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Study of Heat Transfer and Pressure Drop for a Condenser Tube with Inserted Rings.

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Study of heat transfer and pressure drop for a condenser tube with inserted rings

دراسة إنتقال وفقد الضغط لماسورة مكثف باستخدام حلقات مدمجة

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خلاصة البحث :

فى هذا البحث تم عمل دراسة نظرية و معملية لإنتقال الحرارة بالحمل الجبرى و فقد الضغط للمياه التى تمر داخل ماسورة مكثف . يسخن السطح الخارجى للماسورة بواسطة بخار مشبع عند الضغط الجوى. إعتمدت الدراسة على تاثير وجود حلقات بأقطار مختلفة موضوعه داخل الماسورة و على مسافات نسبية (بينية) مختلفة.أجريت الدراسة النظرية باستخدام احد البرامج المدمجة (CFD) وهو ال Fluent . وقد تم حل المعادلات الحاكمة للسريان بالبرنامج وهى معادلة السريان و كمية الحركة والطاقة مع وضع الشروط الحدية. بعد حل هذه المعادلات أمكن الحصول علي توزيع كل من معامل إنتقال الحرارة والضغط على طول ماسورة المكثف وبالتالى تم حساب متوسط معامل إنتقال الحرارة بالحمل الجبرى وفقد الضغط للمياه المارة داخل الماسورة. حساب معامل إنتقال الحرارة والفقد فى الضغط للماسورة بدون حلقات بداخلها.و كذلك فى حالة وجود مجموعة الحلقات داخل الماسورة بأقطار مختلفة و على مسافات نسبية مختلفة . ولاتمام الجزء المعملى تم تصميم وتنفيذ دائرة إختبار تتكون من ماسورة نحاسية يمر بداخلها مياه تبريد ويسخن السطح الخارجى للماسورة باستخدام بخار مشبع عند الضغط الجوى . وضعت حلقات على أماكن مختلفة من طول الماسورة مع تغير أقطار الحلقات.وقد تم تزويد دائرة الإختبار بالحساسات اللازمة لقياس كل من درجات الحرارة و سرعة المياه وفرق الضغط .وقد أظهرت النتائج النظرية و المعملية فى كلا الحالات انه عند استخدام حلقات داخل ماسورة المكثف ادى الى تحسين فى معامل إنتقال الحرا رة بالحمل بنسبة ٦٦%.و قد لوحظ توافق بين النتائج النظرية و المعمليةالتى تم الحصول عليها.

Abstract

Forced convection heat transfer and pressure drop for cooling water which pass into condenser tubes is investigated numerically and experimentally, The study is dependent on the effect of the rings with different inner diameter placed inside the tube at different distances. The outer surface of the tested condenser tube is heated by saturated steam at atmospheric pressure. By using one integrated a computational fluid dynamic software (CFD) is the fluent theoretically investigated. The software is used to solve the governing equations for flow; which are continuity equation, momentum equation and energy equation. the distribution of the heat transfer coefficient and the pressure drop along tube length, can be obtained and consequently the average heat transfer coefficient. The theoretical study was performed for the cases of a tube without rings and with rings. To perform the experimental study, an experimental apparatus is designed and constructed .It consists of a copper tube ,where the cooling water passes inside it and the outer surface of the tube is heated by saturated steam at atmospheric pressure. The apparatus is equipped with the required sensors to measure the tube outer surface temperature. flow rate and pressure drop. The experimental and numerical results showed that the use of rings increasing the heat transfer coefficient rather than that of tube without rings and the rings of inner diameter 8mm increase average heat transfer coefficient up to 66%.There is an acceptable agreement between the numerical and experimental results.

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Keywords: Surface condenser, heat transfer enhancement, and forced convection.

