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Heat Transfer Analysis of Refrigerant Flow in an Evaporator Tube

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Abstract: *the paper aim is to presenting the heat transfer analysis of refrigerant flow in an evaporator tube is done. The main objective of this paper is to find the length of the evaporator tube for a pre-defined refrigerant inlet state such that the refrigerant at the tube outlet is superheated. The problem involves refrigerant flowing inside a straight, horizontal copper tube over which water is in cross flow. Inlet condition of the both fluids and evaporator tube detail except its length are specified. here pressure and enthalpy at discrete points along the tube are calculated by using two-phase frictional pressure drop model. Predicted values were compared using another different pressure drop model. A computer-code using Turbo C has been developed for performing the entire calculation.*

Keywords: *heat transfer, refrigerant flow, evaporator tube, pressure drop model, copper tube, Turbo C*

I. Introduction

Two phase flow of gases and liquids or vapors and liquids in pipes, channels, equipment, etc. is frequently encountered in industry and has been studied intensively for many years. Exact prediction of pressure gradients an d boiling heat transfer phenomenon during the flow of two phase mixtures is an essential step in the design of a great variety of industrial equipment in the power and process industries. Knowledge of boiling heat transfer phenomenon, flow patterns and heat transfer correlations can reduce the cost and avoid the drastic results due to over design and under design of evaporators, boilers and other two-phase process equipments.

1.1 Flow boiling

In the literature, two types of boiling of a saturated fluid are described, pool boiling and flow boiling. In pool boiling, heat is transferred to a stagnant fluid, a pool. During flow boiling, heat is transferred to a fluid having a velocity relative to the surface from where the heat is supplied. In saturated flow boiling heat is transferred by two different mechanisms, nucleate boiling and convective evaporation. Convective evaporation resembles ordinary convective heat transfer in single phase heat transfer; i.e. the main resistance to heat transfer is at the heated wall. This part of the heat transfer is often modeled using heat transfer correlations similar to single phase heat transfer correlations. In nucleate boiling, heat is mainly transferred into the bulk of the gas/ liquid by means of bubbles nucleating on the surface, growing and finally detaching from the surface. This part of the heat transfer is similar to pool boiling and is often modeled as pool boiling. The total heat transfer coefficient is then calculated accounting (by weighing) for both mechanisms.

The weighing is carried out differently, as may be found in the literature. In the most cases, a correction factor is introduced into the pure convective and nucleate parts. The convective part is often said to be enhanced due to the presence of bubbles. In a similar way, the nucleate part is often said to be suppressed due to the fact that the flow of the liquid may suppress bubble nucleation. The combined effect of these two mechanisms is not yet well understood and several different approaches may be found in the literature.

1.1.1. Flow boiling heat transfer correlations

Numerous flow boiling correlations exist in the literature. In this section, some of the better known correlations are listed.

1.1.1.1. Chen (1966) correlation

Chen (1966) published his classical paper on flow boiling, where the evaporating heat transfer coefficient was a sum of macro and micro mechanisms.

$$\alpha = \alpha_{mic} + \alpha_{mac} \tag{1.1}$$

The micro evaporation, nucleate boiling, was calculated as

$$\alpha_{mic} = 0.00122 \left[\frac{k_l^{\ 0.79}\ c_{pl}^{\ 0.45}\ \rho_l^{\ 0.49}\ g^{0.25}}{\sigma^{0.5}\ \mu_l^{\ 0.29}\ h_{fg}^{\ 0.24}\ \rho_g^{\ 0.24}} \right] \Delta t^{0.24}\ \Delta p^{0.75}\ S \qquad (1.2)$$

1.1.1.2 Kandlikar (1990, 2003) correlation

The correlation is $\dfrac{\alpha_{TP}}{\alpha_l} = C_1\ Co^{C_2}\ (25\, Fr_{lo})^{C_5} + C_3\ Bo^{C_4}\ F_{fl}$

The correlation is calculated twice using each set of constants and the greater of the two values is used as the heat transfer coefficient. The Froude number is calculated with the entire flow as liquid.

$$Fr_{lo} = \frac{G^2}{\rho_l^{\ 2}\ g\ d}$$

1.1.1.3 Pressure drop correlations

There are numerous correlations on pressure drop in two-phase flows in the literature. The purpose of the present thesis is not to cover them all; merely the most important contributions will briefly be discussed. A good introduction of adiabatic two-phase flow pressure drop may be found in Chisholm (1983) and in Collier and Thome (1996).

