also be EMC compliant.

SAFETY SWITCHES

If fan/motor isolating safety switches are required … *to allow the ‘safe working’ upon a motor by isolating the electrical supply to that motor and allowing the locking of the switch to avoid inadvertent reconnection* … either standard or EMC compliant safety switches can be fitted.

Special customer specified switch gear can be fitted which may comply with ATEX or NORSOK requirements.

STARTER PANEL

The most basic type of motor control panel.

It houses main isolator/switch, fuses (MCBs) and the starter (motor contactor) and associated controls for an AC motor.

The starter panel may also include a circuit breaker fitted with a current overload to protect the motor.

Alternatively, relays may be included to handle PTO or PTCs fitted to motors for thermal protection purposes

STEP CONTROL – FAN CYCLING

When there is a need to control the fluid leaving temperature from a dry cooler or the condensing pressure (head pressure) for an air cooled condenser, adjustment of the air volume will ensure that the set point conditions can be met.

Turning fans on and off is the simplest mechanism to achieve this control and in the ‘old days’, multiple Danfoss KP77 thermostats were set-up at temperature intervals to provide ‘course‘ control of the desired leaving temperature.

However today, a simple electronic sequencer can be used where the set point condition can be set. Typically, such a controller provides a number (often 4) stages/relays, which in turn can activate individual or groups of fans and can even activate the chosen wet system e.g. EASiSpray or EASiPad if fitted.

When there is no wet system to activate and if each stage is switching groups of equal numbers of fans, the controller is factory set to provide a ‘rotation’ feature, meaning that the order in which the fan/group of fans is activated is randomised, thereby equalising the service life of the fans.

If however, the number of fans in each group is different, then the controller is programmed to operate sequentially.

Furthermore, when a wet system is part of the control functionality the sequential mode is programmed so that it is activated as the last stage to operate and the first stage to terminate.

One drawback with fan cycling capacity control is that the heat exchanger surface associated with the inactive fans experience more or less 'natural convection' and thus the heat flux in this vicinity is dramatically lower than a similar coil section beneath an active fan. Therefore the efficiency of this method the capacity control is far less that when using a variable fan speed control option such as variable voltage, frequency inverter or EC fan control.

Another potential drawback is the far lower sound attenuation achievable when individual or groups of fans are turned off compared with reducing the speed of all the fans together, whist still achieving the same reduced capacity.

VARIABLE SPEED CONTROL OPTIONS

The abbreviation VSD (Variable Speed Drive) or VFS (Variable Fan Speed) can be confusing and some industry sectors perceive it to infer fan speed control via variable voltage, usually associated with the speed control of AC induction motors.

However, others compare this term with VFD (Variable Frequency Drive) i.e. via the use of a frequency inverter to modulate the motor/fan speed.

On occasions the term FSC (Fan Speed Control) is used in place of VSD, but EAS interprets VSD or FSC abbreviations to be a generic term for either a variable voltage (VVD) or variable frequency (VFD) control of the fan speed.

VSD can also be (mis)used when referring to EC or PM motors, which either include an integral EC controller when applied to compact fans utilising external rotor motors or externally located EC controllers, when PM motors are employed. EC motors/fans are inherently a variable speed machine.

VARIABLE VOLTAGE - VVD

Variable voltage drive/controllers were popular before frequency inverters and EC fans became economically viable and common place. Although they are still available, they can on occasion exhibit resonance noise issues plus generate earth leakage currents that may result in bearing longevity related problems.

Nevertheless, supplying AC induction motors with a voltage less than the normal supply voltage will cause the motor to run less efficiently resulting in a lower speed, this in turn reduces the air volume and thus cooling effect, which allows the heat exchanger to provide the desired fluid leaving temperature or maintain the condensing pressure.

FREQUENCY INVERTER - VFD

Frequency inverters provide similar results to the above variable voltage controllers, but in this case modulates the AC supply frequency, which in turn affects the fan speed and thus air volume enabling capacity control to be achieved.

The user interface – programming sequence – and basic control features are very similar for both types of devices.

Inverters overcome the potential problems that voltage controllers exhibit and are electrically more efficient, whilst controlling down to 10-20% of full speed.

Both variable voltage and inverter controllers provide more efficient thermal cooling across the whole heat exchanger surface with the benefit of a lower sound power level generated by the product.

When controlling the frequency/fan speed down to 10% of full speed, the starter panel/control panel should accommodate the necessary relays etc. to handle either motor fitted PTO or PTCs to provide better motor protection than motor contactors with current overloads are able to provide.

VARIABLE SPEED CONTROL OF EC MOTORS

Electronically commutated (EC) motors/fans use permanent magnet motor technology and integral 'intelligent' electronics that can handle a variety of start-up scenarios, error and electrical/thermal overload states, where also, their speed (rpm) can simply be controlled with either a 0-10VDC signal or digitally via Modbus or indeed other control protocols.

By design, EC motors are very efficient compared with AC induction motors and even more so, at reduced speed.

As the EC motor includes the speed control electronics, there is no need for any large voltage or frequency device in the starter panel, just fuses to protect the wiring ... the EC motor electronics handle the error status.

Therefore to provide capacity control for a dry cooler or condenser, sensors measuring temperature or pressure are connected to a simple electronic controller/PLC, which outputs either a 0-10V DC signal or Modbus signal to the motors and adjusts the speed accordingly.

ADIABATIC – WET SYSTEMS

BACKGROUND

Dry coolers and air cooled condensers are often designed for a specific set of conditions and on occasions these can be exceeded. In such scenarios both products will under-perform and either more surface or more air volume is required to rectify the situation.

To enable products to function adequately when the design dry bulb ambient temperatures are exceeded, one method is to pre-cool the incoming airstream, widen the operating temperature difference, enabling the product to meet the design duty.

Pre-cooling of the incoming air stream can be achieved by spraying ‘city’ water directly into it, which has associated disadvantages such as ‘drift’ related issues plus the contamination of the heat exchanger surface and reducing the longevity of the product, plus the concern … in some countries … of the ‘aerosol’ related issues and potential legionella related concerns.

Alternatively, a wetted absorbent pad material can be placed in front of the heat exchanger, through which the air is induced. The resulting adiabatic cooling process ‘depresses’ the dry bulb temperature allowing a greater capacity to be achieved from the same heat exchanger.

EASISPRAY SYSTEM

The pseudo-adiabatic spray technology, known as **EASiSpray** can either be factory fitted or retrofitted to both legacy products and indeed competitor’s products.

When a shortfall in capacity is potentially an issue, a simple remedy is to in effect ‘turbo charge’ the product with the introduction of water into the incoming air stream. Often, this methodology is referred to as ‘**peak load**’ cooling and is limited to a maximum of 200 operation hours per year unless special provision is made to ensure adequate water quality to minimise fouling of the heat exchanger surface.

The EASiSpray system can be supplied as a water distribution system only or complete with water control valves and can be either independently controlled by the End User or interfaced with the capacity control system factory fitted to the product.

Providing the incoming air is not fully saturated, the presence of the introduced moisture will result in a depressed air inlet dry bulb temperature as a result of the adiabatic evaporative cooling effect. Obviously, the depression of the dry bulb temperature will widen the operating TD (temperature difference) of the dry cooler or air-cooled condenser thus allowing the product to match the increased load.

Generally, there are two methodologies for introducing moisture into the incoming air stream ..

- Spraying water directly onto the coil surface and in the direction of the air flow
- Spraying water away from the coil in opposition to the air flow direction

Both methods can produce a degree evaporative cooling however, there are some basic differences described as follows …

The first method; a spin-off of traditional spray coil technology; tends to flood the coil with water, which if not adequately dispersed can result in ‘water logging’ or agglomeration of the heat exchanger surface creating high velocity zones that can result in water carry-over and consequently, ‘drift’ issues.

However, there is a positive effect of the direct application of ‘city’ water; which often has a temperature averaging around +10°C; onto a hot heat exchanger surface, which will exhibit a degree of sensible cooling besides the generation of a depressed dry bulb temperature and perhaps an improved overall performance.

On the negative side and depending upon the fluid temperatures and thus surface temperature of the heat exchanger surface, large amounts of water can result in unacceptable build-up of scale, which over time will decrease the performance of the product. Clearly this aspect of the product behaviour is directly related to the quality of the water. Such systems are often restricted to operating <=200 hours per year.

Furthermore, scaled-up heat exchanger surfaces can often be very difficult to manually clean due to their delicate nature and often not suitable for chemical cleaning because of the normal materials of construction. Thus, the only remedy would be to ensure that softened water of the correct quality be used or regular replacement of the product. Some manufacturers claim that water softening can extend the operation to perhaps 1000 hours per year.

The second method – spraying away from the coil – is the preferred methodology to reduce some of the implications associated with the first method as described above.

Generally, this concept endeavours to take full advantage of the adiabatic evaporative cooling effect resulting from introducing water into a relatively dry air stream.

Note : Any ambient air stream, which is close to or even saturated (100% RH), will not benefit from the depressed dry bulb temperature implications resulting from the introduction of water … this applies to cooling tower technology as well. However, unlike a cooling tower where saturation efficiencies (μ) are extremely high, the water spraying methodologies are only likely to reach μ = 80%. As a consequence, the expectation that a fluid/refrigerant temperature can closely approach the operating wet bulb temperature is just not realistic or indeed feasible.

As a minimum requirement it is recommended to use Hydrophilic coated (ALHy) fin material with a fin pitch of >=2.5 mm. This aluminium fin surface treatment is suitable where fin corrosion issues may be a concern as it promotes film-wise water dispersion and assists in reducing water agglomeration in the heat exchanger.

Refer to the System Components below, which are similar for both the EASiSpray and EASiPad systems.

SYSTEM COMPONENTS

EASiSpray comprises …

- Water sparge pipe distribution system fitted with quick release spray nozzles to suit the product configuration. Unless otherwise informed, the nozzles are rated for 2-3 barg mains water pressure
- Pipe work and nozzles are positioned to naturally drain to the lowest point of the system to avoid water retention and avoid any risk of Legionella
- If requested a valve pack can be supplied comprising two control solenoid valves and a manual isolating ball valve

The suitably sized solenoid valves comprise …

- 'Normally Closed' city water valve
- 'Normally Open' drain valve (mounted at the lowest point in the system)

The 10W control coils for the solenoid valves can be supplied with either a 24 VAC or 230 VAC supply and are wired in parallel to operate simultaneously.

On special request other items such as non-return valves, strainers, pressure regulation valves, pumps, UV-lamps etc. can be supplied.

NOZZLE DETAILS

Nominal flow rate for 90° & 130° Lechler ceramic nozzles …

						l/min		
Angle	Type		Matl			2 bar	3 bar	Part #
90°	302,326	.	51	.	KB	0.40	0.49	616041345
	302,406	.	51	.	KB	1.00	1.22	616041346
	302,486	.	51	.	KB	1.60	1.96	616041347
	302,606	.	51	.	KB	3.15	3.86	616041348
130°	302,368	.	51	.	KB	0.63	0.77	616041349
	302,408	.	51	.	KB	1.00	1.22	616041350
	302,468	.	51	.	KB	1.40	1.71	616041351
	302,488	.	51	.	KB	1.60	1.96	616041352

EASIPAD SYSTEM

The pre-cooling pad technology has been given the name **EASiPad**, which potentially allows the dry cooler/condenser to operate 'wet' for sustained periods; unlike the direct spraying alternative, which typically has a time cap of 200 hours per year.

A further benefit of the EASiPad system is that none of the added water meets the heat exchanger surface, thus eliminating scale and maintaining the longevity of the extended surface. Furthermore, a specially treated (*AlHy - hydrophilic*) extended surface is not required, as is the case with direct spray systems. Additionally, no water carry-over is exhibited and thus no detrimental 'drift' implications are associated with this system.

A spin-off from, in effect, fitting a 'filter' to the incoming air stream ensures that the heat exchanger is kept clean and maintains its performance. Most dry coolers/condensers become dirty or clogged, thus impairing the cooling air volume and/or corroding the extended surface.

EASiPad can either be factory fitted or retro-fitted to both 'legacy' vee-type products and indeed competitor's products. It can also be supplied as a water distribution and pad system only or complete with water control valves and can be either independently controlled by the End User or interfaced with the capacity control system factory fitted to the product.

SYSTEM COMPONENTS

A EASiPad section comprises …

- Casework assembly in either stainless steel 304/316 or AlMg with a sloping drain tray at the base and at the top, fitted with water distribution system …
  - The initial design, which operated for some 8-10 years, utilised a Ø32 mm PVC water pipe distribution system with closely placed Ø2.5 mm holes on the underside to allow water to fall downwards onto the 30 mm deep top distribution section of similar pad material
    - Each pad section was fitted with a flow control 'Setter' to enable adjustment of the water flow rate
  - In around 2022 the American eco-Air variant of the Vee-type product range, developed a distribution set-up utilising flat profile, interfering spray nozzles which provided better distribution, less influenced by product inclination for fully draining equipment and reduce the water flow rate by approx. -50%
    - Rather than using Setters to adjust the water flow, the spray nozzles rely upon a PRV (pressure regulation valve) that ensures the necessary spray pattern
- This distribution portion of the pad material provides for lateral water dispersion prior to it entering the main pad material where axial dispersion is enabled via the angled 'flutes'
- Unless otherwise informed, the water piping is rated for approx. 3 barg mains water pressure and supplied in hydraulically crimped copper pipe and brass fittings

- The pipe work is positioned to naturally drain to the lowest point of the system to avoid water retention and avoid any risk of Legionella
- If requested, a valve pack can be supplied comprising two control solenoid valves and a manual isolating ball valve.

The suitably sized solenoid valves comprise ..

- 'Normally Closed' city water valve
- 'Normally Open' drain valve (*mounted at the lowest point in the system*)

The 10W control coils for the solenoid valves can be supplied with either a 24 VAC or 230 VAC supply and are wired in parallel to operate simultaneously.

On special request other items such as non-return valves, strainers, pumps, UV-lamps etc. can be supplied.

PAD TECHNICAL DATA

GigolA® COOLING & DARKENING PADS

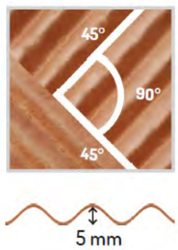
- 600 mm wide pads
- 150 mm deep
- 30 mm thick distribution section
- No sunlight penetration
- UV protected
- Algae, fungus & bacteria protected

Dry weight : 4.4 kg/m^2

Wet weight : 8.8 kg/m^2

Air pressure drop, Pa : 13.4 x [Face Velocity / 2.5]$^{1.35}$

GigolA® 50/90 PADS



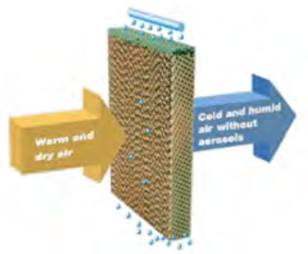
A more recent (circa 2020) replacement pad from the same supplier, where tests have confirmed a saturation Efficiency of 86.5% compared with the Darkening Pad figure of around 70% is the **Gigola's 50/90** pads.

Unlike the 50/90 designation suggests, the 5 mm Flutes are straight and angled at 45° in both directions.

Air pressure drop, Pa : 32 x [Face Velocity / 2.0]$^{1.848}$

PRINCIPLE OF OPERATION

When fitted to a dry cooler or air-cooled condenser, EASiSpray & EASiPad provides a 'final stage' option or 'high ambient' option to match the load requirements of the system.



EASiPad is a total loss water system where water is overfed to ensure that the pad material is kept wet and to ensure the correct adiabatic cooling effect. As a result, Legionella related worries; which are often associated with recirculating systems; are not a concern with this type of system where water retention is eliminated.

Capacity control of a dry cooler or condenser can be achieved by one of the following methods …

- Step control - fan cycling
- Variable voltage fan speed control
- Frequency inverter fan speed control
- EC motor fan speed control

STEP CONTROL

A basic step controller (*also known as fan cycling*) monitors the dry cooler's fluid leaving temperature or condenser head pressure and compares this with the programmed 'Set Point'.

If the measured condition exceeds the Set Point, the fan(s) (*either individually or in groups*) are activated sequentially to provide sufficient cooling to match the load. If the load cannot be matched when all the fans are running, then the final stage of the step control sequencer is to activate the EASiPad valves i.e. shut the drain valve and open the city water valve and thus feed water to the distribution pipework.

The functionality of the variable voltage, frequency inverter and EC systems described below is identical, however, the manner in which the variable fan speed is achieved differs, but this relates only to the underlying electronics of the controller/motors, not its overall functionality.

VARIABLE VOLTAGE/FREQUENCY INVERTER/EC CONTROL

All these controllers either have sensors to monitor the dry cooler fluid leaving temperature or condenser head pressure and compare it with the programmed 'Set Point'. If the measured condition exceeds the Set Point, the fans are increased in speed to provide sufficient cooling to match the load. If the load cannot be matched when all the fans are running at full speed, then when the monitored temperature or pressure drifts past a predefined offset the EASiSpray/EASiPad valves are activated.

In both scenarios above, the application of the water into the air stream will provide a cooling thermal shock to the system and likely provide over-cooling. As a consequence, the monitored temperature or pressure will eventually fall below the Set Point and thus the water valves will be deactivated i.e. the city water valve will close and the drain valve will open allowing the distribution pipework to drain.

Thereafter, in the case of the step controller, fans may be sequentially deactivated thereby reducing the air volume and cooling effect or in the case of the variable fan speed control options, the fan speed and air volume will be reduced resulting in the same effect.

Over time as the effects of the added water lessens, the monitored condition will increase causing fans to be activated or increased in speed until again, the EASiSpray/EASiPad system is reactivated.

If 'short cycling' of the EASiPad system is expected to be a problem, the control system can include for a timer to ensure that the pads are continually wetted for a minimum period. During this period the variable fan speed control options would allow the fan speed to reduce, thereby reducing the consumed electrical power.

During operation of the EASiSpray system, the nozzles provide a homogeneous 'fine mist' spray either beneath a flatbed product or in front of the coils of a vee-type product. The induced air stream will pass through this mist and adiabatically absorb moisture increasing the air stream's relative humidity and depressing the dry bulb temperature.

Adjustment to both the spray nozzle orientation and city water supply pressure can be used to modify the spray pattern.

WATER QUANTITY AND QUALITY

The water quantity/flow rate required to be supplied to the EASiPad system is detailed on the technical data sheet generated by CoilCalc. The figure is derived from the ambient air dry bulb temperature and specified %RH, calculated saturation efficiency plus the air volume of the product. Thus, the amount of water varies depending upon the size of the product and how dry the entering ambient air is. The water flow rate defined in litres/minute will be different for each specific application. As mentioned earlier, unless otherwise defined, the design operating pressure is assumed to be between 2 and 3 barg.

Water supplied to the EASiSpray system should be compliant with drinking water requirements defined in European Directive 98/83/EC.

The recommended water quality parameters to ensure longevity of the system and minimise coil fouling issues are …

- Hardness : 8-12°f (4.5 - 6.7°dH)
- $MgCO_3/CaCO_3$: <100 ppm
- pH : 6.5 - 8.5
- Chlorides : : <20 ppm

- Conductivity : <100 µS/cm

END USER RESPONSIBILITIES

It is the responsibility of the End User / Installer to ensure that a clean, filtered city water supply is provided to the EASiPad system.

If the EASiSpray/EASiPad option has been supplied as a retrofit option, then it is the responsibility of the End User to ensure that the distribution pipework is installed correctly to ensure that it will freely drain when the drain solenoid valve (positioned at the lowest point in the system) is deactivated and thus open.

In the case of the EASiPad system, the End User / Installer should ensure that all the unevaporated water residing in the sloping drain tray is able to drain to the sewer unimpeded.

Local regulations may require the fitting of an in-line non-return valve and it may be wise to fit a pressure regulation value to ensure a constant water pressure is maintained.

The water distribution sparge pipe is designed to operate in a pressure range between 2-3 barg to ensure a correct water distribution.

Furthermore, frost protection precautions should be considered, such as trace heating etc., during winter conditions when the EASiSpray/EASiPad system would not normally be activated. In such low temperature conditions, it may be wise to drain the city water supply line back to a point that is not subject to frost damage.

DISCLAIMER

The EASiSpray system is designed **only** for 'Peak Load' occasions when the original 'dry mode' design/operation of the product is insufficient … perhaps on unusually hot summer days.

As a consequence, the EASiSpray system should <u>not</u> be operational for more than **200 hours per year**.

BACKGROUND

Acoustics is a very broad subject; however, I shall confine this topic to the areas concerning our market sector.

Sound is regarded as the fluctuation of air pressure at a frequency which is audible, typically 20 Hz to 20,000 Hz.

Noise, on the other hand, is often defined as ‘unwanted sound’.

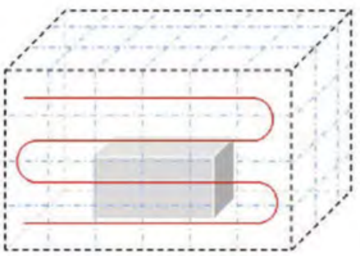
The presentation of sound data for our type of products has evolved over the years. Originally data relating to hemispherical sound propagation was published, but this was often open to ‘artistic licence’, and it was difficult to compare one manufacturer’s data with a competitors.

Eurovent was instrumental is attempting to ‘level the playing field’ with its adoption of EN13487’s parallel-piped envelope methodology to calculate and present overall average sound pressure levels at a distance, derived from the certified sound power level spectrum.

NOISE STANDARDS : EN ISO 3744 & EN13487

EN ISO 3744 is the de-facto standard for determining the sound power level for equipment. A variety of measurement methods are described, which can be conducted in anechoic or semi-reverberant chambers or alternatively free field environments. However, all result in providing the sound power spectrum for the equipment via defined microphone positions and sound pressure level readings.

EN ISO 9614 describes the measurement of sound intensity using specialised twin headed microphones and integrating measurement meters, which ultimately enables the sound power level, L_w to be derived.



Measurement using either discrete points (*EN 9614-1*)or using a scanning method (EN 9614-2) are alternative methods described in the standard.

In view of the somewhat more sophisticated & expensive equipment needed to conduct this type of measurement, ISO 3744 methodology is more commonly performed and has become the preferred method adopted by Eurovent since 2020.

However, there is a consensus that suggests that intensity-based sound power derivation (ISO 9614) results in slightly lower figures than sound pressure based measurements (ISO 3744). However in reality, on-site measurements can restrict the ability to perform intensity measurements.

EN 13487 is a standard and is derived from ISO 3744 and adapted to the more practical ‘in field’ measurement of sound data using the parallel-piped discrete microphone position methodology. In view of its adaption and more practical implementation, Eurovent has adopted this standard as a mechanism to transpose the certified sound power level measurement of a product into an overall average sound pressure level at a prescribed distance from the extremities of the product.

In doing so, it is now feasible to compare competitor’s sound pressure level data calculated from their certified sound power level data on a ‘like for like’ basis, thus eliminating the confusion associated with the various interpretations of the hemispherical propagation methodology.

DEFINITIONS

The fundamental unit of sound is the 'bel' (B), named after *Alexander Graham Bell*, but it is more common to use decibels (dB), which is 1/10th of a bel. Confusingly, both sound power and sound pressure are quoted in decibels (dB), however, each has its own definition.

SOUND INTENSITY

Sound intensity is defined as the amount of energy passing through a unit area per unit time and is denoted by the symbol **I** and expressed in Watts/m^2.

There is a direct relationship between intensity and sound pressure; intensity being directly proportional to the pressure squared.

$$\mathbf{I \propto p^2}$$

where, *I = intensity, W/m^2*
p = pressure, Pa (N/m^2)

It is more usual to express the sound intensity relative to a threshold level, defined as 10^{-12} W/m^2 (1 pW/m^2) and present it in a logarithmic form.

$$\mathbf{L_I = 10 \times \log_{10}(I / I_0)}$$

where, *L_I = sound intensity level, dB*
I = sound intensity of source, W
I_0 = reference sound intensity, 10^{-12} W/m^2

SOUND PRESSURE

Sound pressure is defined as the local pressure deviation from atmospheric pressure caused by a sound wave and denoted by the symbol, $\mathbf{L_P}$ and is expressed in N/m^2 or Pa.

However, it is more usual to express the sound pressure level relative to a reference level, defined as 2 x 10^{-5} Pa and presented in a logarithmic form …

$$\mathbf{L_p = 20 \times \log_{10}(p / p_0)}$$

where, *L_p = sound pressure level, dB*
p = sound pressure of source, Pa (N/m^2)
p_0 = reference sound pressure, 2 x 10^{-5} Pa

Sound pressure is what the human ear responds to and is directly influenced by the environment in which the noise source is located.

Sound pressure levels are distance related, whilst sound power is an energy level and unrelated to distance.

A simple analogy is to consider a 1 kW electric heater … this is equivalent to the sound power level. When stood in front of the heater the temperature that can be measured … this is equivalent to the sound pressure level … will fall as the distance from the heater increases, but the heater is still emitting 1 kW of energy.

Furthermore, if the 1 kW heater is fully enclosed, then there will be no rise in the temperature compared with the surrounding … equivalent to fully attenuating a noise source and thus not being able to hear it.

CONVERSION : SOUND POWER TO PRESSURE

The relationship of the measured sound pressure level at a given distance to the sound power level of the source or alternatively, the calculation of the sound pressure level at a given distance from the source's sound power level, is surface area related and given by …

$$\mathbf{L_p = L_w - 10 \times \log_{10}(A)}$$

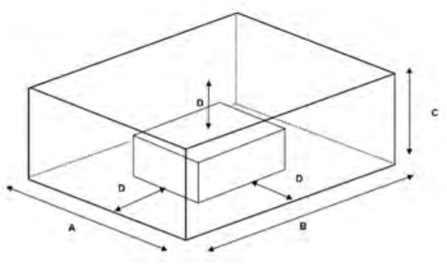
where, *L_p = sound pressure level, dB*
L_w = sound power of source, dB
A = enclosing surface area, m^2

The surface area under consideration is governed by the method defined, which could be hemispherical or parallelpiped, where the latter is preferred for our type of equipment.

Basic acoustic 'point source' theory considers that sound pressure waves emanate in a spherical fashion or hemi-spherically if the source is mounted on or above a flat reflective surface.

Clearly, for our type of product, which is not suspended in mid-space, we always consider the location to be on a flat reflective surface, thus hemispherical. However, when applied to a large product, such a shaped envelope is impractical or difficult to conduct measurements. Therefore, the acoustic standards such as ISO 3744 / EN 13487 offer an alternative surface area scenario.

Our type of equipment can be considers as having rectangular sides, ends and plan elevations and thus lends itself to considering a parallel-piped shaped enclosing envelope.



where, *D = distance at which L_p is required, m*
A = product width + 2D, m
B = product length + 2D, m
C = product height + D, m

Parallel-piped Surface Area = (AB + 2AC + 2BC) … in m²

Note : It is important to understand that the sound pressure level derived from the sound power level is an ***'overall average'*** *figure and not a specific value at the specified distance at all positions around the product.*

Some dry coolers and condensers e.g. Vee type or 'upright' flat-bed products, exhibit directivity characteristics, where the sound pressure level at a given distance from the ends of the product is less than at the same distance from the long open coil side of the product. This can be in the order of 3-4 dBA. Furthermore, the sound pressure level above the product can be up to 6 dBA higher than from the ends.

ADDITION OF NOISE SOURCES

As the whole basis of acoustics in founded upon logarithmic scales, combining noise sources involves the use of logarithmic addition. This applies to both sound pressure levels and sound power levels presented in dB.

Considering two noise sources with identical sound powers, W Watts each, then by definition …

$L_{w1} = 10 \log_{10}(W/W_0)$ — *sound power level for one source only*

$L_{w2} = 10 \log_{10}(2W/W_0)$ — *sound power level with both sources emitting noise*

therefore, performing the maths …

$L_{w2} = 10 \log_{10}(W/W_0) + 10 \log_{10}(2)$

$L_{w2} = L_{w1} + 3$ dB

So two identical noises sources will generate a sound power level 3 dBs higher than when only one is emitting noise.

Applying the same logarithmic rationale to more than one noise source yields …

$\mathbf{L_{w2} = L_{w21} + 10 \times \log_{10}(n)}$ — *where, n = number of noise sources*

As an example, consider a product fitted with 8 identical fans each having a sound power level of 73 dB. The total sound power level with all 8 fans running will be …

$L_{w2} = 73 + 10 \log_{10}(8)$

$= 73 + 9.03$

$= \mathbf{82.03}$ ***dB***

When combining noise sources of different pressure or power levels, then resorting to basic logarithmic addition must be followed …

$$\mathbf{L_T = 10 \times Log_{10}[10^{(L_1/10)} + 10^{(L_2/10)} + 10^{(L_3/10)} + \ldots]}$$

where, *L_T = total sound power or pressure level, dB*
L_1 .. L_3 = power or pressure level of each source, dB

OCTAVE BANDS

When a noise source is sampled using a sound level meter, the captured sound pressure levels can be presented as either a full octave or 1/3rd octave spectrum, depending upon the instrument.

Unless detailed analysis is required to identify a discrete frequency, which may be the cause of a noise nuisance, sound data is usually presented as full octave data (8 frequencies).

