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Method for determining the optimum number of circuits for a fin-tube condenser in a heat pump

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ABSTRACT

Fin-tube heat exchangers are widely used in air-source heat pumps and air-conditioners. Because these heat exchangers are comprised of a number of tubes, there are innumerable ways of organizing the refrigerant circuitry. It is important to determine the number of refrigerant circuits because the circuitry significantly influences the performance of the heat exchangers. A method for determining the optimum number of circuits is proposed in this work. This method is applied to determine the optimum number of circuits for a fin-tube condenser. The performance analyses of fin-tube condensers with various circuit configurations demonstrates that the new method is useful in determining the number of circuits required in a heat exchanger.

1. Introduction

The use of air-source heat pumps and air-conditioners has become popular in both developed and developing countries for space heating and cooling. It has been reported that approximately 40% of the energy consumption of buildings is attributed to air-conditioning and 20% of the total electricity is consumed for air-conditioning in developed countries. Due to this large consumption of energy, various efforts have been made to improve the performance and energy efficiency of air-source heat pumps and air-conditioners [1–4]. Because air-source heat pumps and air-conditioners utilize condensers and evaporators to reject and absorb heat to and from the surrounding areas, the performance of condensers and evaporators significantly affects the heat transfer capacity and energy efficiency of the systems. Accordingly, studies on heat exchangers have been, and continue to be, important to increase efficiency and reduce energy consumption.

Various studies on heat exchangers have been conducted focusing on the air-side performance [5], refrigerant-side performance [6,7], and circuitry design [8–11] of condensers and evaporators. Fin-tube heat exchangers are widely used in air-source heat pumps and air-conditioners. They consist of a number of heat transfer tubes and fins attached to the outside surface of the tubes. Because fin-tube heat exchangers consist of a large number of heat transfer tubes, there are innumerable tube combinations that can compose refrigerant circuitries. In general, the design of refrigerant circuitries requires a long time and high cost because it is conducted relying on the experience of the designer and repeated tests. Furthermore, a generally accepted analytic method to design refrigerant circuitry does not exist, and the optimal number of refrigerant circuits is determined based on the designers' experience rather than an analytic method.

The optimization of fin-tube heat exchangers' refrigerant circuitries has been attempted using various methods. Wang et al. [8] experimentally investigated the effect of the refrigerant circuitry on the fin-tube heat exchanger performance in a condenser that uses air as the heat source. They concluded that, although the performance increases when the refrigerant circuitry is counter-cross arranged, the increase in the performance may be offset by the thermal conduction through the fins. There has been a study that applied the genetic algorithm in order to optimize a refrigerant circuitry [9]. This algorithm can automatically construct every possible refrigerant circuitry in a simulation. This study searched for the refrigerant circuitry with the maximum heat transfer rate among the constructed refrigerant circuitries. This method requires a significant amount of time and it is difficult to confirm whether the circuitry can be practically applied or not. Liang et al. [10] proposed a model that can be used to investigate the heat transfer characteristics and performance evaluation of a refrigerant circuitry through exergy destruction analyses. Although exergy destruction analyses were used to determine which refrigerant circuitry was appropriate for certain operational conditions, it does not provide a direct answer to how many refrigerant

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Nomenclature

A	area (m^2)
D	diameter (m)
G	mass flux (kg m^{-2} s)
L	length (m)
N_p	circuit number
Pr	Prandtl number
$\dot{Q}$	heat transfer rate (W)
R	thermal resistance (m^2 K W^{-1})
$\dot{S}_{gen}$	entropy generation rate (J K^{-1} s^{-1})
Re	Reynolds number
T	temperature (K)
U	overall heat transfer coefficient (W m^{-2} K^{-1})
$\dot{V}$	volume flow rate (m^3 s^{-1})
c_p	specific heat at constant pressure (J kg^{-1} K^{-1})
h	convective heat transfer coefficient (W m^{-2} K^{-1})
k	thermal conductivity (W m^{-1} K^{-1})
p	pressure (N m^{-2})
pr	actual pressure/critical pressure
v	specific volume (m^3 kg^{-1})
x	thermodynamic vapor quality
η	overall surface efficiency
μ	viscosity (Pa s)

Subscripts

a	air
c	cool side
cont	contact
foul	fouling
gen	generation
i	inner
l	liquid
o	outer
r	refrigerant
w	wall

Table 1
Correlations used for simulations.

