Film Condensation on a Vertical Tube at Different Pressures

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Abstract:-

The work presented is a numerical and experimental study film condensation of steam on a vertical tube surface at different pressures, power supplied to the evaporation tank and cooling water temperature. The designed system consists of three parts; cold water, steam supply sub-system and the test rig which consists of vertical copper tube. The calculated parameters were measured by local and average condensation heat transfer rate, local condensation heat transfer coefficient, average condensation heat transfer coefficient, tube surface temperature distribution, film thickness and steam condensation rate. It was observed that there is a gradual increase in condensation rate with increasing steam pressure in the test vessel. It was concluded that when we increase the steam pressure this will lead to an increase of the average heat flux, while local heat transfer coefficient is affected by the condensate layer thickness over tube surface. Temperature of the surface tube decrease's continuously in the length of condenser tube and the average temperature of tube is proportional with its steam pressure as maximum average temperature is reached when this P=0.45 bar. The agreement between the experimental and numerical value of the average heat transfer coefficient appears to be reliable with a deviation of about (2-7%).

Keywords: ***Film Condensation, Different pressures, Different power supplying to the Evaporation, Different Cooling Water Temperatures, Numerical and Experimental Study***.

Introduction

Liquid films of different pressures fluid flow and heat transfer is of certain concern in several mechanical and chemical engineering applications, tempering of glass, drying of textile, industrial applications, and biochemical reactions. Therefore the study of these processes and new techniques for their improvement remains an important area of process engineering.

The first paper is about laminar film condensation of the heat transfer rate on a vertical wall surface is published by Nusselt [12] in 1916, many theoretical and experimental studies have been devoted the difference between film and drop condensation of steam on a vertical tube at different pressures [2, 9], they calculated the condensation heat transfer coefficient for drop and film condensation of steam decreased with decreasing steam pressure. Also, the heat transfer rate was not affected by the small steam flow in the case of film-wise mode, while it was 3-6 times higher in the drop-wise mode also the heat transfer rate was higher in upward facing than in downward facing. [7,8] conduct experimental of steam condensation in a vertical copper tube, the heat transfer coefficient determined by resistance of thermal method and compare result with Wilson method. And his theory has been extended to steam condensation with the presence of a non-condensable gas [3, 4]. A new approach on the determination of condensation heat transfer coefficient was studied experimentally using two different models (frictional pressure drop &void fractional) in a vertical tube is studied by [1].

The purpose of the this study is to investigated experimentally and numerically the steam condensation process on vertical tube condenser at different pressures and different cooling water temperatures which is related to steam production rate. Steam condensation process was used to measure (and predict) the rate of heat transfer to liquids flowing across the surface of a heated tube and coolant fluid used for tube cooling purpose. The present work can be described as follows:-

- Preparing a mathematical model for thermal performance of different sub-systems. Also, the model studies condensation process on vertical copper tube condenser at different pressures and temperature of cooling water.
- Investigate the condensation rate along the length of tube at different pressure, cooling system temperature and water evaporation power.
- Study the relation between condensate film thickness versus position along tube at different pressures, cooling system temperature and water evaporation power.
- Examine the relation between the surface temperature versus position along tube at different pressure, cooling system temperature and water evaporation power.

II EXPERIMENTAL WORKE

A.*General description :*

The designed system is used to supply of water for use in the steam generator. The generator steam steadily at a rate which is adjusted by the power supply input of the electrical heaters. The steam flows enter the vessel through the base and flows upward over the condenser tube. The temperature of the steam is measured via T_7 and the steam pressure in the vessel by P. The cooling water flows through internal passage in the tube from source of cold water. The container can be evacuated using the water jet pump P. For this purpose water is fed to the pump via the control valve V_3. A non-return valve built into the water jet pump prevents water flowing back into the vessel. In order to prevent the escape of steam and thus the loss of water, the suction pipe is fitted with cooling system and a water separator. The water drawn off is fed back into the vessel. The vessel can be filled with distilled water and drained via the valve V_4.

B. Test section:

The apparatus used in the experimental work consists of two pipes (K1, K2) upon which the drop & film condensation can be observed on the tube surface although the present research is concerned with film condensation only. The two pipes are connected to a tank on its upper surface. Cold water flows through a pipe inside the submerged pipes. The heat supplied from the steam to the condensation tube can be determined by knowing inlet & outlet temperature of the cooling water. Also, the cooling water flow rate can be controlled by the valve (V_2). The cooled water enters the tube from the lower end of the condenser tube and rises until reaching the upper end and cooling its inner wall.

Outlet diameter of the condenser tube: 12 mm

Cooled length of the condenser tube: 96 mm

Surface Area cooled: 36.18 cm^2

C. Steam Generator:

The steam water is generated in the lower part of the vessel by the electric heater (H), and the output power of the heater is adjustable (0-3000 W). The distilled water is drawn into the container. Fill the container to the mark (1-2 cm above the heater element).