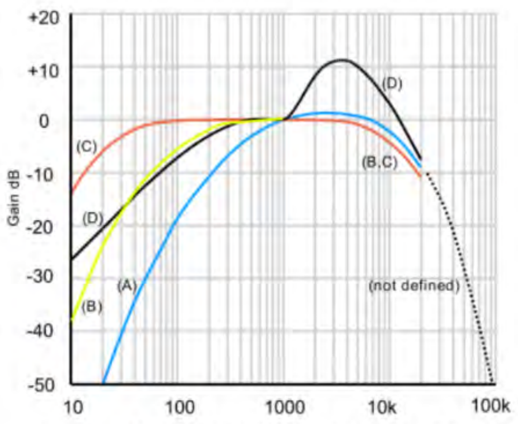
	Ful Octave Bands							
Frequency - Hz	63	125	250	500	1K	2K	4K	8K

Although the audible range is often considered to be 20 Hz to 20 kHz, for the purpose of our industry sector, 63 Hz to 8K is often more than sufficient and the standard format for both measured sound pressure levels and derived sound power levels.

The 1/3rd octave band alternative covers 30 discrete frequencies (25 to 20000 Hz), but whose range actually covers 22 to 22500 Hz.

A-WEIGHTING

A-weightings are applied to measured (full or 1/3rd octave linear) sound pressure levels in an effort to account for the relative loudness perceived by the human ear, as the ear is less sensitive to low audio frequencies compared with frequencies above 1000 Hz.



A-weighting (blue), B (yellow), C (red), and D-weighting (blk)

It is applied by arithmetically adding discrete values, listed by frequency bands, to the measured sound pressure levels in dB. The resulting octave band measurements must then be added (logarithmically) to provide a single A-weighted value describing the sound pressure level and is denoted by **L_pA** and denoted by units of dBA.

Sound pressure level meters can often display both the linear (unfiltered) sound pressure spectrum and A-weighted spectrum plus the overall A-weighted dBA value. It is this latter single value which is most often used when presenting sound pressure data or used when comparing competitor's products.

The following table presents the adjustments that should be applied to a full octave linear sound pressure or power level spectrum.

	Ful Octave Bands							
Frequency - Hz	63	125	250	500	1K	2K	4K	8K
A-weighting - dB	***-26.2***	***-16.1***	***-8.6***	***-3.2***	***0***	***1.2***	***1***	***-1.1***

To serve other industry sectors and applications, alternative weighting systems are available, namely … B, C, D and more recently Z.

SOUND POWER

Sound power is defined as the rate at which sound energy is emitted from a source and denoted by the symbol, **L_w** and is expressed in J/s or Watts.However, it is more usual to express the sound power relative to a threshold level, defined as 10^{-12} W and present it in a logarithmic form …

$$\mathbf{L_w = 10 \times \log_{10}(W / W_0)}$$

where, L_w *= sound power level, dB*
W *= sound power of source, W*
W_0 *= reference sound power, 1 pW (10^{-12} W)*

Situation and sound source	Sound power (W)	Sound power level (dB ref 10^{-12} W)
Saturn V rocket	100,000,000	200
Project Artemis Sonar	1,000,000	180
Turbojet engine	100,000	170
Turbofan aircraft at take-off	1,000	150
Turboprop aircraft at take-off	100	140
Machine gun Large pipe organ	10	130
Symphony orchestra Heavy thunder Sonic boom	1	120
Rock concert Chain saw Accelerating motorcycle	0.1	110
Lawn mower Car at highway speed Subway steel wheels	0.01	100
Large diesel vehicle	0.001	90
Loud alarm clock	0.0001	80
Relatively quiet vacuum cleaner	10^{-5}	70
Hair dryer	10^{-6}	60
Radio or TV	10^{-7}	50
Refrigerator Low voice	10^{-8}	40
Quiet conversation	10^{-9}	30
Whisper of one person Wristwatch ticking	10^{-10}	20
Human breath of one person	10^{-11}	10
Reference value	10^{-12}	0

An indication of the magnitude of typical noise sources is presented which helps explain the reason for using a logarithmic scale to denote the sound level.

Sound power cannot be directly measured but can be derived from sound pressure level measurements or sound intensity measurements, which are governed by a variety of international standards.

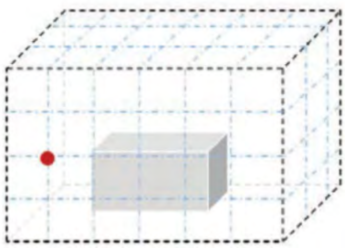
Measurements can be conducted in either …

- Anechoic chambers
- Semi-reverberant rooms
- Free-field

For the purpose of this guide we shall restrict consideration to the Free-field methodology and it is generally accepted that **ISO 3744** – *Acoustics : Determination of sound power levels of noise sources using sound pressure : Engineering method in an essentially free field over a reflecting plane* – is the definitive source for conducting such measurements.

ISO 3744 covers both hemispherical and parallel-piped methods and for practical reasons in our industry sector, the parallel-piped method is preferred.

Additionally, **EN 13487** – *Heat exchangers : Forced convection air cooled refrigerant condensers and dry coolers : Sound measurement* - is practically used to conduct measurements or transpose sound power level into overall average sound pressure level *(at the surface of the envelope)* at a distance.



This standard has been adopted by ***Eurovent*** to ensure that all participants calculate and publish sound data on the same basis.

Note : Sound power levels are independent of distance.

SOUND BEHAVIOUR

Generally, sound is considered a good characteristic whilst noise is considered a nuisance and often needs attenuating.

Coils per say do not emit any noise, unless perhaps they are exhibiting cavitation or water hammer issues, however, products containing coils, which are also fitted with fans do generate noise.

Every type of fan has a unique sound power characteristic and when fitted to a product, issues sound pressure waves that are attenuated by nature of the design of the product. So some equipment is naturally quieter due to the casework construction, which obstructs the emanating sound pressure waves.

Published sound power data for fans is usually divided into the ‘suction side, L_wA_5’ and ‘discharge side, L_wA_6’ sound spectrums. EAS’s products need to consider the logarithmic addition of these two noise sources to enable the analysis of the total sound power generated by a particular piece of equipment to assess the sound pressure level at a given distance.

Regenerative noise is also a consideration, which if the product is badly designed, will increase the expected noise level.

As the sound power level comprises two components, each has the potential for attenuation to enable the overall noise level of the product to be lowered.

Although a significant noise reduction can be achieved by lowering the fan speed, the associated reduced air volume may cause a thermal underperformance and thus not a practical option.

The fitting of attenuators to the suction side or discharge side or indeed both sides, will reduce the sound power level of the fan although there are practical considerations which may negate the use of this methodology.

BACKGROUND

Classical acoustic theory considers point sources, where the sound radiates equally in a uniform spherical divergence and the sound intensity, **I** diminishes in accordance with the inverse square of the distance, **r**.

$\mathbf{I \propto 1 / (4\pi r^2)}$ *for spherical propagation*

When the above is applied to reality and the noise source is placed on a flat reflecting plate i.e. hard ground or roof of a building, then hemispherical analysis applies, which when the mathematics is conducted, results in the following traditional equation for the attenuation in sound pressure level with distance …

$\mathbf{I \propto 1 / (2\pi r^2)}$ *for hemispherical propagation*

Thus …

$\mathbf{L_p = L_W - 20 \times \log_{10}(r) - 8}$ *where,* *L_p = sound pressure level, dB*
L_w = sound power of source, dB
r = distance (radius) to the observer, m
$8 = 10 \log_{10}(2\pi)$

The typical distance (radius) often used to present sound pressure data is **10 meters** and thus the above becomes …

$\mathbf{L_{p(10\,m)} = L_W - 28}$

Furthermore, the above also provides the other, often used, approximation for estimating the sound pressure level at alternative distances …

$\mathbf{L_{p2} = L_{p1} - 20 \times \log_{10}(r_2/r_1)}$ *where,* *L_{p1} = known sound pressure level @ r_1, dB*
L_{p2} = required sound pressure level @ r_2, dB

Caution : the above is strictly only applicable to point source theory and care should be taken if used with our products … which are not point sources !!

One problem with misapplying point source theory to a large product follows from the question 'What is the distance (radius of the hemisphere) that should be used?', when the product is say, 12 m long, 2.5 m wide and 2.8 m high.

If the specification distance was 10 meters, point source theory would consider a 10 m radius hemisphere centred at the mid-point of the product. This results in measurement positions only 4 meters away from the end of the 12 m long product, whilst the same 10 m radius gives a position of 8.25 m away from the long side of the product.

Thus, the fans towards the ends of the product will have a much greater influence upon the measured/calculated sound pressure level at effectively 4 m distance, compared with the influence of more of the fans when located on the long side at 8.25 m. Clearly there is a problem with this approach.

Another approach is to consider that the 'specification distance' of 10 m is taken from farthest edge of the product related to the mid-point of the product. Thus, in this example, the farthest dimension is 6 m to the end of the product. So, if we apply the 10 m distance to this, we are considering a hemisphere of 16 m in radius … a significantly larger surface area and thus a somewhat lower predicted sound pressure level than if we consider the 10m from the mid-point of the product when the product has the same overall power level.

To avoid such confusion, ISO/EN 3744 & EN 13487 and its parallel-pipe methodologies, eliminate the misinterpretation and ensures that all manufacturers are transposing the product's overall sound power level into sound pressure level at a distance, in the same fashion.

Products such as dry coolers and air cooled condensers are generally reasonably large and often fitted with multiple fans … noise sources. Therefore, they cannot be considered as a ‘point source’, but perhaps more correctly, considered as a ‘line of point sources’. Consequently, the sound pressure level attenuation with distance is quite different and there is a real risk of incorrect prediction if the above simple point source equations are applied.

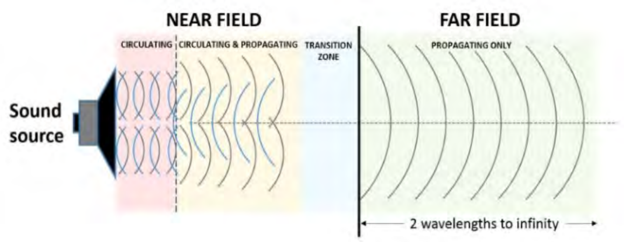
Point source theory states that sound pressure decays by 6 dBs every time the distance is doubled, whilst a line of point sources (motors fitted to a product) more closely emulate ‘line source’ behaviour, where the decay rate is 3 dBs per doubling of distance.

To complicate matter still further, the construction of dry cooler and condenser products involves the use of reasonably large sized areas of sheet metal, which can be considered to act as a ‘finite plane source’.

The acoustic behaviour of a finite plane source has 3 zones …

- Zone I – where the sound pressure level is constant – no decay
- Zone II – where the sound pressure follows the ‘line of point sources’ decay relationship of
  $L_{p2} = L_{p1} - 10 \times \log_{10}(A)$ … 3 dB each doubling of distance
- Zone III – where the sound pressure follows the ‘point source’ decay relationship of
  $L_{p2} = L_{p1} - 20 \times \log_{10}(A)$ … 6 dB each doubling of distance

This latter behaviour is relevant to large products, where moving from 1 meter to 2 meters often indicates no reduction in the sound pressure level reading (*circulating*), whilst moving from 2 meters to say 5 meters follows the Zone II behaviour (*Near field*) and beyond that, the Zone III behaviour begins to apply (far field).



However, the application of ISO 3744/EN13487 allows for the calculation of the parallel-piped enclosing envelope surface area, which when fed into the governing equation provides the overall average sound pressure level at the specified distance based upon the total sound power level of the product.

$$\mathbf{L_p = L_w - 10 \times \log_{10}(A)}$$

So for dry cooler and condenser products the above approach is the only safe way to calculate the <u>overall average sound pressure level</u> at a given distance if the sound power level of the product is known.

Alternatively, the required or minimum sound power level of the product can be predicted if the sound pressure level at a distance is to be met, using this technique.

According to EN 13487, consider firstly a ‘small’ product of say 1 x 1 x 1 meter in size. If the sound pressure level is required at 10 m, then the parallel-piped surface area enacted upon is 1365 m^2 and thus 10 $\log_{10}$(1365) = 31.4, whilst a ‘large’ product fitted with pads of say 12 x 2.8 x 2.8 meter in size results in a surface area of 2132.5 m^2 and thus 10 $\log_{10}$(2132.5) = 33.2

So the difference in the attenuation figure is not so large and an average estimate would be …

$L_p = L_w – 32$ *for a distance of 10 meters*

Note *the difference between the* ***-28*** *adjustment for ‘point source theory’ and the* ***-32*** *for parallel-piped theory … 4 dB !!*

SOUND ATTENUATION

Sound attenuators or silencers entrain the incoming or leaving air stream through either a rectangular or circular sheet metal housing where the internal perimeter is filled with sound absorbing material, such as Rockwool, acoustic foam and even lead lined acoustic material.

RECTANGULAR SILENCERS

Such discharge silencers are simple and only marginally successful due to attenuation only being achieved at the perimeter of the rectangular housing. At best, such silencers may provide -2 dBA attenuation.

CIRCULAR SILENCERS

Circular silencers similar to the above rectangular concept are again mediocre in their performance and might provide a -2 dBA reduction.

However, an improvement to the design of a circular silencer is to use an internal acoustically filled 'pod', creating a double sided acoustically lined anulus through which the air stream passes, which improves the attenuation achievable.

Although for induced draft fan products, the fitting of discharge silencers may be straight forward, and their impact; if they were to eliminate all the discharge sound energy from the fans; would theoretically only be a -3 dBA reduction in the overall sound pressure level of the product.

Some manufacturers claim that discharge silencers can reduce the noise level of a product by upwards of -5 dBA, which is hard to believe !! Unless perhaps, the design of the silencer improves the discharge efficiency of the fan allowing its speed to be reduced to achieve the same air volume, compounded with the acoustic attenuation of the silencer, and then perhaps such reductions are justifiable. Increased efficiency equates to lower noise levels, but the main reason for the improved reduction relates to the reduced fan speed.

For products with induced draft fans fitted on the top of the product, the addition of suction side silencers is often not a practical solution. This would result in the fan being sandwiched between both an inlet and perhaps a discharge silencer, each perhaps 750 mm to 1000 mm in length, resulting in a significant increase in product height … besides the added weight of the product and windage related structural implications.

VERTICAL SPLITTERS

An alternative to suction side silencers in the case of a flat-bed product, is to fit vertical acoustic 'splitters' beneath the coil.

Splitters attenuate the noise emanating from the suction side of the fan, which 'breaks out' through the coil on the underside of the product. The thickness of, and distance between the splitters, governs the attenuation achievable.

Clearly, closely spaced splitters are acoustically more effective, but the imposed pressure drop may affect the thermal performance and may ultimately increase the noise level of the product if the fan speed needs to be increased to overcome the additional static pressure imposed by the splitters.

Again, the levels of attenuation are likely to be relatively small if air restriction implications should be minimal.

ACOUSTIC BARRIERS

By far the best way to reduce the sound level reaching an 'observer' is to erect a barrier /wall between the equipment and observer, but this is often not aesthetically pleasing and can be rather costly.

Furthermore, if the location of the critical noise level requirement is at an elevation, a barrier may not provide a solution.

FAN SPEED REDUCTION

By far the most effective way to reduce the noise level of a product is to reduce the fan speed of all the fans. Slower running fans deliver less air volume, which directly affects the thermal performance. Therefore this option often creates a challenge.

Nevertheless, a 25% reduction in fan speed equates to a -6 dBA noise reduction and a 50% decrease in fan speed equates to a -15 dBA reduction in noise level.

Oversizing the product to be able to meet the capacity demands at a reduced fan speed and thus lower noise level is an expensive option, however, acoustic control via reduced fan speed is viable when night-time low noise level requirements are associated with reduced ambient temperatures or reduced thermal capacity demands, where the product's capacity can be maintained when operating TDs are wider.

INTRODUCTION

Structure-borne noise in building systems can lead to unwanted and unacceptable noise levels in commercial building applications. With this being the case, it is common practice for anti-vibration mounts (AVMs) to be specified in the mounting of capital equipment in air-conditioning applications such as air handling units and chillers. The purpose of these devices is to isolate the vibration of the operating machinery from the building, reducing the structure-borne noise in the building system.

The selection of the AVMs is critical, as an incorrectly specified product can be ineffective in its aim to reduce transmission of structure-borne noise, or cause the equipment to operate in resonance, leading to damage.

RESONANCE

Resonance occurs when a vibration or external force is applied to a system at its natural frequency (the resonant frequency), which causes the vibration to amplify. A common example of this is the shattering of a crystal wineglass when exposed to a tone of the correct pitch; the wavelength of the sound excites the wineglass at its resonant frequency, causing it to vibrate then shatter. Resonance does not cause vibration; it simply amplifies it.

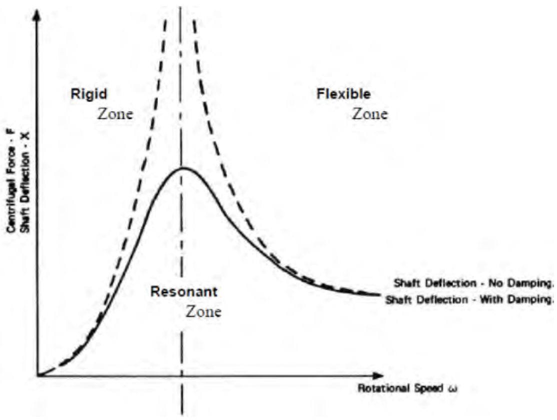
In fans, the speed of the fan is a source of excitation of vibration, and it follows that with higher speeds there are higher frequencies of system vibration. Given that the fan system has a resonant frequency, operating the fan at a speed corresponding to the resonant frequency will result in excessive vibration, which, over time, can lead to structural damage to the fan.

The strategic design of fan systems is required such that the fan speed, and the vibration caused by the rotational speed, does not cause the system to operate in a region of resonance. Fans can be rigidly mounted to solid bases without the need for AVMs, and operation from 0-100% speed will not cause the fan to operate at a resonant frequency.

SELECTION OF ANTI-VIBRATION MOUNTS

Where AVM are specified, their correct selection is crucial, as the addition of AVM changes the speed at which resonance occurs. The new resonant frequency is governed by the AVM selected, which is generally based on the isolation efficiency required, the spring deflection and fan operating speed range. When selecting AVM for use with fans it is imperative that the fan operating speed in the application is known so that a correct AVM selection can be made.

The diagram to the side describes the rigid zone, resonant zone and flexible zone that are present in the operation of an AVM.



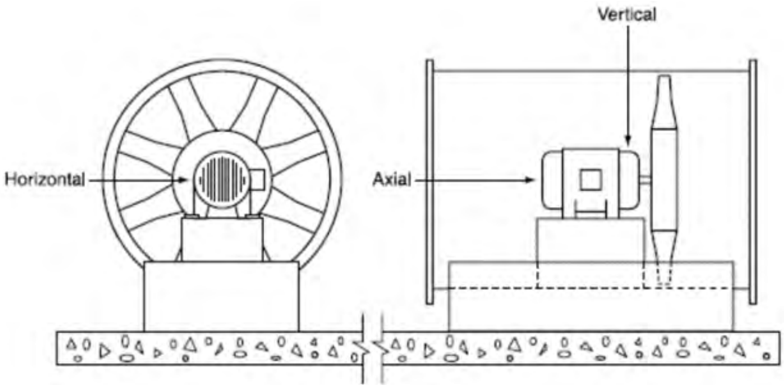
AVMs should be selected based on continuous operation in the flexible zone; it is here that vibration isolation is achieved. In the case of speed-controlled AC or EC fans, it is important to take this into consideration, as operating the speed of the fan varies, and an incorrectly selected AVM could cause the fan to operate in the resonance region as the fan speeds up and slows down.

For fan systems operating at a speed in the rigid zone, there is no vibration isolation and the AVM has no effect.

MEASURING VIBRATION AND VIBRATION LEVELS

The vibration level of the equipment can be measured through the use of a vibration transducer. ANSI/AMCA Standard 204-05 Balance Quality and Vibration Levels for Fans defines the suitable measurement locations as axial, vertical and horizontal. In any installation orientation, “measurement shall always be made in a radial direction and perpendicular to the axis of rotation.” In the case of a vertical reading, the measurement needs to be taken perpendicular to the axis of rotation and perpendicular to a horizontal reading. An axial vibration measurement needs to be taken parallel to the shaft. An axial measurement shall always be made parallel to the shaft (rotor) axis of rotation. An example of each vibration measurement point is shown in the following figure.

The measurement of vibration at all three points is recommended to be undertaken across the full speed range to attain a complete picture of the vibration present in the application.



The balancing grade of many fans is G6.3 and therefore as per the standard ISO 14694 Industrial fans - *Specifications for balance quality and vibration levels*, fan application BV-3 applies.

For flexibly mounted fans, vibration velocity levels of less than 3.5mm/s is recommended (RMS), in line with the BV-3 levels. For rigidly mounted fans, vibration velocity levels of less than 2.8mm/s (RMS) is recommended, in line with the BV-3 levels.

If the vibration levels are exceeded in any measurement orientation, it suggests that the fan is operating in resonance and selection of AVMs or minimum fan speed needs to be reconsidered.

RUBBER MOUNTS

Captive rubber anti-vibration mounts are anti-vibration elements which subject the rubber to shear and compression forces. Their high profile rubber sections produce larger deflections for low natural frequencies. This range of mounts is suitable for applications where vibration isolation is a priority.

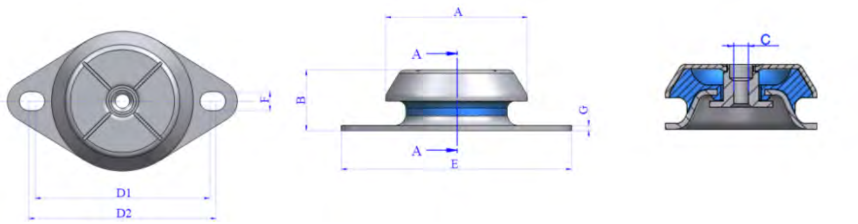
The mount is especially recommended for applications such as isolation of rotating equipment that is continuously subjected to shocks or are exposed to the elements.

APPLICATIONS

These mounts are suitable for the isolation of rotating machines which are exposed to axial and radial shocks or exposure to the weather. They are particularly suitable for applications where a high level of vibration isolation is required.

TECHNICAL CHARACTERISTICS

- The top metal hood protects the rubber from the Ozone, UV rays or oils which could damage the rubber
- The metal parts have a suitable anti-corrosive treatment for outdoor applications. RoHs compliant
- They have an interlocking metal mechanism that provides a fail-safe protection for mobile applications. This device limits vertical movement when the mounting is subject to shock loads
- The mounts are clearly identified with the type and stiffness engraved on the baseplates, making it possible to easily recognise the part even after several years of use
- The top cap has a cross shaped indentation, which enhances rigidity on mobile applications and also permits oils or liquids to flow off the mounting.

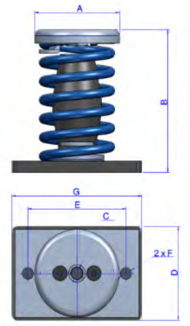


TYPE	A - mm	B - mm	C - mm	D1 (min)	D2 (max)	E - mm	F - mm	G - mm
BRB 50	50	25	M-8	61	70	85	6,5	2
BRB 60	64	35	M-10	76,5	90,5	110	9	2,5
BRB 65	64	35	M-12	76,5	90,5	110	9	2,5
BRB 70	64	35	M-12	100	100	120	11	3
BRB 80	83	35	M-12	108	112	134,8	11	3
BRB 95	92	39	M-12	122	126,6	150	10	3
BRB 110	106	41	M-12	137	150	175	13	3
BRB 125	123	48	M-16	154	162	190	14	4
BRB 150	155	53,5	M-16	176	188	218	14,5	4

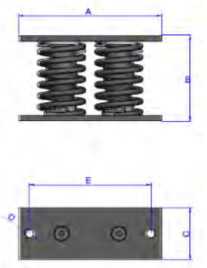
SPRING MOUNTS

Anti-vibration mounts using springs are an alternative to standard rubber mounts when improved vibration isolation is required, such as the location of equipment upon a roof. Springs are particularly efficient at damping low frequency vibrations.

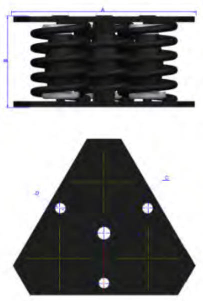
Although these mounts generally provide better isolation characteristics than the rubber alternatives, they are not however, suitable for equipment that may be subject to lateral movement from forces such a wind loads. In these cases the 'restricted' version of the spring mount (AMC AS) is recommended.



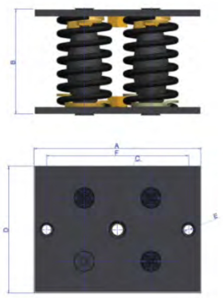
Type	A (mm)	B (mm)	C (mm)	D (mm)	E (mm)	F (mm)	G (mm)
1 AMC 150 - 350	75	122	M-12	80	87	10	115
1 AMC 500 - 750	90	122	M-14	100	120	12	150



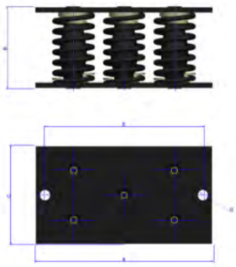
Type	A (mm)	B (mm)	C (mm)	D (mm)	E (mm)	F (mm)	G (mm)
2 AMC 300 - 700	75	122	M-12	80	87	10	115
2 AMC 1000 - 1500	90	122	M-14	100	120	12	150



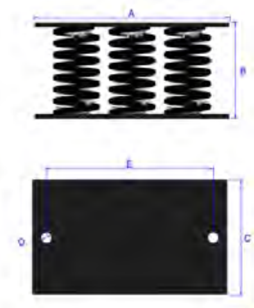
Type	A (mm)	B (mm)	C (mm)	D (mm)
3 AMC 450 - 750	190	124	M-16	12
3 AMC 1050 - 2250	242	124	M-20	14



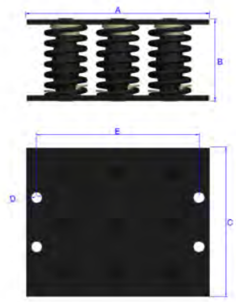
Type	A (mm)	B (mm)	C (mm)	D (mm)	E (mm)	F (mm)
4 AMC 600 - 1000	200	124	M-16	150	12	170
4 AMC 2000 - 3000	250	124	M-16	200	14	210



Type	A (mm)	B (mm)	C (mm)	D (mm)	E (mm)
5 AMC 750 - 1750	280	124	150	16	248
5 AMC 2500 - 3750	350	124	200	18	300/315



Type	A (mm)	B (mm)	C (mm)	D (mm)	E (mm)
6 AMC 900 - 2100	280	124	150	16	248
6 AMC 3000 - 4500	350	124	200	18	300



Type	A (mm)	B (mm)	C (mm)	D (mm)	E (mm)
9 AMC 1350 - 3150	280	124	226	16	248
9 AMC 4500 - 6750	350	124	300	18	310

RESTRICTED/ANTISEISMIC SPRING MOUNTS

Restricted spring anti-vibration mounts are an alternative to the standard spring mounts and contain a mechanical anchoring system to ensure equipment stability when subject to lateral forces, such as wind loads.

As springs are particularly efficient at damping low frequency vibrations, this variant of the standard spring mount ensures improved vibration isolation when equipment is located in sensitive areas such as upon a roof.

1 AMC ANTISEISMIC + RUBBER

Natural frequency: 3-5 Hz

AMC CODE	MOD.(NºSprings)	SPRING COLOR	MAX.LOAD(kg.)	DEFLECTION(mm)
20409	1	PURPLE	305	22
20381	1	GREEN	405	22
20382	1	GREY	540	22
20383	1	WHITE	612	22
20384	1	RED	803	22

2 AMC ANTISEISMIC + RUBBER

Natural frequency: 3-5 Hz

AMC CODE	MOD.(NºSprings)	SPRING COLOR	MAX.LOAD(kg.)	DEFLECTION(mm)
20494	2	PURPLE	610	22
20496	2	GREEN	815	22
20497	2	GREY	1080	22
20498	2	WHITE	1225	22
20500	2	RED	1610	22

4 AMC ANTISEISMIC + RUBBER

Natural frequency: 3-5 Hz

AMC CODE	MOD.(NºSprings)	SPRING COLOR	MAX.LOAD(kg.)	DEFLECTION(mm)
20700	4	PURPLE	1220	22
20696	4	GREEN	1620	22
20697	4	GREY	2160	22
20698	4	WHITE	2448	22
20699	4	RED	3220	22

6 AMC ANTISEISMIC + RUBBER

Natural frequency: 3-5 Hz

AMC CODE	MOD.(NºSprings)	SPRING COLOR	MAX.LOAD(kg.)	DEFLECTION(mm)
20761	6	PURPLE	1830	22
20762	6	GREEN	2430	22
20763	6	GREY	3240	22
20764	6	WHITE	3670	22
20765	6	RED	4820	22

9 AMC ANTISEISMIC + RUBBER

Natural frequency: 3-5 Hz

AMC CODE	MOD.(NºSprings)	SPRING COLOR	MAX.LOAD(kg.)	DEFLECTION(mm)
20961	9	PURPLE	2745	22
20962	9	GREEN	3645	22
20963	9	GREY	4860	22
20964	9	WHITE	5508	22
20965	9	RED	7227	22

BACKGROUND

At the very beginning of the PED Directive 2014/68/EU, Article 1/Scope states … '*This Directive shall apply to the design, manufacture and conformity assessment of pressure equipment and assemblies with a maximum allowable pressure PS greater than 0.5 barg*'.

Thus, any heat exchanger whose maximum allowable pressure/operating pressure, PS < 0.5 barg will fall outside the scope of the PED Directive.

All pressures are relative to atmospheric pressure i.e. gauge pressure, denoted by **barg**.

A '**Fluid**' as referred to by the PED, can be either a liquid, a gas, a vapour or a mixture thereof.