1.INTRODUCTION

Surface condenser is the commonly used term for a water-cooled shell and tube heat exchanger installed on the exhaust steam from a steam turbine in thermal power stations. These condensers are heat exchangers which convert steam from its gaseous to its liquid state at a pressure below atmospheric pressure. Where cooling water is in short supply, an air-cooled condenser is often used. An air-cooled condenser is however significantly more expensive and cannot achieve as low a steam turbine exhaust pressure as water cooled surface condenser. Many researches has been directed to study the enhancement of heat transfer coefficient over the power plant condenser tubes. Chuang and Sue [1], studied performance effects of combined cycle power plant (CCPP) with variable condenser pressure and loading. They found that the CCPP can produce more power output when operating at a lower ambient temperature (or lower condenser pressure),the net power output and heat rate are related to the ambient temperature and condenser pressure, which is controlled, by a number of fans, and plant efficiency is decreased significantly when load is operated below 35% load A lot of studies on full length twisted tape [2-6] have been carried out , and it is found that the use of twisted tapes improve the thermal performance associated with increasing the pressure drop. Leonard et. al [7],investigated experimentally the effect of internal fins with a star-shape cross section on heat transfer and pressure drop in a counter flow heat exchanger using air as a working fluid. They found that heat transfer is increased by about 12-51% over plain tube value. Promvonge and Eiamsa –ard [8], investigated experimentally the influence of twisted tape inserts on heat transfer, Nusselt number, and friction factor in a double pipe heat exchanger. They found that the maximum Nusselt number for using these enhancement devices is 188% higher than that of plain tube. Nothing et al [9] studied the effect of V-shape nozzle turbulators on the heat transfer and friction factor in a circular tube. They found that the nozzle turbulators have significant effect on the heat transfer. Promvonge and Eiamsa – ard [10], investigated experimentally the heat transfer, and friction factor in a circular rings turbulators and twisted tape swirl generator. They found that the average heat transfer rates when using both the conical rings and twisted tape of different twist ratio of 3.75 to 7.5 are found to be 367% and 350%, respectively over the plain tube. Piasarn [11], presented the heat transfer characteristics and pressure drop in the corrugated channel under constant heat flux. He found that the average Nusselt number increased with both Reynolds number ,heat flux and wavy angle. Mehmet et al [12], studied the effect of geometry of the deflecting element in the radial guide vane swirl generator on the heat transfer and fluid friction in decaying swirl flow which has three different configurations: swirl generator with conical deflecting element, with spherical deflecting element and with no deflecting element. They found that the average Nusselt number is increased up to 99%, 119%, and 148%, for the first, second, and third type swirl generator, respectively. Smith et al [13], reported through an experimental work the pressure drop and heat transfer characteristics of flow through circular tube with twisted tape inserts at free spacing of s=2p, 3p and 4p,respectively and Reynolds number varied from 2300 to 7500. They concluded that the free spacing twisted tapes with s=2p gave a heat transfer lower than full length twisted tape around 5-15% while it can be decreases the pressure drop around 90%. Ebru et al [14] studied experimentally the effect of swirl motion on the hot air by placing swirl elements of various diameters and the number of circular holes in straight or zigzag line rows on the entrance region of an inner pipe of a concentric heat exchanger. They found that the heat transfer rate increases with decreasing diameter and number of holes of the swirl element and the maxim enhancement was about 130% in a counter current flow with swirl element having smaller diameter five holes in a narrow zigzag line . Hong et al [15] , studied the heat transfer of a converging –diverging tube with evenly spaced twisted –tapes and found that the twisted- tapes with twist ratio of 4.72and the rotation angle 180°has the best performance . Smith et al [16], investigated experimentally the heat transfer and friction factor employing louvered strips inserted in a concentric tube. Their results confirmed that the increase in average Nusselt number and friction loss for the inclined forward louvered strip were 284% and 413% while those for the back word louvered strip were 263% and 233% over the plain tube, respectively .The aim of the present study is to investigate the forced convection heat transfer and pressure drop for cooling water which pass into condenser tube. inner diameter for rings, heat source, water which pass into condenser tube equations are presented, The outer surface of the tested condenser tube is heated by saturated steam at atmospheric pressure . .The effect of using rings with different inner diameter placed inside the tube at different distance. The experimental included the effect of spacing ratio

2.THEORETICAL MODEL

For steady state, turbulent flow, and two-dimensional, the governing

Continuity equation

$$\frac{1}{r}\frac{\partial(\rho v_z)}{\partial z}+\frac{\partial(\rho v_r)}{\partial r}=0 \tag{1}$$

Momentum equations

z- direction

$$\frac{1}{r}\frac{\partial(\rho v_z^2)}{\partial z}+\frac{\partial(\rho v_r v_z)}{\partial r}=-\frac{\partial p}{\partial z}+\frac{\partial}{r\partial z}\left(\mu_e\frac{\partial v_z}{\partial z}\right)+\frac{\partial}{\partial r}\left(\mu_e\frac{\partial v_z}{\partial r}\right)-\frac{2}{3}\frac{\partial}{r\partial z}(\rho k) \tag{2}$$

R- direction

$$\frac{1}{r}\frac{\partial(\rho v_z v_r)}{\partial z}+\frac{\partial(\rho v_r^2)}{\partial r}=-\frac{\partial p}{\partial r}+\frac{\partial}{r\partial z}\left(\mu_e\frac{\partial v_r}{\partial z}\right)+\frac{\partial}{\partial r}\left(\mu_e\frac{\partial v_r}{\partial r}\right)-\frac{2}{3}\frac{\partial}{\partial r}(\rho k) \tag{3}$$

Energy equation

$$\frac{1}{r}\frac{\partial(\rho v_z T)}{\partial z}+\frac{\partial(\rho v_r T)}{\partial r}=\frac{1}{r}\frac{\partial}{\partial z}\left(\Gamma_e\frac{\partial T}{\partial z}\right)+\frac{\partial}{\partial r}\left(\Gamma_e\frac{\partial T}{\partial r}\right)+\Phi \tag{4}$$

Turbulence equations

K-equation

$$\frac{\partial(\rho v_z k)}{r\partial z}+\frac{\partial(\rho v_r k)}{\partial r}=\frac{\partial}{r\partial z}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{r\partial z}\right]+\frac{\partial}{\partial r}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial r}\right]-g\frac{\mu_t}{Pr_t}\frac{\partial\rho}{\partial r}$$

$$+\mu_t\left\{2\left[\left(\frac{\partial v_z}{r\partial z}\right)^2+\left(\frac{\partial v_r}{\partial r}\right)^2\right]+\left(\frac{\partial v_z}{\partial r}+\frac{\partial vr}{\partial z}\right)^2\right\}-\rho\varepsilon \tag{5}$$

ε -equation

$$\frac{\partial(\rho v_z \varepsilon)}{r\partial z}+\frac{\partial(\rho v_r \varepsilon)}{\partial r}=\frac{\partial}{r\partial z}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{r\partial z}\right]+\frac{\partial}{\partial r}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial r}\right]-c_{1\varepsilon}g\frac{\varepsilon}{k}\frac{\mu_t}{\Pr_t}\frac{\partial\rho}{\partial r}$$

$$+c_{1\varepsilon}\frac{\varepsilon}{k}\mu_t\left\{2\left[\left(\frac{\partial v_z}{r\partial z}\right)^2+\left(\frac{\partial v_r}{\partial r}\right)^2\right]+\left(\frac{\partial v_z}{\partial r}+\frac{\partial vr}{r\partial z}\right)^2\right\}-c_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{6}$$

Turbulence model

The standard $k-\varepsilon$ model is employed and the main equations for both k and ε are:

$$\frac{\partial(\rho u_j k)}{\partial x_j}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\varepsilon \tag{7}$$

$$\frac{\partial(\rho u_j \varepsilon)}{\partial x_j}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{8}$$

