ความดันลดและการสูญเสียกำลังในเครื่องแลกเปลี่ยนความร้อนแบบบัฟเฟิลเคลื่อนที่แบบสลับ

นางสาวสรียา ศรวิบูลย์ศักดิ์

วิทยานิพนธ์นี้เป็นส่วนหนึ่งของการศึกษาตามหลักสูตรปริญญาวิศวกรรมศาสตรมหาบัณฑิต

สาขาวิชาวิศวกรรมเคมี ภาควิชาวิศวกรรมเคมี

คณะวิศวกรรมศาสตร์ จุฬาลงกรณ์มหาวิทยาลัย

ปีการศึกษา 2552

PRESSURE DROP AND POWER DISSIPATION IN OSCILLATORY BAFFLED HEAT EXCHANGER

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A Thesis Submitted in Partial Fulfillment of the Requirements
for the Degree of Master of Engineering Program in Chemical Engineering
Department of Chemical Engineering
Faculty of Engineering
Chulalongkorn University
Academic Year 2009

Thesis Title	PRESSURE DROP AND POWER DISSIPATION IN OSCILLATORY BAFFLED HEAT EXCHANGER
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Field of Study	Chemical Engineering
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สรียา ศรวิบูลย์ศักดิ์ : ความดันลดและการสูญเสียกำลังในเครื่องแลกเปลี่ยนความร้อนแบบบัฟเฟิลเคลื่อนที่แบบสลับ. (PRESSURE DROP AND POWER DISSIPATION IN OSCILLATORY BAFFLED HEAT EXCHANGER) อ. ที่ปรึกษาวิทยานิพนธ์หลัก : ผศ.ดร.สุรเทพ เขียวหอม, 84 หน้า.

ในงานวิจัยนี้ได้ศึกษาความดันลดของเครื่องแลกเปลี่ยนความร้อนแบบท่อสองชั้นชนิดมีแผ่นบัฟเฟิลภายในท่อเคลื่อนที่แบบสั่นสลับไปมา (oscillatory baffled plate) ซึ่งความดันลดเป็นความสูญเสียหลักที่เกิดขึ้นในหน่วยปฏิบัติการนี้ ตัวแปรที่ส่งผลต่อความดันลดได้แก่ สัดส่วนรูภายในของแผ่นบัฟเฟิล สัดส่วนระยะห่างระหว่างแผ่นบัฟเฟิล อัตราการไหลของสายร้อน ความถี่ และแอมปลิจูดของการเคลื่อนที่แบบสลับ โดยจะศึกษาความสัมพันธ์ของตัวแปรดังกล่าวในรูปตัวแปรไร้หน่วยในช่วงเรย์โนลนัมเบอร์ของการเคลื่อนที่ตั้งแต่ 0 ถึง 2,000 เรย์โนลนัมเบอร์ของของไหลสายร้อนตั้งแต่ 100 ถึง 1,000 สัดส่วนรูภายในแผ่นบัฟเฟิลเทียบกับเส้นผ่านศูนย์กลางท่อตั้งแต่ 0.4 ถึง 0.7 สัดส่วนของระยะห่างระหว่างบัฟเฟิลเทียบกับเส้นผ่านศูนย์กลางท่อตั้งแต่ 1 ถึง 2.5 ความถี่ในการเคลื่อนที่แบบสลับตั้งแต่ 0.5 ถึง 2 เฮิร์ทซ์ และแอมปลิจูดในการเคลื่อนที่แบบสลับตั้งแต่ 0.25 ถึง 1 เซนติเมตร โดยพฤติกรรมที่สังเกตได้จะแบ่งเป็นสองช่วงคือเรย์โนลนัมเบอร์ตั้งแต่ 200 ถึง 600 และ 700 ถึง 1,200 และความสัมพันธ์อย่างง่ายที่พัฒนาขึ้นของทั้งสองช่วงการไหล เป็นดังสมการ

$$\frac{\Delta p}{\frac{1}{2}\rho v^2} = \frac{1.19066x10^{11}[\mathrm{Re}\,o]^{0.14758}}{[\mathrm{Re}\,h]^{1.62023}} \qquad \text{และ}$$

$$\frac{\Delta p}{\frac{1}{2}\rho v^2} = \frac{0.15039x10^{10}[\mathrm{Re}\,o]^{0.17981}}{[\mathrm{Re}\,h]^{1.11808}[\alpha]^{0.30682}[\beta]^{0.14299}} \qquad \text{ตามลำดับ}$$

ภาควิชา วิศวกรรมเคมี ลายมือชื่อนิสิต สรียา

สาขาวิชา วิศวกรรมเคมี ลายมือชื่อ อ.ที่ปรึกษาวิทยานิพนธ์หลัก สุรเทพ

ปีการศึกษา 2552

4971520621 : MAJOR CHEMICAL ENGINEERING
KEYWORDS : OSCILLATORY BAFFLED / PRESSUER DROP / HEAT EXCHANGER

SAREYA SORNVIBOONSAK : PRESSURE DROP AND POWER DISSIPATION IN OSCILLATORY BAFFLED HEAT EXCHANGER. THESIS ADVISOR : ASSISTANT PROFESSOR SOORATHEP KHEAWHOM, Ph.D., 84 pp.

