Design and Selection of Industrial Finned Air Coolers for Natural Refrigerants – A Comparison between NH_3 and CO_2

Stefan S. Jensen, B. Sc. Eng. MIE Aust, MAIRAH

Executive Summary

In practical blast freezer applications, the required finned air cooler surface area for an evaporator employing CO_2 is found to be approximately 25% lower than the equivalent unit using NH_3. Gravity flooded CO_2 low temperature evaporators are found to perform 10 to 12 % better than equivalent NH_3 units all other things being equal.

Small bore evaporator geometries as typically used in commercial HFC applications are demonstrated to potentially further improve heat transmission coefficients by around 45% when used with CO_2. This is for an industrial application in comparison with evaporator geometries not specifically optimized for use with CO_2.

Calculation of Air Cooler Performance

The calculation of air cooler performances follows basic heat transfer theory. The methodology is determination of the inside film coefficient, the external film coefficient, fin efficiency and overall heat transmission coefficient also known as k-value or u-value.

For the calculation of overall air cooler performance, it is necessary to determine the overall temperature differential between air and refrigerant. This is also known as logarithmic temperature difference or LMTD.

The refrigerant side pressure drop affects the LMTD. A pressure drop transforms into a temperature change as the refrigerant flows through the air cooler and changes phase gradually throughout this process. The greater the temperature drop, the lower the available LMTD.

The flow pattern between air and refrigerant (counter flow, parallel flow or cross flow) determine how large the impact of the refrigerant side pressure drop is on LMTD and hence on overall cooler performance.

With a relatively large refrigerant side pressure (temperature) drop, it is often advantageous to circuit the air cooler for physical parallel flow (thermodynamic counter flow). This circuiting method can, however, create other practical problems/considerations with respect to defrost, reliable superheat signal if dry expansion feed is used, with liquid distribution uniformity in the case of liquid overfeed or with oil return/drainage.

Stefan S. Jensen is Managing Director of scantec refrigeration technologies pty ltd, Brisbane, Australia.

The dynamics of air cooler performance are complicated to capture. In the great majority of design situations, an air cooler is selected for a specific operating point – usually maximum performance.

This selection method – although very common – can and has caused practical problems in many cases. Typical examples are dry expansion feed applications combined with variable fan speed. Failure at design stage to identify the impact of frost or oil fouling has also been the cause of unsatisfactory coil design.

The performance of an air cooler will alter constantly with changing operating conditions. The common assumption that the overall heat transmission coefficient or k-value remains more or less constant throughout a wide range of operating conditions is incorrect.

The common assumption that the performance of a given air cooler may be extrapolated linearly on the basis of the temperature difference between air and refrigerant is also not reliable. This would assume that the impact of refrigerant side pressure drop for a given air cooler is constant, irrespective of heat flux and this is clearly physically impossible.

Fortunately, natural refrigerant applications are not complicated by the presence of a temperature glide. This eliminates the need to identify at what point in the evaporation process the relationship between pressure and temperature is defined.

Manual calculation of the performance of a finned air cooler is possible, but time consuming in the extreme. Most manufacturers of finned air cooling equipment offer software packages for performance calculation and/or air cooler selection.

These proprietary software packages are reasonably user friendly for the make of air coolers they are designed for. They are, however, generally not readily suitable for relative comparisons between alternative manufacturers.

Computer Modeling

The various comparisons and theoretical calculations presented in this paper are based on a proprietary, universal software package [1]. The package is based on first principles and is capable of predicting performances of most tube/plate fin extended surface type finned air coolers available on the market.

The accuracy of the software package has been the subject of practical tests in real industrial freezing and chilling plant applications over a period of close to two decades for a large variety of refrigerants, geometries and operating conditions.

Most comparisons presented in this paper are of relative nature. Any accuracy shortcomings that the software package employed may have in absolute terms will therefore to a large extent be cancelled out.

Refrigerant Properties

The transport properties of the refrigerant at the prevailing evaporating temperature within the evaporator coil determine the circuiting of the coil. For any given coil circuitry, the transport properties in turn determine the thermodynamic performances i.e. inside film coefficients and pressure drops.

The differences in transport properties between R717 (NH_3) and R744 (CO_2) at an evaporating temperature of -45°C are shown in table 1.

Refrigerant	NH_3	CO_2
Evaporating temperature, °C	-45.0	-45.0
Refrigerant liquid density, kg/m^3	696.17	1135.26
Refrigerant thermal conductivity in liquid phase, W/mK	0.6422	0.1656
Refrigerant dynamic viscosity in liquid phase, kg/ms	$0.3048*10^{-3}$	$0.2140*10^{-3}$
Refrigerant density vapour phase, kg/m^3	0.4865	21.74
Refrigerant latent heat, kJ/kg	1401.06	330.08
Refrigerant temperature/pressure gradient dT/dP, K/bar	32.62	2.08

Table 1. Properties of NH_3 and CO_2

The density of CO_2 is almost twice the density of NH_3 whereas the thermal conductivity of CO_2 in the liquid phase is around four times lower than for NH_3.