Where: σ_k and σ_ε are the turbulence prandtl numbers for k and ε respectively

Table (1) the turbulent model constants

$C_{1\varepsilon}$	$C_{2\varepsilon}$	σ_k	σ_ε	P_{rt}	C_μ
1.44	1.92	1.3	0.9	1	0.09

Boundary conditions as shown in Fig.(1) which are specified as: At inle $T=T_i, v=v_i$ At the tube wall: $T=T_w, v=0$ At exit: $T=T_o, v=v_o$

The problem domain discretization is achieved by dividing the domain boundaries into a finite number of nodes and applying a mesh generation code to computations with minimum error On the other hand, cells of the small properties variation regions let to be large to minimize computations which saving time and memory consumption Number of nodes in mesh generated differs from case. In general nodes number are ranged from 10000to 14548 nodes according to the tube size. In this study, the geometric of condenser tube is investigated for both of the proposed two cases. This study was conducted theoretically, by using Gambit software for drawing the computational domains of the studied cases. Then the governing equations for the problem were solved using Fluent software. The following values which are applicable to create a finite number of cells .The cells number and size are depending on the number of nodes of domain boundaries and mesh type, structured or unstructured .In the problem under investigation, the suitable mesh is unstructured type with a triangle cell shape. Cells sizes around the tubes, where the properties variation is sharp, are small enough to ensure the accurate condenser tube are used as input data in solving the studied problem:

1-Flow is steady

2-Water velocity, v=1m/s

3-Inlet water temperature=296K

4-Wall temperature=373K

5-Tube length =6m

6-Tube inner diameter =27mm

7-Tube thickness=1.5mm

The present study is directed to enhance the heat transfer coefficient for condenser tube by inserting rings into the condenser tube. This can be achieved by inserting rings into the condenser tube from S/D=10 to S/D=40

3.Experimental test rig

The open loop system, in the figure (2) demonstrates the flow of cooling water through a water pump, that discharges inside a copper tube(4) its outer surface used as condensate , cooling water discharges into a calibrated tank, the pressure drop along the tube is measured by using a differential tube manometer (6), there is a control valve inserted on the pump outlet to control the cooling water flow rate .The tested tube (4) is made of copper, 6m long 27mm inside diameter, and 1.5mm thickness. Steam at atmospheric pressure is used as heat source around the condenser tube . An electric boiler (16) of 18 kW power is used to generate the required amount of steam, the inlet and outlet temperature of the cooling water are measured by using two thermocouples type (T) inserted in both tube inlet and outlet of the cooling water flow. The temperature along the condensed tube surface was measured by using (twelve) thermo couples type (T) equally distributed and inserted on the tube outer surface , connected to a temperature recorder (13) Pressure drop across the test tube is measured by using glycerin U- tube manometer (S G 1.276).Cooling water flow rate is measured simply by means of calibrated tank, and consequently the flow velocity of cooling water inside the test tube, is determined, the flow rate of cooling water can be controlled by using the gate valve (12) at the pump discharge side. The steam through pass from the boiler to an insulated box. The box have holes from its top surface and are used as header for passage of steam. The box and header are insulated with 50mm glass wool (14) to decrease the heat loss. The rings (10) are fixed at the specified positions (S/D=10, 20, 30, 40) . A temperature record (13) is used to measure the temperature of all thermocouple used with an accuracy of (0.1) C.

4.Measurements Technique

During the measurements, nearly it was needed to read the steady state condition, which was recorded as the temperature reading change within 0.1^oC with time. All thermocouples are connected to 14 point temperature recorder having full scale of 200^0 C, which has an accuracy of $\pm$ 1% and a resolution of 0.1^0 C. After constructing the experimental test rig as mentioned in Fig (2), and checking the measuring devices, the boiler is switched on and begins to be loaded. The system is allowed to operate for approximately half hour to reach the steady state condition. The steady state is achieved when the variation in the recorded temperature is about $\pm 0.1C$. After the system reaches the steady state condition, , the following measurements are recorded: pressure drop in the test section The inlet and outlet water flow temperature . Temperatures at different locations along the condensing tube surface.

5.Data Reduction

The average convection heat transfer coefficient and pressure drop across the condenser tube can be calculated as follows .

1.Mass flow rate:

The mean water velocity can be calculated by using the following formula

$$u = \frac{V_o}{A_c} \quad \text{m/s} \qquad (9)$$

Where,

more than the effect of spacing ratio. Figure16 shows the average heat transfer coefficient of the rings with different diameters to space ratio.Many test runs are performed at different space ratio of rings(S/D): 10, 20, 30, and 40 for the rings group with diameters (8, 12, 16, and 20) respectively, and water velocity is constant form water source, during the test. -The results of experimental results with using rings at different spacing ratio are shown in figures (17, 19).It was found that the maximum value of the average heat transfer coefficient and pressure drop occurs at spacing ratio (S/D=30), as informed from the theoretical program. From figures (17, 18) it is clear that the higher of distribution for heat transfer coefficient on the tube length with rings inner diameter 8 mm, this is occurs since the water velocity is increased, which makes the flow is turbulence and therefore an increase in heat transfer coefficient.

6.3 Comparison between the theoretical and experimental results:

There are acceptable compare is on between the theoretical and experimental results as shown in figures (17, 18) it was found from analysis of the results that there is coincide between the theoretical and experimental results to a large limit.

7.1 Conclusions

The heat transfer augmentation in a condenser tube with rings inserts at different position has been studied theoretically and experimentally. It can be concluded that :

1- Spacing ratio variation has a little effect on the pressure drop. The values of pressure drop (ΔP) seem to increase with increasing space ratio (S/D).

2- The enhancement of heat transfer coefficient was accompanied with an increase in pressure drop.The percentage increase in pressure drop is dependent on both the spacing ratio and the diameter of rings .

3-The effect of rings diameter on pressure drop is more than that of spacing ratio variation.

