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ABSTRACT

A new method — pinch analysis is introduced to discuss design optimization of the air-conditioning refrigeration system from the point of view of saving energy and saving material, and the evaluation principle of system economy is built up. The method avoids the defect of traditional design method using experience, easy to realize optimum match between system component parts.

NOMENCLATURE

C	constant
C_P	constant pressure specific volume kJ/kg.K
COP	coefficient of performance
F	area m^2
G	mass flow rate kg/s
H	enthalpy kJ/kg
M	penalty factor
P	pressure atm
PBP	increased capital payback period year
Q	heat duty kW
T,t	temperature ℃
Δt_p	temperature difference of pinch-point ℃
V	volume flow rate m^3/h
W	compression work kW
ρ	density kg/m^3
π	pressure ratio
ε	error

Subscripts:

a	air	r	refrigerant
c	condenser	e	evaporator
t	total		

INTRODUCTION

With the development of computer technology, simulation and optimization of refrigeration systems are getting possible, and due to facing two problems — the substitution for CFC and the renewal of design methods in the refrigeration field, studies are gradually turned to system simulation, design optimization and system CAD etc.. For instance, modeling and optimization of refrigeration systems have been listed as prime task in Norway, and many other countries such as America, Japan and German are also active in these aspects.

At present, much work about design, calculation and modeling of refrigeration systems has been done[1,2], but most of them use traditional design method with experience, without considering optimum match of the refrigeration system from the point of view of saving energy and saving material. Here a new method — pinch analysis is put forward to simulate the air-conditioning refrigeration system, when material consumption of heat exchangers is restricted, the system COP can be promoted to the greatest extent. In this way saving energy and saving material are integrated together with an economy index to evaluate the performance of system.

MODELING METHOD

As an energy saving technology, pinch technology has been successfully used in optimization of heat exchanger network[3,4]. Similarly, it can be applied to simulate the air-conditioning system. Fig.1 shows the air-conditioning refrigerant cycle, comprised of four main process: compression process 1-2, condensation process 2-3-4-5, expansion process 5-6 and evaporation process 6-7-1.

In most cases, there exists a pinch point between the refrigerant and air both in the condenser and the evaporator, at which the temperature difference between the refrigerant and air reaches minimum, as shown in Fig.2. Generally, the pinch point in the condenser lies at T_3 and t_{a3}, and T_6 and t_{a6} in the evaporator, so the temperature difference of the pinch point can be written respectively as $\Delta t_{cp}= T_3-t_{a3}$ and $\Delta t_{ep}=t_{a6}-T_6$. They can not only decide the heat exchanger area (or heat exchanger cost), but also decide COP of the refrigeration system. The method proposed in this paper emphasizes the concept of the temperature differences of pinch points, using Δt_{cp} and Δt_{ep} as control parameters to write a program to simulate the air-conditioning refrigeration system in order to achieve optimum match of air-conditioner performance. Fig.3 shows the program flowchart.

During practical design of air-conditioning system, the air is usually required to be cooled from primitive temperature to target temperature, so we can obtain the cooling duty of the evaporator as

$$Q_e = \rho_{a1} V_a^e \left(H_{a1} - H_{a6}\right) \tag{1}$$

If the pinch-point temperature differences in both condenser and evaporator are given, we have

$$T_e = T_6 = t_{a6} - \Delta t_{ep} \tag{2}$$

$$T_k = T_3 = t_{a3} + \Delta t_{cp} \tag{3}$$

Since the temperature t_{a3} is unknown, an initial value is assumed in calculation, then the refrigerant enthalpies at point 2, 3, 4 and 5 can be decided. For the evaporation temperature is known, so the refrigerant mass flow rate is got

$$G_r = Q_e / \left(H_1 - H_6\right) = Q_e / \left(H_1 - H_5\right) \tag{4}$$

According to the heat balance, the heat duty required by the air due to the temperature lift from t_{a5} to t_{a3} is described as

$$Q_{35} = G_r\left(H_3 - H_5\right) = \rho_{a5} V_a^c C_{pa}\left(t'_{a3} - t_{a5}\right) \tag{5}$$

Consequently, the new condenser air pinch temperature can be soluted

$$t'_{a3} = t_{a5} + \frac{G_r\left(H_3 - H_5\right)}{\rho_{a5} V_a^c C_{pa}} \tag{6}$$

Then comparing with the previous value t_{a3}, if $\left|t'_{a3} - t_{a3}\right| > \varepsilon$, another iteration will be required using $t_{a3} = t'_{a3}$ until the convergence condition is satisfied. Thus we can calculate each parameter, such as Qe, W, COP, π, Fe, Fc, Ft.

