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Energy Aspects in the Design of Air Coolers for Indirect Refrigeration Systems

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FOR INDIRECT REFRIGERATION SYSTEMS

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Abstract
In this paper a model for the performance prediction and optimization of air coolers for indirect refrigeration systems is presented. The model applies for finned tube air coolers operating under frosting conditions and considers the transient heat and mass transfer between humid air and air cooler surfaces. The model is extended with an energy consumption analysis for the compression refrigeration system. Further, the properties of several indirect coolants are added to the model so that a comparison of the performance of different coolants is possible. A parameter analysis including both price/performance and energy cost considerations is presented.

INTRODUCTION

Supermarkets pay, in the last years, special attention to indirect systems to preserve their chilled en frozen products. With indirect systems the refrigerant contents of the system may be reduced to 20% of the contents of direct systems (Schmidt [1993]). In some cases CFC's are still substituted by HCFC's in such systems (Groeningen [1993]). But CFC's or HCFC's can be substituted by NH_3 which is kept in a safety controlled machine room.
ASHRAE's Refrigeration Handbook [1990] gives some considerations on secondary coolant selection and design for indirect refrigeration systems. A large problem of the indirect systems is that they lead to an extra heat exchanging level - and so to an increase of the energy consumption. For the designer of indirect systems for chilling and freezing rooms or cabinets is interesting to know, for instance, which temperature driving force for the air cooler will lead to the lowest yearly costs (energy + investment) but also the consequences of selecting a determinated secondary coolant. Infante Ferreira et al. [1994] introduced a model for the performance and price prediction for air coolers used in indirect systems. After a short description of the model, the effect of some parameters on the yearly costs of an air cooler of an indirect refrigeration system will be analysed.

MODEL FORMULATION

The analysis presented here applies for a system as schematically shown in fig. 1 and consists of compressor, condenser, expansion valve, liquid (indirect coolant) cooler and air cooler.

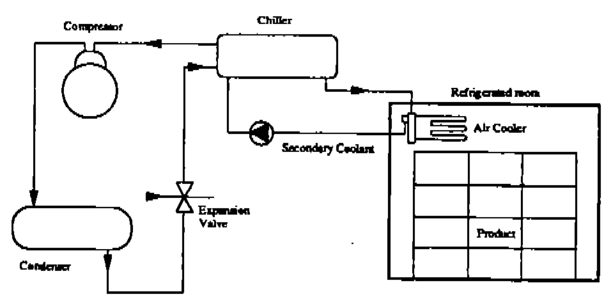

Fig. 1. - Schematic drawing of indirect refrigeration system.

Refrigeration System

The main objective of this paper is to study the effect of design parameters on the performance of the air cooler. Since some design parameters affect the electrical energy consumption (due to compression work) of the refrigeration cycle, the yearly cost analysis of the system must include the study of the energy consumption of the compressor drive. For this purpose the compressor suction pressure is assumed to be the saturation pressure corresponding to a temperature 4 K lower than the secondary coolant temperature leaving the liquid cooler and entering the air cooler. An air cooled condenser is used. The discharge pressure is assumed to be the saturation pressure corresponding to a temperature 10 K above the ambient temperature. The ambient air temperature distribution applies for De Bilt in The Netherlands and varies from -11°C to 29°C. The ambient air temperature is divided in temperature ranges. For these temperature ranges the heat load to the refrigerated/freezing room is calculated according to Infante Ferreira [1988]. Also the COP of the refrigeration system is calculated for each range.
The isentropic compressor work, w_t, is calculated from

$$w_t = p_{suct}\, v_{suct} \frac{k}{k-1} \left[\left(\frac{p_{disch}}{p_{suct}}\right)^{(k-1)/k} - 1\right] \qquad (1)$$

with p_{suct} the suction pressure [kPa], p_{disch} the discharge pressure [kPa], v_{suct} the specific volume of the refrigerant at the compressor inlet [m^3/kg] and $k = c_p/c_v$ the ratio of specific heat at constant pressure and at constant volume [-]. The energy consumption of the compressor drive, P_c, is calculated from

$$P_c = \frac{w_t\, \dot{m}_{ref}}{\eta_i \eta_{mech} \eta_{elec}} \qquad (2)$$

with $\dot{m}_{ref}$ the mass flow rate of refrigerant through the compressor [kg/s], η_i the isentropic efficiency of the compressor [-] (here assumed to be equal to 0.80), η_{mech} the mechanic efficiency of the compressor [-] (here assumed to be equal to 0.95) and η_{elec} the electric efficiency of the electromotor [-] (here assumed to be equal to 0.85).
The COP is calculated from

$$COP = \frac{\dot{Q}_o}{P_c} \qquad (3)$$

with $\dot{Q}_o$ the cooling capacity [kW]. The cooling capacity is linked to the mass flow of refrigerant by

$$\dot{Q}_o = \dot{m}_{ref}\, \Delta h \qquad (4)$$

with Δh the refrigerant enthalpy change between inlet and outlet of the liquid cooler, also depending on the temperature range of the ambient air.

