THE EFFECT OF REFRIGERANT VAPOUR FLOW FLUCTUATION ON CONDENSATIONPARAMETER "AN ANALYTICAL MODEL APPROACH"

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Abstract. The condensation process in the condenser unit is affected by some factors affiliated with heat transfer problems which include fouling resistance effect, corrosion and fluid flow fluctuation among others. Hence this research was carried out to investigate the relationship between refrigerant vapour flow fluctuation and condensation process parameters. In this study, an analytical model was developed to determine the relationships between refrigerant vapor flow rate and condensation parameters which includes flow rate of condensate, rate of condensation dimensionless parameterand condensate film thickness using laws, assumptions and boundary conditions from relevant literatures. The model developed predicted a direct linear relationship between refrigerant vapor flow rate and condensate flow rate, an inverse linear relationship between refrigerant vapor flow rate and rate of condensation dimensionless parameter and an inverse linear relationship between refrigerantvapor flow rate and condensate film thickness. The model was tested using a plant data collected from an LNG company in Nigeria.The plant data plotted agree with the predictions of the model for upper layer film thickness, side layer film thickness, lower layer film thickness and average film thickness with percentage fitting deviations of 6.50%, 4.20%, 94.60% and 4.20% respectively. Also the plant data plotted agree with the predictions of the model for refrigerant condensate flow rate with percentage fitting deviation of 3.40%

Keywords; vapourcompression refrigeration cycle. Condenser.refrigerantvapour flow rate.condensation parameters.

1 Introduction

The condenser, in a vapour compression refrigeration system, is one of the most influential units of the system. Its performance strongly affects the amount of cooling effect experienced by the cooling system and the amount of compressor power required to run the refrigeration system regardless of the cycle configuration.

According to Ajeet et al (2013), the condenser is a heat transfer unit designed to condense fluid from gaseous to liquid state through losing of their latent heat of vaporization to another fluid (condenser coolant). Rajput (2007) viewed Condensers as basically heat exchangers in which the refrigerant undergoes a phase change (from gas phase to liquid phase). A condenser is a device or an appliance in which fluids condense and heat released by fluid is absorbed by another fluid (coolant). Condensers are important heat and mass exchange apparatus in oil refining, chemical engineering, environmental protection, electric power generation, (Master, Chanangad and Pushparathan 2003).. In condensers the refrigerant vapor condenses by rejecting heat to an external fluid, which acts as a heat sink. Condensation set in whenever a saturated vapor comes in contact with a surface whose temperature is lower than the saturated temperature corresponding to the vapor pressure. The main purpose of the condenser in a vapour compression refrigeration cycle is to extract the heat lost by the cooling fluid to the refrigerant during cooling and transfer it to the environment via an indirect heat flow contact between the refrigerant and the cooling water.

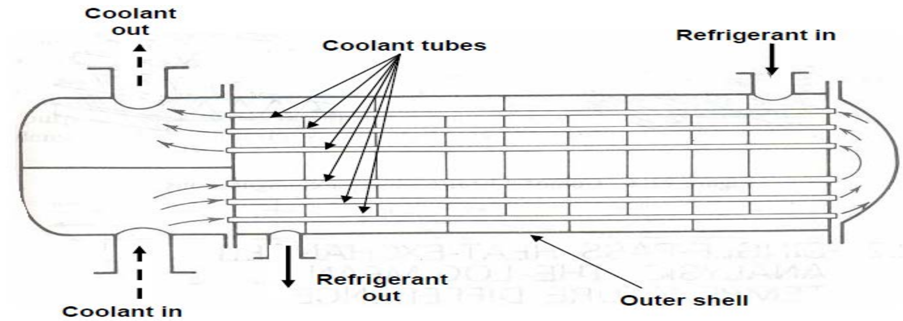


Figure 1; shells and tube condenser, source refrigeration and air condition (Kharagpur 2008)

In a standard condenser unit, the refrigerant usually flows into the condenser unit in a super heated state. It is then de-superheated firstin the condenser unit and then condensed by rejecting heat to a coolant usually an external fluid mainly water. The refrigerant fluid may leave the condenser unit in three forms one, saturated liquid two, sub-cooled liquid, and finally as two phase fluid depending on the temperature of the coolant, design, and configuration of the condenser. Figure 2 shows the variation of refrigeration cycle on T-s diagram. In the figure, the condensation process (heat rejection process) is represented by 2-3'-3-4. The temperature profile of the coolant (external fluid), which is assumed to undergo only sensible heat transfer, is shown by dashed line. It can be seen that process 2-3' is a de-superheating process, during which the refrigerant is cooled sensibly from a temperature T2 to the saturation temperature corresponding condensing pressure, T3'. Process 3'-3 is the condensation process, during which the temperature of the refrigerant remains constant as it undergoes a phase change process. Process 3-4 is a sensible, sub cooling process, during which the refrigerant temperature drops from T3 to T4 (Kharagpur, 2008,)

