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THE BEHAVIOR OF LATERAL AND CONCENTRIC CAPILLARY TUBE-SUCTION LINE HEAT EXCHANGERS USING CFC-12 AND HFC-134a

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ABSTRACT

This work presents a mathematical model for the flow through non-adiabatic capillary tubes. Lateral and concentric capillary tube-suction line heat exchangers are considered. Results for numerical simulations using CFC-12 and HFC-134a as well as the characteristics and some preliminary data obtained on an experimental apparatus are presented.

NOMENCLATURE

α: void fraction.
Cp: specific heat.
CT: capillary tube.
D: diameter.
f: Darcy friction factor.
G: mass flux.
H: specific enthalpy.
HX: heat exchanger.
h: heat transfer coefficient.
L: length.
M: mass flow rate.
Nu: Nusselt number.
P: pressure.
Pr: Prandtl number.
Re: Reynolds number.
s: specific entropy.
SL: suction line.
UA': global heat transfer coefficient per unit of length.
v: specific volume.
X: mass quality.
z: capillary tube position.

Subscripts:

amb: ambient.
crit: critical condition.
ct: capillary tube.
ct1: 1st part of cap. tube.
ct2: 2nd part of cap. tube.
hx: heat exchanger.
i: inside, inlet.
l: liquid.
o: outside, outlet.
sl: suction line.
v: vapor.
w: wall.

INTRODUCTION

The expansion device used in household refrigerators and freezers is the non-adiabatic capillary tube. Lateral and concentric capillary tube-suction line heat exchangers are usually employed. As the current working fluid - CFC-12 - must be substituted possibly by HFC-134a, this device has to be redesigned.

In this way the authors are developing a theoretical and experimental study on the behavior of non-adiabatic capillary tube. Some data and numerical results obtained so far are presented. For the case of concentric-tubes HX, no published data have been found.

LITERATURE REVIEW AND NUMERICAL MODEL

The variables that affect M_{ct} are: $P_{i,ct}$, $T_{i,ct}$ (or $X_{i,ct}$), $P_{o,ct}$ (low pressure reservoir), $T_{i,sl}$, $D_{i,ct}$, $D_{o,ct}$, L_{ct1}, L_{hx}, L_{ct2}, D_{sl}, capillary tube roughness, heat transfer to or from the environment and type of heat exchanger used. Figures 1 and 2 show sketches of the heat exchangers treated here.

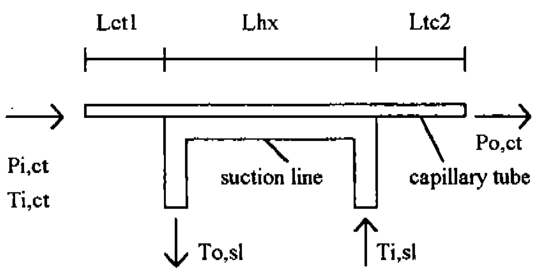

Figure 1: Longitudinal view of lateral heat exchanger.

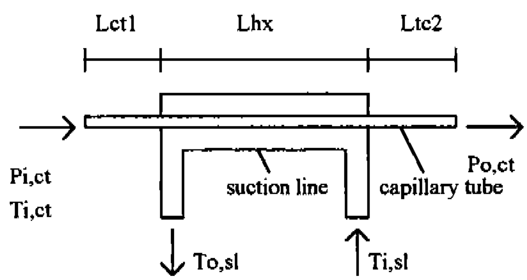

Figure 2: Longitudinal view of concentric-tubes HX.

When developing a mathematical model for the flow through capillary tubes, the first aspect to be considered is the two-phase flow model. For the case of capillary tubes, many technical papers suggest that the homogeneous model is sufficiently good (e.g., Mikol & Dudley, 1964 and Pate, 1982). Some results achieved with more detailed models (Li et al, 1991) confirm that assumption.

Another important matter is the model to be used for the friction factor. As will be shown below the authors have found experimentally that the Moody chart may be used for the liquid flow through capillary tubes, in the same way as for large bore pipes. This follows the results of Mikol (1963), among others. For homogeneous two-phase flow the same friction factor equation (liquid flow) may be used, provided that an equivalent mixture viscosity is available.

Several papers report that a delay of vaporization takes place almost always in adiabatic capillary tubes (e.g. Mikol & Dudley, 1964). For non-adiabatic conditions, one of the few data available is due to Pate (1982), who did not find any evidence of this phenomenon.

For the CT two-phase flow heat transfer coefficient no specific model has been found. In spite of that the difficulty is minimized when one observes that the global heat transfer coefficient is mainly influenced by the suction line side.

