Extended surface steel air coolers for industrial refrigeration

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This paper explains the broad principles of a compact computer simulation program which will predict performances of extended surface steel air coolers of any geometry. The versatility of the software is demonstrated in a number of examples. These include a method of comparing costs of air coolers, the effect of circuiting on the performance of a direct expansion feed cooler, the influence of selection of refrigerant and refrigerant feed, various air/refrigerant flow patterns, design for humidity and the effect of oil fouling.

1. Introduction

In industrial refrigeration applications as in abattoirs, cold stores, fish and chicken processing plants etc., the use of finned air coolers manufactured from steel and hot dipped galvanised after manufacture is very common. They are used for a range of refrigerants including CFCs, various brines, chilled water and ammonia and the refrigerants are fed through the finned air coolers in a number of different ways.

It has been common practice in practical refrigeration engineering to relate required cooler surface areas to the nature of the application without considering differences in air cooler geometries and layout. The assumption that the heat transmission coefficients of all types of coolers remain constant at full load as well as part load is not unusual either, but may nevertheless be quite wrong.

In many situations, it is necessary to view the air cooler as the air/refrigerant heat exchanger it is and consider the exact influences of refrigerant, circuiting, feed, geometry etc. on cooler performance. Such analyses based on first principles of heat transfer and heat transmission are best carried out using a computer.

This paper describes the broad principles of a computer program which will calculate the cooling capacity of any air cooler manufactured from a tube bundle provided with plate fins. The program is small enough to be contained in a portable hand-held calculator and does consequently not boast an exceptionally high degree of accuracy. It is, however, a very powerful tool for the practical refrigeration engineer and can be used for assessing capacities of existing air coolers, selection of new air coolers, evaluation of tenders on an equal basis, optimisation of refrigerant circuiting and a number of other things. The versatility is demonstrated by a series of examples in this paper.

2. The Model

The principle of the model used for the evaluation of cooler performance is shown in the psychrometric chart Fig. 1. (12).

The temperature T1 and T2 represent before and after the cooler respectively — the same applies for the absolute humidity X. The temperature T3 is the evaporating temperature (average temperature inside the cooler tubes) and T4 is the average outside surface temperature of the cooler. The slope of the line X1, T1 — X2, T2 (or line X1, T4 — X4, T4) is determined by iteration considering inside and outside film coefficients, ratio between primary and secondary surface areas, fin efficiencies, fin material, etc.

The overall outside film coefficient is determined based on the principles described in (4, 8, 10) and is of the following final form

A4 = A0*F1/F0+A3*E0*F2/F0 (1)

where

A4	apparent overall outside film coefficient	W/m^2K
A0	tube film coefficient	W/m^2K
F1	tube surface area	m^2
F0	total cooler surface area	m^2

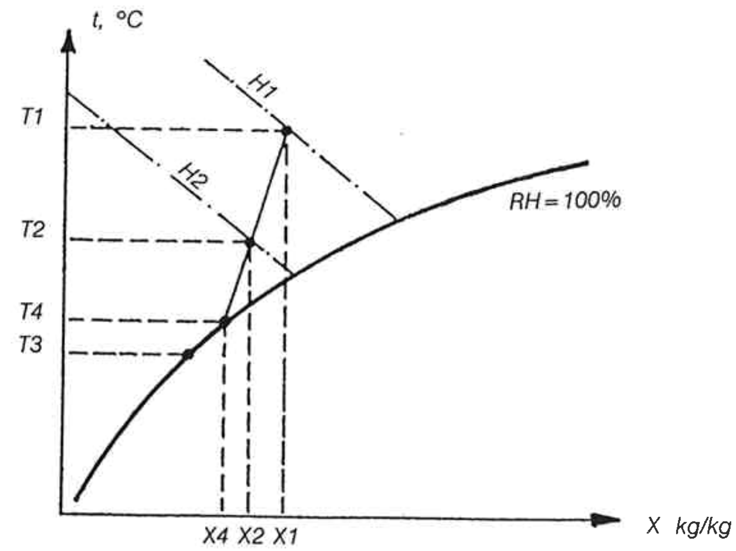

Fig. 1. Process in the psychrometric chart.

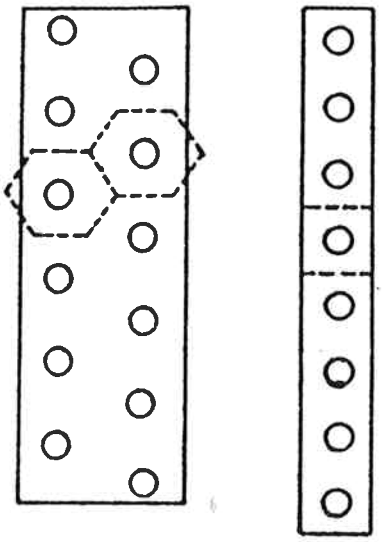

Fig. 2. Typical fin shape for the evaluation of fin efficiency.