Air Cooler

The air cooler is a multi-row fin-tube cross-countercurrent heat exchanger which includes a fan (or fans) for forcing the air through the heat exchanging surface. Fig. 2 shows, schematically, such an air cooler.
The secondary coolant flows inside the tubes. Since the external surface temperature mostly will be lower than the dew point of the surrounding air, mass transfer will take place from air to air cooler surface. In refrigeration applications this will be mostly in the form of frost formation. The frost layer thickness will be a function of time and since the frost layer affects the heat transfer rate and the air flow rate, the air cooler performance will be time dependent. The model is described in Infante Ferreira et al. [1994] so that here only some aspects will be discussed. For the air side the energy equation is solved for the control volume and energy transfer terms shown in fig. 3.
The frost accumulation is assumed to take place inside this control volume so that in the energy equation for the tube wall the time dependency will not be explicitely shown. Application of energy balances for the subsystems and substitution of the continuity equation lead to

$$\dot{q}_{wf} = \frac{\dot{m}_{wf}}{A_f}(h_{wa} - h_{wf}) - \rho_f \frac{\partial h_{wf}}{\partial t}\, \delta z \qquad (5) \qquad \text{and}$$

$$\dot{q}_{wt} = \frac{\dot{m}_{wt}}{A_t}(h_{wa} - h_{wt}) - \rho_t \frac{\partial h_{wt}}{\partial t}\, \delta z \qquad (6)$$

where $\dot{m}_{wf}$ and $\dot{m}_{wt}$ are the frost mass transfer rate, respectively, to the fin surface and to the tube surface [kg/s], h_{wa} the enthalpy of water in the air [kJ/kg], h_{wf} and h_{wt} the water enthalpy for the frost surface conditions, respectively, on the fins and on the tubes [kJ/kg], ρ_f en ρ_t are the frost density at, respectively, fin and tube surface [kg/m^3]. A_f and A_t are the external fin and tube surfaces, respectively [m^2] and t is the time [s].
The second term on the right hand side of eqs. (5) and (6) is small and shall be neglected so that eqs. (5) and (6) reduce to:

$$\dot{q}_{wf} = \frac{\dot{m}_{wf}}{A_f}(h_{wa} - h_{wf}) \quad (7) \quad \text{and} \quad \dot{q}_{wt} = \frac{\dot{m}_{wt}}{A_t}(h_{wa} - h_{wt}). \quad (8)$$

The continuity equation can be written as

$$\rho_f \frac{\partial(\delta_f)}{\partial t} = \frac{\dot{q}_{wf}}{A_f} \qquad (9) \quad \text{and} \quad \rho_t \frac{\partial(\delta_t)}{\partial t} = \frac{\dot{q}_{wt}}{A_t} \qquad (10)$$

with δ_f and δ_t the frost thickness on fins and tubes, respectively, [m].

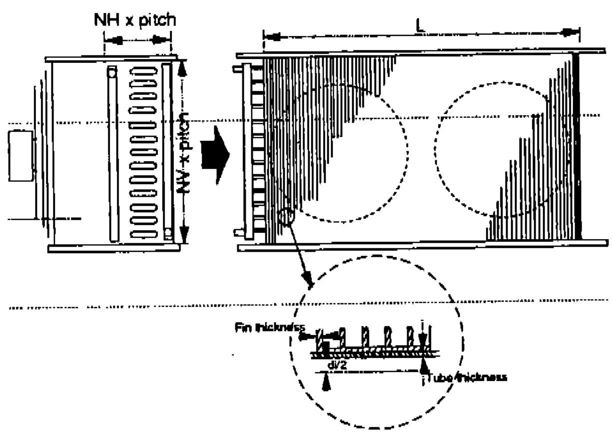

Fig. 2. - Schematic drawing of a fin-tube air cooler.

