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Performance analysis of a refrigerating system with a grooved-tube evaporator

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HIGHLIGHTS

- A grooved-tube evaporator was designed and implemented in a refrigerator.
- Results were compared to those from a regular-tube evaporator.
- Heat exchanger effectiveness of the grooved-tube evaporator was found to be higher.
- COP of the refrigerator was improved with the grooved-tube evaporator.
- Grooves increased the pressure drop but negligibly compared to compressor pressure.

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ABSTRACT

A grooved-tube heat exchanger was developed and applied as an evaporator of a refrigerating system. Studies were carried out to evaluate the performance of the refrigerator. Isobutane (R-600a) was employed as the working fluid. Same studies were performed on the system but with the same size of a regular-tube evaporator. The two refrigerating systems were compared in terms of heat exchanger effectiveness, coefficient of performance (COP), and two-phase pressure drop. The heat exchanger effectiveness of the grooved-tube evaporator was found to be higher compared to the regular-tube evaporator. COP of the system with the grooved-tube evaporator was also observed to be higher. Although the pressure drop for the grooved-tube evaporator was slightly higher than that for the regular-tube evaporator, the compressor demand for this increase was not very significant.

1. Introduction

Demand for enhanced performance from heat exchangers is becoming significant in the cooling systems' industry. Since the larger wetted area in the tubes has the potential to promote heat transfer, internally grooved tubes have been attracting large attention in the area of air-conditioning and refrigeration. Rainieri et al. [1] studied the heat transfer coefficient and pressure drop for spirally enhanced tubes. Bore and envelope diameters, groove depth, helix pitch and type of corrugation (single helix or cross-helix) were the major parameters investigated. The results showed that in spirally enhanced geometries, the transition to turbulent flow can occur at Reynolds numbers lower than 2×10^3. Bergles [2] investigated enhanced surfaces for pool boiling and forced convection boiling/vaporization. Lines [3] studied the advantages of helically coiled heat exchangers. These heat exchangers were found to yield higher film coefficients. The study reported that effective usage of available pressure drop provides more efficient and less expensive systems. Also, helical geometry makes it possible to handle high temperatures and high temperature differences without strong induced stresses or expensive joints. Another study regarding these systems was performed on a double-pipe helical heat exchanger [4]. The study observed little difference between the overall heat transfer coefficients for parallel flow and counter-flow. Increasing the mass flow rate in both the inner tube and the annulus resulted in decreasing temperature uniformity in the other section.

Shikazono et al. [5] experimentally and numerically investigated the heat transfer characteristics of micro-grooved flat plate evaporators employing R-134a with the direction of refrigerant flow being parallel to the groove axis. The predicted heat transfer coefficient was found to be inversely proportional to the quality. Stephan and

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Busse [6] approached the flow in grooved pipes as a two-dimensional heat conduction problem. In addition to the temperature of the refrigerant, the shape of the liquid–vapor interface was taken into consideration in the study. It was reported that if the interface temperature was assumed to be equal to the saturation temperature of the vapor it could result in over prediction of the radial heat transfer coefficient. Suh and Park [7] analyzed the thermal performance of a micro heat pipe with trapezoidal grooves inside it. The grooves did not affect the pressure of the vapor. However the liquid pressure showed a significant drop. Kim et al. [8] studied and compared the critical heat flux (CHF) performance of flow boiling of the refrigerant R-134a in uniformly heated regular tubes and in rifled tubes positioned vertically. The varying parameters were the outlet pressures, mass fluxes and inlet subcooling temperatures of the test loop. The CHF for the rifled tubes was improved by 40–60% compared to the circular tube. The CHF enhancement ratio increased with increasing mass flux and pressure.