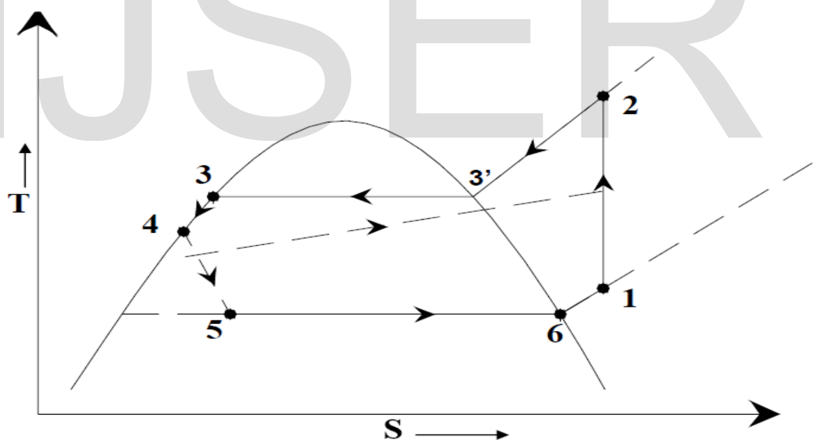


Figure 2; Refrigeration cycle on a T S diagram source Refrigeration and Air conditioning by ITT Kharagpur (2008)

The maain aim of empirical and theoretical studies into the performance of the condenser and heat exchanger in general is to effectively and accurately predict the condensers behavoir with respect to its characteristics so that the designers can develop a smaller, cost effective and reliable condenserswhich will meet the same thermal duty requirement. Over the past few decades, the theoratical studies carried out to analyse the laminar film condensation process over horizontal circular tube is significant and that has contributed emmencely in predicting the heat tranfer behavoir and characteristic of the condenserand as such lead to improvement in condenser design.

According to Adams (1993) majority of the theoretical researches werefocusedand tageted at controlling and reducing the condensate film thickness as it is proved to be the major resistance to heat transfer processwhich controls the condensation process and these studies were carried out using the combined laws on conservation of mass, momentum and energy with appropriate assumptions and boundry conditions. He also pointed out that condensate film thickness is dependent on interfacial shear between the vapour and liquid layer boundry and rate of condensation

The geometry of the tube is another factor that affects the heat transfer performance of a condenser unit. Research carried out by many author has confirmed that elliptical tubes has higher heat transfer performance than circular tubes of similar surface area.

Ota and Nishiyama (1984) in an empirical research carried out to determine the effect of heat exchangers geometry on the heat transfer characteristics and performance concluded that elliptical cylinders of eccentricity 1.5 gave a better heat transfer performance than a circular cylinder counterpart. Moalem and Sideman (1975)in a theoretical research carried out on the horizontal elliptical tube condenser-evaporator unit using the law of conservation of mass and momentum discovered that an elliptical cylinder tube showed 10-20% heat transfer enhancement than the circular cylindrical counterpart.

Researcher like Sparrow and Gregg (1959), Shekrilaze and Gomelauri (1966), Fuji et al (1979), Gaddis (1979) and Rose (1884) independently studied the effect of inertia, interfacial shear, pressure gradient and surface tension on the heat transfer performance of the condenser under different boundary conditions using the laws of conservation of mass momentum and energy. And their individual contributions are very helpful in the design of condenser and heat exchanger.

Other factors that affect the condenser heattransfer performance which will hinder condensation process are(1) fluctuating/changing refrigerant flow rate (2) fouling effect(3) corrosion.

Fluctuating refrigerant flow rate; Every condenser with a fixed heat capacity has a designed optimum refrigerant and coolant flow rate for an effective and complete condensation of the refrigerant for optimum performance of the refrigeration cycle. Any alteration in this designed flow rate effect the degree of condensation of the refrigerant thereby affecting the effectiveness of the condenser. This alteration is usually caused by fluctuation in the power supply to the compressor and pressure drop due to friction along the condenser.

Fouling; When fluid flows through a pipe or tube continually for a long time, there are usually traces of the fluid and dirt particles that will settle on the walls of the pipe or tube forming slippery layers. This process is called fouling effect. When this fouling occur in the inner and outer wall of the condenser due to continua refrigerant flow, they form resistance and by so doing reduce the heat transfer between the refrigerant and the coolant. This in turn reduces the degree of condensation of the refrigerant which is dependent on the amount of heat transfer between the coolant and the refrigerant.(Eastop&McConkey 1993)

Corrosion; corrosion is a gradual deterioration of material as a result of its contact with its immediate environment. Hence there are tendencies of occurrences of corrosion in the inner and outer walls of the condenser as a result of interaction between the metallic walls of the condenser and its environment. These corroded surfaces form a great resistance to heat transfer between the refrigerant and the coolant thereby reducing the amount of heat transfer between the refrigerant and the coolant. Hence, reduction in the degree of refrigerant condensation which result in ineffectiveness of the condenser

