Computational Fluid Dynamics of Change Input Parameters on the Effectiveness of Heat Transfer for Gasoline Vapor Control Unit

Nattadon Pannucharoenwong*, Phadungsak Rattanadecho, Snunkhaem Echaroj, Suwipong Hemathulin and Nattha Kanlayaprasit

Abstract— *An increase in transportation of petroleum related commodity harm the environment due to the release of low boiling point vapors in order to regulate pressure built-up inside the transportation vessel. This research presented the analysis of computational fluid dynamic regarding the influence of input parameters on the efficiency of heat transfer in heat exchangers (HTX) with Two-pass. The input parameters included different cold-water flow rate ranges from 4 to 10 kg/s, hot water from 4 to 40 kg/s and outlet temperature of hot water from 310 K to 410 K. The response observed from these changes in input parameters are the outlet temperature of cold water and pressure drop inside the heat exchanger (HTX). Results indicated that an increase in water's flow rate (cold channel) and hot water's temperature caused the heat transfer to increase. Additionally, an incline in water's flow rate (cold channel) resulted in larger pressure drop. From graphical analysis it was revealed that when the outlet temperature of cold water was 360 K and pressure drop decreased, the flow rate was approximately 7.5 kg/s, Δp was 0.9 kPa and rate of heat transfer 423.4 kW effectiveness 22.5%.*

Keywords— **Heat Exchange, volatility reduction unit, computational fluid dynamics.**

1. INTRODUCTION

The rapid growth and development of the petroleum industry in Thailand is the result of the development and expansion of the country's economy and in response to the increasing energy demand in the country. In addition to the increase in production capacity, transportation of petroleum efficiently and safely is important an important issue for the entrepreneurs. During the transportation of petroleum commodities fuel evaporation, which causes high pressure inside the system, must be released from the system in order to reduce potential explosion. The fuel vapors emitted is harmful to organism and cause air pollution [1]. For this reason, the operator must control the amount of toxins in accordance with the law [2]-[4]. This requires the installation of a fuel control unit (Vapor Recovery Unit; VRU). In the fuel supply system, a conventional filtration method employed to trap oil mist is made of high surface area carbon material. This is disadvantageous because carbon material cannot be re-used. Most entrepreneurs want to use the trap system for capturing fugitive pentane which usually evaporate at 303.15 K [5]. In order for the emitted vapor to be condensed back to liquid form the system must be cooled to a temperature lower than the evaporation of pentane. Such a system required on-site installation and imported technology which is extremely expensive. An example of this type of technology can be found in one of PTT gasoline station. However, the station has difficulty in maintenance when system crashes which caused discontinuity in the operation of the entire system. For this reason, PTT. Ltd (Company) Bangkok developed an idea to create a device that could replace the expensive cooling system with a domestic technology. The system needs to perform as well as the imported technology. The principle of shell and tube-heat exchangers (HTX) was integrated in the VRU system in order to reduce the construction cost. The operator has proposed a joint research and prototype build up. The layout of the developed VRU system is shown in Figure. 1. Initially, the steam condenses resulting in pressure and temperature drop and then flow to the next system. An increase in temperature deviation was the main result of turbulent behavior in the system.

The shell and tube type HTXs are referred as a device where the fluid flow in tubular pipeline and another type of fluid in the shell section which is often divided and attached to uni-directional baffles in order to prevent fluid from flowing into the wrong section, change the direction of fluid flow and assure appropriate distance between the pipes causing turbulent flow to maximize heat transfer. [6], [7]. Variables included for analyzing the heat transfer process, included mass flow rate determined by measuring the fluid velocity changes in kinetic energy. This depends on the pressure difference that forced the liquid through the tube. For this reason, pressure drop (Δp), resulted from an increase in friction of the system, is also an important parameter. Friction depended mainly on the speed of the fluid in the pipe line, pipe length, pipe size and the texture of the pipe (smooth or rough). If the fluid flow at high speed, friction increased which resulted in reduction in pressure. The friction created between the layer of fluid and the

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pipe wall resulted in total energy loss and ultimately ΔP of the whole system [8]. Therefore, HTX should have high flow rate in the shell section, there should be no fouling, low pressure system and high thermal efficiency. Reduction in the pressure and the thermal efficiency will depend on the pattern of the material flow and baffle [9]. The experimental data obtained from monitoring ΔP in the shell-tube section of the HTX with co-current (parallel) flow and counter-current flow with equal dimension and flow rate revealed counter-current type of HTX to have more thermodynamic characteristics than the co-current type. This is due to the superior flow management efficiency resulting in high temperature deviation in specific area inside the HTX. This is also illustrated by setting the overall thermal conductivity (UA), material flow rate, heat capacity and fluid temperature at the entrance (T inlet) of the HTX. In addition, the use of helical baffles proves that the HTX efficiency is better than segmental baffles given the same dimension and mass flow rate [10]-[12].

The study of different helical baffles angles through a periodic model revealed that the efficiency of the HTX depended on the pressure and angle of the baffle (helix angle). The most effective angle was measure at approximately 45◦ [13], [14]. Flow experiment by changing the nature of the baffle revealed segmental baffle to demonstrate dead zone problems due to high pressure drop causing the pump to work at high load. This problem can be solved by switching from segmental baffle to helical type baffle in order to reduce pressure drop [15].

Fluid dynamics analysis in large industries is a time-consuming and costly task. Another interesting alternative method is to create a small prototype or model in computer program by setting the boundary conditions. This method is usually referred to as the Computational Fluid Dynamics (CFD) and it is used to analyze or predict the flow of fluid flowing through the object of interest which is broking into smaller volume mesh to improve reliability of the model and integration of the Navier – Stokes equations. Calculation of net flows from one mesh to another one is estimated. Accuracy of the analysis depend on how user divide the mesh, set configuration and performance of the computer-based calculations.

Variety model with different size is available as HTXs, depending on the needs of various industries. Whether to be used for increasing heat reduction, fluid condensation, and evaporation of fluid. Within the current HTX, sometimes it is employed to contain the heat to improve the heat transfer efficiency, such as using segmented baffles. This will help increase the heat transfer coefficient due to cross flow resulting in more turbulent flow [16]. The use of helical baffle helps to cause turbulence by causing the value of the pressure drop to be less. In the literature review the research team studied by dividing the literature into 3 types as follows.

