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# Computational Science of Vapor Liquid Two Phase Flow Inside Heat Exchanger Tube

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***Abstract*—** A heat exchanger is one of the key components in air-conditioning and refrigeration systems. Condensers and evaporators are typically heat exchangers which are used to condense vapors into liquid and evaporate liquid into vapor respectively also known as phase transition. To obtain the dynamic performance of the heat exchanger system, simulation in transient state is required. For prediction of such performance a homogeneous vapor liquid two phase flow model is used. To obtain the solution of the model science of the computations need to be used efficiently. In the present paper an efficient computational model and method are proposed. The method is capable of predicting the refrigerant temperature distribution, velocity of refrigerant, tube wall temperature as a function of position and time. A single tube heat exchanger with refrigerant R22 as working fluid was chosen as a sample and some tests were carried out to determine its transient response. The examination of results indicates that the computational model provides a reasonable prediction of dynamic response.

***Keywords*—** Vapor liquid two phase flow Homogeneous Model, set of nonlinear equations, finite difference method, and MATLAB program

## I. INTRODUCTION

Analysis of two phase flow inside heat exchanger tube begins with the most general principles governing the behavior of all matter, namely, conservation of mass, momentum and energy. These principles can be expressed mathematically at every point in space and time by local, instantaneous field equations. However the exact solution to these equations is almost impossible, requiring the tracking of many convoluted liquid-vapor interfaces that change continuously in time. Instead, the usual procedure is to average the local, instantaneous equations in either time or space, or both. Although we lose information in the process, the resulting equations yield accurate solutions to a wide variety of practical problems so long as the averaged variables bear some resemblance to the actual situation, that is, so long as the flow is not too chaotic.

In this work, a distributed dynamic model is presented for dry-expansion condensers and evaporator. The model mainly takes account of refrigerant density, mass flow, and refrigerant temperature inside tube. An efficient and stable numerical method is also described to obtain the numerical solution of the model.

During the averaging, the two phases may be treated together to obtain averaged variables for a two-phase mixture; alternatively, treating each phase separately, we obtain averaged variables for both phases. The better procedure yields the Homogeneous model, which is a bit more general and useful. A usual homogeneous model consists of three field equations: averaged mass, momentum and energy equations for the liquid and vapor phases. For example, integrating across the cross section of our heated tube at some particular time, through regions with liquid and regions with vapor, we obtain area averaged conservation equations for the liquid and the vapor. Or, integration over a small volume element provides volume-averaged equations. We could also integrate over a period of time at some particular location in the tube to obtain time-averaged conservation equations. Finally, additional variables are introduced into the averaged conservation equations, namely, the volume (or area) fraction of the vapor a1 and of the liquid a2 for a given region. Because the flowing material is either vapor or liquid, a1 and a2 are not independent. Rather a1 + a2 = 1 [1].

The field equations are usually derived by assuming that the interface separating the phases are zero thickness and zero mass and hence cannot store kinetic and thermal energy. Density (ρ), Enthalpy (h), Vapor quality (X) are computed by the Refprop software for the refrigerant R22.

Moreover, the mass transpose process gradually takes place inside the heat tube. To predict a correct refrigerant mass distribution, the model should be also distributed [2]. Standing from this point of view, the following modelling Strategies can be made: *Strategy 1,* the model to be set up will be distributed in Structure and all balance equations will be solved in the form of partial differential equations. At the manumit, the heat transfer between refrigerant and tubes, tubes and other tubes, tubes and air will be calculated locally. The momentum equation in the two-phase flow region determines mass flow rate correlation with void fraction model. *Strategy 2,* because the refrigerant temperature in the two-phase flow region is almost constant

 

except the effect of pressure drop, the energy balance equation can be easily integrated in this region so that it will be solved lumped. The temperature decrease caused by pressure drop will be added later on. This strategy avoids simultaneous solution of the mass balance and energy balance equations which are otherwise all partial differential equations. In the single-phase region the refrigerant has mass and heat contents and it is possible to consider the refrigerant temperature as a zero-order parameter (no heat and mass accumulations in the region), while the pipe wall of this region still plays a dynamic role.

