Selection of the Capillary Tubes for Retrofitting in Refrigeration Appliances

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Abstract:

Retrofitting the refrigeration appliances using the capillary tube is reviewed in this paper. If the refrigerant is changed in the existing system, the COP of the system will be affected. It is necessary to do some modifications in the existing system and retrofit it. Changes in compressor design are too complicated and expensive. Hence modification in capillary tube dimensions is preferred so as to compensate for the performance of the system.

Keywords: Retrofitting, capillary tube.

Introduction:

To overcome Ozone depletion and global warming problems, replacement of chlorofluorocarbon refrigerant like R12 is required. If the refrigerants are changed in the existing VCRS system, the COP of the system will be affected thereby either reducing cooling effect or increasing the amount of compressor work. The present work is divided in two parts. First one is selection of refrigerant and another one is select proper size of capillary tube.

A) Selection of refrigerant:

There is no general rule governing the selection of refrigerants. It depends upon thermo physical properties, technological and economic aspects, safety and environmental factor, the capacity required for household refrigerators, which have refrigerating capacities of 400-500 W and electrical power input within 100-150 W range. The energy efficiency is far from that achieved using vapour-compression systems. The hydroflurocarbon R134a is non-flammable, difficult to synthesize, has zero ozone depletion and high global warming.

B. Tashtoush et al. (2002) carried out experimental study on the replacement of R12 in domestic refrigerators by new hydrocarbon/hydro fluorocarbon refrigerant mixtures. The results showed that butane/propane/R134a mixtures provide excellent performance parameters, such as coefficient of performance of refrigerator, compression power, volumetric efficiency, condenser duty, compressor discharge pressure and temperature. In addition, the results support the possibility of using butane/propane/R134a mixtures as an alternative to R12 in domestic refrigerators, without the necessity of changing the compressor lubricating oil used with R12.

Zhijing Liu et al. (2003) found that naturally occurring substances such as water, carbon dioxide, ammonia and hydrocarbons are believed to be environmentally safe refrigerants. With the CFC phase-out underway, interest in these environmentally safe refrigerants is growing. The thermodynamic properties of hydrocarbons, such as propane are similar to those of R12 and R22. Table 1 shows that hydrocarbons have lower viscosity and higher thermal conductivity compared to that of CFCs and HFCs. These superior transport properties are believed to contribute to the higher energy efficiency of hydrocarbons as compared to CFCs and HCFs. Table 2 shows that the global warming potential (GWP) of hydrocarbons such as propane (R290), n-butane (R600), and n-pentane (n-c5) is much lower than that of synthetic refrigerants. It also shows that the ozone depletion potential (ODP) of hydrocarbons is zero. Another advantage of hydrocarbons is their solubility in mineral oil, which is traditionally used as a lubricant in the compressors. They also tested the mixture R290/R600 as a drop-in substitute in a 20-cubic-feet, single-evaporator, and auto defrost, top mount, conventional domestic refrigerator/freezer. All the hardware remained the same, only the capillary tube was...
lengthened to achieve the optimum performance. The best result with an optimized R290/R600 blend was 6% savings compared to the baseline test with R12. The optimum performance of the modified unit yielded 14.6% and 16.7% energy savings with binary mixtures R290/n-c5, and R290/R600, respectively. A ternary mixture R290/R600/n-c5 with 17.3 % energy savings proved to be better than the binary mixtures. The superior transport properties of the hydrocarbon mixtures are believed to be responsible for their better test performance.

Table 1. Transport properties of select refrigerants

<table>
<thead>
<tr>
<th>Refrigerants</th>
<th>R12</th>
<th>R22</th>
<th>R123</th>
<th>R290</th>
<th>R600</th>
<th>n_c5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>35°C liquid (µPa/s)</td>
<td>182.0</td>
<td>151.7</td>
<td>386.3</td>
<td>87.19</td>
<td>144.6</td>
<td>200.0</td>
</tr>
<tr>
<td>-20°C vapor (µPa/s)</td>
<td>10.45</td>
<td>10.75</td>
<td>8.86</td>
<td>7.07</td>
<td>6.32</td>
<td>5.79</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>35°C liquid (mW/mK)</td>
<td>63.1</td>
<td>82.1</td>
<td>76.1</td>
<td>99.2</td>
<td>108.9</td>
<td>108.7</td>
</tr>
<tr>
<td>-20°C vapor (mW/mK)</td>
<td>7.89</td>
<td>8.90</td>
<td>7.33</td>
<td>14.38</td>
<td>12.93</td>
<td>11.67</td>
</tr>
</tbody>
</table>