1.1 Literature reviews related to condensation and oil vapor

Li Shi and Weiqiu Huang [17] conducted using the method condensation separation method with condensation, oil vapor by means of cooling steam 4 types S1, S2, S3 and S4 to pass by condensation three stage where the temperature 274 K, 313 K and 163 K. Result has effectively condensed oil vapor of each species were 99.73%, 99.79%, 99.82% and 99.19% respectively, and the amount of fuel vapors out of 2.87 g/m^3, 2.75 g/m^3, 3.04 g/m^3 and 16.98 g/m^3, respectively, and have the fuel vapors and 4 to compare with the condensed first step, with the result that condensation with 3 steps, saving effect of more than 12.23%, 15.68%, 13.96% and 15.65. % respectively.

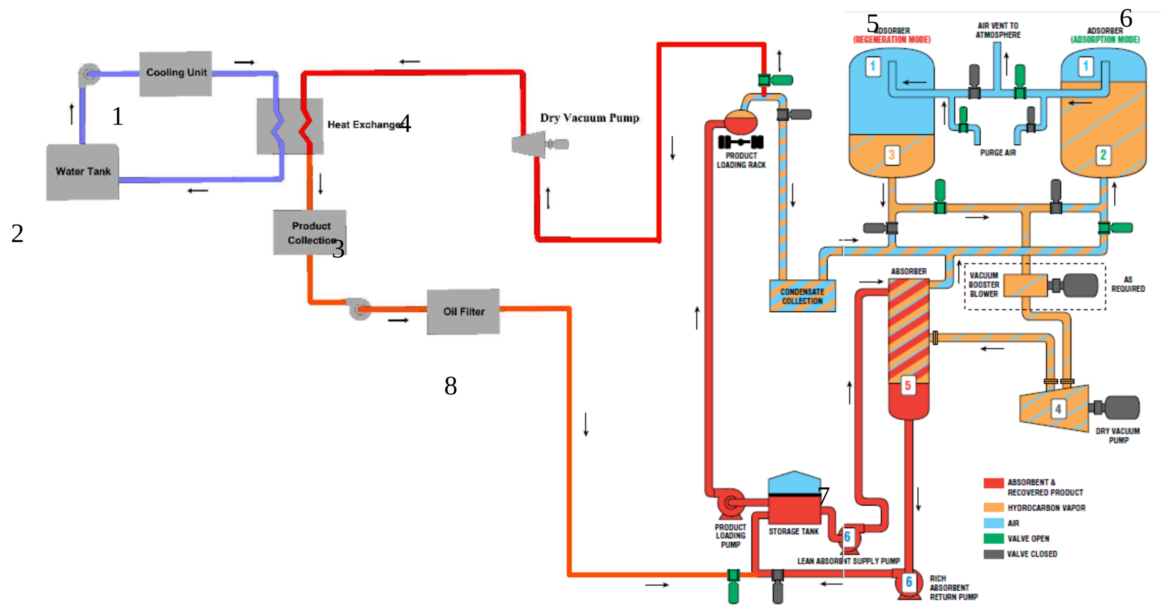


Fig. 1. Schematic of system prototypes with alternative systems VRU and Control volume; 1. Cooling unit, 2. Water tank, 3. Product collection, 4. Heat exchanger, 5. Absorber (Regeneration mode), 6. Absorber (Absorber mode), 7. Storage tank, 8. Oil filter, and 9. Dry vacuum pump.

1.2 Literature related to the HTX.

Yonghua You et al. [18] studied heat and flow efficiency in a hell-and-tube (ST) HTX containing Trefoil-holes baffles (THB-STHXs) in turbulent flow condition. Test data demonstrated that the heat transfer rate is increased efficiency in the shell. The flow resistance increased significantly additionally, heat transfer performance and ΔP increase as Reynolds number of the system increases.

Bin Gao et al. [19] studied the backward flow and heat transfer occurring within the HTX type. Shell-and-tube (ST). Screw discontinuous barrier (Discontinuous helical baffles), by comparing the angle of baffles 8, 12, 20, 30 and 40 degrees respectively angle, the effective maximum is 40 degree of heat was the result of estimation entrants dissipation of entropy generation, and shall have the same conditions as the HTX of this kind would be more productive if the Reynolds number is less in the shell site.

Guo-Yan Zhou et al. [20] lead CFD analysis the ST HTX system with trefoil-hole baffles, which results from a simulation that shows the changes of the fluid flowing through the trefoil-hole baffles from first. The cost to heat and pressure changes periodically axial velocity of the fluid will increase and a higher speed (Jet flow) in close baffles and slower (Secondary flow) when away. The two runs that resulted in an optimized heat transfer.

Jian Wen et al. [21] compared the distribution of the speed and temperature of the shell of the HTX ladder fold baffle with a helical baffle the heat transfer coefficient (α_0) and overall heat transfer coefficient (K) increased from 22.3 to 32.6% and from 18.1 to 22.5%, respectively, with the pressure drop at 0.91-9.08 kPa while pumping power is 2-80 W and the thermal efficiency factor. (TEF) is 18.6 to 23.2%.

Aniket Shrikant et al. [22] studied the heat transfer and pressure drop within the HTX type. Shell-and-tube (ST) Many different type of baffles is available such as single, double, triple segmental baffles, flower baffle and helical baffles simulation computer program Solidworks. Simulated flow results, the size of the shell site flow direction as it turns out that type. The single highest heat transfers while Stagnation zone will appear in the type helical baffles and flower baffle would result in a reduction of stagnation zone.

Anas El Maakoul et al. [23] compared the distribution of the flow heat transfer coefficient and the ΔP in the shell of a HTX helical, trefoil-hole, and the conventional baffles by the numerical model with segmental baffles as a comparison with other finds helical baffles with the thermo-hydraulic efficiency but trefoil-hole with the heat transfer capability with large ΔP compared with normal segmental baffles

Mohammed Irshad et al. [24] compared the slope of segmental baffles in ST HTX the slope ranges from 0° - 40° found that blood pressure was reduced by an average of 6% on the slope increases and found that baffle cut 25% slope of 40 ° with performance is better than 0°, 20° and 30°.

Pranita B et al. [25] studied the comparison of the pressure drop in segmental, double segmental and helical baffles the transition from the $\dot{m}$ 0.0104-0.032 kg / s that the $\dot{m}$ = 0.032 kg / s in double segmental baffles and segmental born dead zone caused the pressure drop at 239.54 and 196.20. Pa, respectively, and the helical baffles to reduce the pressure of 127.21 Pa.