4- The effect of rings diameter variation on the average heat transfer is more that of spacing ratio.

5- The values of the average heat transfer (h_{av}) seem to increase with increasing space ratio (S/D) as shown in Fig (17)

6- The highest heat transfer coefficient is reached at spacing ratio (S/D =30) with diameter of rings d=8mm with about 66 % higher than without using rings.

7-The comparison between theoretical and experimental results shows fairly good acceptance.

Nomenclature

A_c Tube cross-sectional area, m^2

A_s Surface area of tube, m^2

C_{1e} Constant

C_{2e} Constant

C_m Constant

C_D Constant pressure specific heat, K j/ K g.k

H Manometer height differenc across the test section, mm

h Heat transfer coefficient, $w\ / m^2.k$

K Thermal conductivity, w/m. k

energy due to mean velocity

h Heat transfer coefficient, $w\ / m^2.k$

K Thermal conductivity, w/m.k

energy due to mean velocity

t Thickness of tube, mm

L Test section length, m

$m^\cdot$ Mass flow rate, kg/s

p pitch, m

P Pressure, pa

P_{rt} Turbulent prandtl number

P_r prandtl number, (mC_p / k)

ΔP Pressure drop, pa

U Overall heat transfer coefficient

Nu Nusselt number, (hD/k)

Re Reynolds number, (ρ VD/ μ)

Q Total heat transfer rate, W

q Heat flux, W/m^2

S_T Source term of heat,

T Temperature, K

V Flow velocity

u_i Velocity in x-direction

u_j Velocity in y-direction

v_r Radial velocity

v_θ Tangential velocity

$v^\cdot$ Volume flow rate, m^3 / S

S/D Space ratio

D Tube diameter, mm

D Ring diameter, mm

F Frication factor

G Gravity acceleration, $m\ / S^2$

G_b Generation of turbulence kinetic energy due to buoyancy

G_k Generation of turbulence kinetic

Greek symbols

ρ Density

Γ_e Diffusion factor

μ_t Turbulent viscosity

μ Dynamic viscosity

N Kinematic viscosity

S_k Turbulent prandtl number for ε

B Coefficient of thermal expansion

y Stream function

e Turbulent dissipation

Subscripts

av Average

b Bulk

I Inlet

O Outlet

S Surface

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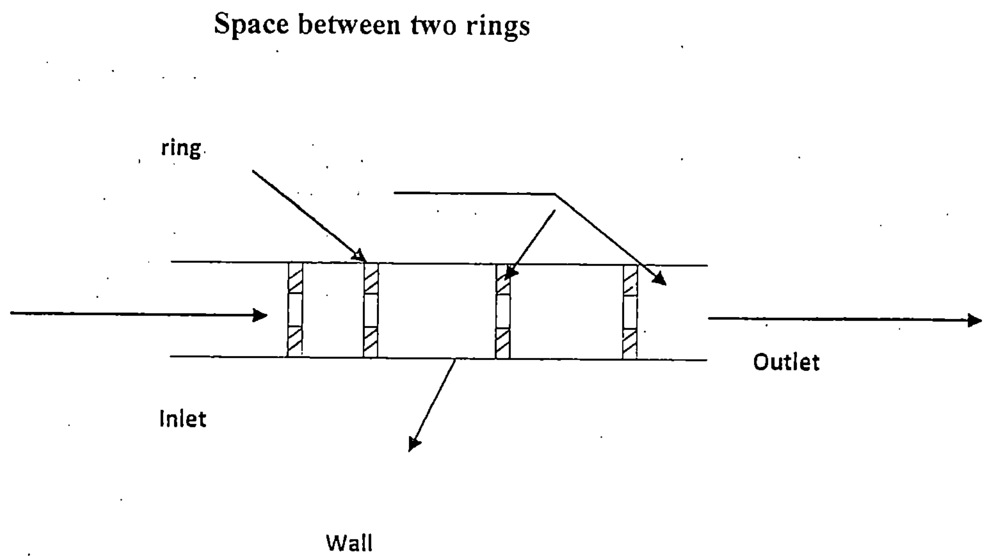

Fig (1) tube condenser with rings

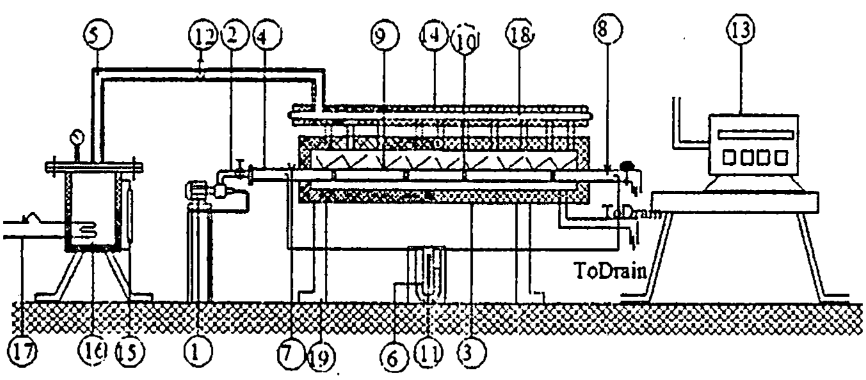

Fig.2 Schematic layout diagram of the experimental test rig.