1.1 Statement of problem

Under ideal and optimum operation of the condenser unit of the vapour compression refrigeration cycle, the refrigerant fluid always enters the condenser as vapour but leaves as complete liquid (fully condensed). But in real operational condition, the outlet fluid leaves with some vapor fraction (partially condensed)

Eastop and McConkey (1993) indentified that one of the reasons for the deviation from ideal to real condition is usually as a result of ineffectiveness of the condenser arising from the reduction of the heat transfer performance of the condenser due to corrosion, fouling effect and refrigerant flow rate fluctuation and this incomplete condensation of the refrigerant has an effect on the performance of the entire vapor refrigeration cycle regardless of the cycle configuration. Eastop and McConkey (1993) detailed the effects of corrosion and fouling effect on the heat transfer performance of the condenser which will lead to incomplete condensation but effect of the fluctuation of the flow rate of the refrigerant on condensation process was only a whisper. Hence we are posed with this question “what is the relationship between refrigerant vapour flow rate and condensation process parameters”?

Nevertheless, the research below was designed to answer the questions comprehensively and explicitly because it is extremely important to understand the factors that affect condensation process in a condenser unit so as to run the condenser unit effectively knowing it’s important in the refrigeration cycles.

1.2 Aim and objectives

The main aim of the research is to demonstrate the effect of refrigerant vapour flow rate on the condensation process parameters. The objectives are to determine the relationship between the refrigerant vapour flow rate and the three main condensation parameters namely rate of condensation dimensionless parameter, condensate flow rate and condensate film thickness.

1.3 Scope of the research

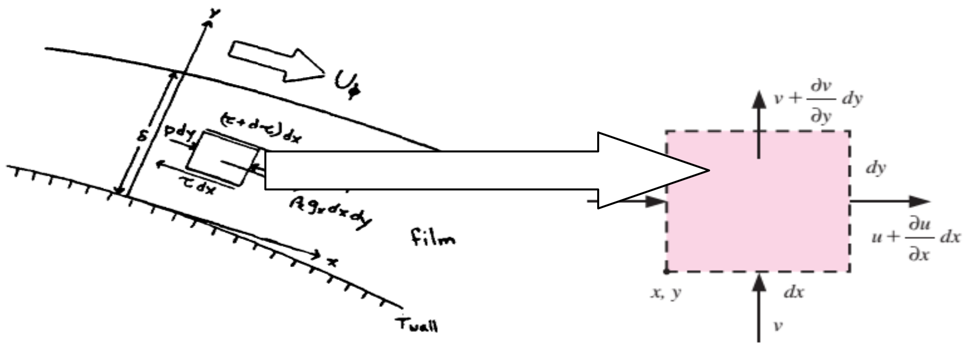
Ananalytical model will be developed based on laws of conservation of mass, momentum and energy on a control volume of thin film condensate under forced convection (Rajput 2007) assumption of Nusselt as used by Adams (1993) and boundary conditions of Shekrilaze and Gomelouri (1966)

2 Model development methods

Consider a pure saturated refrigerant vapor at temperature (T_{sat})flowing with velocity of Uϕ into a shell and tube condenser with tube external surface temperature (T_W) less than the T_{sat} such that the vapor condensed on the outer surface of the horizontally arranged elliptical cylinder tube of eccentrically k. to analyzed the thermodynamic of the phenomena, the following assumption proposed by Nusselt (1916) and used by Adams (1993) is adopted. The assumptions are

a) The vapor is pure and homogenous
b) The condensation is purely film wise condensation
c) The tube wall temperature T_W is uniform and constant.
d) The temperatureT_{sat} of the film/vapor layer is uniform and constant.
e) The condensate film thickness is small compare to the radius of the ellipse
f) The heat transfer through the film is in radial direction only and in linear temperature distribution.
g) The only force acting on the condensate film is viscous force (forced convection)
h) The flow in the condensate film is linear and there is no boundary separation
i) The stream wise velocity U_ϕ >u_y y = δ
j) The interfacial shear stress is assumed to be τ = m(U_ϕ – u_m)

Consider a small section of the condenser tube cut off and the flow analysis is carried out on a finite unit of the condensate film as shown below.



Using the law of conservation of mass on the control volume above

Mass in –mass out = mass stored, considering only flow in direction of condensation on unit cylinder length.