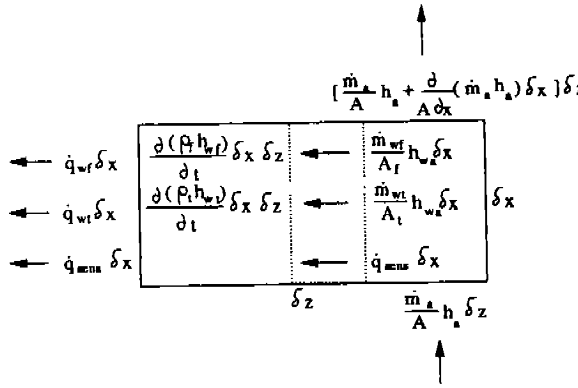

Fig. 3. - Control volume for development of the energy equation for the air side.

Although not shown explicitely, eqs. (7) and (8) are, due to the link to the continuity equation, time dependent. The subdivision of the mass transfer rate of frost into two separate flows was found to be needed because for air coolers with a relatively large tube pitch the fin effectiveness is small so that the surface temperature of the tubes and the average fin temperature can differ for some degrees Kelvin.

The set of equations consists of mass and energy conservation equations for the secondary coolant side, for the tube wall and fin surface and for the air side. The set of equations is completed with other basic equations needed to come to a set of solvable equations. The set of equations can be found in Infante Ferreira et al. [1994] and shall not be discussed here.

The pressure - volume flow and energy consumption - volume flow characteristics of a fan have been added to the model. For each time step, the intersection of the pressure - volume flow characteristic of the fan and of the frosted tube-fin air cooler is determined so that the momentary air volume flow is known.

Numerical Solution of Air Cooler Model

The set of equations which describes the air cooler model has been discretized in such a way that a finite difference approach can be used. In the secondary coolant flow direction a forward discretization method is used while in the air flow direction a finite between-pitch change is used. For the time variable a forward discretization method is applied. Acceptable convergence is obtained with time steps smaller than 900 s (15 minutes). For $t = 0$, steady-state conditions, without frost formation, are assumed.

DESIGN VARIABLES

The design variables consist of two groups - one related to the operating conditions and one related to the geometry of the air cooler. In the group related with the operating conditions the following variables are included:

- the air inlet conditions (temperature and relative humidity in the cooling / freezing room);
- the secondary coolant inlet temperature (4 to 12 K lower than the air inlet temperature);
- the secondary coolant mass flow rate (the temperature change of the secondary coolant through the air cooler should be between 0.1 and 3.0 K);
- the secondary coolant properties. Table 1 gives a list of the secondary coolants considered in this study. Most of the data have been taken from the ASHRAE Fundamentals Handbook [1993]. The ammonia-water data have been taken from Niebergall [1981]. In addition an ice slurry of propylene glycol was considered to improve the performance of this non-toxic secondary coolant. Slurries have been proposed by Kasza et al. [1987] as working fluids for thermal systems in general. Considering the experimental data of Kasza et al. we assume that the heat transfer coefficient at the slurry side is twice as large as for the pure liquid. Further we assume that the slurry temperature remains constant as the fluid passes the air cooler. The freezing point of the coolant is then equal to the slurry temperature.
- the allowed pressure drop on secondary coolant side. All the results presented in this paper have been obtained with 100 kPa as maximum allowable pressure drop, a common design restriction. This restriction leads to a subdivision of the main secondary coolant flow in a larger number of parallel circuits when the pressure should be larger than 100 kPa.
- the allowed liquid velocity in the tube side. All the results presented in this paper have been obtained with 2.0 m/s as the maximum allowable velocity to prevent erosion-corrosion problems.

Table 1. - Secondary coolants.

Propylene glycol	Ethylene glycol	Methanol	Sodium chloride	Calcium chloride
Ammonia water	Trichlorethylene	d-Limonene	Methylene chloride	Polydimethylsiloxane

In the group related to the geometry of the air cooler the following variables are included:

- the finned lengh, the distance between end tube-sheets, L in fig. 2;
- the number of tube rows perpendicular to air flow, NV (see fig. 2). This is directly connected to the vertical tube pitch and the air cooler height. In this analysis only the tube pitch and the air cooler height shall be considered;
- the number of tube rows in air flow direction, NH (see fig. 2);
- the tube pitch;
- the tube pitch arrangement (square or triangular);
- the internal tube diameter, d_i. These are usually discrete values since only some diameters are commercially available;
- the tube wall thickness, δ_{tube}. Normally connected with the tube diameter selection;

-the fin thickness, δ_{fin};
-the fin pitch;
-the manufacturing materials (steel tubes and steel fins or copper tubes and aluminium fins). Copper tubes and aluminium fins generally will lead to air coolers with a better performance/price ratio but some secondary coolants will require the use of steel tubes.
-the use (or not) of heat transfer augmentation techniques on the air side;
-the use (or not) of heat transfer augmentation techniques on the secondary coolant side.

