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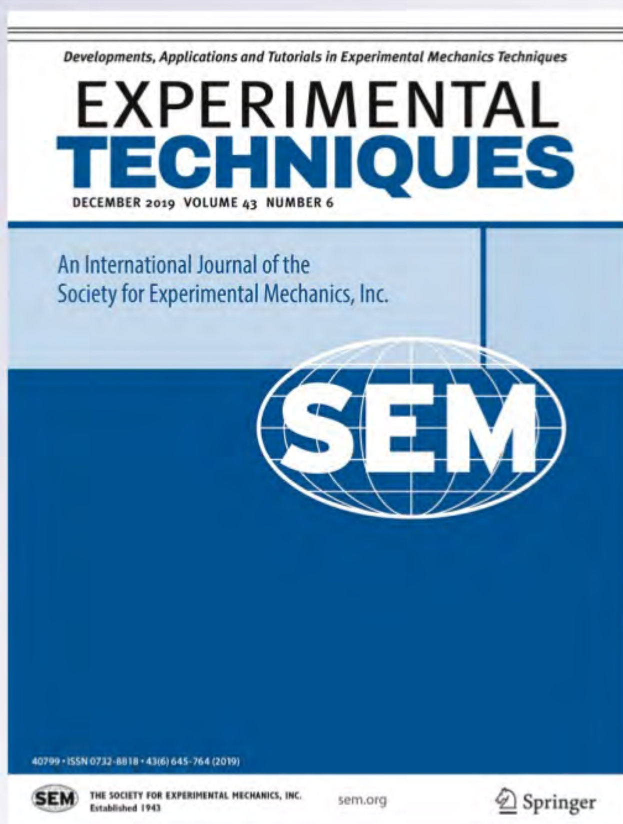

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Experimental Study of Stationary-Head/Channel Cover STHE Prototype Using ε-NTU Method

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Abstract

Our research focuses on the performance evaluation of the small shell-and-tube heat exchanger (STHE) – laboratory type. The experiment used the prototype design of stationary-head/channel cover using the ring rubber, which separate the hot and cold fluid in a chamber. The stationary-head prototypes unusually are designed using low cost manufacture and simple construction, without bolt or nut to join both the stationary-head and shell. The shell has four holes to supply hot/cold fluid, and next to the tube-sheet hole to supply cold/hot fluid, the position both of them are inside the stationary-head. The single and double segmental baffles were used in this study. Calculation of thermal performance and effectiveness of STHE were calculated based on ε-NTU method. The correlation of heat transfer proposed was based on the unique construction of stationary-head design for the effectiveness of STHE. The data were collected from the both single and double segmental baffles, which were investigated by varying flow rate. The investigation including Reynolds and Nusselt number, heat transfer coefficient, and pressure drop which all effects of the shell-and-tube heat exchanger effectiveness. The results show that the ratio of the actual heat transfers for single segmental was higher than double segmental and the average effectiveness of single segmental baffle was 10 to 30% less than the double segmental baffles.

Keywords Heat exchanger · STHE · Baffle · NTU · Stationary-head

Introduction

At present, energy consumption in industrial processes is very important to manage due to the limitation of fossil fuel. Heat exchanger is one of the equipment that is used in the industry to support the production and manufacturing and are related to heat transfer and energy.

Many researchers have been used the heat exchanger to develop and reduce the heat transfer time as well as increase the energy and fuel efficiencies. Many studies discussed about the specific aspect of shell-and-tube heat exchanger. Mica Vukic and Tomic discuss about the effectiveness of shell-and-tube heat exchanger using different variation number of segmental baffle [1]. Bayram and Sevilgen investigate the effect of variable baffle spacing on the thermal performance using numerical method (CFD) [2]. Ozden and Tari observe the shell side of the shell-and-tube heat exchanger using numerically modelling in a small heat exchanger [3]. They investigate the baffle spacing, baffle cut, heat transfer coefficient, and pressure drop with Bell-Delaware method results. Delaware method is also used by Gaddis and Gnielinski to calculate the pressure drop in an ideal tube bank coupled with correction factors [4]. Sparrow and Reifschneider discuss about the effect of interbaffle spacing in the shell-and-tube heat exchanger to determine the response of the heat transfer and pressure drop [5]. The other researchers, Wee and Aicher investigate using 32 different heat exchanger test experimentally. The heat exchanger differs by number of tubes, length, shell-and-tube diameter, nozzle diameter and tube pitch. They confirm that the tube pitch can be neglected in shell-and-tube heat exchanger used in real processes [6].

The heat exchanger used in this study was comparatively small sized. The experiment investigates the effect of both single and double segmental baffles compare with varying

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flow rate. In this study, investigation of the experimental results consisted of Reynolds and Nusselt number, heat transfer coefficient, and pressure drop effect of the shell-and-tube heat exchanger effectiveness.

Method

Thermal Design

Shell is a container where the tube bundle is placed inside the shell. Thermal design of the shell-and-tube heat exchanger includes heat transfer area, number of tube, length and tube diameter, tube pitch, number and type of baffle, and shell-and-tube side pressure drop. The calculation of heat transfer coefficient for cold and hot fluid are assumed without phase change.

Experimental Setup

One of the important parts of heat exchanger are two stationary-heads on the left and right side of the shell, which are joined with the shell. The combined joint between stationary head and shell is commonly fixed using bolt or nut.

The prototypes of stationary-head are designed using low cost manufacture and simple construction, without bolt or nut to join both the stationary-head and shell. The prototype uses the flange joint and the ring rubber that are used to unify the stationary-head and shell as shown in Fig. 1(c). The construction design of the stationary-head is like bonnet-type channel cover (integral cover). In each stationary-head there are two chambers. Chamber 1 for inlet cold fluid (shellside) and chamber 2 for outlet hot fluid (tubeside), which are separated with the rubber at the end of the shell as shown in Fig. 1(a). Both of the stationary-heads (channels cover) are integral with the tube-sheets. One of the benefit is both the stationary heads can be removed easily to clean the tubes, and the leakage of the shellside fluid can be minimized with the ring rubber between the stationary-head and flange joint. The shell has four holes inside the stationary-head next to the tube-sheet for the inflow or outflow of the cold/hot fluid as shown in Fig. 1(b).