Table 2. ODP and GWP of the select refrigerants

<table>
<thead>
<tr>
<th>Refrigerants</th>
<th>R12</th>
<th>R134a</th>
<th>R22</th>
<th>R123</th>
<th>R290</th>
<th>R600</th>
<th>n_c5</th>
</tr>
</thead>
<tbody>
<tr>
<td>ODP</td>
<td>1</td>
<td>0</td>
<td>0.05</td>
<td>0.02</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>*GWP</td>
<td>4500</td>
<td>420</td>
<td>1600</td>
<td>90</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>*GWP of CO₂ =1: time base:100 years</td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>

Douglas G. Westra (2003) found that one option to developing new alternative refrigerants is to combine existing non-CFC refrigerant to form a non-Azeotropic mixture, with the concentration optimized for the given application so that system COP may be maintained or even improved. An Azeotrope is made up of two or more refrigerants and occurs only at a particular composition. An azeotrope behaves as a pure refrigerant, undergoing no temperature change during condensation and evaporation. The thermodynamic properties of an azeotrope are different than those of its two constituent fluids. Common azeotrope refrigerants are R-500 (73.8% R-12 and 26.2% R-152a) and R-502 (48.8% R-22 and 51.2% R-115).

Barbara Minor et al. (2010) found that HFO-1234yf is a low global warming potential refrigerant that has been selected as the preferred option to replace R-134a in automotive air conditioning system. M. Fatouh, et al. (2006) test liquefied petroleum gas (LPG) of 60% propane and 40% commercial butane as a drop-in substitute for R134a in a single evaporator domestic refrigerator with a total volume of 10 ft³ (0.283 m³). Continuous running and cycling tests were performed on that refrigerator under tropical conditions using various capillary tube lengths and various charges of R134a and LPG to satisfy the required freezer temperature of -12°C. The lowest electric energy consumption was achieved using LPG with combination of capillary tube length of 5 m and charge of 60 grams. This combination achieved higher volumetric cooling capacity and lower freezer temperature compared to R134a. Pull down time, pressure ratio and power consumption of LPG refrigerator were lower than those of R134a refrigerator by about 7.6%, 5.5% and 4.3%, respectively. Actual COP of LPG refrigerator was higher than that of R134a by about 7.6%. Lower on-time ratio and energy consumption of LPG refrigerator by nearly 14.3% and 10.8%, respectively, were obtained as compared to those of R134a.

Maclaine-cross et al. (1995) state that LPG refrigerant occur naturally, cause no ozone depletion and negligible global warming. R290 can replace R22 and LPG mixtures replace R12 and R 134a. It leads to lower vapour pressure and 10 to 20% energy savings. Mani K. et al. (2009) carried out an experimental study on the performance of environmentally friendly refrigerants such as R134a, LPG and LPG/R134a mixture with and without the effect of magnetic field as suitable replacements for the refrigerant R12 in a vapour compression refrigeration system.
Lee et al. (2005) concluded that ternary mixtures composed of HFCs and HCs can be used to develop the HCFC-22 alternative refrigerant mixtures. They selected HFC-32, HFC-125, HFC-134a, HFC-143a, HFC-152a, HFC-227ea, and HFC-236ea as HFCs, and propane and isobutane as HCs. The simulator which could predict theoretically the performance of given refrigerant mixtures was developed and tested for various refrigerant systems. Nineteen different kinds of ternary mixtures were chosen for thermodynamic simulation. Among nineteen mixtures, six ternary refrigerant mixtures were selected as candidates for HCFC-22 alternatives. They were R-32/143a/600, R-32/152a/227ea, R-32/134a/236ea, R-32/143a/236ea, R-32/152a/236ea, and R-32/134a/600a. Performance of these mixtures was obtained experimentally by the thermodynamic calorimeter and was compared with that of HCFC-22, R-407C and R-410A.

B) Selection of capillary tube:

Bansal et al. (1998) worked for homogeneous two-phase flow model, CAPIL, which was designed to study the performance of adiabatic capillary tubes in small, vapour compression refrigeration systems, in particular household refrigerators and freezers. The model was based on the fundamental equations of conservation of mass, energy and momentum that are solved simultaneously through iterative procedure and Simpson's rule. The model uses empirical correlations for single-phase and two-phase friction factors and also accounts for the entrance effects. The model uses the REFPROP data base where the Carnahan-Starling-DeSantis equation of state is used to calculate the refrigerant properties. The model includes the effect of various design parameters, namely the tube diameter, tube relative roughness, tube length, level of subcooling and the refrigerant flow rate. The model is validated with earlier models over a range of operating conditions and is found to agree reasonably well with the available experimental data for HFC-134a.

