International Journal of Thermal Technologies
E-ISSN 2277 – 4114
Available at http://inpressco.com/category/ijtt/

Research Article

Modeling Design of Phase Change Mechanism in Condenser

Adil Abbas Alwan* and Hussein Mahmood Jassim

Department of Mechanical Engineering-College of Engineering-University of Babylon-Babylon-Iraq

Accepted 01 Sept 2016, Available online 23 Sept 2016, **Vol.6, No.3 (Sept 2016)**

Abstract

Heat transfer mechanism of phase change phenomena of vapor liquid in fluid through closing unit was simulated by condensation process with high level pressure .This model was simulated with two values of velocity of vapor phase and liquid phase through this process .This model designed by using a hyperbolic equation to predict the spatial and temporal temperature distribution of phase change phenomena in three dimensions depending upon pressure measurement values through different points in domain of condenser using FTCS method in numerical analysis of this model in simulation the temperature distribution in three domain of condense by using FTCS method. In simulation interface moving boundary condition was used to simulate the phase change temperature distribution in three dimensions at interface line saturation temperature. Experimental included design rig and measurement different parameters by using different instruments with high accuracy to verify the theoretical work with time dependent .These measurements were taken after time on hour of working the unit to give more stability and ideal work of phase change phenomena through condensation process.

Keywords: *Condensation; Phase change; heat exchanger; FTCS*

1. Introduction

The heat conduction problems with phase change are receiving increasing attention, mainly because of the broad range of technological applications and applied research fields where this phenomenon has a prevailing role, in addition to the interesting mathematical aspects. In a wider context, they receive the name of Stefan problems, which comprehends general moving boundary phenomena. In civil engineering, for instance, the determination of the freezing time of certain ground regions or the degradation of frozen layers, follow this model. Casting of metals and alloys in metallurgy or solidification of crystals is other classic applications. Condensation is the change of the physical state matter from gas into liquid phase and is reverse of evaporation. Condensation commonly occurs when a vapor is cooled and /or compressed to its saturation limit when the molecular density in gas phase reaches its maximal threshold. Vapor cooling and compressed equipment collects condensed liquids are called a condenser. In systems involving heat transfer a condenser is a device or unit used to condense a substance from its gaseous to its liquid states, by cooling it. (**Rita Szijarto, 2015**) introduced and developed condensation models particularly for the application in the emergency cooling system of a boiling water reactor. The condensation in horizontal pipes is investigated with both one-dimensional system codes (RELAP5) and three-dimensional computational fluid dynamics codes (ANSYS FLUENT). The performance of the RELAP5 code is not sufficient for transient condensation processes; therefore, a mechanistic model is developed and implemented. The model calculates the heat transfer coefficient in a cross section of the flow field considering local parameters in the pipe. The model assumes a stratified flow pattern in the pipe with a laminar liquid film on the upper part of the cross section and an axial turbulent flow at the bottom of the pipe.

They used the modified RELAP5 code to calculate the invert Edward pipe experiment, which consists of a closed, slightly inclined horizontal pipe, resting in a cold water tank. The fast pressurization of the pipe results in a highly transient condensation process on the pipe walls. The simulation predicted well the pressure, void fraction and temperature data for different initial conditions and different locations in the condensation pipe during the transient condensation process. Furthermore, the COSMEA facility, a single tube experiment for flow morphology and heat transfer studies, is simulated with the modified RELAP5 code. The calculations reproduce the experimental temperature and condensation rate results for different initial pressure and mass flow rates. Condensation phenomenon is analyzed using the volume of fluid multiphase method in the

*Corresponding author: **Adil Abbas Alwan**

computational fluid dynamics code. The volume of fluid method solves a unique set of continuity equations in the domain and models the two phases by tracking the interface between them. Four models are developed and implemented into the FLUENT solver for modeling phase change processes. The first technique introduced a mass and energy transfer at the interface, if the temperature of the corresponding cell is different from the saturation value. The equations, through much iteration, achieved the correct boundary conditions at the interface. The second method relies the surface renewal theory. The theory assumes that eddies, forming on the liquid surface, are responsible for the transfer of the latent heat to the bulk liquid. Therefore, the condensation rate depends on the turbulent velocity and length scale. The third model solves the heat flux balance equation at the interface; hence, the resolution of the thermal boundary layer in the linear region is necessary. The fourth technique is derived from the phase field theory, which is a mathematical approach to calculate interface related problems. The four models are tested on the LAOKOON facility, which analyze direct contact condensation in a horizontal duct. The sensitivity study showed that the numerical iteration technique and the phase field model best the experimental results. The COSMEA facility was simulated with the numerical iteration and the phase field models. The simulations captured the developing geometry of the gas-liquid interface in the horizontal pipe and the temperature distribution in the liquid film. **(Kondou,and Hrnjak, 2012)** showed that conventional modeling of condensers always assumes three zones: de superheating, condensation and sub cooling even condensation occurs in de superheating zone at some conditions and that sub cooling occurs during condensation. This paper discusses the actual situation and provides experimental validation of the hypothesis.

The experimental results show heat transfer coefficients of and CO_2 and R410A at mass fluxes 100 to 240 $(kg/m^2.s)$ and reduced pressures from 68% to 100% in a horizontal smooth tube of 6.1 mm inner diameter. Data are compared to correlations proposed for other working fluids or other conditions. Results showed much higher values of HTC than correlation proposed for single-phase turbulent flow in superheated zone. The occurrence of condensation in superheat zone is evident when tube wall temperature is below saturation temperature. The results suggest that simplified calculations of heat rejection in superheated zone significantly oversize the condenser. The semi empirical correlation, which is here proposed as the combination of existing correlations for single-phase turbulent and saturated condensation, satisfactorily predicts the heat transfer coefficient of the superheat zone condensation.(**D. Juric, 1996)** noted that the liquid-vapor phase change problem involves fluid flow as well as heat transfer. This requires the solution of energy equations. However mass transfer across the interface and momentum as well as energy sources at the interface must now be taken into account. Noting that in two-phase flow, additional terms appear in these equations due to the phase change and the fact that the interface is no longer a material interface. The fluid velocity at the interface and the interface velocity are unequal. As before a single set of governing equations is written for both phases. This local, single field formulation incorporates the effect of the interface on the governing equations as sources which act only at the interface. He formulates the phase change problem in terms of variables for each phase with appropriate jump conditions at the moving phase interface. Those local, separate phase formulations form a fundamental basis for all averaged models of two-phase mixtures. In the present work a numerical simulation was presented for analysis the temperature and pressure distribution inside heat exchanger (condenser) in each phase and in the interface of phase change. Also in experimental work the parameters were measured by advance instruments.

