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Power for Fans and Pumps in Heat Exchangers of Refrigerating Plants

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Abstract

The energy consumption to operate auxiliaries such as fans and pumps in heat exchangers have a significant influence on the total energy demand for operating a refrigerating system. With a starting point in a simple entropy analysis a more practical approach is adopted for common cases of air coil fans in evaporators or condensers. Examples are given to illustrate how the power for evaporator or condenser fans will affect the capacity and total energy demand of refrigerating systems.

It is shown that the two different criteria:
- maximum of capacity or
- minimum of energy demand (equivalent to maximum system COP)

will give different optima for the power to be used in fans or pumps.

Simple relations are derived for optimum power in fans or pumps for the two different criteria applicable for many general cases in refrigerating systems.

Introduction

The energy needed to operate fans or pumps is important when considering the *total* energy demand to operate a refrigerating plant or a heat pump. It is not unusual that the electric power of such auxiliaries is in the order of 25 % or more of the power to operate the compressor in a system. The purpose of this paper is to illustrate and exemplify this issue. Simple relations will be derived for optimum power in fans or pumps to reach criteria like maximum capacity or maximum system COP.

Let us exemplify with the application of an evaporator. As a starting point let us assume that we have a given plant where we can adjust the fan speed – in practice perhaps by means of an inverter control. It is obvious that by using a high fan speed the evaporator will operate with smaller temperature differences between the inlet air and the refrigerant evaporating temperature than if low speed is used. This will decrease the temperature lift of the *cycle* and thus decrease the compressor work. However we will have to pay for the fan power and what is of interest is the sum of the power for the compressor *and* the fan. It is obvious that there must exist a certain fan power that we can call optimal from the point of view of energy consumption.

Two different approaches will be used: First a treatment minimizing the entropy generation and, secondly, a more practically oriented way of treatment will be demonstrated. This treatment will concentrate on *refrigerating* applications. Slightly different relations will be obtained for *heat pump operation.* Space limitations prevents a treatment for that case but the practical result for minimum energy demand are quite similar.

Relations between pumping power and temperature difference

The pumping power will influence the temperature difference for a case *with given geometry*. A reasonable assumption is that the overall heat transfer coefficient is proportional to V^{nU} where V is the fluid flow and n_U is an exponent, which in most cases has a value in the range of 0,3 to 0,6. The pressure drop can be set proportional to V^{np} where for turbulent flow $n_p \cong 1{,}8$. The pumping power, E_P, will hence (assuming constant pump efficiency) be proportional to $V^{(np+1)}$. Based on a reasoning indicated we can write the overall temperature difference, θ, as

$$\theta \cong C \cdot (E_P)^{-nE} : \qquad (1)$$

where $n_E \cong \dfrac{n_U}{n_p + 1}$ which hence for most cases will be in the order of 0,1 to 0,2

C is a constant.

If we consider different operating conditions for a given geometry we can express the constant C based on the temperature difference θ_o and pumping power E_{po} in a *reference* case ("o"); thus $C = \theta_o \cdot (E_{po})^{nE}$.

Entropy generation in forced flow heat exchange processes

A thermodynamic approach for the analysis of the heat exchange in a heat exchanger can be used minimizing the entropy generation. (E.g. Bejan, 1996, give thorough discussion on methods. De Jong et al 1997 show an application.) A simple treatment can be made as follows:

The "entropy generation" due to the temperature differences in a heat exchanger where the heat flow Q is transferred between two media of temperatures T_a and T_b is

$$\Delta S_{hx} = \frac{Q}{T_a} - \frac{Q}{T_b} = \frac{Q}{T_a \cdot T_b} \cdot (T_a - T_b)$$

which also can be written

$$\Delta S_{hx} = \frac{Q}{T^2} \cdot \theta$$

where T is the average temperature =($\sqrt{T_a \cdot T_b}$) and θ (= T_a - T_b) is the temperature difference.
However in a system with forced convection the pumping power will be dissipated (at ≅ temperature T) by friction and other irreversibilities so that, in total, the entropy generation is:

$$\Delta S_{tot} = \frac{Q}{T^2} \cdot \theta + \frac{E_p}{T} \qquad (2)$$

Figure 1 shows the relation between the entropy generation and the pumping power for three examples with different reference temperature differences θ_o. (The relation between θ and E_p as given by equation 1 is used.)