The experiment used fixed-tube-sheet heat exchanger design with one shell pass and one tube pass. The number of tube used were inside the shell of 45 tubes with 10 mm outside diameter, 1 mm thickness, 1100 mm length, and the tube pitch is 12.85 mm. The inner diameter of shell is 110 mm with thickness 10 mm. The material of shell and tubes are flexiglass and copper, respectively.

The thermal conductivity of flexiglass (Polymethyl methacrylate (PMMA)) and copper are 0.24 W/m K and 401 W/m C, respectively. The material of stationary-head is POM (Polyoxymethylene). The design of stationary-head is very compact and there is a ring rubber inside the stationary-head to separate between the hot fluid and cold fluid.

The baffle used in the experiment was triangular pattern 30^0 single segmental baffle and rotated triangular 60^0 double segmental baffle as shown in Fig. 2. The number of baffle used in the experiment was 10 baffles ($N_p = 10$) with the baffle cut and spacing of 20% and 100 mm, respectively. The flow rate of shell and tube each was controlled by one valve. Both of them were controlled at the inflow of shell and tube. The flow rate data were read using pulsemeter MP5W-Autonics, which was calibrated by weighting the collected water at certain period of time.

The pressure of shell was measured using differential pressure transmitter ST3000 model STD910 and installed at the inlet and outlet of the shell. The measuring range of the pressure transmitter was −1000 to 1000 Pa with accuracy measurement of linear output based on the equipment of $\pm((0.15 + 0.15 \times (1.0/(\text{greatest range value/lower range value})))\%$ [7].

The temperatures was measured using thermocouple (K type), which was inserted in the inlet and outlet shell and tube of heat exchanger. The temperature and pressure data were collected using NI 9213 and NI 9203, respectively. The accuracy of fluid temperature measurement was ±0.001 °C. Two thermocouples measured the cold fluid temperature at the inlet and outlet of heat exchanger shell side, and the other two thermocouples measured the hot fluid temperature at the inlet and outlet of heat exchanger tube side.

Figure 3 shows the schematic diagram of experimental installation that was used in the experiment. In general, the system has two loops of different fluids, hot and cold fluids. Both fluids were stored in two different tanks. The system also has two pumps and two flowmeters. Four valves were installed at the outflow of both pumps. Two valves were installed to control the flow rate of fluid, and the other valves for emergency bypass purpose. The 1.8 kW of electric heater was installed as the source of heat generation. The hot fluid will be cooled using radiator.

Figure 4 shows the experimental set-up of shell-and-tube heat exchanger.

Experimental Measuring

The fluid flowing in the shell-side and tube-side was water and it was considered with constant properties at the inlet temperature. The experimental investigation was conducted under ambient temperature of 25 to 27 °C. The experiment used constant volume flow rate at each condition, single and double segmental baffle. The inlet constant volume cold fluid flow rate for single and double segmental baffle was $V_{ColdSingle} = 11.75$, 9.75, 7.5, 5.75, and 3.75 L/min, and $V_{ColdDouble} = 12.75$, 10.5, 8.5, 6, and 4 L/min, respectively, with both inlet temperatures were maintained at 25–27 °C.

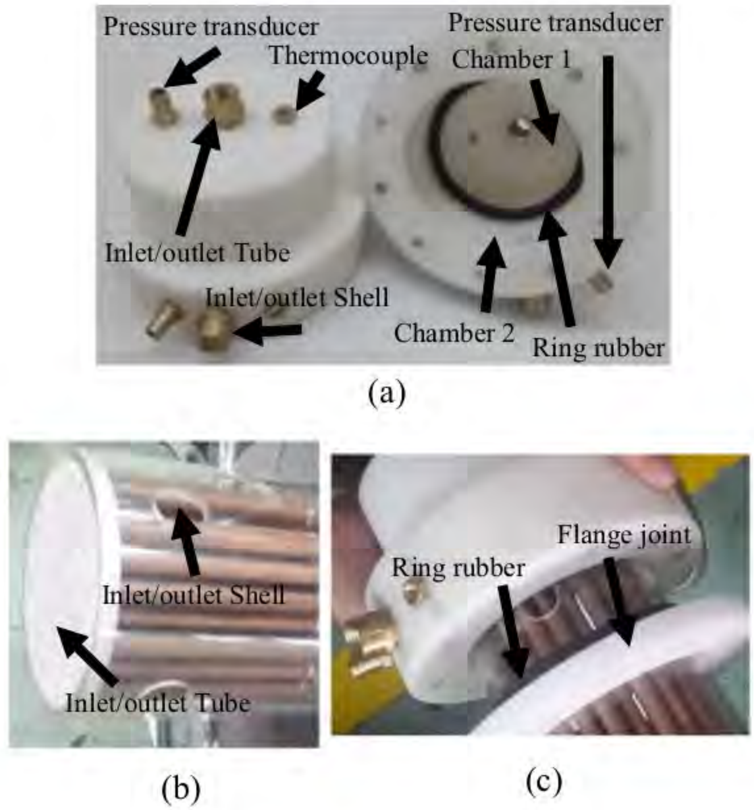

Fig. 1 Design of (**a**) stationary-head, (**b**) inflow and outflow hole position, (**c**) stationary-head assembly on shell

On both single and double segmental baffles, the inlet constant volume hot fluid flow rate was $V_{HotSingle} = 9$, 7.5, 5.5, 3.5, and 1.5 L/min, and $V_{HotDouble} = 11.3$, 8.25, 6, 4, and 2.55 L/min, respectively, with both inlet temperatures were maintained using electric heater.

Single and double segmental baffles was investigated for the 17 and 25 variations of the mass flow rate, respectively. These mass flow rate variations are adjusted with the design and dimension of the prototype. Table 1 shown the mass flow rate of the single and double segmental baffle of hot and cold fluid.