The latter is not an advantage in terms of heat transfer. In the case of overfeed systems, the inside film coefficient is directly proportional with the thermal conductivity of the refrigerant in the liquid phase.

The dynamic viscosity of CO_2 is around 30% lower than for NH_3. This has a favourable impact on the Reynolds number Re. An increased Reynolds number will increase the dimensionless measure for internal heat transfer or the Nußelt number Nu. The impact on the inside film coefficient α_i will therefore be an increase – the overall effect on coil performance will be an improvement.

The vapour density of CO_2 is around 45 times greater than for NH_3. The effect of this is that for a certain tube diameter and refrigerant mass flow, the vapour velocity for CO_2 will be many times lower than for NH_3. The result is that evaporators circuited for CO_2 generally feature smaller tubes and longer circuits than the equivalent evaporators circuited for NH_3.

A comparison of the ratios between the liquid and vapour densities of the two refrigerants is also interesting. These are around 1400 and 50 respectively for NH_3 and CO_2. The effect is that it is easier to obtain uniform distribution in liquid distributors and distributor leads with CO_2 as compared with NH_3. This is significant in dry expansion feed applications.

The latent heat of NH_3 is around four times greater than for CO_2. For the same refrigeration capacity, the circulated refrigerant mass flow in a CO_2 system will therefore be around four times higher than for the equivalent NH_3 plant. In this context it is worth noting that many synthetic refrigerants feature lower latent heat than CO_2.

Finally, the much lower temperature/pressure gradient of CO_2 is of significance in terms of coil circuiting. The inside film coefficient and pressure drop are related. The greater the turbulence, the higher the inside film coefficient and the greater the refrigerant side pressure drop.

In the case of ammonia, even reasonably small refrigerant pressure drops translate into significant changes in refrigerant temperature. Significant changes in refrigerant temperature as the refrigerant evaporates jeopardize the available logarithmic mean temperature difference LMTD and hence overall evaporator performance.

The properties of NH_3 when applied at low evaporating temperatures therefore necessitate employment of relatively large tubes and relatively short refrigerant circuits. The properties of CO_2 are in this context much more favourable. They enable the designer to achieve significantly improved evaporator performances provided the coil geometry and circuiting are selected to take advantage of the properties of the fluid.

Finned Air Cooler Geometry and Liquid Overfeed

The question facing the plant designer is which geometries and circuiting is optimal for a particular fluid and application. To a certain extent the evaporator manufacturers may be relied upon to make suitable recommendations.

However, manufacturers are likely to display bias towards geometries, which are available within their own range. It is in the interest of the plant designer to be able to research the entire range of coil geometries available to determine the optimal geometry and indeed pressure rating for the application in question.

In low temperature, liquid overfeed applications the designer will find that the range of potentially suitable geometries and circuit lengths when using refrigerant NH_3 is narrower than when using CO_2.

This may be illustrated by way of an example. Consider an evaporator coil for an automatic in-line blast freezer for freezing of beef in cartons. The capacity requirement per evaporator coil is around 140 kW. The air on/off temperatures are -35/-38°C respectively.

The required air quantity is given by the blast freezer geometry and has been determined at 30 m³/s per coil. Again dictated by the blast freezer layout, the required finned coil length is 5 to 6 m and the finned height around 1.5 m. This results in a face velocity in the range 3.3 to 4.0 m/s.

The coil geometry used for comparison purposes in table 2 is available from several evaporator coil manufacturers. In the case of both refrigerants, the coil geometries are identical – the difference in performance is a direct result of the difference in refrigerant properties.

Refrigerant	NH_3	CO_2
Location	Blast Freezer	Blast Freezer
Fin thickness, m	0.00035	0.00035
Vertical tube pitch, m	0.050	0.050
Tube outside diameter., m	0.015	0.015
Horizontal tube pitch, m	0.050	0.050
No. of tube rows per fin element	Two	Two
Thermal conductivity of fin material, W/mK	210	210
Tube inside diameter, m	0.0134	0.0134
Thermal conductivity of tube material, W/mK	18	18
Tube pattern	Square	Square
Air/refrigerant flow pattern	Cross	Cross
Air distribution	Free discharge	Free discharge
Air flow direction	Horizontal	Horizontal
Coil material	304 Stainless Steel tubes, Aluminium fins	304 Stainless Steel tubes, Aluminium fins
Refrigerant feed	Liquid recirculation	Liquid recirculation
Headers	2 off, horizontal	2 off, horizontal
Circuit orifices	Not included	Not included
Finned length, mm	5300	5300
Finned height, mm	1500	1500
Rows high	30	30