D. Cooling Water System:

The cooling water flow was taken from the laboratory's cooling water system. Cooling water flow through the inside of the tube is adjusted by control valve V_2. It is measured using a flow meter F_2. Inlet and outlet temperature was measured by utilizing two thermocouples (T_4 and T_5).

III. MATHEMATICAL MODEL

The numerical heat transfer establishment between vapor condensation and the cooling water used to cool tube that simulates the present experimental work is divided into three sub-sections which deal with the fundamentals of heat transfer, hydrodynamics of boundary layer on a vertical tube and the sensible heat transfer calculations together with predictions concerning heat transfer with phase change.

The mathematical model studies steam pressure effect, steam temperature, temperature profile of tube, condensation rate, and condensed steam boundary layer growth along the copper tube length. From the Fig. 1 it can be seen that water steam condensation on a vertical tube and film condensate thickness increases from section x to (x + dx) under the influence of gravity, while cooling water is fed to the inner diameter of the vertical tube from section (x + dx) to x. The film condensation thickness increases starting from x to (x + dx). Because of the permanent condensation at the interface of liquid-steam, it is observed that the condensate rate is increased gradually with increasing length (x + dx).

During the present investigation, the considered assumptions are listed as follow:

1. The following input parameters are based during conducting numerical approach:
   - Thermal conductivity of tube (K).
   - Dimension of physical tube.
   - Physical properties of the condensed steam.
   - Heat transfer between steam and cooled tube is found by proper formula.
   - The flow rate of cold water in the closed tube system.

2. The following parameters are found by utilizing the listed assumptions below:
   - Constant fluid properties are because of narrow temperature range along length of copper tube.
   - At condensate/vapor interface the shear stress is zero.
   - Newtonian and Incompressible liquid flow.
   - Laminar and steady state condensate film flow.
   - Smooth film surface.

The thickness of liquid film condensate at any position x for steam condensation over a vertical flat plate was derived by Nusselt [12], as follow:

$$\delta_x = \left[\frac{4K_l\mu_l(T_{steam} - T_{sur})x}{g\rho_l(\rho_l - \rho_v)h_{fg}}\right]^{1/4} \quad (1)$$

Where ρ_l, ρ_v *is* the water and vapor density (kg/m^3), g is the gravitation due to gravity (m/sec^2), K is the thermal conductivity of the water (W/m K), μ_l is the viscosity of the liquid (Ns/m^2) and h_{fg} (J/kg) the latent heat of condensation.

An improvement to the foregoing result for $\delta_x mm$ was made by Nusselt [12] and [11], who showed that, with the inclusion of thermal advection effects, a term is added to the latent heat of vaporization, in lieu of(h_{fg}), the modified latent heat was defined as [11]

$$h_{fg}{}^* = h_{fg}\left(1 + 0.68C_p(T_{steam} - T_{sur})\right) \quad (2)$$

Also, the local condensation heat flux q_x (W/m$^{2)}$ is given in terms of the temperature different between the saturated steam and surface temperatures in the local x, as follows

$$q_x = h_x(T_{steam} - T_{sur}) \quad (3)$$

The heat transfer coefficient by condensation at certain location is determined by using:

$$h_{cond\,x} = k_l \frac{(T_{steam} - T_{sur})}{\delta_x} \qquad (4)$$

By substituting δ_x in equation (4), $h_{cond\,x}$ (W/m^{2}K) becomes:-

$$h_{cond\,x} = \frac{k_l}{\delta_x}$$

$$= \left[\frac{g\rho_l(\rho_l - \rho_v)h_{fg}{k_l}^3}{4\mu_l(T_{steam} - T_{sur})x}\right]^{1/4} \qquad (5)$$

Lastly, by utilizing the definition of average condensation heat transfer coefficient over entire plate, its value is determined by substituting h_{avg} relation and integrating it along plate length:

$$h_{avg} = \frac{1}{l}\int_0^1 h_x\,dx \qquad (6)$$

$$= 0.943\left[\frac{g\rho_l(\rho_l - \rho_v)h_{fg}{k_l}^3}{4\mu_l(T_{steam} - T_{sur})l}\right]^{1/4} \qquad (7)$$

Equations (1 & 2) for vertical plates can also be utilized to determine the film thickness of condensation. In addition, the one dimensional steady state average heat transfer coefficient is taken on vertical tubes outer surface because the tube diameter is large relative to the liquid film thickness. The film condensate boundary layer originates at the top of the tube and flows downward under the effect of gravity, while the cooling water flows is fed through an immersion tube to the lower end of the condenser pipe and then rises up the inner wall Fig. 1.