Considering these aspects, the main assumptions adopted for the mathematical model presented in this paper are: homogeneous two-phase flow, steady-state flow, stable equilibrium between vapor and liquid phases, pure fluid (no oil), horizontal capillary tube, no axial heat conduction through the walls. The refrigerant mass flow rate may be small, so heat transfer between the expansion device and the environment is considered. For concentric-tubes HX, the conservation equations - conservation of mass, momentum and energy in CT flow, conservation of energy in SL flow, energy balance for CT and SL walls - are presented below:

$$\frac{dP_{ct}}{dz} = -\frac{f_{ct} G_{ct}^2 v_{ct}}{2D_{i,ct}} - G_{ct}^2 \frac{dv_{ct}}{dz} \tag{1}$$

$$\frac{dH_{ct}}{dz} = -\frac{h_{ct} \pi D_{i,ct}\left(T_{ct} - T_{w,ct}\right)}{M_{ct}} - v_{ct} G_{ct}^2 \frac{dv_{ct}}{dz} \tag{2}$$

$$\frac{dT_{sl}}{dz} = \frac{h_{o,sl} \pi D_{i,sl}\left(T_{sl} - T_{w,sl}\right) - h_{i,sl} \pi D_{o,ct}\left(T_{w,ct} - T_{sl}\right)}{M_{sl} Cp_{sl}} \tag{3}$$

$$T_{w,ct} = \frac{h_{ct} D_{i,ct} T_{ct} + h_{i,sl} D_{o,ct} T_{sl}}{h_{ct} D_{i,ct} + h_{i,sl} D_{o,ct}} \tag{4}$$

$$T_{w,sl} = \frac{h_{o,sl} \pi D_{i,sl} T_{sl} + UA'_{amb,sl} T_{amb}}{h_{o,sl} \pi D_{i,sl} + UA'_{amb,sl}} \tag{5}$$

where, on the equations above H_{ct} and v_{ct} are average values weighted by the mass fractions of the phases on the considered section. These equations are valid both for liquid and two-phase flow.

For calculation of the thermodynamic properties of CFC-12 and HFC-134a Martin-Hou equations are used. For the transport properties the data of ASHRAE Handbook (1993) are conveniently interpolated.

The coefficient h_{ct} for the liquid flow is computed by the Dittus-Boelter equation:

$$Nu_{ct} = 0.023 Re_{ct}^{0,8} Pr_{ct}^{n} \tag{6}$$

where, n is 0.4 (heating) or 0.3 (cooling). A modified Dittus-Boelter equation is used for the two-phase flow, with the average mixture velocity and liquid properties (Pate,1982):

$$Nu_{ct} = \frac{h_{ct} D_{i,ct}}{k_{l,ct}} = 0.023 Re_{l,ct}^{0,8} Pr_{l,ct}^{n} \left(\frac{1 - x_{ct}}{1 - \alpha_{ct}}\right)^{0,8} \tag{7}$$

where, for homogeneous two-phase flow:

$$\alpha_{ct} = \frac{x_{ct} v_{v,ct}}{v_{ct}} \tag{8}$$

The isentropic homogeneous equilibrium model is chosen to calculate G_{crit}, since the homogeneous two-phase flow was adopted earlier. It should be pointed out that the mathematical model presented here does not use the linear quality model assumption, because this is not really a physical constraint. Similar equations were obtained for lateral HX but they are not presented here.

Using the mathematical model presented above an algorithm for design (calculation of L_{ct}) and simulation (calculation of M_{ct}) has been developed. In the design version the differential equations are integrated while $P_{o,ct}$ or critical conditions are not achieved. When one of these conditions is obtained the length of the CT is determined. In the simulation version, two variables must be converged, $T_{o,sl}$ and M_{ct}. The calculation is time consuming in some cases, but the convergence is unconditional. On intermediate calculations SL temperatures may decrease to excessively small values, outside the validity range of the property equations and this aspect should be considered when developing the algorithm.

EXPERIMENTAL APPARATUS

In order to validate the numerical model an experimental apparatus was built. The unit runs in a batch process since the refrigerant is stored in a high pressure reservoir and transferred to a low pressure reservoir through the capillary tube to be tested. The time available for the test is limited by the capacity of the reservoirs.

The variables measured are: $P_{i,ct}$, $P_{o,ct}$, $T_{i,ct}$, capillary wall temperatures and SL temperatures along the HX, $T_{i,sl}$, $T_{o,sl}$, M_{ct}, M_{sl}. In order to measure the profile of SL temperatures, air was the fluid chosen to substitute the vapor of refrigerant into the SL. Considering also the difficulties imposed by the HX configuration, the SL had to be cut longitudinally and divided in two parts. The air mass flow rate and the SL diameter had to be modified because the heat transfer rate and the SL temperature profile should be the same as with the use of vapor of refrigerant.

The inside diameter was measured by two methods: filling the tube with Hg and by profile projection of some sections. The latter procedure was made in a sample of the same lot of the capillary tube tested. The difference between the two average values measured was about 1 %.

