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Transient simulation of Vapor Liquid Two Phase Flow inside Single tube Evaporator using Quazi linearization approach

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Abstract

A homogeneous model for prediction of the transient performance of an evaporator with liquid vapor two phase flow inside is presented. The model is capable of predicting the refrigerant temperature distribution, velocity of refrigerant, tube wall temperature as a function of position and time. An efficient finite difference method is proposed to obtain the numerical solution of the model without solving a large set of non-linear equations simultaneously. A single tube evaporator 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 theoretical model provides a reasonable prediction of dynamic response which is useful in designing a controllable compressor to reduce power consumption. Transient behavior of temperature of refrigerant has been obtained using MATLAB programs.

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

NOMENCLATURE

A	heat transfer area per meter
Atub	inside cross-section area of tube
Cp	specific heat
h	enthalpy

mw	tube wall and fin mass per meter
p	pressure
qwf	heat transfer rate from tube wall to refrigerant
t	time
T	temperature
u	refrigerant velocity
U	heat transfer coefficient
z	vapor quality
α	refrigerant void fraction
p	density

Subscripts

a	air
i	inside
o	outside
w	tube wall
f	refrigerant side

INTRODUCTION

An evaporator is one of the key components in air-conditioning and refrigeration systems. Evaporator are typically boilers or heat exchangers which are used to evaporate liquid into vapor also known as phase transition. During transient operation, all the components experience phenomena absent in steady state operation, due to the non-uniformity of conditions within them. The refrigerant mass flow rate, in general, is continuously changing, causing changes in refrigerant distribution in the system and because of two-phase condensing flows inside the tube; the local heat transfer coefficient varies in a great range at different locations. This varying local heat transfer coefficient results in an uneven temperature distribution of the evaporator. In applications, to fully simulate the above process inside the tube demands a transient model of the system[i].

For heat exchangers, a sub-class of transient modeling technique was found[ii] i.e. the phase-independent finite difference methods. In the finite difference approach, the governing conservation equations (Navier-Stokes equations of CFD) are approximated by a finite difference scheme that typically consists of dividing the heat exchangers into a number of elements, and each element is defined with its own state properties. The formulation for any element is phase-independent and therefore identical in all the phases.

In this paper, a general homogeneous model applicable for transient analysis is presented. The model is demonstrated for the analysis of temperature distribution inside a commercial evaporator.

THEORETICAL ANALYSIS

To predict this important phenomenon, the transient model 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.

The common assumptions found to be made in the model are:

- Flow in the heat exchangers is one dimensional
- Axial conduction in the refrigerant is negligible
- Liquid and vapor refrigerant in the heat exchangers are in thermal equilibrium
- Effects of pressure wave dynamics are negligible
- Expansion is isenthalpic
- Compression is isentropic or polytrophic
- 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 modeled using a homogenization[iii]. In the homogenous model, the vapor and liquid 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[iv].

The governing equations for the Homogeneous model are as follows:

Refrigerant Side :-

Continuity Equation :-

$$\frac{\partial \rho}{\partial t}+\frac{\partial \rho u}{\partial x}=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(1)}$$

$$\rho=\alpha\rho_v+(1-\alpha)\rho_l \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2)}$$

$$u = \frac{\dot{m}}{\rho} \qquad \text{.. (3)}$$

$$\alpha = \frac{1}{1 + \frac{\rho_v (1-X)}{\rho_l (X)}}$$, X is vapor quality................................ (4)

Momentum Equation :-

$$\frac{\partial \rho u}{\partial t} + \frac{\partial \beta \rho u^2}{\partial x} = -\frac{\partial P}{\partial x} \qquad \text{.. (5)}$$

Energy Equation :-

$$\frac{\partial \rho T}{\partial t} + \frac{\partial \rho u T}{\partial x} = \frac{A_i}{A_{tub}} q_{wf} \qquad \text{... ..(6)}$$

$$h = \alpha \rho_v h_v + (1-\alpha)\rho_l h_l \qquad \text{.. ...(7)}$$

$$q_{wf} = U_i (T_w - T_f) \qquad \text{... ...(8)}$$

Energy Equation for Wall Side :-

$$c_{pw} m_w \frac{\partial T_w}{\partial t} = \frac{U_o A_o}{c_{pa}} (h_a - h_w) - U_i A_i (T_w - T_f) \qquad \text{.................... (9)}$$