Numerical solutions were obtained by using Mat Lab (version 7.10) method. the inlet cooling water, steam temperature and cooling water rate are used as an input for the program, the calculates surface and cooling water temperature distribution, local heat transfer rate , local heat transfer coefficient and film thickness.

The liquid film condensate thickness of tube includes two unknown parameters, film thickness and surface temperature. This requires making necessary estimation of tube surface temperature $(T_{sur\,x})$ est. The transfer condensation heat (Q_x) is measured by two methods, first method by conduction from the liquid film surface at temperature of (T_{steam}) to calculate the outside tube surface temperature of $(T_{sur\,x})$ est. and the second method, by conduction and convection from an evaluated the outside tube surface at temperature of $(T_{sur\,x})$ est. to the cooling water entering the element with a temperature of $(T_{cool\,x})$, equal two values of heat transfer rate, then the calculated value of the tube wall surface temperature is correct . Otherwise, iteration process made by correcting the $(T_{surx})_{est}$ [6].

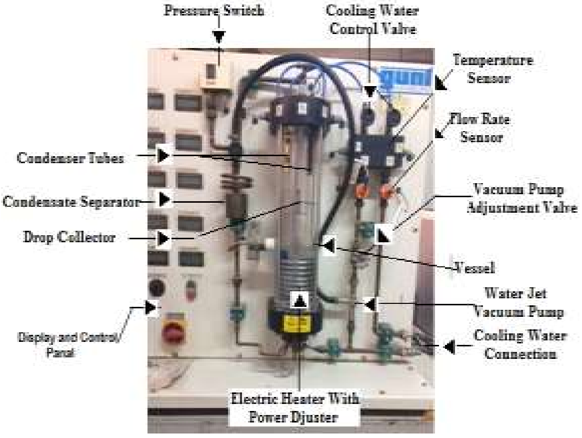


Plate1. Film Condensate Device

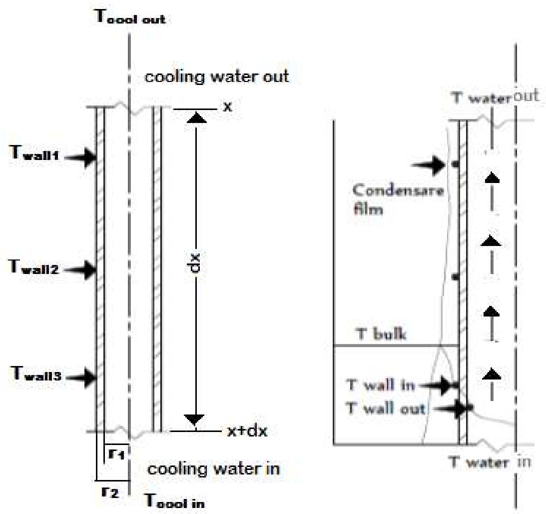


Fig.1. Scheme Diagram of Condenser Tube

the operation steady laminar, the condensation heat transfer rate to a test section is equal to the condensation heat transfer rate from the test section, therefor, transferred condensation heat from film condensation at a T_{steam} (K) by way of elemental length tube dx (mm) with outside surface temperature $T_{sur.}$ (K) is equal to the heat transfer rate from the outside surface temperature $T_{sur.}$ (K) the same element to the cooling water temperature $T_{cool.}$ (K).

$$Q_x = \frac{T_{steam} - T_{surx}}{R_f} = \frac{T_{steam} - T_{sur\,x}}{R_{cond} + R_{conv}} \quad (8)$$

$$R_{cond.} = \frac{ln\left({}^{r_2}/{}_{r_1}\right)}{2\pi k_l dx}$$

$$R_{conv.} = \frac{1}{h_{in} A_{x\,in}} \quad (9)$$

The average Nusselt number N_{um} in equation (11) for cooling water flow along the tube is determined from equation (10) according to Gröber [5]. Also, the heat transfer coefficient of condensation h (W/m^{2}k) of the cooling water on inner surface of tube can be obtained from:

$$N_u = \frac{h_l d_h}{k_l} = 1.86\left[R_e P_r \frac{d_h}{l}\right]_l^{0.33} \left[\frac{\mu}{\mu_{wall}}\right]_l^{0.14} \quad (10)$$

$$h_l = \frac{N_{um} k_l}{d_h} \quad (11)$$

Both the velocity and temperature profile are 'developing' for this case (combined entry lengths) we can use the Sieder and Tale [10] to equation (10) to determine the Nusselt number.