In the research work, the pressure drop in an oscillatory baffled plate double pipe heat exchanger is investigated. The pressure drop is a major lost encountered in this unit operation. The variables affecting the pressure drop include the ratio of the diameter of baffled plate to pipe diameter, the ratio of distance between the baffled plate to pipe diameter, the flowrate of the hot stream, the frequency, and amplitude of the oscillatory movement. We investigate the relationship among these variables in a dimensionless form in the range of Oscillation Reynolds number between 0 to 2,000, Reynolds number from 100 to 1,000, the ratio of the diameter of baffled plate to pipe diameter from 0.4 to 0.7, the ratio of distance between the baffled plate to pipe diameter from 1 to 2.5, the frequency of the oscillatory movement from 0.5 to 2.0 Hz, and the amplitude of the oscillatory movement from 0.25 to 1.0 cm. The behavior observed can be classified into two categories with different range of Reynolds number, from 200 to 600 and 700 to 1,200. The empirical models developed for both categories are

$$\frac{\Delta p}{\frac{1}{2}\rho v^2} = \frac{1.19066x10^{11}[\mathrm{Re}\,o]^{0.14758}}{[\mathrm{Re}\,h]^{1.62023}} \quad \text{and}$$

$$\frac{\Delta p}{\frac{1}{2}\rho v^2} = \frac{0.15039x10^{10}[\mathrm{Re}\,o]^{0.17981}}{[\mathrm{Re}\,h]^{1.11808}[\alpha]^{0.30682}[\beta]^{0.14299}} \quad \text{,respectively.}$$

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Field of Study : Chemical Engineering Advisor's Signature Soorathep Kheawhom
Academic Year : 2009

ACKNOWLEDGMENTS

This thesis will never be completed without the help and support of many people and organizers who are gratefully acknowledged here. Firstly, I would like to express my sincere gratitude to Assistant Professor Soorathep Kheawhom, Ph.D., my advisor, for his suggestions, guidance, warm encouragement and generous supervision throughout my master program. I am also grateful to Assistant Professor Montree wongsri, Dr. Kasidit Nootong, and Dr. Veerayut Lersbamrugsuk for their helpful and many valuable comments.

Of course, I wish to express my thankfulness to all members in The Control Engineering Research Laboratory and SANYO Semiconductor (Thailand) Co., Ltd. for the many nice times and their encouragement during my study. Especially, I can not forget to express my thankfulness to Assistant Professor Soorathep's group research: N'Aor, N'Keng, N'Meen and N'Deaw and My boss: Mr. Wasan Sontipermpoon and Sanyo's friend: P'Nan Moreover, special thanks should be made for the Graduate School, Chulalongkorn University for their financial support.

Most of all, I would like to express my sincere indebtedness to my parents, everyone in my family and my friends for their inspiration and worthy support at all times.

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LIST OF SYMBOLS

A	Area of heat transfer
A^i	Cross section area of hot flow rate
A^o	Cross section area of cool flow rate
A_i	Area of heat transfer of inside pipe
A_o	Area of heat transfer of outside pipe
Cp	Heat capacity
Cp_i	Heat capacity (hot flow rate)
Cp_o	Heat capacity (cool flow rate)
D	Diameter
D_i	Inside Diameter
D_o	Outside Diameter
D_s	Diameter of shell
D_h	Diameter of tube
r	Radius
r_i	Inside radius
r_o	Outside radius
d_o	Diameter of baffle plate
S	Distain between baffle

L	Length of heat exchanger
f	Oscillatory frequency
x_o	Oscillatory amplitude
F_h	Hot flow rate
F_c	Cool flow rate
V_i	Volume in side pipe of hot flow rate
V_o	Volume in side pipe of cool flow rate
v_i	Velocity in side pipe (hot line)
v_o	Velocity in side pipe (cool line)
$\dot{m}_h$	Mass flow rate of hot line
$\dot{m}_c$	Mass flow rate of cool line
ρ	Density of fluid
ρ_i	Density of fluid (hot line)
ρ_o	Density of fluid (cool line)
μ	Viscosity
μ_i	Viscosity (hot line)
μ_o	Viscosity (cool line)
ν	Kinematics viscosity
ν_i	Kinematics viscosity (hot line)
ν_o	Kinematics viscosity (cool line)

Re	Reynolds Number
Re_{osc}	Oscillatory Flow Reynolds Number
Re_{h}	Hot Flow Reynolds Number
Re_{c}	Cool Flow Reynolds Number
α	The ratio inside diameter per inside diameter
β	The ratio of spacing per inside diameter

CHAPTER I

INTRODUCTION

1. Background and important

Heat exchangers have been used in a wide variety of applications such as the district heating stations, the heat recovery systems, food and chemical process systems, and the oil cooling systems. As energy has been concerned, the energy conservation has driven the development in the high efficiency of heat exchangers in recent years. The demand for economical, lightweight, and space saving heat exchangers has influenced the research of compact surfaces. The double pipes heat exchanger is one of the best examples for the advancement and technological development in the compact heat exchanger due to its high thermal efficiency but the system has gained the pressure drop that has an effect on cost of operation.