Each variable has been varied so that its effect on the average performance for the simulation time could be determined. Most geometric data have also effect on the manufacturing costs. For each set of results, the ratio performance per unit manufacturing costs is determined. The optimum value for a variable is obtained when this ratio attains its maximum value.

SIMULATION RESULTS

Two commonly encountered air inlet conditions have been considered in this study: the cooling room conditions (0°C/85% R.H.) and the freezing room conditions (-18°C/85% R.H.). The results will be discussed separately. For the cooling room conditions the cooling period is set equal to 4 hours of operation; for the freezing room this is 8 hours. For both conditions a heat load is assumed leading to 1500 hours of full load operation of the refrigerating plant per year. Since the energy costs will only change when the evaporating temperature is changed, first the parameters affecting initial costs are analysed. The parameters affecting both energy and initial costs are then discussed.
In the simulations is assumed that the freezing point of the coolant lies 10 K below the coolant inlet temperature.

Secondary Coolant Properties (Cooling Room Conditions)

Fig. 4 shows the performance/price ratio for the secondary coolants given in table 1 as a function of the temperature change across the air cooler on the secondary coolant side. This effect was determined by varying the mass flow rate of the coolant. The secondary coolant inlet temperature is maintained constant. TV1 is the difference between the inlet temperatures of air and secondary coolant. The air inlet conditions are 0°C/85% R.H.. It appears that for all the water-based secondary coolants (except propylene glycol) there is a mass flow rate for which the performance/price ratio of the air cooler approaches the same value (0.0069 kW/NLG). The other secondary coolants also may attain acceptable performance/price ratios, depending on the mass flow rate, but they have values which are approximately 5% lower. Propylene glycol remains, for the conditions under study, in laminar flow and its performance is clearly much lower (0.0051 kW/NLG). When a propylene glycol ice slurry is used, then the performance/price ratio is brought to 0.0061 kW/NLG. This is a relatively large performance increase but the performance is still 13% lower than the performance of the water-based secondary coolants. The investment in an ice-slurry production plant will not pay back.

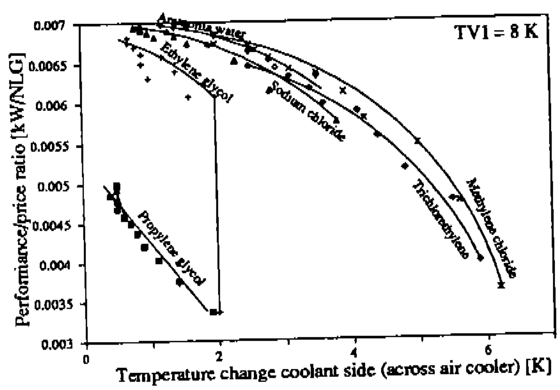

Fig. 4. - Effect of the temperature change of the secondary coolant across the air cooler on the performance/price ratio of an air cooler.

The water-based secondary coolants attain their maximum performance for temperature changes across the air cooler of 0.5 to 1.5 K while the other coolants attain their optimum for changes between 1.5 and 3.0 K. This is associated with the low specific heat capacity of this last group. Table 2 shows the required pumping power per unit cooling capacity for the optimum performance conditions of fig. 4 (TV1 = 8K).

Table 2. - Specific pumping power for optimum performance/price ratio conditions [$kW/kW_{cooling\ capacity}$]

Propylene glycol	0.046	Ethylene glycol	0.028	Methanol	0.024	Sodium chloride	0.016	Calcium chloride	0.015
Ammonia water	0.017	Trichlorethylene	0.019	d-Limonene	0.036	Methylene chloride	0.020	Polydimethylsiloxane	0.037

Fig. 5 shows the effect of secondary coolant selection on the performance/price ratio of an air cooler. The operating conditions are similar to the conditions of fig. 4. The trends of fig.4 can be recognized in fig. 5, also for other temperature driving forces. The