Where

$$W = G_r\left(H_2 - H_1\right) \tag{7}$$

$$COP = \frac{Q_e}{W} \tag{8}$$

$$\pi = \frac{P_2}{P_1} \tag{9}$$

$$F_t = F_e + F_c \tag{10}$$

EXAMPLE AND ANALYSIS

The proposed method has been applied to design optimization of the refrigeration system of type KLD29 train air-conditioner. The known design parameters are as follows: in the evaporator, the air flow volume $V_a^e = 2500 m^3/h$, the air inlet temperature $t_{a1} = 25$ ℃, the air outlet temperature $t_{a6} = 12$ ℃; in the condenser, the air flow volume $V_a^c = 5000 m^3/h$, the air inlet temperature $t_{a5} = 35$ ℃, superheat degree $\Delta t_{sup} = 10$ ℃, subcooling degree $\Delta t_{sub} = 8.3$ ℃, the refrigerant is R22.

Fig.4 demonstrates the changes of system COP and total heat transfer area Ft (including evaporator and condenser) with the two controlling parameters — Δt_{ep} and Δt_{cp}. It can be seen that as Δt_{ep} or Δt_{cp} decreases, COP and Ft both increase smoothly, but more sharply when Δt_{ep} and Δt_{cp} are small. If the system COP is considered separately, the smaller Δt_{ep} and Δt_{cp}, the greater the COP, but at the same time Ft will increase sharply, causing the rapid increment of the capital of heat exchangers, which may balance out even exceed the profits brought by the increase of COP.

EVALUATION PRINCIPLE

The above analysis shows there is a trade-off between COP and heat exchanger area. When pinch method is used to simulate refrigeration systems, Fig.5 gives the change of system COP with total heat transfer area Ft. It can be seen that COP increases with total area, and the curve has a steep start, then getting more and more even as total area continues to increase. Therefore, even if the size of heat exchanger is enlarged further, the improvement of COP would be little, so economically it's very disadvantageous. In a design process, the dependent relation between COP and heat exchangers area should be fully considered in order to realize optimum system economy, so the evaluation principle of system economy — increased capital payback period must be built up.

On the basis of the original refrigeration system, if the size of heat exchangers is enlarged, the capital cost of system will increase, also the system COP will be improved, which lowers the cost of operation. Therefore, we may as well discuss the payback period of the increased capital cost duing to the size enlargement of heat exchangers[5,6].

If we know the increased capital cost ΔP, the annual average cooling duty Q_0, the annual running hours h, the lifespan of system n, the price of electric energy input C_E, the motor efficiency η_{mo}, and the original system performance COP_0, then the yearly increased capital cost is got (the unit is ￥/kw).

$$\Delta P' = C_F / n \tag{11}$$

Where $C_F = \Delta P / Q_0$, represents the increased capital cost per unit cooling duty.

For a refrigeration system, the price of per unit cooling duty can be described as (￥/kw.h)

$$C_A = \frac{C_E}{\eta_{mo} COP} + \frac{\Delta P'}{h} \tag{12}$$

While in the original system, it becomes that

$$C_0 = \frac{C_E}{\eta_{mo} COP_0} \tag{13}$$

Then, the payback period can be deduced as follows:

$$PBP = \frac{C_F}{h(C_0 - C_A)} = \frac{C_F}{\frac{hC_E}{\eta_{mo}}\left(\frac{1}{COP_0} - \frac{1}{COP}\right) - \frac{C_F}{n}} \tag{14}$$

Since the cost of heat exchangers ΔP increases with the heat transfer area ΔF, they should satisfy a certain correlation, supposing

$$\Delta P = C_H \cdot \Delta F \tag{15}$$

Where C_H represents the cost increased by enlarging unit heat transfer area, it varies with different materials.

Now the increased capital payback period for above example will be discussed by incorporating some typical parameters (C_H=200-250 ￥/m², C_E=0.45-0.60 ￥/kw.h, n=10, η_{mo}=0.75, h=6000), and supposing the original pinch-point differences are Δt_{ep} =10 ℃, Δt_{cp} =10 ℃.