Jiraporn Sinpiboon et al. (2002) developed a mathematical model to study flow characteristics in non-adiabatic capillary tubes. The theoretical model is based on conservation of mass, energy and momentum of fluids in the capillary tube and suction line. The mathematical model is categorized into three different cases, depending on the position of the heat exchange process. The first case is considered when the heat exchange process start in the single-phase flow region, the second case is determined when the heat exchange process start at the end of the single-phase flow region, and the last case is considered when the heat exchange process takes place in the two-phase flow region. A set of differential equations is solved by the explicit method of finite-difference scheme. The model is validated by comparison with the experimental data working with alternative refrigerants for design and optimization.

Dongsso Jung et al. (2006) modelled pressure drop through a capillary tube in an attempt to predict the size of capillary tubes used in residential air conditioners. They provided simple correlating equations for practicing engineers. Stoecker's basic model was studied with the consideration of various effects due to subcooling, area contraction, and different equations for viscosity and friction factor, and finally mixture effect. McAdams' equation for the two-phase viscosity and Stoecker's equation for the friction factor yielded the best results among various equations. With these equations, the modified model yielded the performance data that are comparable to those in the ASHRAE handbook. After the model was validated with experimental data for CFC12, HFC134a, HFC22, and R407C, performance data were generated for HCFC22 and its alternatives such as HFC134a, R407C and R410A under operating conditions such as condensing temperature (40, 45, 50, 55°C), subcooling (0, 2.5, 5°C), capillary tube diameter (1.2±2.4 mm), and mass flow rate (5±50 g/s). These data showed that the capillary tube length varies uniformly with the changes in condensing temperature and subcooling. Finally, a regression analysis was performed to determine the dependence of mass flow rate on the length and diameter of a capillary tube, condensing temperature, and subcooling. Simple practical equations yielded a mean deviation of 2.4% for 1488 data obtained for two pure and two mixed refrigerants.

Kim et al. (2000) experimentally investigated the capillary tube performance for R-407. The experimental setup is a real vapor compression refrigerating system. Mass flow rate is measured for various diameters and lengths while inlet pressure and degree of sub cooling are changed. These data are compared with the results of numerical model. The mass flow rate of the numerical model is about 14% less than the measured mass flow rate. They found that mass flow rate and length for R-407C are less than those for R-22 under same conditions. They used the numerical method for finding the diameter and length by using the continuity equation, momentum equation and energy equation.

Mittal et al. (2009) investigated the effect of coiling on the flow characteristics of R-407C in an adiabatic spiral capillary tube. The characteristic coiling parameter for a spiral capillary tube is the coil pitch. The effect of the coil pitch on the mass flow rate of R-407C was studied on several capillary tube test sections. They observed that the coiling of the capillary tube significantly reduced the mass flow rate of R-407C in the adiabatic spiral capillary tube. In order to quantify the effect of coiling, they conducted experiments for straight capillary tube and observed that the coiling of the capillary tube reduced the mass flow rate in the spiral tube in the range of 9–18% as compared with that in the straight capillary tube. A generalized non-dimensional
correlation for the prediction of the mass flow rates of various refrigerants was developed for the straight capillary tube on the basis of the experimental data of R-407C, R-134a, R-22, and R-410A measured by other researchers. Additionally, a refrigerant-specific correlation for the spiral capillary was also proposed on the basis of the experimental data of R-407C.

Akash Deep Singh (2009) developed a mathematical model of diabatic capillary tube. The mathematical model has been developed by using equations of conservation of mass, momentum and energy for predicting the length of diabatic capillary tube. Moody (1944) correlation is used to calculate the friction factor. McAdams et al. (1942) viscosity correlation has been used to evaluate the two-phase viscosity of the refrigerant. Input parameters have been taken from the data of Mendoca et al. (1998). Further, a geometric model is developed in Pro-E and the mesh is created in Ansys ICEM CFD and analysis is carried out in Ansys CFX which has three modules CFX-Pre, Solver and CFX-Post.