2. Mathematical Model

The temperature distribution inside rectangular high pressure heat exchanger was covered by energy equation. Many parameters needed to solve this equation. The geometry and coordinate system of the problem under consideration is described in Fig. 1.The working fluid was changed from vapor to liquid due to the location of boiling point according of temperature and pressure. (**J.P.Holman, 2010)** showed that the energy equation express as:

$$\frac{1}{\alpha_v}\frac{\partial T}{\partial t}+\frac{v_v}{\alpha_v}\frac{\partial T}{\partial x}=\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2} \qquad (1)$$

The assumptions of this case study are:

1- The working fluid is a homogenous and isotropic.
2-The heat exchanger treated as rectangular neglecting the pipe lines.
3 -The heat transfer is controlled by energy equation.
4- The phase change happened somewhere in mid of this rectangular heat exchangers at boiling temperature.
5-The measured velocity of the working fluid enters heat exchanger used as initial value until phase change happens.

(Anderson ,*et al*,1984) showed that the simple Euler method ,equation (1)can be made stable replacing the forward space difference by a backward space difference provided that velocity (v) is positive if the backward space difference is used, then following algorithm of equation (1) result:

$$\frac{T^{n+1}_{i,j,k}-T^{n}_{i,j,k}}{\Delta t}+V_v\left[\frac{T^{n}_{i,j,k}-T^{n}_{i-1,j,k}}{\Delta x}\right]=\alpha_v\Delta t\left[\frac{T^{n}_{i+1,j,k}-2T^{n}_{i,j,k}+T^{n}_{i-1,j,k}}{\Delta x^2}+\frac{T^{n}_{i,j+1,k}-2T^{n}_{i,j,k}+T^{n}_{i,j-1,k}}{\Delta y^2}+\frac{T^{n}_{i,j,k+1}-2T^{n}_{i,j,k}+T^{n}_{i,j,k-1}}{\Delta z^2}\right] \qquad (2)$$

Where$\lambda=\frac{\alpha\Delta t}{\Delta x^2}$ = Fourier number, and $c=\frac{\Delta t V_v}{\Delta x}$ =Courant number

This method is called Upstream (windward) Differencing method

$$T_{i,j,k}^{n+1} = \lambda T_{i+1,j,k}^{n} + (1 - c - 6\lambda)T_{i,j,k}^{n} + (\lambda + c)T_{i-1,j,k}^{n} + \lambda T_{i,j+1,k}^{n} + \lambda T_{i,j-1,k}^{n} + \lambda T_{i,j,k+1}^{n} + \lambda T_{i,j,k-1}^{n} \quad (3)$$

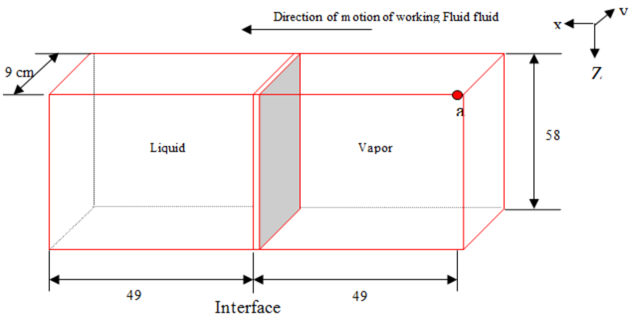


Figure 1 Schematic diagram of the assumption model

3. Numerical Analysis

3.1 Temperature Distribution through Vapor Phase

In vapor phase the three dimensions Cartesian temperature distribution according to energy equation was calculated with assumption of vapor inlet velocity. Which is calculated experimentally by ultrasonic flow meter to be was $v_v = 3.140\ m/s$.

a-The temperature distribution at point (a) represents the starting location of vapor at high pressure heat exchanger. Equation (3) becomes:

$$T_{i,j,k}^{n+1} = 4\lambda T_{i+1,j,k}^{n} + (1 - 6\lambda)T_{i,j,k}^{n} + 2\lambda T_{i,j,k+1}^{n} \quad (4)$$

b- Temperature distribution along x-axis: As the vapor flows in x- direction inside heat exchanger, to find temperature distribution along x – axis equation (3) used as shown in figure (2), equation (3) becomes:

$$T_{i,j,k}^{n+1} = \lambda T_{i+1,j,k}^{n} + (1 - c - 6\lambda)T_{i,j,k}^{n} + (\lambda + c)T_{i-1,j,k}^{n} + 2\lambda T_{i,j+1,k}^{n} + 2\lambda T_{i,j,k+1}^{n} \quad (5)$$

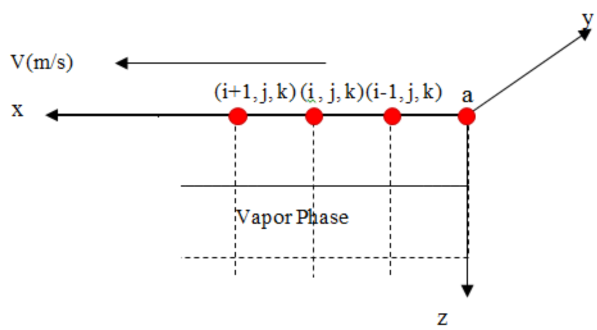


Figure 2 Temperature distribution through x-axis in vapor phase

c. Temperature distribution along y– axis .Equation (3) will be form as:

$$T_{i,j,k}^{n+1} = (2\lambda)T_{i+1,j,k}^{n} + (1 - 6\lambda)T_{i,j,k}^{n} + \lambda T_{i,j+1,k}^{n} + \lambda T_{i,j-1,k}^{n} + 2\lambda T_{i,j,k+1}^{n} \quad (6)$$

d. The temperature distribution through corner end point at y –axis, for i= 1 to 1,j= m to m ,and for k= 1 to 1equation(6) becomes as :

$$T_{i,j,k}^{n+1} = (2\lambda)T_{i+1,j,k}^{n} + (1 - 6\lambda)T_{i,j,k}^{n} + 2\lambda T_{i,j-1,k}^{n} + 2\lambda T_{i,j,k+1}^{n} \quad (7)$$

e. Temperature distribution along z- axis, also the velocity =0 and equation (3) will be form as:

$$T_{i,j,k}^{n+1} = (4\lambda)T_{i+1,j,k}^{n} + (1 - 6\lambda)T_{i,j,k}^{n} + \lambda T_{i,j,k+1}^{n} + \lambda T_{i,j,k-1}^{n} \quad (8)$$

f. The temperature distribution through corner end point at z –axis , equation(8) becomes as :

$$T_{i,j,k}^{n+1} = (4\lambda)T_{i+1,j,k}^{n} + (1 - 6\lambda)T_{i,j,k}^{n} + 2\lambda T_{i,j,k-1}^{n} \quad (9)$$

g. Temperature distribution through x-z plane

$$T_{i,j,k}^{n+1} = \lambda T_{i+1,j,k}^{n} + (1 - 6\lambda - c)T_{i,j,k}^{n} + (\lambda + c)T_{i-1,j,k}^{n} + 2\lambda T_{i,j+1,k}^{n} + \lambda T_{i,j,k+1}^{n} + \lambda T_{i,j_1,k-1}^{n} \quad (10)$$

3.2. Temperature Distribution at Interface Line

(Kim *et al* ,2013) focused their the attention on situations in which evaporation (or condensation) takes place at discrete boiling point temperature and as a result the liquid and vapor phases are separated by sharp interface.

