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A Review on Predicting the Refrigerant Flow Characteristics

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Abstracts

A comprehensive review of the literature on the flow of various refrigerants through the capillary tubes of different geometries viz. straight and coiled and flow configurations viz. adiabatic and diabatic, has been discussed in this paper. This paper give a brief study about throttling expansion of refrigerant, capillary tube, refrigerant property and mathematical correlation for the flow through the capillary tubes of different geometries operating under adiabatic and diabatic flow conditions

Keywords: Refrigeration

Introduction

In the global scenario for the environment preservation, it is bringing new challenges to the design of refrigerationsystems. The research community are making efforts which focused on improving the energy efficiency of systems (in order toreduce the power consumption) and on replacing the harmful artificial working fluids to environmentally friendly refrigerants

Throttling expansion of refrigerant

Similar to liquids, gases can also be expanded from high pressure to low pressure either by using a turbine (isentropic expansion) or a throttling device (isenthalpic process). Similar to throttling of liquids, the throttling of gases is also an isenthalpic process. Since the enthalpy of an ideal gas is a function of temperature only, during an isenthalpic process, the temperature of the ideal gas remains constant. In case of real gases, whether the temperature decreases or increases during the isenthalpic throttling process depends on a property of the gas called Joule-Thomson coefficient, μ_{JT} is given by[1]

$$\mu_{JT} = \left(\frac{\partial T}{dP}\right)_h$$

From thermodynamic relations it can be shown that the Joule-Thomson coefficient, μ_{JT} is equal to:

$$\mu_{JT} = \frac{\left[T\left(\frac{\partial v}{\partial T}\right)_p - v\right]}{C_p}$$

Where 'v' is the specific volume and C_p is the specific heat at constant pressure. From the above expression, it can be easily shown that μ_{JT} is zero for ideal gases (pv = RT). Thus the magnitude of μ_{JT} is a measure of deviation of real gases from ideal behavior. From the definition of μ_{JT}, the temperature of a real gas falls during isenthalpic expansion if μ_{JT} is positive, and it increases when μ_{JT} is negative.

Vapour compression refrigeration systems

These systems belong to the general class of vapour cycles, wherein the working fluid (refrigerant) undergoes phase change at least during one process. In a vapour compression refrigeration system, refrigeration is obtained as the evaporation of refrigerant take place at low temperatures. Mechanical energy is required to run the compressor. Vapour compression refrigeration systems are suitable for the refrigeration capacities ranging from few Watts to few megawatts. A wide variety of refrigerants can be used in these systems to suit different applications, capacities etc. The actual vapour compression cycle is based on Evans-Perkins cycle, which is also called as reverse Rankine cycle.[2-3]

Capillary Tube:

The expansion devices used in refrigeration systems can be divided into fixed opening type or variable opening type. As the name implies, in fixed opening type the flow area remains fixed, while in variable opening type the flow area changes with changing mass flow rates. Capillary tube and orifice belong to the fixed opening type. A capillary tube is a long, narrow tube of constant diameter. [3] The word "capillary" is a misnomer since surface tension is not important in refrigeration application of capillary tubes. Typical tube diameters of refrigerant capillary tubes range from 0.5 mm to 3 mm and the length ranges from 1m to 6 m.

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The refrigerant has to overcome the frictional resistance offered by tube walls. This leads to some pressure drop, and the liquid refrigerant flashes (evaporates) into mixture of liquid and vapour as its pressure reduces. The density of vapour is less than that of the liquid. Hence, the average density of refrigerant decreases as it flows in the tube. The mass flow rate and tube diameter (hence area) being constant, the velocity of refrigerant increases since m = ρVA. The increase in velocity or acceleration of the refrigerant also requires pressure drop.[4]

Several combinations of length and bore are available for the same mass flow rate and pressure drop. However, once a capillary tube of some diameter and length has been installed in a refrigeration system, the mass flow rate through it will vary in such a manner that the total pressure drop through it matches with the pressure difference between condenser and the evaporator. Its mass flow rate is totally dependent upon the pressure difference across it; it cannot adjust itself to variation of load effectively.