The cost-optimized design of heat exchangers requires a fast and reliable calculation of pressure drop on the operation costs. For co-current and counter-current flows, a simple approximating calculation method is presented, taking into account the variation of the fluid properties along the flow path. For any practical case reliable, results are obtained only by calculating both the pressure drop and the local overall heat transfer coefficient at least at two points of the heat exchanger.

The various heat transfer enhancement techniques have been divided by considering increasing the rate of heat transfer in forced convection through reducing the size of the heat exchanger and the effect of energy consumption. That can be categorized to be the active techniques and the passive techniques. The active techniques is to bring out source factors for making turbulent flow such as external power source, mechanical aids, surface vibration, fluid vibration, electrostatic fields, injection or suction of fluid and jet impingement. On the other hand, the passive techniques is to add some equipment for making turbulent likes such as the treated surfaces, rough surfaces, extended surfaces, displaced enhancement devices, swirl flow devices, coiled tubes, additives for liquids and gases, and et cetera. When compare with this two techniques, the passive is relatively low cost method, easy to install and remove the inside tubes for cleaning purpose.

Recently, the technology of the oscillatory baffled flow has been increasingly utilized in various industrial processes such as the suspension polymerization, the crystallization, the paint dispersion, the flocculation and the fermentation. Focusing on a substantial body of scientific papers and engineering applications, currently support the view that the oscillatory baffled flow is an exciting form of reactor and process technology with major commercial potential. However, the significant point is that the passive heat transfer enhancement mostly achieves at a cost of pressure drop increment. Therefore, the evaluating of these techniques requires the knowledge of both heat transfer and pressure drop minimization.

Pressure drop is an economical factor in the heat exchanger design which is required to optimize during heat exchanger equipment design phase. In this study, it is to research in relationships between factors, which affect pressure drop, including with find out the well-matched condition of double baffled plate pipe in terms of oscillatory flow. This study mainly improves heat exchanger efficiency design and improves efficiency of each operating unit such as the chemical reactor, the mixing tank, fluid and et cetera. In addition, the fluid pumping power is required to determine as part of the system design and operation cost analysis.

1.1 Objective of this work

1. To determine the effect of operating parameters including hot flow rate, frequency and amplitude of shaking, phasing of baffled plate and internal hold of baffle plate to pressure drop in oscillatory baffled plate heat exchanger.
2. To develop the empirical model between pressure drop and related parameters including hot flow rate, frequency and amplitude of shaking, phasing of baffled plate and internal hold of baffle plate in oscillatory baffled plate heat exchanger.

1.2 Scopes of this work

The variables have an effect on the pressure drop such as the ratio of the hole in the baffled plate per inside pipe diameter, the ratio of distance between the baffled plate per inside pipe diameter, heat flow and the oscillatory of frequency and amplitude as the value on Table 1.1 in the Reynolds number of oscillatory from 0 to 2,000 and the Reynolds number of the hot flow from 100 to 1,000.

Table 1.1 Condition during heat exchanger machine running

Factor		Unit	Value
Inner pipe	Hot Flow rate	l/min	4-15
	Amplitude	cm	0-1
	Frequency	Hz	0-2
	Inside baffle plate diameter / inside pipe diameter: d_i/D_i	-	0.4-0.7
	Spacing/ Inside pipe diameter: S/D_i	-	1-2.5
Outer pipe	Cold Flow rate	l/min	5-25

CHAPTER II

LITERATURE REVIEW

2.1 Heat exchanger

Choowang P. and Kheawhom S. (2009) study method to increase the heat transfer efficiency of double pipe heat exchanger at laminar flow by using the combined effect of oscillatory motion and turbulence promotion. Thermal boundary layer can be decreased by installing periodically spaced orifice baffles. The oscillatory motion is achieved by moving a set of orifice baffles up and down at the top of the tube. The parameters that have effect on heat transfer efficiency are inner hole diameter of baffle plate, space between baffle plates, hot and cold stream flow rates, frequency and vibration amplitude. The relationships between Nusselt number of inner and outer heat transfer surfaces can be shown as functions of oscillatory flow Reynolds number in rang of 0-2,000, hot flow Reynolds number in range of 100-1,000, cold flow Reynolds number in range of 500-2,500, the ratio of inner hole diameter and tube diameter in range of 0.4-0.7 and the ratio of the space between each baffles and tube diameter in range of 1-2.5 are

$$Nu_i = 0.1995\,\mathrm{Re}_h^{0.3839}\,\mathrm{Pr}_i^{0.9998}\left(1+\frac{1}{\alpha^{0.1956}\beta^{0.2093}}\right)+\frac{0.1077\,\mathrm{Re}_{osc}^{0.4871}\,\mathrm{Pr}_i^{1.000}}{\mathrm{Re}_h^{0.5244}\,\alpha^{0.9748}\beta}-100.0$$ and

$Nu_o = 58.7525\,\mathrm{Re}_c^{4.2271}\,\mathrm{Pr}_o^{0.9639}$, respectively.