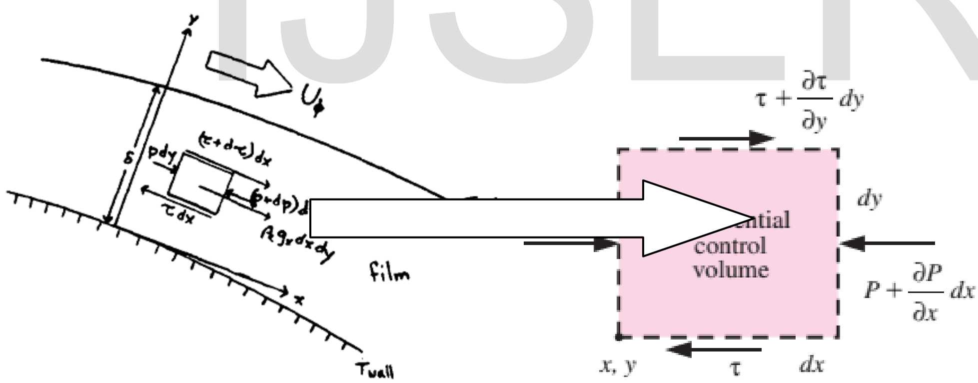
$$\left[\rho_l \int_0^\delta u\ dy\right] dx - \left[\rho_l \int_0^\delta u\ dy + \frac{d}{dx}\left\{\rho_l \int_0^\delta u\ dy\right\}\right] dx + m\ dx = 0 \qquad \textbf{(1)}$$

Where **m** is the local condensate flow rate and $\mathbf{u_m} = \frac{1}{\delta}\int_0^\delta u\ dy$ = mean film velocity, Substituting

$\mathbf{u_m} = \frac{1}{\delta}\int_0^\delta u\ dy$in equation **(1)** aboveWe have

$$\mathbf{m} = \frac{d}{dx}(\rho\ u_m \delta) \qquad \textbf{(2)}$$

Conservation of momentum on the finite confined volume is given below



Assuming that the only force acting on the film element is viscous force, Assuming that the fluid is Newtonian

$$\tau = \eta \frac{\delta u}{\delta y} \qquad \textbf{(3)} \text{and } \textbf{area = dy} \times \textbf{dx} \qquad \textbf{(4)}$$

Net force on the condensate film

$$\mathbf{F} = ((\tau + \Delta\tau) - \tau) \times \textbf{area} \qquad \textbf{(5)}$$

Substituting equation (3) and equation (4) in equation (5) we have

$$\mathbf{F} = \left((\eta \frac{\delta u}{\delta y} + \frac{\delta}{\delta y}(\eta \frac{\delta u}{\delta y})\right) - \eta \frac{\delta u}{\delta y}) \times \mathbf{dydx}$$

Simplifying the equation above, we have

$$\mathbf{F} = \boldsymbol{\eta} \frac{\partial^2 u}{\partial y^2} \mathbf{dydx}$$

But the net summation of the forces on the fluid element is zero, hence the momentum equation becomes

$$\boldsymbol{\eta} \frac{\partial^2 u}{\partial y^2} = 0 \qquad \textbf{(6)}$$

(Assuming that momentum loss due to pressure loss and gravity is zero)

Conservation of energy is the balance of latent heat from condensation (heat of fusion) and heat conducted through the condensate film thickness. The expression of the balance is given below

$$\mathbf{m\ L_f\ dx} = \frac{k}{\delta} (T_{sat} - T_W)\ \mathbf{dx}$$

Making **m** the subject of the formula where $(T_{sat} - T_W) = \Delta T$

$$\mathbf{m} = \frac{k\ \Delta T}{L_f\ \delta} \qquad \textbf{(7)}$$

The boundary condition as used by **shekriladze and Gomelauri (1966**) are as follows

at y $= 0$ $\mathbf{u_m} = 0$ and $\frac{du}{dy} = 0$ (no slip condition)

At $y = \delta$ $U\phi > u_m$ and $\eta \frac{du}{dy} = mU_\phi$

Integrated the momentum equation subject to boundary condition above gives the local velocity function below

$$u_y = \frac{mU_\phi}{\eta} y \qquad \textbf{(8)}$$

Hence, the mean film velocity

$$u_m = \frac{mU_\phi}{\eta_C} \frac{\delta}{2} \qquad \textbf{(9)}$$

Introducing the dimensionless velocity function f (ϕ) used to express the geometry relations for forced convection as proposed by **Adams (1993**) which is given as

$\mathbf{F}\ (\phi) = \frac{1+k}{\sqrt{1+k^2 cos^2\phi}}$ when k = 1, the tube is circular. K > or < k 1, the tube is elliptical, k = 0, vertical flat plate, k = ∞ horizontal flat plateHence

$$u_m = \frac{mU_\phi}{\eta_C} \frac{\delta}{2} \mathbf{F}\ (\phi) \qquad \textbf{(10)}$$

Substituting m in equation (7) in equation (10)

$$u_m = \frac{k\ \Delta T\ U_\phi}{2\eta_C L_f} \mathbf{F}\ (\phi) \qquad \textbf{(11)}$$

Substituting equation (11) in equation (2) I have

$$\mathbf{m} = \frac{d}{dx}\left[\frac{\rho_c K\ U_\phi\ \Delta T\ \delta}{2\ \eta_C\ L_f} \mathbf{F}\ (\phi)\right] \qquad (12)$$