Selection of Capillary Tube:

The both the diameter and length of capillary tube have to be selected such that the balanced point can be achieved by the compressor and the capillary tube at the desired evaporator temperature.[5] For mass production a tube longer tube is installed with the expected result that evaporating temperature will be lower than expected while the tube is shortened to achieve the desired balance point. On the other hand for a single system the tube of slightly shorter length than the design length is chosen.

The coiling of the capillary tube significantly reduced the mass flow rate in the adiabatic spiral capillary tube. In case of straight capillary tube, the coiling of the straight 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. [M. K. Mittal et al. (2009)] A generalized non dimensional correlation for the prediction of the mass flow rates of various refrigerants has been developed for the straight capillary tube for the R-407C, 134a, R-22, and R-410A measured. The coil pitch is the also significantly influences the mass flow rate of through the adiabatic spiral capillary tube. The effect of providing pressure taps on the capillary tube surface has a negligible effect on the mass flow rate through the capillary tube. For the different refrigerants M.K. Khan et al. (2008) concluded that the effect of coiling of capillary tube reduces the mass flow rate by 5–15% as compared to those of the straight capillary tube operating under similar conditions but for different refrigerant.[6]

The analysis of flow through a capillary tube

In a capillary tube the flow is actually compressible, three-dimensional and two-phase flow with heat transfer and thermodynamic meta-stable state at the inlet of the tube. However, in the simplified analysis, the assumption is for steady, one-dimensional and in single phase or a homogenous mixture. Mass and momentum conservation for a control volume

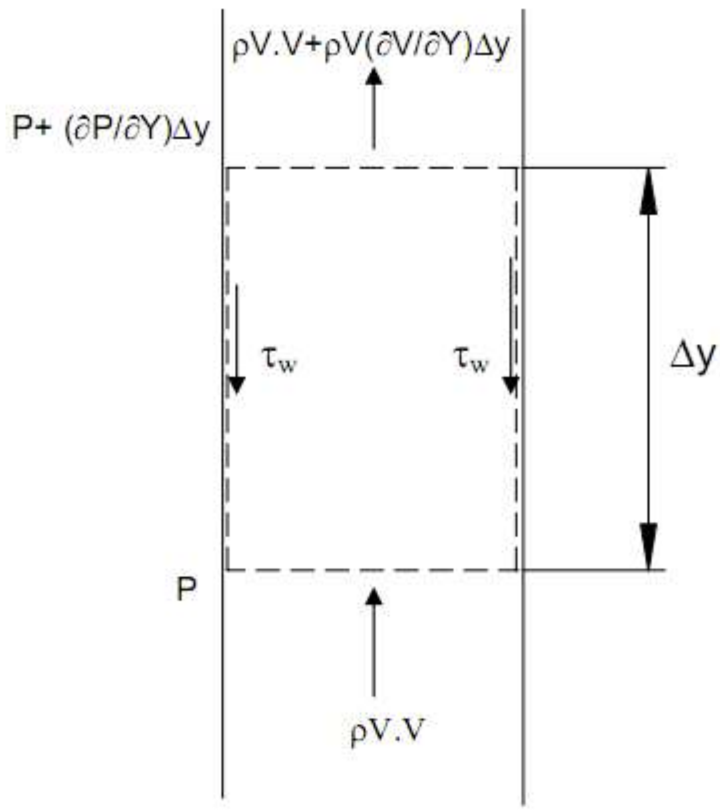

Figure 1 A small section of a capillary tube considered for analysis

$$\rho V = Const$$

Momentum Conservation:

The momentum theorem is applied to the control volume. According to this

$[\text{Momentum}]_{out} - [\text{Momentum}]_{in}$ = Total forces on control volume

$$\pi R^2\left[\rho V^2 + \rho V\frac{\partial v}{\partial y}\Delta y\right] - \pi R^2[\rho V^2] = -\pi R^2\frac{\partial p}{\partial y}\Delta y - \rho_{avg} g\pi R^2\Delta y - 2\pi R\Delta y\tau_w$$