1-The temperature of adjacent phases should be equal the boiling temperature at interface line.

i.e. $T_v = T_l = T_{boiling}$ at $x = \delta_{(t)}$

2-Energy balance must be satisfied at interface.

$$[-q_l - (q_v)] = \rho L \frac{\partial \delta(t)}{dt} \quad at \ \ x = \delta(t) \quad (11)$$

Then the interface energy – balance equation as:

$$k_l \frac{\partial T_l}{\partial x} - k_v \frac{\partial T_v}{\partial x} = \rho L \frac{\partial \delta(t)}{dt} \quad at \ \ x = \delta(t) \quad (12)$$

For condensation process equation (11) is expressed as:

$$-k_v \frac{\partial T_v}{\partial x} = \rho L \frac{\partial \delta(t)}{dt} \quad at \ \ x = \delta(t) \quad (13)$$

This is the moving boundary condition (M.B.C) which supplies in energy equation to calculate the temperature distribution through the phase change where $\frac{\partial \delta(t)}{dt}$ the velocity of interface is in positive x-direction then:

$$\frac{\partial \delta(t)}{dt} = V_* \quad (14)$$

Then the moving boundary condition is expressed as:

$$-k_v \frac{\partial T_v}{\partial x} = \rho L V_* \; at \;\; x = \delta(t) \tag{15}$$

The velocity is of phase change calculated according to **(Steen and Kamalu ,1983),** which depends on heat flux and latent heat which equal the enthalpy change (h_{fg}). All the properties being taken from the standard tables depending on previous temperature values.

$$V_* = \frac{Q_{flux}}{[\rho_v L_v + \rho_v c_v (T_v - T_l)]} \tag{16}$$

a. Phase change at point c

The dynamic moving boundary condition is deeded in energy equation representing in this problem in order to solve it. The dynamic moving boundary condition is represented as:

$$-k_v (\frac{\partial T}{\partial x}) = \rho_v L_v v_* \tag{17}$$

$$T_{i+1,j,k} = T_{i-1,j,k} - \frac{\rho_v L_v 2\Delta x v_*}{k_v} \tag{18}$$

The temperature is change in three directions; the governing equation is represented as:

$$\frac{1}{\alpha_v}\frac{\partial T}{\partial t} = \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \tag{19}$$

Where $\lambda_1 = \frac{\alpha_v \Delta t}{\Delta x^2}$ as assuming $\Delta x = \Delta y = \Delta z$

$$T_{i,j,k}^{n+1} = \lambda_1 T_{i+1,j,k}^n + (1 - 6\lambda_1) T_{i,j,k}^n + \lambda_1 T_{i-1,j,k}^n + \lambda_1 T_{i,j+1,k}^n + \lambda_1 T_{i,j-1,k}^n + \lambda_1 T_{i,j,k+1}^n + \lambda_1 T_{i,j,k-1}^n \tag{20}$$

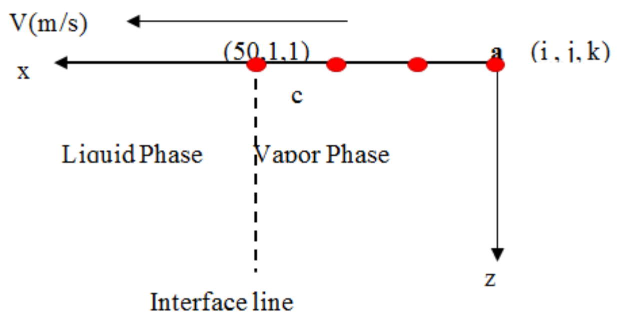


Figure 3 Temperature distribution of phase change at point c

Substituting the dynamic boundary condition in equation (18), and then the equation (20) becomes:

$$T_{i,j,k}^{n+1} = 2\lambda_1 T_{i-1,j,k}^n + (1 - 6\lambda_1) T_{i,j,k}^n + 2\lambda_1 T_{i,j+1,k}^n + 2\lambda_1 T_{i,j,k+1}^n - \lambda_1 \left[\frac{\rho_v L_v 2\Delta x V_*}{k_v}\right] \tag{21}$$

b. Phase change through z-axis

The phase change happened along z-axis, equation (3.28) becomes:

$$T_{i,j,k}^{n+1} = 2\lambda_1 T_{i-1,j,k}^n + (1 - 6\lambda_1) T_{i,j,k}^n - \lambda_1 \left[\frac{\rho_v L_v 2\Delta x V_*}{k_v}\right] + 2\lambda_1 T_{i,j+1,k}^n + \lambda_1 T_{i,j,k+1}^n + \lambda_1 T_{i,j,k-1}^n$$

c. The phase change temperature of the coroner end point at z-axis.

$$T_{i,j,k}^{n+1} = 2\lambda_1 T_{i-1,j,k}^n + (1 - 6\lambda_1) T_{i,j,k}^n - \lambda_1 \left[\frac{\rho_v L_v 2\Delta x V_*}{k_v}\right] + 2\lambda_1 T_{i,j+1,k}^n + 2\lambda_1 T_{i,j,k-1}^n \tag{23}$$

d. Phase change through y-axis

The phase change happened along y-axis

$$T_{i,j,k}^{n+1} = 2\lambda_1 T_{i-1,j,k}^n + (1 - 6\lambda_1) T_{i,j,k}^n - \lambda_1 \left[\frac{\rho_v L_v 2\Delta x V_*}{k_v}\right] + \lambda_1 T_{i,j+1,k}^n + \lambda_1 T_{i,j-1,k}^n + 2\lambda_1 T_{i,j,k+1}^n \tag{24}$$

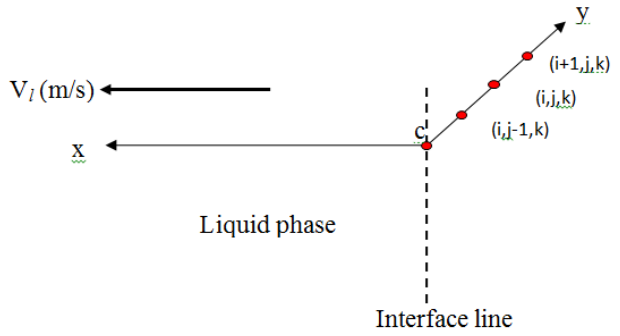


Figure 4 Temperature distribution of phase change through y-axis

e. The phase change temperature of the coroner end point at y-axis

$$T_{i,j,k}^{n+1} = 2\lambda_1 T_{i-1,j,k}^n + (1 - 6\lambda_1) T_{i,j,k}^n - \lambda_1 \left[\frac{\rho_v L_v 2\Delta x V_*}{k_v}\right] + 2\lambda_1 T_{i,j-1,k}^n + 2\lambda_1 T_{i,j,k+1}^n \tag{25}$$

f-Phase change through z-y plane

These phase change through z-y plane

$$T_{i,j,k}^{n+1} = 2\lambda_1 T_{i-1,j,k}^n + (1 - 6\lambda_1) T_{i,j,k}^n - \lambda_1 \left[\frac{\rho_v L_v 2\Delta x V_*}{k_v}\right] + \lambda_1 T_{i,j+1,k}^n + \lambda_1 T_{i,j-1,k}^n + \lambda_1 T_{i,j,k+1}^n + \lambda_1 T_{i,j,k-1}^n \tag{26}$$