Lockhart and Martenelli (1949) presented their classical paper where they introduced a new parameter, later denoted the Lockhart- Martenelli parameter, defined as

$$X^2 = \frac{\left(\Delta p_l / \Delta l \right)}{\left(\Delta p_g / \Delta l \right)} \qquad (1.26)$$

They graphically correlated the two phase multiplier with the Lockhart- Martenelli parameter. The two-phase multiplier was defined as

$$\phi_l^{\ 2} = \frac{\left(\Delta p_{TP} / \Delta l \right)}{\left(\dfrac{\Delta p_l}{\Delta l} \right)} \qquad (1.27)$$

Baker (1954) presented a paper investigating pressure drop of simultaneous flow of oil and gas. The data does not correlate well with the Lockhart- Martenelli correlation. He observed a distinct change in slope when plotting the data in the Lockhart- Martenelli chart. He concludes that something radical changed the flow. He suggests different correlations for different flow regimes and stresses the importance of taking into account the actual flow pattern when correlating the pressure drop.

Chisholm and Laird (1958) revisited the work by Lockhart and Martenelli (1949) and suggested that the two-phase frictional data could be correlated with reasonable accuracy as

$$\frac{\Delta p_{TP}}{\Delta p_l} = 1 + C / \hat{X}^m \qquad (1.28)$$

For rough tubes. The value of C and m depends on the tube surface and liquid flow rate. The variable $\hat{X}^m$ differs from the definition by Lockhart and Martenelli.

Later, Chisholm (1967) conducted a more refined analysis. However, at the end of the paper he states that the derived equations are too complicated for practical calculations and suggests

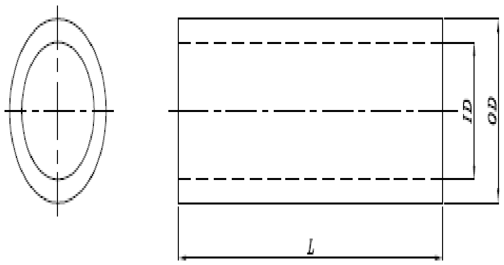
$$\phi_l^{\ 2} = 1 + C / X + C / X^2 \qquad (1.29)$$

Where he now used the definition of X according to Lockhart- Martenelli. He also included the classical values for the Chisholm parameter, C.

II. Problem Description

2.1. Problem

The problem involves refrigerant flowing through a straight, horizontal copper tube over which water is in cross flow. Inlet conditions of both the fluids and evaporator tube detail except its length are specified. The main objective is to find the length of the evaporator tube for a pre defined inlet refrigerant state such that the refrigerant at the tube outlet is super heated.



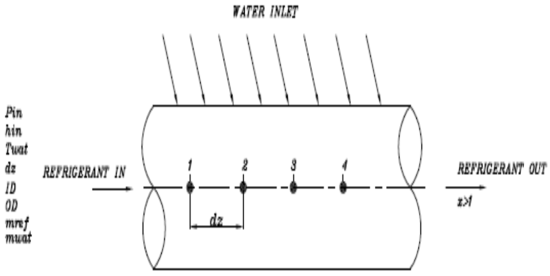


Fig.2.1. Geometry of evaporator tube

Fig.2.2. Problem Visualization

2.2. Input parameters to be specified

1. Refrigerant inlet pressure
2. Refrigerant inlet enthalpy
3. Name of the refrigerant
4. Mass flow rate of the refrigerant
5. Water velocity
6. Water temperature
7. Inner diameter of the tube
8. Outer diameter of the tube
9. Space between two nodes

2.3. Solution approach

We require two properties to fix the state of the refrigerant i.e., pressure and enthalpy. In this analysis pressure and enthalpy can be found at discrete points along the length of the tube. For calculating enthalpy one ordinary differential equation is required. This can be obtained by balancing energy, which flows through an elementary strip as shown below. Energy balance can be defined as “The amount of energy entering into the strip is equal to the energy leaving from that strip”.