## II. THEORATICAL ANALYSIS

To predict the important fluid phenomenon inside heat exchanger, the transient model we required should be fast enough to be practical. This requires the identification of suitable assumptions that can simplify the mathematical form without loss of relevant detail. Efficient numerical techniques are also necessary to reduce the computation time, thereby allowing the model to run as close to real-time as is possible, while also constraining errors to acceptable limits.

Some of the common assumptions found to be made in the model are:

i. Flow in the heat exchangers is one dimensional
ii. Axial conduction in the refrigerant is negligible
iii. Liquid and vapor refrigerant in the heat exchangers are in thermal equilibrium
iv. Effects of pressure wave dynamics are negligible
v. Expansion is isenthalpic
vi. Compression is isentropic or polytrophic
vii. Thermal resistances of metallic elements in the system are negligible in comparison with their capacitances.

In addition to the above classifications, the flow of two-phase refrigerant can be modelled using a homogenization. In the homogenous model, the liquid and vapor phases are considered to be in thermal equilibrium and moving at the same velocity. This assumption is based on the belief that differences in the variables will promote momentum, energy, and mass transfer between the phases rapidly enough so that equilibrium is reached.

Consequently, a heat exchanger model consists of

(1) an overall material balance
(2) component material balances
(3) energy balances
(4) heat transfer equations

## III. *DYNAMIC MODEL OF HEAT EXCHANGER*

Begin with the material balances. A total material balance can be written for vapor and liquid phase

$$\frac{\partial A\rho}{\partial t} + \frac{\partial A\dot{m}}{\partial x} = 0 \quad (1)$$

But the tube has constant volume and is filled with steam, so we can revise this to

$$\frac{\partial \rho}{\partial t} + \frac{\partial \dot{m}}{\partial x} = 0 \quad (2)$$

$$\rho = \alpha\rho_v + (1-\alpha)\rho_l \quad (3)$$

$\rho_v = vapor\ density, \rho_l = liquid\ density$

$$u = \frac{\dot{m}}{\rho}, \dot{m} = mass\ flow \quad (4)$$

$$\alpha = \frac{1}{1+\frac{\rho_v(1-x)}{\rho_l(x)}}, x\ is\ vapor\ quality. \quad (5)$$

Component (Solute) Balance:

$$\frac{\partial m}{\partial t} + \frac{\partial \beta \dot{m} u}{\partial x} = -\frac{\partial P}{\partial x} \quad (6)$$

$$\beta = x^2\left(1 + \frac{\rho_l(1-\alpha)}{\alpha\rho_v}\right) + (1-x^2)\left(1 + \frac{\alpha\rho_v}{\rho_l(1-\alpha)}\right) \quad (7)$$

A solvent balance could also be written, but would not be a mathematically independent equation. Note that the solute is treated as non-volatile, so that none leaves with the vapor. This is true for most heat exchanger applications [3].

Energy Balance:

$$\frac{\partial \rho T}{\partial t} + \frac{\partial \dot{m} T}{\partial x} = \frac{A_i}{A_{tub}} q_{wf} \quad (8)$$

$$q_{wf} = U_i(T_w - T_f) \quad (9)$$

$A_i$, is the inner tube wall area,

$A_{tub}$ , is inside cross-section area of tube.

The heat transfer terms are $q_{wf}$, heat transferred from the tube wall to the process fluid.

$$U_i = \frac{1}{\frac{1}{h_o} + \frac{1}{h_i}}, overall\ heat\ transfer\ coefficient. \quad (10)$$

$h_o$, is the heat transfer coefficient between the gas and the solid surfaces,

$h_i$, is the convection coefficient between the liquid and the tube wall.

The convection coefficient is found by application of convection heat transfer correlation. For the present work Chen's correlation [6] is adopted to estimate the local heat transfer coefficient of two-phase flows.

For the present work the out-side heat transfer coefficient $h_o$ is calculated by the Turaga *et al.* correlation [5].

Thermal properties of the refrigerant (R-22) and air are calculated directly from the Refprop software.

Energy Balance -- Tube Wall:

$$C_{pw}M\frac{\partial T_w}{\partial t} = \frac{U_o A_o}{C_{pa}}(h_a - h_w) - U_i A_i (T_w - T) \quad (11)$$

$C_{pw}$, the conductivity of tube wall,

$C_{pa}$, the conductivity of air,

$M$, the mass flow rate of air,

$h_a$ , the enthalpy of air,

$h_w$ , the enthalpy of tube wall.