Finned depth, mm	600	600
Rows deep	12	12
No. of tube passes/circuits	10/36	10/36
Fin spacing, mm	10.0	10.0
PERFORMANCE DATA:		
Face velocity, m/s	3.77	3.77
Evaporating temperature at the wet return header, °C	-41.5	-41.5
Refrigerant liquid density, kg/m^3	691.93	1121.62
Refrigerant thermal conductivity in liquid phase, W/mK	0.6340	0.1611
Refrigerant dynamic viscosity in liquid phase, kg/ms	$0.2937*10^{-3}$	$0.2031*10^{-3}$
Refrigerant density vapour phase, kg/m^3	0.5974	24.746
Refrigerant latent heat, kJ/kg	1391.25	323.994
Refrigerant temperature/pressure gradient dT/dP, K/bar	27.83	1.81
Internal fouling resistance (oil), m^{2}K/W	0	0
External fouling resistance (frost), m^{2}K/W	0.0050	0.0050
Air on/off, °C	-35.0/-36.8	-35.0/-38.1
Mean air temperature, ºC	-35.9	-36.5
Relative humidity on/off, %	80/93	80/93
Circulation rate	2 to 1	2 to 1
Air flow per evaporator, m^3/s	30.0	30.0
Air side coil pressure drop lightly frosted, Pa	110	110
Total cooling capacity/unit, kW	81.1	140.3
Sensible cooling capacity/unit, kW	80.6	138.3
Refrigerant pressure drop, K	5.35	0.19
Refrigerant pressure drop, bar	0.19	0.104
α_i, W/m^{2}K	1633	2150
α_o, W/m^{2}K	48.2	48.5
Overall k-value clean, W/m^{2}K	35.2	37.7
Overall k-value service, W/m^{2}K	29.9	31.7
Safety margin for fouling, %	15.1	15.9
Primary unit surface area, m^2	86.8	86.8
Secondary unit surface area, m^2	856.6	856.6
Total surface area, m^2	943.4	943.4
Refrigerant volume, m^3	0.31	0.31

Table 2. Two identical evaporator coils with NH_3 and CO_2 refrigerants

It may be argued – and correctly so – that the evaporator coil employing refrigerant NH_3 is incorrectly circuited for the application. It is obvious that the main impediment to achieving optimal performance for the NH_3 coil is refrigerant pressure drop.

To improve performance of the particular NH_3 coil used in the example in table 2, it is therefore necessary to reduce circuit length i.e. reduce the number of refrigerant passes to a number less than the present value of ten. This may be achieved in a number of different ways.

The most common way is by introducing multiple, horizontal liquid inlet and wet return headers. This is convenient from the coil manufacturers point of view, but raises piping and valve station costs on the part of the plant installer. Provided the liquid inlet header is situated at the bottom of the coil, horizontal headers require no orifices for refrigerant distribution.

In the present example where the coil is 30 rows high, the introduction of multiple horizontal headers is not without difficulty. The simplest way is to reduce the number of passes from ten to six, refer fig 1.

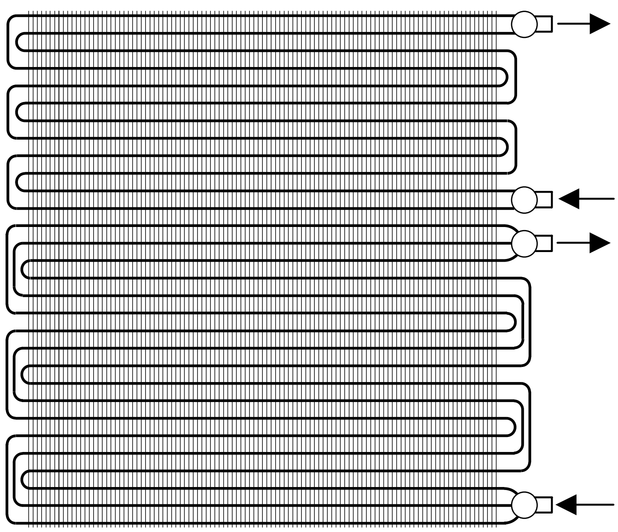


Fig. 1. Introduction of multiple headers for circuit length reduction

In simple terms, the evaporator block, following the introduction of a second set of headers, needs to be considered as two individual evaporator blocks. This has a number of consequences with respect to piping and valve station design not to mention the multiple oil drains, which will be required.

A reduction in the number of refrigerant passes from ten to six will halve the temperature drop. The corresponding overall coil performance improvement is ~25%, which is not insignificant, but the improved capacity remains around 23% less than the requirement of 140 kW.

It may be argued that this circuit length reduction can also be achieved without the introduction of multiple headers. This is correct, but only very few coil manufacturers – if any – would be in a position to offer headers feeding five rows and do so at a reasonable price.

Reducing the number of refrigerant passes further is possible by altering the header orientation from horizontal to vertical. This is not an uncommon measure, but this will require the introduction of individual circuit orifices in the liquid inlet headers to ensure uniform refrigerant distribution.

A coil as detailed in table 2, but featuring four refrigerant passes, vertical headers and physical counter flow between air and refrigerant is shown in fig. 2 in plan view. The corresponding additional performance improvement is approximately 8-9%, which is directly attributable to a further reduction in refrigerant pressure drop to around 1.3K.