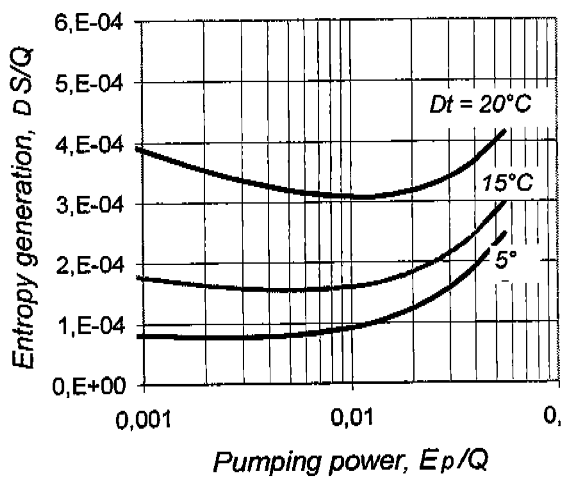

Figure 1. Entropy generation (ΔS_{tot} /Q) for heat exchange for three cases with reference temperature differences θ_o = 5; 10 and 20 K. (For all cases n_E =0,15).

It is obvious that a certain pumping power will result in a minimum entropy generation. Given the assumptions made it can be shown that the following value of the pumping power will result in this minimum entropy generation:

$$\left(\frac{E_p}{Q}\right)_{\Delta S\ min} = n_E \cdot \frac{\theta}{T} \qquad (3)$$

Notice that the larger the temperature difference is, the larger fan power should be used to minimize entropy generation.

Example: Let us assume that the temperature difference in the operating point considered is θ = 10°C, T = 273K and exponent n_E = 0,15 then we find that

$$\left(\frac{E_p}{Q}\right)_{\Delta S\ min} = 0{,}15 \cdot 10 / 273 \cong 0{,}005$$

This means that a fan power of only about 5 Watts per kW heat transferred should be spent!! Can we trust the results form the entropy approach also for practical applications? Let us also a more practical approach:

A practical approach

Let us consider a complete refrigerating system as shown in figure 2 (compare Granryd, 1973) *We can assume that system exists, with given components. Let us also assume that we have a possibility to adjust the fan speed to any level we desire without other changes in the plant.* In figure 3 a number of parameters are exemplified versus the air velocity in the evaporator fan coil for a given system.

It is obvious that there is a certain velocity resulting in a maximum net cooling capacity of the plant and still another velocity that will give minimum energy demand. Which speed should we choose? It is obvious that we can have different criteria in mind:

- Maximum cooling capacity with a given set of components
- Lowest total energy demand for a given cooling load duty.

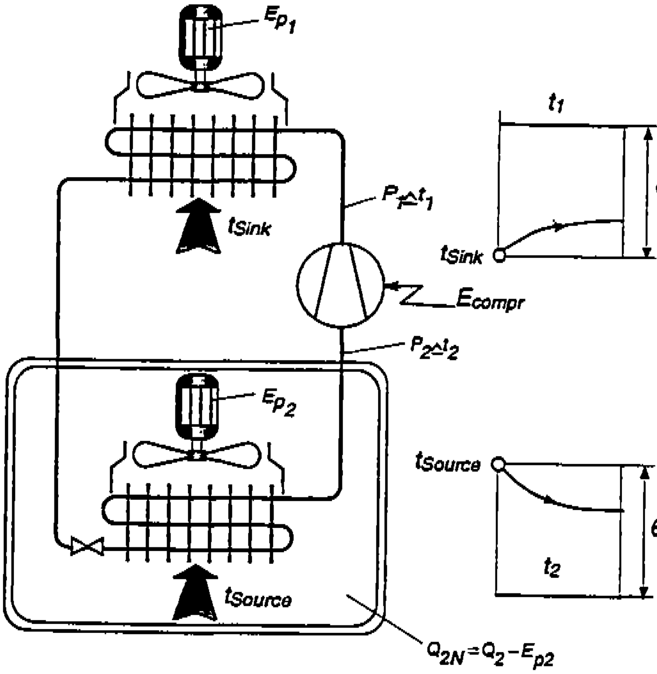

Figure 2. Refrigerating system and schematic temperatures in evaporator and condenser