PED IMPLICATIONS

EAS is approved to manufacture Stainless Steel coils that fall into the following categories as defined in the PED 2014/68/EU and with a maximum **PS <= 120 barg** & **TS <= 300°C** (*BVC & Force Jan. 2025*) …

• Article 4/Para 3		*Shall NOT bear CE mark*	*Sound Engineering Practice*
• Category I • Category II • Category III • Category IV	Module B + D	*Must bear CE mark + ID 0062*	*Approved certification*

Furthermore, for Copper, Cupronickel, Aluminium & Titanium tube heat exchangers, EAS is approved to manufacture such coils that fall into the following categories as defined in the PED 2014/68/EU. The above pressure and temperature limits can be exceeded providing SEP (Sound Engineering Practice) has been considered.

• Article 4/Para 3	*Shall NOT bear CE mark*	*Sound Engineering Practice*
• Category I / Module A	*Must bear CE mark*	*Internal production control*

The PED categorisation of a coil involves the knowledge of …

- **PS**, Maximum Allowable Pressure (*may be the Design Pressure if this is the maximum limit*) in **barg** (gauge pressure … above atmospheric pressure)
- **TS**, Maximum Allowable Temperature in **°C**
- **V**, Volume of the largest header, V_H (Guideline 2/4) or total coil section volume, V_T including both inlet & outlet headers in **Litres** or **dm³**
- **DN**, Connection size in **mm**
    - o *Non-DN rated sizes or 'plain tail connections' are referred to via their **Inside Diameter** in millimetres.*

The fluid used inside the tubes; which can be either a single-phase liquid or gas or indeed a two-phase mixture; needs to be qualified as either **Fluid Group 1** (Dangerous) or **Fluid Group 2** (Safe) as this directly affects the categorisation.

Dangerous Fluids Group 1	Safe Fluids Group 2
* Explosive * Highly flammable * Flammable (where TS > Flashpoint) * Toxic * Oxidizing	* Inert * Non flammable * Non toxic
Examples ..... **Explosive** - R2, R3 **Flammable** - R11, R12, R17, R290, R600, R717 **Very Toxic** - R26, R27, R28, R39 **Toxic** - R23, R24, R25, R39, R48 **Oxidising** - R7, R8, R9	Examples ..... Air R728 - Nitrogen R744 - Carbon Dioxide R717 - Water Ethylene & Propylene glycol Hycool Aspen temper Refrigerants not in Group 1 Most thermal oils (TS < Flashpoint)

IMPORTANT NOTES

- *Fluids defined as 'flammable' according to Directive 67/548/EEC and thus denoted as Fluid Group 1 do not belong to this group when the maximum allowable temperature (TS) is <= Flash Point of the fluid. In this case the fluid is considered as Fluid Group 2 ~ Safe fluid.*
- *Some fluids, such as heat transfer oils, are considered safe i.e. Fluid Group 2 because their Flash Point is above 55°C and not considered dangerous at normal ambient conditions. However, according to Directive 67/548/EEC they may be deemed to belong to Fluid Group 1 if the maximum allowable temperature (TS) is > Flash Point. Such an oil corresponds with the definition of Article 9, section 2.1 and thus constitutes a 'flammable' Group 1 Fluid. See also PED Guideline B-20 for further clarification.*

DEFINITIONS

- **Gas** - includes gases, liquefied gases, gases dissolved under pressure, vapours and those liquids whose vapour pressure, P_{sat} at the maximum allowable temperature, TS is > 0.5 barg.
- **Liquid** - includes liquids whose vapour pressure, P_{sat} at the maximum allowable temperature, TS is <= 0.5 barg.
- **Boiling point** - is the temperature where the vapour pressure of a liquid equals the pressure of its surroundings and then the liquid changes into a vapour
- **Flash point** - is the lowest temperature at which a flammable liquid vaporises into a gas, which can be ignited with the introduction of an external source of fire
  The flash point is often used as one descriptive characteristic of a liquid fuel, but it is also used to describe liquids that are not used intentionally as fuels
- **Flame/Fire point** - a slightly higher temperature, typically +5/10K, is defined as the temperature at which the vapour continues to burn for at least 5 seconds after being ignited
- **Ignition temperature** - unlike the flash point, the ignition temperature does not need an ignition source. In other words, the ignition temperature is the lowest temperature at which a volatile liquid will vaporise into a gas and ignite without the help of any external flame or ignition source. As a result, the ignition temperature is of course higher than the flash or flame/fire points

STARTING THE CATEGORISATION PROCESS

The temperature at which the coil is operating is not directly part of the calculation process, however, knowledge of **TS**, 'Maximum allowable temperature' is required to determine the fluid's **Vapour Pressure**, $\mathbf{P_{sat}}$ - also known as **Saturation Pressure** - to ascertain whether the fluid is to be considered a liquid or a gas, dangerous or safe and which categorisation selection chart should be used.

Furthermore, TS is an important consideration regarding the strength of the materials of construction because the tensile stress reduces with temperature.

If a fluid has a Vapour Pressure, $\mathbf{P_{sat}}$ greater than 0.5 barg at the Maximum Allowable Temperature, TS it shall be treated as a **gas**, otherwise it shall be treated as a **liquid**.

Remember …

- It is not the operating temperature that is important, but the Vapour Pressure, $\mathbf{P_{sat}}$ at the maximum allowable temperature, TS. This vapour pressure, $\mathbf{P_{sat}}$ dictates which selection Chart 1 to 9 should be used
- **Only** if the purpose of the coil is to heat water to temperatures > 110°C, should **Chart 5** (Steam Generators) be used. For example, an industrial application using high temperature exhaust gas which is required to be cooled down using water (under pressure) inside the tubes and as a result is heated above 110°C … **otherwise Charts 1-4 & 6-9 must be used**.
- Furthermore, as mentioned at the beginning of this document, if PS <= 0.5 barg then the heat exchanger is outside of the scope of the PED Directive, but for documentation purposes, would be classified as an Article 4/3 … *Sound Engineering Practice* … and would not bear the CE mark
- If a coil comprises more than one discrete section e.g. has 2 or more sections, each with its own headers and connections, then each discrete section is considered as a 'pressure vessel' and needs to be categorised individually using the volume of the largest header or total volume of the discrete section.
  If these discrete sections are classified as Category I to IV, then each shall have its own nameplate and shall bear the CE mark.

1. Determine the type of pressure equipment being considered : Vessel or Piping System
2. Determine the state of the fluid in the pressure equipment : Gas or Liquid
3. Determine the hazard group of the fluid in the pressure equipment : Dangerous - Group 1 or Safe - Group 2
4. Select the appropriate hazard category chart : Charts 1 to 9
5. Determine the maximum allowable pressure and the defining volume or pipe size of the pressure equipment : PS, V or DN
6. Determine the PED hazard category : Article 4/3, Cat I, II, III or IV

COILS WITH MAXIMUM ALLOWABLE PRESSURE, PS > 0.5 BARG

Article 4/Para 3 coils (SEP)

Generally, for any heat exchanger manufactured by EAS, which shall be …

- Considered as a **Vessel** or **Piping System**
- Using a safe **Group 2 Fluid**
- Where the vapour pressure, $\mathbf{P_{sat}}$ of the fluid at the maximum allowable temperature, TS is <= 0.5 barg
- **Vessel** : PS x Defining Volume, V <= 10000 barL or **Piping System** : DN <= 200

… shall be classified as an Article 4/Para 3 coil and EAS can design and manufacture it following 'Sound Engineering Practice' (SEP) to ensure safe use. Consequently, such a coil **must not** bear the **CE** mark.

As an example, water is defined as a safe liquid falling into Fluid Group 2 and has a saturation temperature of 111.35°C at 0.5 barg. (*Generally, **110°C** is used as the limit …. to simplify matters*)

Thus, if the operating fluid temperatures are below this limit, the coil or Dry Cooler will be an Article 4/Para 3 and thus we do not need to consult the PED selection charts.

However, the pressure at which the coil can operate safely is dictated by the strength of the materials used for construction. So clearly, to comply with the SEP obligations, we would need to perform some calculations to ensure that the tube, header, end cap thicknesses and fittings (e.g. flanges) are sufficiently strong enough to cope with the pressure and temperature that the coil may be subject to.

The same would apply to any other Group 2 fluid (safe) if the saturation vapour pressure, $\mathbf{P_{sat}}$ at TS is <= 0.5 barg … even when PS <= 50 barg.

HOW TO CATEGORISE A COIL

The PED applies to pressure equipment where the Maximum Allowable Pressure, PS is > 0.5 barg. If PS <= 0.5 barg then the equipment is outside the scope of the PED and not deemed to be a pressure vessel, but SEP would still apply.

Where PS exceeds 0.5 barg, the PED must be a consideration and a decision made as to whether the equipment shall be classified as a Vessel or Piping as specified in the Directive. The following extracts are taken from the PED 2014/68/EU …

- **Article 1**
  - Para 2.1 - 'Pressure equipment' means vessels, piping, safety accessories and pressure accessories.
  - Para 2.1.1 - 'Vessel' means a housing designed and built to contain fluids under pressure including its direct attachments up to the coupling point connecting it to other equipment. A vessel may be composed of more than one chamber.
  - Para 2.1.2 - 'Piping' means piping components intended for the transport of fluids, when connected together for integration into a pressure system. Piping includes, in particular, a pipe or system of pipes, tubing, fittings, expansion joints, hoses or other pressure-bearing components as appropriate. ***Heat exchangers consisting of pipes for the purpose of cooling or heating air shall be considered as piping.***

- **Article 4**
  - Para 1.1 - 'Vessels' … *defines limits for gases and liquids falling into Fluid Group 1 & 2*

- Para 1.2 - Fired or otherwise heated pressure equipment with the risk of overheating intended for generation of steam or super-heated water at temperatures higher than 110 °C having a volume greater than 2 litres
- Para 1.3 - 'Piping' … *defines limits for gases and liquids falling into Fluid Group 1 & 2*

- ***Para 3*** *- Pressure equipment and/or assemblies below or equal to the limits in Sections 1.1, 1.2 and 1.3 and Section 2 respectively must be designed and manufactured in accordance with the* ***Sound Engineering Practice*** *of a Member State in order to ensure safe use. Pressure equipment and/or assemblies must be accompanied by adequate instructions for use and must bear markings to permit identification of the manufacturer or of his authorized representative established within the Community. Such equipment and/or assemblies* ***must not bear the CE marking*** *referred to in Article 15.*

Following the introduction of the PED into European law, certain problems with the implementation became apparent and a whole list of working guidelines (*currently more than 200*) were ratified to be used in association with the PED.

The PED Directive implies that a pressure equipment operating above 0.5 barg should be considered as a **Vessel**, which would have resulted in many tube and fin tube coils becoming Category coils and thus invoking manufacturing issues and additional costs. However, Article 1/para 2.1.2 provides a 'loophole', which was particularly exploited by Sweden resulting in **PED Guideline B-04**, previously known as PED Guideline 2/4

Guideline B-04, clarifies exactly how to classify a coil as either a **Vessel** or **Piping** in relation to its application and thus the resulting hazard category once certain calculations are performed and selection Charts 1 to 9 are consulted. *See the selection charts at the end of this document*.

Guideline B-04

Pressure Equipment Directive PED 2014/68/EU
Commission's Working Group "Pressure"

Guideline related to: Article 2(2) and (3)

Question	**Which type of pressure equipment is a heat exchanger ?**
Answer	Heat exchangers are considered to be vessels. As an exception, heat exchangers which consist of straight or bent pipes which may be connected by common circular header(s) made also from pipe are classified according to Article 2(3) last sentence as piping if, and only if, the three following conditions are met: - air is the secondary fluid, - they are used in refrigeration systems, in air conditioning systems or in heat pumps, - the piping aspects are predominant. For such heat exchangers with headers, the piping aspects are pre-dominant if $Cat_p \geq Cat_v$ where: Cat_p = Abstract category that would be applicable according to the PED if the heat exchanger were classified as piping using DN of the biggest header. Cat_v = Abstract category that would be applicable according to the PED if the biggest header, without the connecting piping, were classified as a vessel (i.e. for determining Cat_v, not the total volume V of the heat exchanger is taken into account, but only the volume V_H of the biggest header). When the result is $Cat_v > Cat_p$, the appropriate vessel classification shall be determined by using the volume of the entire heat exchanger (headers plus connecting tubes). The abstract category approach for determining the predominant aspect is limited to this specific application dealt with in Article 2(3). The use of this concept outside this context is not supported by the directive and thus is not permissible.
Reason	
Note	Piping heat exchangers which do not meet the requirements of the exception are not to be classified according to the last sentence of Article 2(3) as piping; they are to be classified as vessels. For example: - Heat exchangers which are not used in refrigeration systems, in air conditioning systems or in heat pumps, and for which the main purpose is to heat or cool the contained fluid by using the surrounding air; - Half-pipe coil or a similar « jacket » construction that heat or cool a vessel;

SPECIAL CASES

- In relation to Guideline B-04 …
  - ‘**Piping aspects are predominant**’ in the case of all EAS coils (*excluding tube-plate & header-box coils*)
  - Air is almost always the ‘secondary fluid’
  - If the coils are used in **air conditioning systems**, **refrigeration systems** or in **heat pumps**, then ….
    EAS coils can be considered as Piping, where the vessel aspects relate to the volume of the largest header
  - Coils which <u>**are not**</u> associated with **air conditioning systems**, **refrigeration systems** or in **heat pumps**, <u>must</u> be considered as **Vessels**

- Refrigerant evaporators and condensers …
    - Unless the maximum allowable pressure, PS is stipulated by the Client (*in writing*), we are obliged to invoke EN 378-2 to ascertain PS based upon the refrigerant's saturation pressure at a temperature defined by the standard …
        - Evaporators : +43°C
        - Condensers : +63°C
    - For strength validation purposes, plus PED nameplate requirements, the minimum and maximum allowable temperatures are required to be known and documented
        - In the case of condensers, TS is <u>not necessarily</u> the condensing temperature but the maximum possible hot gas compressor discharge temperature that the heat exchanger may experience. *For example, an Ammonia condenser fed via a reciprocating compressor may be condensing at 35°C, but the hot gas inlet temperature is likely to be around 120°C. Thus, in this case, TS = 120°C*
- Superheated water/steam heat exchangers …
    - The PED stipulates that if the heat exchanger is for use in 'Fired' or 'otherwise heated' boiler applications where either steam or superheated water is generated at temperatures <u>exceeding</u> 110°C and the total coil internal volume $\mathbf{V_T}$, exceeds 2 litres ….
        - In almost all cases the coil will exceed 2 litres and <u>must</u> therefore be considered as a 'Vessel'
        - Multiply **PS** by $\mathbf{V_T}$ (*total volume including both headers and connection tubes*) and compare the result with **Chart 5**
        - Identify Category
- Single circuit coils …
    - Coils with only one circuit or where the (number of circuits) / (number of sections) = 1 and therefore <u>do not</u> have any headers, shall be classified as **Piping** and thus **Charts 6 to 9** should be used for categorisation purposes. ***No Vessel aspects need to be considered.***

CATEGORISATION METHODOLOGY

Coil considered as Piping …

If and only if, the coil shall be used as part of an ….

- **Air conditioning system**
- **Refrigeration system**
- **Heat pump**

then according to **Guideline B-04** the coil can be considered as **Piping** where the **Vessel** aspect of the pressure equipment is the **largest header** …

- Determine $\mathbf{V_H}$, the internal volume of the largest header including the connection tube length in litres (dm^3)
- Multiply $\mathbf{V_H}$ by **PS** (Maximum allowable pressure in barg) and compare the result with **Charts 1 to 4** depending upon whether the fluid is a gas or liquid and whether the Fluid Group is 1 or 2
- Determine the Category Cat^V

Now …

- Consider the coil as **Piping** and determine the connection DN size or internal diameter in millimetres … *if the connection tube is not DN rated*
- Multiply **DN** by **PS** (Maximum allowable pressure in barg) and compare the result with **Charts 6 to 9** depending upon whether the fluid is a gas or liquid and whether the Fluid Group is 1 or 2
- Determine the Category Cat^P

If $\mathbf{Cat^P \geq Cat^V}$ then the coil is deemed to be **Piping** and the Category is that dictated by the Cat^P calculation, otherwise the coil **must** be considered as a **Vessel** and a totally new calculation performed using the total volume, $\mathbf{V_H}$ (including both headers).

If $\mathbf{Cat^V > Cat^P}$ (*following on from above*) or **the coil must be considered as a Vessel** e.g. used in an industrial process or outside of the scope of Guideline B-04, then …

- Considering the equipment as a Vessel, determine **V_T**, the Total Internal Volume of the coil section (including both headers) in litres (dm^3)
- Multiply **V_T** by **PS** (Maximum allowable pressure in barg) and compare the result with **Charts 1 to 4** depending upon whether the fluid is a gas or liquid and whether the Fluid Group is 1 or 2
- Determine the Category Cat^V

Thereafter …

- All coils classified as Article 4/3 **shall not** bear the **CE** mark
- All coils classified as Category I, II, III or IV **shall** bear the **CE** mark
- Additionally, **Stainless Steel** coils categorised as Cat I, II, III or IV **shall** also bear our Notified Body's Identification Code – **0062**

Historically, FORCE Technology was our Notified Body and oversaw manufacture of our category coils up to Cat III.

*The need to manufacture Cat IV coils as from 2017 resulted in **FORCE** Technology (ID 0041) Type Approving a Cat IV design according to Module B. Subsequently in October 2017 EAS attained Module D Approval (Quality Assurance) via **Bureau Veritas**, who has since become our official Notified Body (ID 0062) relating to Module D. FORCE Technology now advise on design, strength and welding aspects of the construction.*

Now EAS has Bureau Veritas approval to manufacture **Stainless Steel** coils that fall into the following categories …

• Article 4/Para 3		*Shall NOT bear CE mark*	*Sound Engineering Practice*
• Category I	Module B + D	*Must bear CE mark + ID 0062*	*Approved certification*
• Category II	Module B + D	*Must bear CE mark + ID 0062*	*Approved certification*
• Category III	Module B + D	*Must bear CE mark + ID 0062*	*Approved certification*
• Category IV	Module B + D	*Must bear CE mark + ID 0062*	*Approved certification*

Furthermore, for Copper, Cupronickel, Aluminium & Titanium tube heat exchangers, EAS is approved to manufacture such coils that fall into the following categories …

• Article 4/Para 3	*Shall NOT bear CE mark*	*Sound Engineering Practice*
• Category I / Module A	*Must bear CE mark*	*Internal production control*

COILS OPERATING AT ELEVATED TEMPERATURES

If the heat exchanger is categorized as an Article 4/3 coil then the operating temperature only influences the choice of materials taking into consideration the design and manufacture in accordance with Sound Engineering Practice to ensure that the product is 'fit for purpose' and will not pose a Health & Safety risk whilst in operation.

However, coils classified as a Category I, II, III or IV must be accompanied by associated proof calculations; via an EAS developed PED Design Tool; to verify the material strength for the selected materials and method of construction at the design temperature & pressure.

TEST PRESSURES

The PED defines the test pressure as …

PT = max. value [(PS x 1.43) *or* **(PS x 1.25 x Stress @ 20°C / Stress @ TS)] barg**

Although PT = PS x 1.43 is often sufficient for normal applications, elevated temperatures can adversely affect the results and the PED Design Tool should always be used to determine the test pressure.

PED DESIGN TOOL

All category coils must be accompanied by supporting documentation to verify that the material thicknesses and associated components are both strong and safe enough for their intended purpose at the stipulated design conditions.

The EAS PED Design Tool provides a mechanism to simply perform the necessary strength calculations upon all the materials and joint options associated with the applicable construction. Generally, all calculations are in accordance heat exchanger related design codes EN 13480-3, EN 10253-2/ASME B16.9 & EN 13445-3.

Data entry requires knowledge of …

- Maximum allowable pressure, PS (*January 2025 – Force approved maximum pressure is 120 barg*)
- Maximum allowable temperature, TS (*Current approved range : -50 / +300°C*)
- Maximum air temperature
- Tube & header material, diameter and wall thicknesses
- Type of welded joints and end caps
- Flanges and ancillary fittings

The PED Design Tool will provide a validated calculation summary sheet including the required Test Pressure, PT or a multipage document detailing all the strength calculations of all elements chosen for the heat exchanger construction.

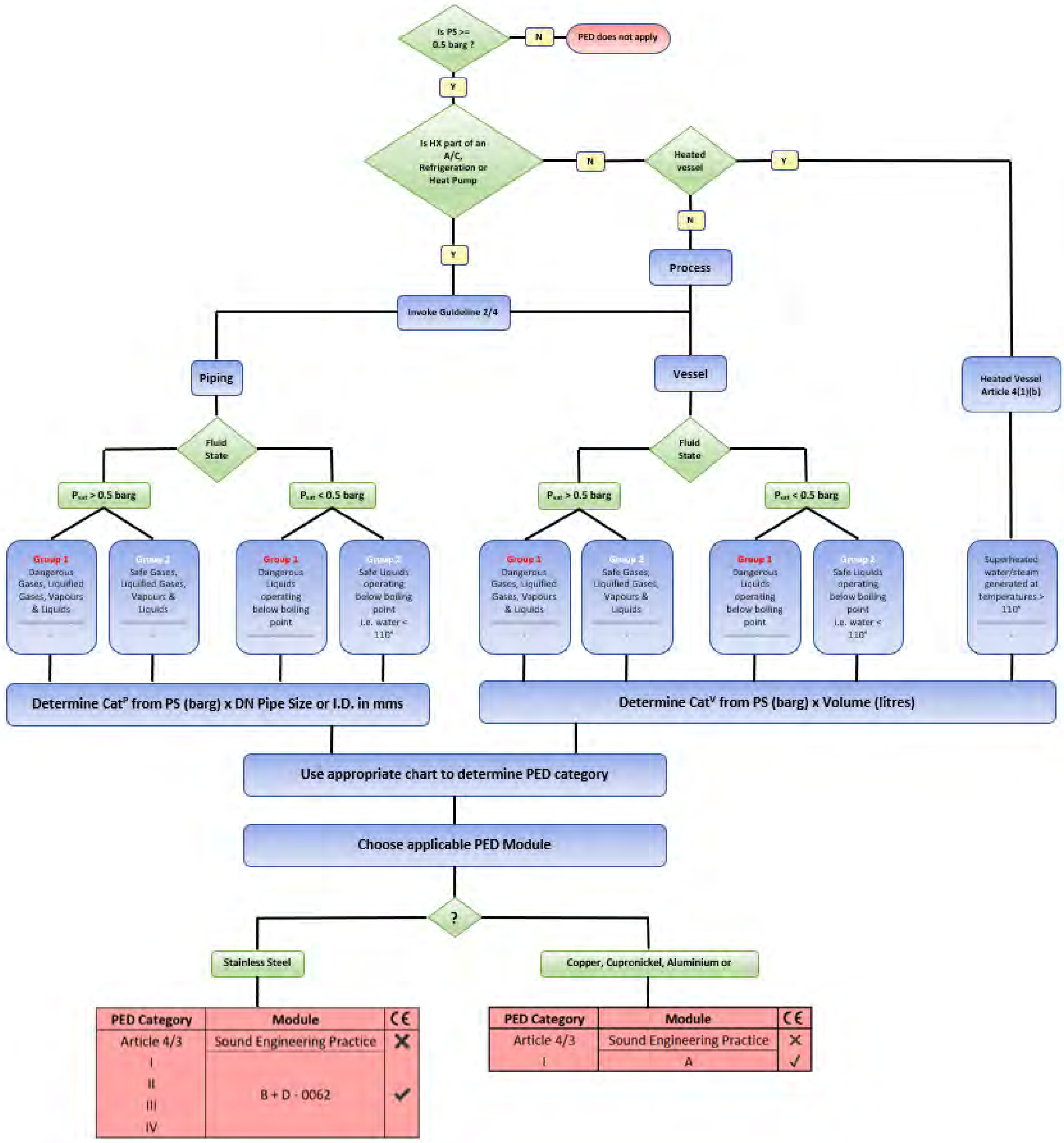
RECOMMENDATIONS

One should endeavour to manufacture Article 4/3 coils as this imposes the least restrictions etc. upon the design and manufacture. However, there will be numerous cases where, due to the fluid and operating conditions, the coil will become a Category I, II, III or even IV and thus require the necessary supporting documentation and potentially NOB involvement.

Except in the case of fired/boiler related applications (Chart 5) where the coil is always considered as a vessel, focus should be given to the size of the 'largest' header and the number of sets of connections/sections when performing the PED calculations.

Where a calculation will initially suggest that the coil is a Category Coil, a little 'artistic licence' such as considering a smaller header diameter or splitting the coil into 2 discrete sections with 2 sets of connections/headers, may well lower the classification of the coil resulting in an Article 4/3 solution.

Clearly the above is desirable to ease manufacture and reduce the associated documentation necessary to accompany the coil.



PED 2014/68/EU – EXTRACT

Article 4

Technical requirements …

1. *The following pressure equipment shall satisfy the essential safety requirements set out in Annex I:*

*(**a**) vessels, except those referred to in point (b), for:*

*(**i**) gases, liquefied gases, gases dissolved under pressure, vapours and also those liquids whose vapour pressure at the maximum allowable temperature is greater than 0,5 bar above normal atmospheric pressure (1 013 mbar) within the following limits:*

- *for fluids in Group 1 with a volume greater than 1 L and a product of PS and V greater than 25 barL, or with a pressure PS greater than 200 bar (Annex II, table 1),*
- *for fluids in Group 2, with a volume greater than 1 L and a product of PS and V is greater than 50 barL, or with a pressure PS greater than 1 000 bar, and all portable extinguishers and bottles for breathing apparatus (Annex II, table 2);*

*(**ii**) liquids having a vapour pressure at the maximum allowable temperature of not more than 0,5 bar above normal atmospheric pressure (1 013 mbar) within the following limits:*

- *for fluids in Group 1 with a volume greater than 1 L and a product of PS and V greater than 200 barL, or with a pressure PS greater than 500 bar (Annex II, table 3),*
- *for fluids in Group 2 with a pressure PS greater than 10 bar and a product of PS and V greater than 10 000 barL, or with a pressure PS greater than 1 000 bar (Annex II, table 4);*

*(**b**) fired or otherwise heated pressure equipment with the risk of overheating intended for generation of steam or super- heated water at temperatures higher than 110 °C having a volume greater than 2 L, and all pressure cookers (Annex II, table 5);*

I piping intended for:

*(**i**) gases, liquefied gases, gases dissolved under pressure, vapours and those liquids whose vapour pressure at the maximum allowable temperature is greater than 0,5 bar above normal atmospheric pressure (1 013 mbar) within the following limits:*

- *for fluids in Group 1 with a DN greater than 25 (Annex II, table 6),*
- *for fluids in Group 2 with a DN greater than 32 and a product of PS and DN greater than 1 000 bar (Annex II, table 7);*

*(**ii**) liquids having a vapour pressure at the maximum allowable temperature of not more than 0,5 bar above normal atmospheric pressure (1 013 mbar) within the following limits:*

- *for fluids in Group 1 with a DN greater than 25 and a product of PS and DN greater than 2 000 bar (Annex II, table 8),*
- *for fluids in Group 2 with a PS greater than 10 bar, a DN greater than 200 and a product of PS and DN greater than 5000 bar (Annex II, table 9).*

3. Pressure equipment and assemblies below or equal to the limits set out in points (a), (b) and (c) of paragraph 1 and in paragraph 2 respectively shall be designed and manufactured in accordance with the Sound Engineering Practice of a Member State in order to ensure safe use. Pressure equipment and assemblies shall be accompanied by adequate instructions for use.

Without prejudice to other applicable Union harmonisation legislation providing for its affixing, such equipment or assemblies shall not bear the CE marking referred to in Article 18.