Equation (10) is used with the following conditions:

$$\begin{bmatrix} 0.48 \le p_r \le 16.700 \\ 0.004 \le \left(\frac{\mu}{\mu_l}\right) \le 9.750 \end{bmatrix}$$

The condensation heat transfer coefficient at xth local position can be calculated as

$$h_x = \frac{Q_x}{(T_{steam} - T_{sur\,x})(2\pi d_x r_2)} \qquad (12)$$

While the average condensation heat transfer coefficient $h_{cond.}$ (W/m^{2}k) can be calculated as

$$h_{cond\,av.} = \frac{Q}{(T_{steam} - T_{sur})(2\pi L r_2)} \qquad (13)$$

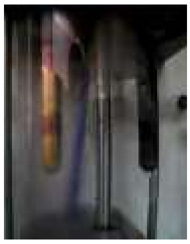
IV. RESULTS AND DISCUSSION:

The experiment and numerical work concluded that above calculated parameters affect the condensation rate as follows:-

1. Steam pressure, P rang (0.15, 0.2, 0.25, 0.35, 0.45 bar).
2. The power consumed by the evaporation unit ranges (900, 1200 and 1500Watt).
3. Cooling water temperature $T_{cool.}$ rang (17, 18 and 19°C).

Plate 2, shows the effect of steam pressure on condensation rate using electronic camera type (high speed digital camera, Samsung WB 2000) and this effect is illustrated graphically in Fig. 2 estimate the cumulated film condensate on the copper surface tube at constant power supply to the evaporation tank $P_{steam} = 1500\ Watt$ at different steam pressure and cooling water temperatures, the heat transfer is proportional to the steam pressure and inversely proportional to the cooling water temperature. It concluded that the values of steam pressure increase to be observed the increasing heat flux across the tube, while local heat transfer is affected by condensate film thickness over the tube surface. Any increase in condensate flow rate across section will increase film thickness at all length tube. As greater clarified before higher condensate rate when $T_{cool.} = 17$°C but condensate rate when $T_{cool.} = 18$°C and $T_{cool.} = 19$°C is lower. Fig. 3 and 4 the similar behaviors Fig. 2 with different evaporation power P_{steam} (1200 and 900 Watt) it could be concluded also that condensate rate decreases with decreasing power supply, by gradually decreasing the steam pressure.

Steam pressure = 0.45 bar



Steam pressure = 0.15 bar

Plate.2 Comparison of Condensation Steam of Surface at Different Pressures (0.15 and 0.45 bar), Constant Power Supply P_{steam}= 1500 Watt and Cooling Water Temperature $T_{cool.}$ = 17°C

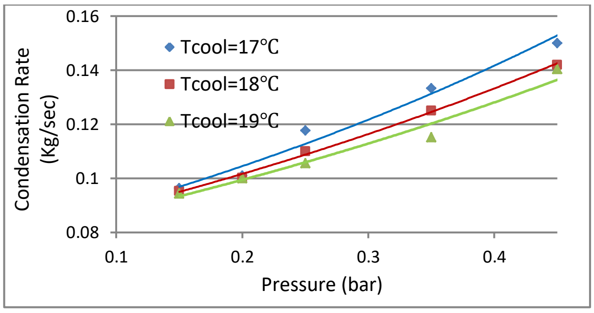


Fig. 2 Variation condensation rate with steam pressure for different cooling water at a constant evaporation power P_{steam}=1500 Watt

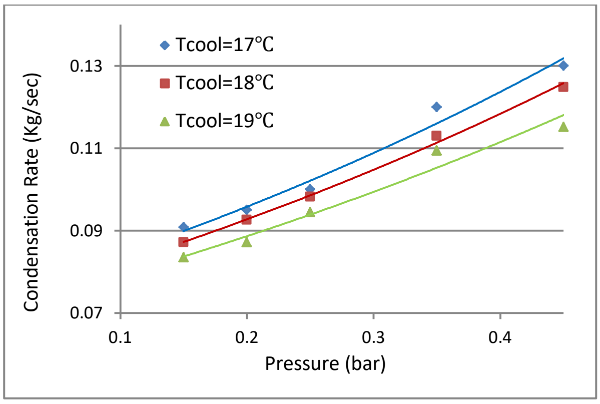


Fig.3. Variation condensation rate with steam pressure for different cooling water at a constant evaporation power P_{steam}=1200 Watt

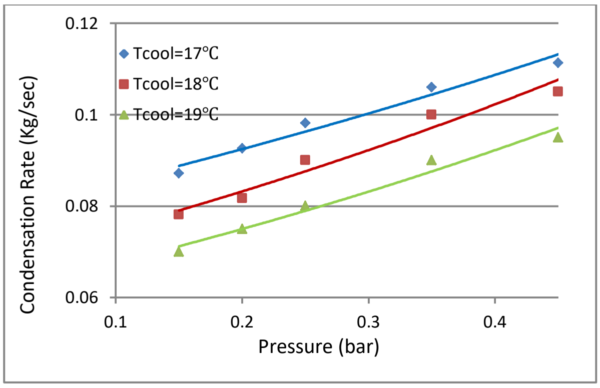


Fig.4. Variation condensation rate with steam pressure for different cooling water at a constant evaporation power P_{steam}=900 Watt