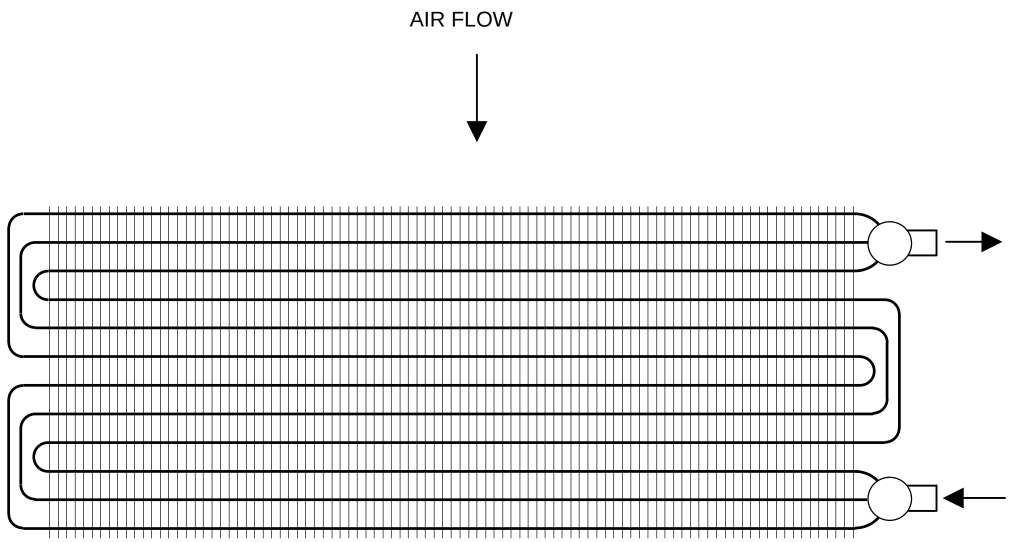


Fig. 2. Circuit length reduction by employing vertical headers

The diameter of the circuit orifices required for refrigerant distribution is in the order of 1.8 to 2.2 mm. The smallest orifices are at the lower end of the headers to compensate for the effect of gravity on the driving force of the liquid in the header.

Quite apart from the tendency of circuit orifices to be susceptible to blocking by impurities in the system, they may also extend the defrost time if hot gas defrost is used. Imperfections in orifice diameters often also necessitate greater circulation ratios in order to ensure complete utilization of the available coil surface area.

Without special measures with respect to coil design, the hot gas - or the condensate that forms during defrost - will need to pass through the orifices. The flow restrictions in the orifices and the corresponding non-uniform frost removal from the coil face, will often lead to longer defrost times.

Vertical headers enable the introduction of physical parallel flow. Due to the refrigerant pressure (temperature) drop, this translates into thermodynamic counter flow. In reality the coil arrangement will be as per fig. 2, but with the air flowing in the opposite direction.

The performance improvement in this case amounting to around 4 to 5% is a direct result of the improvement in LMTD.

None of the circuit length reduction measures introduced for the NH_3 coil will provide the necessary capacity of 140 kW. With limited possibilities of altering coil height and length in this case, the only option remaining is to increase coil depth and achieve design capacity this way.

Increasing coil depth to 14 rows as opposed to 12 rows is more or less adequate to achieve performance. The total tube number in the coil then increases to 30x14=420 off.

Retaining 4 refrigerant passes with an even number of distributions per circuit remains possible at 420/4=105 distributions in the headers. Circuit layout is not straightforward and many would elect to increase the depth to 16 rows for simplicity whilst retaining 4 passes. This would also provide a suitable safety margin for non-uniform refrigerant distribution – a common, but not very desirable feature of vertical headers.

The additional safety margin would also compensate for the presence of oil fouling internally within the evaporator. The combination of NH_3 and a non-miscible oil can have a significant impact on overall coil performance.

The lower the evaporating temperature, the greater the equilibrium oil film thickness on the inner surfaces of the evaporator tubes [2]. An oil film 0.05 mm in thickness will give rise to an internal fouling resistance of approximately 0.0004 m^2K/W. The relative effect of this on performance is shown in table 3.

The step-by-step measures required for the NH_3 coil to achieve the performance of the CO_2 evaporator are summarized in table 3. These measures are a direct result of the less favourable properties of refrigerant NH_3 as compared with CO_2 at this operating condition.