1.3 Literature associated with finding the right balance.

Jian Wen et al. [26] created a process for mix model via Kriging response surface and multi-objective genetic algorithm (MOGA) to determine the most appropriate heat transfer configuration. Optimization of STHXsHB by adjusting the angle of the baffles (β) were performed. The proportion of overlap and flow into the differential volume V is considered a parameter for the optimization of the design. The study concluded that the usage of helical baffles in STHXsHB can give better results in terms of efficiency and cost reduction.

Literature review above regarding changes in the input parameters demonstrated how output are significantly affected. But these studies were conducted with different types of HTXs. It is impossible to determine the effect of changing the parameters of the HTX, two of which were made to the design. The researchers want to know how the variable inlet of the HTX was designed to replace the master control unit, fuel vapor will affect the effectiveness of the HTX and variable output.

In this study, analysis of heat transfer behavior and the total heat transfer coefficient were performed. The STHTX with a plate holding a continuous spiral were simulated and validated by changing the pitch and rate of fluid flow for heat and cold channel inside the HTX, which was designed to replace the fuel vapor control unit as shown in the figures below. Design parameters complied with the software packages Workbench 17.2 according to the design standards [27], [28]. Due to confidential issue PTT. Ltd (Company) decided not to release some of the important design conditions. The company will disclose these parameters when the project has been approved as a prototype from trial experiments. Researchers need to use cold and hot water for use in this study represented a benzene oil for quick analysis.

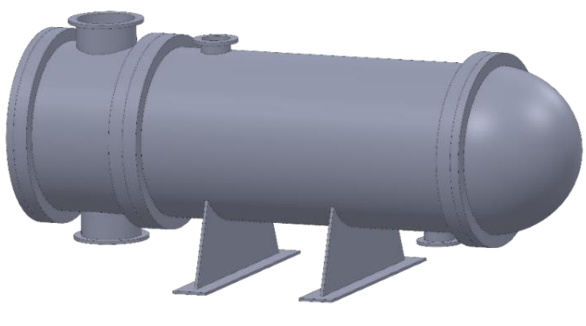


Fig. 2. ST HTX with a plate holding a continuous spiral used in the analysis.

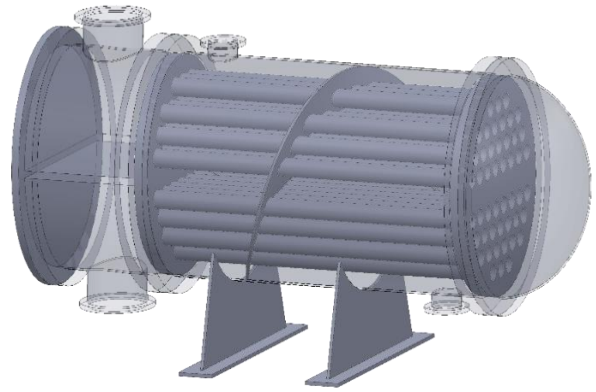


Fig. 3. Analysis within the shell and tube HTX with a plate holding and a continuous spiral.

1.4 Advantages of Helical Baffle type HTX

Helical baffle is a novel kind of baffle used to overcome the problems of HTXs. The first group of scientists to developed a helical baffle was 1990 [29]. In this research the pattern of flow was produced and observed for different shape of helical baffle and also with different angles. A plug flow type of flow was observed fort helical baffle HTX, due to a reduction in pressure drop for the shell-side and an enhancement of heat transfer.

Stehlik et al. [30] employed the Bell-Delaware technique to study ΔP and heat transfer correction for optimization of HTX with segmental baffle and helical baffle. Kral et al. [31] reported the efficiency of HTXs containing difference shape of helical baffles. It was realized that the helix angle is a significant parameter leading to an effective STHXHB.Currently, test data obtained in term of shell-side heat transfer coefficient compared with shell-side pressure drop demonstrated 1 segmental and 5 helical baffles from an oil-water HTX. Experiment shown that the pumping power (unit shell-side fluid pressured drop) can compromise the performance of the HTX even with a helical baffle. A continuous helical baffle demonstrated an increase in heat transfer coefficient per unit ΔP in the HTX.

Table 1 Pentenes Properties and Identifiers.

Properties	
Chem. formula	C_5H_{10}
MW	70.135 $g \cdot mol^{-1}$
Density	0.64 g/cm^3
Melting point	-165.2 ° C
Boiling point	30 ° C
Magnetic susceptibility (X)	-53.7 × 10-6 cm^3/mol

It is important to design a continuous helical baffle in order to avoid vibration from turbulent flow and reduce fouling. An incline of 10% in heat transfer coefficient was reported when a helical baffle (STHXs) was used instead of the normal segmental baffles under the same magnitude of ΔP. Additionally, data monitored revealed various non-dimensional correlations for pressure drop and heat transfer coefficient. These data can be used to improve the application of continuous helical baffle HTX in the industry.

2. ENERGY AND EXERGY ANALYSES

2.1 Algorithm model

This research employed the Ansys Fluent v14.0 program for simulation of the corrugated the shell and tube HTX [34], [35]. Experimental data revealed that a turbulence flow behavior is possible even for a flow with Reynold number of about 400 given the chevron-angle (φ) of 60º. Additionally, Gherasim et al. [36] compared data of heat transfers monitored from the shell and tube HTX with CFD simulation results. It was also reported that a turbulence flow can be achieve at Reynolds number of 400. For this reason, this research proposed that the Reynolds number shown be larger than 400. However, when the flow is in laminar mode the Reynold used was below 400, according to equation (1) - (5).

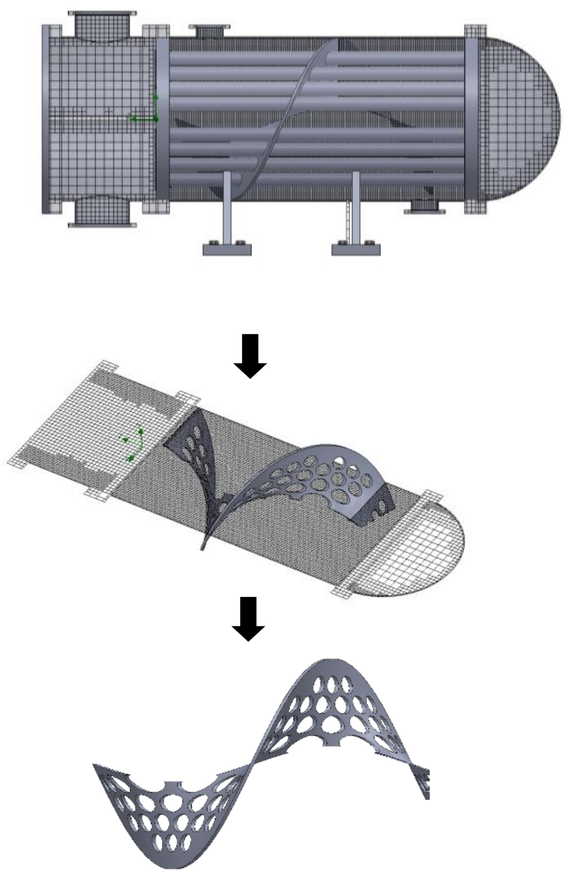


Fig. 4. The geometry and generated grid.