3.3 Temperature Distribution through Liquid Phase

The energy equation was used in liquid phase as form:

$$\frac{1}{\alpha_l}\frac{\partial T}{\partial t} + \frac{v_l}{\alpha_l}\frac{\partial T}{\partial t} = \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \tag{27}$$

a- Equation (27) is can be written in numerical form along x- axis as in same technique in vapor phase:

$$T_{i,j,k}^{n+1} = \lambda_2 T_{i+1,j,k}^n + (1 - 6\lambda_2 - c_1) T_{i,j,k}^n + (\lambda_2 + c_1) T_{i-1,j,k}^n + 2\lambda_2 T_{i,j+1,k}^n + 2\lambda T_{i,j,k+1}^n \tag{28}$$

Where here $c_1 = \frac{\Delta t V_l}{\Delta x}$, $\lambda_2 = \frac{\alpha_l \Delta t}{\Delta x^2}$

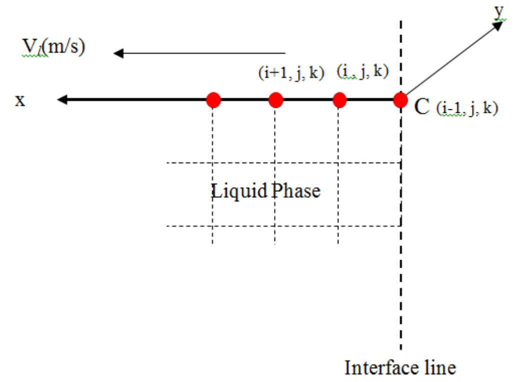


Figure 5 Temperature distribution through x-axis in liquid phase

b. Temperature distribution of the end corner at x –axis in liquid phase:

$$T_{i,j,k}^{n+1} = (1 - 6\lambda_2 - c_1)T_{i,j,k}^{n} + 2(\lambda_2 + c_1)T_{i-1,j,k}^{n} + 2\lambda_2 T_{i,j+1,k}^{n} + 2\lambda T_{i,j,k+1}^{n} \qquad (29)$$

c. Temperature distribution through x- z plane

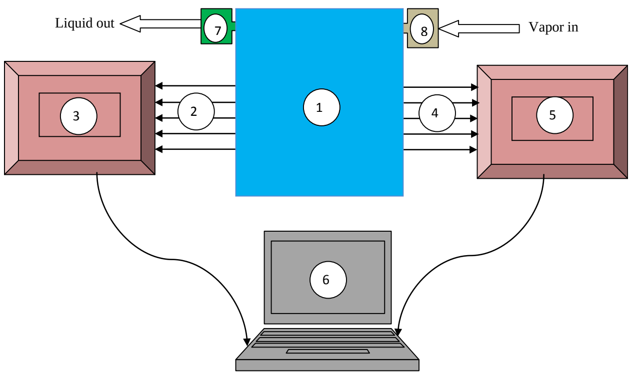
$$T_{i,j,k}^{n+1} = \lambda_2 T_{i+1,j,k}^{n} + (1 - 6\lambda_2 - c_1)T_{i,j,k}^{n} + (\lambda_2 + c_1)T_{i-1,j,k}^{n} + 2\lambda_2 T_{i,j+1,k}^{n} + \lambda_2 T_{i,j,k+1}^{n} + \lambda_2\, T_{i,j,k-1}^{n} \qquad (30)$$

d. Temperature distribution through x- y plane:

$$T_{i,j,k}^{n+1} = \lambda_2 T_{i+1,j,k}^{n} + (1 - 6\lambda_2 - c_1)T_{i,j,k}^{n} + (\lambda_2 + c_1)T_{i-1,j,k}^{n} + \lambda_2 T_{i,j+1,k}^{n} + \lambda_2 T_{i,j-1,k}^{n} + 2\lambda_2 T_{i,j,k+1}^{n} \qquad (31)$$

4. Experimental Apparatus and Procedure

The condensation process occurs under high pressure condition .High pressure heat exchanger receives vapor over boiling point, a high pressure heat exchanger was used in this study. Freon used as working fluid .The air cooled heat exchangers for refrigeration system was used. This type is forced convection finned condenser. In forced convection type condenser the circulation of air over the condenser surface is maintained by using a fan or a blower. These condensers normally use fins on air-side for good heat transfer. The dimension of condenser which is 58 cm in width, 98 cm in length and 9 cm thickness. This is a part of a cold storage unit has capacity of five tons. To study condensation process an experimental rig was built .The main part was a high pressure heat exchanger. The **viols turbine** flow meter used to measure the volume flow rate of a liquid that fills a closed pipeline and flows continuously in it. In actual application, Type KF500 turbine flow meter integrates with relevant flow indication instrument-XSJ Series flow integrating instrument. Also, it may combine with KF500 /TBS converter display to form a integral type to measure and indicate flow rate and quantity of a liquid, to pulse output signals and/or standard current signals and to provide power-off data protection. The velocity of vapor phase enters the high pressure heat exchanger and leaves the low pressure heat exchanger was measured by an **ultrasonic** flow meter when was a type of flow meter that measured the velocity of a Freon with ultrasound to calculate volume flow. It used the propagation of sound waves in moving fluid to determine flow rate by measuring the time required for the waves to traverse inside it. This device consists of a pair of transducers which emit and receive ultrasonic impulses. Waves are emitted both with and against fluid flow. The difference in transit time of the upstream and downstream waves is proportional to the velocity of fluid. Meters can be placed within the pipe or clamped to its exterior surface. The most important parameter measured is temperature .Many locations need to know the work fluid temperature in it. The flowing instruments were used to measure temperature. **Copper-constantan thermocouples type (K)** was extensively used for temperature measurements in the range -200°C to +1250°C evaporator and condenser. **High pressure transducer sensors** were used to record the pressure at different points on the condenser pipe .The pressure transducer sensors range is (0 to 50) bar Figure (6) shows a schematic diagram for condensation process test rig .