Fig. 5, describer the effect of different steam pressures on film thickness versus axial tube length at constant power supply of the evaporation and cooling water temperature. It shows increasing the film condensation with the axial tube length. Also, the film thickness has been effected proportional by the steam pressure during the test section. The cooling water temperature distribution curves along the tube at different values steam pressures are shown in Fig. 6, it is observed that for gradual decrease in cooling water temperature leads increase in film thickness versus long tube. While Fig. 7 related to power supply of the evaporation in tank (900, 1200 and 1500 Watt) but constant cooling water temperature $T_{cool.}$ = 17℃ and pressure supply in vessel P=0.35 bar. It can be seen that there is clear difference between the film thicknesses for three power supply. As might have been anticipated, the film thickness is highest for the P_{steam}=1500 Watt, slightly reduced for the P_{steam}=1200 Watt and very much reduced for the P_{steam} Watt.

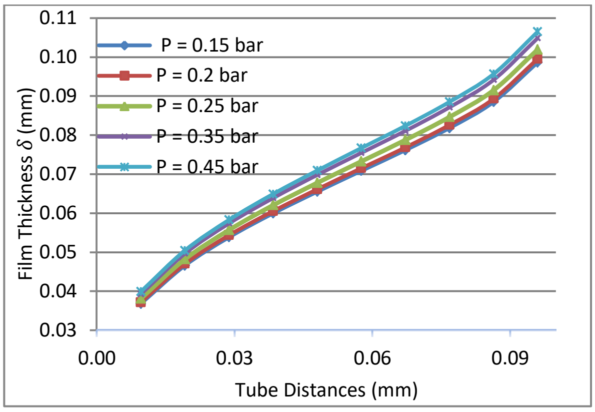


Fig.5. Film thickness distribution along the Tube of different Pressure at constant cooling water temperature $T_{cool.}$= 17℃ and evaporation power P_{steam}= 1500 W

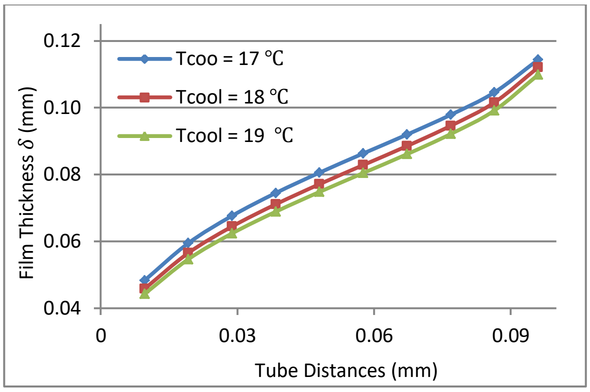


Fig.6. Film thickness distribution along the tube of different cooling water temperature at constant evaporation power P_{steam}= 1500Watt

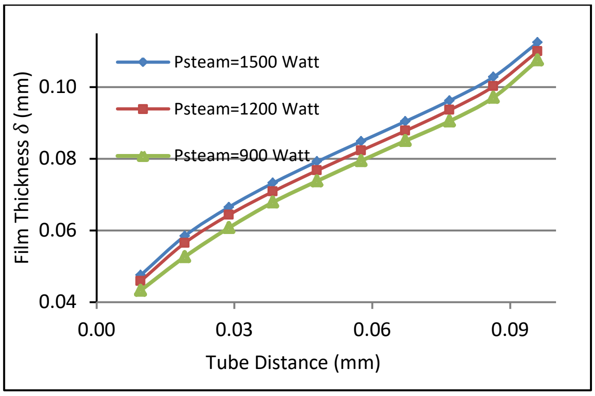


Fig.7. Film thickness distribution along the tube of different evaporation power at constant cooling water temperature $T_{cool.}$= 17 ℃ and P = 0.35 bar

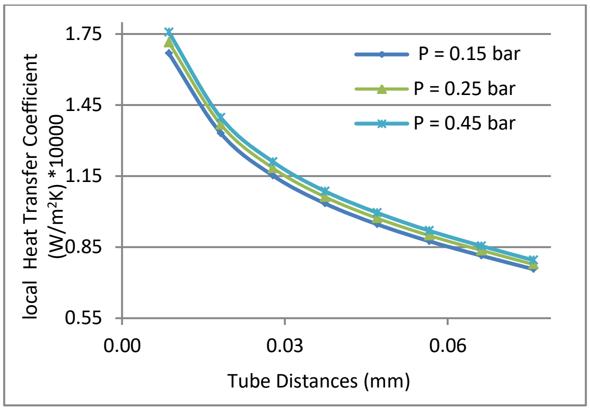


Fig.8. Local heat transfer coefficient variation with tube distances at cooling water Temperature $T_{cool.}$ = 17℃ and evaporation power P_{steam} = 1500 Watt