At the face y + Δy, Taylor series expansion has been used for pressure and momentum and only the first order terms have been retained. The second order terms with second derivatives and higher order terms have been neglected. If the above equation is divided by $\pi R^2\Delta y$ and limit $\Delta y \to 0$ is taken; then all the higher order terms will tend to zero if these were

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included since these will have Δy or its higher power of Δy multiplying them. [7-10]Also, ρ_{avg} will tend to ρ since the control volume will shrink to the bottom face of the control volume where ρ is defined. Further, neglecting the effect of gravity, which is very small, we obtain:

$$\rho V.\frac{\partial v}{\partial y} = -\frac{\partial p}{\partial y} - 2\frac{\tau_w}{R}$$

The wall shear stress may be written in terms of friction factor. In fluid flow through pipes the pressure decreases due to shear stress. The Darcy's friction factor is for fully developed flow in a pipe. In fully developed flow the velocity does not change in the flow direction. [11] In case of a capillary tube it increases along the length. Still it is good approximation to approximate the shear stress term by friction factor. For fully developed flow the left hand side of above equation is zero, hence the frictional pressure drop Δp_f may be obtained from the following equation:

$$\tau_w = R\Delta P_f/2(\Delta y)$$

The friction factor is defined as

$$\Delta P_f = \rho f - \frac{\Delta y}{D}.\frac{V^2}{2}$$

Now,

$$\tau_w = \rho f V^2/8$$

Substituting for τ_w

$$\rho V.\frac{\partial V}{\partial y} = -\frac{\partial p}{\partial y} - \frac{\rho f V^2}{2D}$$

For laminar flow the effect of wall roughness in negligible and friction factor is given by

$$f = 64/Re$$

For turbulent flow the friction factor increases with increase in roughness ratio. Moody's chart gives the variation of friction factor with Reynolds numbers for various roughness ratios. Blasius Correlation expression for the smooth pipe [12]

$$f = 0.3164Re^{-0.25}$$

It is assumed in the above analysis that the expansion is a constant enthalpy process. This is strictly not true inside a capillary tube since there is a large change in kinetic energy due to change in velocity along the length due to flashing of refrigerant liquid. In fact kinetic energy increases at a very fast rate as the velocity becomes sonic and the flow becomes choked.

Thermodynamic and Thermo-Physical Properties of Refrigerants:

a) Suction pressure: At a given evaporator temperature, the saturation pressure should be above atmospheric for prevention of air or moisture ingress into the system and ease of leak detection. Higher suction pressure is better as it leads to smaller compressor displacement

b) Discharge pressure: At a given condenser temperature, the discharge pressure should be as small as possible to allow light-weight construction of compressor, condenser etc.[13-14]

c) Pressure ratio: Should be as small as possible for high volumetric efficiency and low power consumption

d) Latent heat of vaporization: Should be as large as possible so that the required mass flow rate per unit cooling capacity will be small.

e) Isentropic index of compression: Should be as small as possible so that the temperature rise during compression will be small

f) Liquid specific heat: Should be small so that degree of sub-cooling will be large leading to smaller amount of flash gas at evaporator inlet

g) Vapour specific heat: Should be large so that the degree of superheating will be small

h) Thermal conductivity: Thermal conductivity in both liquid as well as vapour phase should be high for higher heat transfer coefficients

i) Viscosity: Viscosity should be small in both liquid and vapour phases for smaller frictional pressure drops