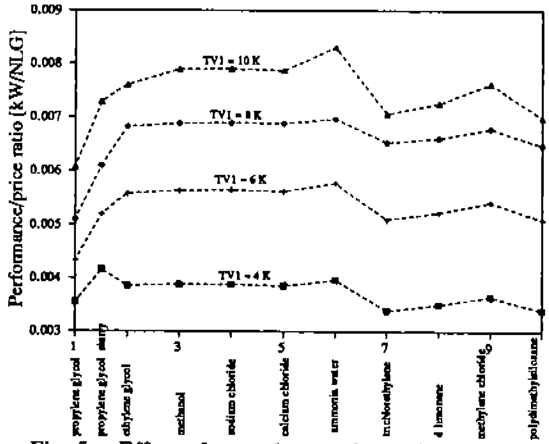

Fig. 5. - Effect of secondary coolant selection on the performance/price ratio of an air cooler (air inlet conditions 0°C/85%R.H.).

values shown apply for the mass flow rate of secondary coolant associated with the pumping power given in table 2. This means that the temperature change across the air cooler of the secondary coolant will depend on the temperature driving force between air and coolant.

Calculation of the electrical energy consumption of a 100 kW indirect refrigeration system as shown in fig. 1, with the assumption of an energy price of 0.12 NLG/kWh, gives the yearly energy costs. Assuming an amortization period of 10 years, an interest rate of 8%, and adding the energy costs, the yearly air cooler owning + compressor and pump drive energy costs are, for some of the secondary coolant systems, given in fig. 6. The optimum temperature difference is 6 K for all the secondary coolants. From fig. 6 it appears that the use of propylene glycol slurries leads to a better economic performance of propylene glycol. The advantages are not sufficiently large to support the investment in an ice-slurry production plant. Ammonia water shows clearly the best economic performance.

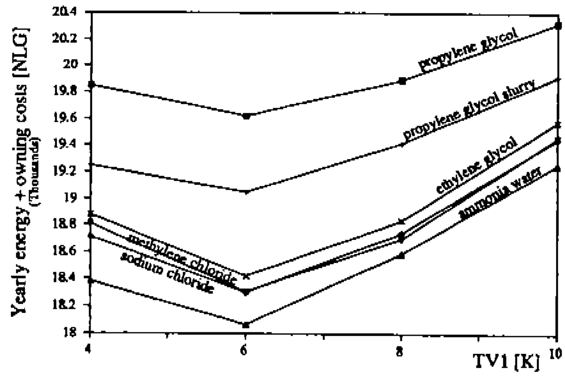

Fig. 6. - Yearly owning + energy costs for a 100 kW indirect refrigeration system with different secondary coolants (air inlet conditions 0°C/85%R.H.).

Secondary Coolant Properties (Freezing Room Conditions)

Fig. 7 is similar to fig. 5 but applies for air inlet conditions of -18°C/85%R.H.. Sodium chloride is outside its application range and is not shown. Propylene glycol has a relatively poor performance and is also not included. The non-water-based secondary coolants show now, specially for the large temperature differences, a much better performance than the water-based secondary coolants. As an exception, ammonia water shows the best performance.

Fig. 8 shows the yearly air cooler owning + compressor and pump drive energy costs for the conditions of fig. 7. Ammonia water and methylene chloride show the best economic performance. The optimum temperature difference is now 4 K.

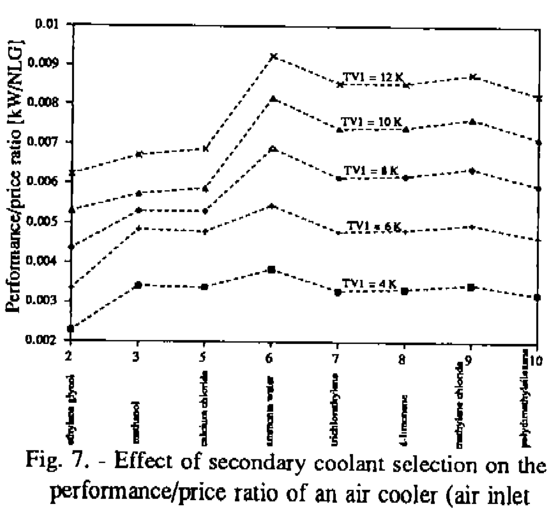

Fig. 7. - Effect of secondary coolant selection on the performance/price ratio of an air cooler (air inlet conditions -18°C/85%R.H.).