Fig.6 gives the changes of PBP with Δt_{ep},Δt_{cp} when C_H=200 and C_H=250. It is seen that as Δt_{ep} decreases, PBP descends sharply at first, and smoothly when $\Delta t_{ep} \leqslant 6$ ℃; whereas the greater Δt_{cp}, the more advantageous, because PBP descends as Δt_{cp} increases. Considering the overall effects of Δt_{ep}, Δt_{cp} on PBP, the values of Δt_{ep} and Δt_{cp} can be decided to lower PBP to the greatest extent. For this example, choosing Δt_{ep} =5 ℃ and Δt_{cp} =8 ℃ is more suitable. The filled areas show the changes of PBP when C_E floats within 0.45-0.60 ￥/kw.h, and the upper, the lower boundary curve is drawn respectively when C_E=0.45 ￥/kw.h and C_E=0.60 ￥/kw.h, so the fact that PBP becomes shorter when C_E increases is seen. Besides, PBP is much affected by C_H, which gets longer when C_H increases.

It is noted that when the size of heat exchangers increases, the refrigerant mass flow rate and compressor capacity will decrease if the cooling duty is kept identical, so the motor and compressor with smaller capacity can be chosen. Since the above calculation didn't allow for this, the results are conservative.

OPTIMAL DESIGN

As shown in Fig.5, the system COP increases with total area of heat exchangers, and the improvement of COP would have a great potential even if within the same total area (or the same total cost). It is seen that when Ft=120m², the COP value at Δt_{ep}=2 ℃ is 15.3% higher than that at Δt_{ep}=10 ℃, which mainly results from the distribution of the total area between evaporator and condenser. Although the total area is given, the area of evaporator or condenser could change through controlling the evaporator and condenser pinch-point temperature differences, accordingly the system COP also changes. Therefore, the optimization target can be built up as follows: the optimal COP of the refrigeration system is acquired under certain total area (or under the fixed cost).

This optimization aims at lowering energy consumption, therefore the objective function can be defined as the reciprocal of COP, that is

$$f = \frac{1}{COP} \tag{16}$$

While COP is related to Δt_{ep} and Δt_{cp}, then the mathematical expressions can be written as

$$\begin{cases} \min f\left(\Delta t_{ep}, \Delta t_{cp}\right) \\ s.t.\ F_t\left(\Delta t_{ep}, \Delta t_{cp}\right) = C \end{cases} \tag{17}$$

Equation (17) is a constrained problem to minimize $f\left(\Delta t_{ep}, \Delta t_{cp}\right)$, which can be converted to an unconstrained problem with exterior penalty function strategy. Here the penalty function is introduced[7]

$$F\left(\Delta t_{ep}, \Delta t_{cp}, M\right) = f\left(\Delta t_{ep}, \Delta t_{cp}\right) + M\left(F_t\left(\Delta t_{ep}, \Delta t_{cp}\right) - C\right)^2 \tag{18}$$

So long as the unconstrained minimum value of $F\left(\Delta t_{ep}, \Delta t_{cp}, M\right)$ is solved, the optimization solution is then got. This is a two-dimensional, non-linear optimization problem with one equality constraint. Owing to the existence of constraint condition, there is only one independent parameter of the two variables —Δt_{ep} and Δt_{cp}. The value of COP can be obtained by searching Δt_{ep} and Δt_{cp} within the boundary range respectively.

The optimal design of system COP with different total heat transfer area is discussed in details, as shown in Fig.7. It can be seen that, although the total area is given, the system COP may change by controlling the evaporator and condenser pinch-point temperature differences. The three-dimensional curves show the changes of COP with Δt_{ep} and Δt_{cp}, when Δt_{ep} is small and Δt_{cp} is great, the refrigeration system can acquire high COP. Therefore, the optimal design can be achieved by diminishing the evaporator pinch-point temperature difference as possible as we can.

Fig.8 illustrates both evaporator and condenser heat transfer area changes with Δt_{ep} under certain total area. As Δt_{ep} decreases, the evaporator fin area increases, while the condenser fin area decreases. For small Δt_{ep} can bring high COP, the size of evaporator should be enlarged further to promote system COP to the greatest extent if the total heat transfer area is definite.

CONCLUSIONS

Pinch analysis not only can be applied to optimization of heat exchanger network, but also can be applied to design optimal refrigeration system. Pinch analysis is used in this paper to discuss optimal design of the air-conditioning refrigeration system from a point of view of saving power and saving material. When material consumption of heat exchangers is constant, the COP value can be promoted to the greatest extent to save energy. Accordingly the evaluation principle of system economy is built up to guide optimal design of an air-conditioning system. The method aims at saving power and saving material in a refrigeration system, and avoids the defect of traditional empirical design method, which is easy to realize optimum match between component parts. The program is simple, and satisfactory results can be obtained when the method is used to the matched design of an air-conditioning refrigeration system.