To close the system of equations, constitutive equations heat transfer coefficients and thermal properties of the refrigerant and air are included. The single phase heat transfer coefficient is calculated by the Turaga et al. correlation[v]. Chen's correlation[vi] is adopted to estimate the local heat transfer coefficient of two-phase boiling flows. Thermal properties of the refrigerant (R-22) and air are calculated directly from the Refprop software.

Initial conditions

In this study, we are more interested in the dynamic response of the evaporator[vii] to variations of the system operation parameters; the evaporator 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 Equations (1), (5), (6) and (9) are set to zero. The solutions of the basic governing equations in the equilibrium state are used as the initial conditions of the system.

Boundary Conditions

The boundary conditions applied are the refrigerant conditions at the tube inlet and air conditions onto the evaporator coil:

mf / x = 0 = mf in

pf / x = 0 = pf in

α / x = 0 = α in

T / x = 0 = T in

NUMERICAL SOLUTION

To solve the above mathematical model numerically, the governing nonlinear equations are linearized by using quazi linearization and after that discretized using implicit finite difference scheme[viii] which results in algebraic equations as follows:

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

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

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

$$C_{p_w}m_{w}\frac{(T_{w})_{j}^{n+1}-(T_{w})_{j}^{n}}{\Delta t}-\frac{U_{0}A_{0}}{C_{p_a}}(h_{a}-h_{w})+U_{i}A_{i}(T_{wj}^{n}-T_{fj}^{n})=0$$

Where, $r=\frac{\Delta t}{\Delta x}$.

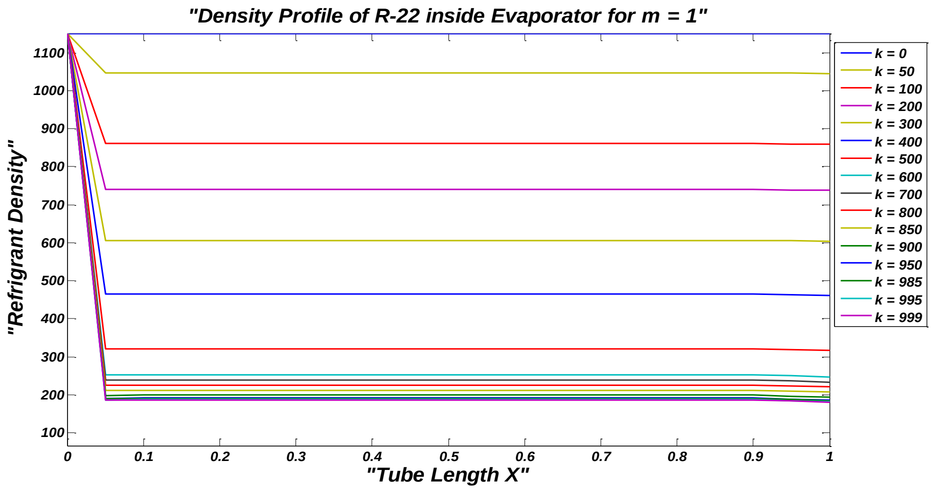
Above algebraic equations are in tridiagonal matrix form which is solved by using LU decomposition in the present work.