McAdams et al. (1998) concluded that some parameters such as friction factor, dryness fraction and Reynolds number affect the required length and diameter of a capillary tube for a given refrigeration capacity. The friction factors were based on the dryness fraction ranges between 0.05 - 0.85% and 0.5 - 1.9% above that of ASHRAE under the same conditions. Furthermore, both McAdams’ and Duckler’s equations for two-phase viscosity were employed so that the deviation in the estimated lengths could be compared. The tube length generated by combining various friction factor models with McAdams’ equation are much closer to that of ASHRAE standard than those of Duckler’s equation. The estimated lengths using McAdams’ and Duckler’s equations exceeded ASHRAE standard by 1.65% and 4.13% respectively. The required capillary tube length for a specified condenser condition depends on both Reynolds number and dryness fraction and not on either alone and these two factors should not be in exponential form. The generated lengths approach the ASHRAE requirement as the degree of sub cooling is increased.

M.A. Akintunde (2008) investigated the effects of various geometries of capillary tubes based on the coil diameters and lengths alone, with no particular attention placed on the effect of coil pitch. This paper examined the effects of pitches of both helical and serpentine coiled capillary tubes on the performance of a vapor compression refrigeration system. Several capillary tubes of equal lengths (2.03 m) and varying pitches, coiled diameters and serpentine heights were used. Both inlet and outlet pressure and temperature of the test section (capillary tube) were measured and used to estimate the COP of the system. In the case of helical coiled geometries, the pitch has no significant effect on the system performance. In the case of serpentine geometries, both pitch and height affect the system performance. Performance improves with both increase in the pitch and the height. Correlations were proposed to describe relationships between straight and coiled capillary tube and between helical coiled and serpentine coiled capillary tubes. The COP obtained was 0.9841 for mass flow rates of helical and serpentine with straight tubes, 0.9864 and 0.9996 for mass flow rates of serpentine and helical coiled tube respectively. This study investigated the performance of capillary tube geometries having R-134a as the working fluid.

Corberan et al. (2008) found a numerical method used to calculate mass flow rate in a capillary tube. The proposed method solves the conservation laws (continuity, momentum and energy) in 1D mesh. An iterative process is performed for a guessed value of the mass flow rate and it is followed until critical flow conditions are obtained. The resulting length is compared with the capillary tube length and a new guess of the mass flow rate is imposed. The iteration is repeated until convergence. In two phase flow, a separated flow model is assumed. Both, two-phase friction factor and viscosity models were determined by Lin correlation and void fraction from Zivi correlation. This model is included in ‘IMST-ART’, software for simulation and design of refrigeration equipment. The addition of capillary tube model allows calculating the superheat at the evaporator giving the capillary tube geometry. A simulation with different operative conditions and capillary tube geometry is presented and the results are compared with those given by ASHRAE correlation with the integration of the conservation equations (mass, momentum and energy) over individual control volumes. The model includes non critical and critical flow conditions. The model assumes thermodynamic equilibrium and one-dimensional two-phase flow. The metastable condition is not taken into account.

Peixoto et al. (2007) concluded that the finite difference model shows reasonably good agreement with published data for both adiabatic and diabatic capillary tubes. The agreement is better than obtainable with the widely-used charts published in the ASHRAE handbook for CFC-12 and HCFC-22. This model can be used to predict the performance of alternative refrigerants. To develop a test matrix for the required validation experiments, the model was run in both design and simulation mode for a wide range of operating conditions. The mass flow rate of HFC-134a is expected to be about 5% lower than CFC-12 in an adiabatic capillary tube. Adding a capillary tube-suction line heat exchanger increases mass flow by about 20% for both refrigerants. For a given mass flow rate, the calculations suggest that HFC-134a will require substantially a smaller suction line diameter and heat exchange length. Finally the model illustrates how the effectiveness of a capillary tube-
the range 30 to 500°C. These tests were conducted for three positions of the heat exchanging section and for each performed at evaporating temperatures in the range -5 to -25 °C corresponding to condensing temperatures in another of the capillary tube. The length of the heat exchange region was fixed at 1000 mm. Tests were varying evaporator and condenser pressures and with the heat exchange region changeable from one end to and flow condition.

Experimental data was supplemented by numerical data for R22, R410A and R407C. The relations show that the mass flow rate is the power function of the geometries. Based on the approximate analytic solutions, influences of geometrical parameters (inner diameter and length) and inlet operating parameters (pressure, subcooling or quality) on the mass flow rate through an adiabatic capillary tube have been intensively studied in this work. Some simple theoretical relations were developed. The relations show that the mass flow rate is the power function of the geometries. Experimental data was supplemented by numerical data for R22, R410A and R407C.