Data Processing

The first step of the present study, we check the accuracy of the test procedure, we should know the overall energy conservation between the cold side and the hot side heat transfers. Heat load of heat exchanger can be estimated from heat balance,

$$Q_{Shell} = Q_{Tube} \tag{1}$$

$$Q_{Shell} = \left[\acute{m} \, . \, Cc \, (Tc_{out} - Tc_{in})\right]_{Shell} \tag{2}$$

$$Q_{Tube} = \left[\acute{m} \, . \, Ch \, (Th_{in} - Th_{out})\right]_{Tube} \tag{3}$$

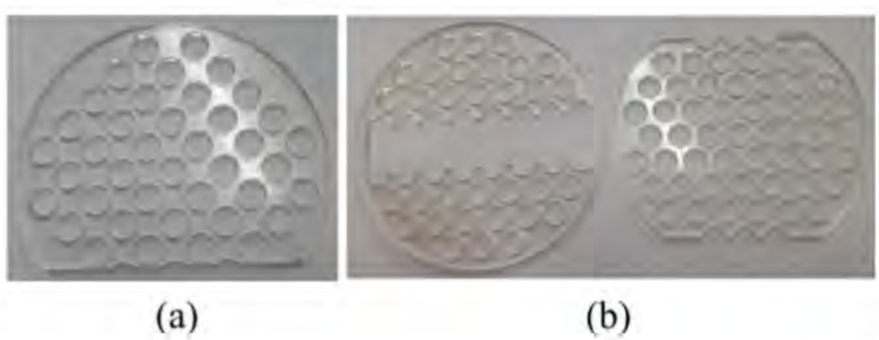

Fig. 2 Design of heat exchanger baffles. (**a**) triangular pattern 30° - single segmental baffle, (**b**) rotated triangular 60° - double segmental baffle

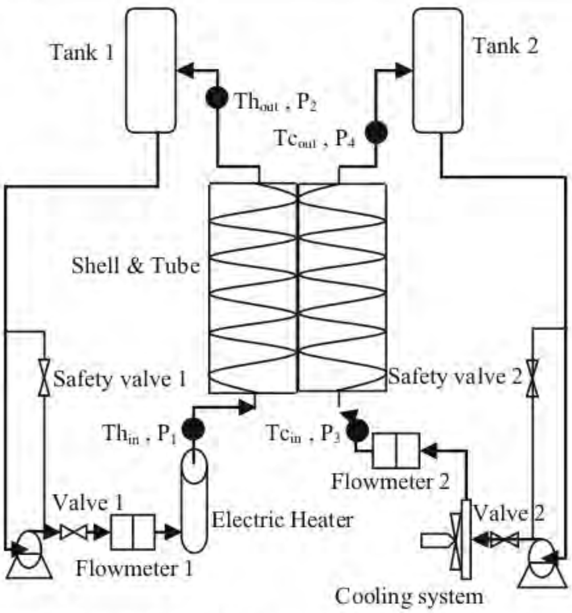

Fig. 3 Schematic diagram of experimental installation

The temperature data of single and double segmental baffle are used to check the heat balance between hot and cold fluid.

The experimental data of mass flow rates, temperatures, dimensions of the equipment were collected and used to calculate the convective heat transfer coefficient, overall heat transfer coefficient, Reynolds, Prandtl, Nusselt number, and effectiveness of heat exchanger. Hence the flow was laminar, with Reynolds Number below 2300 and Prandtl Number bigger than 0.48. The Nusselt Number of tube (Nu_{Tube}) was recommended to be calculated using Kays and Hausen correlation [8, 9],

$$Nu_{Tube} = 3,66 + \frac{0,0668\left(\frac{d_{In\ \ Tube}}{L_{Tube}}\right)Re_{Tube}.Pr_{Tube}}{1 + 0,04\left[\left(\frac{d_{In\ \ Tube}}{L_{Tube}}\right)Re_{Tube}.Pr_{Tube}\right]^{2/3}} \quad (4)$$

Generally, the average heat transfer coefficient for the entire tube bundle inside the shell can be calculated using tube arrangement in a bank staggered condition. The flow across the tube bundle composed of 45 rows with Reynolds Number between 10 to 100, and Prandtl Number between 0.7 to 500. The Nusselt Number of shell (Nu_{Shell}) was recommended to be calculated using an empirical correlation due to Hilpert [10], and Zhukauskas correlation [11] with $C = 0.9$ and $n = 0.4$ (staggered condition),

$$Nu_{Shell} = C.Re_{Shell\,Max}{}^{n}.Pr_{Shell}{}^{0.36}\left(\frac{Pr_{Shell}}{Pr_{Wall}}\right)^{1/4} \quad (5)$$

Calculation of the outer and inner area contact fluid of tube ($A_{Out\ Tube}$ and $A_{In\ Tube}$) depends on the outer and inner diameter ($d_{Out\ Tube}$ and $d_{In\ Tube}$), tube length (L_{Tube}) and number of tube (N_{Tube}) so,

$$A_{In\ \ Tube} = \pi.d_{In\ \ Tube}.L_{Tube}.N_{Tube} \quad (6)$$

$$A_{Out\ \ Tube} = \pi.d_{Out\ \ Tube}.L_{Tube}.N_{Tube} \quad (7)$$

Four variables of temperature were measured from location of inlet hot fluid (Th_{In}), outlet hot fluid (Tc_{Out}), inlet cold fluid (Tc_{In}), and outlet cold fluid (Tc_{Out}). Cross flow was used in the experiment.