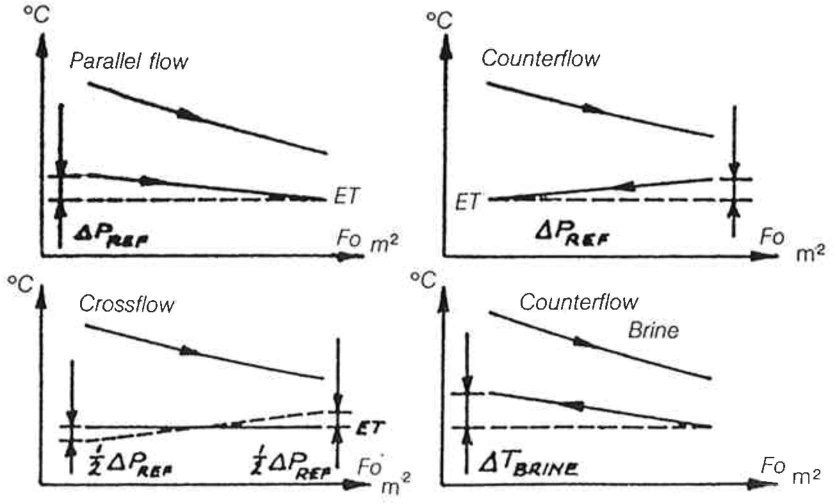

Fig. 3. Logarithmic mean temperature differences for Different Air/Refrigerant Flow Patterns.

A3	fin film coefficient	W/m²K
E0	fin efficiency	
F2	finned surface area	m²

The correction of the outside coefficient for "wet" operation is of the form

$$A3 = A2/S0 \quad (2)$$

where

A3	"wet" fin film coefficient	W/m²K
A2	"dry" fin film coefficient	W/m²K
S0	sensible/total heat ratio	

The fin efficiency of the plate fins is determined in accordance with the principles described in (8) and in Fig. 2 below.

With tubes arranged in triangular pattern as shown in Fig. 2, the fin efficiency is approximated by assuming a fin of hexagonal shape so that it will "fit" the standard equations described in the literature. The differences in local fin transfer coefficients are ignored which in most practical situations gives satisfactory results (2).

The model handles all refrigerants and brines, but requires the user to type in the necessary thermophysical and transport properties. A subroutine for calculation of thermodynamic and transport properties of humid air is part of the computer model (11). Refrigerant feed can be either forced liquid recirculation (incomplete evaporation), direct expansion (complete evaporation) or single phase forced flow as in the case of brine. Three different flow patterns may be selected namely parallel flow, counterflow or crossflow.

A detailed description of the correlations used for estimation of inside film coefficients and refrigerant pressure drops is beyond the scope of this paper, literature references are provided (7, 9).

The heat transfer correlations for incomplete and complete evaporation of refrigerants inside the tubes are of the general form

$$N1 = C*RE^{a}*K^{b} \quad (3)$$

where

N1	Nusselt number	-
C,a,b	Constants	-
RE	Reynold's number	-
K	"Boiling" factor	-

For turbulent flow of brines, the heat transfer correlation is of the general form

$$N1 = C*RE^{a}*PR^{b} \quad (4)$$

where

PR Prandtl number

and other symbols as in equation (3). Corrections for laminar flow are included.

For operation with refrigerants, three different air/refrigerant flow patterns may be selected. For operation with brine, only counterflow is used. The principle of calculating logarithmic mean temperature differences is indicated in Fig. 3.

All of the above methods of estimating LMTDs are approximations. The exact calculation of temperature differences is complicated by the fact that most finned air coolers are of the cross-parallel flow, cross-counterflow or 1-2 counterflow type.

However, when operation with relatively low refrigerant pressure drops (or relatively low brine temperature difference), the errors are relatively small.