No material balance is necessary for the tube metal, but heat is transferred into and out of the tube walls, so an energy balance is needed.

These four first order non-linear partial differential equations and two algebraic equations comprise a

 

fundamental, dynamic model of a single tube heat exchanger.

**Physical Initial conditions**

There are two different initial conditions that may be applied, i.e. starting-up and steady-state initial conditions. In this study, we are more interested in the dynamic response of the heat exchanger to variations of the system operation parameters; the heat exchanger is therefore assumed to be in the steady state or in equilibrium position initially. In the equilibrium state, the system should also obey the basic governing equations, and the only difference from the transient state is that all terms involved with time derivative in differential equations are set to zero. The solutions of the basic governing equations in the equilibrium state are used as the initial conditions of the system.

**Physical Boundary conditions**

The boundary conditions applied are the refrigerant and air inlet conditions for the heat exchangers. For the refrigerant side, two boundary conditions are required for determining the thermal response, and one more boundary condition is required for determining the pressure response. Thus, for the solution of condenser and evaporator models, the boundary conditions are set by the output parameters of the compressor and the capillary models. For a condenser, the boundary conditions are compressor mass flow rate, discharge enthalpy (or temperature), and the capillary mass flow rate. For evaporator, the boundary conditions are capillary and compressor mass flow rates and the enthalpy (or temperature) at the exit of a capillary tube. Mathematically, it can be shown as follows:

$\rho$ / x = 0 = $\rho$ in  $\rho$ / x = L = $\rho$ out

$\dot{m}$ / x = 0 = mf in  $\dot{m}$ / x = L = mf out

p/ x = 0 = p in

T / x = 0 = T in  T / x = L = T out

## IV. NUMERICAL SOLUTION OF MODEL

Let us consider first the finite difference method and assume that necessary grids have been generated using transformation to a suitable coordinate system. The transformed plane, known as the computational plane, is assumed to be rectangular. The governing partial differential equations are also transformed to the computational plane. The partial differential equations as well as those occurring in the boundary and/or initial conditions are then approximated by partial difference quotients of the unknown function at each mesh point. In practice, the partial derivatives of the unknown function at a given mesh point are approximated by difference quotients involving a finite number of ordinates. The term "finite" in "finite difference" is used in contrast with the term "infinitesimal" used in "infinitesimal Calculus" implying small quantities approaching zero in the limit. In method like the finite difference, the concerned elements involved are not infinitesimal but finite.

So, for each mesh point of the domain, an algebraic equation is obtained. Thus, we obtain a system of algebraic equations, the number of unknowns being equal to the number of equations, each being equal to the number of mesh points. If the partial differential equation as well as the boundary and/or initial conditions are linear, then the resulting system of algebraic equations is also linear; otherwise the system is nonlinear. For the nonlinear equations some kinds of approximations leading to linearization of the basic equations are required to introduce in order to make the problems tractable. This algebraic system is then solved on a digital computer using standard numerical procedures.

To solve the above mathematical model numerically, the tube is divided into small cells. For each cell, the governing differential equations can be discredited by using the implicit finite difference scheme as mentioned in point.

$$-\frac{r}{4}u_{j-1}^{n}\rho_{j-1}^{n+1}+\rho_{j}^{n+1}+\frac{r}{4}u_{j+1}^{n}\rho_{j+1}^{n+1}=\rho_{j}^{n}-\frac{r}{2}\left((\rho u)_{j+1}^{n}-(\rho u)_{j-1}^{n}\right)-\frac{r}{4}u_{j-1}^{n}\rho_{j-1}^{n}+\frac{r}{4}u_{j+1}^{n}\rho_{j+1}^{n} \quad (12)$$

$$-\frac{r}{4}u_{j-1}^{n}(\rho u)_{j-1}^{n+1}+(\rho u)_{j}^{n+1}+\frac{r}{4}u_{j+1}^{n}(\rho u)_{j+1}^{n+1}=(\rho u)_{j}^{n}-\frac{r}{2}\left(\left(\left(\beta\rho u^{2}\right)_{j+1}^{n}-\left(\beta\rho u^{2}\right)_{j-1}^{n}\right)-\left((P)_{j+1}^{n}-(P)_{j-1}^{n}\right)-\frac{r}{4}u_{j-1}^{n}(\rho u)_{j-1}^{n}+\frac{r}{4}u_{j+1}^{n}(\rho u)_{j+1}^{n} \quad (13)$$