1 Condensation heat exchanger3 Temperature Recorder 5 Interface 7 Liquid flowmeter
2 Thermocouples 4 High pressure transducers 6 Computer 8 Vapor ultrasonic flowmeter

Figure 6 Schematic diagram of condensation rig

The location distribution of thermocouples was chosen to cover maximum points that have to catch the temperature distribution along axis and planes. A new method was used to support thermocouple .The previous studies presented that an acceptable reading of thermocouple ±1% when it supported on the surface of pipe. In this study pipes drilled with 5 mm holes to support a 5mm copper pipe contain the thermocouple junction bead, the pipe head was closed to prevent fluid leakage in same time its attaches the fluid inside pipe was shown in figure (7). This method is compared continuously with conventional method .The difference between two readings was about 3°C. All measurement equipment's were calibrated by measuring actual readings with different instruments and found that the percentage errors between .4$\mp$4% to $\mp$ 3%.

From experimental work some parameters can calculated such as heat generation which was depending upon the volume of heat exchanger (condenser) .This value about 344000 w/m³ .Heat flux which depending upon the area of condenser which about 30928,923 w/m² .Also , the velocity of working fluid leaves the condenser was calculated depending upon the volume flow of liquid phase which measured by turbine flow meter and area of outlet pipe which about 1.326m/s .This value calculated after calibrated the measuring value of flow meter

Figure 7 Thermocouple fixing

5. Results and discussion

Figure (8) shows the isothermal contour of temperature distribution through (x-z) plane in vapor phase at 2100 seconds.

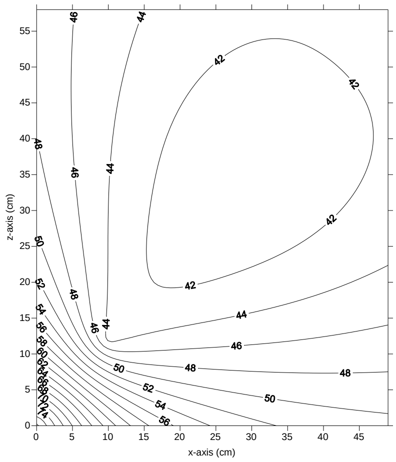


Figure 8 Isothermal contour of temperature distribution through (x-z) plane

The high temperature region which represented the enter temperature of working fluid .This temperature value was near 80°C which it indicate that the vapor working fluid was super-heated. The temperature decreases as distance increasing .That is mean when going forward in condenser domain there was a reduction in temperature value ,but still in vapor phase. This value about (40 – 38) °C, above the of boiling temperature of working fluid (R-22) at the certain pressure which was measured through this process. The pressure value which was about (21.5) bar.

Figures (9) and (10) show a relationship between z –axis, y-axis and temperature at middle of high pressure heat exchanger

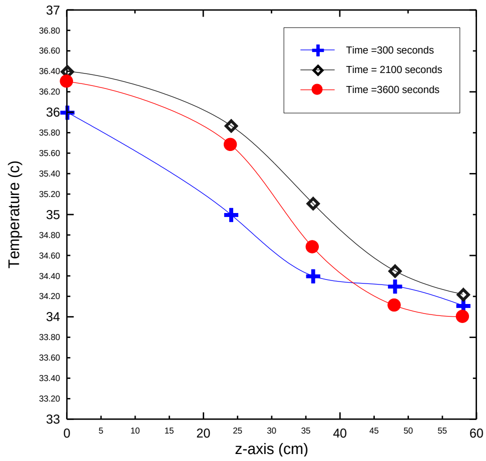


Figure 9 Temperature distribution through z-axis at phase change in three time intervals

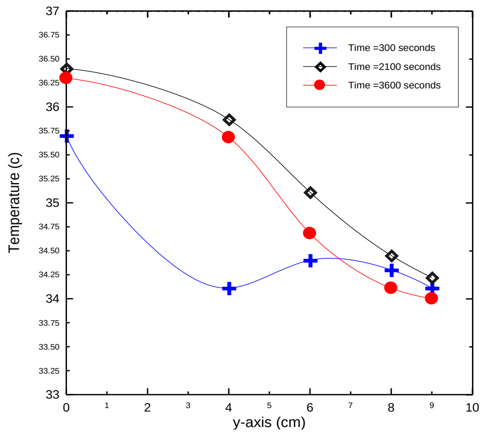


Figure 10 Temperature distribution through y-axis at phase change in three time

This line represents the phase change location. The values of temperature assisted between (34.5 to 36.5)°C. These values are the saturation temperatures (boiling) temperatures. At these values the phase change phenomena was happened. This means the working fluid transferred from vapor phase to liquid phase. These values of temperature were confirmed from standard tables depending upon the value of pressure which was measured at these points by pressure transducers .This value between (13.21- 13.9) bar which indicated from standards tables for boiling point of the working fluid R-22 in condensing process. These figures represented the values of phase change temperature in three time levels .It has been shown alternative decrease in temperature and rapidly decreases in temperature between the start and the end of axes.

Figure (11) describes the isothermal contour of temperature distribution through x-z plane in liquid phase at two time intervals.

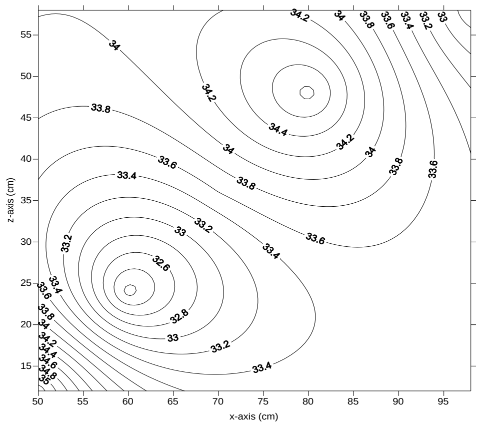


Figure 11 Isothermal contour of temperature distribution through (x-z) plane in liquid phase at time 2100 seconds

It has been shown the temperature of liquid phase from 34.8 °C and reduction as gone forward in the plane until 32 °C .The pressure at this phase was measured between 13.10 to 13.32 bar .It was indicated that also the pressure reduction in liquid phase than vapor phase in condenser. That was proved an experimental results than another workers the pressure in condenser was not constant .This is due to in this experimental work was measured the temperature and pressure of working fluid directly.

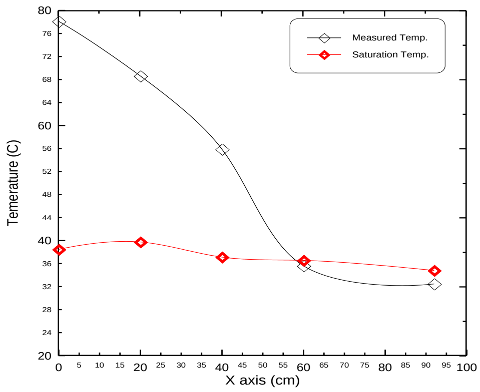


Figure 12 Measured and saturation temperature distribution through x-axis at 300 seconds

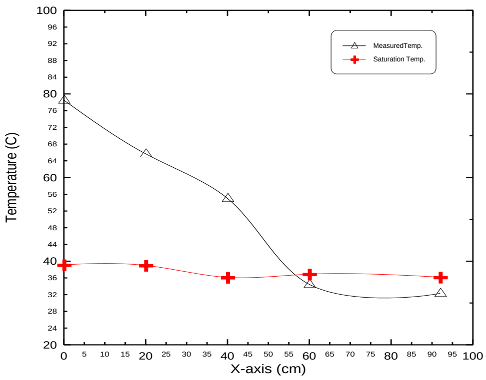


Figure 13 Measured and saturation temperature distribution through x-axis at 2100 seconds