The calculation of the overall heat transmission coefficient is of the general form:

$$K6 = 1/((1/A5+F6+(D4-D6)/2/L4*F3/F1)*F0/F1+1/A4) \quad (5)$$

where

K6	Overall heat transmission coefficient	W/m²K
A5	Inside film coefficient	W/m²K
F6	Inside fouling resistance	m²K/W
D4	Outside tube diameter	m
D6	Inside tube diameter	m
L4	Thermal conductivity of tube	W/mK
F3	Inside surface area of tube	m²
F1	Otuside surface area of tube between the fins	m²
F0	Total outside cooler surface area	m²
A4	Apparent overal outside film coefficient	W/m²K

3. Various Air Cooler Geometries And Materials

Various manufacturers in Australia and overseas produce finned air coolers manufactured from hot dipped galvanised steel. Finned air coolers for ammonia with tubes and fins manufactured from aluminium are to the knowledge of the author only produced outside Australia.

There are nearly as many cooler geometries as there are cooler manufacturers. Appendix 1 summarises geometric and detailed performance

Cooler	A	B	C	D	E	F
Heat transmission coeff. W/m²K	35.7	30.6	40.8	28.0	19.9	20.7
Fin efficiency, %	49.7	52.3	66.7	46.7	36.5	39.8
Ratio sec./prim. surf. areas	8.24	11.9	10.2	12.3	16.6	16.9
Relative cooling capacity, %	93.1	85.7	100	77.1	61.5	63.1

Table 1. Performance comparison between air coolers of different geometries and materials.

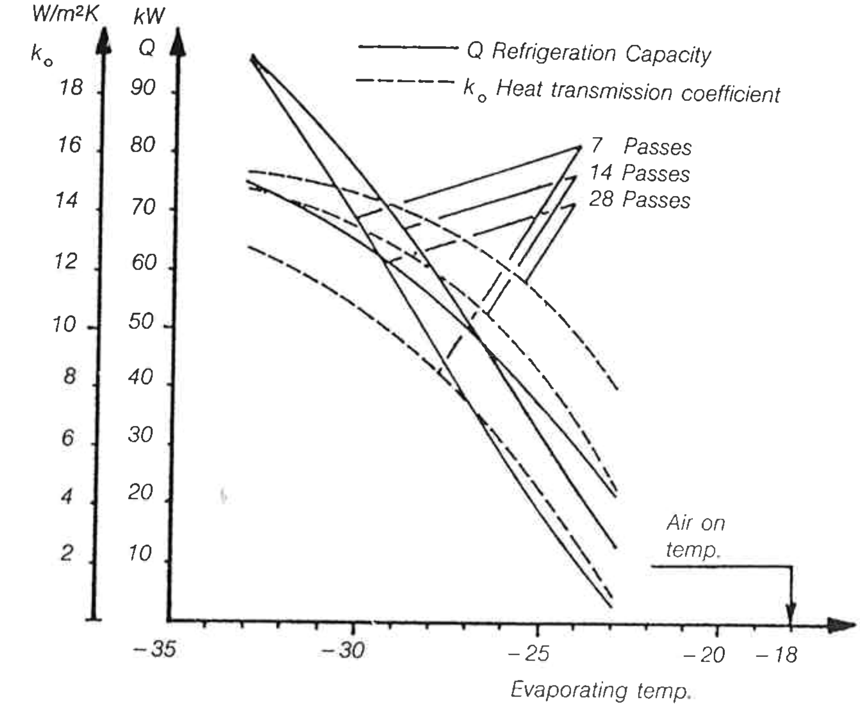

Fig. 4. Effects of circuiting on cooler performance.

data of a selection of different air coolers from Australian and overseas manufacturers. In this performance comparison, the refrigerant is ammonia, the evaporating temperature is -40°C and the mass flow density inside the tubes has been selected to provide identical inside film coefficients of 2000 W/m²K for all coolers. The differences in performance are a direct result of the differences in geometries and materials. Extracts from the performance comparison are shown in Table 1.

Cooler C is manufactured from aluminium whereas all other units are manufactured from mild steel, hot dip galvanised. Coolers D and F have rectangular tube pattern, the others have triangular tube arrangement. A variation of a factor of 1.6 in performance is evident over this range of units. It is obvious that the good thermal conductivity of aluminium as compared with galvanised mild steel has a very positive influence on cooler performance. The correlation between high ratio of secondary to primary surface (finned area to tube area) and cooler performance is also very clear — compare coolers A and E.

The above comparison shows that selection of coolers as "cooler surface area per carcass", "cooler surface area per m³ cold store capacity" or "cooler surface area per carton of meat to be frozen" can lead to greatly varying plant performances and should be used only as rough estimates.

It follows from the above that comparison of cooler cost on the basis "cost per unit area of cooler surface" can be misleading. This is demonstrated in Table 2. Here the term "cost per kW cooling capacity over LMTD" is used ($/(kW/K)). This is calculated by first dividing the cooling capacity of the cooler by the logarithmic mean temp. difference (LMTD) across the cooler considering the refrigerant side pressure drop. The cost of the cooler is then divided by the result of the first calculation.