For the condensate film at steady state, the expression $\frac{\rho_c K\ U_\phi \Delta T \delta}{2\eta_C L_f}$ is constant, then equation (12) becomes

$$\mathbf{m} = \frac{\rho_c K\, U_\phi \Delta T \delta}{2\eta_C L_f} \frac{d}{dx}[\mathbf{F}\,(\phi)] \qquad (13)$$

Using the dimensionless parameter G as defined and proposed by **Fuji et al (1979)** used to express rate of condensation below

$$\mathbf{G} = \frac{K\,\Delta T}{\eta_C L_f}\sqrt{\frac{\rho_c\,\eta_C}{\rho_v \eta_v}} \qquad (14)$$

substituting $\mathbf{H} = \sqrt{\frac{\rho_c\,\eta_C}{\rho_v \eta_v}}$ in equation (14) above, we have

$$\frac{\mathrm{G}}{\mathrm{H}} = \frac{K\,\Delta T}{\eta_C L_f} \qquad \mathbf{(15})$$

Substituting $\frac{K\,\Delta T}{\eta_C L_f}$ in equation(13) with $\frac{\mathrm{G}}{\mathrm{H}}$in equation (15), we have

$$\mathbf{m} = \frac{\rho_c\, G\, \delta\, U_\phi}{2H} \frac{d}{dx}[\mathbf{F}\,(\phi)] \qquad (16)$$

Making $U_\phi$the subject be dividing both sides by $\mathbf{m}U_\phi$ in equation (16) above we have

$$\frac{1}{U_\phi} = \frac{\rho_c\, G\, \delta}{2Hm} \frac{d}{dx}[\mathbf{F}\,(\phi)] \qquad (17)$$

Using that mass flow rate of refrigerantvapour in the condenser is given as

$$\mathbf{M} = \rho_V U_\phi A,$$

Making $U_\phi$the subject and substituting it in (3-17) above, I have

$$\frac{1}{M} = \frac{\rho_c\, G\, \delta}{2\,\rho_V\, H\, m\, A} \frac{d}{dx}[\mathbf{F}\,(\phi)] \qquad (18)$$

Substituting the value of $\mathbf{F}\,(\phi)$in equation 3-18 above, I have

$$\frac{1}{M} = \frac{\rho_c\, G\, \delta}{2\,\rho_V\, H\, m\, A} \frac{d}{dx}\left[\frac{1+k}{\sqrt{1+k^2 cos^2\phi}}\right] \qquad (19)$$

(Note, the above is the modeled only for elliptical condensers of eccentricity k and potential flow angleϕ. At k=1, the equation become model for circular condensers.)

3 Model testing method

The model will be tested using data collected from an LNG company in Nigeria and the percentage fitting deviation between the model developed above and the plant data result will be calculated using the R^2 values of the graphs in the equation below

$$\text{Fitting deviation \%} = -\frac{R^2_{MODEL} - R^2_{IND\ DATA}}{R^2_{MODEL}} \times \frac{100}{1} \qquad 20$$

Using that the R^2 value of the model is always one which represent a perfect fitting of the relationship between the parameters, In that case, the equation above becomes

$$\text{Fitting deviation \%} = \frac{1 - R^2_{IND\ DATA}}{1} \times \frac{100}{1} \qquad 21$$

4 Results and discussions

The theoretical model developed to describe the relationship between mass flow rate of refrigerant vapor and condensation parameters is given below.

$$\frac{1}{M} = \frac{\rho_c \, G \, \delta}{2 \, \rho_V \, H \, m \, A} \frac{d}{dx} \left[\frac{1+k}{\sqrt{1+k^2 cos^2\phi}} \right] \quad (19)$$

Where **M** is the mass flow rate of refrigerant vapor, **Condensation parameters** are **G** rate of condensation dimensionless parameter, $\boldsymbol{\delta}$ condensate film thickness, and **m** the mass flow rate of the refrigerant condensate**. Fluid parameters** are $\rho_V$the density of refrigerant vapor**,** $\rho_c$density of refrigerant condensate and **H** the dimensionless fluid parameter. **Geometry parameters** are **A** the shell inlet cross sectional area **K** the eccentricity of the shape. Assuming that the fluid parameters and the geometry parameters are constant, the model could be expressed as follows

$$\frac{1}{M} = \frac{G \, \delta}{m} \, \mathbf{c} \quad (22)$$

Rearranging the equation above for easy linearization, we have

$$\mathbf{M} = \frac{m}{G \, \delta} \mathbf{C} \quad (23)$$

M is the independent variable while **G, m** and $\boldsymbol{\delta}$are the dependent variables. Making the dependent variables the subjects in equation (4-3) above, we have

$$\mathbf{m} = \mathbf{M} \frac{G \, \delta}{C} \quad (24)$$

$$\mathbf{G} = \frac{m}{M \, \delta} \mathbf{C} \quad (25)$$

$$\boldsymbol{\delta} = \frac{m}{MG} \mathbf{C} \quad (26)$$

Where C is constant representing geometric and fluid parameters Comparing the equation (4-3), (4-4), (4-5) and (4-6) above with the general equation of straight line **y=mx + c,** through linearization, the analysis will be as follows