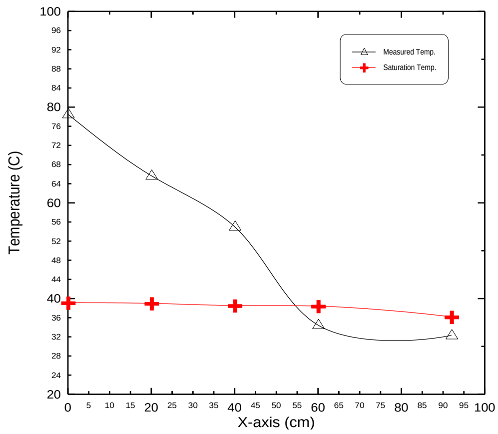


Figure 14 Measured and saturation temperature distribution through x-axis at 3600 seconds

Figures (12), (13) and (14) demonstrate the condensation process. It has been shown the variable temperature in vapor phase and liquid phase through the front view of condenser (high pressure heat exchanger). It has been shown how the temperature distribution through the condensation process from high temperature at vapor phase which was about 80 °C. Then ,the temperature reduction in liquid phase which was about 32°C. Also, indicated the saturation temperature (boiling temperature) which was represented the phase change line which variable with time. It has been started at middle of condenser. In figure (12), it has been started at 59 cm in x-axis and the boiling temperature value was between 36.5 to 34.5 °C. In figure (13) It has been started at 53 cm in x-axis, while in figure (14) started at 51 cm in x-axis with time intervals 300, 2100, and 3600 seconds respectively. These results confirmed with (**Fadhil *et al*, 2013).**

Figure (15) demonstrates the isothermal contour of condensation process in high pressure heat exchanger at two time values .It has been shown all the stages of condensation process indicated by temperature values. These stages starting from vapor phase and how the phase change was happened at interface line, and how the vapor change to liquid due to the condensation process.

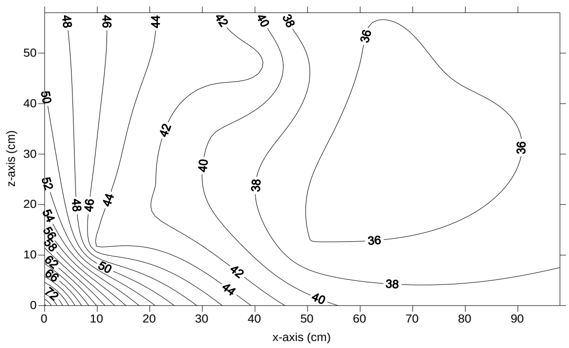


Figure 15 Isothermal contour of condensation process through front view of high pressure heat exchanger at time 2100 seconds

Figure (16) notes the vector velocity distribution in high pressure heat exchanger through condensation process. The long line vector represented the high value of velocity entering in the condenser which was measured experimentally by ultrasonic flow meter, which was3.14 m/s. While the small vector represented 1.326 m/s. It has been shown the velocity value of vapor phase was reduced after the interface line in liquid phase.

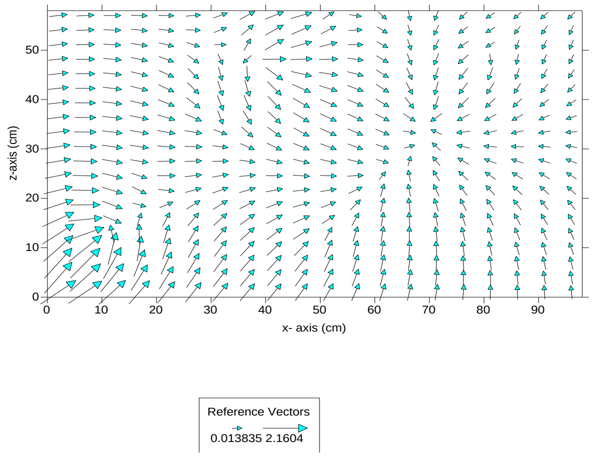


Figure 16 Vector velocity distribution in high pressure heat exchanger through condensation process

Figure (17) shows the pressure distribution through the domain plane of high pressure heat exchanger (condenser) after the condensation process happened in heat exchanger .It has been shown that the pressure values of working fluid variables and has different values through x-axis at different time interval of measuring. It has been used pressure transducer sensor as used pressure measurement instrument to measure the pressure values of condenser domain through condensation process .That means when the phase change phenomena was happened .Through this process the vapor of working transformed to liquid inside the surface area of condenser.

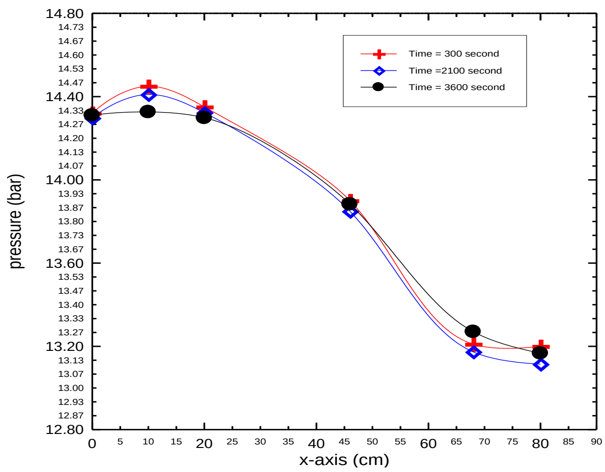


Figure 17 Pressure variation along high pressure heat exchanger

It has been recorded with different time intervals, pressure values in area of vapor phase of vapor phase, phase change location and area of liquid phase. All these pressure values recorded a variable values of pressure between 14.47 to 13 bar .These values of pressure proportional to temperature. This figure points that pressure value was between 14.47 to 14.27 bar in the space area of condenser at vapor phase with high level of temperature values as superheated .While it has been measured values between 13 to 12.9 bar in space area at the end length of condenser .These values proportional to temperature of working fluid .It has been found the temperature values lower than in space area of beginning the length of condenser . These values of temperature and pressure indicated the working fluid in liquid phase. Also, other pressure transducers were fixed in middle of condenser in order to measure the saturation pressure which was pointed the boiling point. This spaces was represented the phase change location. The values of pressure in this location was recorded between 13.2 to 13.4 bar with different intervals time. These pressure values were indicated the boiling point of working fluid R-22 through condensation process from standard table of properties. It has been found the boiling temperature of R-22 (36.5) depending upon the values of pressure measurement in this location. Figure (18) describes the contour map of pressure distribution in x-z plane at time level of 300 seconds.