Moreover, a non-equilibrium wall function in a realizable k-ε model was proposed by various researches regarding flow pattern in HTX [37,38]. Even though this model employed variable Cμ, the model does not produce negative normal stress in case of large strain.

For this reason, the proposed model should offer better efficiency in term of flows including separation and spinning mode. Additionally, this model entailed a modified ε equation that demonstrate a reliable simulation of both planar and axisymmetric jets. After calculation and prediction of the complicated flows including boundary layer, separated and rotating shear types, it was demonstrated that the modified k-ε model gave better results than the standard k-ε model. For this reason, in this research the modified k-ε model was employed for simulation of turbulence flow behavior. Equation used for incompressible flow in turbulence mode including mass, momentum and energy balance are illustrated below.

$$\frac{\partial\left(\rho\overline{u_i}\right)}{\partial x_i}=0 \tag{1}$$

$$u_j\frac{\partial\left(\rho\overline{u_i}\right)}{\partial x_i}=\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_i}\left(\left(\mu+\mu_t\right).\frac{\rho\overline{u_i}}{\partial x_i}\right) \tag{2}$$

$$C_p.u_j\frac{\partial\left(p.T\right)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\left(k+k_t\right)\frac{\partial T}{\partial x_j}\right) \tag{3}$$

Kinetic equation terminology of the turbulent flow κ and its dissipation rate ε with the following equation.

$$u_j\frac{\partial\left(p.\kappa\right)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\left(\mu+\frac{\mu_t}{\sigma_\kappa}\right).\left(\frac{\partial\kappa}{\partial x_j}\right)\right)+G_\kappa+G_b-\rho.\varepsilon-Y_M+S_\kappa \tag{4}$$

$$u_j\frac{\partial\left(p.\varepsilon\right)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right).\left(\frac{\partial\varepsilon}{\partial x_j}\right)\right)+pC_1S_\varepsilon+G_b-\rho.C_2.\frac{\varepsilon^2}{\kappa+\sqrt{\nu.\varepsilon}} \tag{5}$$

The assumptions used in the analysis grid. 1

Table 2 Terms used in the analysis of HTX.

Order	Hypothesis
1	Analysis is stabilized (Steady State)
2	To qualify for the analysis. Lahore Fluid temperature properties of the water. The fluid properties of high temperature steam.
3	Analysis is 3-D
4	Analysis by a combined force of gravity.
5	Viscosity model equation. Standard k-epsilon (2 eqn).
6	Equation Pressure-Velocity Coupling A SIMPLE Scheme.

2.1.1 Definition of meshes and geometry of the tube

The TEMA standards were employed to design ST HTX [39] as shown in the tube layout provided in Fig. 5.

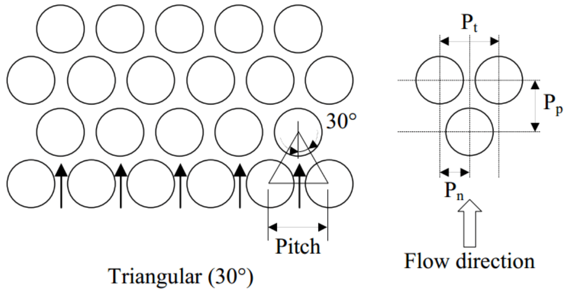


Fig. 5. Tubular dimension coding and arrangment including tube parallel pitches and tube normal to flow (Pt, Pp and Pn) is typically shown for equilateral triangular layout.

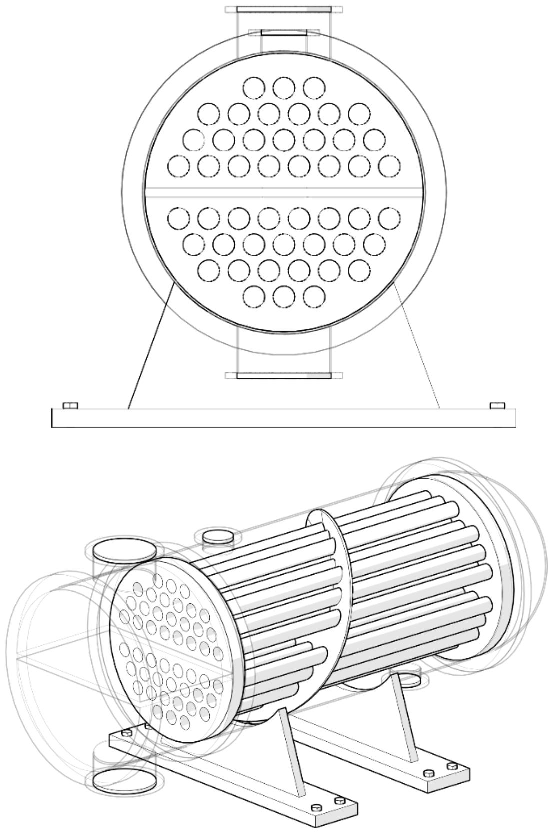
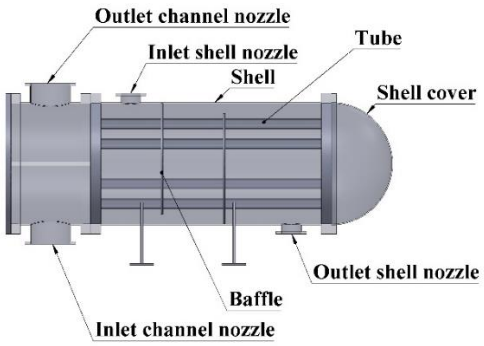


Fig. 6. Configuration of baffles and tubes in isometric view with baffle inclination.