Cooler	I	II
Cost, A$/m²	15.54	11.71
Cooling cap./LMTD, kW/K	20.52	17.03
Cost, A$/(kW/K)	482	669

Table 2. Cooler cost comparison

Evaluation on the basis of cost per unit surface area would be in favour of II whereas the more correct evaluation on the basis of $/(kW/K) favours cooler I.

4. Special Practical Considerations

Circuiting

The inside film coefficient in the cooler is often by designers considered to have negligible influence on cooler performance. This is true in some cases, but in the cases of for example direct expansion and in flooded (gravity fed) low temperature coolers, circuiting of the units must be considered very carefully. Incorrect circuiting (incorrect no. of circuits and circuit lengths) can cause dramatically reduced performance due to low inside film coefficients or insufficient "wetting" of the inner surfaces. This is demonstrated in Fig. 4 which shows performance data for a mild steel cooler 16 rows high and 7 rows deep with an outside surface area of approx. 600m². The cooler is operating with direct expansion of refrigerant R22. With a total of 7 × 16 = 112 tubes, the number of tube passes may be selected 112, 56, 28, 14, 7, 4, 2 or 1. Some of these are not practical and Fig. 3 shows cooler performances with 28, 14 and 7 passes corresponding to 4, 8 and 16 circuits respectively.

The optimum circuiting depends on the evaporating temperature the cooler will operate at most of the time. At low evaporating temperatures (high temperature differences), the choice is between 7 and 14 passes. At low temperature differences, 28 passes should be selected.

When determining optimum circuiting, it is also necessary to consider operation at low loads (low temperature differences). In low load situations it benefits plant operating economy to retain an efficiently operating cooler.

Finally Fig. 4 clearly shows that despite what is often assumed by designers, the heat transmission coefficient K_o is far from being constant over the range of temperature differences shown.

Refrigerant

The selection of refrigerant for a system depends on a variety of factors ranging from personal preferences of the engineer over physiological considerations to availability of the fluid at the actual location. It is outside the scope of this paper to address all of these factors — only the effect of refrigerant selection on the performance of coolers will be discussed.

Most manufacturers of finned air coolers apply correction factors to rate the units for different refrigerants. The coolers can for example be rated for refrigerant ammonia, liquid recirculation. For other refrigerants and feeds, the capacities are then reduced by a factor which is usually constant over a range of operating conditions. However, in reality there is for every cooler only a limited number of options for circuiting available. Practical considerations such as header location, oil return, distributor arrangement, flow pattern and location of expansion valve sensor will often limit the possibilities of the designer to optimise and in many cases he will have to compromise. In these cases it is a more accurate approach to return to basics and calculate the cooler from start as described in the beginning of this paper rather than applying approximate average correction factors. In the cases of more unusual refrigerants or operating conditions, the above approach becomes a necessity because manufacturers do generally not publish design data for these situations.

In Table 3 is shown how the coolers detailed in Appendix 1 would perform with different refrigerants and refrigerant feeds. The comparison is made for -40°C evaporating temperature and -32°C air on temperature. For direct expansion feed, the liquid inlet temperature has been taken at 0°C. In the case of all coolers, the practical circuiting possibility which provides the highest capacity has been used. However, some of the coolers only have a comparatively limited number of practical circuiting possibilities and are therefore somewhat disadvantaged, but this is also the way it is in practice. For this reason the relative capacities in Table 3 should not be used generally. With refrigerants which are not highly suitable for low temperature, the relative cooler capacity is quite low (R12 and R500). At higher evaporating temperatures, the relative performances would be different. Coolers with large bore tubes appear to be somewhat disadvantaged in direct expansion feed applications.

It is shown very clearly in Table 3 that ammonia is a superior refrigerant from a cooler performance point of view. This is an important consideration on the part of the plant designer. He/she must realise that the required cooler surface area may vary by a factor of more than two depending on the refrigerant and refrigerant feed specified. This naturally also has an effect on the capital cost of the plant.

Cooler		A	B	C	D	E	F
Refrigerant	Feed						
NH_3	LR	100	100	100	100	100	100
	DX	90	86	87	90	87	84
R22	LR	84	76	80	80	78	78
	DX	59	51	53	55	49	49
R12	LR	64	55	57	62	55	55
	DX	37	29	30	32	27	26
R13B1	LR	84	78	82	86	79	79
	DX	62	52	57	59	47	47
R500	LR	71	62	65	70	63	63
	DX	44	35	37	39	33	33

Table 3. Relative cooler capacities in per cent with various refrigerants and refrigerant feeds. LR = liquid recirculation, recirculation rate 4. DX = direct expansion. Clean coils, no fouling, no frost.