Chun-Lu Zhang et al. (2000) had done parameter analysis, an insight into the flow characteristics of capillary tubes. Based on the approximate analytic solutions, influences of geometrical parameters (inner diameter and length) and inlet operating parameters (pressure, subcooling or quality) on the mass flow rate through an adiabatic capillary tube have been intensively studied in this work. Some simple theoretical relations were developed. The relations show that the mass flow rate is the power function of the geometries. Experimental data was supplemented by numerical data for R22, R410A and R407C.

Worachest et al. (2004) provided new selection charts for the sizing of adiabatic capillary tubes operating with alternative refrigerants. The mathematical model is based on conservation of mass, energy, and momentum of fluids in the capillary tube. After the developed model is validated by comparison with the experimental data reported in literature, selection charts that contain the relevant parameters are proposed for sizing adiabatic capillary tubes. The selection charts are presented for some alternative refrigerants and a wide range of operations. These newly developed selection charts can be practically used to select capillary tube size from the flow rate and flow condition or to determine mass flow rate directly from a given capillary tube size and flow condition.

Christensen (1967) performed a series of tests with a capillary tube-suction line heat exchanger at varying evaporator and condenser pressures and with the heat exchange region changeable from one end to another of the capillary tube. The length of the heat exchange region was fixed at 1000 mm. Tests were performed at evaporating temperatures in the range -5 to -25 °C corresponding to condensing temperatures in the range 30 to 50°C. These tests were conducted for three positions of the heat exchanging section and for each test the refrigerant (CFC-12) mass flow rate was measured. It was verified that the evaporating pressure has little influence while the condensing pressure has a great effect on the refrigerant mass flow rate. Mass flow rates increased as the free length of the capillary tube was moved from the condenser towards the evaporator for fixed condenser and evaporator pressures.

Pate and Tree (1983, 1984 and 1987) publications were based in the Pate's Ph.D. research (1982) on the flow of CFC-12 though capillary tube-suction line heat exchanger. These papers described the experimental apparatus used, the results of the tests performed, a mathematical model for simulation and a study of the choked flow conditions at the capillary tube outlet. Model results for length calculations showed fair agreement with the experimental data. However to simplify the computational procedure, the authors assumed a linear profile of the quality in the heat exchanger region. As pointed out by Yan and Wang (1991), this assumption simplified the calculation but the actual refrigerant quality distribution in the capillary tube depends on the rate of heat exchange with the suction line.

The interaction between the capillary tube-suction line heat exchanger and the whole system was studied by Pereira et al. (1987). They analyzed experimentally the thermal performance of a domestic refrigerator, using CFC-12 as working fluid, as the heat exchanging conditions at the capillary tube and the
suction line were modified. They documented the improvement in overall performance of the refrigerator and verified that the effect of the heat exchange with the surrounding air was negligible. The ct-slhx simulation presented by Q.Yan and X.L.Wang (1991) was based on the fundamental fluid dynamics and heat transfer equations. The results for mass flow rate, quality profile for CFC-12 for a fixed geometry over a range of inlet and outlet pressures and inlet subcooling were obtained. They showed that small changes in mass flow can produce large changes in the quality along the slhx length.

Domanski et al. (1992) presented a theoretical evaluation of the performance effects resulting from the addition of a liquid line-suction line heat exchanger to a standard vapor compression refrigeration cycle. It demonstrated installation of a liquid line-suction line heat exchanger for improvement in COP and volumetric capacity in case of fluids that perform poorly in the basic cycle. The benefits obtained depend on the combination of the operating conditions and fluid properties with the liquid and vapor specific heats being the most important. The improvement with HFC-134a is 2.5% higher than for CFC-12. The results presented are for ideal cases and do not consider some factors which present in the real systems and affect their performance. For household refrigerators, the heat exchange is not between the liquid line and the suction line but between the capillary tube, where the refrigerant is flashing, and the suction line. The usual approach of idealizing the ct-slhx as an isobaric non adiabatic section followed by an isenthalpic expansion is a simplification of the real process (A. D. Little, 1982 and ASHRAE, 1988).

Conclusions

Earlier studies have shown that use of alternative refrigerants play an important role in forming problems such as global warming and ozone depletion. The coefficient of performance of refrigeration appliances improves in case of retrofitting the capillary tube. It is possible to obtain the effective size (diameter & length) of capillary tube by using of mathematical techniques and by maintaining proper pressure equalization between condenser and evaporator.

References