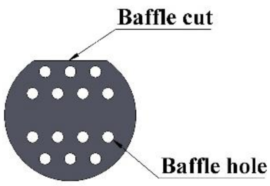
Table 2 ST HTX dimension for simulation

Length of HTX, L	600 mm
Inner Diameter of the Shell, Di	90 mm
Outer Diameter of tube, do	20 mm
Tube Bundle Geometry and Pitch Triangular	30 mm
# of Tubes, Nt	7
# of Baffles, Nb	6
Central Baffle Spacing, B	86 mm
Baffle Inclination Angle, θ	0 to 40

The configuration of the HTX stated in Table 2 is shown in Figure 7 below. The heat exchange is composited of channel inlet and outlet nozzle, shell nozzle (inlet and outlet), shells, tubes and a shell cover as shown in Figure 7a. Figure 7b depicted the circular plate or the baffle used to hold the tubes in their places.



(a)



(b)

Fig. 7. Typical parts of HTX designed (a) and the baffle (b).

2.1.2 Mesh formation

The next step is the discretization of the three-dimensional model in ICEM CFD. This is because it is important to demonstrate velocity and thermal boundary layers for the whole model. A hexahedral mesh unit was employed because it is more precise and took lower computation time. Tuning of the wall surface in the hexahedral mesh provide an accurate boundary layer. The whole three-dimensional model is separated into three fluid domains including Fluid_Inlet, Fluid_Shell and Fluid Outlet. Additionally, the model also included 6 solid domains (Solid_Baffle 1 to Solid_Baffle 6) for six baffles respectively.

In order to enhance the stability of the algorithm the mesh HTX is separated into solid and fluid domains with multiple nodes. An increase in discretization of the fluid mesh resulted in a finer simulation node which demonstrate a more accurate heat transfer phenomenon. Figure 8 illustrated the 3 different fluid domains included in the model. The primary height of the cell in the fluid domain is controlled at 100 microns in order acquire the thermal and velocity boundary layers. Validation of the discretized model revealed the minimum angle of 18 ° and min determinant of 4.12. After the discretized meshes are error less and reached a minimum required quality, the model is then sent to the pre-processor program (ANSYS CFX).

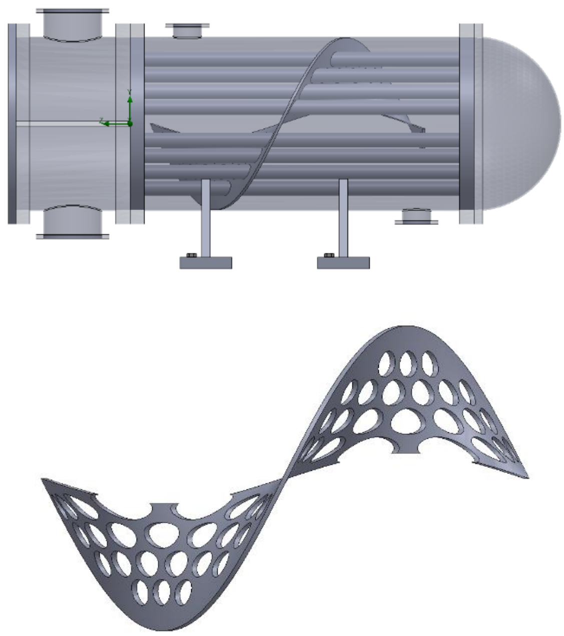


Fig. 8. The entire model of baffle used in the HTX simulation.

2.1.3 Discretization

At first the model contained approximately 1.8 million mixed mesh nodes with both triangular and quadrilateral area at the outer layer. Next the hexahedral structured cell is separated into many different parts via the automatic function in ANSYS software. In order to decrease the numerical diffusion by remodelling the mesh generated with emphasis on the, location near the wall region. Next, a mesh with 5.65 million cells was created from the previous set of meshes. This resulted in a finer mesh and a more accurate edges especially in the high temperature area at different pressure gradients.The discretization process is illustrated in Figure 9, Figure 10 and Figure 11.

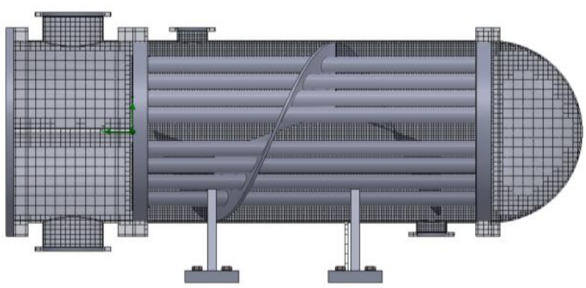


Fig. 9. Meshing illustration of shell and tube HTX.

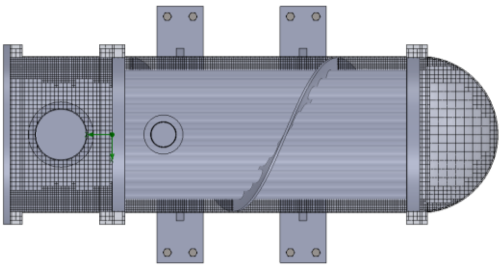


Fig. 10. HTX containing helical baffle, tube bundles and surface meshes.

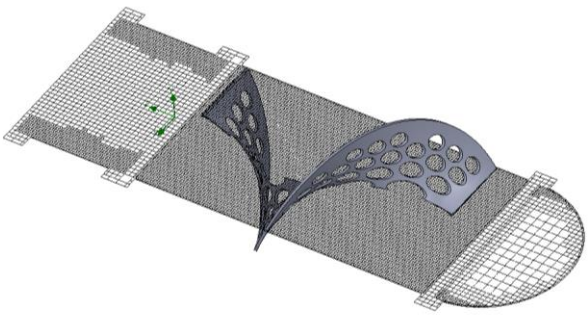


Fig. 11. Surface mesh with Helical Baffle.