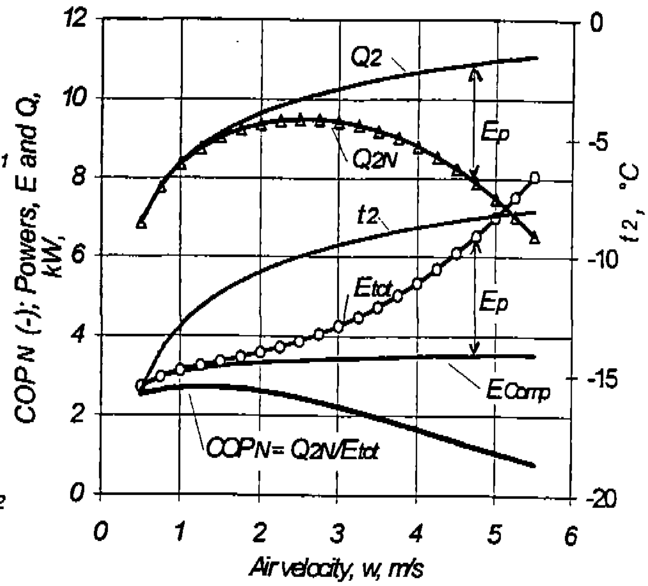

Figure 3. Influence of air velocity in evaporator (Data: $w_o = 2{,}5$ m/s; $Q_{2o} = 10$ kW; $E_{fo} = 0{,}5$ kW; $t_{room} = 0$°C; $t_{2o} = -10$°C; $t_1 = 30$°C; $k_q = 0{,}055$; $nE = 0{,}15$)

Net cooling capacity.

To derive the net useful cooling capacity Q_{2N} we must, from the compressor cooling capacity Q_2, deduct the fan power, E_p, which will be dissipated in the cooled space:

$$Q_{2N} = Q_2 - E_p \tag{4}$$

Often the cooling capacity for a vapor compression system can be set

$$Q_2 = Q_{2o} \cdot (1 - k_q \cdot (t_2 - t_{2o})) \tag{5}$$

where $k_q = \dfrac{\partial(Q_2)}{\partial(t_2)}$ a parameter indicating how capacity changes with the evaporating temperature.

Often k_q is in the order of 0,05 (5% increase in cooling capacity per degree increasing evaporating temperature), somewhat depending on the type of compressor and operating temperatures.

Inserting $t_2 = t_{Room} - \theta$ with θ as given by eq. (1) we will (with Q_{2o} and E_{po} denote the cooling capacity and fan power in the reference case in which we have temperature difference θ_o) have:

$$Q_{2N} = Q_{2o} \cdot \left[1 + k_q \cdot \theta_o \cdot (1 - (E_{po} / E_p)^{nE}\right] - E_p \tag{6}$$

The net cooling capacity compared to that for the reference case (Q_{2N}/Q_{2No}) is exemplified in Figure 4 as a function if the fan power in the evaporator. Three reference cases are shown with different temperature differences in the evaporator. Changes in the cooling capacity will for a practical case with given net cooling load be compensated by changes in the relative running time of the compressor (provided that we have an on-off type capacity modulation).