Items	Zone	Correlations
Heat transfer in matrix	Air side	Wang et al. [12]
	Single-phase refrigerant	Gnielinski [13]
	Two-phase refrigerant	Shah [14]
Pressure drop in matrix	Air side	Wang et al. [12]
	Single-phase refrigerant	Churchill [15]
	Two-phase refrigerant side	Friedel [16]
Pressure drop in U-bend	Single-phase refrigerant	Ito [17]
	Two-phase refrigerant	Chen et al. [18]

circuits should be used. A model that can be used to minimize the refrigerant-side exergy destruction of a condenser was presented using the entropy generation minimization theory [11]. Although this method is useful, the calculation process is complicated and the refrigerant circuitry cannot be designed intuitively.

Studies on heat exchangers have been conducted for numerous years. However, it is not yet clear how to obtain the optimal number of refrigerant circuits. In particular, there is a significant shortage of studies that consider the divergence and mergence of refrigerant circuits. In this paper, the existing evaluation methods for heat exchangers are discussed and a new method is proposed for determining the appropriate number of circuits. This method is applied to a fin-tube heat exchanger and compared with the simulation results in order to prove its validity. This method is also used to review where to diverge or to merge circuits and how many circuits should be diverged and/or merged.

2. Numerical simulation model

It is difficult to experimentally measure the performance of a large number of heat exchangers with different refrigerant circuitries due to the limitations of time and cost. In the present work, a numerical simulation model is used to investigate the performance of the heat exchangers.

In order to evaluate the performance of these condensers, correlations associated with heat transfers and pressure drops are necessary. The correlations used in the analysis are listed in Table 1. The analysis is conducted with five assumptions: (1) the effect of oil inside the tube is negligible, (2) the internal surface of the tube is smooth, (3) the flows of the working fluids are steady, (4) the longitudinal conductive heat transfer through the tubes and fins is negligible, and (5) the curved tube parts (U-bends) do not involve heat transfers but they involve pressure drops. The analysis was conducted such that the refrigerant mass flow rate of each circuit was determined in order to equalize the outlet pressure of each circuit while the total refrigerant flow rate and pressure at the entrance remained constant. Prior to using this numerical simulation model, the performance of some heat exchangers with different circuitries were experimentally measured and compared with the simulation results in order to examine the validity of the simulation model.

A representative fin-tube heat exchanger matrix was selected for this study. The selected heat exchanger matrix had a tube configuration of 22 steps and 2 rows. It was used as a heat pump and had a designed capacity of 3.5 kW. The detailed specifications of the heat exchanger matrix are presented in Table 2. Regarding the operation conditions, the median values of the operation parameters required in order to satisfy the load were selected and they are listed in Table 3. Thirty refrigerant circuitries were constructed; these are illustrated in Fig. 1. They are identified using the number of circuits and flow configuration as in Table 3. Seven refrigerant circuitries of 2CUe, 2CNe, 2PUe, 3CUV, 3CUA, and 4Cue were used to verify the simulation model.

Table 2
Specifications of the fin-tube heat exchanger matrix.

List	Value
Rated capacity	3.5 kW
Coil type	Fin-tube HX
Tube configuration	Staggered 22 steps × 2 rows
Tube length	653.2 mm
Tube OD	7 mm
Tube thickness	0.25 mm
Tube horizontal spacing	12.7 mm
Tube vertical spacing	21 mm
Tube material	Copper
Fin type	Louver
Fin thickness	0.1 mm
Fin spacing	18 fpi
Fin height	0.7 mm
Fin material	Aluminum