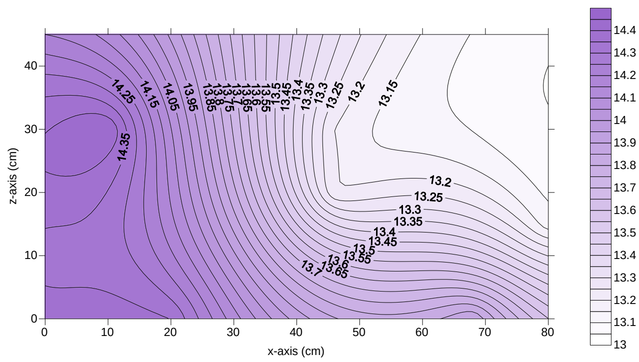


Figure 18 Pressure contour map through x- z plane in high pressure heat exchanger at time 300 seconds

It has been found the values the pressure at high level at entering space area and the value starting redacting at the end of condenser domain .That represented the level at vapor phase to lower level at liquid phase.

Figure (19) demonstrates the isothermal contour of temperature distribution through (x-z) plane in vapor phase at time 2100 seconds .It has been shown that the temperature reduction as going forward in condenser domain. The temperature as superheated vapor entering the condenser at different values of time between 80°C to 76°C depending upon the time being measured from experimental work and these reduction values to 39°C, which was still in vapor phase.

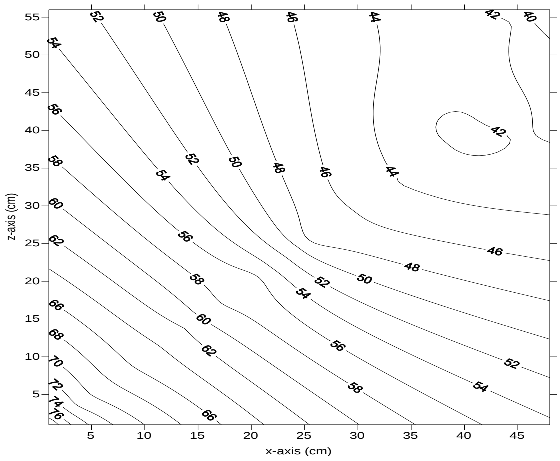


Figure 19 Isothermal contour of temperature distribution through (x-z) plane in vapor phase at time 2100 seconds

Figure (20) observes the temperature values of phase change phenomena, which was occurred at interface line as assumed in middle the condenser domain with values of time level through z-axis. It has been associated between 35.5°C to 35.5°C at three time levels.

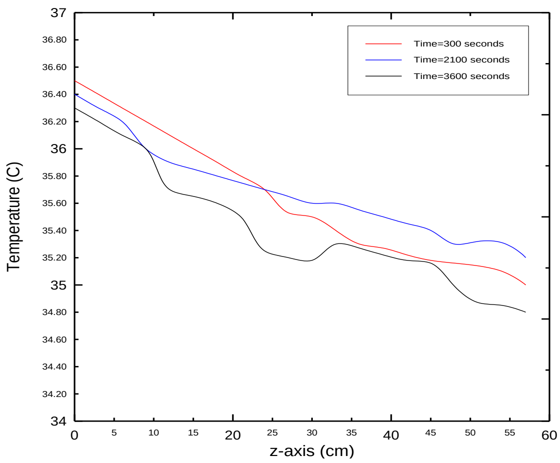


Figure 20 Temperature distribution through z-axis at phase change in three time intervals

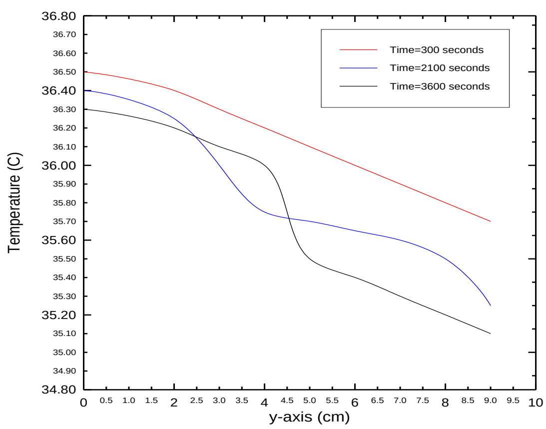


Figure 21 Temperature distribution through y-axis at phase change in three time intervals

While figure (21) introduces the temperature values of interface line at y-axis .These figures show that values of phase change temperature (boiling point) distribution in point z-axis equal zero x-axis equal 50 cm through the z and y axis .These values of temperature were the boiling points (boiling temperatures) of working fluid R-22. These were approved from standard tables of thermal properties of R-22 depending upon the pressure values which were measured experimentally at this space of condenser domain.

Figure (22) describes the isothermal contour through x-z plane in liquid phase in condenser at two time levels.

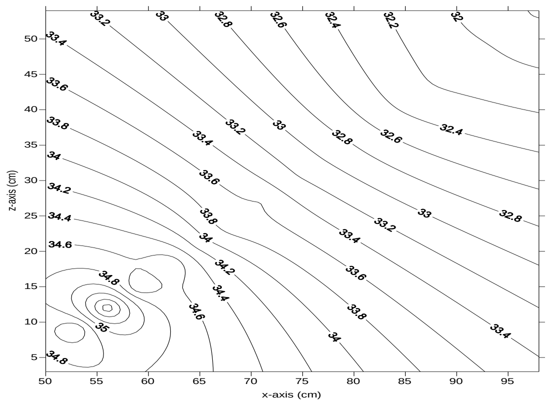


Figure 22 Isothermal contour of temperature distribution through (x-z) plane in liquid phase at time 2100 seconds

These values of temperature predicted after the interface line of the domain assumption of condenser. The liquid phase was produced after the phase change phenomena happened. That means the vapor temperature reduction and all the molecules of vapor changed from vapor to liquid after across the boiling point of R-22 .It has been noted that the working fluid in condenser still high pressure but at different values of pressure in vapor, interface, and liquid .These values of temperature below the boiling point of working fluid R-22.The working fluid properties depending upon the value of pressure and the case of processing .It has properties in condensing process different than other process.

Figure (23) shows isothermal contour of temperature distribution through (x-z) plane in all domain of condenser through condensation process at two time levels.

This process represented by three stages firstly, the vapor phase with super heat temperature and the vapor started to reject the heat energy through this process. Then, the phase change to every molecules of vapor which was lost high level energy in vapor phase and go to second stage of interface line at the boiling point. The third was the liquid phase that means the transform of the vapor phase to liquid phase. It has been shown that the temperature values represented the vapor, interface line and the liquid phase.