Refrigerant feed

The relative differences in performance between liquid recirculation and direct expansion are indicated in Table 3 above. Another very commonly used refrigerant feed method is gravity flooded based on the thermosyphon principle, see Fig. 5. The equilibrium circulation rate is determined by the driving force (drum height "H") and the pressure drops in the circuit. Again correct circuiting is very important, but also correct drum height. Putting the drum at high level ensures good recirculation rates, but also high refrigerant pressure drops which penalise performance. With the drum height too low, the recirculation rate becomes too low and some circuits of the cooler may boil dry. Lowest possible air temperature from the cooler for a given saturated surge drum temperature must be the goal. Fig. 6 illustrates the calculated performance as a function of drum height "H" of a 460 m² finned air cooler for a freezer. Refrigerant is R22, the saturated surge drum temperature -43°C, refrigeration capacity 115 kW and two different circuiting methods with 6 and 8 tube passes are shown. If a minimum circulation rate of 2.5 is considered necessary to avoid that any of the circuits boil dry, circuiting with 6 passes will clearly provide the lowest air temperature.

Air/refrigerant flow pattern

In heat exchangers with no change of phase, counterflow normally provides the highest logarithmic mean temperature difference and consequently highest capacity. In finned air coolers with boiling refrigerants, however, physical parallel flow translates into thermodynamic counterflow. The benefits or parallel flow in terms of capacity gain compared with physical counterflow or crossflow are greater the greater the refrigerant side pressure drop, see also Fig. 7. This illustrates the performances of a large 720 m² finned air cooler with parallel flow and counterflow, evaporating temp. -50°C and refrigerant ammonia, liquid recirculation.

In direct expansion feed air coolers, physical parallel flow creates some difficulties in obtaining a good superheat signal for the expansion valve.

The situation can be improved by placing the last one or two tube passes in the relatively warm inlet air stream of the cooler.

Air/refrigerant crossflow is shown in Fig. 8. In coolers of this type, the heat flux on the tubes placed in the relatively warm inlet air stream is higher than on the tubes in the outlet stream. This type of circuiting is unsuitable for direct expansion feed with a single conventional thermostatic expansion valve because the high flux circuits will receive insufficient refrigerant supply. Liquid recirculation feed coolers of this type may suffer similar problems if they are operating with relatively high diffusion (high air temp. drop across the cooler). In these situations it is either necessary to provide orifices in the inlet to each circuit or circulate sufficient

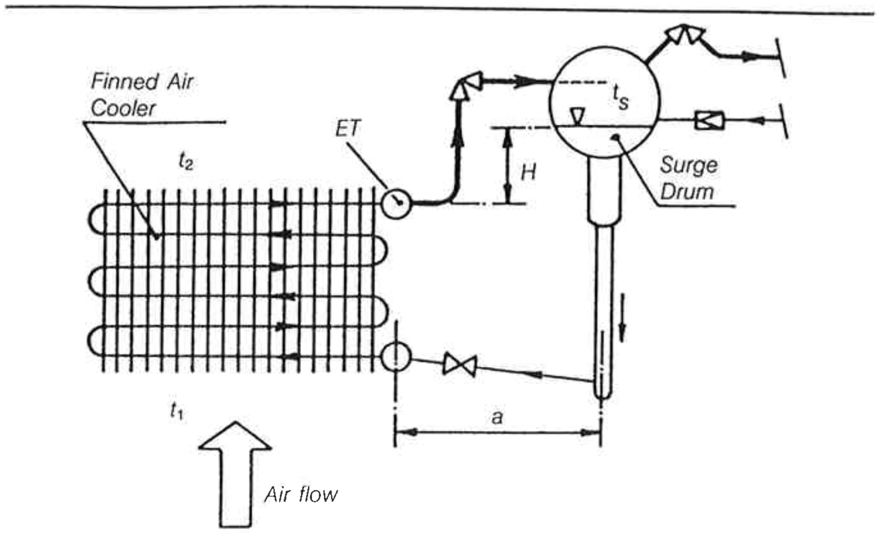

Fig. 5. Gravity flooded cooler.

