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Analytical expressions for optimum flow rates in evaporators and condensers of heat pumping systems

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Dedicated to Professor Dr.-Ing. Dr.h.c.mult. Karl Stephan on the occasion of his 80th birthday.

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ABSTRACT

The flow velocities on the air or liquid side of evaporators and condensers in refrigerating or heat pump systems affect the system performance considerably. Furthermore the velocity can often be chosen rather freely without obvious first cost implications. The purpose of the paper is to show analytical relations indicating possible optimum operating conditions.

Considering a base case where the design data are known, simple analytical relations are deduced for optimum flow rates that will result in highest overall COP of the system when energy demand for the compressor as well as pumps or fans are included. This optimum is equivalent to the solution for minimum total energy demand of the system for a given cooling load. It is also shown that a different (and higher) flow rate will result in maximum net cooling capacity for a refrigerating system with fixed compressor speed.

The expressions can be used for design purposes as well as for checking suitable flow velocities in existing plants. The relations may also be incorporated in algorithms for optimal operation of systems with variable speed compressors.

Expressions analytiques pour déterminer les écoulements optimaux dans les évaporateurs et les condenseurs des systèmes à pompe à chaleur

Mots clés : Système frigorifique ; Pompe à chaleur ; Optimisation ; Débit ; Évaporateur ; Condenseur ; COP ; Ventilateur

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Nomenclature

- A analogy number $= f/(8 \cdot J)$
- A_w area for fluid flow, m^2
- A_Q heat transfer area, m^2
- C_T temperature dependence factor introduced in eq. 22, K^{-1}
- $C_{T1} = \frac{1-(T_{10}-T_{20}) \cdot k_E}{T_{10}-T_{20} \cdot (1-\eta_{Ct})}$ applies to condenser side, defined in eq. 22b
- $C_{T2} = \frac{T_{10}}{\eta_{Ct} \cdot T_{20}^2}$ applies to evaporator side, defined in eq. 22a
- COP coefficient of performance
- d hydraulic diameter, m
- E power demand, W
- f friction factor
- h heat transfer coefficient, W m^{-2} K^{-1}
- J Colburn factor $= St \cdot Pr^{2/3}$
- $k_q = \partial(Q_2/Q_{20})/\partial T_2$ = cooling capacity dependence of evaporating temperature, t_2, K^{-1}
- $k_E = \partial(\eta_{Ct}/\eta_{Ct}0)/\partial T_1$ Carnot efficiency dependence of condensing temperature, t_1, K^{-1}
- n exponent
- Pr Prandtl number
- Q cooling capacity, W
- St Stanton number $= h/(w \cdot \rho \cdot c_p)$
- T temperature, K
- t temperature, °C
- U overall heat transfer coefficient, W m^{-2} K^{-1}
- w velocity, m s^{-1}
- $\eta_{Ct} = COP_2/(T_2/(T_1-T_2))$ = total Carnot efficiency
- ρ density, kg m^{-3}
- θ temperature difference, K

Index:

- 0 refers to base case
- 1 refers to condenser side
- 2 refers to evaporator side
- h heat transfer (on air or brine side)
- c compressor
- COP_{max} conditions for maximum overall COP
- f friction factor
- in inlet (to heat exchanger)
- Is isentropic
- N net
- p pump or fan
- pm pump or fan including motor
- Q_{max} conditions for maximum net cooling capacity
- Si sink
- So source
- tot total
- TM average temperature (in heat exchangers logarithmic average)
- Tin inlet temperature

1. Introduction

Fig. 1a depicts a simple general air to air refrigerating system which is in focus for the present paper. The temperature differences in the evaporator and condenser have a significant influence on the performance of the system. For a system with given compressor and given designs of heat exchangers, the operating temperature differences are strongly affected by the air velocity, which often can be chosen rather freely. For instance, *increasing the velocity on the evaporator side* will increase the evaporating temperature, thus *increasing the capacity and the COP of the compressor* but the cost of this is a *higher power consumption of the evaporator fan*. It is obvious that there must be a velocity which gives the most favorable operation, representing an optimum velocity. The purpose of the paper is to derive relations for optimum velocity (or fan power), simple enough to be used in practice. Also in indirect systems, as shown in Fig. 1b, similar questions related to the influence of the flow rates arise, and similar relations can be derived.

The influence of the fans (and pumps) on the total operating energy of refrigerating systems has been treated by Granryd (1974, 1998) with similar methodology as here. Results from second law approach minimizing total exergy loss in heat exchangers are given for instance by Bejan, 1996, DeJong et al., 1997 and Kuenth, 1986. Examples of experimental results are given by Waller (1988).

The choice of fan speed (or flow rate) does only marginally influence the first cost of the system. Thus the question of optimal flow rates is mainly of a *technical* character and the solutions are not influenced by investment strategies, interest rates or energy prices. This simplifies the problem to derive *general relations for optimum flow velocity* for applications in refrigerating, air conditioning or heat pump systems.

2. An example

For the following treatment it is assumed that we have a given system design (comprising evaporator, compressor and condenser). In a “*base case*” the fluid velocity is chosen to a certain value (arbitrarily or by experience) and for a given application the operating temperatures depends on the performance of the evaporator and condenser.

Figs. 2 and 3 gives an example of temperature difference and the pressure drop versus air velocity based on relations for a fin coil evaporator with geometries as indicated (estimated by relations given by Granryd, 1964). Fig. 2 shows temperature differences that can be expected for different frontal air velocity if this evaporator is used in a system with given capacity, Q_2. Curves are shown for θ_M (the logarithmic *mean* air temperature difference) and θ_{in} (the difference between *incoming* air temperature and refrigerant saturation temperature at the exit of the coil). As expected, the inlet temperature difference, θ_{in}, is much more influenced by the air velocity than the logarithmic average, θ_M. Which temperature difference that is of interest depends on the application.

In Fig. 3 the estimated air side pressure drop versus air speed is shown for the same fin coil. In most cases the fan flow