- All pressures are in **barg**
- P_{atm} = Atmospheric Pressure, 0 barg (1 barA, 101.325 kPa)
- P_{sat} = Saturation Pressure @ Design Temperature, TS
- SEP = Sound Engineering Practice => **Article 4/3**
- If non-DN connection size use DN => I.D. in mms

CHART 1

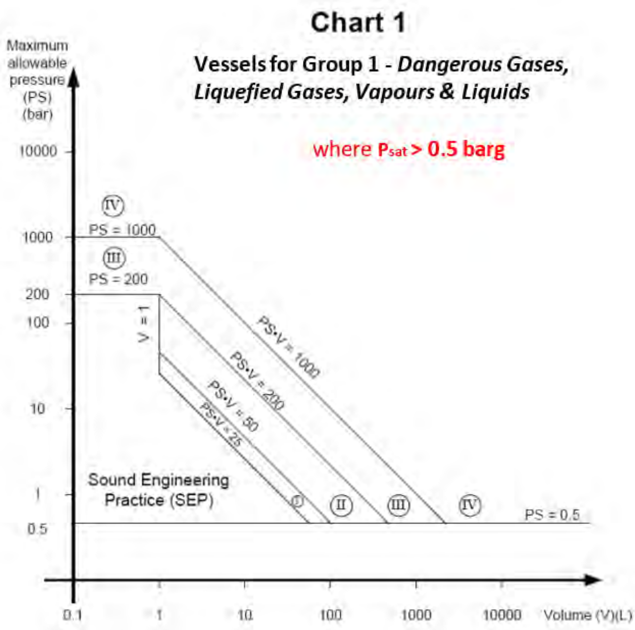
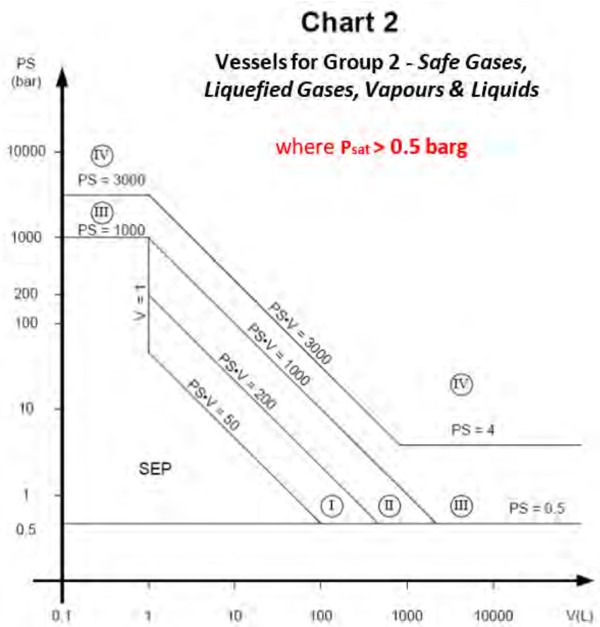
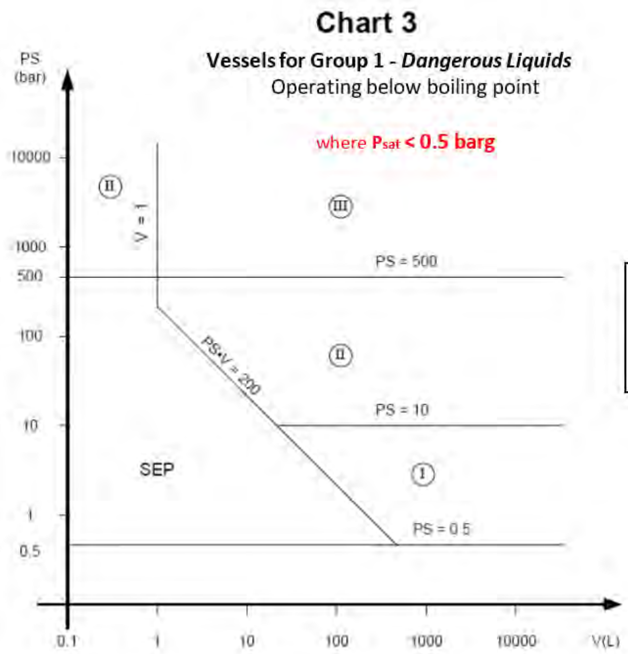


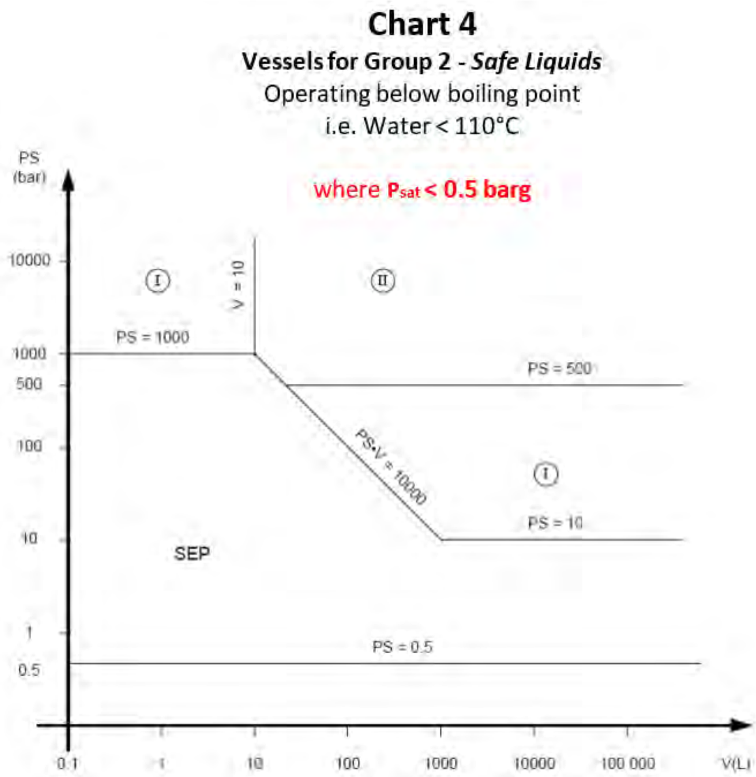
CHART 2





Note : *Use for liquids such as oils where TS < boiling point but TS >= Flame Point, are considered Dangerous Group 1*

CHART 4



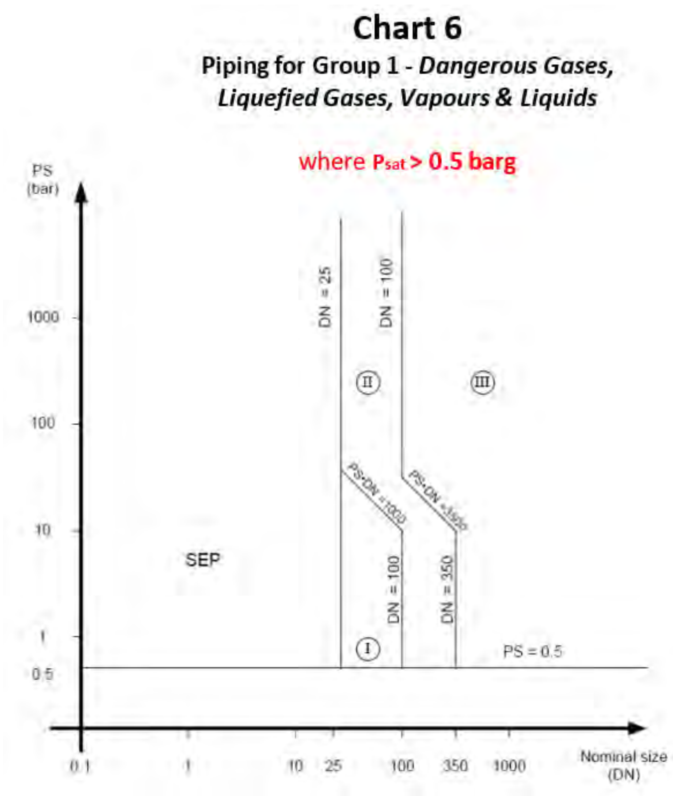


Chart 6
Piping for Group 1 - Dangerous Gases,
Liquefied Gases, Vapours & Liquids
where Psat > 0.5 barg
PS
(bar)
1000
100
10
1
0.5
SEP
DN = 25
DN = 100
II
III
PS·DN =1000
PS·DN =3500
DN = 100
DN = 350
I
PS = 0.5
0.1
1
10
25
100
350
1000
Nominal size
(DN)

CHART 7

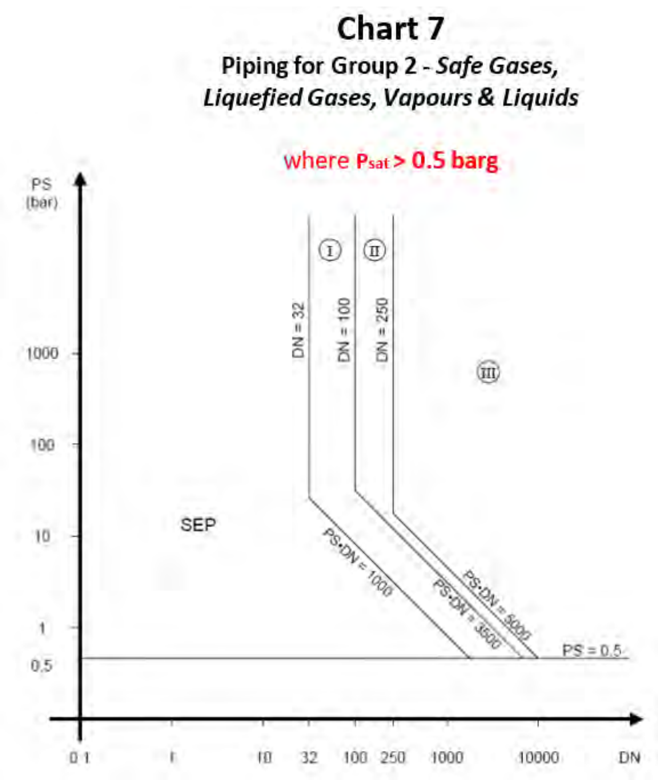


Chart 7
Piping for Group 2 - Safe Gases,
Liquefied Gases, Vapours & Liquids
where Psat > 0.5 barg
PS
(bar)
1000
100
10
1
0.5
SEP
I
II
DN = 32
DN = 100
DN = 250
III
PS·DN = 1000
PS·DN = 3500
PS·DN = 5000
PS = 0.5
0.1
1
10
32
100
250
1000
10000
DN

CHART 8

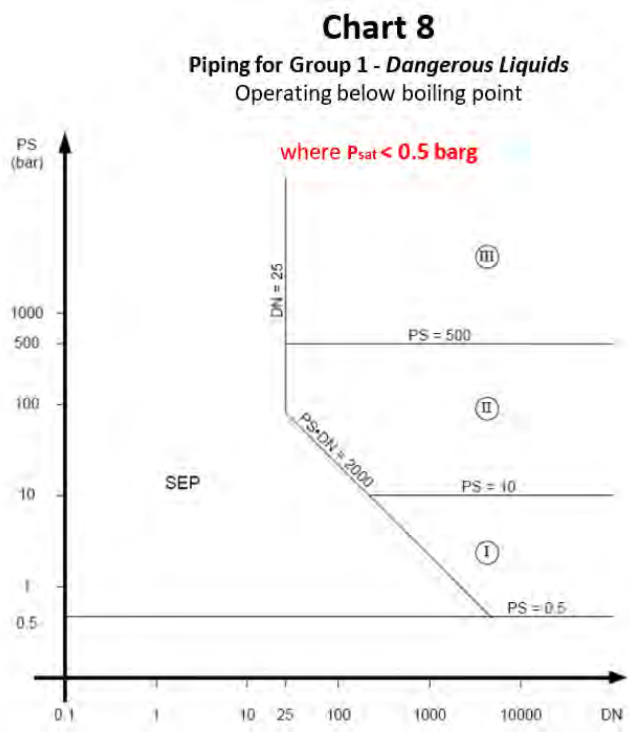
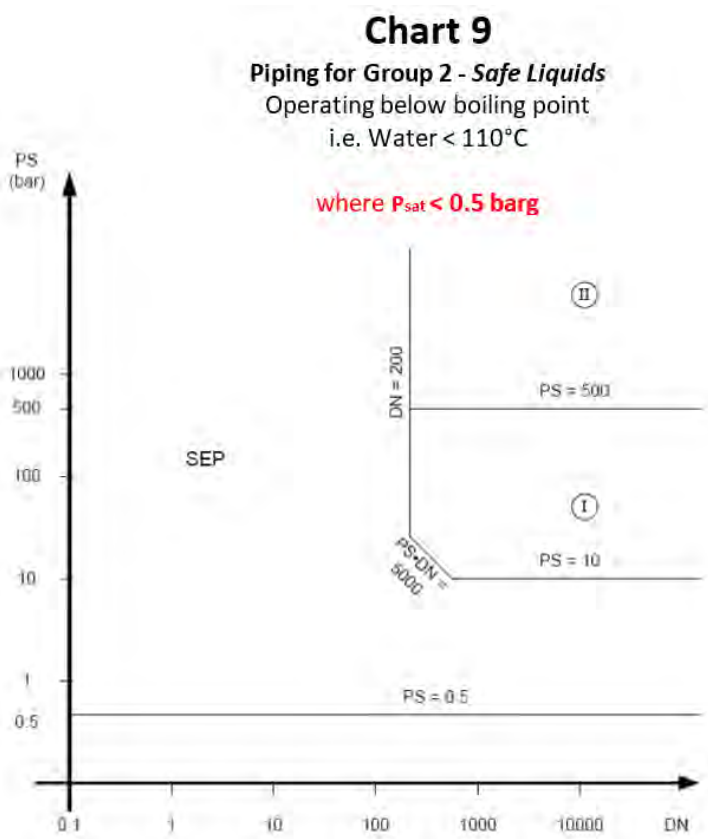
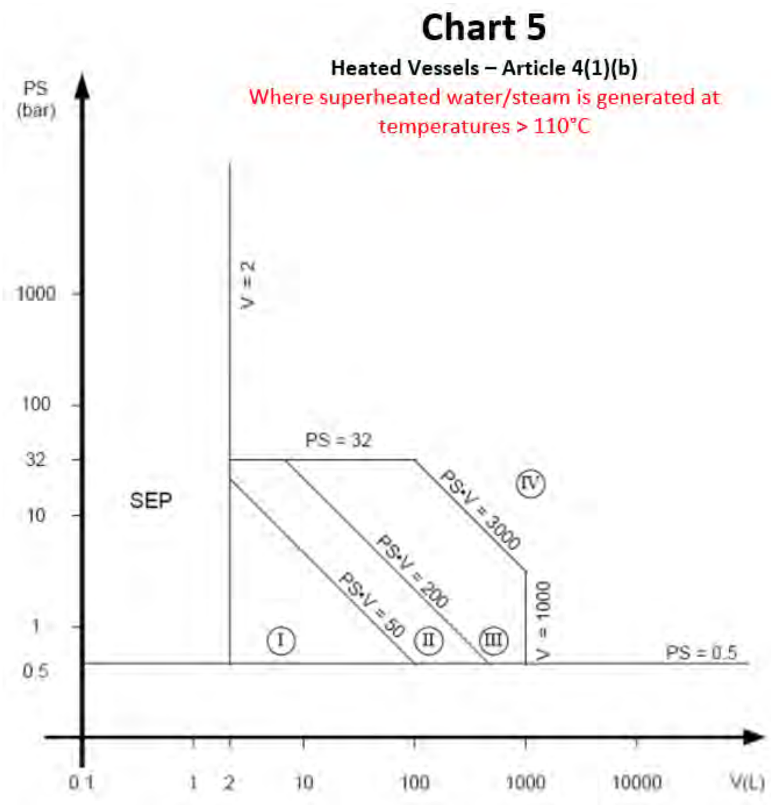


CHART 9





WORKED EXAMPLES

See *Appendix 2* for step by step worked examples.

Performing the PED categorisation of a coil, which can be classified as either a 'Vessel' or perhaps as 'Piping'; in accordance with PED Guideline B-04; necessitates an accurate estimate of the internal volume of the header assemblies and circuitry tubes.

The internal volume of the tubes within the fin block is straight forward to calculate, however, the internal volume of the header assemblies is dependent upon the header pipe diameters, wall thicknesses, header lengths plus connection tube lengths … for some coil variants, the inlet and outlet connection sizes and/or header sizes can vary.

Headers can be manufactured in copper, K65, aluminium, titanium, SS304L or SS316L and in the case of stainless steel, both DIN/EN standard header pipe sizes are used as well as ASME Sch 10, 20, 40, 80 & potentially 160 covering DN10 to DN150 (¼" to 6"). Therefore the envelope of permutations is vast and currently, CoilCalc only considers a small portion of the options.

Besides the header assembly material permutations, there are a multitude of 'connection fitting' options too, e.g. …

- Plain tail/end
  - No fittings, connection tube cut to length and possibly plugged with plastic cap
  - BFW – connection tube end prepared for welding (**B**evelled **F**or **W**elding)
- Threaded connection
  - BSP (male/female) parallel thread
  - EN 10226-2 : BSP (male/female) tapered thread
  - American MPT & NPS threads – male & female
- Flanged connection
  - EN 1092-1/11 : Weldneck flanges for PN 6 to PN 160 in sizes DN 10 to DN 150
    - Galvanised mild steel
    - SS304 or SS316 stainless steel
  - EN 1092-1/02 : Loose (Spinning) flanges - alternatives covering above DN & PN size/rating
    - Galvanised mild steel
    - SS304 or SS316 stainless steel
  - ANSI B16.5 standard flanges – weldneck & loose
  - Screw-on flanges matching BSP (male) connection
- Victaulic fitting
- Special non-standard Customer specified fittings

CoilCalc is not expected to consider all the above options but should consider the most common, such as …

- Plain tail/end
- BSPT(M) – male tapered thread, either achieved via TIG welded steel nipple or brass/bronze braze fitting
- EN standard weldneck flanges
- EN standard loose flanges

Other alternatives must be manually specified.

Note : Loose or spinning flanges comprise two parts, the stub-end (Type 32) which is brazed or welded onto the connection tube and the loose flange itself (Type 02)

Our industry sector still accommodates both Imperial & metric pipe and header sizes. However, apart from the Refrigeration industry sector, the traditional imperial connections sizes, such as ¼" through to 6", match the European DN sizes 10 to 150 and generally, the pipe O.D. are given in millimetres.

When considering copper coils with copper headers, there are typically two grades of piping namely, water and refrigeration. When copper is used for single phase fluid heating & cooling applications, a thinner wall pipe is used, which is typically available in metric sizes, Ø10 to 159 mm that mate with the applicable threaded of flange fitting.

BSPT EN10226-2 & Weldneck EN1092-1/11	Loose Flange EN1092-1/02	Connection Size	Copper		
-	-	10.0	10.0	1.0	611721037
-	-	12.0	12.0	1.0	611721012
-	-	15.0	15.0	1.0	611721028
DN10 - ¼"	DN15 - ½"	18.0	18.0	1.0	611721015
DN15 - ½"	DN20 - ¾"	22.0	22.0	1.0	611721016
DN20 - ¾"	DN25 - 1"	28.0	28.0	1.2	611721017
DN25 - 1"	DN32 - 1¼"	35.0	35.0	1.5	611721018
DN32 - 1¼"	DN40 - 1½"	42.0	42.0	1.5	611721019
DN40 - 1½"	-	48.0	48.0	1.5	611721020
-	DN50 - 2"	54.0	54.0	1.5	611721021
-	-	60.0	60.0	1.5	611721022
DN50 - 2"	-	64.0	64.0	2.0	611721023
-	-	66.7	66.7	2.0	611721029
DN65 - 2½"		76.1	76.1	2.0	611721024
DN80 - 3"		88.9	88.9	2.0	611721025
DN100 - 4"		108.0	108.0	2.0	611721026
DN125 - 5"		133.0	133.0	2.0	611721030
		133.0	133.0	3.0	611721031
DN150 - 6"		159.0	159.0	3.0	611721027

However, for the generally higher pressure applications associated with the refrigeration industry sector, a thicker wall thickness is used and such pipe is available in Imperial sizes …

		Copper		
	Connection Size	Conn. Tube mm	Wall Thk mm	Part #
Refrigerant - DX & CD	⅜"	9.52	0.76	611721034
	½"	12.70	0.85	611721001
	⅝"	15.88	0.90	611721002
	¾"	19.05	0.81	611721035
	⅞"	22.22	1.15	611721033
	1⅛"	28.58	1.27	611721003
	1⅜"	34.93	1.45	611721004
	1⅝"	41.28	1.72	611721005
	2⅛"	53.98	2.24	611721006
	2⅝"	66.68	2.77	611721008
	3⅛"	79.38	1.65	611721038
	4¼"	107.95	2.50	611721009

Usually, copper refrigerant coils are supplied with Plain Tail connections because they are usually brazed to the refrigeration system's pipework. On occasion, special flare fittings may be used, but these are usually provided and fitted by the Customer/Contractor.

For CO2 gas cooler applications, typically operating at 90 to 120 barg or some of the newer zeotropic refrigerant applications, the use of a high tensile copper alloy is used. Again this copper pipe variant is usually only available in Imperial sizes …

		K65 - CuFe2P		
	Connection Size	Conn. Tube mm	Wall Thk mm	Part #
Refrigerant - DX & CD	⅜	9.52	0.65	-
	½"	12.70	0.85	611721039
	⅝"	15.88	1.05	611721040
	¾"	19.05	1.30	611721041
	⅞"	22.22	1.50	611721042
	1⅛"	28.58	1.90	611721043
	1⅜"	34.93	2.30	611721044
	1⅝"	41.28	2.70	611721045
	2⅛"	53.98	3.50	611721046

When we consider the coil block tube diameters and materials used in heat exchanger manufacture, the following indicates the options available …

O.D. mm	Material	Wall Thk mm	Part #
9.52	Copper	0.28 0.50	611125006 611125002

15.00	Copper	0.38 0.75	611121003 611121005
	Aluminium	1.25	611111010
	Titanium	0.60	61135021
	SS304L	0.60	611131002
	SS316L	0.60 1.50	611135002 611735032

EAS DEFINITION OF COIL SECTIONS & SETS OF CONNECTIONS

The whole topic of coils split into separate sections with potentially multiple sets of connections, can be a bit of a minefield.

In our terminology, a coil (heat exchanger) will have at least one section and thus at least one circuit, where the inlet and outlet to this one section/one circuit coil may require some form of connection/coupling e.g. typically a threaded or flange fitting.

So we need to clarify the terminology …

- Number of **sections**
- Number of **sets of connections**

A coil comprising one section could have more than one set of connections but a coil with one set of connections will only comprise one section !

In some industry sectors, what we know as a ‘section’ can be referred to as a ‘circuit’ … especially in the refrigeration industry sector, where a refrigeration circuit infers a complete system comprising the compressor, condenser, evaporator and expansion device. So when a coil serves two or more independent refrigeration systems, we need to split the coil into discreet sections, each handling an individual system.

In our terminology, one ‘circuit’ can comprise a single tube e.g. inlet and outlet at opposite ends of the coil, such as a vertical tube steam coil or a number of interconnected tubes, say 4 tubes per circuit … also referred to by Evapco Inc. as **4 passes**.

In relation to the PED, splitting a coil into multiple discreet sections, where each discreet section has its own inlet and outlet connections or possibly more than one inlet/outlet connection, affects the Categorisation of each discreet 'pressure equipment' i.e. vessel or possibly, in our case, piping … Guideline B-04.

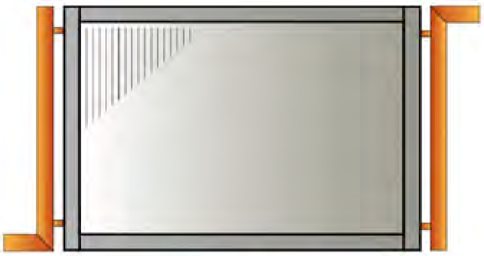
When we talk about 'connections' we are usually inferring the physical 'fitting' that may or may not be brazed or TIG welded onto the plain connection tail or header pipe.

These 'fittings' cover a multitude of types (DIN/EN or ANSI), sizes (DN or Imperial), pressure rating (PN or ANSI Class) and materials either dictated by the Client or the need to match the material of the header pipe and/or connection tube.

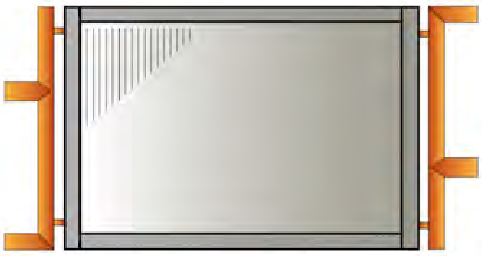
Usually, the volume flow rate of the fluid apportioned to each section dictates the physical size/diameter of the required fitting. This in turn, dictates the diameter of the connection tube and thus header pipe diameter.

If the resultant size/diameter of the required header pipe is too large to be accommodated withing the space available, one option is to use two or more sets of connections on a common header pipe which allows the header pipe diameter to be reduced.

For example, a coil requiring 1 x 4" connection and matching 4" header pipe can also be fulfilled by using a 2½" header pipe fitted with 2 x 2½" connection fittings …



1 section : 1 x 4" inlet/outlet



1 section : 2 x 2½" inlet/outlet

Both coils might be fed with the same volume flow rate, but the coil to the right with the 2 sets of connections allows a smaller header pipe of 2½" to be used because each connection is only handling 50% of the total flow rate. If sized correctly, the overall fluid pressure drop for both scenarios will be similar.

A variation to the above single section with two sets of connections is to split the coil into 2 discreet sections, each with its own inlet & outlet connection …

This coil solution would behave and function identically to the above coil with 2 sets of connections on a common header. However, from a PED perspective, now we have a coil with 2 discreet sections and thus half the internal volume per section. Therefore, depending upon the maximum allowable pressure (PS), maximum allowable temperature (TS), Fluid Group and application of the coil, will affect the PED Category calculation. Usually in such cases, the category will be lower and may even fall within the scope of a PED § Article 4/3 categorisation.

2 sections : 2 x 2½" inlet/outlet

The coils discussed so far are typical fluid heating and cooling coils fitted with same sized inlet and outlet header pipe/connections and to a lesser extent, refrigerant condenser coils, albeit that the latter variant would typically have a smaller condensate outlet header/connection size than the larger inlet hot gas header/connection.

However, when considering DX evaporators; unless the coil has a single circuit or a single circuit per section and thus no outlet (suction) header; then the inlet header shown in the above sketches is replaced with a distributor and distribution leads (capillaries).

Such DX coils can have one or more sections each with their own inlet distributors and leads plus outlet suction headers. However, there can be cases where a DX coil is required to have 2 uneven sections where Section 1 serves 1/3rd of the capacity, whilst Section 2 serves 2/3rd of the capacity. This is a clever way of achieving 33%, 67% & 100% capacity by only having 2 uneven refrigerant sections.

Another complication with DX coils with multiple sections, is that the sections can be arranged as follows to provide capacity control …

- Split in face : *One section above the other as indicated in the above sketch*
- Interlaced : *Where the circuits of each section are evenly distributed across the full face area of the coil*

There is a 3rd option, Split in Depth, but this is quite rare and is often handled by separate coils mounted in series.

Thermal performance wise, the alternatives behave similarly, but operational implications of the system may suggest one methodology is preferable.

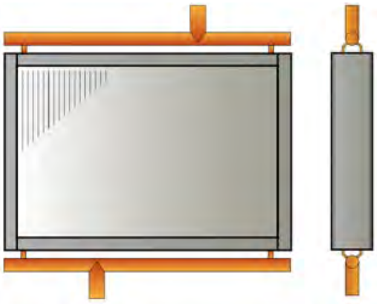
Interlaced sections, with each section's circuits spanning the coil face, necessitates full length headers and thus more material and associated cost compared with the 'split in face' option, which for a 2 section coil would use half height suction header lengths and thus half the header material consumption. Clearly, the distributor leads would be longer for the interlaced variant, but the implications are minimal.

Furthermore, there will also be an effect upon the coil weight where the interlaced variant will be heavier and the longer headers and thus greater internal volume may elevate the PEC category.

STEAM COILS

Steam coils are an oddity in so far that typically they are designed with vertical tubes and opposite end connections and thus the headers, instead of being vertical, are horizontally mounted at the top and bottom of the coil. In essence the coil is turned through 90° compared with other types of coils.

Other variants of steam coils are horizontal tube options using either hairpins or alternatively a 'transfer header' to provide a 'same end connection' solution. Additionally, another opposite end connection solution with horizontal tubes can be achieved by mounting the fin block on an incline inside the casework. These horizontal tube steam coil solutions are usually avoided.



1 section : 1 inlet/outlet

Steam coils are in essence condensers, however, due to the nature of their application involving the condensing of dry saturated steam into a saturated liquid (condensate), often operated cyclically (On/Off), they lend themselves to a vertical tube arrangement.

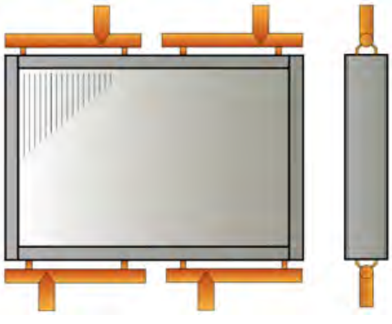
The above sketch of a 2 row, single section steam coil shows horizontal headers with the inlet & outlet connection tubes mounted as some location along the header. This is not necessarily typical and the header(s) can be extended horizontally to provide the connections. Thus, the steam enters the end of the header horizontally and may exit in a similar manner at the bottom of the coil. So the connection orientation is dictated by the Client's requirements.

Related to the above comments, we impose restrictions on the maximum header pipe length and also the number of rows that a header can distribute steam to …

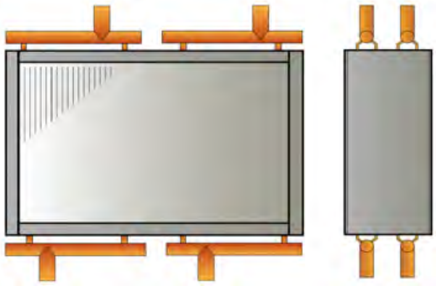
- Maximum finned height/header length : 1500 mm
- Maximum rows per header : 2 rows

These restrictions will automatically require the coil to be split into discreet sections, which favourably influences the discreet section PED categorisation.

2 section : 2 inlet/outlet



As an example, a 4 row deep vertical steam coil with a finned height of 2500 mm, thus header length > 1500 mm, will result in 4 sections. At least 2 sections are required with a header length of 1250 mm and 2 sections in the depth because only 2 rows can be fed by a single header. Thus a total of 4 sections.



In the case of an odd number of rows e.g. 5 rows, then if the header length is <1500 mm the coil would have 3 inlet headers and 3 outlet headers … one header to serve rows 1 & 2, a second header to serve rows 3 & 4 and a third header serving row 5.

This arrangement of multiple headers/connections is often considered a ‘plumbers’ nightmare’ and may affect whether the Client wants to accept our solution, but our experience with especially steam pressures above 3 barg, have prompted us to invoke the header length and rows per header restrictions.

The reason for imposing a maximum of 2 rows feeding each header stems from ensuring that the header legs experience similar thermal related stresses.