The total thermal energy resistance (R_{Th}) can be calculated using heat transfer coefficient of hot (h_h) and cold (h_c) fluids respectively, thermal conductivity of tube (k), as well as the inner and outer area contact

Fig. 4 Experimental set-up of shell-and-tube heat exchanger

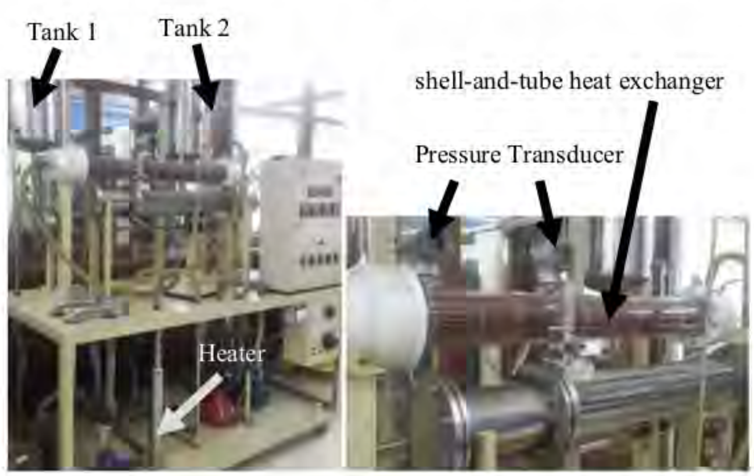

Table 1 Mass flow rate of single and double segmental baffle

No	Single Segmental Baffles (L/min)		Double Segmental Baffles (L/min)	
	Hot Fluid	Cold Fluid	Hot Fluid	Cold Fluid
1	9	11.75	11.3	12.75
2	7.5	9.75	8.25	10.5
3	5.5	7.5	6	8.5
4	3.5	5.75	4	6
5	1.5	3.75	2.55	4
6	9	9.75	11.3	10.5
7	5.5	5.75	8.25	8.5
8	1.5	5.75	6	6
9	9	7.5	4	4
10	5.5	3.75	2.55	12.75
11	1.5	11.75	11.3	8.5
12	9	5.75	8.25	6
13	5.5	11.75	6	4
14	1.5	9.75	4	12.75
15	9	3.75	2.55	10.5
16	5.5	9.75	11.3	6
17	1.5	7.5	8.25	4
18			6	12.75
19			4	10.5
20			2.55	8.5
21			11.3	4
22			8.25	12.75
23			6	10.5
24			4	8.5
25			2.55	6

fluid of tube respectively. Thermal resistance of inner (R_{Di}) and outer (R_{Do}) tube respectively depends on the operating temperature, fluid velocity, and tube length.

$$R_{Th} = \frac{1}{h_h.A_{In\ \ Tube}} + \frac{\ln\left(\frac{d_{Out\ \ Tube}}{d_{In\ \ Tube}}\right)}{2\pi.k.L_{Tube}} + \frac{1}{h_c.A_{Out\ \ Tube}} \quad (8)$$

The overall heat transfer coefficient (U_{Total}) depends on the diameter of inlet and outlet, heat transfer coefficient and fouling factor or thermal energy resistance (R_{th}) of hot/cold fluids, respectively [8, 9], so

$$\frac{1}{U_{Total}} = \frac{1}{h_{In}D_{In}} + \frac{D_{Out}\ln\left(\frac{D_{Out}}{D_{In}}\right)}{2\,k} + \frac{1}{h_{Out}} + \frac{R_{D\,In\ \ Tube}D_0}{D_{Out}} + R_{D\,Out\ \ Tube} \quad (9)$$

ε-NTU

The solution approach is the calculation using ε - NTU (effectiveness - Number of Transfer Unit) method. This method using three dimensionless parameter; heat capacity rate ratio (C_R) - the ratio of the minimum to the maximum value of heat capacity rate for hot and cold fluids (C_{min}/C_{max}), number of transfer units (NTU), and effectiveness (ε). The C_R is calculated for both hot and cold fluids as the product of the mass flow rate and specific heat capacity of the fluid. The number of transfer units (NTU), is the ratio of the thermal capacity (UA) to the fluid's minimum ability to absorb heat (C_{min}). The last parameter, the ratio of the actual heat transfer rate ($\dot{Q}$) to the maximum possible heat transfer rate ($\dot{Q}_{max}$) for exchanger is determined to define the effectiveness of the heat exchanger (ε), [8].

$$\varepsilon \equiv \frac{\dot{Q}}{\dot{Q}_{Max}} \quad (10)$$

$$\varepsilon = f\left(NTU, \frac{C_{Min}}{C_{Max}}\right) \tag{11}$$

$$NTU = \frac{UA}{C_{Min}} \tag{12}$$

and

$$\varepsilon = \frac{C_h\left(T_{h,In}-T_{h,Out}\right)}{C_{Min}\left(T_{h,In}-T_{c,In}\right)} \tag{13}$$

or

$$\varepsilon = \frac{C_c\left(T_{c,Out}-T_{c,In}\right)}{C_{Min}\left(T_{h,In}-T_{c,In}\right)} \tag{14}$$

Result and Discussion

As the first step of the present study, in the steady state condition and assumes no phase change in any of the fluids, the overall heat transfer from the hot side to the cold side should be balanced. Heat load of the hot and cold fluids in single and double segmental baffles at the various flow rates are showed in Figs. 5 and 6. The figures show the heat load of hot and cold fluids has different value. It is caused by the baffles type and the properties of the stationary-head materials that are used. The material of stationary-head which are made of Polymethyl-methacrylate (PMMA) have thermal conductivity very small of 0.24 W/m K.

The difference of heat load between hot and cold fluid for single segmental baffle is greater than double segmental baffle as shown in that figure, it is caused by the type and design of the baffles and heat transfer of double segmental baffles better than single segmental baffles ones.

The experimental data also can be used for calculating the parameter such as Nusselt and Reynolds number, effectivity, heat transfer coefficient and thermal resistance that can be compared with the correlation based ones. Figures 7, 8, and 9 have correlation with eq. (4) and (5), it is caused by the Nusselt number is one of the important numbers to evaluate the capability of heat exchanger. The Nusselt number is influenced by the convective heat transfer coefficient, characteristic length and thermal conductivity of the materials are used. Therefore, we use and compare the experimental data of the single and double segmental baffles.

Figure 7 shows the relationship of Nusselt number with Reynolds number for the experimental data investigation. It was observed for the single and double segmental baffles, in which both Nusselt number of the tube-side (hot fluid) and shell-side (cold fluid) increase, as both the Reynolds number of the tube-side and shell-side increase. This indicates that the trend phenomena of the results are in accordance with the theory, which are the Nusselt number directly proportional with Reynolds number. The diagram shows that the average of single segmental baffles is above the double segmental baffles. The correlation of Reynolds and Nusselt number help to put in order of experimental results and compare them to the numerical results (in the next research). The Nusselt number almost constant under developed conditions in the laminar flow and depend on the shape of the geometry of the system.