Refrigerant	NH_3	NH_3	NH_3	NH_3	NH_3	NH_3
Location	Blast Freezer	Blast Freezer	Blast Freezer	Blast Freezer	Blast Freezer	Blast Freezer
Air/refrigerant flow pattern	Cross	Cross	Cross/ counter	Cross/ parallel	Cross/ parallel	Cross/ parallel
Air distribution	Free discharge	Free discharge	Free discharge	Free discharge	Free discharge	Free discharge
Headers	2 off, horizontal	4 off, horizontal	2 off, vertical	2 off, vertical	2 off, vertical	2 off, vertical
Circuit orifices	Not included	Not included	Included	Included	Included	Included
Finned length, mm	5300	5300	5300	5300	5300	5300
Finned height, mm	1500	1500	1500	1500	1500	1500
Rows high	30	30	30	30	30	30
Finned depth, mm	600	600	600	600	700	800
Rows deep	12	12	12	12	14	16
No. of tube passes/circuits	10/36	6/60	4/90	4/90	4/105	4/120
PERFORMANCE DATA:						
Face velocity, m/s	3.77	3.77	3.77	3.77	3.77	3.77
Evaporating temp. at the wet return header, °C	-41.5	-41.5	-41.5	-41.5	-41.5	-41.5
Internal fouling resistance, m²K/W	0	0	0	0	0	0.0004
External fouling resistance, m²K/W	0.0050	0.0050	0.0050	0.0050	0.0050	0.0050
Air on/off, °C	-35.0/ -36.8	-35.0/ -37.4	-35.0/ -37.6	-35.0/ -37.7	-35.0 -38.1	-35.0/ -38.1
Relative humidity on/off, %	80/93	80/93	80/93	80/93	80/95	80/96
Circulation rate	2 to 1	2 to 1	3 to 1	3 to 1	3 to 1	3 to 1
Air side coil pressure drop lightly frosted, Pa	110	110	110	110	128	147
Total cooling capacity/unit, kW	81.1	108.3	118.2	123.7	138.8	139.3
Sensible cooling capacity/unit, kW	80.6	107.0	116.6	122.1	136.9	137.5
Refrigerant pressure drop, K	5.35	2.5	1.3	1.4	1.3	1.1
Refrigerant pressure drop, bar	0.19	0.090	0.048	0.052	0.048	0.038
α_i, W/m^{2}K	1633	1415	1544	1616	1554	1365
α_o, W/m^{2}K	48.2	48.4	48.4	48.4	48.7	48.9
k-value, clean, W/m²K	35.2	33.9	34.8	35.2	35.0	33.8
k-value, service, W/m²K	29.9	29.0	29.6	29.9	29.8	25.5
Surface area, m_2	943.4	943.4	943.4	943.4	1100.6	1257.8

Table 3. Performance improvement measures for NH_3 evaporator coil

The conclusions, which may be derived from table 3, are in summary:

- For the NH_3 air cooler circuited for a refrigerant side temperature drop of 1.1K, the inside film coefficient α_i is approximately 60% of the value estimated for the CO_2 air cooler circuited for a temperature drop of ~0.2K or one fifth of the NH_3 cooler temperature drop. This translates into a heat transmission advantage in favour of the CO_2 evaporator of 11% in clean condition. Further optimization of circuiting and geometry used for the CO_2 unit could increase this advantage further.

- The NH_3 air cooler, when selected with appropriate allowances for internal fouling resistances caused by oil and circuited with a refrigerant side temperature drop around 1K, requires a coil depth of 16 rows as opposed to the 12 rows required by the CO_2 evaporator. The difference in required surface areas amounts to approximately 25%.

- For identical refrigeration capacities and fouling allowances to suit the refrigerants employed, the air side pressure drop of the NH_3 air cooler is approximately 25% greater than the air side pressure drop of the CO_2 unit. Using the product of air flow and pressure drop and applying an arbitrary fan/motor efficiency of 0.6, this translates into an increase in absorbed fan power of ~1.9 kW per unit or ~1½ % of capacity.

- The internal fouling allowance of 0.0004 m^2K/W accounting for the presence of a 0.05 mm oil film on the internal tube surfaces of an NH_3 air cooler operating at this evaporating temperature reduces heat transmission by around 12%.

The CO_2 evaporator will require greater design pressures than the equivalent NH_3 air cooler. The manufacturer of the particular coil configuration used in the example above provides for this increase by altering the tube wall thickness from 0.6 to 0.8 mm.

An increase in tube and header material wall thickness will, all other things being equal, increase manufacturing costs of CO_2 evaporators. However, this cost increase is usually more or less offset by the savings associated with simplified circuitry.

Other Refrigerant Feed Methods

Common alternatives to liquid overfeed are gravity flooded feed and dry expansion.

The principle of gravity flooded refrigerant feed is shown in fig. 3.

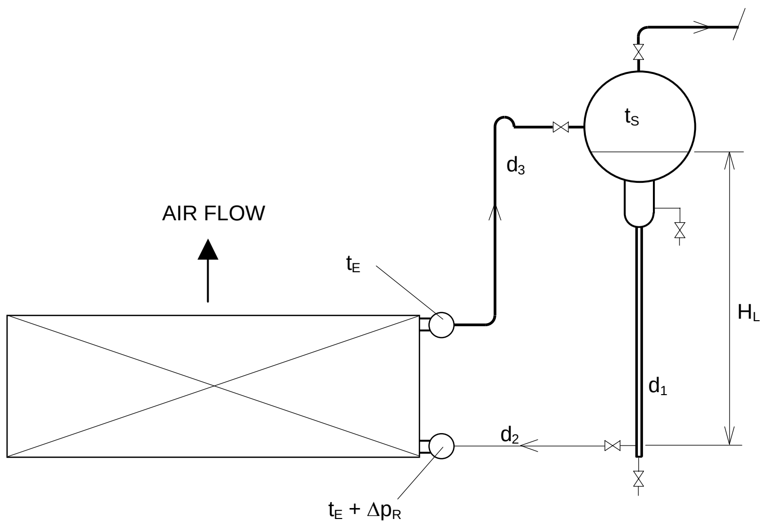


Fig. 3. Evaporator with gravity flooded refrigerant feed.