Ni, Mackley and Stonestresst (2003) is stated to the previously research, which is use for gain more efficiency on each substance to be the better condition. Start from 1940-1950, on industrial where would like to extract the substance from Plate or Packed column. To gain more productivity on extract, the new way is oscillation flow. There are 2 types of oscillation Flow. First, Pulsing the contents of the column (PPC) is making liquid inside of column oscillation by use in-out pump, during the fluid in column. Second, Reciprocating plate column (RPC) is to make baffled plate up and down which the reflection will attack and carry fluid between baffled plates shaking. This research study on oscillation flow which have 2 types of baffle.

Strong point of 2 baffled oscillation flows are

1. Add more efficiency of heat transference.

2. Using of reactor type pipe flow.

3. Add more efficiency of mass transfer between gas and liquid.

4. Add more efficiency of solid separation.

5. Control droplets in liquid separation

6. Crystallization process.

7. Polymerization process.

8. Add more efficiency of sediment mixing process.

9. Gain more efficiency of batch reactor process.

Ru Yang and Fan Pin Chiang (2001) study in heat exchanging in double pipe which inner is waviness. By using heat exchanger coefficient of Wilson plot way, to study in factor which could effected to heat exchanger. Dean Number (Da), Prandtl Number (Pr), Reynolds number (Re) and friction factor (f) find out heat transference's coefficient. In high Dean Number, heat transfer is also high and will increase 100% when compare with direct flow. The friction factor decrease 40% and also get low Reynolds number which is less than 600 and adapt into another method.

NI, Brogan, Struthers, Bennet and Wilson (1998) is experiment on factor of mass transfer coefficient on mixing time and find out the proper parameter in double bending pipe of oscillation baffled. The study on ratio between inner hold of baffled compare with diameter (d_i/D_i), value 11-51%. The ratio between baffle compare with diameter, value 1-2 and thickness of baffle is 1-48 mm. The conductivity measurement find the last mixing time by using NaCl. The frequency is on 1-10 times per second. Amplitude is 1-20 ml.

The result of during experiment, ratio between inner hold baffled compared with pipe diameter in differ frequency and amputate. The frequency is 10 rounds per second and amplitude is 10 mm. can get the least mixing time. The conclusion is in the best mixing and can get the proper parameter are

1. Ratio between hole of baffled and diameter (d_i/D_i) is on 20-22 %.

2. Ratio of baffle compared with pipe diameter, proper value is 2.

3. Proper of baffled thickness is 2-3 mm.

Mackley and Stonestresst (1995) study on gain more efficiency of heat transfer and reduce flow of oscillation baffled, Which is study from double pipe oscillation in pulsation causes from cylinder's pusher. In pipe also construct with baffle plate phasing between baffle plate and pipe diameter is 7.5 on net flow Reynolds number (Re_n) and Oscillatory flow Reynolds number (Re_{os}) on Laminar Flow 0 - 1,400 and 0 - 1,000 so on. The conducted value compare with heat exchanger coefficient (Nusselt Number) to compare with Laminar flow equations which is the equation of Key and Neddernman in 1985. In pipe of non-baffle and oscillation flow of baffled. Furthermore, to get the low pressure which is from oscillating inner baffled, to find power or massive of efficiency compare with volume.

The power density data in this researches, at small amplitudes of oscillation and higher frequencies. The power density is greater than predicted by the quasi-steady model. They believe the device is operating in a non-quasi-steady regime. They told non-quasisteady operating regime offers a different type of fluid mechanics to either quasi steady or turbulent flow, and we believe that significant process advantage can be found by operating in this regime.

2.2 Pressure drop

Tsai, Liu and Shen (2009) is investigations of the pressure drop and flow distribution in a chevron-type plate heat exchanger which they work hydrodynamic characteristics and distribution of flow in two cross-corrugated channels of chevron plate heat exchangers. The numerical results have been validated with the measurements taken by laboratory experiments. A three-dimensional model with the real-size geometry of the two cross-corrugated channels and accounting the inlet and outlet ports has been conducted for the numerical study. The local flow characteristics around the contact points have been discussed. The velocity, pressure and flow distribution of the fluid among the two channels of the plate heat exchanger have also been presented. The computational results of pressure drop have been validated by the experiment data. The experimental results of pressure drop are 20% higher than the predictions by CFD.

Akhavan-Behabadi, Salimpour, and Pazouki (2008) study pressure drop increase of forced convective condensation inside horizontal coiled wire inserted was performed to investigate the increment of the pressure drop during condensation of R-134 a vapor inside a horizontal tube with different coiled wire inserts. Investigating the results for the coiled wire inserted tubes revealed that insertion of helically coiled wires inside horizontal tubes was found to augment the pressure loss from 260% to 1600% above the plain tube values on a nominal area basis. Also, influence of coiled wire geometry on the pressure drop was investigated. Based on the collected data, Pressure loss decreases as coiled wire diameter reduces and coiled wire pitch increases. To predict the pressure loss in coiled wire inserted tubes, a new correlation was developed based on the experimental data of the present study. A new correlation was developed based on Chisholm correlation to predict the pressure loss in plain tube. Finally, the performance evaluation of the coiled wire inserted condensers was done.