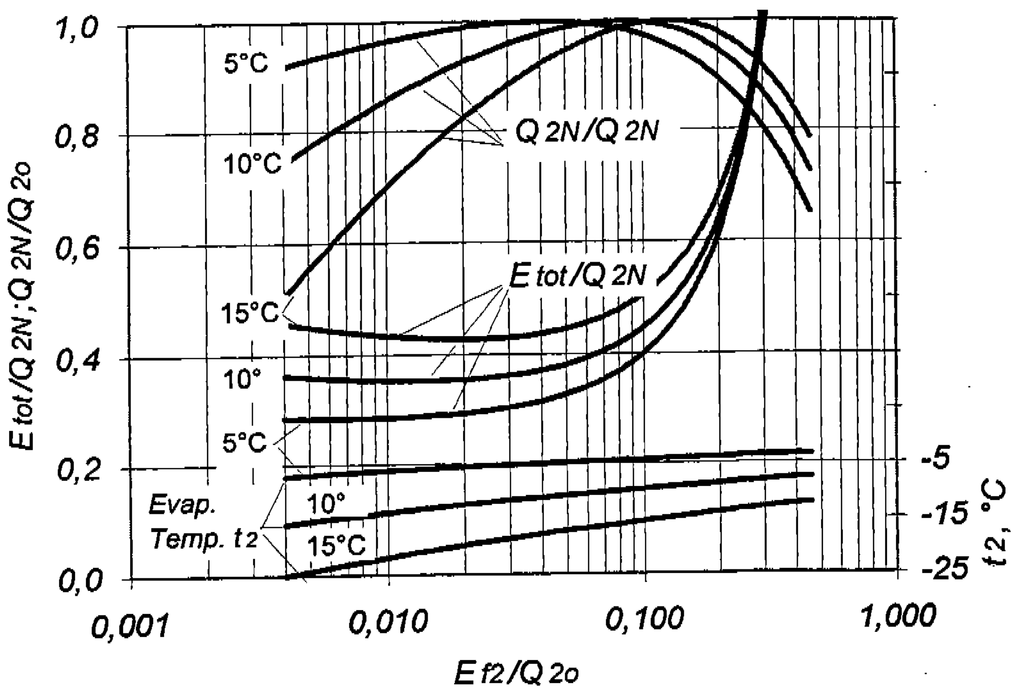

Figure 4. Influence of evaporator fan power on net cooling capacity, Q_{2N}/Q_{2No}, and total energy, E_{tot}/Q_{2N}. (Example with $t_1 = 30$ °C; $t_{Room} = 0$ °C; $k_q = 0{,}055$; $n_E = 0{,}15$; $\eta_{Ct} = 0{,}5$. Also the evaporator temperature, t_2, is indicated in the lower part of the diagram.)

Total energy demand

For many operating conditions the plant is oversized and the user does not need maximum cooling capacity for the cooling duty. The plant may thus be operated on-off or with reduced capacity. The user wants naturally to operate his plant as efficiently as possible, saving energy and money. Let us examine how the pumping power influences the energy demand.

The power to operate the plant is basically the power for the compressor and for the pumps or fans. We can, with reference to figure 2, write the total energy demand (denoting the compressor cycle performance $COP_2 = Q_2/E_{compr}$):

$$E_{tot} = \left[\frac{Q_2}{COP_2} + E_{p2} + E_{p1}\right] \tag{7}$$

or, which is more interesting, the total energy demand in relation to net cooling capacity, Q_{2N}:

$$\frac{E_{tot}}{Q_{2N}} = \frac{Q_2}{Q_{2N}} \cdot \left[\frac{1}{COP_2} + \frac{E_{p2}}{Q_2} + \frac{E_{p1}}{Q_1} \cdot \frac{Q_1}{Q_2}\right] \tag{8}$$

The compressor COP_2 of the system is influenced by the evaporating and condensing temperatures. Let us express this relation by introducing a total Carnot efficiency of the system, by which:

$$COP_2 = \eta_{Ct} \cdot \frac{T_2}{(T_1 - T_2)} \tag{9}$$

where η_{Ct} = total cycle Carnot efficiency (including refrigerant cycle as well as compressor efficiencies)

T_2 = *evaporating temperature* = $T_{source} - \theta_2$

T_1 = *condensing temperature* = $T_{sink} + \theta_1$

T_{source} is the temperature in the heat source (or the room to be cooled)
T_{sink} is the temperature of the heat sink (ambient temperature or cooling water)
θ_2 and θ_1 are the temperature differences in evaporator and condenser.

Figure 4 exemplifies the influence of the fan power on the net cooling capacity Q_{2N}/Q_{2No} as well as the total energy demand , E_{tot}/Q_{2N}, of a plant. Notice that the minimum of E_{tot}/Q_{2N} (corresponding to maximum of an overall COP of the system) is achieved with considerably lower fan power than what is needed to reach maximum cooling capacity! As is seen the curves are quite flat around the optima. It might in practice for simplicity be beneficial to choose a fan power somewhere between the two criteria. Simple relations can however be derived for the two optima as follows:

Evaporator fan power for maximum net cooling capacity

From the equations (6) and (1) it can be shown that the following very simple relation give the optimum evaporator fan power to reach maximum net cooling capacity

$$(E_{p2} / Q_2)_{Q\,max} = n_{E2} \cdot k_q \cdot \theta_2^{**} = C_{2Q\,max} \cdot \theta_2^{**} \qquad (10)$$

where θ_2^{**} is the temperature difference prevailing for the given plant in conditions for the criteria Q_{max}.

Example: With typical values $n_{E2} = 0{,}15$ and $k_q = 0{,}05$

the result is: $(E_{p2} / Q_2)_{Q\,max} \cong 0{,}15 \cdot 0{,}05 \cdot \theta_2^{**} \cong 0{,}0075 \cdot \theta_2^{**}$

and for design temperature differences:

	$\theta_2^{**} =$	5	10	15 °C
this means:	$(E_{p2} / Q_2)_{Q\,max} \cong$	0,038	0,075	0,11

It is thus beneficial to spend larger fan power the larger the temperature differences in the evaporator. Commercial evaporators are often equipped with fans in the order of 5 to 8 % of the nominal capacity. This seem to coincide quite well with the criteria for maximum capacity for cases where the temperature differences are about 10 °C.