Irrespective of the refrigerant used, the calculation principles are the same. In equilibrium, the sum of the pressure gains and the pressure drops in the refrigerant circuit must be zero.

In table 4 are shown the key data for an NH_3 and a CO_2 gravity flooded evaporator. The evaporator configuration is identical to the 4-pass unit used in table 2 except for the orientation. The airflow is now vertically up as opposed to horizontal.

Refrigerant	NH_3	CO_2
t_E, °C	-41.5	-41.5
t_S, °C	-42.4	-41.8
$t_E + \Delta p_R$, °C	-39.7	-41.46
Air on/off, °C	-35.0/-37.8	-35.0/-38.2
Relative humidity on/off, %	80/93	80/93
Circulation rate	5.2 to 1	5.6 to 1
Total cooling capacity, kW	124.8	143.2
Overall heat transmission coefficient service, W/m²K	31.6	32.1
d_1, m	0.10226	0.0779
d_2, m	0.0779	0.0627
d_3, m	0.1282	0.0779

Table 4. Key data for NH_3 and CO_2 flooded evaporators

In both cases, the evaporator coils and surge drum heights are identical and the interconnecting refrigerant pipe line sizes are selected in accordance with good design practice. The differences in saturation temperatures shown are therefore more or less attributable to the differences in refrigerant properties.

The impact of the difference in temperature/pressure gradients between refrigerants NH_3 and CO_2 is very evident in table 4. Relatively small changes in pressure for NH_3 cause much greater changes in saturation temperature than is the case for CO_2.

The practical consequence of the greater temperature changes of NH_3 is lower LMTD across the NH_3 evaporator than across the CO_2 unit. All other things being equal this translates into a performance reduction.

Dry expansion refrigerant feed is generally quite common - presumably due to simplicity and lower capital cost compared with other methods such as liquid overfeed or gravity flooded feed.

Some characteristics of dry expansion refrigerant feed are, however, less favourable than the characteristics of liquid overfeed. These less favourable characteristics become evident at part-load of the evaporator.

Evaporator part-load is a likely occurrence in many industrial and commercial refrigeration applications. Reduced coil face velocity, reduced entering temperature difference and external frost accumulation are common reasons for the evaporator performing less than design capacity.

Varying the evaporator face velocity in a controlled manner by employing either dual speed fans or variable frequency drives makes good technical sense. The main reasons for implementing this feature in refrigeration plant control system are energy efficiency and noise control.

Varying the entering temperature difference (air entry minus evaporating temperature) is a common method of capacity control. The air entry temperature remains constant for varying evaporating temperature.

A comparison between dry expansion feed evaporators and liquid overfeed evaporators is provided in fig. 4 and fig. 5 for NH_3 and CO_2 respectively. The comparisons show the cooling capacity of typical coils at varying loads. For both NH_3 and CO_2 the coil surface areas and geometry are identical, but the number of refrigerant circuits is selected to suit the fluid used. For the DX units, the air/refrigerant flow pattern is counter flow. The NH_3 overfeed coil employs vertical headers and parallel flow, the CO_2 overfeed evaporator is fitted with horizontal headers and the air/refrigerant flow pattern is cross flow.

The evaporator circuited for CO_2 refrigerant generally provides higher cooling capacities and has a wider operating range than the equivalent NH_3 coil. This is mainly due to the reduced impact of refrigerant side pressure drop on the available LMTD across the CO_2 heat exchanger.