2.2 Energy Assessment

The analysis for the ability to exchange HTXs heat stabilized. (steady state) [32], [33] Can be analyzed from the equation 6 to Equation 11 Calculate pressure drop (Pressure Drop; ΔP) From Equation 6 on that note, Coefficient of Friction (f) The length (L) The average speed of the flow (V) And the diameter of the pipe (di)

$$\frac{\Delta p}{\rho} = f\,\frac{L}{d_i}\,\frac{V^2}{2} \tag{6}$$

From Equation 1 Find the average speed of the flow. When the flow rate ($\dot{m}$) density (ρ) And the inside diameter of the pipe (di)

$$V = \frac{\dot{m}}{\frac{\pi}{4}\rho d_i^{\,2}} \tag{7}$$

The rate of heat exchange (Rate of net heat transfer).Obtained from the equation 8 on that note, The rate of heat exchange ($\dot{Q}$) Flow rates ($\dot{m}$) Specific heat capacity (c_P)Latent Heat (L) The total heat transfer coefficient (U_o) And log mean temperature difference (ΔT_{lm})

$$\dot{Q} = \dot{m}\,c_p\Delta T + \dot{m}\,L = AU_0\Delta T_{lm} \tag{8}$$

by log mean temperature differenceObtained from the equation 9

$$\begin{aligned} \Delta T_1 &= T_{hot,in} - T_{cold,out} \\ \Delta T_2 &= T_{hot,out} - T_{cold,in} \\ \Delta T_{lm} &= \frac{(\Delta T_1 - \Delta T_2)}{\ln\frac{\Delta T_1}{\Delta T_2}} \end{aligned} \tag{9}$$

The effectiveness of HTXs can be obtained from the equation 10

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{10}$$

when $\dot{Q}$ is refered to as heat transfer rate of the HTX. $\dot{Q}_{max}$ is the rate of heat transfers of HTX.

$$\dot{Q}_{max} = C_{min}\left(T_{hot,in} - T_{cold,in}\right) \tag{11}$$

when $C_{min} = \dot{m}\,c_p$ is less than In the hot and cold water

3. 3. EXPERIMENTAL SET-UP AND MEASUREMENTS

Simulation was carried out in Solidworks (Flow Simulation). The popular Navier-Stokes equations were computed by the Solidworks program simulation by creating mass, momentum and energy conservation laws for fluid flows. The simulation technique was performed in steady state condition. The energy calculation performed during simulation given the viscosity of the fluid, standard k_e, standard wall function (k-epsilon as shown in 2 eqn.). Water-liquid mode was employed in the cell zone oft he simulated HTX. Copper and aluminum was chosen in the simulated model where the inlet and outlet boundary condition were given.

In 0.25 kg/s mass flow inlet was set at 353 K of temperature. Parameters used in the simulation included 1 m^2, density equaled to 998 kg/m^3, enthalpy of 229,485 J/Kg, length 1 m, temperature of 353 K, velocity of 1.44 m/s, ratio of specific heat 1.4 was considered for the analysis of two sections: variation in pitch and flow rate of the mass of water as shown in Table 3.

The model used in the analysis is the simulation of a tube and shell HTXs. The inner containment shell plate holding screw with a long pitch. p = 950 mm, the total length of the plate, hold the equivalent Lbaffle 950 mm diameter and 600 mm D is the model used in the analysis is shown in Figure 12 and Figure 13.

Table 3 The symbols and terms used in the analysis

Order	Symbol	Condition
1	P800 m0.25	Long-pitch equals 800 mm flow of cold water equivalent of 0.25 kg/s.
2	P850 m0.25	Long-pitch equals 850 mm flow of cold water equivalent of 0.25 kg/s.
3	P900 m0.25	Long-pitch equals 900 mm flow of cold water equivalent of 0.25 kg/s.
4	P950 m0.25	Long-pitch equals 950 mm flow of cold water equivalent of 0.25 kg/s.
5	P950 m0.50	Long-pitch equals 950 mm flow of cold water equivalent of 0.50 kg/s.
6	P950 m1.00	Long-pitch equals 950 mm flow of cold water equivalent of 1.00 kg/s.
7	P950 m2.00	Long-pitch equals 950 mm flow of cold water equivalent of 2.00 kg/s.
8	P950 m5.00	Long-pitch equals 950 mm flow of cold water equivalent of 5.00 kg/s.

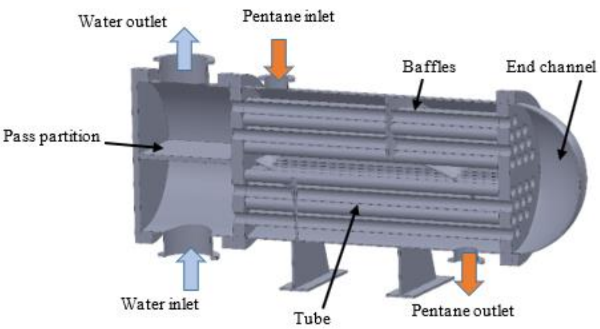


Fig. 12. Transverse cross-sectional model of the HTX.

Fig. 13. Incontinence pads Screw (Helical Baffle).

The study is divided into 2 parts: the first part was to observe the effect of the change in pitch affect the performance of the HTX shell and tube. The second part was to study the impact of changing the water's flow rate that affects the performance of the HTX. Let the oil vapor heat a fluid (hot fluid), the rate of fluid flow, heat mhot equal to 0.25 kg/s flow into the shell's head. And flows out the shell's bottom. The cooling fluid is water (cold fluid), which has a flow rate equal to the initial mcold 0.25 kg / s flow into the header into the bottom and flows to the header at the top. Containment shell inside pipe diameter is equal to 1-inch long pipe with a 48-inch long tube.

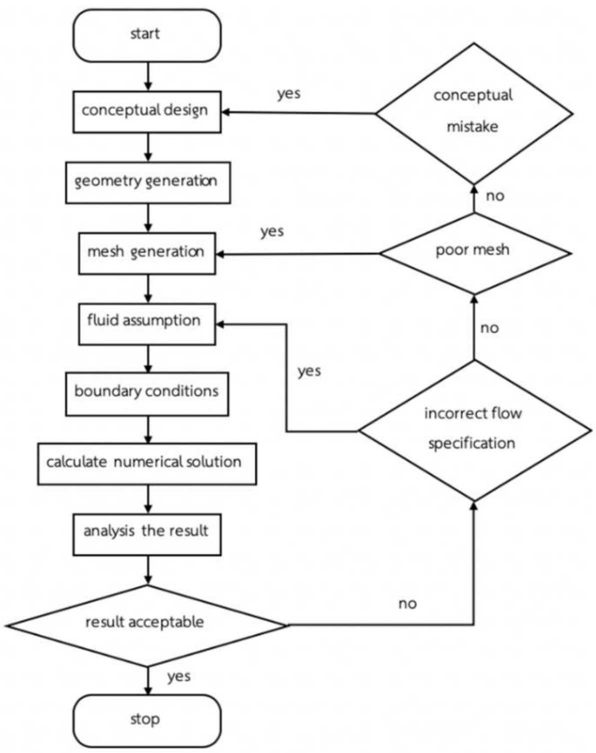


Fig. 14. Map conducting studies.