1) The relationship between mass flow rate of refrigerant vapor (**M)** and all the three condensation parameters $(\mathbf{G\delta \ and \ m})$ is linear.
2) There is a direct relationship between mass flow rate of the refrigerant vapor **M** and mass flow rate of refrigerant condensate**m**. This is because increase in mass flow rate of the refrigerant vapor leads to increase in velocity distribution gradient within the condensate film layer. This increase in the velocity gradient will result to an increase in sweep within the vapor / condensate contact layer leading to increase in the condensation of the refrigerant vapour
3) There is an inverse relationship between mass flow rate of refrigerant vapor **M** and rate of condensation**G**. This is because increase in the mass flow rate of the vapor in the condenser result to reduction in the contact time between the vapor and the walls of the condenser thereby resulting in reduction in rate of condensation and hence reduction in condensation.
4) There is an inverse relationship between mass flow rate of refrigerant vapor **M** and condensate film thickness$\boldsymbol{\delta}$. This is because increase in mass flow rate of refrigerant vapor causes a greater sweep on the contact between the vapor/condensate layer which leads to reduction in the thickness of the film and also resulting to increase in the rate of heat transfer between the vapor and the walls of the condenser therefore leading to increase in vapour condensation.

4.1 Testing of condensate film thickness model,

The theoretical model show a linear and inverse relationship between the flow rate of refrigerant vapor and film thickness which means that the graphical representation of the relationship between vapor flow rate and condensate thickness will give a straight line graph sloping downward from left to right with R^2value of 1 and negative slope.Comparing this with the graphical representation of the relationship between refrigerant vapor flow rate and condensate film thickness using plant data of an elliptical condenser from an LNG company in Nigeria as shown below

The data show that refrigerant condensate film thickness is measured in three main positions namely the upper layer, the side layer and the lower layer. Then the average layer thickness was also calculated and plotted. The graphs of these parameters are shown below.

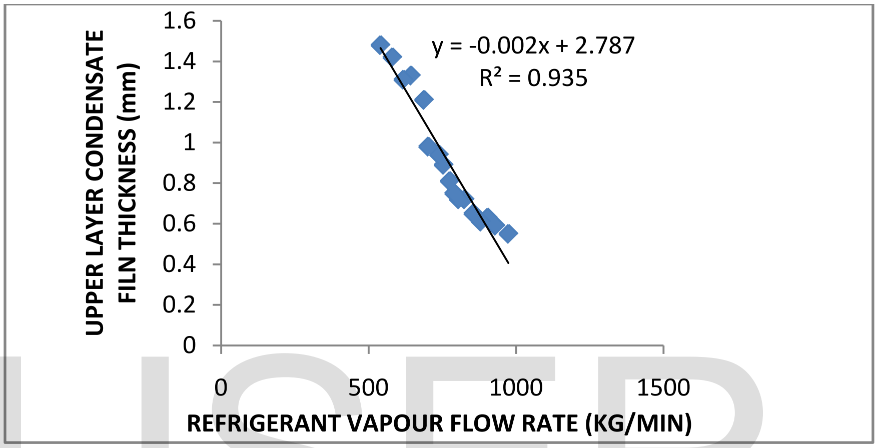


Figure 3;graph of upper layer condensate film thickness against refrigerant vapour flow rate