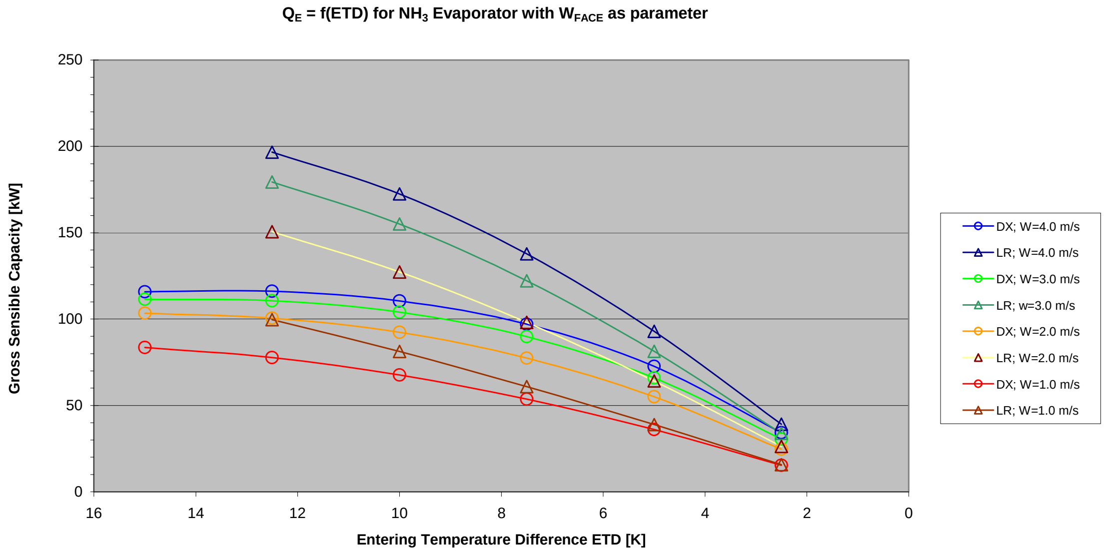


Fig. 4. Capacity for NH_3 evaporator as a function of entering temperature difference and face velocity. Dry expansion (DX) and liquid overfeed (LR), air on constant at -35.0°C.

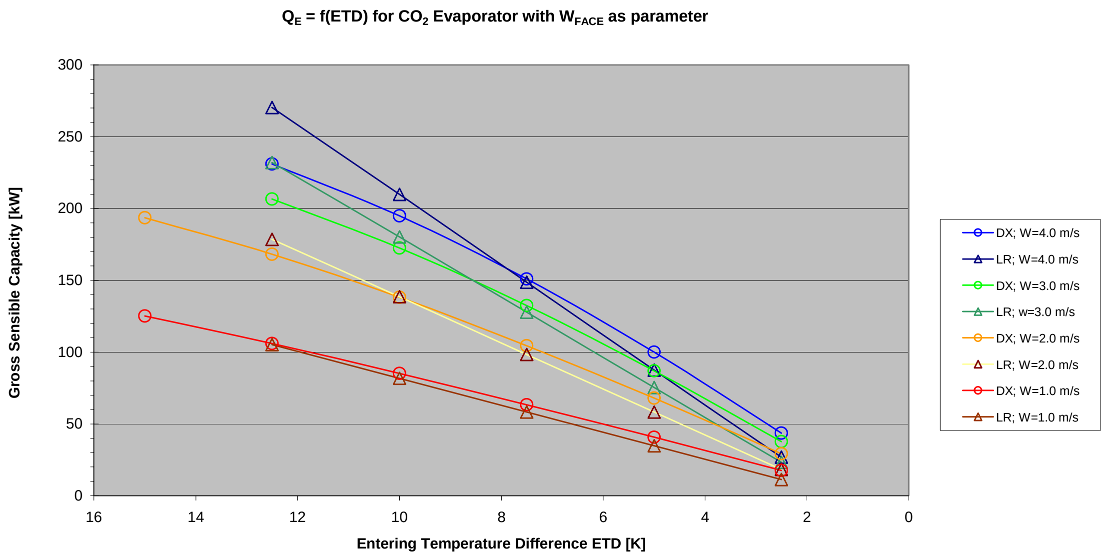


Fig. 5. Capacity for CO_2 evaporator as a function of entering temperature difference and face velocity. Dry expansion (DX) and liquid overfeed (LR), air on constant at -35.0°C..

Another problem with NH_3, which is not evident in fig. 4, is the difficulty associated with uniform refrigerant distribution. Large latent heat combined with a relatively much larger ratio between liquid and vapour densities for NH_3

as compared with CO_2 creates practical difficulties with respect to uniform distribution in liquid distributor and distributor leads.

Air to Refrigerant Flow Pattern

Cross flow, parallel flow and counter flow between air and refrigerant have been discussed in the preceding sections.

Another circuiting method particularly relevant to dry expansion feed when using a refrigerant such as NH_3 displaying a relatively large temperature versus pressure gradient, is reversed suction return. This circuiting method is shown in plan view in fig. 6.

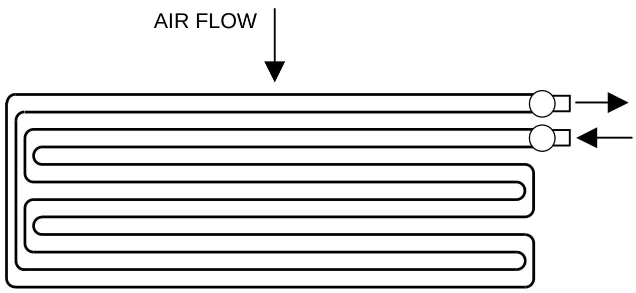


Fig. 6. Reversed suction return

The advantages of this circuiting method are greater LMTD than for counter flow and improved superheat signal for controlling the refrigerant injection.