Fig. 8, illustrates the effect of different values of pressure on local heat transfer coefficient (h_x) at constant evaporation power P_{steam}= 1500 Watt and cooling water temperature $T_{cool.}$ = 17°C. It was noticed that the local heat transfer coefficient decreases according to its position long the length, due to the effect caused by the thinning of the film condensate. From the same figure it was clear that more heat transfer as the pronounced effect pressure increases. This increase was not caused by an increase in $T_{sur.}$ but in this case the steam velocity actually increased as the pressure increased. Fig. 9 describe the distribution curve of the local heat transfer coefficient along the length of the tube at different cooling water temperature and constant steam pressure P = 0.35 bar, from which, it is noticeable that decreasing cooling water temperature decreases heat transfer coefficient along the whole tube, which may explained by the condensate film thickness increased gradually and increased rapidly with decreasing cooling water temperature ($T_{cool.}$).

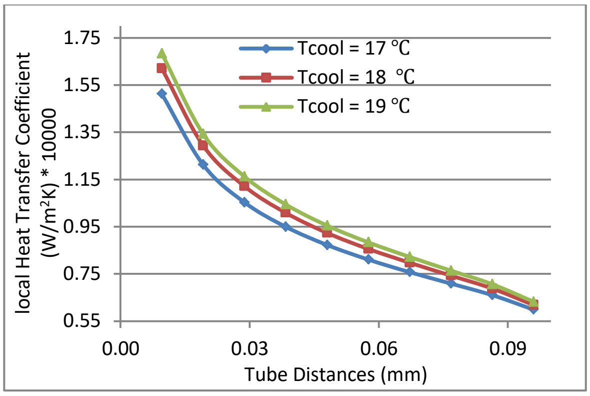


Fig.9. Cooling water temperature effect on the local condensation heat transfer coefficient at constant evaporation power P_{steam} = 1500 Watt and steam pressure P = 0.35 bar

As can be seen from Fig. 10 the pattern of behavior with steam supply is quite different. There is a systematic reduction in condensation heat transfer coefficient rate over the length of tube surface, while the rate of condensation heat transfer coefficient are clearly higher with higher power supply. However, it was noticed that the experimental value of average condensation heat transfer coefficient rate is lower than the numerical value. This difference is predicted to be (2-6.8%) below the experimental value average heat transfer coefficient rate depending on steam pressure, cooling water temperature and steam power supply, as shown in Fig. 11 The good agreement between experimental and numerical and this deflection are justified by the accuracy of experimental device, according to the assumption used in the theoretical approach. Fig. 12, local heat transfer rate of the condensing film is presented for tube length. These data are measured for a power supply P_{steam}=1500 Watt, water cooling temperature $T_{cool.}$ = 17°C and different steam pressures. For the given operational conditions, it can be seem that higher steam pressure gives a higher rate of condensation and also the heat flux at along tube in the test facility. It was found that a heat transfer are slightly too at the beginning of the tube and too high at its end. This might be caused by water cooling temperature have a maximum at its entry to the condenser tube at the depth. Fig. 13 are plotted for same boundary conditions of previous of Fig. 12 except that water cooling temperatures are different $T_{cool.}$ (17, 18 and 19°C) at constant steam pressure P = 0.35 bar. An interesting feature of the results obtained is that with steam condensation on a vertical tube the heat transfer rate increases very rapidly as water cooling temperature is decreased. Fig. 14 describes the experimental results related to varying steam power supply on rate of heat transfer at constant steam pressure P_{steam} = 1500 Watt and water cooling temperature $T_{cool.}$ = 17°C. It is noticeable that the heat transfer rate increases with increasing evaporation power supply because of the increase of steam temperature.

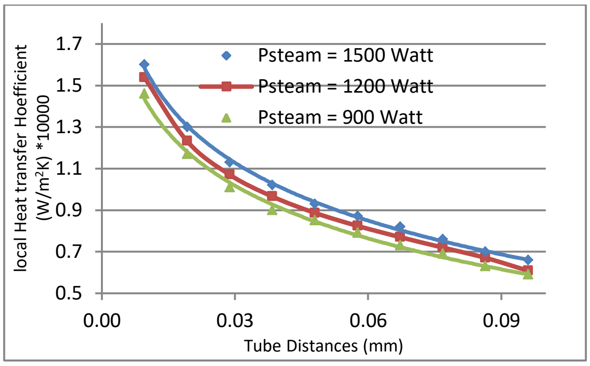


Fig.10. Evaporation power effect on the local heat transfer coefficient at constant cooling water temperature $T_{cool.}$= 17 °C and pressure P = 0.35 bar