Figure 8 show the relationship of heat transfer coefficient and Reynolds number. Generally, the graphs show the value of heat transfer coefficient (h) for both hot and cold fluids in the single and double segmental baffles increase proportional to the Reynolds number increases. The h value for both hot and cold fluids of double segmental baffles increase faster than single segmental baffles. It is caused by the mass flow rates in both single and double segmental baffles is very low,

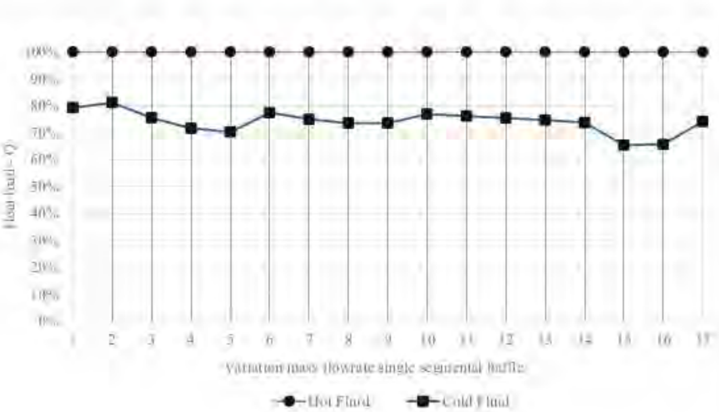

Fig. 5 Heat load of hot and cold fluid for single segmental baffles

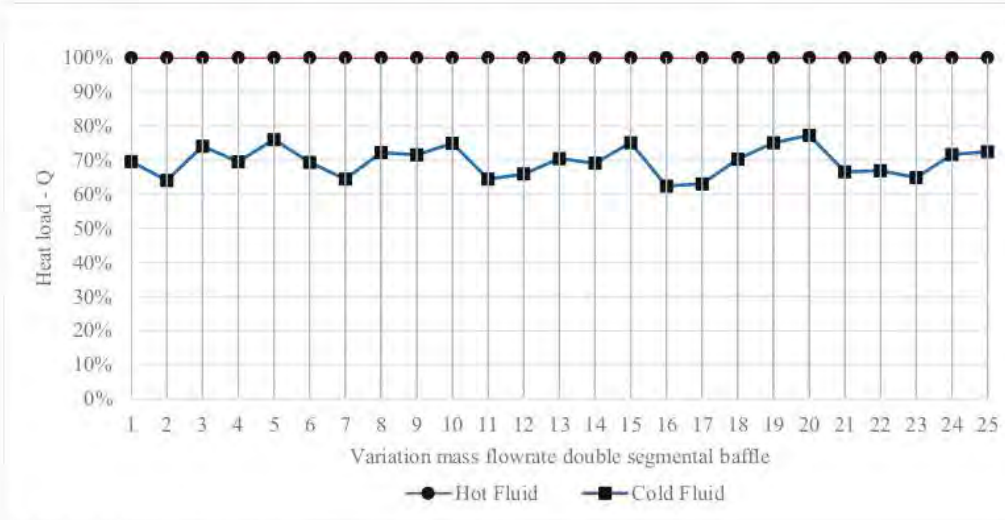

Fig. 6 Heat load of hot and cold fluid for double segmental baffles

resulting the laminar flow. The baffles type which are used in the double segmental baffles indicates suitable at the very low Reynolds number result in laminar flow. At the low Reynolds number, the value of heat transfer coefficient (h) of hot fluid in the double segmental baffles lower than single segmental baffles, but increases significantly after the Reynolds number above than 400. It same conditions for heat transfer coefficient (h) of cold fluid.