Fan power for minimum energy demand

Evaporator side

Based on the equations (1) and (8) to (9) it can be shown that the minimum energy demand for a given net cooling capacity will be obtained if the pumping power is chosen to satisfy the following relation:

$$(E_{p2} / Q_{2N})_{E\,min} \cong C_{2E\,min} \cdot \theta_2^{*} \qquad (11)$$

where θ_2^{*} is the temperature difference prevailing for the given plant in conditions for the criteria E_{min}.

$$C_{2E\,min} \cong \left[n_{E2} \cdot \frac{\varphi_2}{T_1 - T_2 \cdot (1 - \eta_{Ct})} \cdot \frac{T_1}{T_2} \right]$$

φ_2 is a factor which for practical cases can be set to ≅ 1,25 (strictly a function of T_1, T_2; η_{Ct} and k_q, The interested reader is referred to Granryd, 1973)
T_1 and T_2 are the condensing and evaporating temperatures, expressed in absolute temperature (K)
η_{Ct} = the total Carnot efficiency of the cycle

Condenser side

Minimum total energy demand for a given net cooling capacity is similarly achieved if the fan power is chosen to

$$(E_{p1} / Q_1)_{E\,min} \cong C_{1E\,min} \cdot \theta_1^{*} \qquad (12)$$

where θ_1^{*} is the condenser temperature difference for the given plant in conditions for the criteria E_{min}

$$C_{1E\,min} \cong \left[n_{E1} \cdot \frac{\varphi_1}{T_1 - T_2 \cdot (1 - \eta_{Ct})} \right]$$

φ_1 is a factor (in analogy to φ_2) which for practical cases can be set ≅ 1,05

Example:

Let us use the following temperatures and data which can be representative for a typical practical example:

Temperatures: $T_1 = 303\text{K}\ (= 30°\text{C})$ $T_2 = 263\text{K}\ (= -10°\text{C})$;
and data: $k_q = 0{,}055\ 1/\text{K}$ $n_{E1} = 0{,}15$ $n_{E2} = 0{,}15$ $\eta_{Ci} = 0{,}5$

The result is, for the different criteria:

Maximum net cooling capacity	Minimum energy demand: Evaporator side	Condenser side:
$C_{2Qmax} \cong 0{,}008\ (1/°\text{C})$	$C_{2Emin} \cong 0{,}0012\ (1/°\text{C})$	$C_{1Emin} \cong 0{,}001\ (1/°\text{C})$

Assuming temperature differences in the evaporator and condenser in the order of ***10°C*** the result is (by equations 10, 11 and 12 respectively):

$(E_{p2}/Q_2)_{Qmax} \cong 0{,}08$ or ***8%***; $(E_{p2}/Q_2)_{Emin} \cong 0{,}012$ or ***1,2%*** $(E_{p1}/Q_1)_{Emin} \cong 0{,}01$ or ***1%.***

Discussion and some conclusions

Many commercially sold evaporators have often, as mentioned, fan power installed in the range of 5 to 8% of the nominal capacity at a temperature difference of 10 °C. This corresponds for many cases quite closely to the criterion of maximum cooling capacity but it is 6 to 8 times higher than what strictly would correspond to a maximum of system COP. The ratio of fan power for maximum capacity to that for maximum system COP corresponds roughly to what is achieved if the full fan speed (used in conditions where max. capacity is needed) would be cut in half to save energy in conditions when the capacity is large enough for the demand.

For advanced fan speed control e.g. by inverters the relations derived here may be beneficial to use for an intelligent microprocessor control of the fan (or pump) speed at different conditions to reach optimal control for different criteria and in different operating conditions. As is seen from figure 4 the curves are quite flat around the optima and it might in practice for simplicity be beneficial to make a compromise between the two criteria and choose the fan power somewhere in between.

Slightly different relations will be obtained for *heat pump operation* but the final result for minimum energy demand will be quite similar.

Summary

Relations for the fan power resulting in maximum net cooling capacity and minimum energy requirement for a given net cooling load can be summarized by the following relations:

For ***maximum net cooling capacity*** the fan power in the ***evaporator*** should be chosen as:

$$(E_{p2} / Q_{2N})_{Q\,max} \cong C_{2Q\,max} \cdot \theta^{**}$$

and ***minimum energy demand for given net cooling capacity*** the fan power should be chosen as::

$$(E_p / Q)_{E\,min} \cong C_{E\,min} \cdot \theta^{*}$$

where θ^{**} and θ^{*} are temperature differences (in operating point with optimal fan power).

$C_{2Q\,max} \cong 0{,}005 - 0{,}01\ 1/°\text{C}$

$C_{E\,min} \cong 0{,}001 - 0{,}002\ 1/°\text{C}$ which is roughly applicable for both evaporators as well as condensers.

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