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Thermal-hydraulic design of fan-supplied tube-fin condensers for refrigeration cassettes aimed at minimum entropy generation

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ABSTRACT

The method of minimum entropy generation is used to assess the effects of several design parameters on the performance of fan-supplied tube-fin condensers for light commercial refrigeration applications (heat transfer duty ~1 kW). A simplified mathematical model is put forward to simulate the thermal-hydraulic behavior of the condenser, and validated against experimental data obtained elsewhere, showing a good agreement between calculated and measured counterparts. The dimensionless rate of entropy generation due to heat flow across nonzero temperature difference and to viscous fluid flow is calculated for different condenser designs (number of fins and tubes, tube spacing and outer diameter) at fixed operating conditions (heat transfer duty, flow rates, inlet temperatures). It is shown that there do exist optima values for the face velocity, fin density, tube diameter and heat exchanger effectiveness that minimize the production of entropy, so that some design guidelines are proposed.

1. Introduction

In general, tube-fin heat exchangers consist of one or more rows of copper tubes with external aluminum fins through which air flows either by free convection or supplied by a fan. This type of heat exchanger has been widely used in many HVAC-R engineering applications, remarkably in light commercial refrigeration appliances (cooling capacities ~1 kW), which consume around 1% of the electricity produced worldwide [1]. In such applications, the refrigerant flows inside the tubes whereas air flows externally over the bundle of tubes and fins. Given the importance of small-capacity refrigerators for the global energy matrix, efforts have been devoted to reduce the energy consumption of such systems. For instance, Waltrich et al. [2] analyzed the impact of the components (compressor, condenser and evaporator) on both overall system cost and COP, concluding that the heat exchangers have a wide margin for performance improvement with little or no cost penalty.

The design of tube-fin heat exchangers involves both geometric (*e.g.* tube diameter, tube spacing, number of tube rows, fin thickness, shape and spacing) and operational parameters (*e.g.* flow rates and temperatures of the air and refrigerant streams) in order to accomplish a certain heat transfer duty at the penalty of fan pumping power. Some of these parameters affect quite significantly the thermo-hydraulic performance of the heat exchanger whilst others may be relaxed. In general, performance indicators (*e.g.* heat transfer rate and pumping power) respond similarly to changes in design parameters, *i.e.* a heat transfer enhancement (desired effect) is usually followed by an increase in pressure drop and then in pumping power (undesirable effects). Therefore, the heat exchanger should be designed in such a way as to balance the heat transfer enhancement and the pumping power increase trade-offs.

A method devised for this purpose consists of counteracting the thermodynamic losses associated with irreversible heat transfer across a finite temperature difference with the irreversibilities associated with viscous fluid flow [3]. As their effects on the rate of entropy generation oppose each other, there does exist a design (geometry and running conditions) that yields the overall entropy generation towards a minimum. This method, named *Entropy Generation Minimization* (EGM) after Bejan [3], has been widely used in geometric optimization of various types of heat exchangers [4–9].

However, studies of the application of the EGM to tube-fin heat exchanger design are scarce. For instance, Saechan and Wongwises [10] applied the EGM as the objective function to conduct a computational optimization of a tube-fin condenser for air conditioning applications (cooling capacity ~10 kW, air flow rate ~10,000 m^3/h). Recently, Pussoli et al. [11] employed the EGM to size peripheral fin heat exchangers, a new heat exchanger concept for household refrigeration applications (cooling capacity ~0.1 kW, air flow rate ~100 m^3/h). Nevertheless, it has not been found in the open literature studies focusing on the design and optimization of

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Nomenclature

Roman

A	heat transfer surface, m^2
C	thermal capacity, W/K
c_p	isobaric specific heat, J/kg K
D_t	tube outer diameter, m
f	friction factor
G	mass flux, kg/m^2s
h	specific enthalpy, J/kg
L	length, m
m	mass flow rate, kg/s
N_f	number of fins
N_{lo}	number of longitudinal tubes
N_S	dimensionless entropy generation
N_{tr}	number of transversal tubes
N_{tu}	number of transfer units
p	pressure, Pa
P_{lo}	longitudinal tube spacing, m
P_{tr}	transversal tube spacing, m
Q	heat transfer rate, W
Re	Reynolds number $(=\rho_a V_{face} D_t/\mu_a)$
s	specific entropy, J/kg K
S_{gen}	entropy generation rate, W/K
t	temperature, K
UA	conductance, W/K
V	velocity, m/s