$$-\frac{r}{4}u_{j-1}^{n}(\rho T)_{j-1}^{n+1}+(\rho T)_{j}^{n+1}+\frac{r}{4}u_{j+1}^{n}(\rho T)_{j+1}^{n+1}=(\rho T)_{j}^{n}-\frac{r}{2}\left((\rho uT)_{j+1}^{n}-(\rho uT)_{j-1}^{n}\right)-\frac{r}{4}u_{j-1}^{n}(\rho T)_{j-1}^{n}+\frac{r}{4}u_{j+1}^{n}(\rho T)_{j+1}^{n}+\frac{U_{i}A_{i}}{A_{tub}}\left(T_{wj}^{\ n}-T_{j}^{n}\right) \quad (14)$$

$$C_{pw}M_{w}\frac{(T_{w})_{j}^{n+1}-(T_{w})_{j}^{n}}{\Delta\tau}=\frac{U_{o}A_{a}}{c_{pa}}(h_{a}-h_{w})-U_{i}A_{i}(T_{wj}^{\ n}-T_{j}^{n}) \quad (15)$$

Above algebraic equations are in tridiagonal matrix form which is solved by using LU decomposition in the present work.

## V. RESULTES

To simulate the problem the tube is considered of the unit length, inside diameter 0.0093 fetes and outside diameter 0.011 fetes. Now to start simulation, we apply the equilibrium state conditions together with the boundary conditions & heat transfer effects i.e. 1) Only vapor 2) Vapor & liquid 3) Only liquid. First of all fluid enters in the tube in its superheated vapor form, and then by condensation it reduces temperature by releasing the heat. Fluid comes to its saturated vapor form at which it comes in two phase vapor liquid zone. Two phase zone is finished when vapor quality becomes very low; at that point fluid is in saturate liquid state.

The simulation is done by using MATLAB program which gives the following outputs:

 

**Graphs showing the unsteady to steady behavior of density, velocity, Temperature of refrigerant inside Condenser for the m=1**