4. RESULTS AND DISCUSSION

The analysis by CFD in the comparison of the effect of the change in pitch is shown in Figure 15. The inlet hot water's temperature and cold water as well. Long-pitch equal to 950 mm, the water's flow rate was 0.25 kg/s with a hot water outgoing temperature and cold water as low as possible.

Figure 16 demonstrated resulted observed after variation in pitch 800, 850, 900 and 950 respectively. At the pitch of 950 mm, the water's flow rate was 0.25 kg/s with an Enthalpy lowest 361,417.73W rest will be Enthalpy. equally 361,438.68W.

Figure 17 demonstrated the calculated heat transfer coefficient on the change in Overall pitch. Found at pitches of 850 mm, the water's flow rate was 0.25 kg/s is the overall heat transfer coefficient most 77.63 W / m^{2}K and a long pitch of 950mm flow of cold-water equivalent of 0.25 kg/s is the lowest overall heat transfer coefficient equal to 48.39 W/m^{2}K.

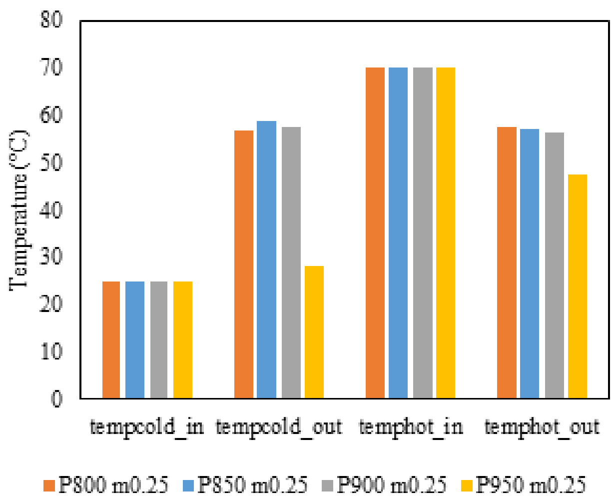


Fig. 15. The temperature of the fluid at each position on the change in pitch.

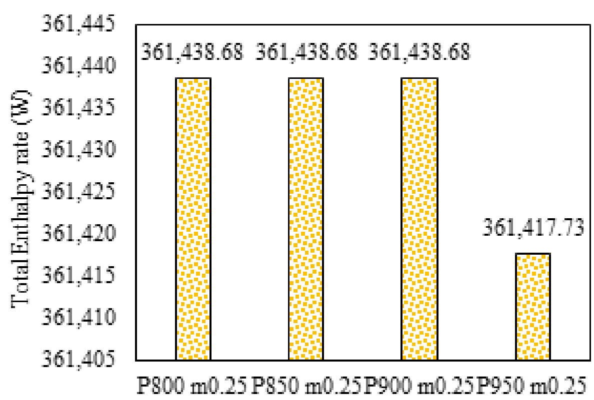


Fig. 16. Enthalpy of fluid in each of the models on the change in pitch.

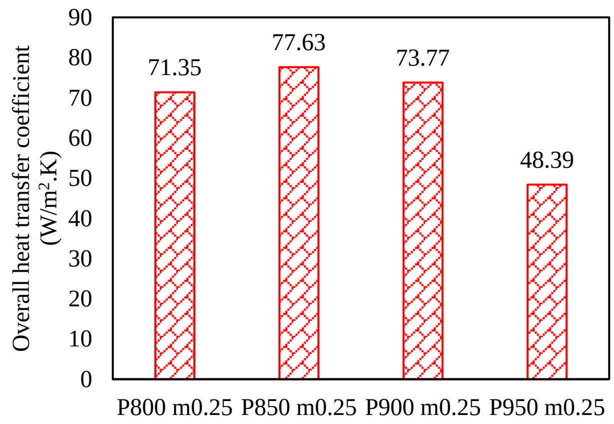


Fig. 17. Effect of changes in pitch on the overall heat transfer coefficient of the HTX on the.

Figure 18 shows the Effectiveness the HTX on the change in pitch. Found at pitches of 850 mm, the water's flow rate was 0.25 kg/s valuable Effectiveness. And at most 1.16 Pitch 950 mm equivalent water's flow rate was 0.25 kg/s with the lowest effectiveness 0.54.

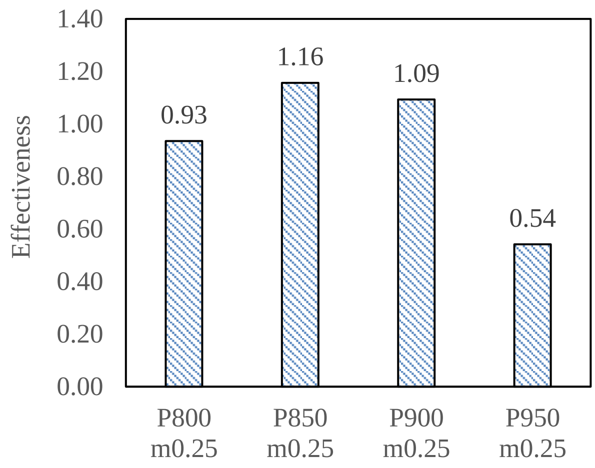


Fig. 18. Cost effectiveness The HTX on the change in pitch.

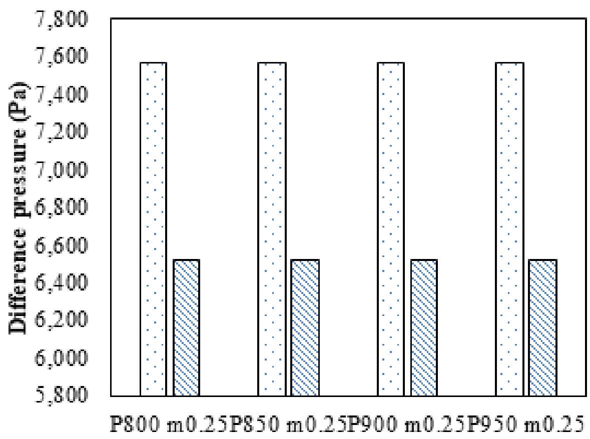


Fig. 19. Effect of pitch variation due to changes in pressure difference.

Figure 19 shows the pressure difference of the material effected by the variation in pitches. Results have shown that the variation in pitch do not affect the pressure difference in the HTX. Then, in second part 2 a comparison of performance caused by the effect of changing the mass flow rate of water which can be shown in Figure 20 and

Figure 20 demonstrated the temperature of the fluid mass flow rate of the water. Found that the cold water's flow rate is equivalent of 0.25 kg/s and 2 kg/s, temperatures of fluid out below. The flow of water is equaled to 0.5 kg/s and 1 kg/s.