Mushtak Al-Atabi and Chin (2006) study pressure drop in laminar and turbulent flows in circular pipe with baffles – An experimental and analytical study. That flow in a circular pipe fitted with segmental baffles may be treated as a shell without-tube system. Its pressure drop has been calculated by adapting the Kern correlation for pressure drop in the shell side of shell-and-tube heat exchangers. The Kern correlation is essentially based on the Hagen – Poiseuille equation for laminar flow. However flow visualization results presented here show that enhanced mixing and turbulence-like flow may be present at Reynolds numbers (based on the pipe diameter) as low as 50. The model accounted for the geometry of the baffles and used well known correlations to combine the effects of turbulent and laminar flows. In addition, the model equations were solved algebraically, which makes it very convenient to use. An analytical model has been developed which predict the pressure drop across pipe with baffles system in nominally laminar flow condition. Flow visualization suggested that the baffles induce turbulence-like flow structure at Reynolds number much lower than the critical value where flow transition is expected to occur. The model was solved algebraically for flow in pipe with three baffle arrangements and the results were validated by experimental data. The present pressure drops model shows much better agreement with the experimental data when compared with of Kern on various clearance to diameter ratios for Reynolds number between 50 and 600.

Cheng and Chen (2006) study of single phase flow heat transfer and friction pressure drop in a spiral internally ribbed tube which the experimental results of single-phase flow heat transfer and pressure drop experiments in the turbulent flow regime in a spirally ribbed tube and a smooth tube are presented. The ribbed tube has an outside diameter of 22 mm and an inside diameter of 11 mm (an equivalent inside diameter of 11.6 mm) and the smooth tube has an outside diameter of 19 mm and an inside diameter of 15 mm. Both tubes were uniformly heated by passing an electrical current along the tubes with a heated length of 2,500 mm. The working fluids are water and kerosene, respectively. The experimental Reynolds number is in the range of 10^4–5 x10^4 for water and is in the range of 10^4–2.2 x10^4 for kerosene. The experimental results of the ribbed tube are compared with those of the smooth tube. The heat transfer coefficients of the ribbed tube are 1.2–1.6 fold greater than those in the smooth tube and the pressure drop in the ribbed tube is also increased by a factor of 1.4–1.7 as compared with those in the smooth tube for water.The corresponding values for kerosene are 2–2.7 and 1.5–2, respectively. The heat-transfer enhancement characteristics of the ribbed tube are assessed.

Roetzel (1973) study calculation of single phase pressure drop in heat exchangers considering in the change of fluid properties along the Flow Path which the cost-optimized design of heat exchangers requires a fast but reliable calculation of pressure drop which makes a major contribution to running costs. For concurrent and countercurrent flow a simple approximating calculation method is presented, taking into account the variation of the fluid properties along the flow path. For any practical case, reliable results are obtained only by calculating both the pressure drop and the local overall heat transfer coefficient at least at two points of the heat exchanger. In the special case of a gas in a turbulent flow and as usual, the major resistance to heat transfer is caused by the gas, it is sufficient to calculate only the pressure drop and at one point only. Pressure drops calculated exactly or by the proposed approximation compare well.

CHAPTER III

THEORY

Fluids need to be pumped through the heat exchanger in most applications. It is essential to determine the fluid pumping power required as part of the system design and operation cost analysis. The fluid pumping power is proportional to the fluid pressure drop, which is associated with fluid friction and other pressure drop contributions along the fluid flow path. The fluid pressure drop has a direct relationship with exchange heat transfer, operation, size, mechanical characteristics, and other factors, including economic considerations. The objective of this chapter is to outline the methods for pressure drop analysis in heat exchangers and related flow devices.

3.1 Importance of pressure drop

The determination of pressure drop Δp in a heat exchanger is essential for many applications for at least two reasons: (1) The fluid needs to be pumped through the exchanger, which means that fluid pumping power is required. This pumping power is proportional to the exchanger pressure drop. (2) The heat transfer rate can be influenced significantly by the saturation temperature change for a condensing /evaporating fluid if there is a large pressure drop associated with the flow. This is because saturation temperature changes with changes in saturation pressure and turn affects the temperature potential for heat transfer.

Let us first determine the relative importance of the fluid pumping power $\mathbf{P}$ for gas flow vs. liquid flow in a heat exchanger. $\mathbf{P}$ is proportional to Δp in a heat exchanger and is given by

$$\mathrm{P} = \frac{\dot{v}\Delta p}{\eta_p} = \frac{m\Delta p}{\rho \eta_p} \tag{3.1}$$

Where $\dot{v}$ is the volumetric flow rate and η_p is the pump/fan efficiency. Now introduce as following relationships:

$$\dot{m} = GA_o \qquad \Delta\rho \approx f\frac{4L}{D_h}\frac{G^2}{2g_c\rho} \qquad \mathrm{Re} = \frac{GD_h}{\mu} \tag{3.2}$$

Where G is referred to the baffle *mass velocity* ($G = pu_{m'}$). A_o is the minimum free flow area, f is the Fanning friction factor and Re is the Reynolds number as defined in Eq.(3.2) The Δp expression in Eq. (3.2) is for the baffle frictional pressure drop and is derived later. Substituting the expressions of Eq. (3.2) into Eq. (3.1) and simplifying results in

$$\mathrm{P} = \frac{\dot{m}\,\Delta p}{\rho\eta_\rho} \approx \begin{cases} \dfrac{1}{2g_c\eta_p}\dfrac{\mu}{\rho^2}\dfrac{4L}{D_h}\dfrac{m^2}{D_h A_0}(f.\mathrm{Re}) & \text{For fully developed laminar flow} \quad (3.3a) \\[2ex] \dfrac{0.046}{2g_c\eta_p}\dfrac{\mu^{0.2}}{\rho^2}\dfrac{4L}{D_h}\dfrac{m^{2.8}}{A_0^{1.8}D_h^{0.2}} & \text{For fully developed turbulent flow} \quad (3.3b) \end{cases}$$