Using water into the CT, several pressure drop measurements were performed. For laminar flow, friction factors measured agree nicely with the theoretical ones. For turbulent flow, tests to achieve the equivalent roughness were performed. Geometrical measurements of the profile showed average values that agree closely with the ones hydrodynamically obtained.

Then the heat transfer coefficient in the annular passage (SL) was determined. Experimental results were up to 15 % greater than the ones obtained with the numerical model. This difference is acceptable in view of the uncertainty of the results.

Tests with refrigerants have been carried out. Tests have been performed only for adiabatic CT, so far.

NUMERICAL AND EXPERIMENTAL TESTS

Figure 3 presents numerical results for lateral and concentric tubes HX using CFC-12 and HFC-134a. The influence of the sub cooling degree is analyzed. Figure 4 presents a comparison between numerical and experimental results for adiabatic CT.

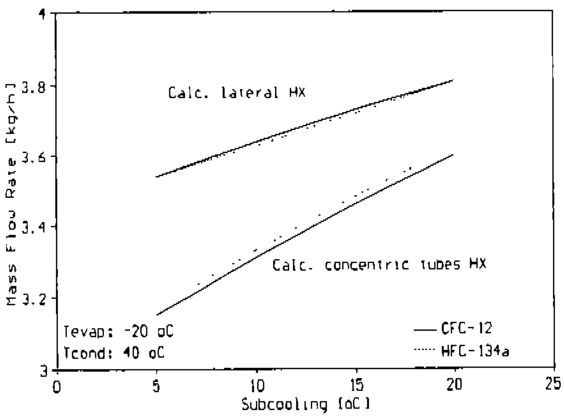

Figure 3 - Influence of sub cooling on the mass flow rate for two heat exchanger configurations and for CFC-12 and HFC-134a.

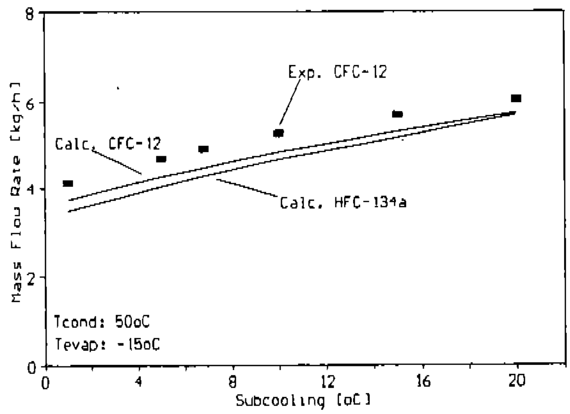

Figure 4 - Comparison between experimental and numerical results for an adiabatic capillary tube.

CONCLUSIONS

All the preliminary tests performed on the experimental apparatus have shown the reliability of the methods employed in the research.

For non-adiabatic CT some numerical results are presented. The effect of inlet sub cooling is analyzed and as expected mass flow rate increases as sub cooling is raised. For the cases analyzed, when CFC-12 is substituted by HFC-134a numerical results show a very slight decrease in the mass flow rate for the lateral HX and a slight increase for the concentric-tubes HX.

For adiabatic CT, numerical results for mass flow rate are about 10 % smaller than the experimental ones. This is possibly due to the delay of vaporization, which is not considered in the numerical model. When CFC-12 is substituted by HFC-134a, the mass flow rate decreases up to 7 % for the case here considered. That difference decreases for greater sub cooling degrees.

ACKNOWLEDGMENTS

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BIBLIOGRAPHY

1. ASHRAE Handbook of Fundamentals, 1993.
2. Collier, J.G., "Convective Boiling and Condensation", McGraw-Hill, 1981.
3. Kakaç, S. et al., "Handbook of Single-Phase Convective Heat Transfer", J.Wiley, 1987.
4. Li, R.Y. et al., "Numerical Modeling of Thermodynamic Non-Equilibrium Flow of Refrigerant Through Capillary Tubes", ASHRAE Trans., pp. 542-549, 1991.
5. Lin, S. et al., "Local Frictional Pressure Drop During Vaporization of R-12 Through Capillary Tubes", Int. J. Multiphase Flow, Vol.17, No.1, pp. 95-102, 1991.
6. Mikol, E.P. & Dudley, J.C., "A Visual and Photographic Study of the Inception of Vaporization in Adiabatic Flow", Journal of Basic Eng., pp. 257-264, Jun., 1964.
7. Mikol, E.P., "Adiabatic Single and Two-Phase Flow in Small Bore Tubes", ASHRAE J., Vol.5, No.11, Nov., 1963.
8. Pate, M.B., "A Theoretical and Experimental Analysis of Capillary Tube-Suction Line Heat Exchangers", PhD.Thesis, Purdue Univ., 1982.