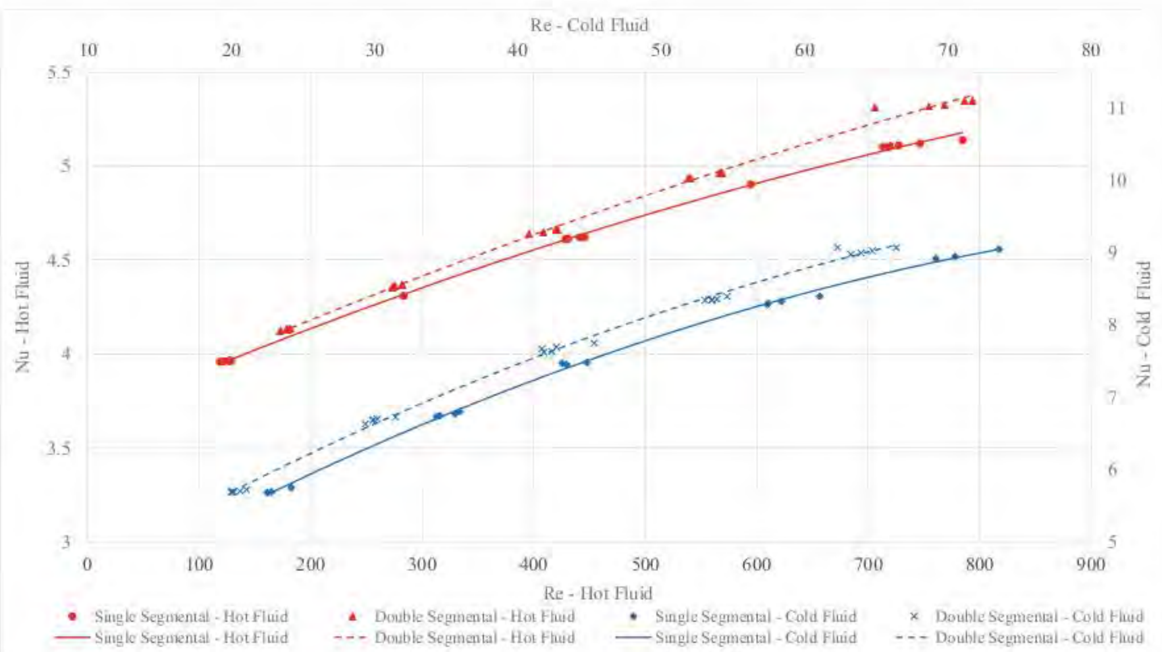

Fig. 7 Relationship between Nusselt number and Reynolds number

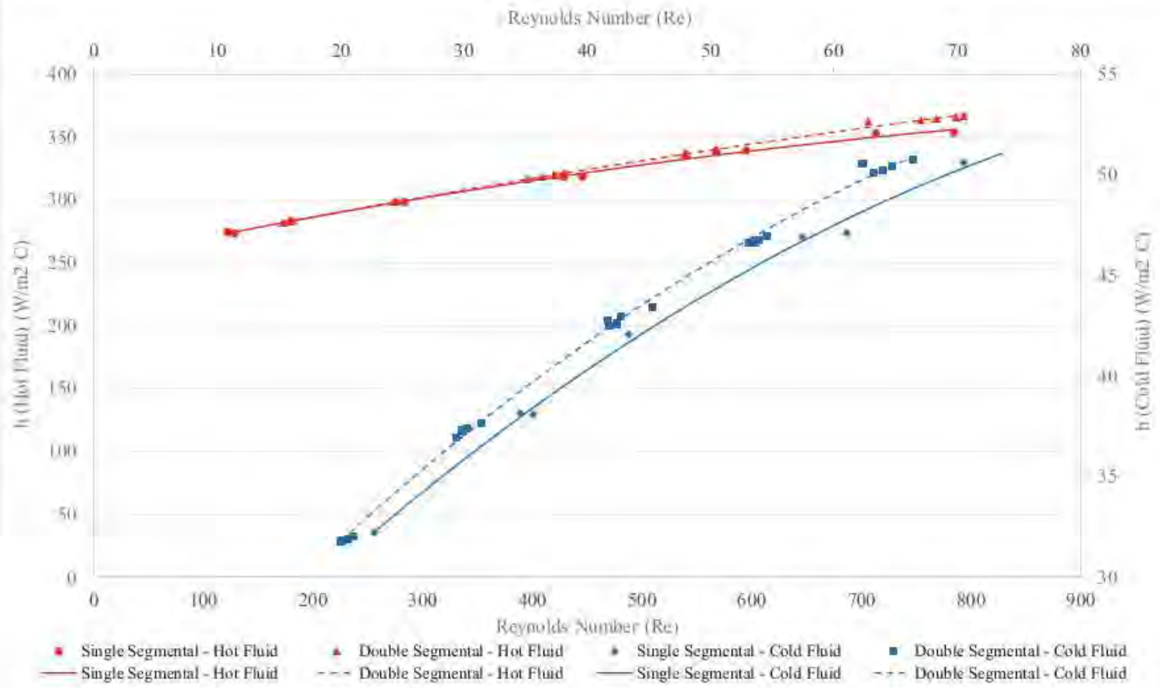

Fig. 8 Relationship between heat transfer coefficient and Reynolds number

Figure 9 shows the pressure drop increases as Reynolds number increases. This results are in line with Moody or Darcy diagram friction factor. The pressure drop increases due to the interaction of pressure force with inertial force along the boundary layer, and partly from friction and baffle type. Yildiz observes the same phenomena using helical pipe construction. [12]. The graphs show the pressure drop of hot fluid in the single and double segmental baffles almost same at the Reynolds number of 100 to 400, but increases after the Reynolds number above than 400 for single segmental baffles ones. The pressure drop of cold fluid of the single segmental baffles was higher than double segmental baffles at the same Reynolds number. The difference value between the hot fluid and the cold fluid of both pressure drop and Reynolds number are almost 10× .

Fig. 10 shows the experimental results of the ratio (h/ΔP) for both single and double segmental baffles. The relative efficiency of heat exchanger can be evaluated by comparing the ratio of heat transfer coefficient to the energy pressure loss (h/ΔP) [13]. The ratio of heat transfer coefficient to the pressure loss is very useful to compare the performance of the heat exchanger. The ratio of the heat transfer double segmental baffles has higher ratio than the single segmental one. This phenomenon also shows the overall performance of the double segmental baffles that is better, because in this case the value of ratio will be affected by the value of pressure drop. The graphs show the performance of heat exchanger is influenced by the mass flow rates, it is indicated with many large scatter at the various mass flow rates. At the hot fluid of the double segmental baffle, performance at the low mass flow rate lower than the hot fluid of the single segmental baffles.

Figure 11 shows the relationship between effectiveness of the single and double segmental baffles versus flow rate of the hot fluid. The trend of graphs shows both of the single and double segmental baffles have the effectiveness decrease as the flow rate increases, it indicates that the system and heat transfer are influenced by the amount and variations of mass flow rates which are flow inside the shell and the tube. The design and characteristic of the stationary-heads also responsible and indicate that the design of the stationary-heads has limited mass flow rates variations. The suitable range variations of the mass flow rate very important to get the high effectiveness.

Figure 12 shows the relationship between effectiveness and Number Thermal Unit (NTU) for single and double segmental baffles. The effectiveness (ε) of single and double segmental baffles are obtained based on various flow rate ratio. It can be seen that average of double segmental baffles has better