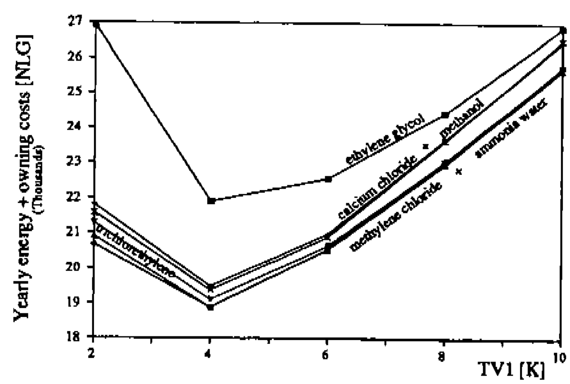

Fig. 8. - Yearly owning + energy costs for a 100 kW indirect refrigeration system with different secondary coolants (air inlet conditions -18°C/85%R.H.).

Fin Pitch (Cooling Room Conditions)

We concentrate now in sodium chloride. Other coolants show similar results. Fig. 9 shows the performance/price ratio as a function of the fin pitch with the secondary coolant inlet temperature as a parameter. The solid lines apply for the average performance after 4 hours of continuous operation. The smaller fin pitches are not feasible in combination with a large frosting rate (large temperature difference). The dotted line applies for a coolant inlet temperature of -12°C at the beginning of the frosting period (the frost thickness is 0 mm). The optimum fin pitch is then 5 mm but the large frosting rate leads rapidly to a blockage of the air flow passage area and so to a very large performance decrease. Running times of ± 4 hours before defrosting are commonly used and normally lead to performance close to the optimum effective performance (Infante Ferreira [1991]. The optimum fin pitch, for a cooler operating under these conditions during 4 hours, is 5 mm for the small temperature differences and 8 mm for the larger ones. For the optimum temperature difference (TV1=6K) is the optimum fin pitch 6 mm. Notice that the optimum is not sharp so that fin pitches from 5 to 7 mm give almost the same performance.

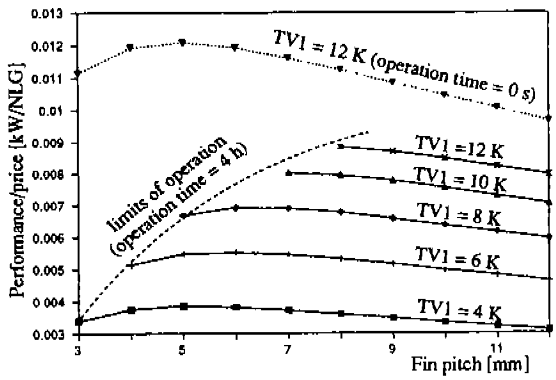

Fig. 9. - Performance/price ratio of an air cooler as a function of fin pitch with the secondary coolant inlet temperature as a parameter.

Other Parameters

In the previous sections the optimum values for the tube pitch (square 50 mm), rows in air flow direction (6), finned length (0.80 m), cooler height (0.60 m), internal tube diameter (15.6 mm micro finned tube), fin thickness (0.40 mm, with heat transfer augmentation techniques) in combination with the selected fan characteristics have been used. The effect of these parameters shall not be discussed here.

CONCLUSIONS

A model for quasi-steady-state simulation of the frost formation process on the external surface of air coolers for indirect refrigeration systems has been developed. The model has been coupled with a model for the energy consumption prediction of a compression refrigeration system. With this package some simulation runs have been executed to evaluate the performace of the secondary coolants. For the conditions studied, the following conclusions have been drawn:

a. The optimum temperature difference between air inlet and secondary coolant inlet is 6 K for an air inlet temperature of 0°C and 4 K for an air inlet temperature of -18°C. The difference between air inlet temperature and evaporating temperature is then 10 K for 0°C and 8 K for -18°C. For the freezing room conditions this is approximately the value which applies for direct refrigeration systems. For the cooling room conditions this implies an increment of the energy consumption (the optimum difference for direct systems would be ±6 K (Infante Ferreira [1988])).

b. Ammonia water shows the best economic performance followed by methylene chloride, calcium chloride and methanol. At the higher temperature level, sodium chloride and ethylene glycol have also acceptable economic performance.

c. For an inlet temperature difference of 8 K and air inlet temperature of 0°C, the water-based secondary coolants attain their maximum performance for temperature changes across the air cooler of 0.5 to 1.5 K while the non-water-based secondary coolants attain their optimum for temperature changes between 1.5 and 3.0 K.

d. The optimum fin pitch depends on the operating conditions. In general, a fin pitch of 7 mm will allow operation close to the optimum for a large range of temperature differences between air and coolant.

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