Here $f = 0.046\mathrm{Re}^{-0.2}$ is used in the derivation of the right hand-side expression of Eq. (3.3*b*) for fully developed turbulent flow. Note also that f Re in Eq. (3.3*a*) is constant for fully developed laminar flow. To determine the order of magnitude for the fluid pumping power requirement for gas vs. liquid flow, let us assume that the flow rate and flow passage geometry are given (i.e., m, L, D_h, and A_o are specified). It is evident from Eq. (3.3) that $\mathrm{P} \propto 1/\rho^2$ (i,e., strongly dependent on ρ in laminar and turbulent flows): $\mathrm{P}\,\alpha\,\mu^{0.2}$ (i,e., weakly dependent on μ in turbulent flow). For high-density moderate-viscosity liquids, the pumping power is generally so small that it has only a minor influence on the design. For laminar flow of highly viscous liquids in large L/D_h. Exchangers, the fluid pumping power is an important constraint. In addition, the pumping power is an important consideration for gases, in both turbulent and laminar flow, because of the great impact of $1/\rho^2$. For example, the density ratio for liquid water vs. air at ambient conditions is approximately 800: 1, which indicates that the pumping power for airflow will be much higher than that for water if $\Delta\mathrm{P}$ is to be kept the same. Hence, typical design values of $\Delta\mathrm{P}$ for water and air as working fluids in a heat exchanger are 70 kPa (10 Psi) (a typical value in shell-and-tube exchanger) and 0.25

kPa (1 in. H_2O) (For compact exchanger with airflows near ambient pressures), respectively, to maintain the low fluid pumping power requirement for exchanger operation.

3.2 Fluid pumping devices

The most common fluid pumping devices are fans, pumps, and compressors. A *f*an is a low-pressure air- or gas-moving device, which uses rotary motion. There are two major types of fans: axial and radial (centrifugal), depending on the direction of flow through the device. Fans may be categorized as blowers and exhausters. A *blower* is a centrifugal fan when it is used to force air through a system under positive pressure, and it develops a reasonably high static pressure (500 Pa or 2.0 in H_2O). An *exhauster* is a fan placed at the end of a system where most of the pressure drop is on the suction side of the fan. A *pump* is a device used to move or compress liquids. A *compressor* is a high-volume centrifugal device capable of compressing gases [100 to 1500 kPa (15 to 220 psi)] and higher.

Fans and pumps are volumetric devices and commonly used to pump fluids through heat exchanger. This means that a fan will develop the same dynamic head [pressure rise per unit fluid (gas) weight across the fan; Eq. (3.4)] at a given capacity (volumetric flow rate) regardless of the fluids handled, with all other conditions being equal. This means the pressure rise across a fan will be proportional to the fluid density at a given volumetric flow rate for all other conditions being equal. Note that the *head, dynamic head of velocity head* is referred to as the kinetic energy per unit weight of the fluid pumped, expressed in units of millimeters of inched (feet). Thus the pressure rise across a fan (which is mainly consumed as the pressure drop across a heat exchanger) can be expressed in terms of the head *H* as flows:

$$\frac{\Delta p}{pg / g_c} = H = \frac{\mu^2{}_m}{2g} \tag{3.4}$$

Since fans and pumps are generally head limited, the pressure drop in the heat exchanger can be a major consideration.

3.3 Major contributions to the heat exchanger pressure drop

The pressure drop associated with a heat exchanger is considered as a sum of two major contributions: Pressure drop associated with the baffle of matrix, and pressure drop associated with fluid distribution devices such as inlet/outlet headers, manifolds, tanks, nozzles, ducting, and others. The purpose of the heat exchanger is to transfer thermal energy from on fluid to the other; and for this purpose, it requires pressure difference (and fluid pumping power) to force the fluid flow over the heat transfer surface in the exchanger. Hence, ideally most of the pressure drop available should be utilized in the baffle and a small fraction in the manifolds, headers, or other flow distribution devices. However, this ideal situation may not be the case in plate heat exchangers and other heat exchangers in which the pressure drop associated with manifolds, headers, nozzles, and so on, may not be a small fraction of the total available pressure drop.

If the manifold and header pressure drops are small, the baffle pressure drop dominates. This results in a relatively uniform flow distribution through the baffle. All heat transfer and baffle pressure drop analyses outlined here and in preceding chapters presume that the flow distribution through the baffle uniform. A serious deterioration in performance may result for a heat exchanger when the flow through the baffle is not uniformly distributed.