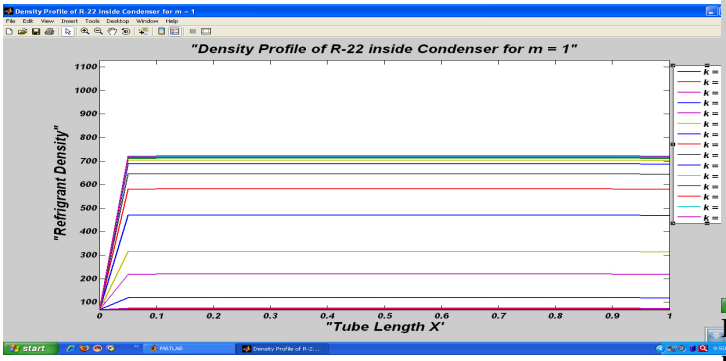


**Fig 1 Density Profile of R-22 inside Condenser for m = 1**

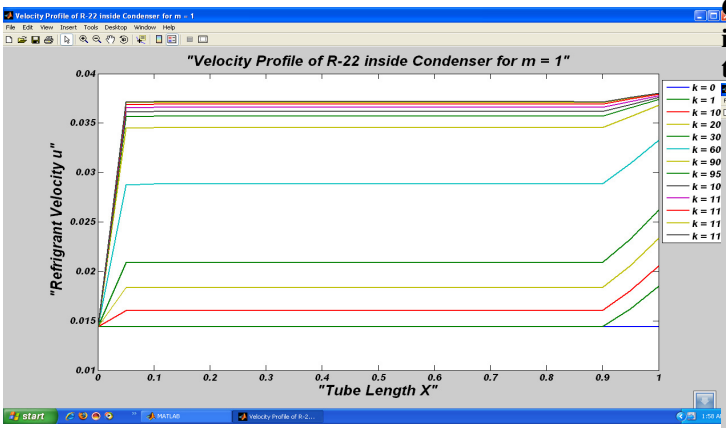


**Fig 2 Velocity Profile of R-22 inside Condenser for m = 1**

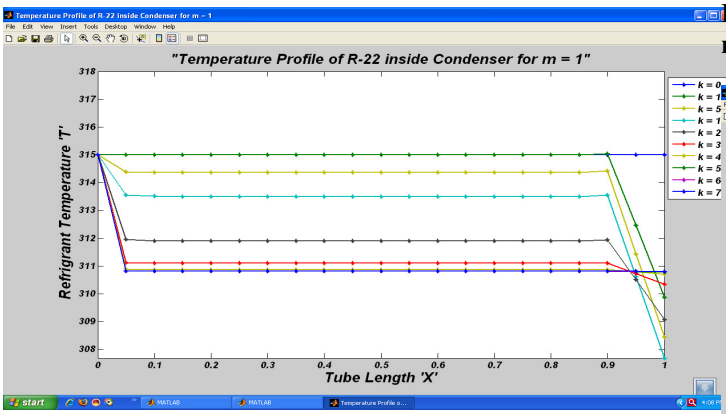


**Fig 3 Temperature Profile of R-22 inside Condenser for m = 1**

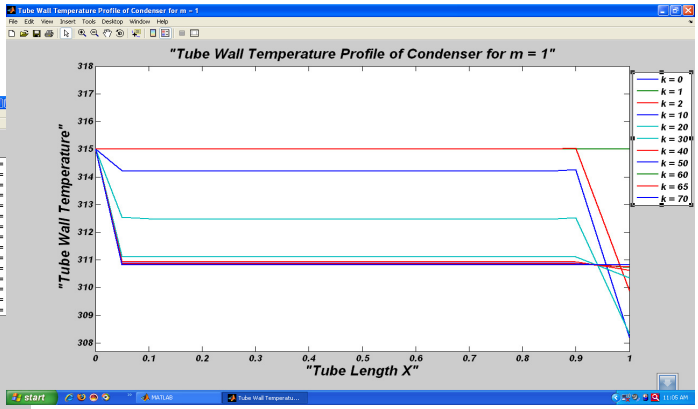


**Fig4 Tube Wall Temperature Profile of Condenser for m = 1**

**Graph showing profile of refrigerant temperature after increasing mass flow rate (m = 1.3 & 0.7) for the same time step.**

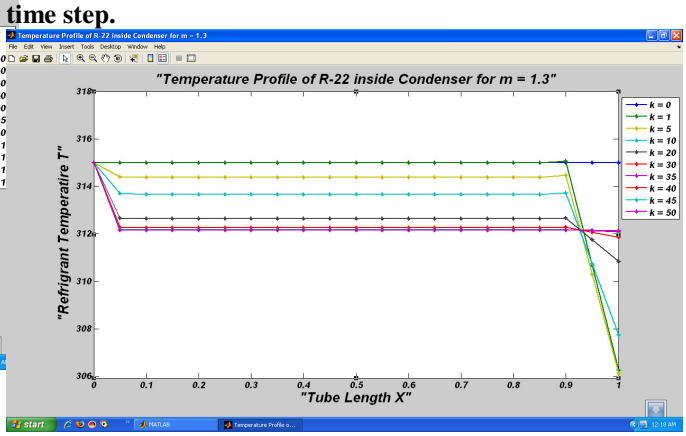


**Fig 5 Temperature Profile of R-22 inside Condenser for m = 1.3**

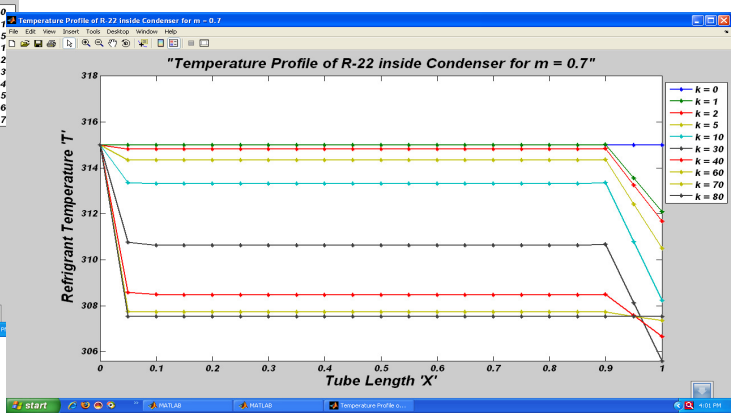


**Fig 6 Temperature Profile of R-22 inside Condenser for m = 0.7**

The governing equations for the Transient (Homogeneous) model of Evaporator and their discretization using implicit method are similar to equations and discretization of

 

Homogeneous model of condenser. Here only the initial and boundary conditions are changing.