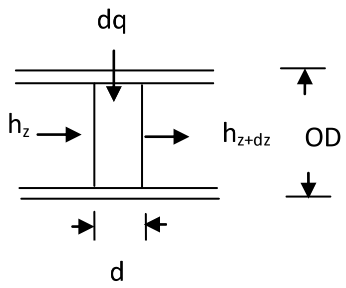


Fig.2.3. Energy balance through strip of length dz

Energy balance for a strip of length dz

Energy input = Energy output

$$m_f h_z + dq = m_f h_{z+dz} \qquad (2.1)$$

where m_f is the mass flow rate of the refrigerant.

After simplification equation 2.1 becomes

$$h_{z+dz} - h_z = \frac{2\pi(T_{wat} - T_{ref})dz}{m_f\left(\frac{1}{\alpha_i r_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{k} + \frac{1}{\alpha_o r_o}\right)} \qquad (2.2)$$

In equation (2.2) α_i and α_o are inside and outside heat transfer coefficients. The inside heat transfer coefficients can be found from boiling heat transfer correlations by Klimenko and outside heat transfer coefficient is calculated from the correlations of flow over bodies. Then enthalpy at next node can be found by using equation

For calculating the pressure we require another differential equation is required which gives the pressure drop between two nodes. Pressure drop includes both frictional pressure drop as well as acceleration pressure drop. Frictional pressure drop can be obtained from Chisholm's model. Acceleration pressure drop can be obtained from momentum equation along axial direction.

By using pressure and enthalpy state of the refrigerant can be fixed at next node. Length of the tube can be found by marching from one node to another until the state of the refrigerant is super heated.

III. Boiling Heat Transfer

Boiling process occurs when the temperature of a liquid at a specified pressure is raised to saturated temperature at that pressure. It is considered to be one form of convection heat transfer as it involves fluid motion (such as rise of bubbles to the top). It differs from other forms of convection in that it depend on latent heat of vaporization of the fluid and surface tension at the liquid-vapor interface, in addition to properties of the fluid in each phase. The heat transfer coefficients associated with boiling are typically much higher than those encountered in other forms of convection processes that involve single phase.

3.1. Introduction

Many familiar engineering applications involve boiling and condensation heat transfer. In the household refrigerator, for example, the refrigerant absorbs heat from the refrigerated space by boiling in the evaporator section and rejects heat to air by condensing in the condenser section. Some electronic components are cooled by boiling by immersing them in a fluid with an appropriate boiling temperature. Boiling is a liquid-to-vapor change process just like evaporation, but there significant differences between the two.

3.2. Types of boiling

Based on the presence of bulk fluid motion boiling is classified into two types. They are

I. Pool boiling
II. Flow boiling

Boiling is called pool boiling in the absence of bulk fluid motion and flow boiling or forced convection boiling in presence of it. In pool boiling, fluid is stationary, and any motion of fluid is due to natural convection currents and motion of bubbles under the influence of buoyancy. In flow boiling the fluid is forced to move in a heated pipe or over a surface by external means such as pump. Pool and flow boiling are further classified into two types depending on the bulk liquid temperature. They are

- Sub cooled boiling
- Saturated boiling

IV. Pressure Drop

4.1. Introduction

During two phase flow in a non horizontal channel, the pressure drop is made up of contributions due to geodetic changes in position, acceleration and friction. Because of very complicated phenomena occurring in two phase flow empirical or semi empirical relationships or physical models are used for calculating pressure drop. These physical models are correlated with the experimental results. They are broadly classified into three categories

- Homogeneous flow model
- Heterogeneous flow model
- Hybrid flow model

Out of all, homogeneous model is the simplest. This assumes that the liquid and the gas or vapor are uniformly distributed over the flow cross section and in the flow direction, so that the mixture can be regarded as a single phase flow with suitably defined mean values of the thermodynamic and hydrodynamic properties of the two phases. It is frequently used as a reference model because of its ease of manipulation.