A one row coil will use identical length straight legs and a 2 row, two similar length bent legs, whilst a 3 row would need to accommodate two bent and one straight leg. In such a case, for a given temperature rise, the straight leg would linearly expand slightly less than the slightly longer bent legs, thus imposing uneven stresses upon the tube-to-tube but more importantly, the tube-to-header welded joints. Over time and subject to the typical ON/OFF cyclic mode of operation of steam coils, fatigue can become an issue, resulting in stress fractures and/or pin hole leaks.

SUMMARY

Although a single phase fluid or DX/condenser coil may not be fitted with any headers when it has either …

- A single (one) circuit *e.g. has 1 (one) section*
- Number of Circuits / Number of Sections = 1 *e.g. has 4 circuits and 4 sections*

In the case of vertical tube steam coils the logic for determining the number of discreet sections the coil needs to be split into is as follows …

Let, X = INT(Rows / 2 – 0.01) +1

Y = INT(Fin Height / 1500 -0.01) + 1

Number of Sections = X * Y

usually, Number of Sets of Connections = Number of Sections … as standard

As an example, consider a steam coil with a Fin Height = 3200 mm … *the actual horizontal width of the coil, which dictates the header length* … and has 5 Rows.

Thus, X = 3 and Y = 3 and hence Number of Sections = 9 resulting in 9 x inlet and 9 x outlet connections, which would typically have weld-neck flanged connections.

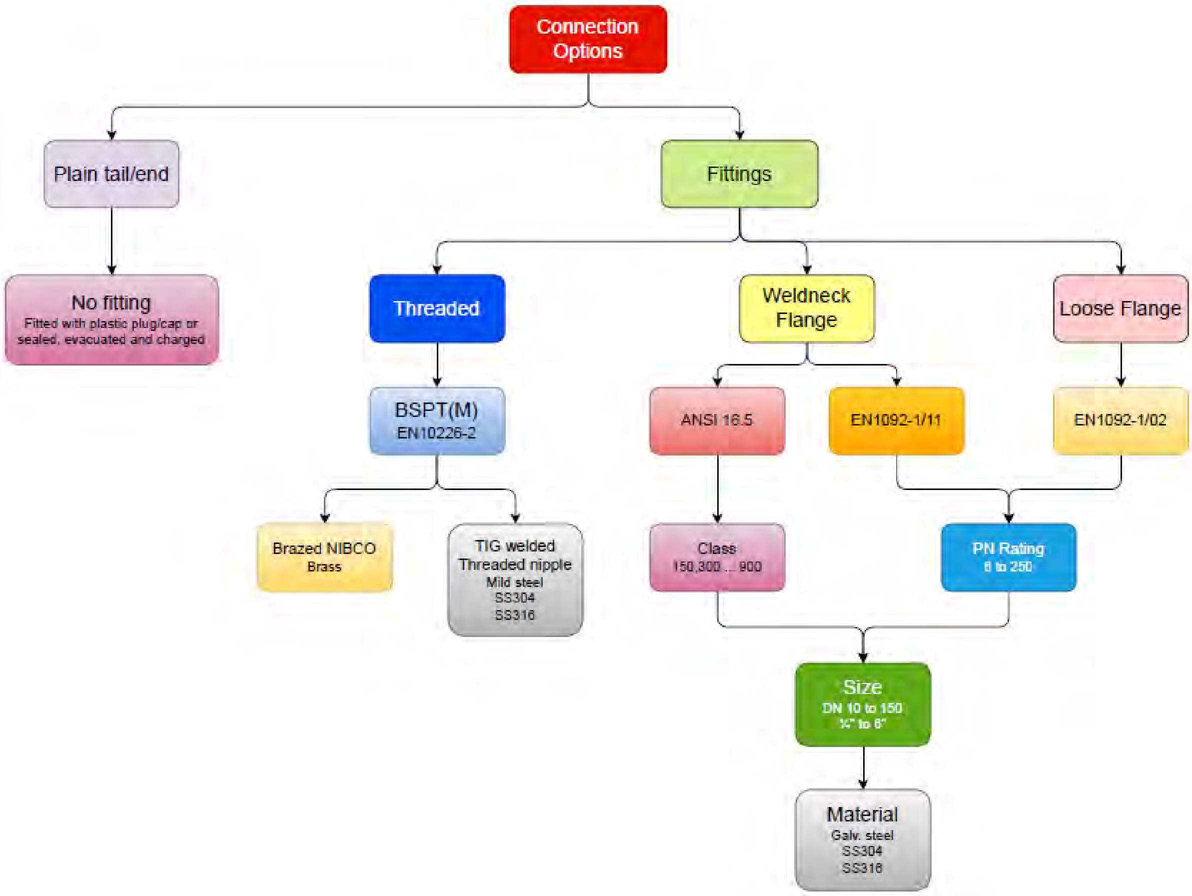
In such a case, and if the coil construction followed a similar layout as shown above for the 4 section steam coil, then if weld-neck flanged connections were required, then the vertical connection tubes accommodating the flanges would need to be offset accordingly to avoid any clashing implications.

Similarly, if the dry saturated steam was to enter through the end of the top inlet headers and perhaps exit in a similar manner, there may be reason to manufacture the fin block in sections of a maximum of 2 row coil blocks and space them apart from one another within the casework, again to avoid clashing of the weld-neck flanges fitted to the end of the headers.

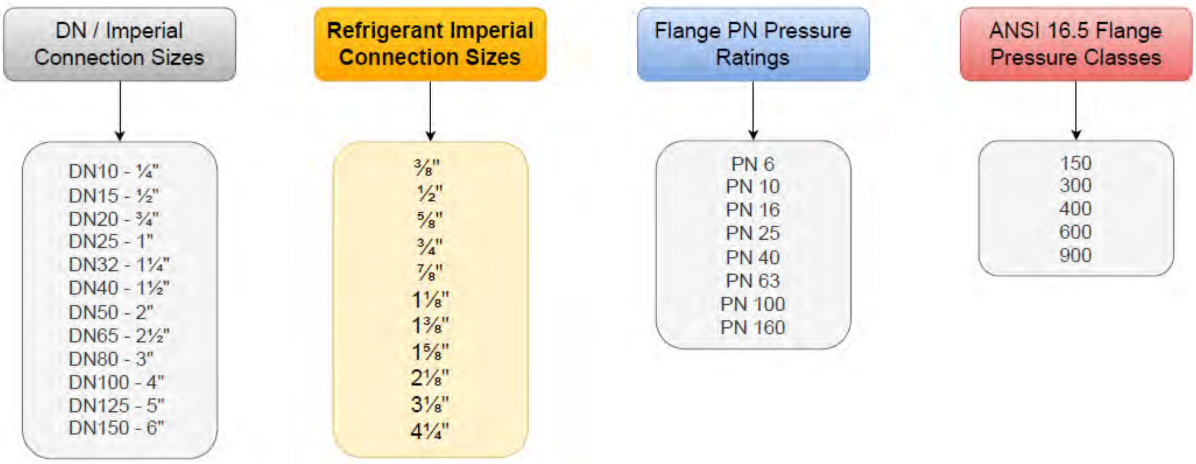
Furthermore, the Customer might specify the number of sections the coil must be split into, but the above calculation dictates the ‘minimum’ number of sections.

Remember, in a similar fashion to fluid, DX & condenser coils, the number of sets of connections per header can exceed one if the application calls for this or there is a benefit in reducing the header pipe size for space saving reasons.

Connection Option Flowchart



Connection Sizes & Pressure Ratings



DISCUSSION OF HEAT REJECTION SYSTEMS

EVAPCO Inc. manufactures a broad range of heat rejection equipment comprising …

- Open-circuit cooling towers
- Closed-circuit coolers and condensers
- Hybrid wet/dry closed-circuit coolers and condensers
- Adiabatic dry air fluid coolers / condensers
- Dry air liquid coolers / air cooled condensers

These constitute a **full spectrum** of products ranging from fully 'wet' to fully 'dry' equipment and the following describes some of the key points of each system in relation to process fluid cooling and indeed, refrigerant condensing applications.

Whilst the majority of current water-cooled applications utilise open-circuit cooling towers due to their low cost, small footprint and low fan power, certain design situations justify the consideration of these other technologies.

Site-specific design situations may be driven by water cost and availability, climate - including concern over winter operation, system size, maintenance and availability of natural cooling sources.

CLOSED-CIRCUIT COOLING TOWERS

Closed-circuit cooling towers, often referred to as 'fluid coolers' operate on the same basic principle of evaporative cooling as open-circuit cooling towers. The exception is that the process fluid to be cooled is kept in a clean, closed loop isolated from the open loop spray water.

The heat load to be rejected is transferred from the process fluid (the fluid being cooled) to the spray water through a heat exchanger, typically a tubular coil, and then from the spray water to the ambient air. The coil serves to isolate the process fluid from the spray water and outside air, keeping it clean and contaminate-free in a closed loop.

This creates two separate fluid circuits …

- an external circuit, in which spray water circulates over the coil and mixes with the outside air
- an internal circuit, in which the process fluid circulates inside the coil

A closed loop system protects the quality of the process fluid, significantly reduces fouling, lowers system maintenance and provides operational flexibility at a cost comparable to an open-circuit cooling tower/heat exchanger combination.

A closed condenser water loop, like the closed evaporator loop, easily accommodates variable speed pumping, conserving energy with virtually unlimited turndown capability. Unlike an open-circuit cooling tower, closed loop units can also be located below the heat source, allowing greater layout flexibility.

Maintaining the open loop spray water system of the closed-circuit design is very similar to that of an open circuit cooling tower, except that there is typically a much lower volume of water to treat. Higher cycles of concentration, resulting in lower blowdown rates, can usually be tolerated on closed-circuit designs with more corrosion resistant materials of construction, such as hot dip galvanized steel coils and stainless steel cold-water basins, as opposed to open loop systems which must consider the metallurgy of the entire cooling loop.

There are two main types of closed-circuit cooling towers …

- coil only
- coil/fill towers *'Fill' referring to the high surface area media to promote better adiabatic cooling*

The coil only cooling towers are counterflow in nature with air moved by axial or centrifugal fans in a forced draft or induced draft manner. Coil/fill versions utilise cooling tower fill to significantly boost the performance of the closed loop coil. This is accomplished by using the fill to pre-cool the spray water before it flows over the coil surface, thereby increasing the unit capacity.

Additionally, because the majority of the evaporation occurs in the fill section, the coil surface tends to stay cleaner and thus performs at a higher level over time. The lower spray water temperature compared to conventional closed-circuit cooling towers also tends to reduce scaling.

Closed-circuit cooling towers are typically equipped with an integral spray pump to recirculate the open circuit water over the heat transfer surfaces. As the unit fan is typically operated on a variable speed drive (VSD), these fixed speed pumps are often the predominate energy draw on these units.

Note that the spray flow over the coils should not be reduced to avoid dry areas on the coils, which can lead to scaling and corrosion. Dual spray pumps are available for smaller installations to provide additional redundancy.

Closed-circuit cooling towers are typically chosen over open-circuit towers for data centres for three primary reasons.

- The reliability and lower maintenance of the clean, closed loop is an operational advantage. If an application must produce high capacities throughout the year, maintaining a clean, reliable system loop can be critical as there is little downtime available for maintenance. By reducing fouling, the system also operates more efficiently, offering attractive economic advantages in terms of energy savings coupled with lower maintenance costs throughout the year.
- Closed-circuit cooling towers can operate in an economizer mode during the winter without the need for an intermediate heat exchanger. This saves the maintenance of having to break down and clean fouled plate and frame heat exchangers. For fluid coolers, any scaling that occurs on the outside surface of the tubes can be handled through the water treatment program and the inside of the tubes only see the clean closed-loop fluid.
- Closed-circuit cooling towers have the ability to run dry in colder temperatures. This can be advantageous for sites that are concerned with water cost and availability, plume in colder climates or the potential for icing of the cooling tower in very cold weather.

Units can also be selected to operate dry in the event of an unexpected cut-off of the facility’s water supply, potentially reducing the need for a back-up water supply. Typically, traditional coil-only units are used for this application because they have a larger sensible heat exchanger surface than coil/fill designs. Note that coil type closed-circuit cooling towers typically have no airflow over the coil and as such have no dry capacity. Remote sumps are ideal for this application as they allow the units to switch between wet and dry operation without the need to drain and refill the cold-water basins.

Higher dry operation switch points enable the closed-circuit cooling towers to have more useful hours of economisation in the dry mode. It should be noted, however, that if the goal is increased economiser hours, staying in wet mode will result in more economiser hours than switching to dry mode. This is a classic example of the trade-off between low energy consumption, which requires more water, and low water use, which increases energy use due to higher approach temperatures and lower economiser switch points. Economiser duty in dry operation is reserved for relatively cold climates and is not practical for moderate climates. For instance, a condensing duty cooling water from 35°C to 29.4°C may have an economical dry operation switch point 4.4°C using a typical coil only unit. In contrast, an economising duty of 12.8°C to 7.2°C would likely require a selection with a dry switch point below –17.8°C. Higher data centre design operating temperatures would help extend the application range of this equipment.

The coils in a closed-circuit cooling tower can also be finned to increase the hours of dry operation by lowering the switch point at which the condensing and/or the economiser load can be handled in the dry mode. The minimum ambient dry switch point for closed-circuit cooling towers is 0°C in order to avoid wet operation in sub-freezing weather. However, units selected with switch points in the range of 7.2°C to 12.8°C are preferred in order to minimise having to switch from dry to wet operation and back again in the shoulder seasons.

In cold climates, there is a risk of coil freezing when the unit is idle. In data centre applications, heat loads are high and constant, reducing (but certainly not eliminating) this threat. If coil freezing is a concern, the use of positive closure damper hoods is recommended to hold the heat in the coil during shutdowns. In addition, a minimum flow should be maintained through the coil along with a small auxiliary heat load to maintain a fluid temperature of approximately 7.2°C leaving the coil anytime the ambient temperature falls below 0°C. An emergency dump system can also be considered in the case of a power failure; however, this will introduce a potential failure point in the system.

Alternatively, the use of an aqueous glycol solution can be considered. Basin heaters can be used to prevent the cold-water basins from freezing during idle periods. Indoor remote sumps, as mentioned earlier, can also be considered.

HYBRID WET/ DRY COOLING TOWERS

Hybrid designs are available that combine both dry and wet cooling to reduce water use, some as much as 70% or more compared to conventional wet units on an annual basis. The use of these designs can be advantageous in areas where water is expensive or in short supply. Hybrid units offer the low process temperatures of conventional open-circuit cooling towers but at a considerable savings in water use. These units can be selected for relatively high switch points between wet and dry operation. Typically, a dry finned coil section is combined in series with a prime surface coil and an evaporative section. The dry finned section handles as much of the load as possible, with the unit able to operate completely dry at a reduced ambient.

One such hybrid wet/dry cooling tower can operate in three modes: combined wet/dry, adiabatic only and dry mode. During wet/dry operation, both evaporative and sensible cooling is utilised. In wet/dry mode, water would flow to the dry finned coil and then to the prime surface evaporative coil. Spray water is drawn from the cold-water basin and pumped above the prime surface coil. Water flowing through the wet prime surface coil is evaporatively cooled by the sprayed water. The spray water falls through the wetted media, enhancing the evaporative heat transfer by further cooling the spray water. Air is drawn through both the prime surface coil and the wetted media where it is saturated and picks up heat. As a consequence of the adiabatic cooling, the air is still cold enough to achieve significant cooling within the finned coil located near the discharge of the unit. The use of sensible cooling provides water savings over typical open or closed-circuit cooling towers.

During adiabatic mode, the evaporative prime surface coil is completely bypassed. No heat is rejected from this coil and the recirculating spray water merely serves to saturate and adiabatically pre-cool the incoming outside air. This scenario consumes much less water than the combined wet/dry operation mode.

Finally, in dry mode, the spray water system is turned off. The fluid to be cooled is fed through both the dry finned coil and prime surface coil where it is sensibly cooled. This scenario consumes no water while offering a low dry switch point.

The principal drawbacks of these units are a high first cost due to the combination of wet and dry cooling sections along with a relatively high fan energy requirement as compared to open circuit cooling towers. The specific advantages on a given site must justify their use. As these water saving designs are typically closed-circuit, the reduced maintenance and higher overall system performance of closed loop operation can be a plus. Keep in mind that glycol is usually required in the loop in freezing climates, especially given the large amount of finned surface that is utilised.

Finally, hybrid designs have the added benefit of reducing or even eliminating the visible, fog-like exhaust air discharge, also known as plume, from the tower under most operating conditions which can be an advantage on some sites.

DRY COOLERS

Dry coolers are often used on smaller data centre facilities (~350 kW and below) to provide closed loop, sensible heat rejection for water cooled CRAC units. Although not offering the lowest system energy use, they do provide a convenient way to reject the heat outside the building. They require little regular maintenance, making them preferred by small data centre facilities without a regular maintenance staff. Dry coolers can provide economisation when the CRAC units are equipped with a separate economizer coil and the internal piping, control valves, and integrated control sequencing for the economizer mode.

A dry cooler consists of a finned coil, typically with copper tubes and aluminium fins, fan motors, a mounting frame, and support legs. The heat is transferred from the process fluid to the cooling airstream in a sensible manner through the finned tube.

As stated earlier, dry coolers are primarily used for heat rejection from water-cooled CRAC units or as direct refrigerant condensers on relatively small installations. Unless there is absolutely no chance of sub-freezing weather, dry coolers should be protected with a suitable aqueous glycol solution for freeze protection. Another type of dry cooler employs a thermosyphon refrigerant loop between a freeze protected shell and tube evaporator and an air cooled condenser. While a higher first cost than a typical dry cooler, this design allows water to be safely used as the cooling fluid in sub-freezing weather conditions.

Individual dry cooler modules with multiple fan motors can be controlled with a single VSD or have EC fans with a common speed controller. In addition, all dry coolers with VSDs on a given load can be controlled in unison to simplify control while minimizing energy usage.

ADIABATIC FLUID COOLERS

Adiabatic fluid coolers are essentially dry coolers with a wetted precooling section that adiabatically lowers the air temperature entering the finned coil section. These units can be used virtually anywhere that dry coolers are being considered. Whereas the high density finned coil bundles are subject to fouling on standard dry coolers, this risk is drastically reduced on adiabatic coolers due to the additional air “filtration” effect of the pre-cooler section. As a result, the main maintenance point becomes the wetted pads and the water distribution header, which can typically be serviced from outside the unit.

As with regular dry coolers, an aqueous glycol solution of the appropriate concentration must be used for the process fluid in sub-freezing climates.

Adiabatic fluid coolers work in the following basic manner: evaporative cooling pads are wetted with water, which is evenly distributed over the top of the evaporative pads. As the air passes through the pads, a small portion of the water is evaporated into the airstream and the air is both humidified and adiabatically cooled to temperatures that are 1 – 3K above the entering wet-bulb temperature. Such substantial depression of the dry bulb temperature of the air results in either a major increase in dry cooling capacity or lower process fluid temperature. The evaporative water is allowed to drain back to a small sump to be reused a few times before being drained to the sewer. Another design drains the water directly from the unit to the sewer on a ‘once through’ basis. While no recirculating pump is required, water use increases with this type of unit, however legionella related issues can be eliminated.

The evaporative cooling pads need to be replaced periodically as they collect dirt and scale. As such, spare pads should be readily available. Controls are typically included with the unit to place it in either dry or adiabatic mode, stage the adiabatic cooling (i.e., run water over one side or both sides), and regulate the fan speed to maintain the desired leaving process fluid or condensing temperature.

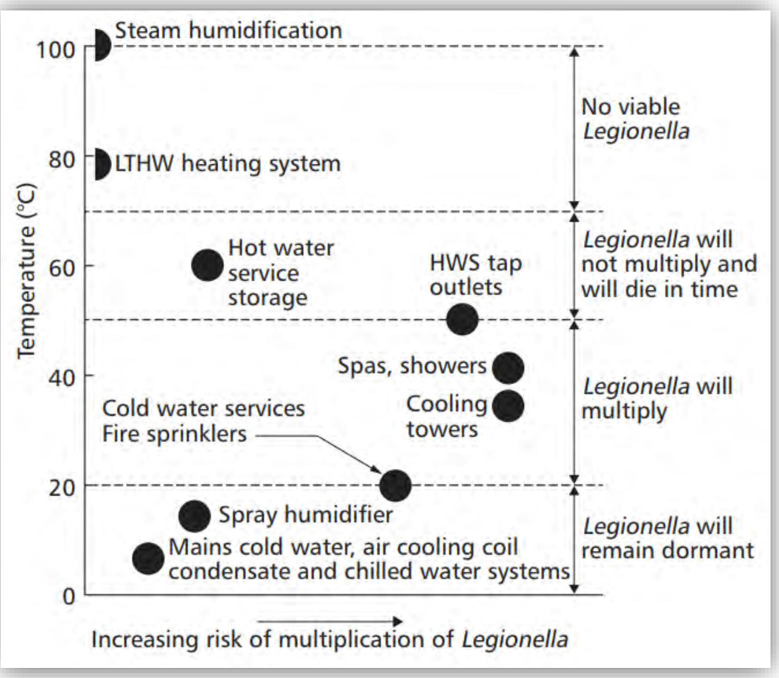
Adiabatic units have high dry operation switch points, typically above 18°C and thus operate dry the majority of the year. Further control enhancements are available that can modulate the switchover point based on weather or load to optimise energy or water savings, depending on the preference of the site. Note that these units cannot operate wet below 0°C wet-bulb or the water on the pads will freeze, blocking airflow. As such, they must operate dry in sub-freezing temperatures, which must be programmed into their control sequence. During sustained cold weather operation, the evaporative water supply can be shut off, the lines and sump drained, and the pads removed to increase the airflow rate through the unit and thus improve the economiser performance.

LEGIONELLA AND COOLING TOWERS

Legionella is a naturally occurring virus first identified following a large outbreak of pneumonia amongst elderly servicemen who attended an American Legion Convention in Philadelphia in 1976. It is normally contracted by inhaling legionella bacteria, either in tiny droplets of water (aerosols), or in droplet nuclei (the particles left after the water has evaporated) contaminated with legionella.

Governing legislation includes CIBSE TM13 & ASHRAE Standard 188p.

Successful legionella control is as much about water temperature management as it is the technical requirements for preventing the disease multiplying in different environments. The diagram below indicates the relationship …



Other forms of control include Ultraviolet Light (UV) but these types od system will tend to reduce Legionella in the area local to the treatment and if not continually applied, they can be less effective.

There are two categories of cooling towers, Direct Contact and Indirect Contact, where closed circuit evaporative coolers and condensers, fall into the latter category. Nevertheless, both necessitate a good water treatment programme to address the threat of corrosion, scaling, microbiological and fouling within the cooling tower.

To meet the stringent bacteriological control requirements of Health & Safety legislation, most systems now use an oxidising biocide such as bromide or chlorine as the primary biocide with a non-oxidising type as a secondary or back-up biocide.

Note : hand dosing of chemicals or suspended basket chlorine tablets are unlikely to give the desired control and will not normally satisfy an HSE Inspector.

Legionella legislation usually states that hot water systems present the greatest risk in environments which support the proliferation of Legionella, for example …

- At the base of the calorifiers where the incoming cold water merges with the existing hot water. This water collects sedimented organic and mineral deposits which support bacterial growth, including Legionella – this can then be distributed throughout the system to colonise its periphery, especially where optimum temperature and stagnation occur e.g. in infrequently used outlets
- Water held in pipes between a recirculating hot water supply and an outlet e.g. tap or shower, particularly when not in use, as they may not be exposed to biocides and high temperatures

System design also plays a major role in the prevention of Legionella. Storage capacity and recovery rate of the calorifier should be selected to meet the normal daily fluctuations in hot water use without any drop in supply temperature. Where more than one calorifier is used, they should be connected in parallel. If temperature is used as a means of control each calorifier should deliver water at a temperature of least 60°C.

If temperature is used as a means of controlling Legionella, the hot water circulating loop should be designed to provide a return temperature to the calorifier of 50°C or above. The pipe branches to the individual hot taps should be of sufficient size to enable the water in each of the hot taps to reach 50°C within 1 minute of turning on the tap.

USEFUL DESIGN GUIDELINES FOR DUMMIES

Often insufficient information is provided to perform a coil calculation and thus certain assumptions are needed to start the process. These 'rules of thumb' are not hard and fast but general indicators to enable a selection to be performed and a solution offered.

GENERAL 'RULES OF THUMB' - ALL COILS

- Fin block support divider plates located at 1100 mm intervals
- Include for intermediate drain pans at 1200 mm height intervals for latent cooling applications
    - It is difficult to 'clear away' (avoid water agglomeration) with high latent cooling generating large amounts of condensate. Therefore a large, tall coil may generate excessive water carry-over if the condensate from the upper portion of the coil cannot be adequately drained away. Hence the need for intermediate drain trays, each handling a portion of the coil
- Avoid approach temperatures differences < 2K
- Design for cross-counter flow circuitry pattern
- Circuitry and connection positions :
    - EAS standard …
        - For single phase fluid coils e.g. water …
            - Circuitry pattern should avoid any water traps and be 'free draining' i.e. is an up-feed pattern from the inlet of the circuit to the outlet. Conversely, the circuit will have a tendency to progress backwards and downhill from the upper outlet of the circuit to the lower inlet of the circuit, which is connected to the inlet header
            - Inlet connection at bottom of the inlet header
            - Outlet connection at top of the outlet header
            - For vertical coils with horizontal air flow, handing A1 or A2
    - Evapco Inc. standard …
        - For single phase fluid coils e.g. water …
            - Circuitry pattern should avoid any water traps and be 'free draining' i.e. is a down-feed pattern from inlet of the circuit to the outlet. Thus, there is a tendency to progress forwards and downhill from the upper inlet of the circuit to the lower outlet of the circuit, which is connected to the outlet header
            - Inlet connection at top of the inlet header
            - Outlet connection at the bottom of outlet header
            - For vertical coils with horizontal air flow, handing A5 or A6
            - Note : This philosophy/pattern works for condenser applications too
    - Condensers :
        - Down feed circuitry
        - Inlet at top or centre of header
        - Outlet at bottom – ensure no condensate trapping
    - DX Evaporators :
        - Up feed circuitry – ensure refrigerant liquid does not free-flow to suction header
        - Consider down feed only for evaporating temperatures <-20°C to ensure oil return – depends upon refrigerant. However, if the circuit loading is reasonably high, suggesting a decent refrigerant velocity, then retain the preferred up feed circuitry pattern
        - Suction header should always be placed on the air on side of the coil. Furthermore, the last few passes of the circuit should be focused upon the air inlet side of the pattern and should ideally progress upwards. Endeavour to design each circuit to have a similar pattern
        - Try to avoid circuit 'shadowing'
        - Suction header outlet should be as low as possible to avoid oil and any liquid retention related issues

GENERAL DATA

- Thermal conductivity of materials …
  - Aluminium : 221.7 W/m/K
  - Copper : 385 W/m/K
  - SST 304/316 : 16.2 W/m/K
  - Titanium : 20 W/m/K
- Air density …
  - European @ 15°C : 1.225 kg/m^3
  - American @ 21°C (70°F) : 1.2 kg/m^3
  - Normal @ 0°C : 1.29 kg/m^3
- Air specific heat (dry) …
  - @ 20°C : 1.006 kJ/kg/K
  - @100°C : 1.011 kJ/kg/K
- Air thermal conductivity …
  - @ 20°C : 0.02587 W/m/K
  - @100°C : 0.03162 W/m/K
- Air dynamic viscosity …
  - @ 20°C : 0.0182 mPa.s
  - @100°C : 0.0219 mPa.s
- Water density : 1000 kg/m^3
- Water specific heat …
  - 5-10°C : 4.2 kJ/kg/K
  - >50°C : 4.18 kJ/kg/K

HEATING COILS – HOT WATER & STEAM

- Fin pitch : >= 1.6 mm
- Air velocity :
  - Duct mounted coils : 3 - 4 m/s
  - AHU coils : 2.5 - 3.5 m/s
  - Dry coolers : 2.5 - 3.5 m/s
- Fluid velocity limits :
  - Copper & Aluminium tubes
    - Water <= 60°C : <= 2.0 m/s
    - Water > 60°C : 1.0 - 1.5 m/s
    - High viscosity fluid (oils) : > 2.5 m/s
    - Gases : < 15 - 20 m/s
  - Stainless tubes
    - Water : < 5.0 m/s
    - Gases : < 15 - 30 m/s
  - Titanium tubes
    - Sea water : > 2.5 m/s
- Guideline flow rate per circuit :
  - Ø3/8” tubes : 0.09 l/sec per circuit
  - Ø15 mm tubes : 0.25 l/sec per circuit
- Fluid pressure drop limits :
  - Small coils : 15 - 30 kPa
  - Large coils : 40 - 60 kPa
  - Dry coolers
    - 80 - 100 kPa
    - Sweden : 25 - 40 kPa
- Connections size :
  - BSP (inches) = sqrt[1.32 x l/sec] … for 1.5 m/s

- BSP (inches) = sqrt[l/sec] … for 2.0 m/s
- DN (mm) = sqrt[625 x l/sec]

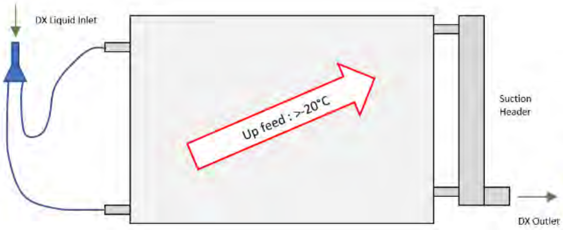
- Steam coils :
  - Only consider dry saturated steam inlet quality
  - Preferably <= 3 barg but 7 barg as an upper limit
  - Vertical tube (single pass) orientation
  - Inlet connection velocity : <= 30 m/s
  - Outlet connection : Often the same size as the inlet or more usually, one size smaller than inlet
  - Be aware that low pressure steam has a high specific volume and thus requires somewhat larger headers and connections sizes

COOLING COILS – CHILLED WATER

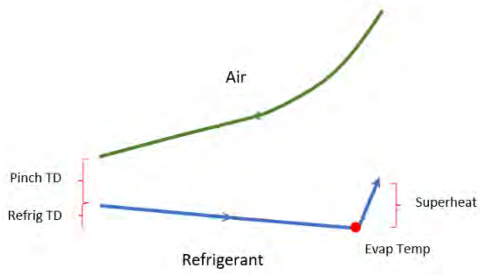
- Fin pitch :
  - Sensible/dry cooling : >= 2.0 mm
  - Latent/wet cooling : >= 2.5 mm
- Air velocity :
  - Sensible cooling coils : < 3.5 m/s
  - Latent cooling coils (to avoid water carry-over) : < 2.5 m/s
- Fluid velocity limits :
  - Copper & Aluminium tubes
    - Water : <= 2.0 m/s
    - Gases : < 15-20 m/s
  - Stainless tubes
    - Water : < 5.0 m/s
    - Gases : < 15 - 30 m/s
  - Titanium tubes
    - Sea water : > 2.5 m/s
- Fluid pressure drop limits:
  - Small coils : 15 - 30 kPa
  - Large coils : 40 - 60 kPa
- Connections size :
  - BSP (inches) = sqrt[1.32 x l/sec] … for 1.5 m/s
  - BSP (inches) = sqrt[l/sec] … for 2.0 m/s
  - DN (mm) = sqrt[625 x l/sec]
- Single phase fluid : Aim for a free draining pattern
- Drain pan connection size :
  - BSP (inches) = 1.85 x $[l/sec]^{0.37}$

EVAPORATORS

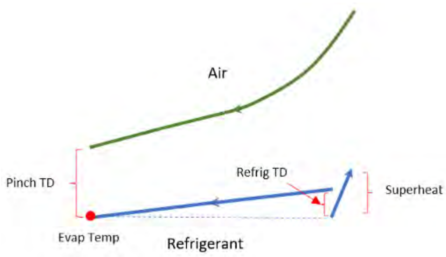
- DX coils:
  - Above -20°C evap. temp. : Up-feed in counter flow
  - Below -20°C evap. temp. : down-feed in counter flow to assist oil return to compressor
  - Refrigerant temperature drop resulting from pressure drop should be minimised



    - Azeotropic refrigerants : aim for 1- 2 K
    - Zeotropic refrigerant temperature glide can be used to advantage
  - Circuit loading : Coil size and refrigerant dependant
    - Ø15 mm tube : 0.5 - 7 kW
    - Ø9.5 mm tube : 0.2 - 2.5 kW
  - Circuit pressure drop : For low evaporating temperature applications using Azeotropic refrigerants, often operating at small TDs, the exhibited pressure drop and thus associated refrigerant temperature drop, will detrimentally impact upon the LMTD due to the higher refrigerant inlet temperature compared with the design evaporating temperature … *the refrigerant falls in temperature because of the pressure drop in the circuit*.