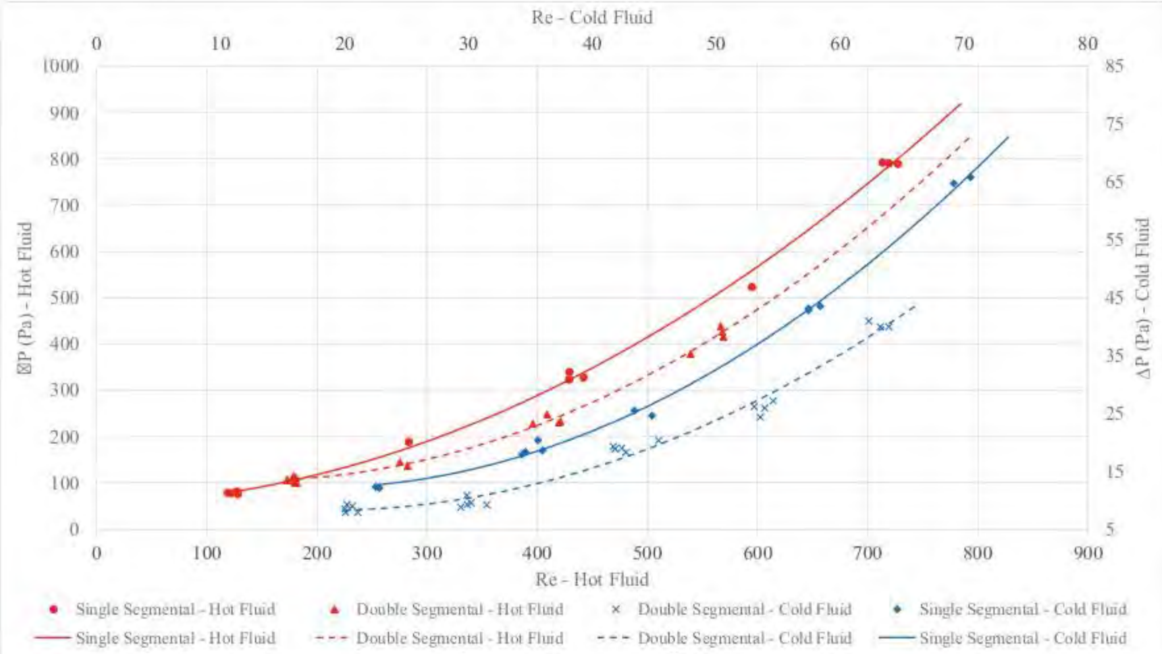

Fig. 9 Relationship between pressure drop and Reynolds number

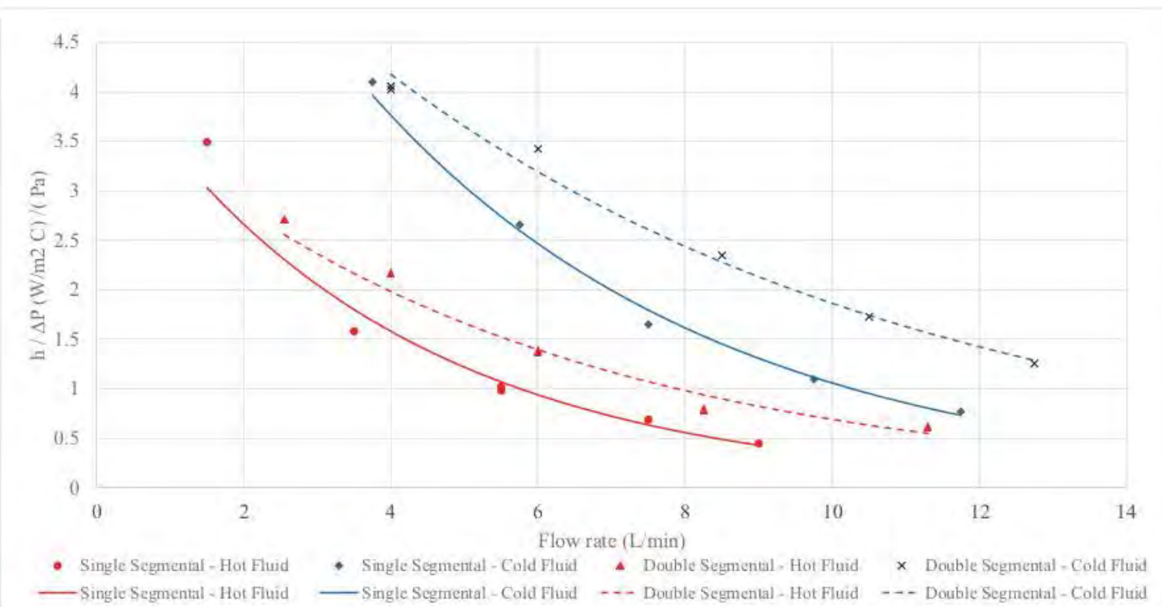

Fig. 10 Ratio of heat transfer to pressure drop for single and double segmental baffles

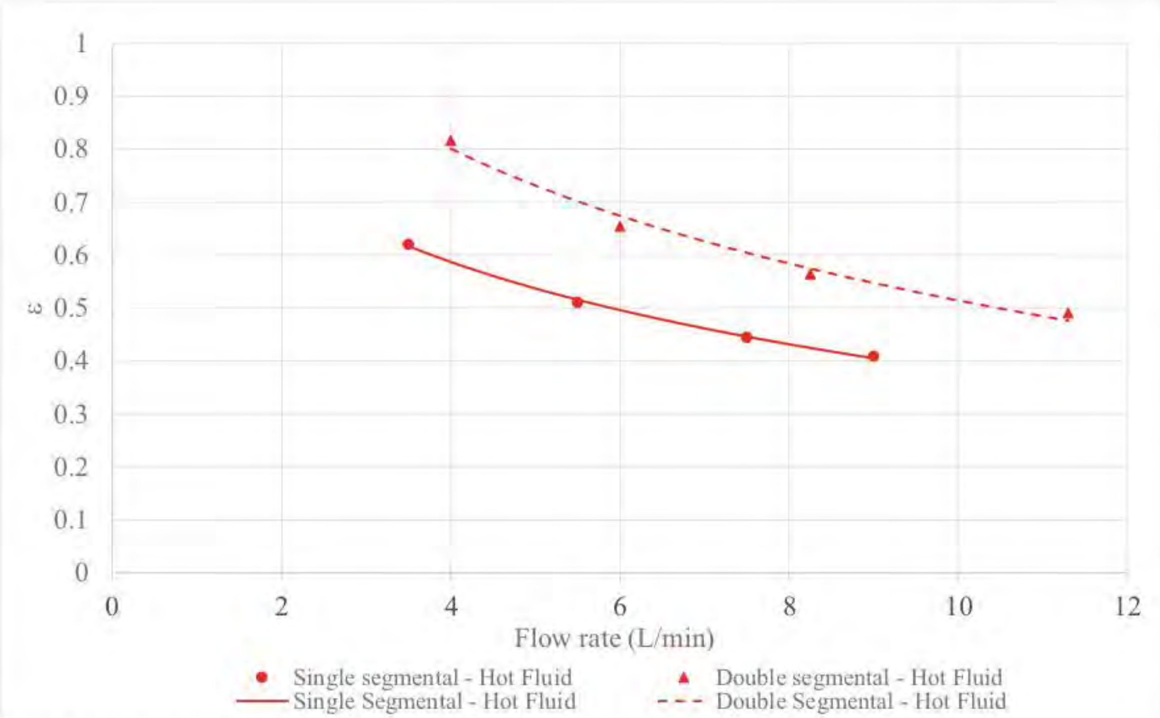

Fig. 11 Relationship between effectiveness and flow rate (Hot Fluid)