The friction factor and thermal stresses occurring around the grooves of internally enhanced heat exchangers should also be studied for cost-effectiveness and reliability. Yucel and Dinler [9] performed a numerical study of laminar and turbulent flows through a pipe having internal fins. The effects of fin heights and number of fins were studied for different fluids. The study found that for laminar flows, the mean Nusselt number decreased and the friction factor increased for increasing fin number. For turbulent flow conditions, both the mean Nusselt number and the friction factor increased with number of fins. It was also found that increase in the height of fins increased the friction factor for all Reynolds numbers studied. Webb et al. [10] investigated the heat transfer and friction characteristics of internal helical-grooved pipes. Number of ribs, helix angle and rib height were the parameters varied in the study. Correlations were developed for the heat transfer and friction. The tested tubes showed the characteristics of both rough tubes and internally finned tubes which provide enhancement by local flow separation and surface area increase. Ozceyhan and Altuntop [11] studied the heat transfer and thermal stress analysis in grooved tubes. The study found that in grooved tubes, the maximum thermal stresses were found to be almost twice that of circular tubes for varying fluid inlet velocities. The maximum thermal stresses occurred mostly close to the grooved parts of the tubes because of higher temperature gradient and they increased as the volumetric flow rate of the fluid increased. Kuntysh et al. [12] studied the energy efficiency improvement and material requirement of air-cooled heat exchangers having externally-grooved tubes. Experiments were performed for varying conditions. Results showed that the heat transfer per unit weight of a tube having ribs with a bent base crimpled into longitudinal grooves was 1.57 times higher than for rolled-in ribs and 1.38 times higher than that for knurled ribs. Kaji et al. [13] compared smooth-tube evaporators with two different grooved-tube evaporator configurations used in a CO_2 heat pump system. Heat transfer performance was enhanced by 17.7% with the herringbone grooved tubes with respect to the smooth tubes.

This present study investigates the fluid flow and thermal phenomena in the evaporator of a refrigeration system. An evaporator of the same size but having cross-helical grooved tubes were fabricated and used in the system. The studies were conducted on a domestic refrigerator. The results obtained from studying the two systems were compared in terms of evaporator effectiveness, system coefficient of performance (COP), and two-phase pressure drop.

2. Theory

Fig. 1 illustrates the two-phase flow model in a horizontal evaporator outlet of the refrigerating system. Separated flow model instead of the homogeneous model was employed due to the nature of the two-phase refrigerant flow being stratified. Following the concepts used by Chiou and Wang [14], Celik and Nsofor [15] and Wang [16], conservation equations for the refrigerant flow in the evaporator are developed.

Considering steady flow, the conservation of mass equations for the liquid and the vapor phases can be written respectively as shown in equations (1) and (2):

$$\frac{\partial}{\partial z}[(1-\alpha)\rho_l V] = \Gamma \tag{1}$$

$$\frac{\partial}{\partial z}(\alpha\rho_v V) = -\Gamma \tag{2}$$

These can be written in terms of mass flow rate as:

$$M_l + M_v = M(\mathrm{const.}) \tag{3}$$

The momentum equations are written respectively as:

$$-\frac{dp}{dz}(M_l v_l + M_v v_v) = M\frac{dE}{dz} + \frac{d}{dz}\left[\frac{1}{2}\left(M_l v_{z,l}^2 + M_v v_{z,v}^2\right)\right] \tag{4}$$

$$-\frac{dp}{dz} = \frac{1}{A}\frac{dF}{dz} + \frac{M^2}{A}\frac{d}{dz}\left[\frac{1}{A}\left(\frac{x^2 v_v}{\alpha} + \frac{(1-x)^2 v_l}{1-\alpha}\right)\right] \tag{5}$$

where α, void fraction, is:

$$\alpha = \frac{A_v}{A} \tag{6}$$

The mass flux, G, for the vapor and liquid phases are written respectively as:

$$G_l = G\left(1-x\right) = \frac{M_l}{A} \tag{7}$$

$$G_v = \frac{M_v}{A} \tag{8}$$

Superficial velocities for both phases are determined using:

$$v_{z,l} = \frac{G(1-x)}{\rho_l} \tag{9}$$

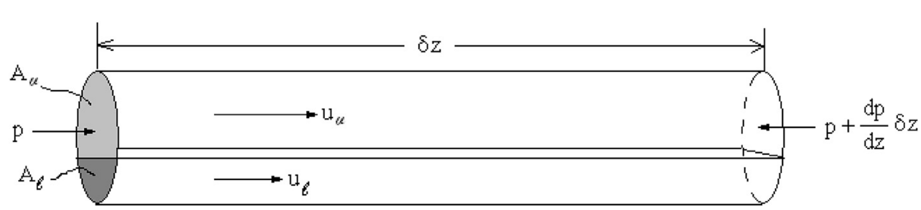

Fig. 1. Control volume for the two-phase flow field.