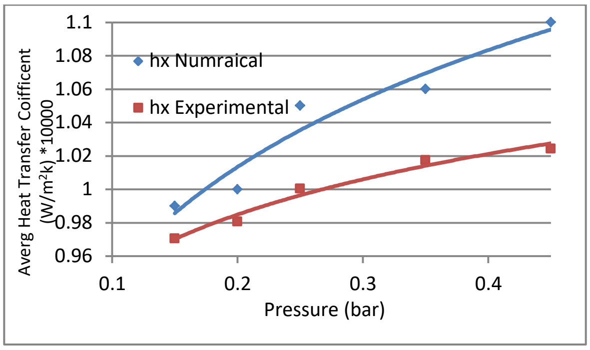


Fig.11. Comparison of have versus pressure in test section at cooling water temperature $T_{cool.}$= 17°C and evaporation power P_{steam}= 1500 Watt

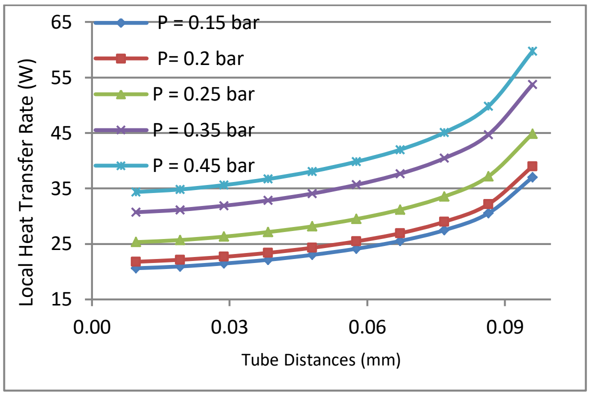


Fig.12. Local heat transfer rate of condensing film versus axial tube length at cooling water temperature $T_{cool.}$=17 °C and evaporation power P_{steam} = 1500 Watt

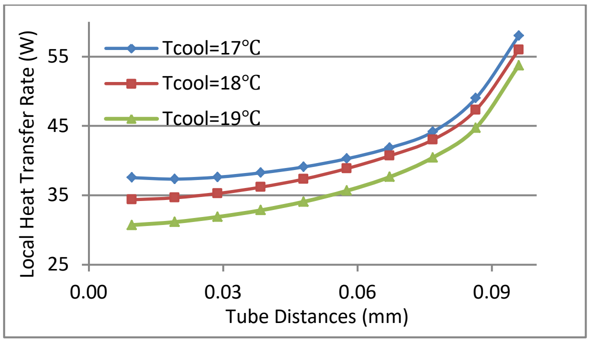


Fig.13.Variation of local heat transfer rate with axial length tube for different cooling water temperature at constant evaporation power P_{steam}=1500 Watt and pressure P = 0.35 bar

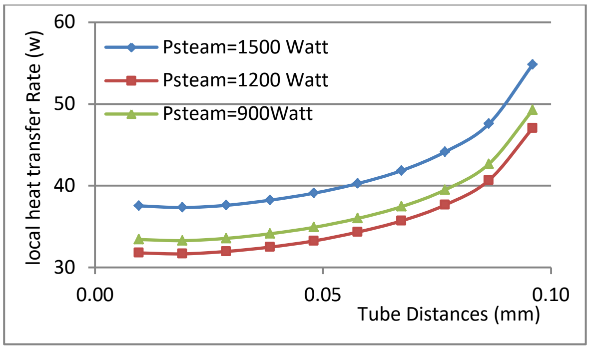


Fig.14. Evaporation power supply effect on the local heat transfer rate at constant cooling water temperature $T_{cool.}$= 17 ℃ and pressure P = 0.35 bar

Cooling water temperature distribution an along axil tube in a vessel at different pressure steam at constant evaporation power supply P_{steam}= 1500 W and cooling water temperature $T_{cool.}$= 17 °C are shown in Fig. 15 from which, it is clear that higher pressure steam in vessel gives a higher cooling water temperature along axil condenser tube, this result may be refer to the fact that cooling water is fed through an immersion tube and then rises up the inner wall, put steam flow inter the vessel in upward. Experimental to study the effect increasing pressure steam in condenser tube were performed next. It can be seen at from Fig. 16 that the reduction of surface temperature rate along the tube at constant vapor temperature. Steam temperature in upward condenser tube is high compared to the temperature of the tube surface. In addition, when the difference temperate is high results a linear decreasing in the surface temperature. Fig. 17, Shows that the temperatures measured values decrease continuously in the axil length tube at constant pressure steam P = 0.35 bar and cooling water temperature $T_{cool.}$= 17 ℃ at different evaporation power ranging. This effect could be observed as surface temperature increase proportionally with evaporation power supply, which results from a higher tube surface temperature at high steam temperature.