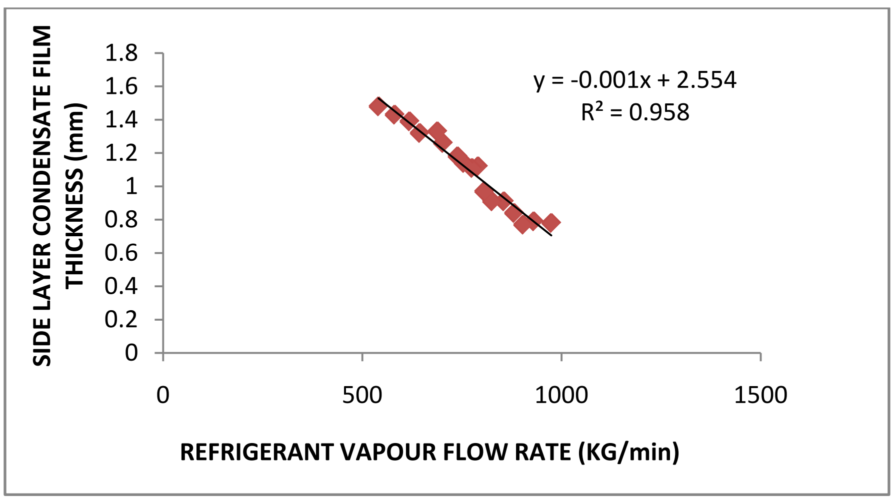


Figure 4;graph of side layer film thickness against refrigerant vapour flow rate

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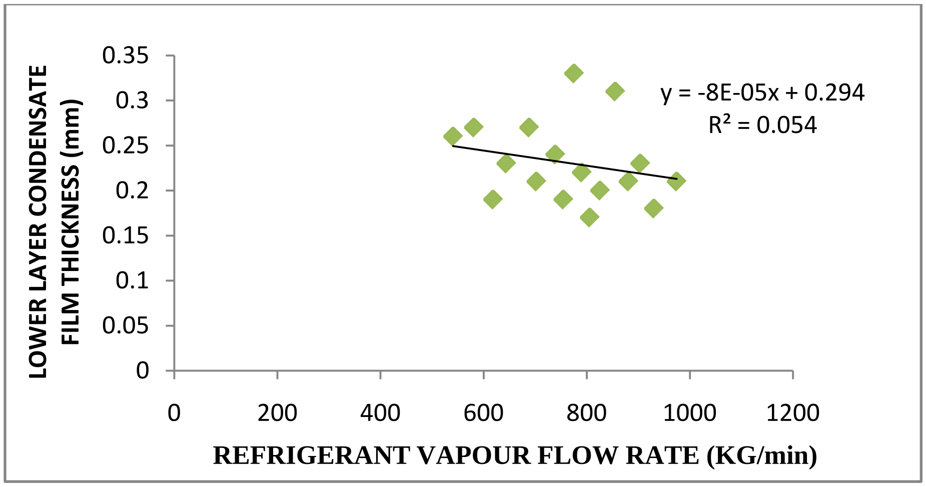

Figure 5;graph of lower layer condensate film thickness against refrigerant vapour flow rate

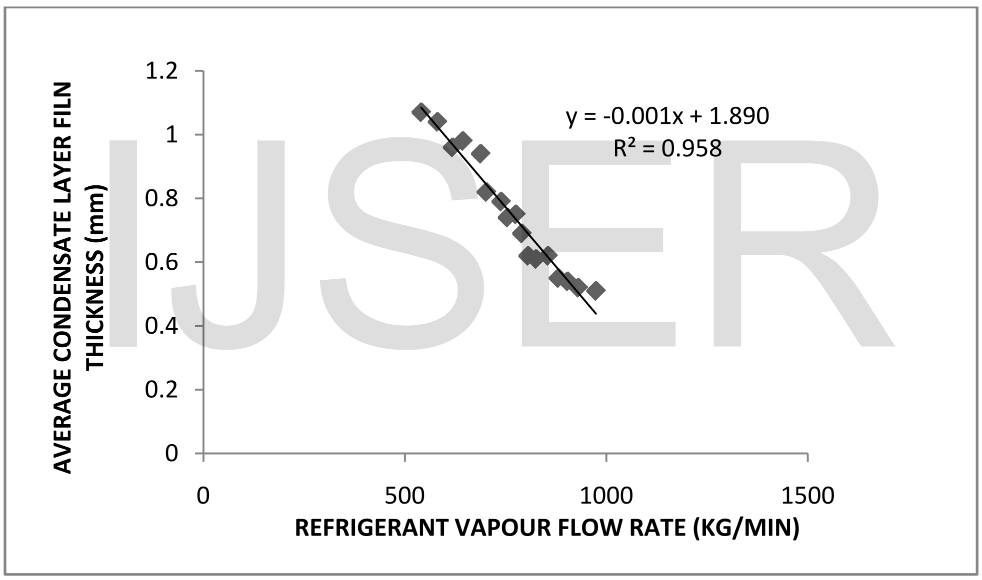

Figure 6;graph of the average condensate film thickness against refrigerant vapour flow rate

From the graphs of refrigerant condensate thickness against refrigerant vapour flow rate shown above, the data plotted agree with the predictions of the model for upper layer thickness, side layer thickness, lower layer thickness and average thickness with percentage fitting deviation of 6.50%, 4.20%, 94.60% and 4.20% respectively. The high degree of fitting deviation experienced by the lower layer film thickness is because the lower part of the tube experienced only small forced convection and much of free convection due to little contact between the lower part of condenser tube and the flowing refrigerant vapour.

4.2 Testing of condensate flow rate model

The theoretical model show a linear and direct relationship between refrigerant vapor flow rate and condensate flow rate which implies that the graphical representation of the relationship between refrigerant vapour flow rate and condensate flow rate will be a straight line graph sloping upward from left to right with positive slop and R^2 value of 1 (one). Comparing this with the graphical representation of the relationship between refrigerant vapor flow rate and condensate flow rate for an elliptical condenser using data from an LNG company in Nigeria shown below