Greek

ε	heat exchanger effectiveness
α	heat transfer coefficient, W/m^2 K
ρ	specific mass, kg/m^3
μ	viscosity, Pa s
δ_f	fin thickness, m
δ_t	tube thickness, m
ΔS	entropy variation rate, W/K

Subscripts

a	air side
cond	condensing temperature
evap	evaporating temperature
f	fin
face	heat exchanger face area
i	inlet
max	maximum
min	minimum
o	outlet
r	refrigerant side
sat	saturated
sub	subcooled
sup	superheated
t	tube
w	wall

tube-fin heat exchangers for light commercial refrigeration appliances (cooling capacity ~1 kW, air flow rate ~1000 m^3/h).

In this context, this paper is aimed at investigating the thermal-hydraulic performance of tube-fin condensers for light commercial applications with focus on the entropy generation minimization. For doing so, a mathematical model to simulate the thermal-hydraulic performance of tube-fin condensers operating with refrigerant R-134a as working fluid was developed and validated against experimental data obtained in the literature [12]. The model was then used to explore the effects of various geometric and operating parameters on the condenser performance.

2. Mathematical model

In tube-fin condensers, the refrigerant flows inside the tubes whereas air flows transversally through the bundle of finned-tubes, receiving heat from the refrigerant, as depicted in Fig. 1. The refrigerant enters the condenser as superheated vapor, it is then condensed and exits as subcooled liquid, as illustrated in Fig. 2. In both the refrigerant and air streams, irreversibilities due to heat transfer with finite temperature difference and fluid friction take place, so that the condenser model was divided into two sub-models, namely thermal and hydraulic, in order to account for both effects.

2.1. Thermal model

The thermal model is concerned with the calculations of the heat transfer rate, and the air and refrigerant temperatures at the outlet ports. The former is calculated separately for each flow region (superheated, saturated and subcooled), so that the overall heat transfer rate is obtained from:

$$Q = Q_{sup} + Q_{sat} + Q_{sub} = m_r(h_{r,i} - h_{r,o}) \tag{1}$$

Gonçalves et al. [13] showed that for high air flow rates, such as the figures found in refrigeration cassettes (~1000 m^3/h), the three-dimensional influence of the refrigerant circuit on the thermal performance (*i.e.* heat exchanger effectiveness) can be neglected. This means that the heat exchanger can be linearized for sake of heat transfer calculations, as depicted in Fig. 2. Thus, the heat transfer rates in each flow region can be calculated from:

$$Q_{sup} = m_r(h_{r,i} - h_v) = \varepsilon_{sup} m_r c_{p,r,v}(t_{r,i} - t_{a,v}) \tag{2}$$

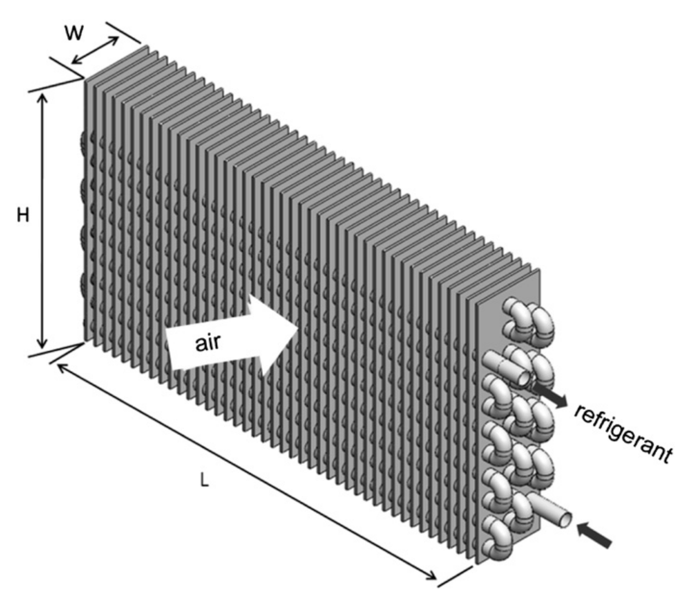

Fig. 1. Schematic representation of a tube-fin condenser for refrigeration cassetes.