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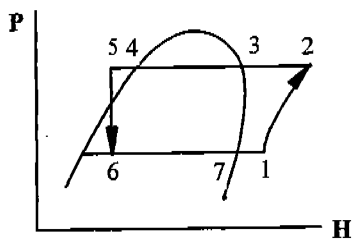


Figure 1 The Air-Conditioning Refrigerant Cycle

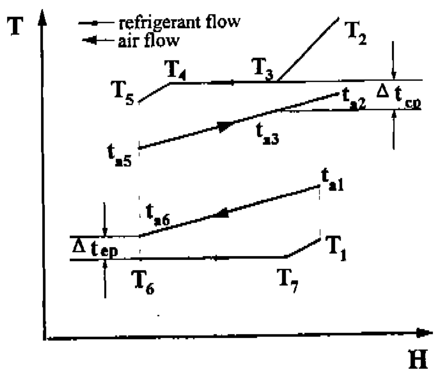


Figure 2

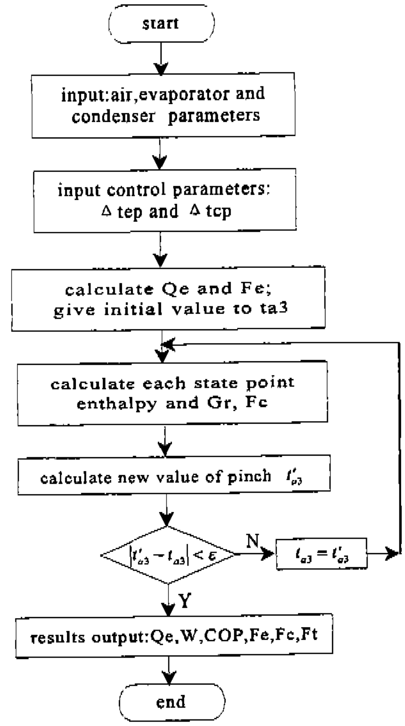


Figure 3 The Program Flowchart

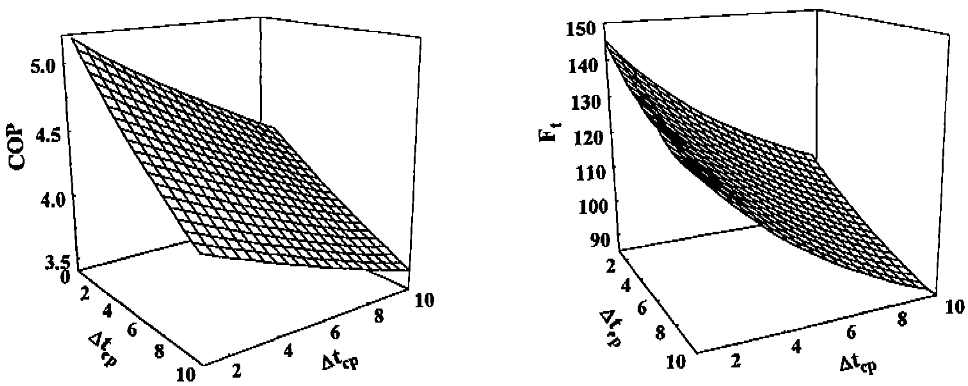


Figure 4 The Changes of COP and Ft with Δt_{ep}, Δt_{cp}

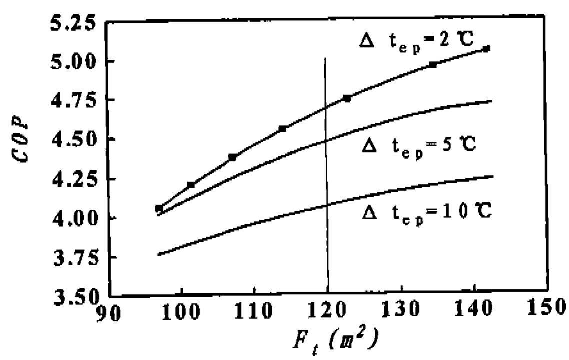
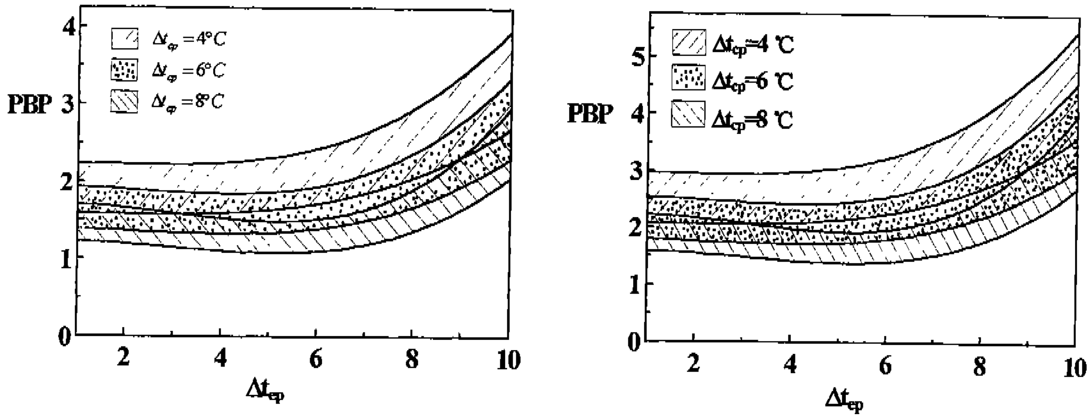
