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HEAT TRANSFER ENHANCEMENT AND PRESSURE GRADIENT INCREASE DURING CONDENSATION IN A MICROFIN TUBE: A NEW APPROACH

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ABSTRACT

Experimental heat transfer coefficients during condensation of R134a in a microfin tube were reported by Cavallini et al. (2002). The test tube had a 7.69 mm inside diameter to the tip of the grooves and 60 fins with 0.2 mm fin height and 13° helix angle. Heat transfer measurements were compared against the heat transfer coefficient of an equivalent smooth tube under the same operating conditions, to show the heat transfer coefficient increase in the microfin tube as compared to the smooth tube.

It was found that the heat transfer enhancement depends on mass velocity and vapour quality: the maximum value of the enhancement factor for R134a at 40°C saturation temperature was obtained at 200 kg/(m^{2}s) mass velocity, where it reached up to 2.8. As the area enhancement for the test microfin tube was equal to 1.8 ($A_{MICROFIN}/A_{SMOOTH}$ = 1.8), it could be seen that heat transfer enhancement was not merely due to the area enhancement, and other effects were important to be accounted for. For values of mass velocity higher or lower than 200 kg/(m^2 s), the enhancement factor decreased when mass velocity increased or decreased.

It came out that there is an optimal value of mass velocity with respect to the heat transfer performance of a microfin tube. This may be due to the flow pattern in the tube. In fact, at low values of mass velocity, micro-fins could promote the annular flow pattern, leading the maximum enhancement for a certain value of the mass velocity.

More recently pressure drop data have also been taken during two-phase adiabatic flow of R134a inside the same microfin tube. Pressure change in the tube was measured by means of a differential pressure transducer. The test section was 1.45 m long.

The measured frictional pressure gradient is reported in this paper and compared to the pressure gradient in the smooth tube at the same operating conditions. In the end, by adopting a microfin tube to replace a plain tube, higher heat transfer coefficients are obtained at the expense of a higher frictional pressure gradient. However, there is no need of running the condenser at the same value of mass velocity within smooth and microfin tubes. More conveniently, when a different tube is adopted, the heat exchanger should be optimized and the mass velocity in the tube should be adequately chosen, to balance the heat transfer enhancement and the pressure drop increase.

As shown by Cavallini et al. (2000), from an energy performance point of view, what is important to consider is not the frictional pressure gradient itself, but the associated saturation temperature drop Δt_{sat} in the length of tube necessary for complete condensation of the refrigerant. This term in fact adds up to the compression head in the refrigeration cycle.

The advantage of using a microfin tube instead of a plain tube can be enlightened by using the *Penalty Factor* of a condensation process: this parameter was recently proposed by Cavallini et al. (2000), as a benchmark for performance comparison among different fluids in two-phase heat transfer processes. The *Penalty Factor* accounts for the saturation temperature change in the heat exchanger. The same parameter can be used to compare different tubes.

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