In the heterogeneous model or slip model it is assumed that the gas or vapor and the liquid flow separately as continuous phases with distinct mean velocities within different parts of the flow cross section or flow channel.

Generally, the actual flow behavior of a two phase mixture lies between these two limiting cases. Only at very small mass fractions of vapor does the condition occur in which the velocities of the two phases are the same. As a result, a series of mixed or hybrid models have been developed such as variable density model, drift-velocity model.

Total pressure drop constitutes

1. Pressure drop due to change in level
2. Acceleration pressure drop
3. Frictional pressure drop

The first two components describe a reversible change of pressure, since a part of the energy or momentum of the flow only appears in another form and can be converted back again without loss. On the other hand, the frictional pressure drop is an irreversible change of pressure resulting from the energy dissipated in the flow by friction, eddying, etc.

In the flow of a two phase mixture through a pipe with a constant flow cross section, there is always an increase in the volumetric flow due to the reduction in pressure caused by friction, and in a single-component mixture also due to flash evaporation. This results in an increase in the velocity of both phases. The resulting momentum changes make themselves felt as a pressure drop due to acceleration. When vaporizing mixtures are heated, there is an increase of the vapor fraction resulting in further increase in the volumetric flow and an additional pressure drop due to acceleration.

In general, frictional pressure drop contributes most significantly to the total pressure drop. However, even today its calculation is still quite imprecise. Experimental results show that under comparable conditions, frictional pressure drop in two phase flow may be considerably larger than in single flow. The ratio of two pressure drops indicates how many times larger the frictional pressure drop in two phase flow is than in single phase flow. It is usually referred to as the two phase friction multiplier. The determination of frictional pressure drop or friction multiplier is not possible by theoretical means alone, since the individual phenomena occurring such as in momentum transfer between two phases, wall friction or shear at the phase interface , can still not be specified quantitatively. In practice, using of relationships based on different frictional pressure drop models, which are corrected or correlated by measurements

4.1.1. Pressure drop due to change in level

The pressure drop as a result of changes in geodetic position is given by the relationship

$$\left(\frac{dp}{dz}\right)_h = \left[\varepsilon\,\rho_g + (1-\varepsilon)\,\rho_f\right] g\,(\sin\phi) \tag{4.2}$$

Where ϕ denotes the angle between the pipe axis and the horizontal. The pressure drop disappears for a horizontal pipe since $\sin\phi = 0$. At small mean void fractions of gas or vapor and large density ratios this component may form the largest contribution to the overall pressure drop in a non horizontal flow. In the more usual case, in which the geodetic pressure drop is very small, calculations with the homogeneous flow model are sufficient, i.e., with the volumetric flow quality instead of with the mean void fraction.

4.1.2. Acceleration pressure drop

An exact calculation of the pressure drop due to acceleration in two phase flow is not possible, since this requires a knowledge of local phase velocities or mass flow rates, which can be only incompletely approximated by the mean phase velocities. If homogeneous flow of the two phases is assumed, the pressure drop due to acceleration can be calculated from

$$(dp)_{acc} = G^2 \int_1^2 \frac{d}{dz}\left(\frac{x^*}{\rho_g} + \left(\frac{1-x^*}{\rho_l}\right)\right)dz \tag{4.3}$$

Compared to the equation for the pressure drop due to acceleration when heterogeneous flow is assumed:

$$(dp)_{acc} = G^2 \int_1^2 \frac{d}{dz}\left(\frac{x^{*2}}{\rho_g\,\varepsilon} + \left(\frac{(1-x^*)^2}{\rho_l\,(1-\varepsilon)}\right)\right)dz \tag{4.4}$$

The first equation leads in most cases to a more accurate conservative form. The assumption of a homogeneous flow is valid for only for special flow conditions. Acceleration pressure drop can be calculated by solving momentum equation. In unheated two phase flows the pressure drop due to acceleration can often be neglected. A simple rule for estimating the pressure drop in flows of refrigerants consists of comparing the frictional pressure drop with saturation pressure.