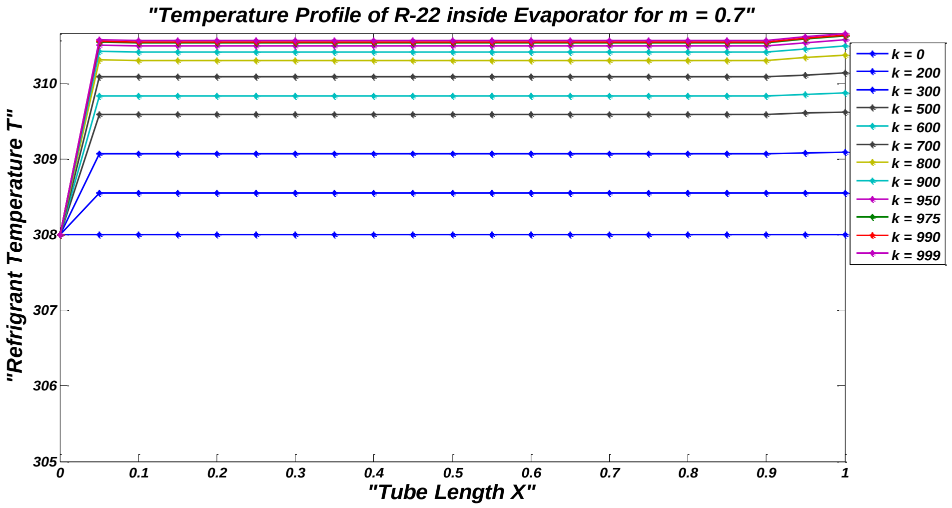
To simulate the problem the tube is considered of the unit length, inside diameter 0.0093 fetes and outside diameter 0.011 fetes. Now to start simulation, we apply the equilibrium state conditions together with the boundary conditions & heat transfer effects i.e. 1) Only Liquid 2) Liquid & Vapor 3) Only Vapor[ix]. First of all fluid enters in the tube in its saturated liquid form, and then by evaporation it increases temperature by receiving the heat. Fluid comes to its saturated vapor 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 superheated vapor state[x].

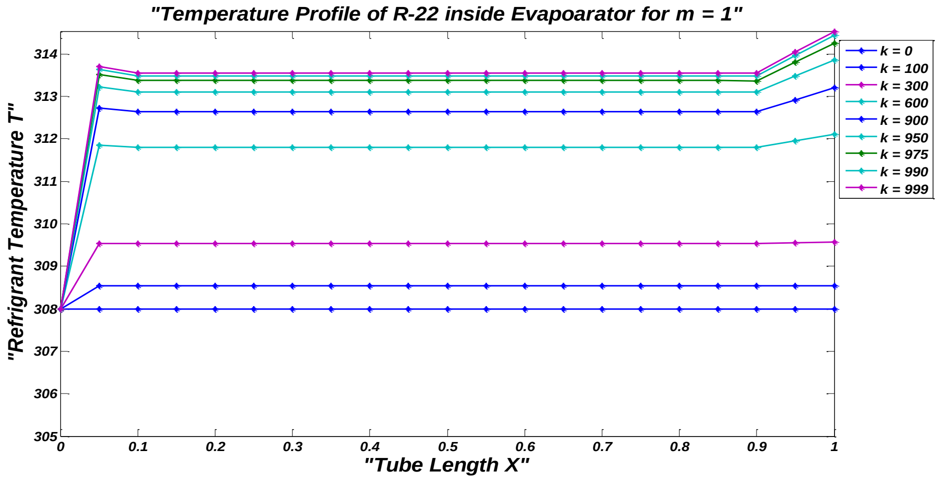
The above simulation is done by using programs based on MATLAB platform.