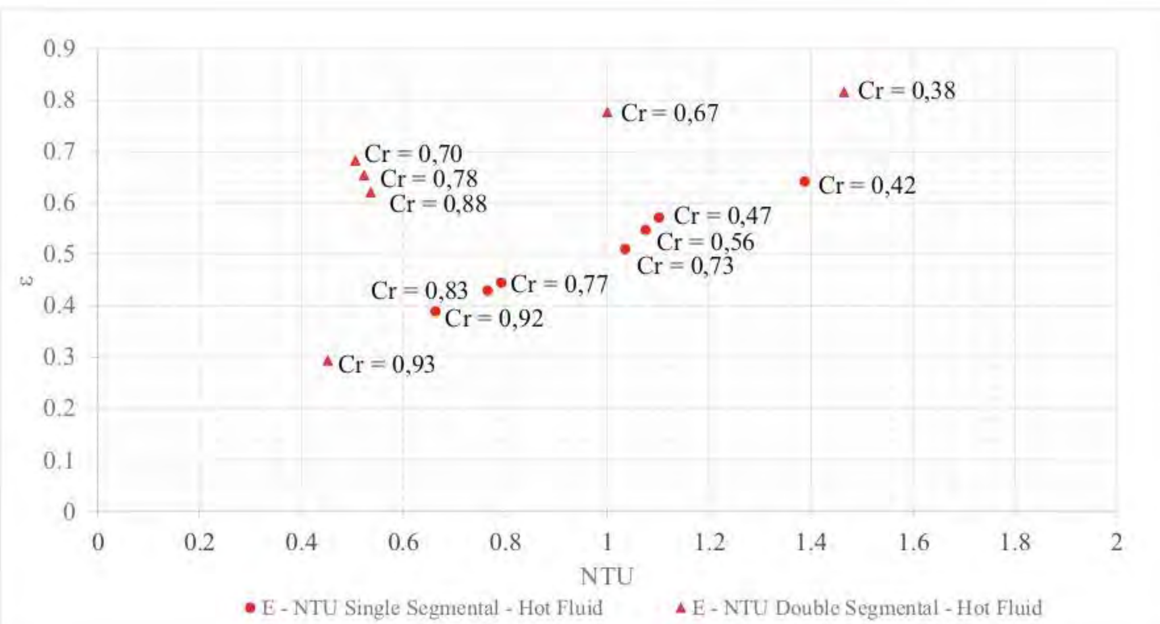

Fig. 12 Relationship between effectiveness and NTU

efficiency than the single segmental baffles for high flow rate. Effectivity of the single segmental increases as the flow rate decreases, and the effectivity of the double segmental baffles decreases as the flow rate decreases. This phenomenon might be caused by the effect of the design and geometry of stationary-head that are affected by the Reynolds number. Alternative approach that can be used is the calculation using ε - NTU (effectiveness - Number Transfer Unit) method. The capacity ratio (Cr) of the double segmental baffles and single segmental baffles, can be seen at the graph relationship between effectiveness and NTU.

Conclusions

In this research, the effects of single segmental baffles and double segmental baffles on the thermal effectiveness have been investigated using experimental method. Based on the results, the double segmental baffles have better effectivity than the single segmental baffles, and the average effectiveness of single segmental baffles is 10 to 30%, so it is less than double segmental baffles.

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Nomenclature Nu_{tube}, Nusselt Number of tube; Nu_{shell}, Nusselt Number of shell; $d_{in\,Tube}$, Inner diameter of tube; $d_{Out\ Tube}$, Outer diameter of tube; L_{Tube}, Length of tube; N_{Tube}, Number of tube; Re_{Tube}, Reynolds number of tube; $Re_{Shell\ Max}$, Reynolds number Max of shell; Pr_{Tube}, Prandtl number of tube; Pr_{Shell}, Prandtl number of shell; Pr_{Wall}, Prandtl number of wall; C_h, Specific heat capacity of hot fluid; C_c, Specific heat capacity of cold fluid; C_{Min}, Minimum heat capacity rate; C_{Max}, Maximum heat capacity rate; C_R, Heat capacity rate ratio; $\dot{m}$, Mass flow rate; $\dot{Q}$, Actual heat transfer rate; $\dot{Q}_{Max}$, Maximum possible heat transfer rate; U_{Total}, Total of heat transfer coefficient; A, Area; A_{Total}, Total area; $A_{In\ Tube}$, Area of inner tube; $A_{Out\ Tube}$, Area of outer tube; R_{th}, Thermal energy total; $R_{Din\ Tube}$, Thermal energy of inner tube; $R_{DOut\ Tube}$, Thermal energy of outer tube; NTU, Number Transfer Unit; Th_{in}, Temperature of inlet hot fluid; Th_{out}, Temperature of outlet hot fluid; Tc_{in}, Temperature of inlet cold fluid; Tc_{out}, Temperature of outlet cold fluid; h_h, Heat transfer coefficient of hot fluid; h_c, Heat transfer coefficient of cold fluid; k, Thermal conductivity; h_{In}, Heat transfer coefficient of inlet fluid; h_{Out}, Heat transfer coefficient of outlet fluid; D_{In}, Inner diameter; D_{Out}, Outer diameter; ε, Effectiveness

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