The operating pressures, temperatures and latent heat of vaporization can be summarized by Clausius-Clapeyron Equation

$$In(P_{sat}) = -\frac{h_{fg}}{RT} + \frac{s_{fg}}{R}$$

Comparison between different refrigerants

R 11 (CFC 11), R 12 (CFC 12), R 22 (HCFC 22), R 502 (CFC 12+HCFC 22) etc are the commonly used Synthetic refrigerants. However, these refrigerants have to be phased out due to their Ozone Depletion Potential (ODP). Prior to the environmental issues of ozone layer depletion and global warming, the most widely used refrigerants were: R 11, R 12, R 22, R 502 and ammonia. Among of these, R 11 was primarily used with centrifugal compressors in air conditioning applications. R 12 was used primarily in small capacity refrigeration and cold storage applications, while the other refrigerants were used in large systems such as large air conditioning plants or cold storages. Among the refrigerants used, except ammonia, all the other refrigerants are synthetic refrigerants and are non-toxic and non-flammable. Though ammonia is toxic, it has been very widely used due to its excellent thermodynamic and thermo-physical properties.[13]

In view of the environmental problems caused by the synthetic refrigerants, opinions differed on replacements for conventional refrigerants. The alternate refrigerants are categorized into two broad groups i.e. Non-ODS, synthetic refrigerants based on Hydro-Fluoro-Carbons (HFCs) and their blends and the second one are Natural

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refrigerants including ammonia, carbon dioxide, hydrocarbons and their blends. Table 1 is useful to find out a list of refrigerants being replaced and their replacements.[11]

Table 1 Refrigerants, their applications and substitutes

Refrigerant	Substitute suggested Retrofit(R)/New (N)	Application
R 11(CFC) NBP = 23.7ºC hfg at NBP=182.5 kJ/kg Tcr =197.98ºC Cp/Cv = 1.13 ODP = 1.0 GWP = 3500	R 123 (R,N) R 141b (N) R 245fa (N) n-pentane (R,N)	Large air conditioning systems Industrial heat pumps As foam blowing agent
R 12 (CFC) NBP = -29.8ºC hfg at NBP=165.8 kJ/kg Tcr =112.04ºC Cp/Cv = 1.126 ODP = 1.0 GWP = 7300	R 22 (R,N) R 134a (R,N) R 227ea (N) R 401A,R 401B (R,N) R 411A,R 411B (R,N) R 717 (N)	Domestic refrigerators Small air conditioners Water coolers Small cold storages
R 22 (HCFC) NBP = -40.8ºC hfg at NBP=233.2 kJ/kg Tcr =96.02ºC Cp/Cv = 1.166 ODP = 0.05 GWP = 1500	R 410A, R 410B (N) R 417A (R,N) R 407C (R,N) R 507,R 507A (R,N) R 404A (R,N) R 717 (N)	Air conditioning systems Cold storages
R 134a (HFC) NBP = -26.15ºC hfg at NBP=222.5 kJ/kg Tcr =101.06ºC Cp/Cv = 1.102 ODP = 0.0 GWP = 1200	No replacement required * Immiscible in mineral oils * Highly hygroscopic	Used as replacement for R 12 in domestic refrigerators, water coolers, automobile A/Cs etc.
R 717 (NH3) NBP = -33.35ºC hfg at NBP=1368.9 kJ/kg Tcr =133.0ºC Cp/Cv = 1.31 ODP = 0.0 GWP = 0.0	No replacement required * Toxic and flammable * Incompatible with copper * Highly efficient * Inexpensive and available	Cold storages Ice plants Food processing Frozen food cabinets
R 744 (CO2) NBP = -78.4ºC hfg at 40ºC=321.3 kJ/kg Tcr =31.1ºC Cp/Cv = 1.3 ODP = 0.0 GWP = 1.0	No replacement required *Very low critical temperature * Eco-friendly * Inexpensive and available	Cold storages Air conditioning systems Simultaneous cooling and heating (Transcritical cycle)
R718 (H2O) NBP = 100.ºC hfg at NBP=2257.9 kJ/kg Tcr =374.15ºC Cp/Cv = 1.33 ODP = 0.0 GWP = 1.0	No replacement required * High NBP * High freezing point * Large specific volume * Eco-friendly * Inexpensive and available	Absorption systems Steam jet systems

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