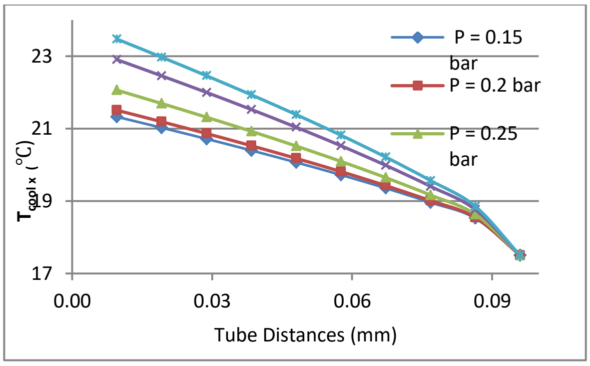


Fig.15. Insignificant impact cooling water temperature on axial length tube at different pressure steam and constant power supply P_{steam}= 1500 Watt

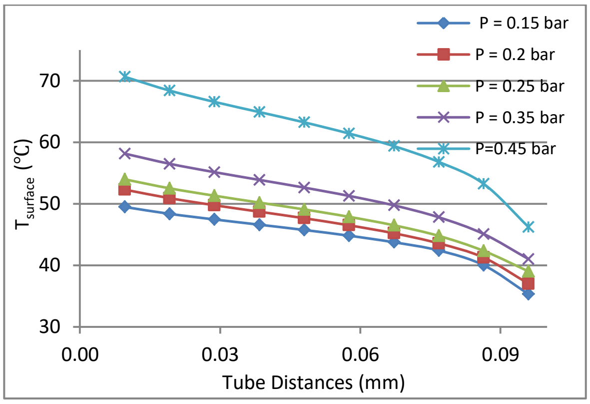


Fig.16. Tube surface temperature variation versus axial length tube at a cooling water temperature $T_{cool.}$=17 ℃ and power supply P_{steam}=1500 W

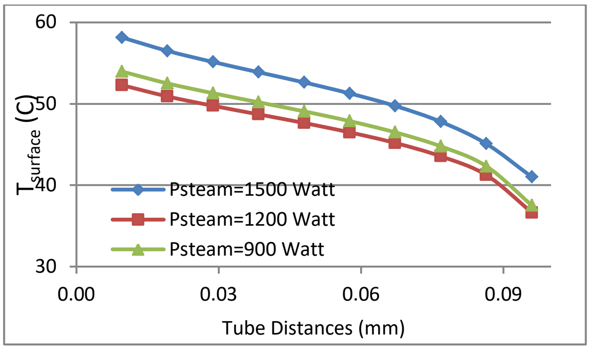


Fig.17. Tube surface temperature variation versus axial length tube at a constant cooling water temperature $T_{cool.}$=17 ℃ and pressure P = 0.35 bar

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تكثيف بخار الماء الغشائي على سطح أنبوب عمودي بضغوط مختلفة

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الخلاصة:-

يقدم البحث الحالي دراسة عملية ونظرية لتكثيف بخار الماء الغشائي على سطح أنبوب عمودي نحاسي وما تأثير أختلاف الضغط , كمية مختلفة لبخارمتدفق الى خزان الأختبار وكذلك أختلاف درجات الحراة الماء البارد على سطح أنبوب. الجهاز يتكون من ثلاث اجزاء, منظومة الماء البارد والجزء الثاني منظومة تولد البخار أما الجزء الثالث منظومة الاختبار تتكون من انبوب نحاسي عمودي .تم بناء برنامج لحساب معدل أنتقال الحرارة الموضعي والكلي, معامل أنتقال الحراة الموضعي بالتكثيف ومعامل أنتقال الحرارة المتوسط بالتكثيف, توزيع درجات حرارة على سطح الأنبوب,توزيع سمك طبقة السائل المتكثف المتاخمة وكمية البخار المتكثف. ولقد وجد أن زيادة الضغط يؤدي الى زيادة معدل البخار المتكثف ومعدل أنتقال الحراة بينما معامل أنتقال الحرارة يتأثر بسمك الطبقة السائل المتاخمة على سطح الأنبوب. درجة حرارة سطح الأنبوب تنخفض على طول أنبوب التكثيف ومتوسط درجة الحرارة يكون أعلى قيمة عند الضغط P=0.45 bar . أظهرت الحسابات النظرية توافق الى حد كبير ومعقول مع الجزء العملي عند حساب معدل معامل أنتقال الحرارة ونسبة الخطأ كانت تتراوح بين (%2-7).

الكلامات المفتاحية:-

تكثيف بخار الماء الغشائي, أختلاف الضغط, أختلاف معدل تدفق البخار, أختلاف درجات حرارة الماء البارد, دراسة عملية ونظرية.