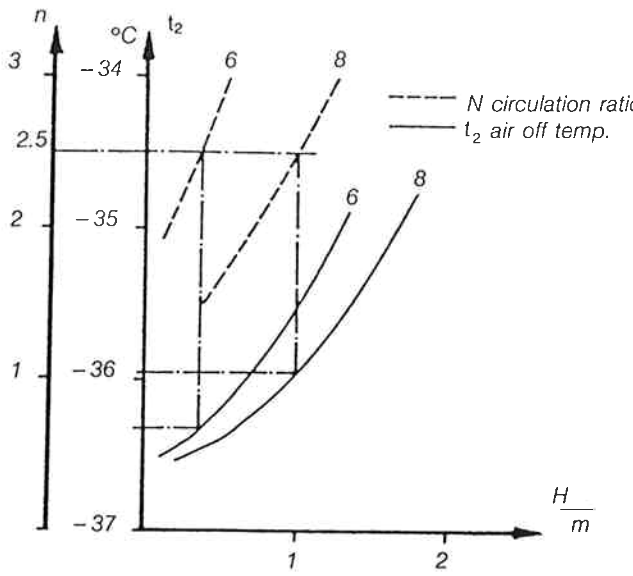

Fig. 6. Performance of a gravity flooded freezer evaporator as a function of surge drum height "H" with two different circuiting methods.

refrigerant to wet all circuits internally. Fig. 9 shows the estimated equilibrium differences in refrigerant flow through the circuits of a crossflow liquid recirculation cooler used for dehumidification of ambient air. The assumption is simply that the driving force across each circuit is identical and equal to the pressure difference between bottom supply header and top return header. In this case it is estimated that to maintain a minimum recirculation rate of 1.2 in the front circuit, the average recirculation rate without orifices must be a minimum of about 5 to 1. Lower average recirculation rates are likely to cause the front circuit to boil dry and thus reduce overall performance.

Design for humidity

Very often the quality of foods cooled and stored in refrigerated spaces is directly influenced by the relative humidity maintained in the air surrounding the product. To select a cooler to suit a room condition, the engineer must first estimate a room heat load including the estimated ratio between total and sensible heat. Then a cooler must be selected which is capable of removing this heat load at the desired total/sensible heat ratio. Some manufacturers of galvanised steel finned air coolers are reluctant or incapable of stating the total/sensible heat ratios which their coolers will do. This, of course, makes it difficult for the engineer to select the right equipment.

The estimated effect of cooler geometry on the total/sensible heat ratio is illustrated in Table 4. The coolers A to F are identical to those described in Appendix 1. They are all operating at an inlet air temperature of 10°C, a total/sensible heat ratio of 1.44 and a total cooling capacity of 58.3 kW. Refrigerant is ammonia, liquid recirculation at a rate of 4 to 1. The evaporating temperatures and inlet equilibrium relative air humidities vary as shown. In this comparison it is important to note that all coolers have approx. identical total surface areas, identical fin spacing, approx. identical face areas, air flows and finned depths. The differences in equilibrium relative humidities are a direct result of differences in geometry. The humidity differences shown may appear insignificant, but will affect product quality in vegetable storage applications and product weight loss in carcass chilling situations.

Also, two coolers of identical basic geometries, but with different general layouts may have large differences in equilibrium relative humidities of the inlet air. Normally, deep coolers with small face areas are cheaper than shallow coolers with large face areas. However, the differences in equilibrium relative humidities may be substantial, see Table 5. This shows a comparison between two coolers of identical basic geometries. Cooler A is 16 rows deep whereas cooler B is only 4 rows deep, but has 4 times the face area of cooler A. The two coolers consequently have identical surface areas.

The decision which cooler to use depends on the product to be stored/chilled and how much the additional weight loss or product deterioration which will be caused by cooler A compared with cooler B will cost over the life of the plant.