1-Motor	6-U-tube manometer	11-measure ruler	16-electric boiler
2- water inlet	7- water inlet thermocouple	12-control valve	17-electric main heater
3- box	8-water outlet thermocouple	13-temperature recorder	18-steam pipe supply
4- test tube	9-thermocouple	14- thermal insulation	19-movable base
5-steam inlet	10-a ring	15-indicator glass level	

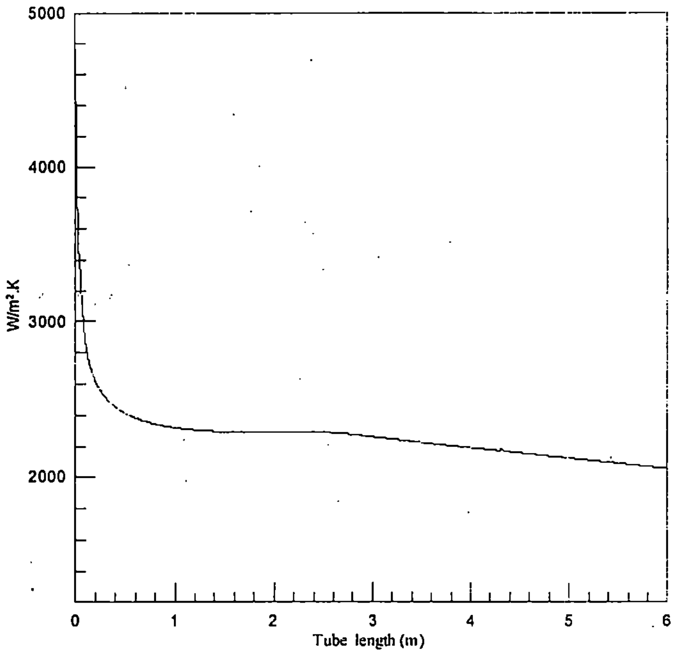

Fig.3 Heat transfer coefficient distribution on the tube length without rings

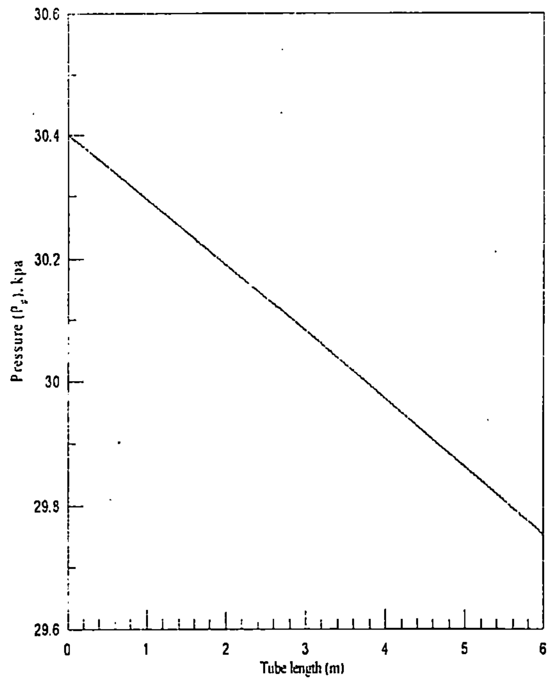

Fig.4 Gauge pressure distribution in the tube length without rings

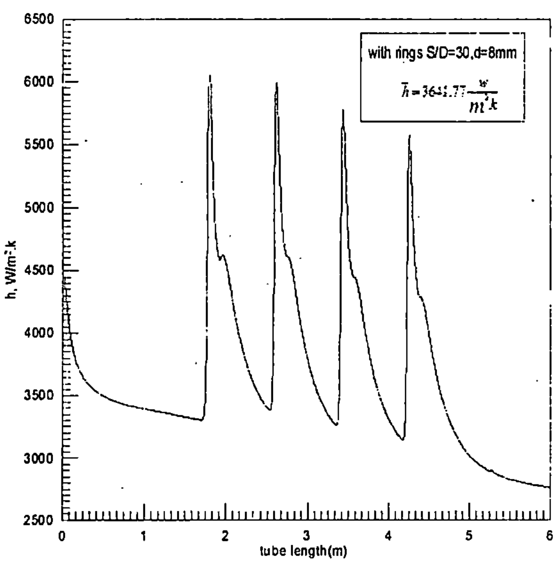

Fig.5 Heat transfer coefficient distribution on the tube length with rings