4.1.3. Frictional pressure drop

In general, frictional pressure drop contributes most significantly to the total pressure drop. However, even today its calculation is still quite imprecise. Experimental results show that under comparable conditions, frictional pressure drop in two phase flow may be considerably larger than in single flow. As a rule the frictional pressure drop in two- phase is referred to that of pure liquid flow at the same total mass flow rate.

$$\phi^2 = \frac{\left(\frac{dp}{dz}\right)_{2,ph}}{\left(\frac{dp}{dz}\right)_{1,ph,o}}$$

The ratio of two pressure drops indicates how many times larger the frictional pressure drop in two phase flow is than in single phase flow. It is usually referred to as the two phase friction multiplier. The determination of frictional pressure drop or friction multiplier is not possible by theoretical means alone, since the individual phenomena occurring such as in momentum transfer between two phases, wall friction or shear at the phase interface , can still not be specified quantitatively.

In practice, using of relationships based on different frictional pressure drop models, which are corrected or correlated by measurements.

4.1.3.1. Frictional pressure drop models

The frictional pressure drop models given below come under heterogeneous model category. They are classified as

1. Chisholm model
2. Friedel model

V. Pressure Drop Models

5.0.1. Chisholm model

$$\phi^2 = 1 + \left(\tau^2 - 1\right)\left[\left(B\, x^{0.875}\, (1-x)^{0.875}\right) + x^{1.75}\right] \tag{5.1}$$

$$\left(\frac{dp}{dz}\right)_{go} = \frac{2\, f_{go}\, G^2}{D\, \rho_g} \tag{5.2}$$

$$f_{go} = 0.079\, \left(\mathrm{Re}_{go}\right)^{-0.25} \tag{5.3}$$

$$\mathrm{Re}_{go} = \frac{G\, D}{\mu_g} \tag{5.4}$$

$$\tau^2 = \frac{\left(\frac{dp}{dz}\right)_{go}}{\left(\frac{dp}{dz}\right)_{lo}} \tag{5.5}$$

$$f_{lo} = 0.079\, \left(\mathrm{Re}_{lo}\right)^{-0.25} \tag{5.6}$$

$$\mathrm{Re}_{lo} = \frac{G\, D}{\mu_l} \tag{5.7}$$

$$\left(\frac{dp}{dz}\right)_{fric} = -\phi^2 \left(\frac{dp}{dz}\right)_{lo} \quad (5.8)$$

$$\left(\frac{dp}{dz}\right)_{lo} = \frac{2\ f_{lo}\ G^2}{D\ \rho_l} \quad (5.9)$$

5.0.2. Friedel model

$$\psi_{lo} = E + \frac{3.24\ F\ H}{Fr^{0.045}\ We^{0.035}} \quad (5.10)$$

$$E = (1-x)^2 + x^2\ \frac{\rho_l\ f_{go}}{\rho_g\ f_{lo}} \quad (5.11)$$

$$F = x^{0.78}\ (1-x)^{0.24} \quad (5.12)$$

$$H = \left(\frac{\rho_l}{\rho_g}\right)^{0.91} \left(\frac{\mu_g}{\mu_l}\right)^{0.19} \left(1-\frac{\mu_g}{\mu_l}\right)^{0.7} \quad (5.13)$$

$$Fr = \frac{G^2}{g\ d\ \rho_H{}^2} \quad (5.14)$$

$$We = \frac{G^2\ d}{\sigma\ \rho_H} \quad (5.15)$$

$$\frac{1}{\rho_H} = \frac{1-x}{\rho_l} + \frac{x}{\rho_g} \quad (5.16)$$

VI. Procedure

Step 1: Specify the inlet refrigerant conditions.
Step 2: Calculation of pressure at node 2
a) Frictional pressure drop
b) Acceleration pressure drop