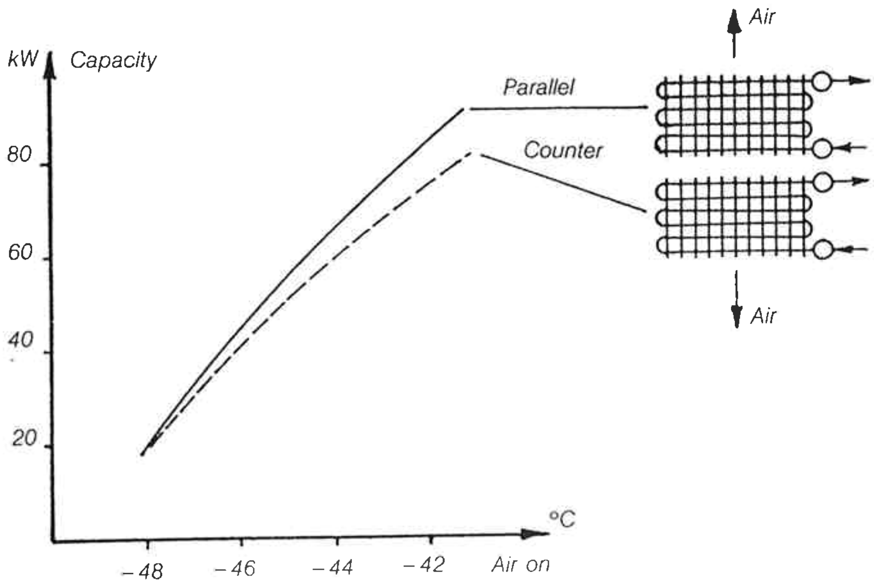

Fig. 7. Comparison of performances of a large finned air cooler with air/refrigerant in parallel flow and counterflow.

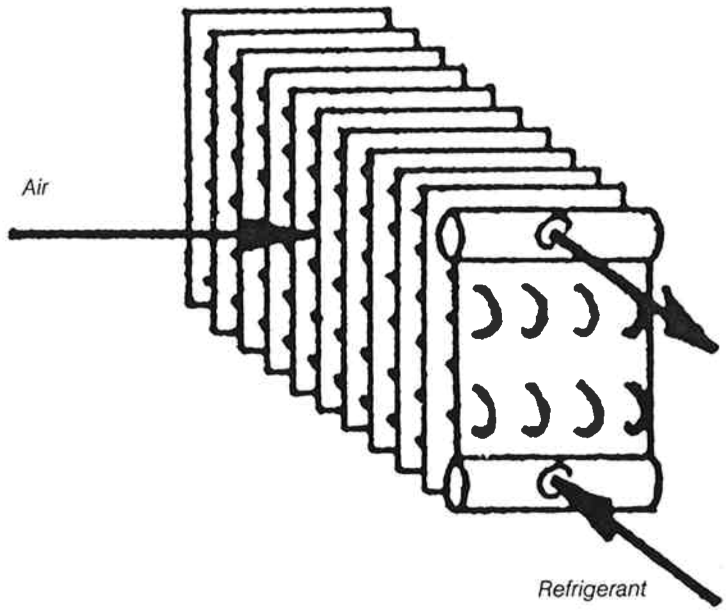

Fig. 8. Air/refrigerant crossflow.

Fouling

Fouling resistances are additional ther-

Cooler	A	B	C	D	E	F
Total/sens. heat ratio	1.44	1.44	1.44	1.44	1.44	1.44
Total cooling capacity, kW	58.3	58.3	58.3	58.3	58.3	58.3
Evap. temp. °C	3.0	2.3	3.5	2.5	0.4	0.6
Equilibrium rel. air humidity at cooler inlet, %	80.0	79.3	81.6	79.0	75.1	75.5
Air inlet temp., °C	10.0	10.0	10.0	10.0	10.0	10.0

Table 4. Estimated equilibrium relative air humidities for various cooler geometries.

mal resistances which can be located on all the heat exchange surfaces of a heat exchanger and so reduce the heat transmission coefficients. In finned air coolers they may occur on both the inside

Cooler	A	B
Total/sens. heat ratio	1.187	1.187
Total cooling capacity, kW	24.4	24.4
Evap. temp. °C	0	2.0
Equilibrium relative air humidity at cooler inlet, %	80.0	88.1
Air inlet temp. °C	5.0	5.0
Air flow, m^3/s	4.1	16.3

Table 5. Comparison between two coolers of identical basic geometries. Cooler A is 16 rows deep, cooler B 4 rows deep. Cooler B has 4 times the face area of cooler A. Both coolers have identical surface areas.

tube surfaces and on the fins.

In coolers using ammonia refrigerant, inside fouling is usually in the form of an oil film. Ammonia is an organic fluid and does not readily mix with oil. Particularly in low temperature ammonia coolers, the design fouling resistance should be around 0.00017 to 0.00034 m^2K/W (3), but may reach even higher values after long periods with no defrost and no oil drainage (5). Oil fouling in CFC systems is not significant provided the oil return to the compressor functions well. Inside design fouling resistances for coolers using brine refrigerant range from about 0.0001 m^2K/W for clean systems with chilled water or glycol to 0.0005 to 0.0008 m^2K/W for systems with chloride or carbonate brines and heavy fouling.