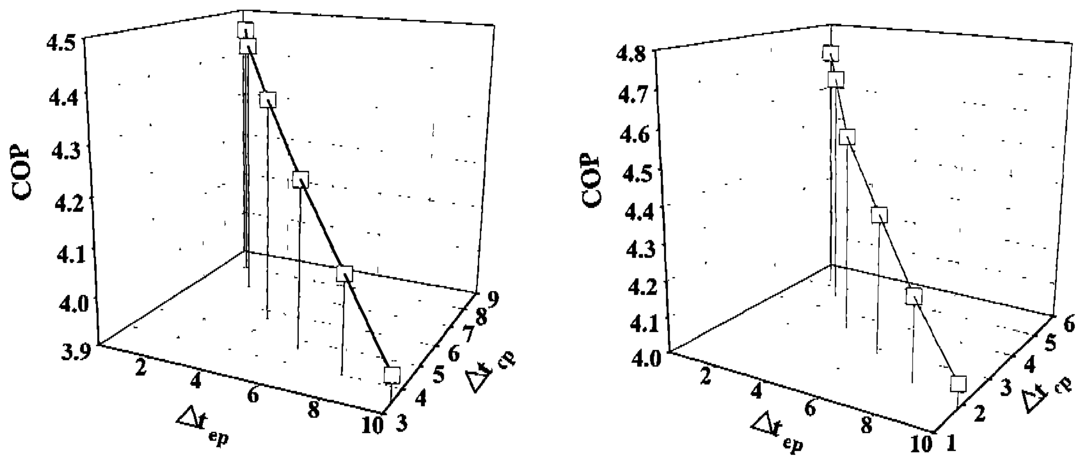


Figure 5 The Change of COP with Ft



C_H=200 C_H=250

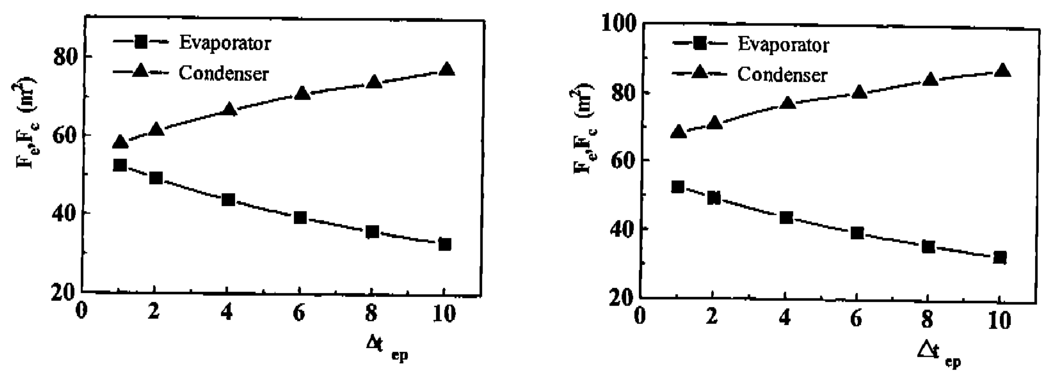
Figure 6 The Increased Capital Payback Period



(a) (b)

Figure 7 The Change of COP with Δt_{ep}, Δt_{cp} When the Total Area is Definite

(a) Ft=110m^2 (b) Ft=120m^2



(a) (b)

Figure 8 The Changes of Evaporator and Condenser Fin Area with Δt_{ep}

(a) Ft=110m^2 (b) Ft=120m^2