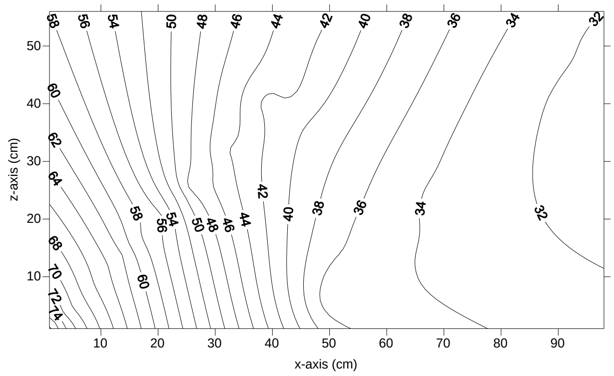


Figure 23 Isothermal contour of condensation process through front view of high pressure heat exchanger at time 2100 seconds

Figure (24) describes the comparison between the experimental and theoretical work through condensation process.

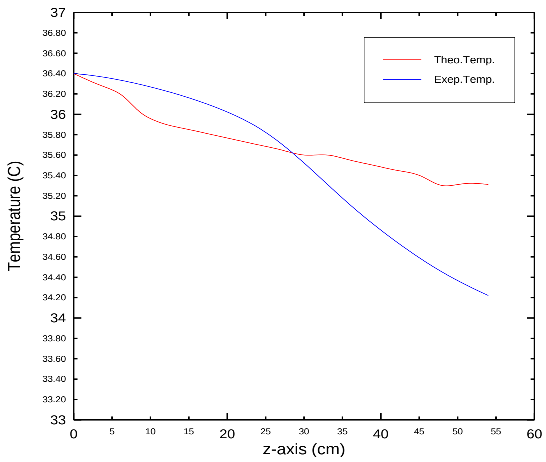


Figure 24 Comparison between the experimental and theoretical work of phase change temperatures through condensation process at time 2100 seconds

It has been indicated the temperature distribution of phase change through z-axis in condenser at interface line which was predicted in middle of domain at 2100 seconds. It has been noted that the theoretical and experimental results are consistence together. That means the experimental results prove the simulation prediction on model of temperature distribution of phase change phenomena. It has been shown the percentage error between 6% to 10%. Figure (25) illustrates the comparisons between experimental and theoretical work through the condensation process at two time levels .It has associated all the stages which were happened in the front view of condenser .That means through x-axis. It has been described how the vapor reduction from high temperature value 80°C in vapor phase to low value of liquid phase 30°C. Also, indicated the boiling temperature which was the point when the phase change happened from vapor to liquid phase. It has been noted that the results of experimental work were confirmed the theoretical model prediction .It has been illustrated the percentage error between the theoretical and experimental/ works between 4% to 10 %.

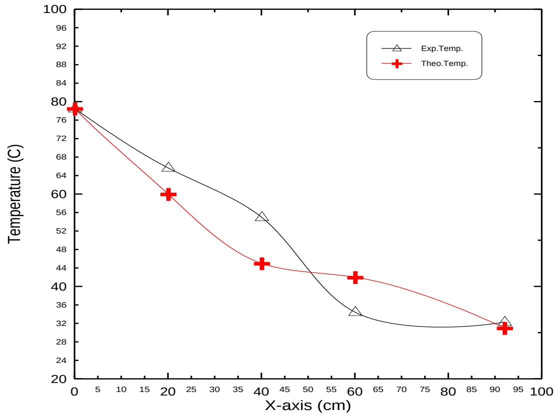


Figure 25 Comparison between the experimental and theoretical work of phase change temperatures through condensation process at time 2100 seconds

Conclusions

1. Temperature distribution through the condensation process was calculated in different points of condenser domain at three stages of energy i.e. vapor phase, phase change (interface) and liquid phase.
2.The numerical solution of FTCS method gives more reliability of stability for solving the theoretical model .
3. Through experimental work the temperature, velocity, pressure and volume flow rate were measured by advance instruments with high accuracy .These were calibrated in order to correct the readings results through condensation process.
4. The velocity of phase change is calculated theoretically depending upon heat flux, latent heat ,difference temperature between two phases ,thermal and physical properties of working fluid.
5. Theoretical results of simulation model are consistent with experimental work and with **(Anghai and Dang ,1996)** work.
6.Boiling point was found depending upon the pressure values measurement which was about 36.5°C.This value was found from standard tables of refrigerant properties (R-22) .**(Standard tables, 2005).**

Nomenclatures

A-Area (m^2)
C-Courant number
c_p-Specific heat capacity at constant pressure (J/kg.°C)
k-Thermal conductivity (w/m. °C)
P-pressure (N/m^2)
t-Time (second)
V_v-Velocity of vapor phase (m/s)
V_l-Velocity of liquid phase (m/s)
L-number of nodes in x- direction
M-Number of nodes in y- direction
N-number of nodes in x- direction

Δx-Increment distance through x-axis (m)
Δy-Increment distance through y-axis (m)
Δz-Increment distance through z-axis (m)
Δt-Increment time (second)
K_l-Thermal conductivity of liquid phase (w/m. °C)
k_v-Thermal conductivity of vapor phase (w/m. °C)
V_* Velocity of interface from vapor to liquid condenser (m/s)
α_v-Thermal diffusivity of vapor phase (m^2/s)
α_l-Thermal diffusivity of liquid phase (m^2/s)
ρ_l -Density of liquid phase (kg/m^3)
ρ_v-Density of liquid vapor (kg/m^3)
λ- Fourier number
$\delta(t)$-The location of interface line m
L_v-Latent heat of vaporization (J/kg)
T_v-Temperature of vapor (°C)
T_l -Temperature of liquid (°C)

References

Rita SZIJÁRTÓ (2015), Condensation of steam in horizontal pipes ; model development and validation Ph.D. ,Thesis ,Budapest university

Kondou,C.,and rnjak, (2012), Heat Rejection in Condensers: Desuperheating, Condensation in Superheated Region and Two Phase Zone, 1–10.International Refrigeration and Air Conditioning Conference at Purdue, July 16-19, p.2503

Juric, D. (1996), Computations of Phase Change.A thesis Doctor of Philosophy (Mechanical Engineering) in The University of Michigan.

Holman J.P, (2010), Heat Transfer Book , published by McGraw -Hill ,USA ,Tenth Edition

Andersion ,D.A.Tennchile ,J.C. , and Plectcher ,R.H., (1984) Book,Computional Fluid Meechanics and heat transfer,Book, Second Edition McGraw Hill

Kim .y, Hossain .A.,and Nakamura.y. (2013),Numerical study of phase change material enhanced by deformation of material of a liquid-gas interface , *International journal of heat and mass transfer* .Vol .63 pp.101 -122 .

Steen,W.M., and Kamalu j.N., (1983),laser cutting Book, laser processing Bass ,M.,ed.,north –Holland publishing company.

Fadhl, B., Wrobel, L. C., & Jouhara, (2013), Numerical modelling of the temperature distribution in a two-phase closed thermosyphon.*J. Applied Thermal Engineering*,60(1-2),pp.122–131.

Anghaie, S., & Ding, Z. (1996),Modeling of Bulk Evaporation and Condensation, Technical report prepared for Lewis Research center under contract NAS3-Z6314.

Standard tables (2005), Thermodynamic Properties of Du point Feron (R-22) Refrigrant .