Frictional pressure drop is calculated by using Chisholm model. Acceleration pressure drop is calculated by using the following equation.

$$dp_{acc} = \left(\frac{G^2}{\rho_1} - \frac{G^2}{\rho_2}\right) \quad (6.1)$$

But for that equation we require the density of refrigerant at state 2. For that one we assume pressure at node 2 as the calculated pressure by considering frictional pressure drop only.

Step 2.1: Calculation of enthalpy at node 2(Est.)
Enthalpy is calculated by using the following equation

$$h_2 - h_1 = \frac{2\pi(T_{wat} - T_{ref})dz}{m_f \left(\frac{1}{\alpha_i r_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{k} + \frac{1}{\alpha_o r_o}\right)} \quad (6.2)$$

Step 2.1.1. Calculation of alpho (α_o) is calculated by using the following equation.

$$Nu = C * \mathrm{Re}_D{}^m * \mathrm{Pr}^{0.333} \quad (6.3)$$

All the relevant properties are calculated at mean film temperature by assuming $T_{wo}=T_{ref.}$
Step 2.1.2. Calculation of alphi (α_i) is calculated by using the boiling heat transfer equations.
T_{wi} is calculated by using the following equation.

$$\frac{T_{wi} - T_{ref}}{\left(\dfrac{1}{\alpha_i A_i}\right)} = \frac{T_{wat} - T_{ref}}{\left(\dfrac{1}{\alpha_i A_i} + \dfrac{\ln\left(r_o / r_i\right)}{2\pi k_c L} + \dfrac{1}{\alpha_o A_o}\right)} \tag{6.4}$$

By using the iterative procedure we will refine the alphi and T_{wi} values until the error (%) should be less than or equal to 0.0001.By using the value of (α_o) outside wall temperature (T_{wo}) is calculated from the following equation.

$$\frac{T_{wo} - T_{wi}}{\left(\dfrac{\ln\left(r_o / r_i\right)}{2\pi k_c L}\right)} = \frac{T_{wi} - T_{ref}}{\left(\dfrac{1}{\alpha_i A_i}\right)} \tag{6.5}$$

Step.2.1.3. α_o is refined from the outside wall temperature (T_{wo}) by the equation
Step.2.1.4. At the end of step 2.1.3., we will get all refined values of α_i and α_o . We can fix the estimated state of the refrigerant at next node.
Step2.2: Calculate the acceleration pressure drop using the equation 6.1.
Step2.3: Find the pressure at node 2 considering both frictional and acceleration pressure drops
Step 3: Calculation of enthalpy at node 2 by repeating the step.2.1.
Step 4: At the end of step2 we will get pressure at node 2 and at the end of step 3 we will get enthalpy at node 2. By using these two values we can fix the exact state of refrigerant at node 2.
Step5: Repeat the steps from 1 to 4 until the condition of the refrigerant reaches superheated condition
Step6: Finally check for law of conservation of energy.

VII. Results and Discussion

Length of the evaporator tube was determined for a pre-defined refrigerant inlet state and finally checked for law of conservation of Energy. Energy is conserved for all refrigerant inlet states and for all combination of pressure drop, boiling heat transfer models. This chapter is divided into two sections. In the first section, variation of parameters along the length for same mass flow rate of 60 kg/hr and different combination of pressure drop and boiling heat transfer correlations are discussed. In the second section, comparison of length of the evaporator tube at different mass flow rates and two different combinations of models is compared. Variation of different parameters along length of the tube for different mass flow rates, for different initial states of the refrigerant for the two different combinations were observed. Those investigations are not producing here.