    A trick to use this temperature drop as a coil design benefit is to arrange the circuit pattern to flow in 'parallel' with the air direction for perhaps 2/3rd of the coil depth and then 'loop back' to the air inlet face of the coil to focus the last few passes of the circuit to take advantage of the widest TD available in this area of the coil to provide the superheat necessary. Now that the fall in refrigerant temperature is in sympathy with the falling air temperature, the 'pinch TD' has widened, improving the LMTD.



  - Superheat : 3 – 5K required to control of the TEV … often 5 – 7K required for real-life stable control, albeit that modern electronic expansion devices tend to control with a smaller superheat that the 'old style' TEVs
  - Distributor body
    - Aim for 0.5 to 1.0 bar pressure drop
    - There are typically 2 types of distributor design …
      - Venturi : Danfoss, Alco etc.
      - Removable nozzle : Sporlan
        - Provides an option to, on site, 'fine tune' the liquid distribution behaviour
        - This type of distributor design also allows for a side inlet/side exit option for providing either …
          - Hot gas injection downstream of the nozzle to provide capacity control
          - Reverse cycle heat pump applications where the DX coil operates as a condenser and where hot gas from the compressor is fed to the DX suction header, condenses within the circuit and the liquid is passed backwards through the distributor leads into the distributor body and exits through the side connection before

experiencing the pressure drop that would be imposed had it had to pass through the nozzle orifice

- Distributor leads (Capillaries)
  - Aim for around 1.0 bar pressure drop across the leads
  - Minimum length : 300 mm
  - Maximum length : 2000 mm … *the shorter the better*
    - If necessary, split the coil into multiple sections each fed via its own distributor (shorter leads) and either collect the evaporated refrigerant in a single suction header or individual suction headers
  - Nominal length : 2/3rd refrigerant section height + 70 mm
  - Allow additional 75 mm for 'reverse cycle' coils where the leads pass through the header and into the tubes
  - Correctly sized leads can continue to behave in a stabile fashion over a duty or mass flow range of +/-25% of the design condition
  - Lead diameter : If in doubt, undersize … *creating a larger than expected pressure drop in the leads may improve distribution, especially if the section height (static head) is large and thus the leads are long. Too much static head can create unequal liquid distribution, which in turn, will cause unstable behaviour and under performance. So, deliberately creating a high pressure drop in the leads will lessen the propensity for the liquid portion of the liquid/flash gas refrigerant mixture to flow to the lowest portion of the coil and cause unstable behaviour*
- Connection size
  - Liquid inlet : 1.5 m/s (pressure drop ~0.5 – 1.0K to minimise liquid hammer issues)
  - Suction gas outlet : 4.5 – 20 m/s (affected by evaporating temperature to aim for 0.02 K/m pressure drop)
  - Suction header size : If in doubt, oversize
- Hot gas defrost
  - Size hot gas header for 5 – 10 m/s
  - Distributor leads usually pass through this header directly into the tubes of the circuit by around 75 mm

- Flooded / Pump circulated
  - Pump rate : 1.5 to 5
  - Up feed circuity vertically
  - Number of circuits is a factor of the number of rows deep
    - ½ rows deep
    - 1 x rows deep
    - 2 x rows deep
    - etc.
  - Liquid tube velocity
    - Ammonia: ~0.1 m/s
    - CO_2, HCFC & HFCs : <0.75 m/s

CONDENSER COILS

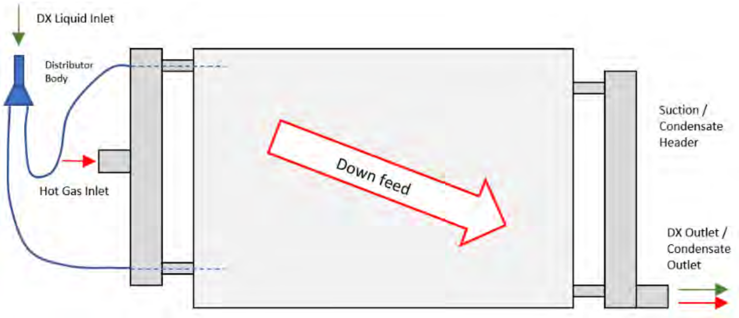
- Down-feed in counterflow
- Refrigerant temperature drop resulting from pressure drop should be minimised
  - Azeotropic refrigerants : aim for <2K
  - Zeotropic refrigerant temperature glide adds to the pressure drop related temperature drop and thus can negatively affect the sub-cooling achievable and reduce the thermal performance
- Circuit loading : Coil size and refrigerant dependant
  - Ø15 mm tube : 3 - 15 kW
  - Ø9.5 mm tube : 0.5 - 5 kW
- Hot gas inlet gas velocity : 10 -18 m/s (Pressure drop ~0.02 K/m)
- Condensate outlet velocity : 0.5 m/s
- Ammonia typical allowable pressure drop : 10 - 20 kPa (0.1 - 0.2 bar)
- If in doubt, oversize the Hot Gas inlet header

- Traditionally, condensate liquid outlet is one size smaller than the hot gas inlet if the liquid velocity is not used to determine the diameter
    - *Evapco Inc. tends to make the hot gas inlet and condensate outlet the same size*

DUAL APPLICATION COILS

EVAPORATOR/CONDENSER – HEAT PUMP OR HOT GAS DEFROST – SAME FLOW DIRECTION

This arrangement is offered to allow the injection of hot gas from the compressor into an additional header which is also connected to the inlet of each circuit. This design necessitates the distributor leads passing through the hot gas header to protrude some 75 mm into the circuit.



This concept allows for capacity control via hot gas injection or more normally, passing hot gas from the compressor directly into the DX circuitry during a defrost cycle to clear the coil from frost and ice.

Furthermore, this design allows a coil primarily designed for operation as an evaporator to 'dual purpose' as a condenser coil for a heat pump application.

Contrary to the preferred 'up-feed' circuitry pattern that EAS promote for applications above -20°C, this variant needs to employ a 'down-feed' approach to ensure that refrigerant liquid does not run backwards into the hot gas header, this also assists the condensed liquid refrigerant to drain towards the dual purpose suction/condensate outlet header/connection.

In this mode; as with the case when a defrost cycle is activated; the hot gas enters the 'hot gas header' connection and passes through the coil in the same direction as the liquid refrigerant when the coil is operating as a DX coil.

Furthermore, in both DX and CD modes of operation the refrigerant and air side flow direction are optimal i.e. counter flow.

HOT GAS INJECTION – CAPACITY CONTROL

Capacity control can be achieved using hot gas injection where a portion of the hot gas from the compressor is injected into and mixed with the cold liquid refrigerant leaving the TEV and entering the distributor.

Although this can be simply achieved by using a tee-piece mounted to the inlet of a simple venturi type distributor, the disadvantage is that both the liquid and now, hot gas, must both pass through the 'neck' of the distributor, which can create a 'chocking' effect and result in unstable behaviour.

By using a specifically designed side inlet nozzle distributor, where the hot gas is injected downstream of the nozzle orifice, then only the liquid portion encounters the high pressure generated by the orifice.

REVERSE CYCLE HEAT PUMP – REVERSE FLOW DIRECTION

Similar in concept to the heat pump and hot gas defrost detailed above, but in this case the refrigerant passes through the circuitry in opposite directions depending upon whether the coil is operating as a DX evaporator or condenser.

The circuitry design is a compromise because although we might wish to up-feed the circuitry for DX operation, this may result in liquid refrigerant running back into the condensate header, accumulating in the lower portion of the coil, reducing the effectiveness of this surface area and affecting the performance.

Down-feeding the DX circuitry may help matters, but when the system reverses, the coil functions as a condenser and the condensed liquid now has to travel up hill to exit the coil, which may cause problems of its own.

Another disadvantage of this concept is that if the circuit is designed to be cross-counter flow for DX operation, it becomes cross-parallel flow when in condenser mode … clearly not ideal !

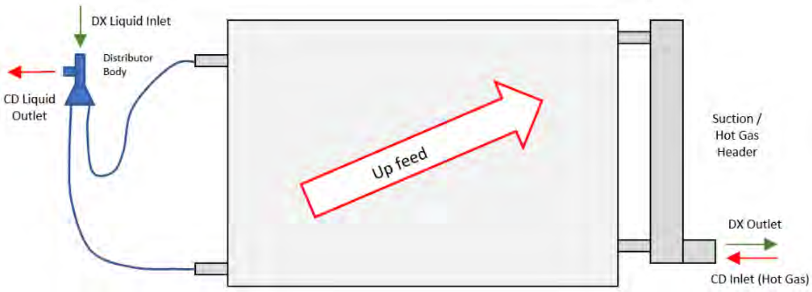
However, if the condenser mode is only a minor portion of the system's operation, then the arrangement may be adequate.

A final drawback of this concept is that switching the direction of the refrigerant flow through a system can dislodge any build-up of debris and may transport it to either the leads or distributor to clog them or back to the compressor and cause a seizure.

REVERSE CYCLE HEAT PUMP – SIDE EXIT DISTRIBUTOR

An alternative, simpler and cheaper reverse cycle coil arrangement is to implement a side exit distributor in place of the CD liquid outlet header.

This time the coil is circuited in the preferred up-feed arrangement to ensure that the refrigerant stays as long as possible within the circuit to fully evaporate and superheat.



When the system reverses and the coil functions as a condenser, the hot gas is fed into the otherwise suction header and now flows downhill through the circuitry towards the distributor leads. It then flows up through the leads to the side exit distributor and avoids the nozzle which would impose an unnecessary pressure drop.

Although this concept has the same drawback as the previous arrangement, i.e. cross-parallel flow in condenser mode, if the primary operation for the coil is as an evaporator, then the 'less than ideal' condenser mode can be handled by ensuring that sufficient surface area is allowed for.

Furthermore, the wider TDs that condensers typically experience along with the reality that the condensing temperature is usually fairly constant, partially mitigates the downside of the parallel flow arrangement and advantageous economic cost savings associated with this 'simple' coils design and manufacture.

PRODUCT SELECTION

When insufficient information is available to perform a calculation or select a product, then certain assumptions must be made to enable an assessment of the product size required to meet the duty expectations.

Examples of sensible assumptions … rules of thumb … are as follows …

- Assume a coil or product face velocity to be 2.5 m/sec to determine an air volume that might be required if the coil size is either given or can be assumed …
    - The air volume is thus …

$\dot{V}$ = FA x FV *where,* $\dot{V}$ *= air volume, m^3/s*

FA = face area, m^2
FV = face velocity, m/s

- Determine the minimum air volume required by assuming an air leaving temperature if the capacity and the air entering temperature is known
  - The air leaving temperature, worst case, could be assumed to be 2K less than the fluid inlet temperature for an air heating application

    AirOutlet = (FluidInlet – 2)

  - Rearranging the basic airside duty and using the known 'worst case' TD in the equation will now provide the minimum air volume requirement. Assume for simplicity that the air density is1.2 kg/m^3 and the specific heat is 1.005 kJ/kg/K …

    $\dot{V} = Q_T / \rho / C_p$ / (AirInlet – AirOutlet)

    where, Q_T *= Total duty, kW*
    ρ *= Air density, 1.2 kg/m^3*
    C_p *= Air specific heat, kJ/kg/K*

  - For a product selection, the calculated minimum air volume will provide an indication of how many fans might be required. This information will provide a clue as to the size of the product and indeed, the number of products that might be necessary to meet the design requirements
- For large scale projects which involve multiple products, a quick assessment of how many products and thus fans that are needed can be conducted by estimating the total air volume to meet the total capacity (kWs), as described above, and then using the following maximum nominal air volumes that either a Ø910 mm EC compact fan or Ø1500 mm industrial AC fan can deliver against a typical coil air pressure drop …
  - Ø910 EC fan 3.2 kW @ 1080 rpm => **9.5** m^3/s
  - Ø1500 Industrial impeller with 5.5 kW 8 pole motor (725 rpm) => **22** m^3/s

  Furthermore, the fan kW indicated will also provide an indication as to whether the Client might accept the product solution suggested, from an electrical perspective

  Note : The power (kW) indicated for an EC fan is P_1, the consumed power, whilst for the industrial option, the AC induction motor shaft power, P_2 is indicated. Therefore, if job site electrical efficiency calculation should be performed, then P_2 should translated into an approximate P_1 value by dividing P_2 by 0.85 (85% efficiency). Thus the consumed power for this 5.5 kW motor would be approximately 6.5 kW
- Again for a large scale project, to determine whether our product type is applicable for the job i.e. that perhaps the tubes are large enough diameter to handle the total fluid flow rate. For single phase fluids such as water and water/glycol mixtures …
  - Ø15 mm tube geometries (A & C fin) : **0.25** l/sec per tube
  - Ø3/8" (Z fin) : **0.09** l/sec per tube

  Thus, dividing the project's total volume flow rate requirement by one of the above figures will indicate the minimum number of circuits required to handle to volume flow rate.

  It is then a simple calculation to assume a suitable product type, Vee or flat-bed with perhaps a 6 row coil requirement to determine the number of total tubes available in one cooler, and then assuming 2 or even 4 tubes per circuit, then the minimum number of dry coolers required can be determined.

  Clearly if the quantity of products is unreasonable, the client can be advised accordingly and advised to seek a manufacturer who can offer a more suitable product variant to fulfil his needs.

APPENDIX 1 - CONVERSION FACTORS

NUMERICAL PREFIXES

Prefix	Symbol		Factor
Yotta	Y	-	10^{24}
Zetta	Z	-	10^{21}
Exa	E	-	10^{18}
Peta	P	-	10^{15}
Tera	T	-	10^{12}
Giga	G	-	10^{9}
Mega	M	-	10^{6}
Kilo	k	-	1000
Hecto	h	-	100
Deca	da	-	10
Deci	d	-	0.1
Centi	c	-	0.01
Milli	m	-	0.001
Micro	µ	-	10^{-6}
Nano	n	-	10^{-9}
Pico	p	-	10^{-12}
Femto	f	-	10^{-15}
Atto	a	-	10^{-18}
Zepto	z	-	10^{-21}
Yocto	y	-	10^{-24}

COMMON UNITS

Category	Units	x	Conversion	=	Units
Length	inch	x	25.4	=	mm
	foot	x	0.3048	=	m
	yard	x	0.9144	=	m
	mile	x	1609.344	=	m
	mile	x	1760	=	yard
	nautical mile	x	1853.836	=	m
Area	in^2	x	645.16	=	mm^2
	ft^2	x	0.09290304	=	m^2
Volume	dm^3	x	1	=	litre
	cm^3	x	1	=	ml
	in^3	x	16387	=	mm^3
	in^3	x	0.016388	=	litre
	ft^3	x	0.02832	=	m^3
	ft^3	x	28.32	=	litre
	pint	x	0.56825	=	litre
	pint	x	0.125	=	imp. gallon
	imp. gallon	x	4.54609	=	litre
	U.S. gallon	x	3.785	=	litre
Mass	metric tonne	x	1000	=	kg
	imp. ton	x	1016	=	kg
	U.S. ton	x	907.185	=	kg
	kg	x	2.205	=	pound
	pound	x	0.45359	=	kg
	pound	x	16	=	ounce
	pound	x	7000	=	grain
	imp. Ton	x	2240	=	pound

Velocity	ft/s	x	0.3048	=	m/s
	ft/min	x	0.00508	=	m/s
	knot	x	0.5148	=	m/s
	mile/hr	x	0.447	=	m/s
	mile/hr	x	1.6093	=	km/hr

Acceleration	g	x	32.1741	=	ft^2/s
	g	x	9.80665	=	m/s^2

Volume Flow	ft^3/min (cfm)	x	0.000471947	=	m^3/s
	m^3/hr	x	0.000278	=	m^3/s
	imp. gpm	x	0.07577	=	l/s
	U.S. gpm	x	0.06309	=	l/s

cfm ≈ m^3/s x 2119
m^3/s = m^3/hr / 3600
gpm (imp) ≈ l/s x 13.2
gpm (US) ≈ l/s x 15.85

Mass Flow	pound/hr	x	0.000126	=	kg/s
	imp. gpm (water)	x	0.07577	=	kg/s

Force	Newton (N)	x	1	=	$kg.m/s^2$
	kg force	x	9.805	=	N
	pound force (lbf)	x	4.4482216	=	N
	poundal	x	0.1383	=	N

Force = Mass * Acceleration
$F = kg * m / s^2 \Rightarrow N$

Pressure	N/m^2	x	1	=	Pa (Pascal)
	kPa	x	1000	=	Pa
	bar	x	100000	=	Pa
	mbar	x	100	=	Pa
	MPa	x	1000000	=	Pa
	kg/cm^2	x	0.9805	=	bar
	Atm	x	101.325	=	kPa
	Atm	x	1.013	=	bar
	Atm	x	14.69595	=	psi
	psi (lb/in^2)	x	6894.757	=	Pa
	psi	x	0.068947	=	bar
	bar	x	14.504	=	psi
	in.H_2O / in.wg (4°C)	x	249.1	=	Pa
	mm.H_2O / mm.wg (4°C)	x	9.803	=	Pa
	m.H_2O / m.wg (4°C)	x	9.803	=	kPa
	psi	x	27.67	=	in.wg
	Torr	x	1	=	mmHg
	Torr	x	133.3	=	Pa

Pressure = Force / Area

$P = N / m^2$
$= kg.m/s^2 / m^2$
$= kg/m/s^2$

Pressure in terms of 'Head of Water' also known as 'Water Gauge (wg)' is expressed in inches, millimeters or meters.

Energy	Btu	x	1055.05585	=	Joule (J)
	Btu	x	0.252	=	kcal
	kcal	x	4187	=	J
	kWh	x	3600	=	kJ

Heat Flow	J/s	x	1	=	Watt (W)
	kW	x	1000	=	W
	Btu/hr	x	0.293071	=	W
	Btu/hr	x	0.252	=	kcal/hr
	kcal/hr	x	1.163	=	W
	HP (work)	x	550	=	ft.lbs
	HP (work)	x	745.7	=	W
	HP (elec.)	x	746	=	W
	ch (metric HP)	x	735.5	=	W
	TR (ton refrigeration)	x	12000	=	Btu/hr
	TR	x	3516	=	W

kW ≈ Btu/hr / 3412

kW ≈ kcal/hr x 860

Btu/hr (Condenser) ≈ TR x 1500

Density, ρ	g/cm^3	x	1000	=	kg/m^3
	kg/litre	x	0.001	=	kg/m^3
	g/litre	x	1	=	kg/m^3
	lb/ft^3	x	16.01846	=	kg/m^3
	lb/imp. gallon	x	9.978	=	kg/m^3

ρ = Mass / Volume

$\rho = kg / m^3$

Specific Heat, Cp	kcal/kg/C	x	4186.8	=	J/kg/K
	Btu/lb/F	x	4186.8	=	J/kg/K

Thermal Conductivity, K	kcal/hr/m/C	x	1.163	=	W/m/K
	Btu/hr/ft/F	x	1.7307	=	W/m/K

Dynamic Viscosity, μ	Ns/m^2	x	1	=	Pa.s	***μ = ν * ρ***
	kg/m/s	x	1	=	Pa.s	***cP = cSt * ρ / 1000***
	mPa.s	x	0.001	=	Pa.s	
	cP (centiPoise)	x	1	=	mPa.s	
	lb/ft/hr	x	0.0004134	=	Pa.s	
	$lbf.s/ft^2$	x	47.88		Pa.s	

Kinematic Viscosity, ν	m^2/s	x	10000	=	Stoke	***ν = μ / ρ***
	cm^2/s	x	1	=	Stoke	***cSt = 1000 * cP / ρ***
	Stoke	x	100	=	cSt (centiStoke)	
	mm^2/s	x	1	=	cSt	
	m^2/s	x	1000000	=	cSt	
	ft^2/s	x	92901	=	cSt	
	in^2/s	x	645.2	=	cSt	
	in^2/s	x	0.0006452	=	m^2/s	
	ft^2/s	x	0.0929	=	m^2/s	

Enthalpy	kcal/kg	x	4187	=	J/kg
	Btu/lb	x	2326	=	J/kg

Heat Transfer Coeff.	$kcal/hr/m^2/C$	x	1.163	=	$W/m^2/K$
	$Btu/hr/ft^2/F$	x	5.67826	=	$W/m^2/K$

Temperature	(°F - 32) / 1.8			=	°C
	°C	+	273.15	=	K
	°F	+	459.67	=	°R

WIND SPEED

	Force	km/h	Name	
Beaufort Scale	0	<2	Calm	
	1	1-5	Light air	
	2	6-11	Light breeze	
	3	12-19	Gentle breeze	
	4	20-29	Moderate breeze	
	5	30-39	Fresh breeze	
	6	40-50	Strong breeze	
	7	51-61	Near gale	
	8	62-74	Gale	
	9	76-87	Strong gale	
	10	88-102	Storm	
	11	103-118	Violent storm	
	12	>118	Hurricane	***Category 5 > 16 km/h***

IMPERIAL & AMERICAN GAUGE SIZES

Comparison of Imperial & American Sheet Metal Gauges

Gauge	Imperial Standard Wire Gauge - SWG		American Gauges					
			Carbon Steel		Galvanised		Stainless Steel	
	inch	mm	inch	mm	inch	mm	inch	mm
6	0.192	4.877	0.194	4.935	-	-	-	-
8	0.160	4.064	0.164	4.176	0.168	4.270	0.172	4.366
10	0.128	3.251	0.135	3.416	0.138	3.510	0.141	3.571
12	0.104	2.642	0.105	2.657	0.108	2.753	0.109	2.779
14	0.080	2.032	0.075	1.897	0.079	1.994	0.078	1.984
16	0.064	1.626	0.060	1.519	0.064	1.613	0.063	1.588
18	0.048	1.219	0.048	1.214	0.052	1.311	0.050	1.270
20	0.036	0.914	0.036	0.912	0.040	1.006	0.038	0.953
22	0.028	0.711	0.030	0.759	0.034	0.853	0.031	0.787

APPENDIX 2 - PED WORKED EXAMPLES

EXAMPLE #1

A hot water heating coil mounted in an Air Handling Unit operating at 50°C and 10 barg is required to withstand a maximum allowable pressure of 20 barg.

The Cu/Al coil is 1920 mm high with full height 4” headers.

Solution

Because the maximum allowable pressure (PS = 20 barg) of the equipment is above 0.5 barg, the PED will apply.

Following the PED classification methodology:

- Determine the type of pressure equipment being considered
- Determine the state of the fluid in the equipment
- Determine the hazard group of the fluid in the equipment
- Select the appropriate hazard category chart
- Determine the maximum allowable pressure and the defining dimension of the equipment
- Determine the PED hazard category

Step 1

The coil is part of an air conditioning system and can thus be considered as **Piping** as defined in Guideline B-04.

Step 2

Water at the maximum allowable temperature (TS = 50°C) has a vapour pressure P_{sat} = -0.877 barg, which is below 0.5 barg and thus constitutes the fluid as being a liquid.

Alternatively, as we are dealing with water and the maximum allowable temperature is < 110°C, this document has earlier concluded that such applications fall under SEP – Article 4/3, so we could stop at this point. But for completeness, we shall continue and follow the rules !!

Step 3

Water is a safe fluid and thus falls into Group 2.

Step 4

The appropriate hazard category charts to use are **Chart 4** when considering the **Vessel** aspects (the headers) of the coil and **Chart 9** when considering the coil as **Piping**.

Step 5

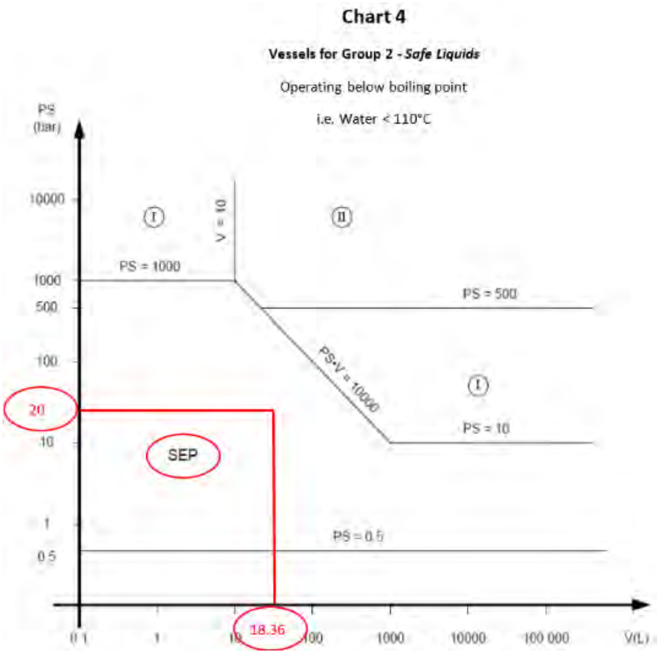
The maximum allowable pressure is defined as PS = 20 barg.

The defining dimensions are …

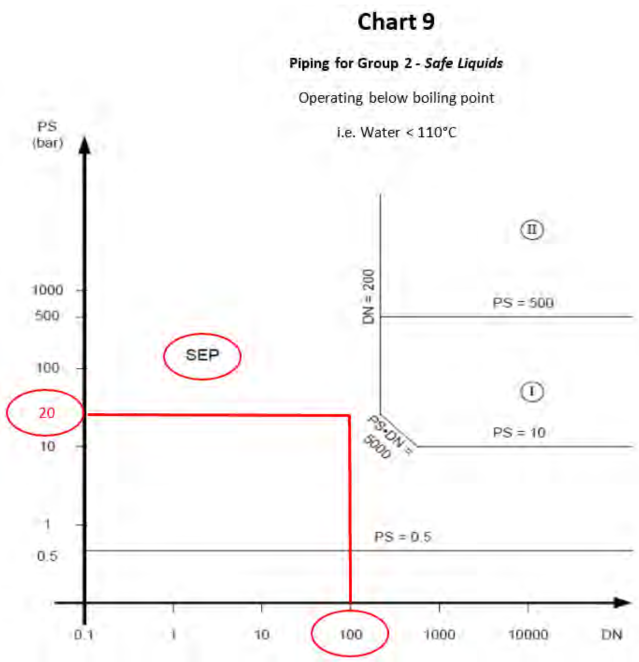
- Considering the **Vessel** aspect of the 4” header, its volume V_H = 9.56 dm^3/m x 1.92 m = 18.36 dm^3 or litres thus Cat^V = PS x V_H = 367.1 barL
- Considering the coil as **Piping** is the header size i.e. 4”/DN 100 thus Cat^P = PS x DN = 2000

Step 6

When considered as a vessel the point corresponding to 20 barg and 18.36 L on **Chart 4** is in the **SEP – Article 4/3 area**.