The baffle pressure drop is determined separately on each fluid side. It consists of one or more of the following contributions, depending on the exchanger construction: (1) frictional losses associated with fluid flow over the heat transfer surface (this usually consists of skin friction plus form drag), (2) momentum effect (pressure drop or rise due to the fluid density changer in the baffle), (3) pressure drop associated with sudden contraction and expansion at the baffle inlet and outlet, and (4) gravity effect due to the change in elevation between the inlet and outlet of the exchanger. The gravity effect is generally negligible for gases. For vertical liquid flow through the exchanger, the pressure drop or rise due to the elevation change is given by

$$\Delta p = \pm \frac{p_m g L}{g_c} \tag{3.5}$$

Where the " + " sign denotes vertical up flow (i.e., pressure drop), the " - " sign denotes vertical down flow (i.e., pressure rise or recovery), g is gravitational acceleration, L is the exchanger length, and p_m is the mean fluid mass density calculated at bulk temperature and means pressure between the two points where the pressure drop is to be determined.

3.4 Assumptions for pressure drop analysis

The following are the major assumptions made for the pressure drop analysis presented

1. Fluid flow is isothermal steady flow and fluid properties are independent time.
2. Fluid density is only dependent on the local temperature which is treated as a constant value (inlet and outlet are different density).
3. The point pressure at in the fluid is direction independent. If a shear stress is present, the pressure is defined as the average of normal stresses at that point.
4. Body forces are only caused by gravity (i.e, magnetic, electrical, and other fields do not contribute to the body forces).
5. If the flow is not irritation, the Bernoulli equation is valid only along a streamline.
6. There are no energy sources or sinks along a streamline; flow stream mechanical energy dissipation is idealized as zero.
7. The friction factor is considered as a constant with passage flow length.

3.5 Quasi-steady theory for power density

For oscillatory flows it has been a common practice to assume "quasi-steady" behavior (Jealous and Johnson, 1955). This model assumes that the frictional pressure drop in a time-periodic flow at a certain instantaneous velocity. It is assumed to be identical to the pressure drop that would be obtained at a steady velocity of the same magnitude of the instantaneous velocity. The maximum frictional pressure drop is obtained according to the standard pressure drop relation across an orifice (at high Reynolds numbers), i.e.:

$$\Delta p_{fo} = \frac{n\rho(x_0\omega)^2(1/S^2-1)}{2C_0^2} \tag{3.6}$$

Where n is the number of baffles and C_0 orifice coefficient (usually assumed to be 0.6). The power density for quasi-steady flow is given by:

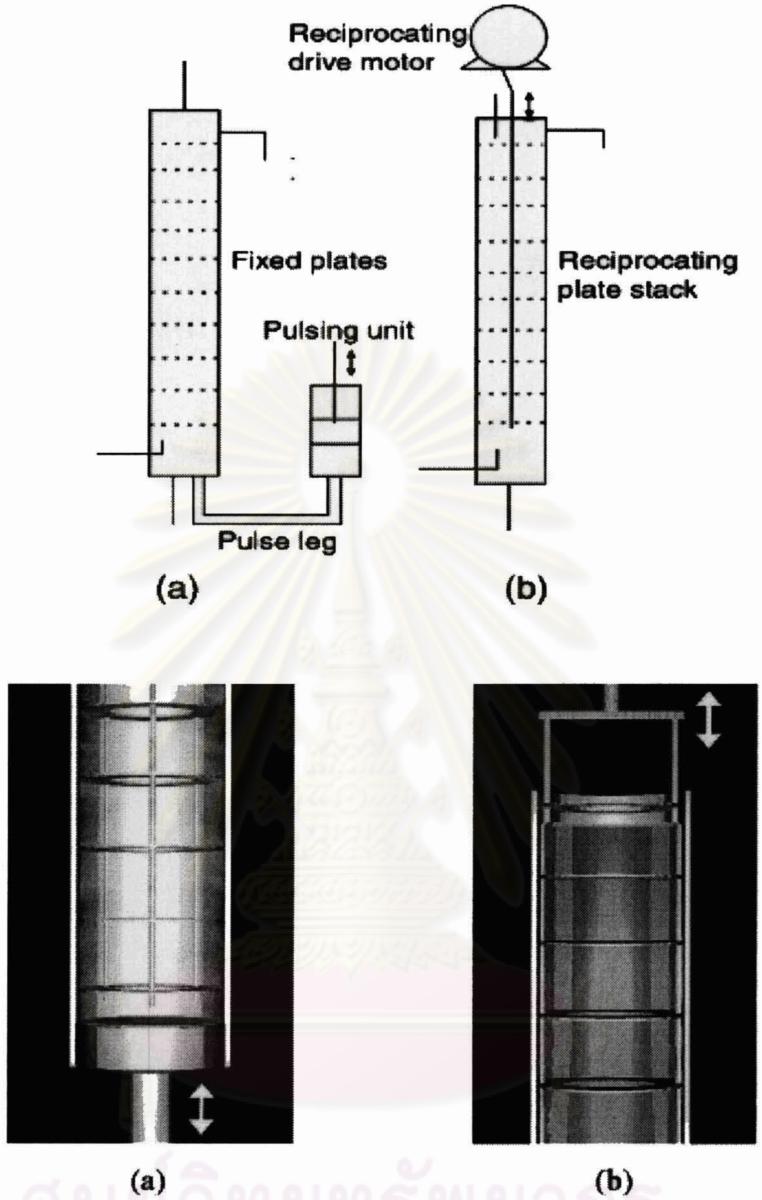
$$\varepsilon_v = \frac{2n\rho(\omega x_0)^3(1/S^2-1)}{3\pi C_0^2 Z} \tag{3.7}$$

3.6 Oscillation Flow

The process for add more efficient by mixing substance during 2 types of fluid. While substance is mixed. It will have the mass and heat transference. Start from 1940 – 1950 (Ni et al. 2003) one of industry want to separate on substance from Plate or packed column in massively, to increase the efficiency on extract, there is move investigation on increasing efficiency by oscillation flow which have 2 type as the figures 3.1A and 3.1B.