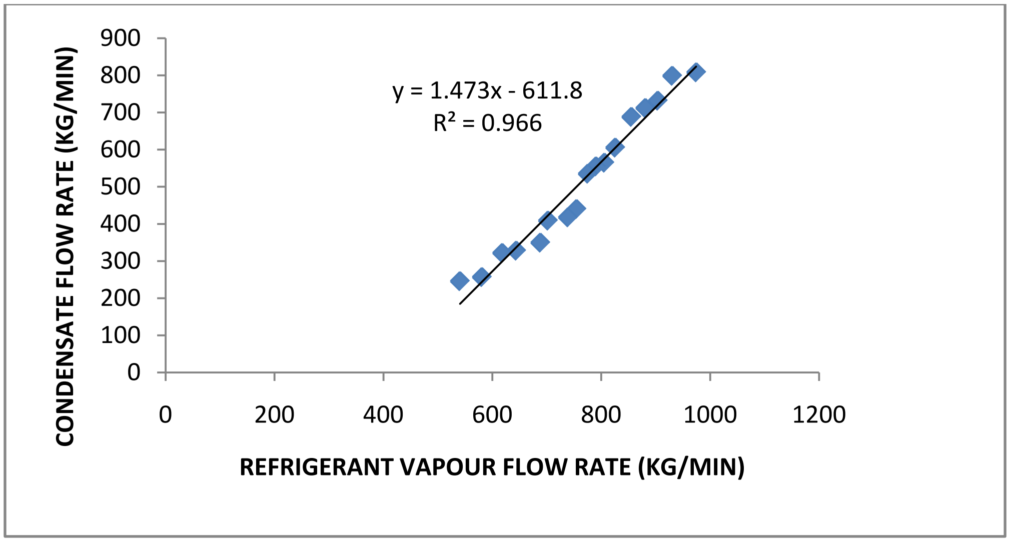

Figure7; graph of refrigerant condensate flow rate against refrigerant vapour flow rate

From the graph of refrigerant condensate flow rate against refrigerant vapour flow rate shown above, the data plotted agree with the model prediction with percentage fitting deviation of 3.40%

4.3 Testing of rate of condensation model,

There is no industrial or empirical data present to test the theoretical model relating refrigerant vapor flow rate and rate of condensation. This is because rate of condensation has little or no operational significant in the industry and is considered as a microscopic phenomenon with much concern to the scientist. Moreover, Fujii et al (1979)are of the opinion that rate of condensation like rate of boiling, cannot have an absolute value like rate of refrigerant flow and rate of condensate flow. They therefore proposed a dimensionless parameter that can quantify the rate of condensate for every condensing vapor. This dimensionless parameter represented with **G** as proposed by Fuji et al (1979)is given as

G = $\frac{K\,\Delta T}{\eta_C L_f}\sqrt{\frac{\rho_c\,\eta_C}{\rho_v \eta_v}}$ (see the nomenclature for the parameters definition)

Hence, for every condensation process, the higher the value of G, the higher the rate of condensation and the lower the value of G the lower the rate of condensation

Finally, the fitting deviations seen in the result are due to assumptions made in development of the model.

5 conclusions

A theoretical model to demonstrate the effect of refrigerant vapour flow fluctuation on the condensation process parameters was developed. The models predictions were tested using plant data collected from an LNG company operating in Nigeria.The model developed predicted direct linear relationship between refrigerant vapour flow rate and condensate flow rate, an inverse linear relationship between refrigerant vapour flow rate and rate of condensation dimensionless parameter and an inverse linear relationship between refrigerant vapour flow rate and condensate film thickness. The model was tested using a plant data collected from an LNG company operating in Nigeria and a percentage fitting deviation of 4.20% and 3.40% was observed with respect to average condensate film thickness and condensate flow rate respectively

The paper maintain that there is infarct a relationship between refrigerant vapour flow rate and condensation process parameters, and as such suggested that an optimization analysis should be carried out on condensers used in the vapour compression refrigeration cycle to ascertain the optimum refrigeration vapour flow rate for maximum vapourcondensation in the condenser unit so as to have an effective and efficient operation of a vapour compression refrigeration cycle. Moreover, an experimental model should be designed to validate and authenticate the claims of the relationships between the flow rate of refrigerant vapor and rate of condensation.

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NOMENCLATURES

A Cross sectional area (m^2)
$\boldsymbol{\tau}$ Interfacial shear stress (N/m^2)
m mass flow rate of condensate film (g/s)
M mass flow rate of refrigerant vapor (g/s)
$\mathbf{U_\varphi}$ Refrigerant vapor stream wise velocity (m/s)

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u_φ Refrigerant condensate film velocity (m/s)
ρ_c Refrigerant condensate film density (kg/m^3)
ρ_v Refrigerant vapor density (kg/m^3)
η_c Refrigerant condensate viscosity
η_v Refrigerant vapor viscosity
F(φ) Geometry relation dimensionless parameter
K Thermal conductivity of the condensate film (w/m°C)
k eccentricity of condenser shape
δ Refrigerant condensate film thickness (mm**)**
L_f Specific latent heat of fusion
G Rate of refrigerant vapor condensation dimensionless parameter
H Fluid property dimensionless parameter
T_{Sat} Saturation temperature of refrigerant vapor ℃)
T_w External wall temperature of the condenser (℃)
Δ**T** change in temperature (℃)