Now when considered as **Piping** the point corresponding to 20 barg and DN 100 on **Chart 9** is also in the **SEP – Article 4/3** area.



In accordance with Guideline B-04 and as Cat^P >= Cat^V the category is associated with the category defined as **Piping**.

The required PED category for the equipment is **Article 4/3 – No CE mark**.

EXAMPLE #2

A coil included as part of an Industrial process uses a thermal oil with an operating temperature of 250°C. The Design & Maximum allowable pressure is 12 barg.

The stainless steel coil is 1080 mm high with full height 4” headers and a total coil volume of 130 litres.

Solution

Because the maximum allowable pressure (PS = 12 barg) of the equipment is above 0.5 barg, the PED will apply.

Following the PED classification methodology:

- Determine the type of pressure equipment being considered
- Determine the state of the fluid in the equipment
- Determine the hazard group of the fluid in the equipment
- Select the appropriate hazard category chart
- Determine the maximum allowable pressure and the defining dimension of the equipment
- Determine the PED hazard category

Step 1

The coil is outside the scope of Guideline B-04 and must be considered as a **Vessel**.

Step 2

The thermal oil has boiling point of 280°C and thus at the design temperature of 250°C, is below the boiling point and thus considered as a liquid.

Step 3

The thermal oil has a Flash Point of 130°C, so the fluid is considered as a Dangerous liquid and thus falls into **Group 1**.

Step 4

The appropriate hazard category chart to use is Chart 3 when considered as a Vessel.

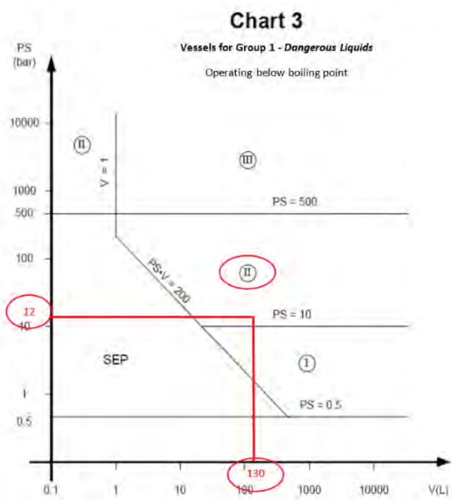
Step 5

The maximum allowable pressure is defined as PS = 12 barg.

The defining dimension is the **Vessel’s** total volume, V_T = 130 litres thus $Cat^V = PS \times V_T$ = 1560 barL

Step 6

When considered as a vessel the point corresponding to 12 barg and 130 L on **Chart 3** is in the **Cat II** area.



The required PED category for the equipment is **Category II/Module B + D – ID 0062 & CE marked**.

EXAMPLE #3

Hot air from a process is required to heat water to 190°C at an operating pressure of 10 barg. The maximum allowable pressure is specified as 15 barg and a maximum allowable temperature is 200°C. The stainless steel tube coil selection results in a coil internal volume of 100 litres.

Solution

Because the maximum allowable pressure (PS = 15 barg) of the equipment is above 0.5 barg, the PED will apply.

Following the PED classification methodology:

- Determine the type of pressure equipment being considered
- Determine the state of the fluid in the equipment
- Determine the hazard group of the fluid in the equipment
- Select the appropriate hazard category chart
- Determine the maximum allowable pressure and the defining dimension of the equipment
- Determine the PED hazard category

Step 1

The coil is defined as being used in a 'process' and thus should be classified as a **Vessel**.

The coil is designed to heat water at 10 barg to 190°C.

At 10 barg the equivalent saturation temperature of water is 184.1°C. The equipment type is therefore classed as a **Heated Vessel - Article 4(1)(b)**, which is a special case, also defined as a Vessel, when designed to raise superheated water or steam to temperatures above 110°C.

Step 2

The state of the fluid is not applicable in the case of a **Heated Vessel - Article 4(1)(b)**.

Step 3

Classification of the fluid group is not required for a **Heated Vessel**. However, for completeness, steam is not classified as a dangerous substance and is therefore a Group 2 fluid.

Step 4

The appropriate hazard category chart to use is **Chart 5**. This chart specifically applies to **Heated Vessel - Article 4(1)(b)**

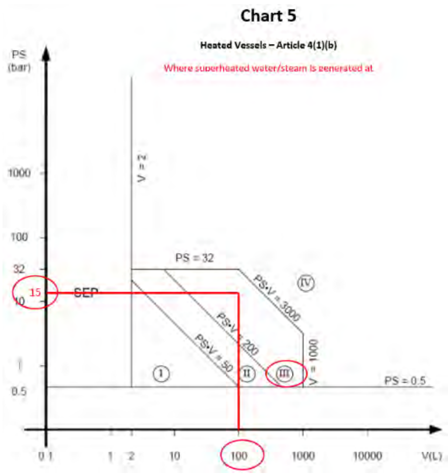
Step 5

The maximum allowable pressure of the steam generator is 15 barg.

The defining dimension of a **Heated Vessel** is the total volume, which is defined as 100 litres.

Step 6

The point corresponding to 15 barg and 100 litres (i.e. PS x V_T = 1500) on **Chart 5** is in the **Category III** area.



The required PED category for the equipment is Category III/Module B + D – ID 0062 & CE marked.

EXAMPLE #4

An ammonia (NH_3 – R717) condenser has a condensing temperature of 35°C and a hot gas inlet temperature of 80°C fed from a screw compressor.

The coil tubes are stainless steel and the finned height is 1920 mm high. Both the inlet and outlet connection sizes are 2½" (DN65). The total coil volume of 55.2 litres.

Solution

As the Client has not specified that the system is fitted with a safety valve set to some pre-determined pressure, we are obliged to refer to EN 378-2 to determine the maximum allowable pressure based. The standard invokes a temperature of 63°C at which the refrigerant saturation pressure should be determined. For ammonia, this equates to 27.1 barg.

Because the maximum allowable pressure (PS = 27.1 barg) of the equipment is above 0.5 barg, the PED will apply.

Following the PED classification methodology:

- Determine the type of pressure equipment being considered
- Determine the state of the fluid in the equipment
- Determine the hazard group of the fluid in the equipment
- Select the appropriate hazard category chart
- Determine the maximum allowable pressure and the defining dimension of the equipment
- Determine the PED hazard category

Step 1

The coil is part of a refrigeration system and can thus be considered as **Piping** as defined in Guideline B-04.

Step 2

The entering ammonia hot gas temperature is 80°C, which is above the saturation temperature of 35°C (its boiling point) and thus considered as a gas.

Step 3

Ammonia is considered as a Dangerous fluid and thus falls into **Group 1**.

Step 4

The appropriate hazard category charts to use are **Chart 1** when considering the **Vessel** aspects (the headers) of the coil and **Chart 6** when considering the coil as **Piping**.

Step 5

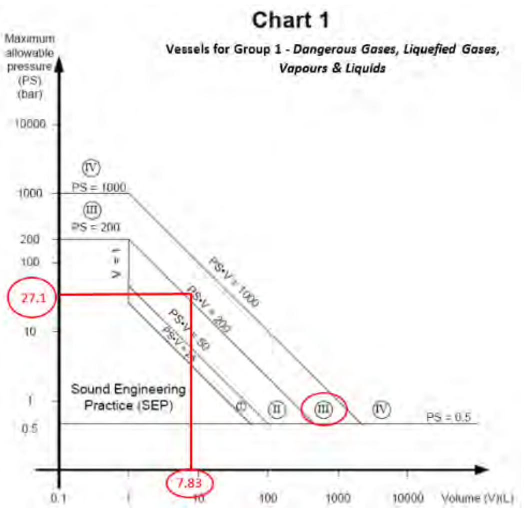
The maximum allowable pressure is defined as PS = 27.1 barg.

The defining dimensions are …

- Considering the **Vessel** aspect of the 2½" header, its volume is V_H = 4.08 dm^3/m x 1.92 m = 7.83 dm^3 or litres thus Cat^V = PS x V_H = 212.2 barL
- Considering the coil as **Piping**, is the header size i.e. 2½"/DN 65 thus Cat^P = PS x DN = 1761.5

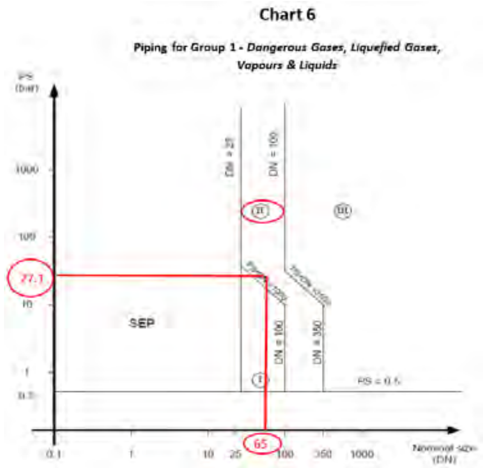
Step 6.1

When considered as a vessel the point corresponding to 27.1 barg and 7.83 L on **Chart 1** is in the **Cat III** area.



Step 6.2

Now when considered as **Piping** the point corresponding to 27.1 barg and DN 65 on **Chart 6** is in the **Cat II** area.

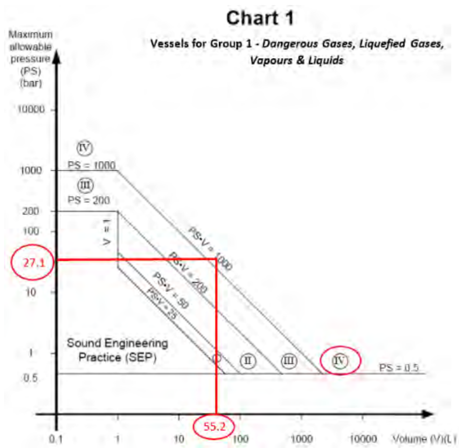


Now we have a conflict …

Because now $Cat^V > Cat^P$, Guideline B-04 **does not** apply and the heat exchanger **must** now be considered as a **Vessel** using **Chart 1** and its total volume V_T = 55.2 litres, so $Cat^V = PS \times V_H$ = 1495.9 barL

Step 6.3

Now when reconsidered as a vessel the point corresponding to 27.1 barg and a total volume of 55.2 L on **Chart 1** is in the **Cat IV** area.



The required PED category for the equipment **is Category IV/Module B + D – ID 0062 & CE marked**.

EXAMPLE #4 – WORK AROUND

The conclusion of Step 6.2 was that $Cat^V > Cat^P$ and thus Guideline B-04 **did not** apply and the heat exchanger had to be considered as a **Vessel** as per Step 6.3

But what if we can make Step 6.1 – *considering the largest header as the Vessel aspect, Cat^V* – fall into Cat II, then Cat^P would be >= Cat^V and the heat exchanger would be classified as **Piping**.

Looking at Chart 1, the upper limit for Cat II is $PS \times V_H$ = 200 barL. Therefore, the maximum PS value to comply with this limit is .. PS = 200 / 7.83 = 25.5 barg.

In which case, the coil becomes classified as Cat^V relating to the largest header volume, as Cat II. Now $Cat^V <= Cat^P$ and thus Guideline B-04 **does** indeed apply and the required PED category for the equipment is **Category II/Module B + D – ID 0062 & CE marked**.

If the Client can accept (*and provide in writing*) that he is responsible for ensuring that the pressure equipment does not exceed a maximum allowable pressure (PS) of 25.5 barg (may be with the fitting of a safety valve), then the coil can be categorised as Cat II (Piping) rather than Cat IV (Vessel).

As an alternative to reducing the Design Pressure to get the product of PS x V < 200, another option would be to divide the pressure vessel into 2 sections and fit 2 sets of connections. Thus, each header would be half the length and if the header size was maintained, then the header volume would also be halved.

Now, $PS \times V_H = 27.1 \times 3.91$ = 105.9 barL and thus Cat^V = **Cat II**

Such workarounds or use of legitimate 'artistic licence' are allowable, providing the PED rules are not breached.

AIR HEATER

Background

The interpretation of what is meant by a 'heater' or 'heating coil' is dictated by the industry sector that one works in.

In the HVAC sector, a heater is usually referred to when air is required to be heated by some form of hot fluid circulating through the tubes of a coil. Such fluids would normally be water or perhaps a glycol/water mix to provide frost protection, but steam or thermal oils might be an alternative 'heating agent'.

Such coils might be further classified by the water temperatures e.g. LPHW, MPHW & HPHW meaning Low, Medium and High Pressure Hot Water …

- LPHW : <90°C and traditionally 82/71°C [*180/170°F*] but more recently, with the popularity of wider operating TDs … 80/60°C or 50/30°C
- MPHW : 90 – 120°C with operating TDs between 25-35K
- HPHW : >120°C with operating TDs between 45-65K

Different countries and global zones may interpret the above 3 categories with alternative temperatures, but as far as EAS is concerned, the above is a good indicator.

Conversely, the 'process' industry sector may apply the terminology 'heater' to the fluid being processed in the tubes. Thus, if the fluid is to be heated by air, then the air may be the high temperature exhaust gas from another process, which is cooled down whilst heating up the process fluid circulating through the tubes of the coil.

So, depending upon 'which side of the fence' you are standing, a 'heater' may refer to the heating of the air or the heating of the fluid … which in this latter interpretation, infers that the air is being cooled, and thus referred to as an air cooling coil … *see next section.*

For the purpose of this design manual and generally in the 'tube & fin' heat exchanger/coil industry, a 'heater' relates to the heating of the air i.e. 'air heater' so, the fluid temperatures will always be higher than the air inlet temperature. Furthermore, by definition, a steam or refrigerant condenser is also an air heater, where the internal tube fluid has a higher temperature than the air inlet temperature.

Discussion of the design philosophy for a LPHW coil

Let us consider the step-by-step calculation of an air heater using LPHW.

The first hurdle to negotiate is the identification of the basic data required to conduct the calculations both relating to the airside and, in this case, the waterside.

HVAC air heater requirements will often either specify the required heating capacity in kWs or dictate the air inlet and outlet temperatures and available air volume.

On the other hand, the Process industry sector air heater requirements will usually be considering the water flow rate and water temperatures or perhaps the fluid cooling duty in kWs.

On occasions, both the airside and waterside data is provided and it is important that there is continuity between the calculated airside and waterside duties, which is often not the case. In such scenarios, the Client will need to confirm which set of data should be used as the basis for the coil design !!

The airside data can be provided in a number of ways and the 'unknowns' calculated …

- Duty, air volume & air inlet temperature → calculate **air outlet temperature**
- Duty, air inlet temperature & air outlet temperature → calculate **air volume**
- Air volume, air inlet temperature & air outlet temperature → calculate **Duty**

Similarly, the waterside data can be provided in a number of ways too and the 'unknowns' calculated …

- Duty, water mass or volume flowrate & water inlet temperature → calculate **water outlet temperature**
- Duty, water inlet temperature & water outlet temperature → calculate **water mass or volume flowrate**
- Water mass or volume flowrate, water inlet temperature & water outlet temperature → calculate **Duty**

Other information needing confirmation is the face size of the coil required to meet the required Duty.

Often the finned height and finned length are provided and this allows the calculation of the airside face velocity. However, there are occasions when perhaps only one of the dimensions is critical and the other left to our discretion to determine.

In such a case, typical air heating coils are often designed with a face velocity in the range 2.5 to 3.0 m/sec and sometimes a little higher. So if the air volume is known and one dimension given, plus an assumption of say, 3.0 m/sec face velocity, the unknown dimension can be calculated.

Clearly, if the finned length is unknown, then the above calculation will provide a dimension that can be manufactured. However, if the finned height is unknown, the calculated finned height must be a factor of the tube pitch i.e. increments of 60 mm for A-fin, 40 mm for C-fin and 25 mm for Z-fin. So when the finned height is adjusted accordingly, the 'actual' face velocity can be recalculated.

Another less common scenario is when only the air volume is given and the Client is not concerned what the fin height or fin length should be but will accommodate whatever we provide. This situation is unsolvable without making some assumptions !

The simplest, layperson's solution, is to assume that the coil is square and thus the face area of the coil would be X^2, where X = length of one of the unknown sides of the square, in metres.

It is the surface area required and not the shape of a coil that determines a coil's behaviour, so whether the shape is square or rectangular does not affect the theoretical thermodynamic behaviour. However in reality, the air velocity distribution implications of unusually shaped coils can have an impact upon performance. So if an axial fan is being used to induce or force air through a coil, then striving for a 'close to square' coil size will be preferable to a narrow and long coil with the same face area.

Thus, the required face area of the coil is the volume flowrate divided by the assumed face velocity, which we shall set to 3.0 m/sec. Thus combining the above, X = Sqrt(volume flowrate / 3.0)

But a face area requirement shaped as a square is <u>not</u> the most cost effective aspect ratio for a coil. It is cheaper to manufacture a coil with a greater finned length than the finned height. This results in shorter headers and fewer tube to bend joints. Incidentally for small HVAC coils, the copper headers can be the 'lion share' of the cost of the coil, so reducing this can be paramount.

So a good compromise is to consider a 3:2 or (1.5:1) aspect ratio, which for the above example provides X = Sqrt(volume flowrate / 3.0 / 1.5) and thus a smaller finned height. So knowing the face area and dividing by X will provide the required (longer) fin length. *See **Coil Related Terminology / Aspect Ratio** for a worked example.*

Once the coil face size has been chosen and the fin pitch and tube & fin materials have been decided upon, plus the Duty, both the air volume and water flowrates plus all four temperatures are known, we can begin to calculate the internal (water) and external (air) heat transfer coefficients to provide the overall heat transfer coefficient, which combined with the surface area per unit face area per row and finally the log mean temperature difference (LMTD), will provide the theoretical number of rows deep the coil needs to be to meet the design data.

This theoretical figure is increased to the nearest whole number of rows, which provides a margin of safety in the calculation.

AIR COOLER

The

STEAM HEATER

The

REFRIGERANT DX EVAPORATOR

The

REFRIGERANT CONDENSER

The

WHAT ASSUMPTIONS ARE MADE WHEN STATING THE RATED CAPACITY OF A PRODUCT?

Rated capacities assume uninterrupted air access to the coil and no hot air recirculation.

Air entering temperatures are defined as the temperature at the inlet face of the heat exchanger and not the surrounding ambient temperature.

Depending upon the installation and environment, the air inlet temperature to the coil may be up to 6K higher than the local ambient temperature.

HOW OFTEN SHOULD THE HEAT EXCHANGER BE CLEANED?

This will depend upon the cleanliness of the environment, both in terms of airborne contaminants that may cause surface corrosion and dust, dirt, leaves etc. that may be induced into the incoming cooling air stream.

Nevertheless, any fouling or corrosion will decrease the effectiveness of the product and so when such a situation arises, the coil should be cleaned. Light brushing of the finned surface (in the direction of the fins with a soft brush) plus an application of a weak detergent is recommended.

It is also possible to use a steam or water pressure system providing that extreme care is taken. It is important that the spray impinges at 90° to the coil face and a safe distance is kept, to avoid deforming the fins.

HOW IS NOISE LEVEL AFFECTED WHEN USING FAN SPEED REGULATION?

The reduction is product related but as an indication, a 25% speed reduction would represent a noise reduction of approximately -6dBA, whilst running at 50% full speed results in around -15dBA.

So clearly, when a 12 fan unit is considered, the lowest noise reduction attainable is when 11 of the fans are switched off. However, this equates to only a -11dBA reduction, which is not as much as when running all he fans at 50% full speed and more importantly, provides only 10% capacity as opposed to perhaps 50% capacity when all the fans are running at a reduced speed.

WHAT ARE THE CONSIDERATIONS WHEN POSITIONING/LOCATING PRODUCTS?

Refer to the EAS website for positioning and location diagrams for recommended distances between horizontal and vertically mounted adjacent products, spacing from walls etc.

Where possible, products should be placed on low ‘solar gain’ roofs and preferably in the shade to avoid effects that can elevate the air inlet temperature by up 6K or more.

ARE THERE ANY SPECIAL CONSIDERATIONS RELATING TO UPRIGHT UNITS?

Upright units with horizontal airflow should be orientated to minimise prevailing wind direction implications such as fan ‘wind milling’ and it is thus recommended that care should be taken when placing product variants fitted with 8 pole and certainly 12 & 16 pole motors. Refer to our website for diagrams suggesting recommended product spacing’s.

When fans are wind milling backwards at perhaps a greater than full speed rpm, activating an AC motor driven fan, will struggle to overcome the backward inertia and the resulting ‘spike’ in the starting Amps will likely overload the motor relays and the overloads will drop out.

To prevent wind milling, AC fans/motors can be fitted with anti-backward rotation brake devices, but even so, low speed motors may still struggle to start if the strength of the prevailing wind cannot be exceeded by the low motor power.

Modern EC motors contain integral intelligent electronics that invoke electrical braking if wind milling is detected and have an inherent ‘soft start’ regime, which will usually eliminate the wind milling issues.

DURING OFF-CYCLE PERIODS, SHOULD MOTORS BE RUN-UP?

We recommend that motors should be run for a minimum of 5 hours per month to avoid motor bearing stiction problems (Brinelling) and to reduce internal condensation build-up.

WHY CAN WATER BUILD-UP INSIDE MOTORS?

Unless a motor is hermetically sealed … and even ATEX rated motors are not … air is able to migrate from the inside to the outside and vice versa, through the oil seals and glands … which are not airtight.

Ambient air, if its relative humidity is greater than 0%, will contain a degree of water, which exists in the form of superheated steam at the water vapour's partial pressure. In such a case, this mixture of dry air and water vapour will have a particular 'dew point', below which the water vapour component will condense into water droplets.

Hopefully, if the motor is fitted with drain holes, such condensation will safely drain away. However, if the drain hole becomes blocked or the drain plug is not removed, then problems will ensue.

So, if a motor that has perhaps been running during the day at its working temperature, is subject to night-time temperatures below the dew point of the inside 'contained air', whilst the motor is off-line, then the condensation generated as the whole motor (and contained air) cools to match the low night-time temperature, will remain inside the motor until it is started again. Then as the motor reaches working temperature, the increased temperature has the effect of 'driving' out and drying out the internals of the motor.

Problems arise when a motor is stationary for long periods, and the cyclic temperature fluctuations cause a build-up of water inside the motor because the drain plug has not been removed or the motor is ATEX or similar rated and not drain hole is provided. Usually, in such cases, anti-condensation heaters are provided to ensure that the motor remains 'warm' and condensation issues do not arise.

In worst case scenarios with no drain hole, blocked drain hole or non-activated anti-condensation heaters, may result in the water level inside a motor reaching halfway up the rotor. For an AC induction motor, water inside the motor (*but not the electrical terminal box*) is not a problem and it will still operate. The issue is that the motor bearings are not designed to operate in water and will, over time, fail often causing the whole bearing assembly to seize and rotate in the softer material 'end shield's, which eventually result in a motor failure.

WILL LOW AMBIENT TEMPERATURES IMPACT UPON COMPONENT ELECTRICAL BEHAVIOUR?

In severe low temperature environments both motors and electrical panels/switchgear will benefit from the fitting of anti-condensation heaters. Excessive moisture build-up where electrical switchgear is concerned, inside motor housing and bearings etc. can cause component failure or reduced life cycle.

For temperatures below -40°C, special materials may need to be considered and in the case of motors, special lubricants/grease used in the bearings.

At low temperatures fan motor start-up power/current will be affected because of the higher air density. As an example, the impeller absorbed power (kW) increases by 25% at -40°C compared with +20°C.

ARE THERE ANY PIPEWORK RELATED RECOMMENDATION?

Discharge line dampers/compensators for condensers or flexible couplings for dry coolers should always be fitted to avoid over stressing the heat exchanger tubes and/or headers.

IS IT RECOMMENDED TO FIT ANTI-VIBRATION MOUNTS (AVM)?

All rotating equipment should be both statically and dynamically balanced and thus vibrations are minimal. However, depending upon the location and environment, products can be mounted upon anti-vibration pads/springs to either avoid exceeding published noise levels or preventing transmission of noise or vibrations through the roof or building structure.

If AVMs are used, ensure appropriate flexible pipe work connections are included to allow for product movement.

CAN SUPPLY PIPE WORK BE DIRECTLY CONNECTED TO THE PROVIDED CONNECTIONS?

It is recommended that flexible connections are always used between the supply pipe work and the connection fittings supplied on the product.

In the case of industrial dry coolers, where high fluid temperatures are often encountered, thermal expansion considerations must be accounted for to ensure that the product is not adversely affected.

The supply pipe work should always be self-supporting to avoid over-stressing the product connections and heat exchanger tubes.

For condensers, it is advisable to use hot gas discharge line dampers to isolate the product from the high frequency pressure pulses often associated with reciprocating compressors.

CAN A FORKLIFT BE USED TO HANDLE THE PRODUCTS?

Consult the Operating and Maintenance instructions to clarify this issue.

Depending upon the product size and weight plus the length of the forks on the forklift, some products can be safely handled providing care is taken to avoid damage to the underside of the product and in particular, the coil (heat exchanger). However, products should never be handled or lifted via their headers. Always use the lifting lugs provided in accordance with the O&M instructions.

CAN THE PRODUCT BE DAMAGED WHEN CONNECTING TO THE SUPPLY PIPEWORK?

Condensers are not usually an issue unless excessive heat is applied when brazing/welding the pipework.

For dry coolers, care should be taken when tightening screwed (BSP) connections, to avoid over stressing the coil tubes and headers.

All supply and return pipework should be independently supported and never hung from or in any way supported by the product, especially if AVMs are fitted to the product.

WHAT CAN HAPPEN IF AMBIENT CONDITIONS DROP BELOW FREEZING?

Adequate frost protection measures should be taken to avoid coil failure from frost damage. These may include using a suitable fluid that will not freeze, fitting of a ‘winterization’ kit to ensure that fluid temperatures remain above freezing or provision of a system that automatically evacuates and drains the system when temperatures become critical.

ARE DRY COOLER COILS ‘SELF-DRAINING’?

In theory, yes, the circuitry is designed to drain freely. However, the tube length, product orientation, circuitry pattern and surface tension considerations of the fluid may prevent the coil(s) from fully self-draining. In this event, ensure that the fluid is evacuated under pressure.

If sufficient fluid remains in a particular circuit, serious frost damage can result from low ambient temperatures.

Under certain circumstances, the circuitry for dry coolers can be arranged with additional headers to replace return bends. Each header is fitted with air vents and drains and if the product is installed on a positive incline, when the vents & drains are opened, the whole coil will self-drain. However, it is always advised that the emptying of the system is assisted by air pressure to ensure complete evacuation.

WHAT IS MEANT BY CAPACITY CONTROL?

When applied to air cooled condensers and dry coolers, is a generic term usually referring to alternative methods of adjustment of the air volume to match the desired capacity of the product, which may vary throughout the day.

In the case of condensers, the controlling criterion is usually the ‘Head Pressure’ or condensing pressure monitored at the inlet to the condenser and for dry coolers, the criterion is usually the fluid leaving temperature – also known as the ‘process entering’ temperature.

The sensor monitoring the head pressure or fluid leaving temperature provides a signal to one of the following air flow control systems which endeavours to match the ‘set point’ condition…

- Fan step controller - also referred to as a fan cycling controller - starts and stops individual or pairs/groups of fans in sequence or randomly, resulting in a variation in the cooling load. This ‘wide band’ control methodology imposes different cooling behaviour over the different sections of heat exchanger surface as fans are activated and deactivated. The method is simple and usually cheap but does not necessarily provide ‘close control’
- Fan speed regulation relates to varying the speed of all the fans simultaneously, thus adjusting the cooling load equally over the entire surface of the heat exchanger. Clearly this is a preferable control methodology but requires a more sophisticated and thus more expensive control system.

Currently the favoured alternatives are frequency inverter solutions or using EC fan/motors and controlling them via a 0-10V signal or digitally via Modbus. Both result in the ability to adjust the speed of the motors/fans and hence reduce the air volume / cooling load to match the set-point condition.

HOW IS CAPACITY CONTROLLED VIA A STEP/FAN CYCLING CONTROLLER?

A fan step/cycling controller in its simplest form can comprise a number of thermostats (dry cooler) or pressure stats (condenser), each with its own sensor and which are adjusted to slightly different incremental set-point conditions. As each set-point condition is reached the thermo/pressure stat activates one or more motor contactors, thus providing additional (incrementally stepped) air cooling to meet an increasing thermal load.

Although for ‘course’ capacity control this system operates adequately, the inherent hysteresis and required ‘dead zone’ between each step, does not make it suitable for ‘close control’ scenarios.

However, today it is more common to provide electronic step controllers utilising only one sensor and typically offering 2 through to 8 steps all integrated into one electronically programmable device. Thus, somewhat closer control is achievable.

Furthermore, the programming functionality allows for sequential, rotational or randomised triggering of the motor relays, which ensures more even usage of the motors fitted to the product and helps equalise the motor lives.

HOW IS CAPACITY CONTROLLED VIA FAN SPEED REGULATION?

Unlike a step controller, speed regulation provides capacity control by infinitely varying the air flow across the heat exchanger; by simultaneously modulating the fan speed of all the fans fitted to the product; to match the required thermal load.

The speed regulation can be achieved by either varying the voltage to the motors, by modulating the AC supply frequency via an inverter or for EC motor, a scaled voltage or digital signal via Modbus.

For IEC/NEMA AC motors, frequency modulation provides a very efficient method for adjusting the fan speed without the inherent electrical inefficiencies associated with voltage modulation.