Now to start simulation, we apply the equilibrium state conditions together with the boundary conditions & heat transfer effects i.e. 1) Vapor & liquid 2) Only vapor. First of all fluid enters in the tube in its saturated liquid form, and then by evaporation it increases temperature by absorbing the heat. Fluid comes to its saturated liquid form at which it comes in two phase liquid vapor zone. Two phase zone is finished when vapor quality becomes very high; at that point fluid is in only dry vapor form.

The simulation is done by using MATLAB program which gives the following outputs:

**Graphs showing the unsteady to steady behavior of density, velocity, Temperature of refrigerant inside Evaporator for the m = 1**

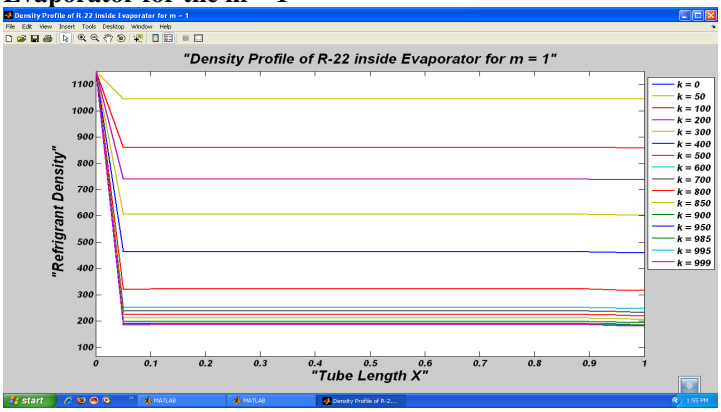


**Fig 7 Density Profile of R-22 inside Evaporator for m = 1**

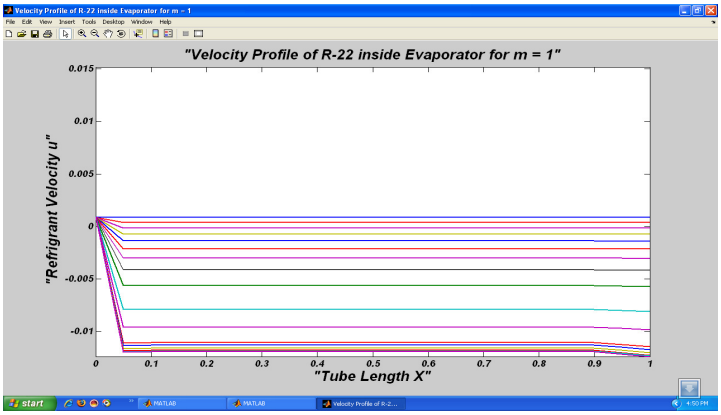


**Fig 8 Velocity Profile of R-22 inside Evaporator for m = 1**

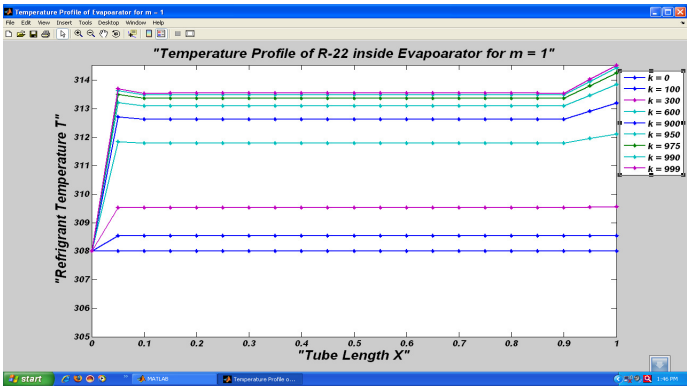


**Fig 9 Temperature Profile of R-22 inside Evaporator for m = 1**

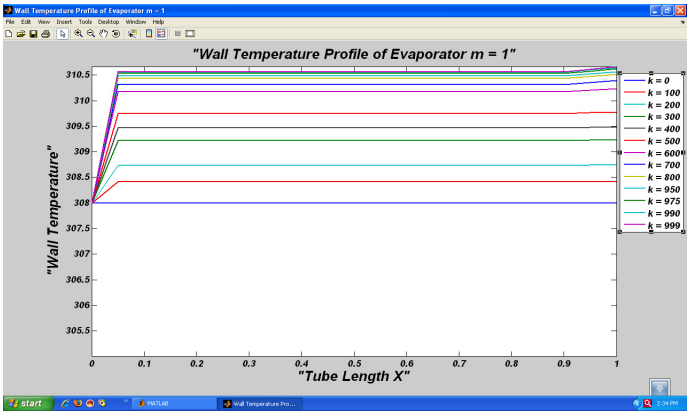


**Fig 10 Tube Wall Temperature Profile of Evaporator for m = 1**

**Graph showing profile of refrigerant temperature after increasing mass flow rate (m = 1.3) for the same time step.**

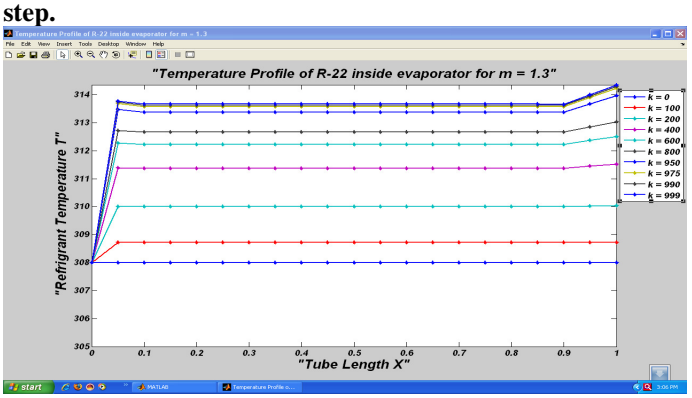


**Fig 11 Temperature Profile of R-22 inside Evaporator for m=1.3**

**Graph showing profile of refrigerant temperature after decreasing mass flow rate (m = 0.7) for the same time step**

 

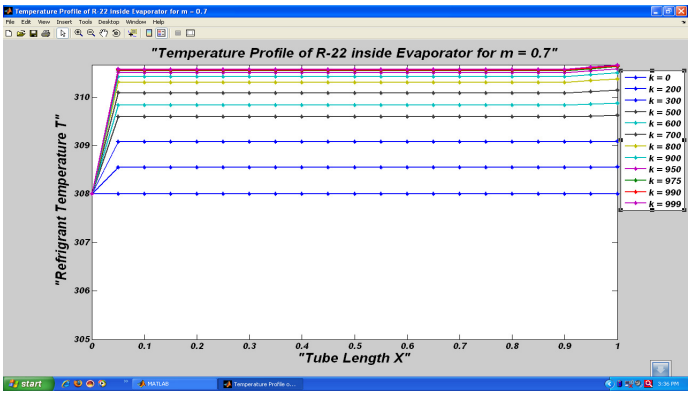