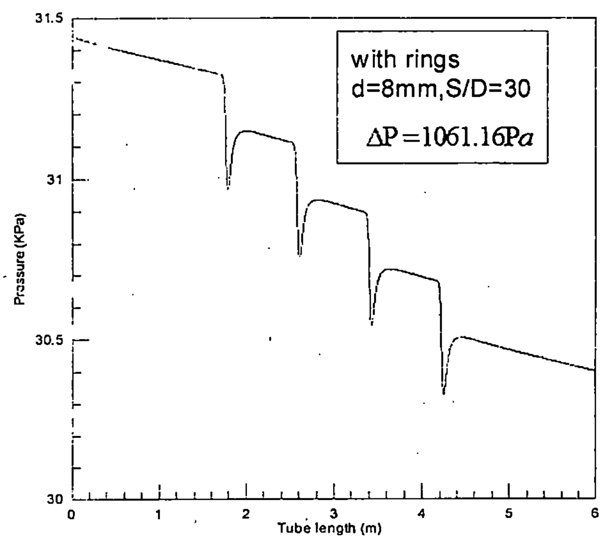

Fig.6 Gauge pressure distribution in the tube length with rings

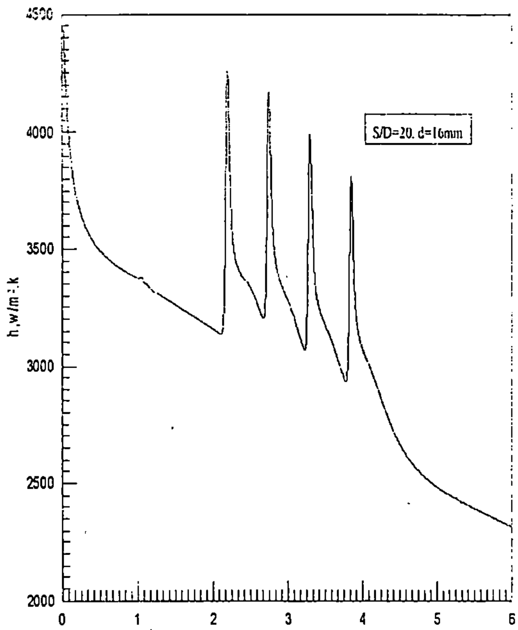

Fig.7 Heat transfer coefficient on the tube length with rings

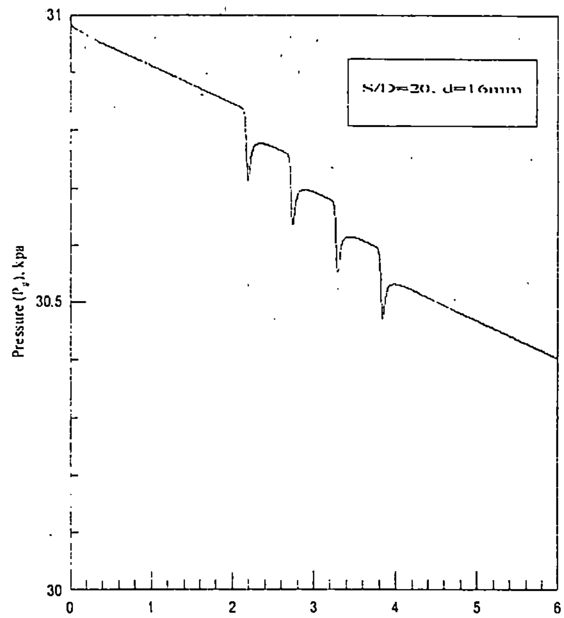

Fig.8 Gauge pressure distribution in the tube length with rings

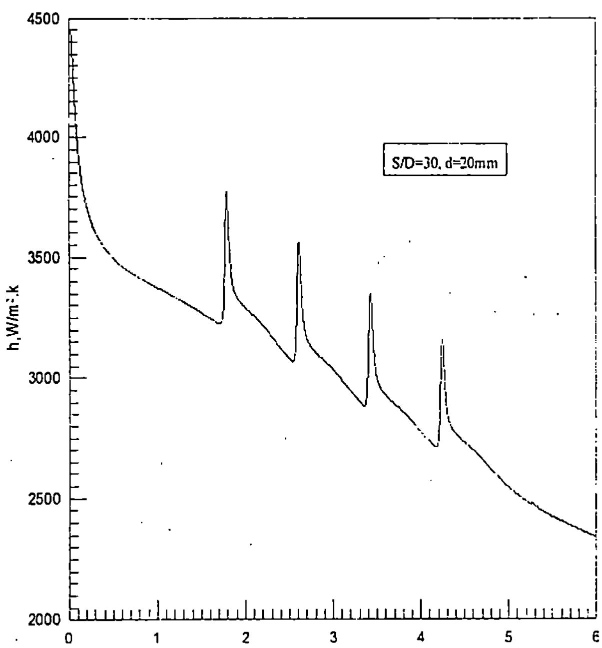

Fig.9 Heat transfer coefficient distribution on the tube length with rings

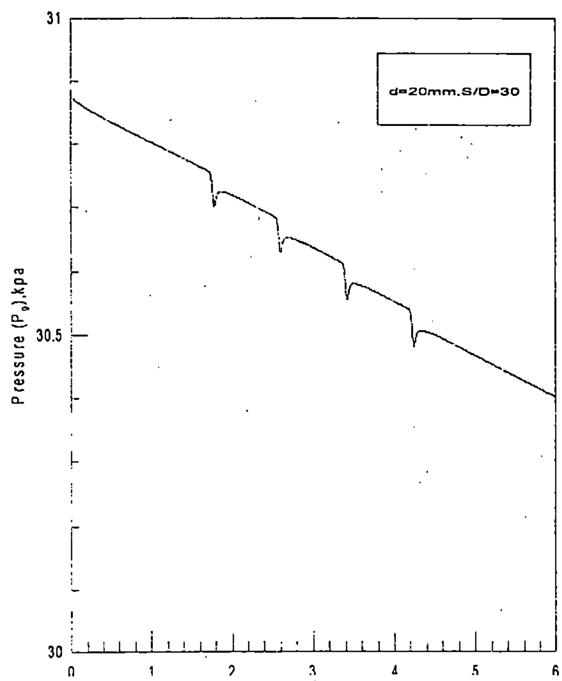

Fig.10 Gauge pressure distribution in the tube length with rings

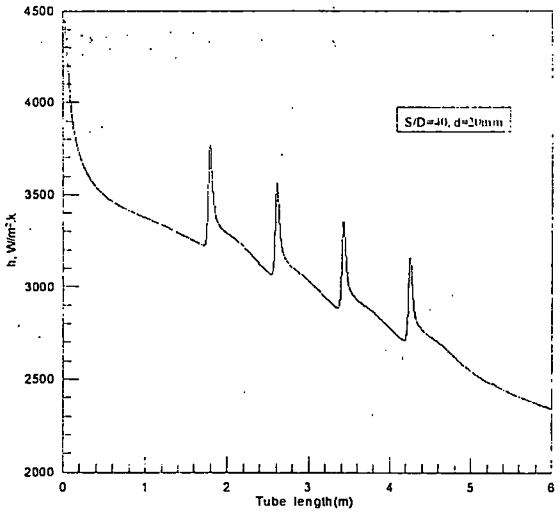

Fig.11Heat transfer coefficient distribution on the tube length with rings

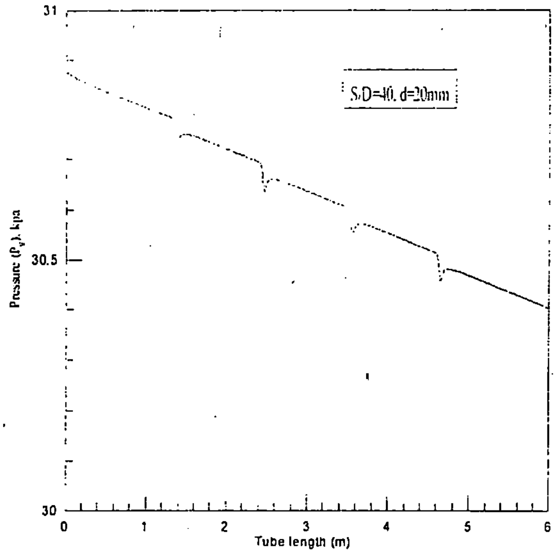