Figure 21 illustrated the enthalpy of the fluid mass flow rate of the incoming water. Once found, the flow rate Enthalpy of fluid does not change. Thus, the flow rate did not affect the Enthalpy.

Figure 22 demonstrated the overall heat transfer ecoefficiency on the mass flow rate of incoming water is found at 2 kg/s is the overall heat transfer coefficient as possible so that the flow rate of 2 kg/s were exchanged best hot.

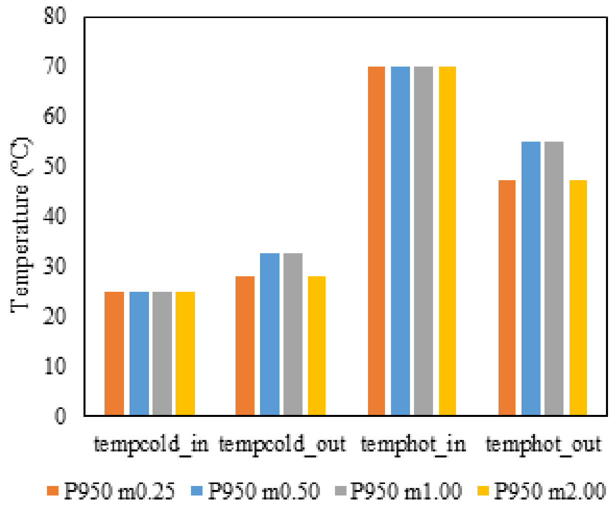


Fig. 20. The temperature of the fluid at each position when changing the mass water's flow rate.

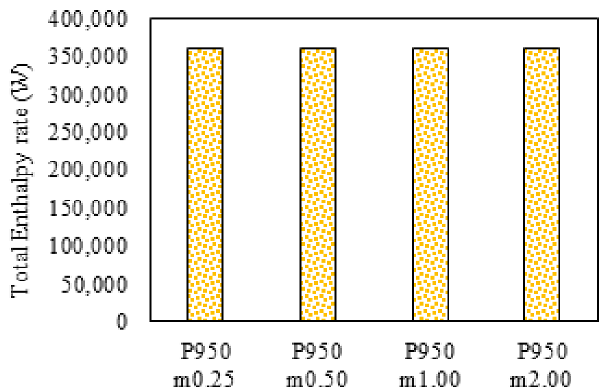


Fig. 21. Enthalpy of fluid in each model when changing the water's flow rate.

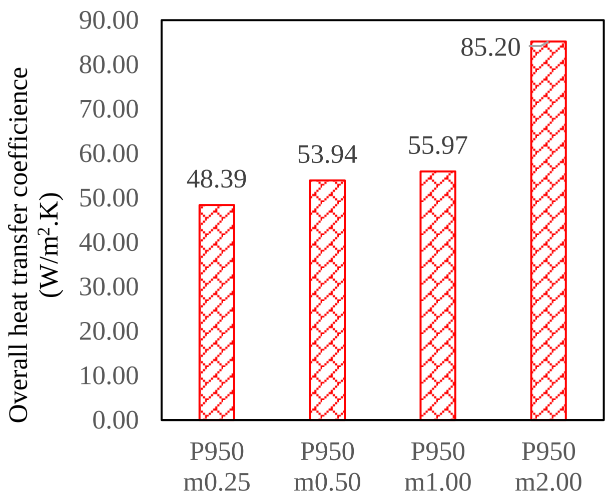


Fig. 22. Effect of water's flow rate on the overall heat ransfer coefficient oft he HTX.

Figure 23 shows he effectiveness The HTX When changing the flow rate of the incoming water. The flow rate was 0.25 kg/s and 2 kg/s, which is equaled to the effectiveness over the flow rate of 0.50 kg/s and 1 kg/s.

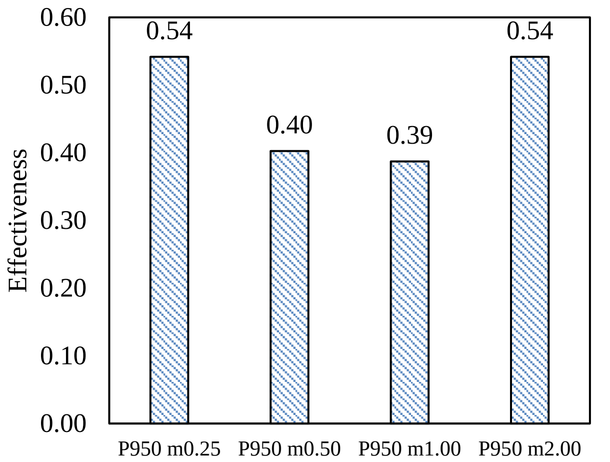


Fig. 23. Cost effectiveness When the HTX, the water flow rate.

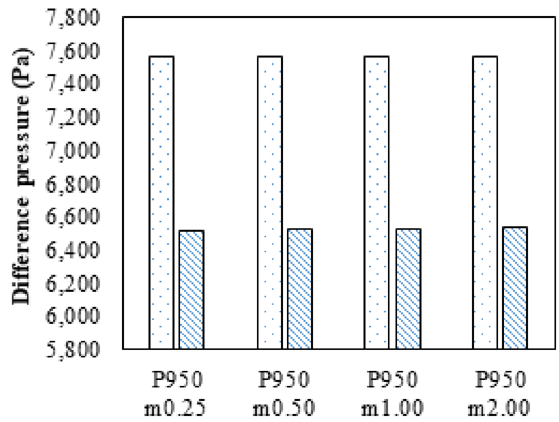


Fig. 24. The difference in pressure of the fluid mass flow rate changes on water.

Figure 24 Comparison of pressure changes on the flow rate. Once found, the flow rate, the differential pressure is the same. The flow rate is not affected by the difference in pressure.

5. CONCLUSIONS

Using the methods of computational fluid dynamics analysis, the ability to exchange heat when shifting pitch and mass flow rate of the study are as follows.

1. When change the pitch, the term pitch 850 mm Coefficient of heat transfer as possible. most

2. When changing the flow rate of the fluid mass flow rate was found. 2 kg/s Coefficient of heat transfer as possible. most

It was concluded that The term pitch 850 mm And the flow rate 2 kg/s Suitable to be used as a fuel vapor control system.

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