**Fig 12 Temperature Profile of R-22 inside Evaporator for m=0.7**

## VI. CONCLUSION

Resulted graphs of condenser simulation show the convergence of the used scheme even for the significantly large time steps. Equations without stiff source term like Conservation of mass and Conservation of momentum converges even for the small time steps but it takes more computational time. The speed of the vapor through the heat pipe is limited by the rate of condensation at the cold end and far lower than the molecular speed.

Simulation of Evaporator is computationally more complicated as it required large numbers of iteration for the convergence. In terms of the motion of molecules evaporation can be explained as at any given temperature above $0^{o}$K (*absolute zero*) the molecules of a liquid move-some slowly, some at intermediate speed, and some very rapidly. An average velocity is calculated. A rapidly moving molecule near the surface of the liquid may possess necessary kinetic energy to overcome the attraction of the surrounding molecules and escape (**EVAPORATE**) to the space above the liquid. Other fast moving molecules will leave the liquid phase and appear in the gaseous phase above the liquid as the process of evaporation continues.

Condenser and evaporator are modelled by simplifying the one-dimensional formulation of the conservation laws of mass, momentum and energy. Both steady-states as well as dynamic behavior are well predicted. But the pressure in both the condenser and the evaporator are not very well predicted. The transient behavior of the mass flow rate is not well captured during large disturbances. The cause of these discrepancies is analyzed and explained, resulting in suggestions for further research.

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