The LMTD advantage increases with increasing refrigerant side pressure drop. The circuiting method is therefore less relevant when using CO_2 than when using NH_3 because the refrigerant side temperature drops in CO_2 applications are several times lower than for NH_3.

Finned Air Cooler Materials and Pressures

Material compatibility with CO_2 is better than with NH_3 – only ferrous materials can be used with the latter whereas copper is not compatible. With respect to CO_2, this widens the range of coil geometries, which may be used also for low temperature duties.

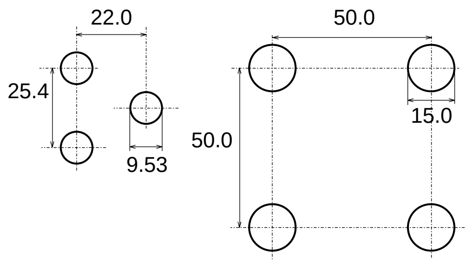
Most evaporator manufacturers will have few difficulties achieving maximum operating pressures around 28 bar with standard copper/aluminium coil designs with copper headers.

A maximum operating pressure increase for a CO_2 coil to around 40 bar as dictated by operating conditions and defrost method usually requires the use of stainless steel headers. Most manufacturers would still consider it acceptable to manufacture the tubes from copper at this pressure.

Further increases in maximum operating pressure to around 52 bar as required in hot gas defrost applications with CO_2 would, according to most manufacturers, require the use of stainless steel tubes and headers within the evaporator.

A challenge to evaporator manufacturers is the application of CO_2 in evaporators with small tube diameters. These evaporator geometries have so far been limited to smaller commercial applications using HFC refrigerants.

As shown in table 5, smaller tube diameters may be employed to greatly reduce evaporator coil dimensions, operating mass and refrigerant volume in CO_2 liquid overfeed applications as compared with the types of evaporators commonly available at present.



Air/refrigerant flow pattern	Cross/parallel	Cross
Coil material	Copper tubes, aluminium fins	304 Stainless Steel tubes, aluminium fins
Refrigerant feed	Liquid recirculation	Liquid recirculation
Headers	2 off, vertical	2 off, horizontal
Finned length, mm	5217	5300
Finned height, mm	1524	1500
Finned depth, mm	352	600
Fin spacing, mm	10.0	10.0
Evaporating temperature at the wet return header, °C	-41.5	-41.5
External fouling resistance (frost), m^2K/W	0.0050	0.0050
Air on/off, °C	-35.0/-38.2	-35.0/-38.1
Air flow per evaporator, m^3/s	30.0	30.0
Total cooling capacity/unit, kW	144.4	140.3

Refrigerant pressure drop, K	0.3	0.2
Overall heat transmission coefficient service, W/m²K	57.0	31.7
Primary unit surface area, m²	146.2	86.8
Total surface area, m²	555.8	943.4
Refrigerant volume, m³	0.26	0.31
Block volume (LxWxH), m³	2.80	4.77

Table 5. Impact of coil geometry on overall coil dimensions, k-value and volume

Conclusion

Refrigerant CO_2 is an excellent refrigerant particularly for low temperature applications. Despite the lower thermal conductivity of CO_2 compared with NH_3, evaporators designed for common blast freezer applications and liquid overfeed of CO_2 require around 25% less heat exchanger surface area compared with NH_3 evaporators for identical applications.

The reduction in required heat transfer area translates into reduced air pressure drop. For blast freezer applications, this will in most practical situations enable the fan motors to be reduced one size. Except in extreme low temperature applications, use of CO_2 will eliminate internal oil fouling in evaporators. In low temperature applications using NH_3 with non-miscible oil, internal oil fouling is a common problem causing cooling capacity loss.

In gravity flooded refrigerant feed applications, the more favourable temperature versus pressure relationship of CO_2 compared with NH_3 reduces the temperature changes in the thermosyphon loop. The performance improvement for a typical industrial application as a result of this is 10 to 12%.

The practical operating range of dry expansion evaporators employing CO_2 is significantly wider than the equivalent air cooler using NH_3. The practical benefits of this are reduced risk of liquid flood-back to the compressor at part load, improved suitability for fan speed control and greater energy efficiency.

Most evaporator manufacturers have retained the evaporator geometries developed for traditional refrigerants for use in CO_2 applications and simply elevated the operating pressures. Development of evaporator geometries specifically for CO_2 refrigerant remains a challenge. The potential rewards associated with the use in CO_2 air coolers of small-bore geometries so far reserved for commercial HFC applications are substantial. Increases in heat transmission coefficient by around 45% and corresponding reductions in evaporator dimensions and weights appear achievable.

References:

[1] "Extended Surface Steel Air Coolers for Industrial Refrigeration." S.S. Jensen. AIRAH Journal, August 1989.

[2] Energy Savings in Ammonia Refrigeration Plant by Using Oil Scrubbers. IR. G.J. Koster, Grenco, s'Hertogenbosch, Netherlands. Proceedings Institute of Refrigeration 1984-85. 4.1