Fig.12 Gauge pressure distribution in the tube length with rings

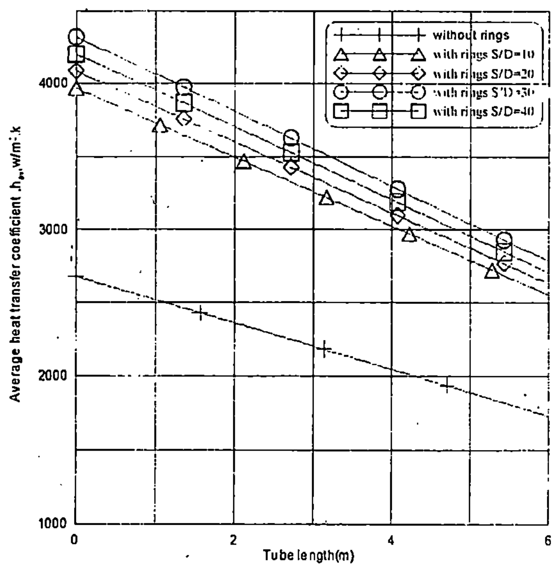

Fig. 13 Average heat transfer coefficient on the tube length without and with rings (d=8mm) theoretical model

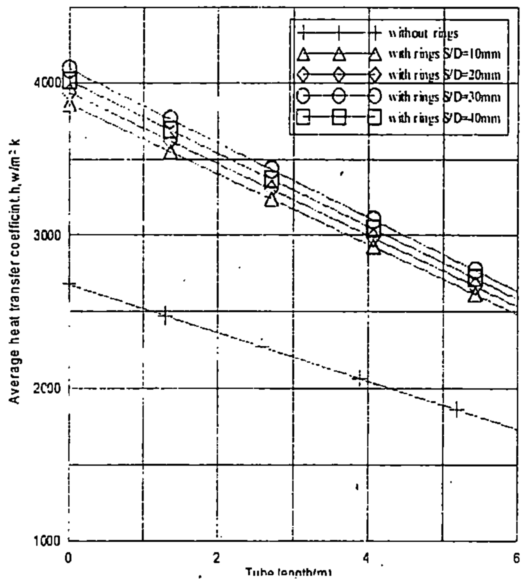

Fig. 14 Average heat transfer coefficient on the tube length without and with rings (d=12mm) theoretical model

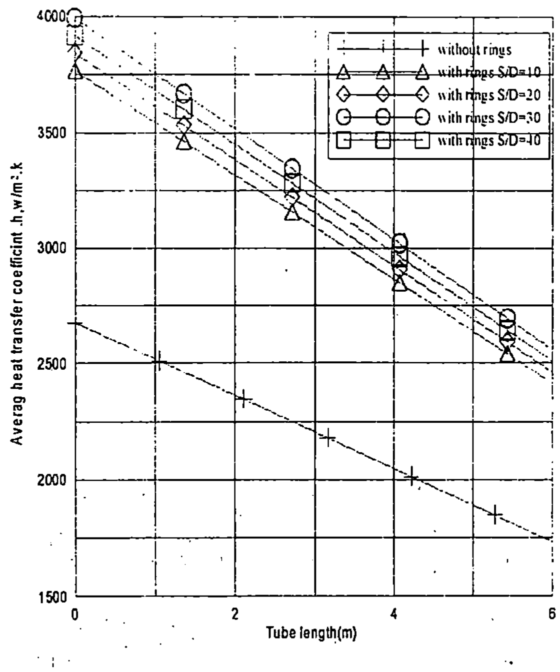

Fig.15 Average heat transfer coefficient on the tube length without and with rings (d=16mm) theoretical model

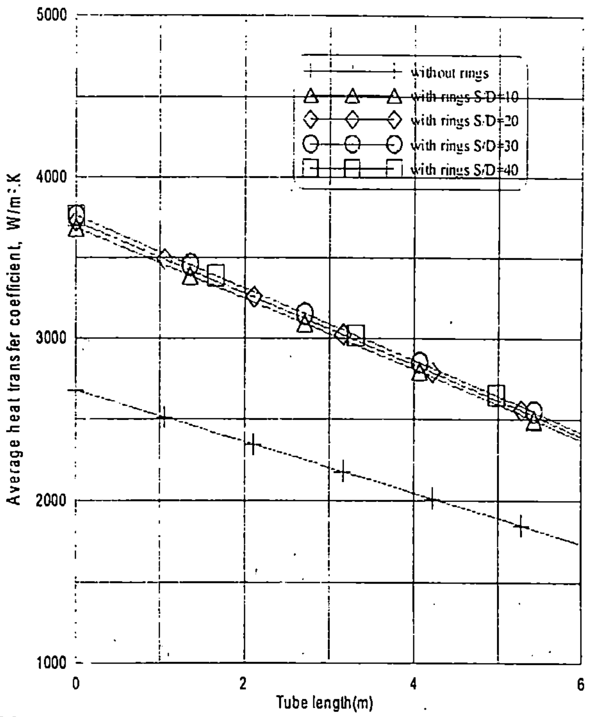

Fig. 16 Average heat transfer coefficient on the tube length without and with rings (d=20mm) theoretical model