As shown in Fig.7.1. For the same mass flow rate of 60 kg/hr of R-22, pressure drop for Klimenko-Chisholm combination is high for a given length. Pressure drop depends on the state of the refrigerant. Two-phase flow multiplier is large for Chisholm's model when compared to Friedel's model. Therefore frictional pressure drop is large, in Chisholm's model. Slope of the Klimenko-Chisholm curve is larger than Dembi-Friedel curve, so pressure drop is large for Klimenko-Chisholm curve.

As shown in Fig.7.2. Enthalpy variation is more or less constant through out length of the tube for the same mass flow rate of R-22 and for both combinations. Both curves merge one over other as. Strictly speaking enthalpy difference is a little bit high for Dembi- Friedel curve.

As shown in Fig.7.3. Saturated temperature variation is large in Klimenko- Chisholm combination, since the saturated pressure difference is large for that combination. Slope of the curve is larger for that combination. As shown in Fig.7.4. For the same mass flow rate dryness fraction varies linearly for both types of combinations and both the curves overlap one over other. And their slopes are more or less constant.

As shown in Fig.7.5. Inside heat transfer coefficient increases along the length of the tube for both types of combination. But their range is different. In Klimenko's model single phase forced convection heat transfer is incorporated and convective boiling number is used to distinguish between nucleate boiling and forced convection boiling. As the flow progresses single phase forced convection heat transfer coefficient decreases and nucleate boiling heat transfer increases in Klimenko's model. But in Dembi's model forced convection heat transfer increases much faster than nucleate boiling. The contribution of forced convection in Dembi's correlation is large when compared to single phase forced convection in Klimenko's correlation. So heat transfer coefficient is high in Dembi's model.

As shown in Fig.7.6. Outside heat transfer coefficient decreases along the length of the tube and both the curves are parallel to each other .The variation of outside heat transfer coefficient is very less in these two models, as it depends on outside wall temperature.

In the second section length of evaporator tube for different mass flow rates and different combinations are compared and finally checked for energy conservation. Energy is conserved for all sets of input data as shown in Table.7.1.It was observed that if the mass flow rate increases, length of the tube was increasing due to larger pressure drop and less increment in enthalpy.

S.No	Mass flow rate(kg/hr)	combination	Heat lost by water(W)	Heat gained by refrigerant(W)	Length(m)
1	60	Dembi- Friedel	3583.51	3583.58	3.13
		Klimenko-Chisholm	3581.72	3581.73	3.19
2	50	Dembi- Friedel	2991.12	2991.12	2.63
		Klimenko-Chisholm	2986.95	2986.96	2.67
3	40	Dembi- Friedel	2392.55	2392.55	2.12
		Klimenko-Chisholm	2393.29	2393.29	2.14

Table.7.1. Comparison of energy balance and Length of evaporator tube for different mass flow rates and different combination

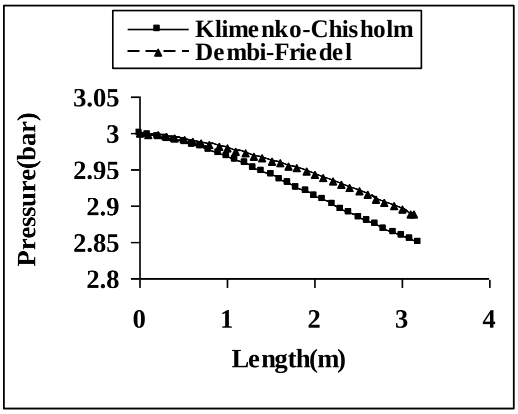


Fig.7.1. Pressure along length of tube for same mass flow rate and for different combinations

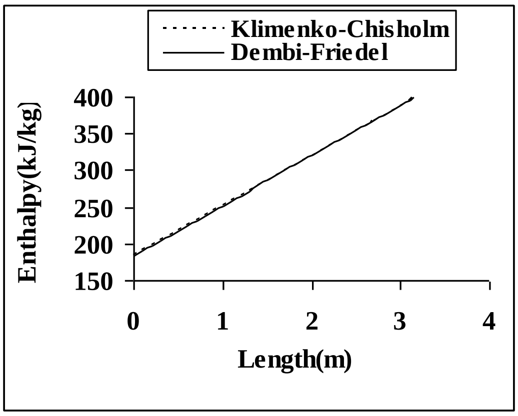


Fig.7.2. Enthalpy variation along length of tube for same mass flow rate and for different combinations