A sensor (temperature or pressure) monitors the operating conditions of the dry cooler or condenser and electronically compares the signal with a set-point. The EC motor or AC motor inverter microprocessor translates the differential into the necessary signal resulting in a motor speed increase or reduction. This in turn, modulates the cooling air flow to maintain a stable operating conditions.

Generally, the electronic and programming functionality provided by an EC controller or AC inverter allows for a high degree of system customisation and control behaviour besides the ability to interface with Building Management Systems (BMS), which are becoming commonplace.

WHAT IS A MOTOR THERMAL CONTACT?

For AC motor installations, a simple ‘starter control panel’ for dry coolers and condensers often use current overloads fitted to the motor contactors to provide protection in the event of a motor related problem. Often this is more than sufficient if the motor runs at full speed when operational.

However, if the motor is speed regulated and thus can operate anywhere between zero and full speed, then when running at reduced speeds (*10% of full speed is the recommended lower limit*), it is possible that the protecting current overload may not sense a problem and thus not ‘trip’ even though the motor windings become excessively hot. Overheated motor winding can result in either a breakdown in the winding insulation, reduced motor life or in the worst case, motor burn-out and failure.

To avoid such a scenario, ‘thermal contacts’ can be embedded into the windings which can be connected to the motor control circuitry to isolate the motor in the event of excessive winding temperatures, either when the motor is running at low speed or if a fault occurs.

Thermal contacts are usually small bi-metallic relays (PTO) which react to temperature. When the temperature exceeds the set-point, the contacts open and provide a break in electrical integrity. On its own, a thermal contact will not do anything, but when connected to a control relay or in series with the supply voltage to a motor contactors solenoid coil, will provide a break in the electrical continuity when it overheats. Thus, the motor will be isolated and if provisioned for, an appropriate alarm signal generated.

WHAT IS A KLIXON?

A klixon is a term often incorrectly used to infer a thermal switch, often referred to as a PTO.

In fact, Klixon is both a company name, product range and a trade name for ‘snap-acting bi-metal discs’ used as electrical switch gear, often built into hermetic compressors. However, the common misuse of the phrase by this industry has resulted in klixon, bi-metallic contact, thermal contact and PTO referring to the same functional device.

WHAT IS A THERMISTOR?

A thermistor, often referred to as a PTC is a small semiconductor device which is a type of resistor used to measure temperature. Depending upon its makeup, it can either provide an increase (PTC) or decrease (NTC) in the resistance as the temperature rises.

Unlike the thermal contact, it cannot be directly connected in a circuit to provide a physical break in electrical integrity but must be powered and wired to some other device that can sense the positive or negative change in resistance before an action can take place. Therefore, depending upon the sophistication of the control circuitry required, a thermistor-based system tends to be a more expensive option or provides less sophistication when considering individual motor protection.

WHAT ARE THE IP RATINGS FOR ELECTRICAL EQUIPMENT?

Generally, AC motors for dry coolers and condensers are either rated as IP54 or 55 (incl. drain hole), whilst brine unit coolers and evaporators are often rated at IP44 or 54.

Lockable safety switches are nominally rated at IP65, but as a matter of necessity often include a drain hole to allow the egress of any condensation. Thus, such electrical switch gear is effectively rated at IP54.

Electrical control panels housing equipment such as speed regulation, step control, ‘wet system’ controls etc. are initially IP65 or indeed IP67 prior to inclusion of the electrical equipment and on occasions, cooling fans. Thereafter, the enclosure’s IP rating drops to typically IP55 or even 54 or whatever the lowest IP rating of any such component that is fitted to the control panel, which may affect the overall IP rating.

WHAT IS THE EASISPRAY SYSTEM?

EASiSpray (water spray technology) is the terminology used when the heat exchanger element of a dry cooler or air cooled condenser is sprayed with water to depress the normal dry bulb air inlet temperature (pseudo dry bulb), thus widening the effective operating TD (temperature difference) resulting in a boost in the capacity or alternatively, requiring less surface to dissipate the thermal load. Often this technology is used to meet abnormal ‘peak load’ performance from a product sized for normal ambient conditions negating the need for oversized or redundant dry coolers.

This particular ‘sprayed coil’ technology should not be confused with adiabatic cooling technology where dry bulb temperature can be depressed close to saturation (wet bulb) conditions e.g. cooling towers.

Distribution of the water is achieved by a matrix of ‘hollow cone’ spray nozzles served by a series of plastic or copper sparge-pipes which spray the water in opposition to the direction of air flow and thus <u>not</u> directly onto the coil surface. The nozzles are sized to provide the correct amount of water in relation to the available mains water pressure.

Mains water pressures from 0.5 to 10 barg can be accommodated, however, 2 to 3 barg is more typical.

The water spray process is intentionally designed as a ‘total loss’ system providing typical water droplet sizes of 100-240 microns which is significantly greater than the 5 micron generally considered as the limit for human inhalation. Both these factors are important in relation to eradicating Legionella bacteria related issues.

Spraying the coil does indeed depress the air inlet temperature, but not as efficiently as with a truly adiabatic solution. Thus, we do not claim that we can depress the air dry bulb temperature to equal the wet bulb temperature i.e. reach saturation.

However, we are able to reach saturation efficiencies of between 60 and 80% depending upon the air inlet temperature, fans speed/air flow. Nevertheless, significant performance improvements can be achieved.

The water spray variant of a dry cooler or air cooled condenser is a solution to accommodate abnormal peak load ambient variations - which seem to be occurring more frequently these days. Thus, if adopted, it is recommended that hydrophilic coated fins are used to resist lime scale build-up (deposition of minerals suspended in the sprayed water) onto the finned surface which over time, can affect the product performance.

Consequently, it is recommended that such a system should be operational for less than 200 hours per year during which time, regular inspections and cleaning of the heat exchanger surface should be scheduled.

One negative aspect of this technology is the possibility of ‘drift’ under extreme conditions. However the products are designed to eradicate this problem.

WHAT IS THE EASIPAD SYSTEM?

EASiPad (pre-cooling pad technology) is the terminology used when the product is fitted with pre-cooling pads (wet absorbent media) in front of the heat exchanger, through which ambient air is induced.

The resulting adiabatic cooling process ‘depresses’ or lowers the dry bulb temperature, widening the operating TD and allowing a greater capacity to be achieved from the same heat exchanger.

The use of this Pad technology allows the dry cooler/condenser to operate ‘wet’ for sustained periods … unlike the EASiSpray alternative, which involves water sprays and thus water in contact with the heat exchanger surface. Typically, such a system has a recommended time cap of 200 hours per year. Furthermore, it is recommended to use a hydrophilic coated aluminium fin material with the EASiSpray system … resulting in both a cost implication and a marginally lower performing extended surface.

A further benefit of the Pad system is that none of the added water comes into direct contact with the heat exchanger surface, thus maintaining the longevity of the extended surface. Additionally, no water carry-over is exhibited and thus no detrimental ‘drift’ implications are associated with this ‘wet’ system.

A spin-off of, in effect, fitting a ‘filter’ to the incoming air stream ensures that the heat exchanger is kept clean and maintains its performance. Most dry coolers/condensers become dirty or clogged, thus impacting upon the cooling air volume, and/or corroding the extended surface.

HOW IS AN EASISPRAY/EASIPAD SYSTEM CONTROLLED?

Wetting the pre-cooler pad or spraying city water into the incoming air stream is the last stage of capacity control to match the cooling demand when either all the fans are activated via a step controller on a multiple fan product, or alternatively all the fans are running at full speed in the case of a variable speed regulation system. Thus, the solenoid valves that activates the water spray system are interlocked such that they will not operate until the full air volume is achieved … unless overridden by the controller if the ‘energy efficient’ mode is selected.

When the product is running at full design air volume and the cooling demand can still not be met, the pad or spray system is triggered. This imposes a ‘thermal shock’ to the system often resulting in over-cooling and thus the control system will sense this behaviour and shut the mains water solenoid valve. Also interlocked with the mains solenoid valve is a drain valve that allows the water in the distribution system to naturally drain from the pipework - eliminating any Legionella related ramifications.

Once the ‘wet’ system is deactivated, the residual cooling thermal inertia may cause the step controller to turn off one or more fans or in the case of a speed regulation controller, may cause all the fans to reduce in speed. However, over time, the insufficient cooling provided by lower speed fans will cause the ‘wet’ system to be re-activated and the whole cycle repeats.

Reduced system hysteresis can be achieved by using a staged spray or pad system where increasing portions of the coil surface are affected to better match the cooling demand and minimise over-shoot scenarios.

The deactivated water system is designed to be free from any residual water, however the incoming water mains supply pipework should either be adequately insulated and provided with ‘trace’ heating plus an automated evacuation/’blow down’ system to drain the water should be allowed for to eliminate frost damage. These latter considerations are not included in the scope of supply of the water spray equipment and thus are the responsibility of the End User.

ARE THERE ANY WATER QUALITY CONSIDERATIONS FOR THE EASISPRAY SYSTEM?

A EASiSpray control package ordered in addition to the fitted or perhaps 'supplied loose' sparge pipe system and nozzles, comprises a filter, solenoid valves, regulation valves and pressure/temperature sensor, but we expect that the incoming water feed is clean, free from waterborne particles and preferably 'soft' rather than 'hard'.

Water quality levels should comply with …

- pH levels between and 7.5
- Hardness 6.5-8°fH (3.5-4.5°dH)
- Ca / Mg carbonate levels < 120 ppm

Should it be determined that the water is 'too hard', it is recommended to fit a water softener to the mains water supply line.

WHAT IS WIND-MILLING?

Wind-milling refers to the condition where axial fans that are 'off line' and should be stationary are in fact rotating due to localised air circulation or prevailing winds. This condition is usually associated with upright units (horizontal air flow) but can affect Vee type products.

If the undesirable prevailing air flow is through the coil and thus emulating the normal mode of operation, then wind-milling is not usually a problem.

However, if the air flow direction is against the normal operating mode, then the fan(s) will wind-mill backwards and on occasions, faster than their normal operating speed.

In such cases, when the fan(s) are required to start they must overcome the 'backward' inertia before they can run-up to full speed in the correct direction. This situation can become critical if 12 & 16 pole motors are fitted to the product and the resultant start-up current drawn exceeds the electrical overload settings. The tripping of the current overloads result in some or indeed all the fans remaining idle and thus no forced draught cooling is available.

Consequently, the set-point conditions cannot be fulfilled and in the case of a condenser, the system may trip-out on high pressure or for a dry cooler, the fluid leaving temperature exceeds the allowable operating temperature resulting is system shut-down.

AC motor fan assemblies can be fitted with 'anti-wind milling' brakes, which prevent the fans from freewheeling backwards. This can help the motor to start, but in extreme conditions, the starting torque to overcome the reverse direction air force, may not be sufficient to allow the fan to start.

EC motors or driven fans have the 'intelligence' to electronically brake the fan and perform a 'soft start', so such fans are less prone to wind milling related issues.

WHAT IS A 'SANDWICH' COOLER?

When a process produces a hot fluid or fluids at more than one temperature that should preferably be handled in the same cooling device, then the resultant product/dry cooler is often referred to as a sandwich cooler.

Usually, the product is a variation of a standard dry cooler where perhaps the 6 rows of coil and associated surface area are divided into 2 rows for the high temperature process and 4 rows allocated to the low temperature process.

Usually the cooling (ambient) air stream passes through the low temperature section first and then through the high temperature section.

Clearly, in this mode of operation it is not possible to accurately control each process fluid leaving temperature by air volume modulation via speed regulation, alone. If the primary process temperatures are control via the air flow capacity control system and the secondary process controlled via 3-way valve fluid mixing, then some degree of dual-mode control is viable.

If, however, the processes are not critical, then capacity regulation controlled via the primary process temperature may be good enough.

HOW TO CONTROL A PRODUCT WITH BOTH AN HT & LT SECTION?

The simple answer is … with difficulty !

Firstly, if this less than ideal scenario is the only viable possibility, one of the operating sections must be defined as the 'master' section and its fluid outlet temperature determines the behaviour of the capacity control system. Therefore the 'slave' section performance will follow accordingly.

If both processes call for 'close control' then a 'sandwich' cooler concept is not a viable option. The alternatives are...

- Each process should be handled by a dedicated dry cooler with its own capacity control system
- Alternatively, use a product with 2 lines of fans and 2 discrete heat exchanger sections where each line of fans is controlled independently by the heat exchanger lying beneath

This latter solution has its own shortcomings relating to heat transfer by conduction between the adjacent high and low temperature sections. Thus often, an oversized product must be provided to match the combined capacity requirements.

CAN AMMONIA (R717 – NH3) BE USED WITH STANDARD PRODUCTS?

Standard product ranges utilising copper tubes are not suitable as Ammonia is highly aggressive towards copper. However, in such a case alternative tube materials such as stainless steel or aluminium can provide a solution.

WHAT IS THE DIFFERENCE BETWEEN EN13487, HEMISPHERICAL & FREE FIELD SOUND DATA?

EN13487 is the preferred sound level standard adopted by Eurovent to present sound pressure levels at a distance in a free field environment.

This methodology provides for the correction to the certified/approved overall A-weighted sound power level (L_wA) with respect to parallel-piped enveloped constructed around the product, typically at a 10 meter distance, resulting in the overall average A-weighted sound pressure level (L_pA).

Thus, $L_pA = L_wA – 10 \log_{10} A$, where A is the surface area (m^2) of the constructed parallel-pipe envelope.

Depending upon the size of the product in question, the surface area reduction component for a distance of 10 m is typically between -31 and -33 dBA.

The **Hemispherical** approach is to apply exactly the same correlation as above, but in this case the surface area (A) of the envelope under consideration is a hemisphere with a radius (R) of a defined distance, typically 10 m.

Note that this methodology assumes that the noise source is a 'point source' … clearly not the case with dry coolers or condensers with more than one fan! Indeed, this product type is neither a line source, but a 'line of point sources' and thus subject to more rigorous analysis.

However, this is where 'artistic licence' is often applied because the normal defined radius of 10 m can either be assumed to be from the centre point of the product or more often and thus commercially more acceptable, a radius equating to the distance from an observer's position 10 m away from the extremities of the product to the centre point of the product. Clearly, this latter scenario is a greater distance and thus a greater surface area and hence lower noise level !

Thus, considering point source theory applied to the centre point,

$L_pA = L_wA – 10 \log_{10}(2\pi R^2)$ which further reduces to $L_pA = L_wA – 20 \log_{10} R – 8$

Now when R = 10, $L_pA = L_wA – 28$ …. apparently +3 to +5 dBA noisier than the EN13487 prediction!

Alternatively for a product size 4 m x 2 m x 1.5 m and a distance 10 m from the end of the product, the effective radius would be R = 12 m and then the above relationship results in $L_pA = L_wA – 30$ … mysteriously becoming 2 dBA quieter.

Both solutions are correct, but it depends upon the definition of R that dictates the theoretical noise level.

The **third method** often used to present noise levels is referred to as just 'free field' and is usually an empirical variant of the theoretical prediction and takes into consideration the directivity and constructional characteristics of the product.

If one was to plot a 3D sound pressure level map around a real product, then it would be clear that the sound level changes circumferentially around the product and indeed at various elevations.

The measured values at a given distance from the header/return bend end (short side) of the product will be slightly lower than measurements from the long side at the same distance. Furthermore, the measured sound level above the unit increases still further. These variations are generally not identified when presenting overall average sound pressure data because these are derived from overall average sound power levels derived in accordance with ISO 3744 or ISO 9614-2, which by definition are included within the overall average figures.

Nevertheless, sound levels presented as ‘free field’ at 10 m from the header end (short side) of the product can be up to -3 dBA lower than figures derived from overall average sound power levels.

Finally, certain product variants are subject to far greater directivity implications; for example, Vee type products that have vertically inclined coils and thus radiate sound horizontally rather than reflection from the ground. Furthermore, the sound levels at the header end of the product can be up to -4 dBA lower than from the open coil side of the product. This phenomenon results from the increased attenuation from the sheet metal end sections.

WHAT REDUCTION IN SOUND LEVEL CAN FANS FITTED WITH SILENCERS GIVE?

The best case scenario is ‘not a great deal’, only -3dB.

If a product was able to be suspended in free space, then the measured sound level would be -3 dB less than the ‘real’ scenario, where there is indeed reflection from the ground.

Noise emitting equipment such as a fan will generally radiate sound energy and thus ‘pressure waves’ in all directions. Clearly the sound energy is often channelled vertically through the fan orifice and thus a proportion of the noise exits in this fashion. But approximately 50% of the noise energy travels backwards through the coil and is reflected off the surface below.

Clearly the ‘absorption coefficient’ of the surface upon which the product is mounted, will govern the strength of the reflected sound pressure waves. Thus, an observer at a distance from a product receives one sound pressure wave directly from the discharge of the fan/motor and a second - partially attenuated - pressure wave reflected from the underside surface of the product. Hence if the discharge (fan side) noise level is fully attenuated, then the observer still receives the reflected noise level … approximately -3 dB lower than without attenuation.

To provide levels of attenuation greater than -3 dB, the inlet side of the product must be attenuated in some way or have an improved inlet efficiency. Marginal improvements can be achieved by acoustically insulating the inside of the fan deck assembly but if the break-out through the coil is not addressed, there will be no significant attenuation.

Clearly, placing acoustic louvers etc. on the air inlet tract to the product invariably creates pressure drop and thus reduces the cooling air volume and capacity and thus is often not a solution. Furthermore, changing the fan operating condition may decrease its efficiency and thus ’increase’ the generated noise level, partially offsetting the whole purpose of fitting attenuators!

Erecting barriers (even acoustically lined) around the product is a simple method of providing significant attenuation. However, if positioned too close to the product, there may be an impact upon the product’s performance.

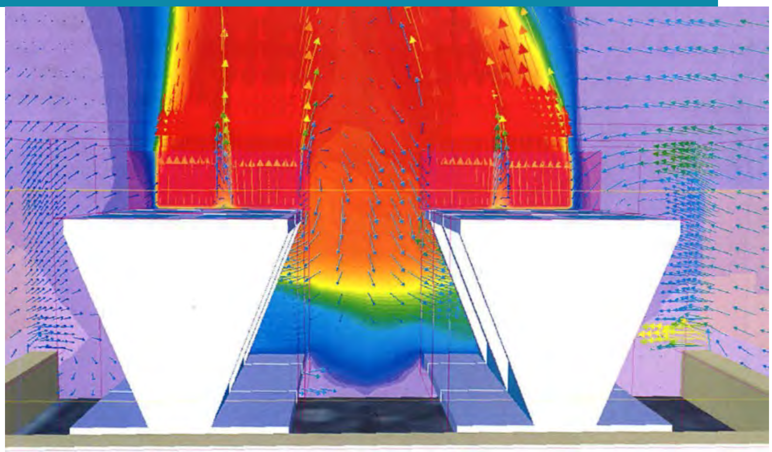
In conclusion, fan discharge sound silencers claiming greater than 3 dBs attenuation should be viewed with suspicion. However, if the silencer is in the form of a ‘static regain’ device, which improves the fan operating efficiency, then the fan speed reduction that can be invoked, will increase the ‘effective’ attenuation by perhaps a further 1-2 dBs.

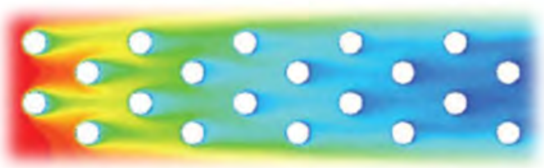
TOPICS TO WRITE ABOUT

- *** TODO ***
- Cooling at high temp. and high humidity – new high temp. psychrometric algorithms
- Describe nominal air volume based on duty and max feasible air TD … air off = fluid inlet - 2K
- Theory behind condensation & evaporation
- Water, deionised, demineralised
- White rust - galv sheet metal
- Strength calculations. Implications of temperature and drop in tensile strength.
- AOM temperatures
- Cable routing and deration at elevated temperatures

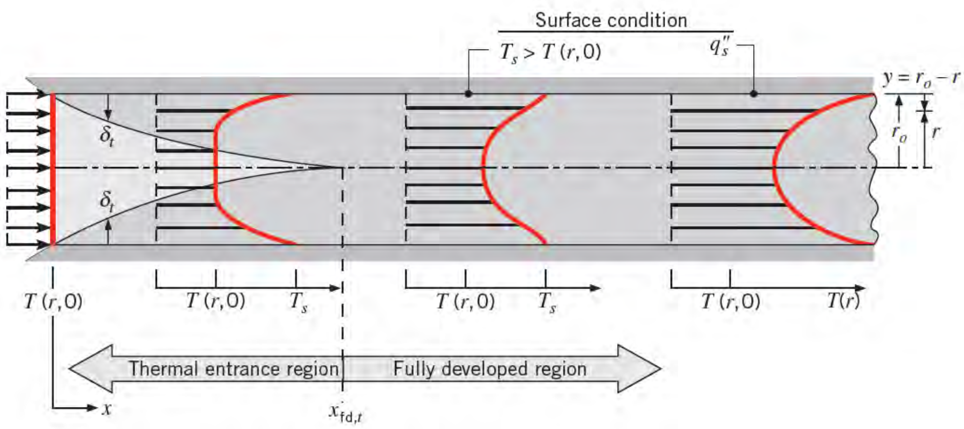
MISC

Air recirculation & thermograph of fin surface temperature

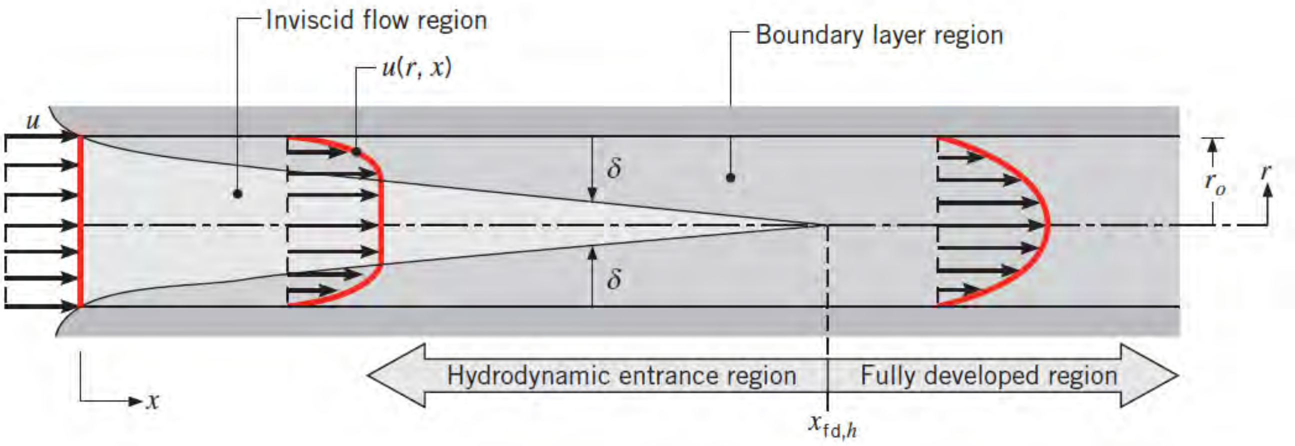


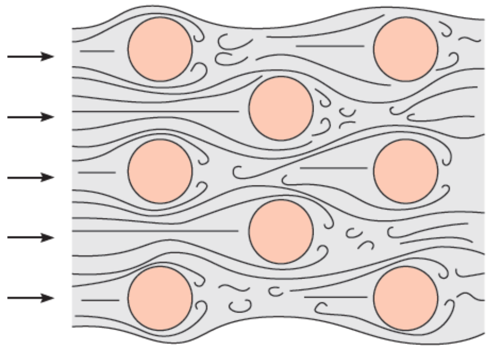


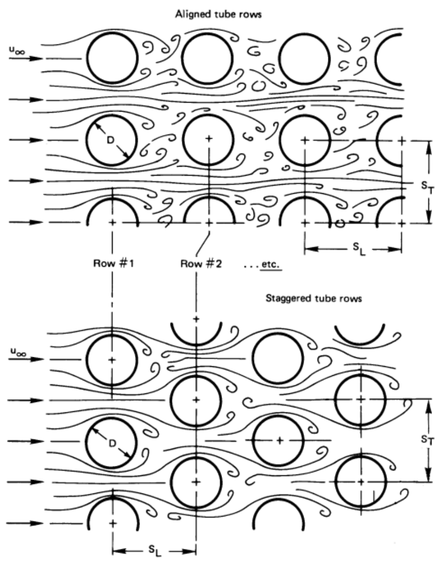
Thermal boundary layer development in a heated circular tube



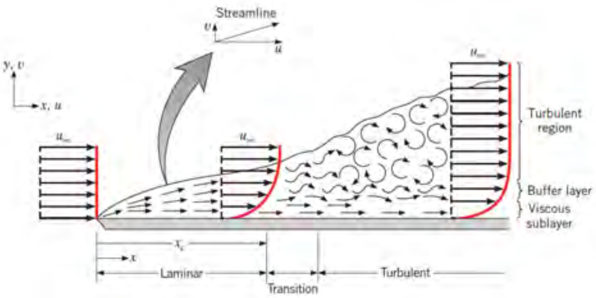
Laminar, hydrodynamic boundary layer development in a circular tube



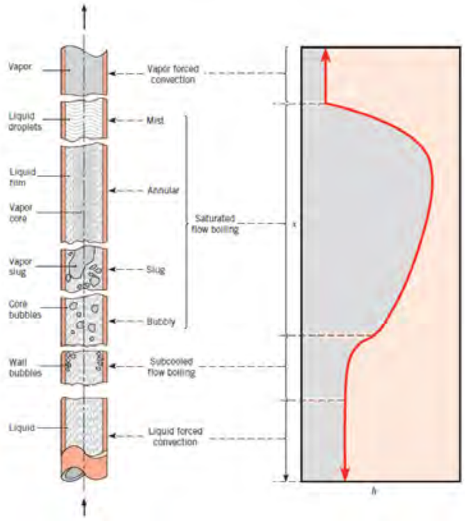


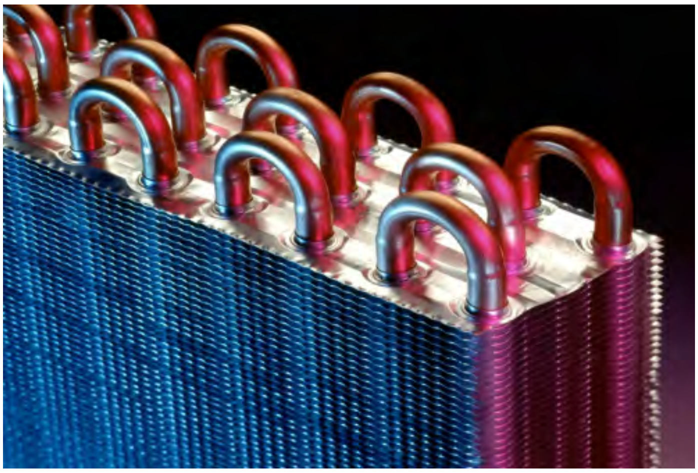


Velocity boundary layer development



Flow regimes for forced convective boiling in a tube





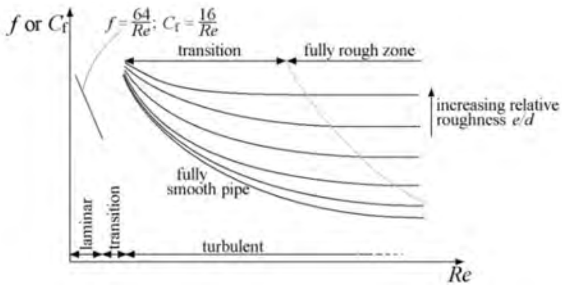
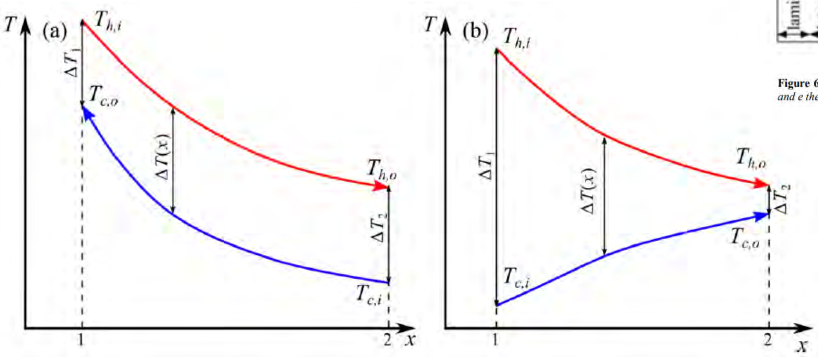
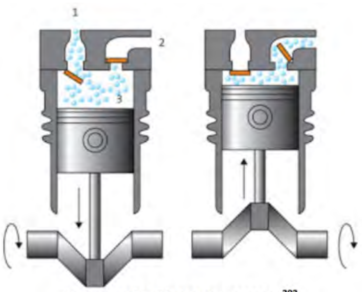
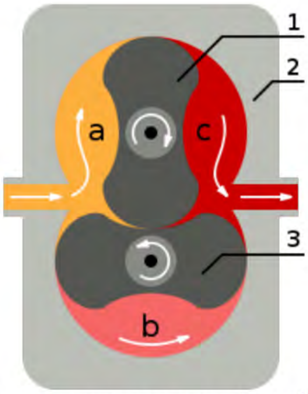


Figure 6.4. *Illustration of a Moody's char; C_f is the Fanning friction factor, f the Darcy friction factor, and e the absolute surface roughness.*





Reciprocating Piston Compressor[293]



Roots blower compressor mechanism[294]

Centrifugal compressor impeller[296]

Lysholm screw compressor mechanism[295]