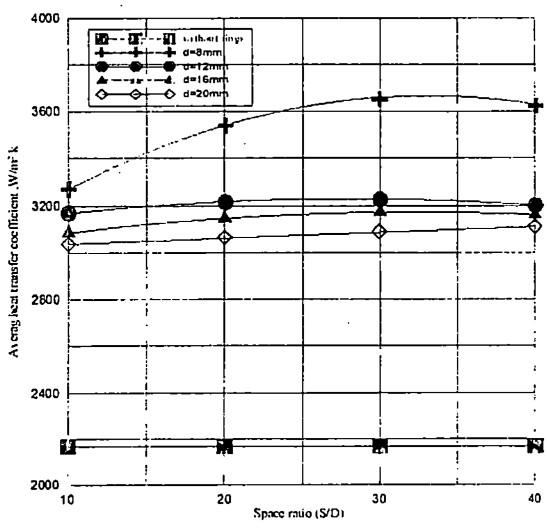

Fig.17 Average heat transfer coefficient at various space ratio(theoretical model)

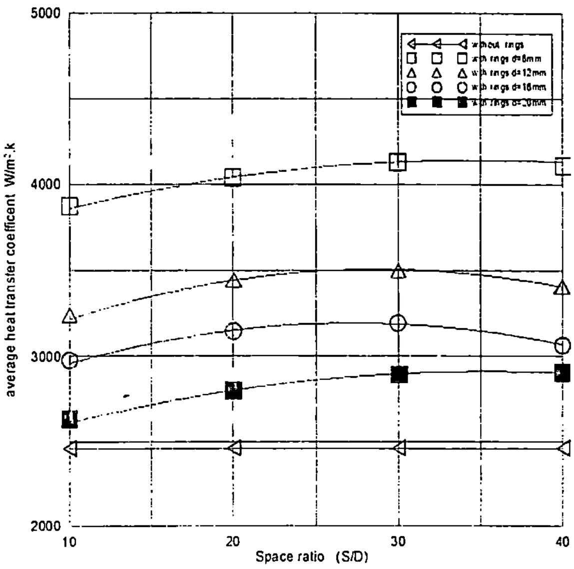

Fig.18 Average heat transfer coefficient at various space ratio(experimental)

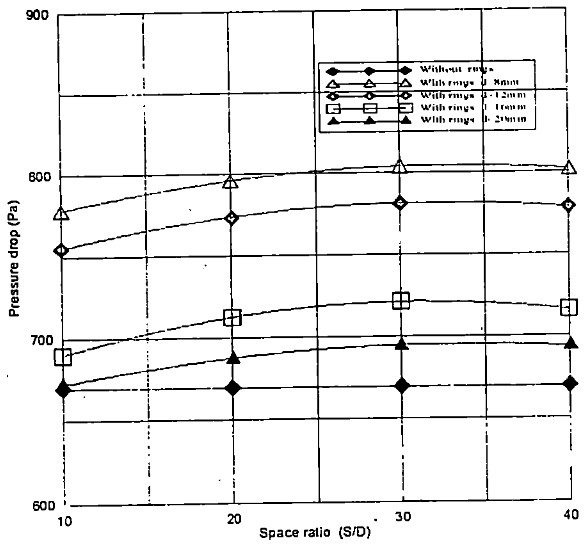

Fig.19 pressure drop at various space ratio(theoretical model)

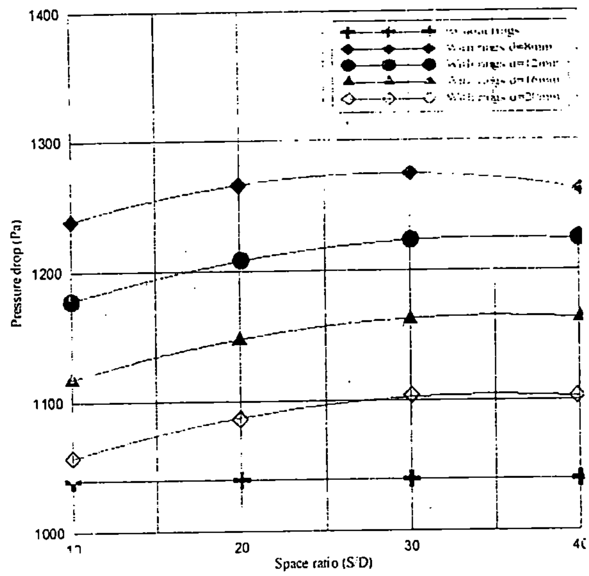

Fig.20 pressure drop at various space ratio(experimental)

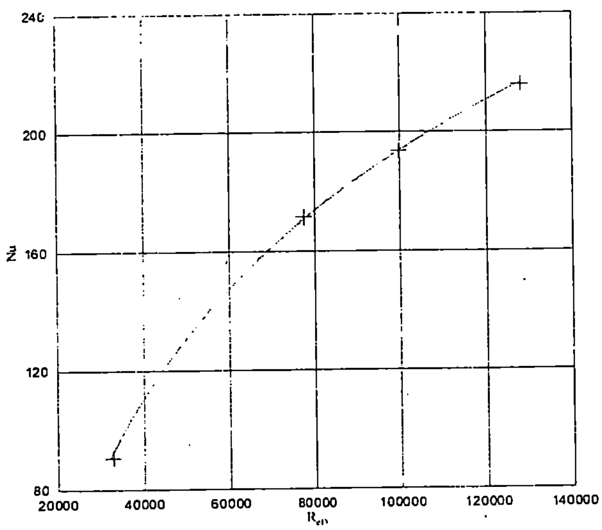

Fig.21 variation of Nusselt with Reynold number for condenser tube with rings of d=8mm, S/D=30

$$\overline{Nu} = 0.0923 R_{eD}^{\ 0.661}$$