Fouling resistances on the fins are usually in the form of frost, but may also be dust particles mixed with condensed moisture etc. Frost on the fins reduces the heat transmission coefficient of the cooler and therefore the cooling capacity. For the engineer it is important to know how the frost affects the performance of the cooler and how long it takes for the performance to decrease to a level where defrost is required.

Accurate prediction of frost growth in finned air coolers is very complex. However, to obtain the relative differences in performance of coolers A to F in Appendix 1 when operating under identical frosting conditions, a simplified method as described below is proposed.

Average frost accumulation on the coil surface during a time interval of Y secs.:

$$D = Q*(1-S0)/R/G*Y/F0 \qquad (6)$$

where

D	Frost thickness	m
Q	Total cooling capacity	kW
S	Sensible/total heat ratio	
R	Latent Heat of sublimation	kJ/kg
G	Average frost density	kg/m^3
F0	Total cooler surface area	m^2

For each time interval, the estimated frost thickness is added to the layer estimated during the previous interval and the cooler performance is re-calculated each time with the external fouling resistance.

$$F9 = D/L \qquad (7)$$

where

F9	External fouling resistance	m^2K/W
L	Thermal conductivity of frost	W/mK

To make the model more realistic, each performance calculation for each time interval is carried out at constant air velocity between the fins, i.e. the total air flow through the cooler decreases proportionally with increasing frost thickness.

For density and thermal conductivity of frost, the values 150 kg/m^3 and 0.1 W/mK may be used for the coolers listed in Appendix 1, see also ref. (6).

The effect of an internal fouling resistance of 0.0003 m^2K/W on the coolers A to F of Appendix 1 when operating under those conditions is listed in Table 6. Performance of each cooler with no fouling is 100 per cent.

Using the greatly simplified frost accumulation model described above, the relative differences in performances between coolers A to F when operating under identical frosting conditions may be established. The relative cooling capacities after 72 hours of operation at an inlet air relative humidity of 90 per cent, air on temperature of -32°C and a constant evaporating temperature of -40°C are shown in Table 7. The performance of each cooler with no fouling is 100 per cent.

As expected, the coolers with relatively low heat transmission coefficients

Cooler	A	B	C	D	E	F
Cooling capacity in per cent with internal fouling resistance of 0.0003 m^2K/W.	92.8	91.6	91.6	91.1	91.7	91.4

Table 6. Effect of internal fouling on the performances of the coolers specified in Appendix 1.

Cooler	A	B	C	D	E	F
Relative cooling capacity after 72 hours, %	49	52	46	57	69	67
Frost thickness, mm	2.4	2.2	2.4	2.1	1.8	1.8
Frost accumulation per unit refrigeration energy provided, kg/MWh	35	35	34	36	37	37

Table 7. Effect of frost on cooler performances.

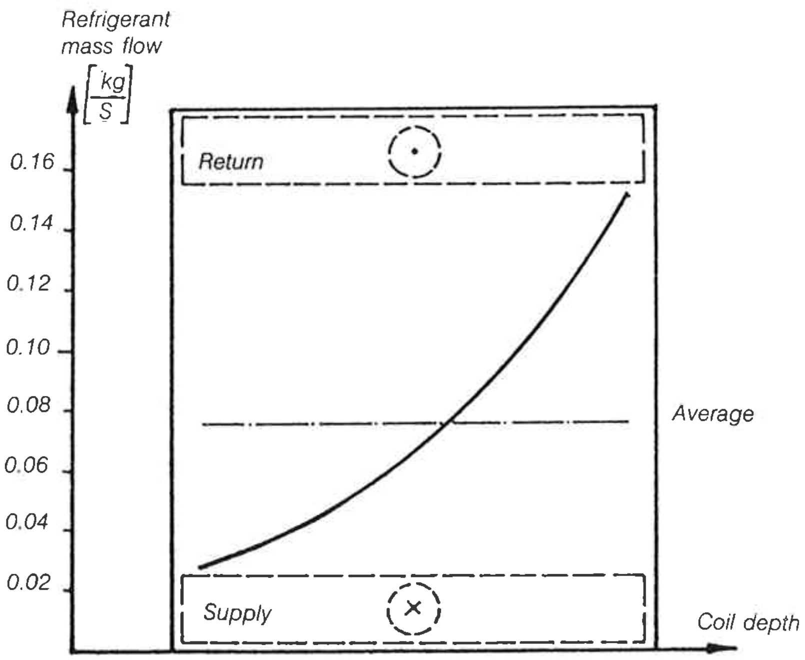

Fig. 9. Equilibrium refrigerant flows through the circuits of a crossflow finned air cooler, ammonia, liquid recirculation.