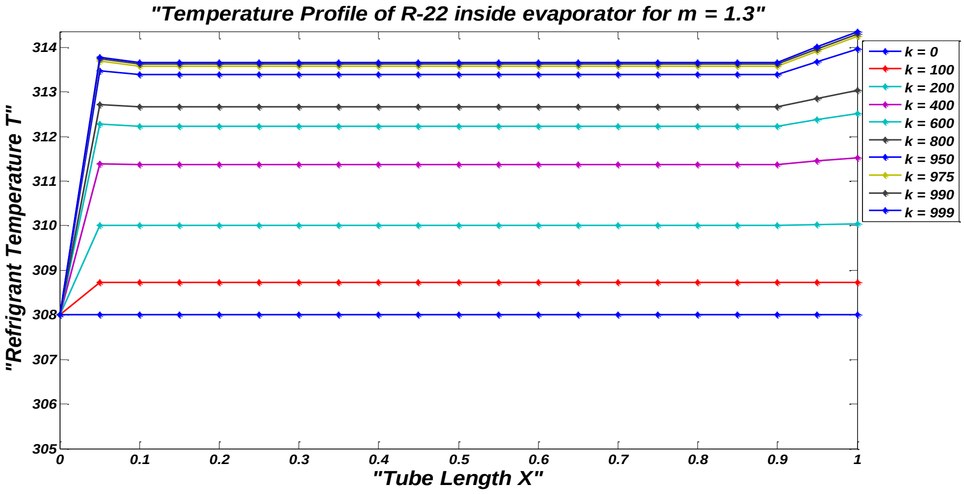
RESULT & DISCUSSION

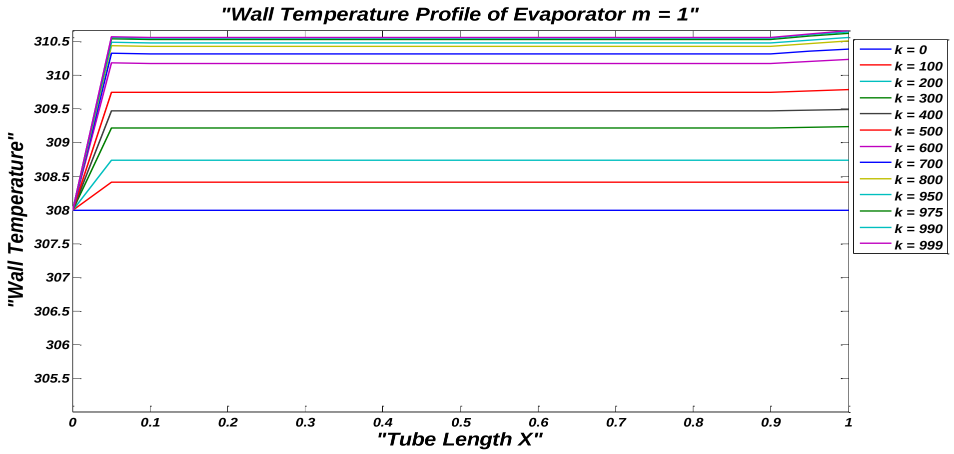
With the Matlab program for certain inputs graph of temperature inside tube on length scale is obtained for different time steps. These time steps are chosen to compute the difference between time steps of unsteady to steady behavior of evaporator. After obtaining the steadiness for the same time step graph of temperature inside tube is obtained by giving certain changes in mass flow inputs. Resulted Graphs are shown as follows:




"Temperature Profile of R-22 inside Evaporator for m = 0.7"
"Refrigrant Temperature T"
"Tube Length X"
305
306
307
308
309
310
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
k = 0
k = 200
k = 300
k = 500
k = 600
k = 700
k = 800
k = 900
k = 950
k = 975
k = 990
k = 999


"Temperature Profile of R-22 inside Evapoarator for m = 1"
"Refrigrant Temperature T"
"Tube Length X"
305
306
307
308
309
310
311
312
313
314
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
k = 0
k = 100
k = 300
k = 600
k = 900
k = 950
k = 975
k = 990
k = 999


"Temperature Profile of R-22 inside evaporator for m = 1.3"
"Refrigrant Temperature T"
"Tube Length X"
305
306
307
308
309
310
311
312
313
314
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
k = 0
k = 100
k = 200
k = 400
k = 600
k = 800
k = 950
k = 975
k = 990
k = 999



CONCLUSION

Computations of the application of the homogeneous model to vapor-liquid flow for predicting the transient performance of an evaporator is presented. The simulation of the model is capable of predicting distributions of the refrigerant temperature in both position and time domain without solving large number of nonlinear algebraic equations. The dynamic behavior of the evaporator is investigated with a step increases and step decreases in the inlet refrigerant flow rate after obtaining steady state which results in variation of time steps and two phase length that is clearly obtained by using the present simulation. This knowledge of the dynamic characteristics of an evaporator is important to the design and control of many air-conditioning and refrigeration system.

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