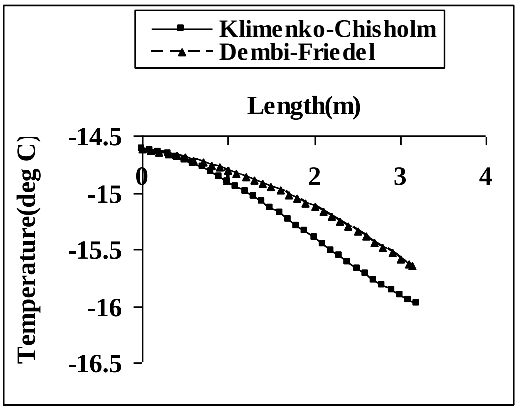


Fig.7.3. Temperature along length of tube for same mass flow rate and for different combinations

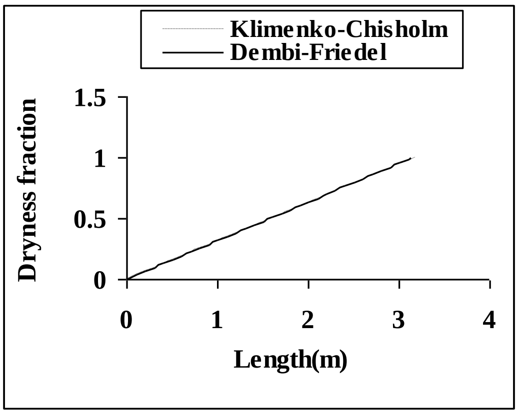


Fig.7.4.Dryness fraction along length of tube for same mass flow rate and for different combinations

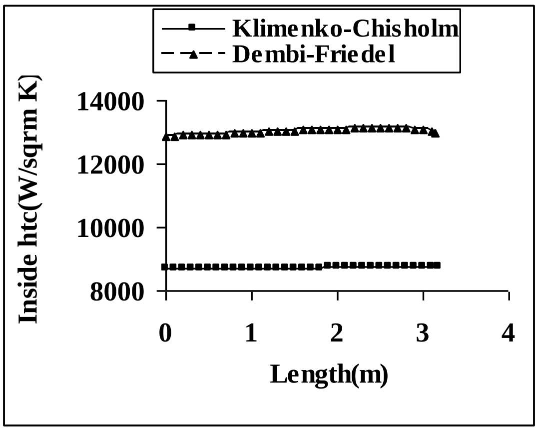


Fig.7.5. Inside heat transfer coefficient along length of tube for same mass flow rate and for different combinations

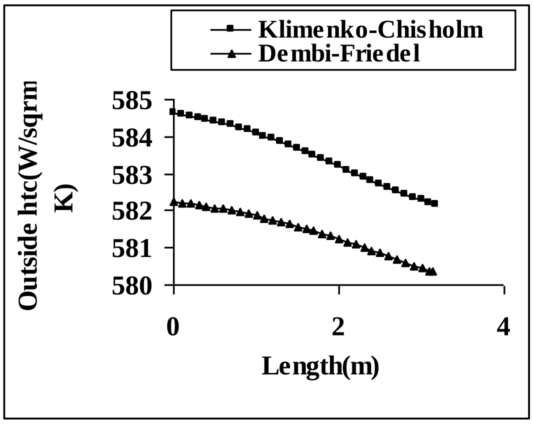


Fig.7.6.Outside heat transfer coefficient along length of tube for same mass flow rate and for different combinations

VIII. Conclusion

The obtained results were satisfactory and energy was conserved for all refrigerant inlet states. There is good agreement between the two combinations of models when they compared and yield results with minimum deviation.

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