1. Pulsing the contents of the column is to make oscillation in column by using pump in-out during flow of substance in column.

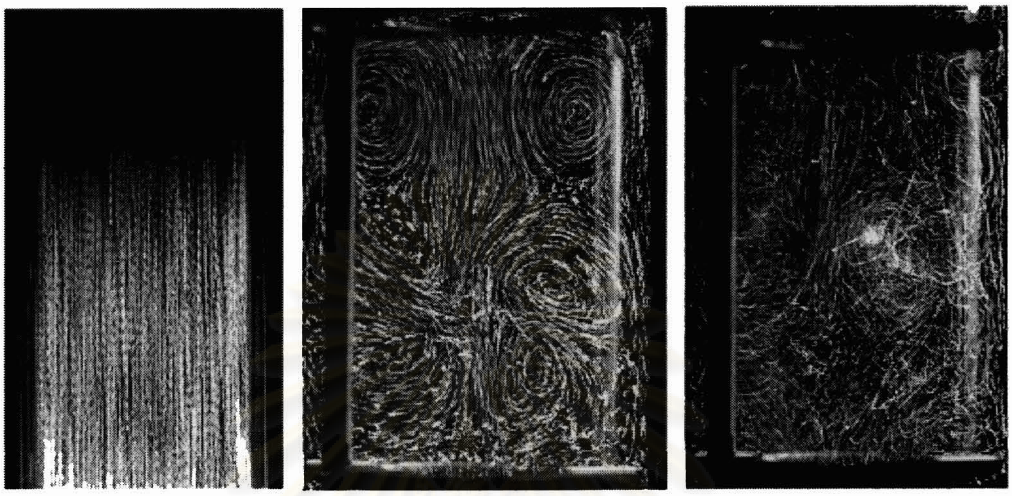
2. Reciprocating plate column (RPC) is to move baffled plate in column up and down. The result is to make fluid between baffled plates shaking.



Figures 3.1 Type of oscillation flow

(a) Pulsing the contents of the column. (b) Reciprocating plate column (RPC).

The fluid flowing of baffle type use by mixed more fluid to gain more transfer effectively between heat and mass. As the figures 3.3



(c) (d) (e)

Figures 3.2 Oscillation flow

(c) No baffle (d) and (e) Have baffle.

3.7 Technical Overview

The knowledge can be adapt on Heat transfer, Plug flow, Mass transfer, Solid suspension, Polymerization, Crystallization, Hydrogenation The Oscillatory Flow is cause of mass transfer and heat transfer get more good condition and made fluid still be in column, which suit for lamina flow or slowly flow. In case of turbulent while mixing of fluid in more effective, use on Reciprocating plate column (RPC) is most popular than Pulsing the contents of the column because can use wide amplitude.

3.8 Parameter of flowing fluid related in term of dimensionless

Reciprocating plate column (RPC) have many factors composed (Mackley and Stonestreet, 1995) by the oscillation fluids, which have equation of dimensionless. The Oscillation of dimensionless consist of amplitude and frequency is Re_o (Oscillatory Reynolds number) or Peak Reynolds number as follow equation 3.1

$$\mathrm{Re}_o = \frac{2\pi f x_o D}{\nu} \tag{3.8}$$

by x_o = Amplitude from Centre-to-peak

f = Frequency of oscillation

D = Diameter

ν = Kinematics viscosity

Net flow Reynolds number in the tube (Re_n) is vary velocity as the equation. 3.9

$$\mathrm{Re}_n = \frac{UD}{\nu} \tag{3.9}$$

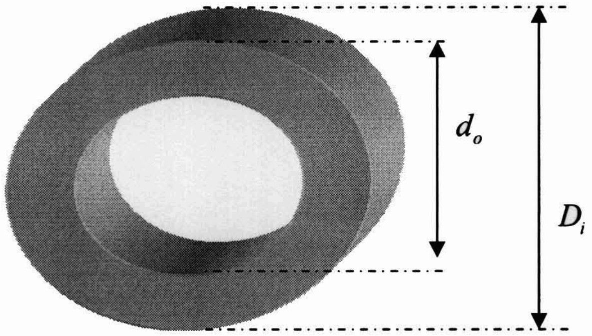
U = Average velocity

Strouhal number (Sr) is vary amplitude as the equation.3.10

$$Sr = \frac{D}{4\pi x_o} \tag{3.10}$$

3.9 Relation of dimensionless

The characteristic of heat exchanger on oscillatory double pipe heat exchanger which have same characteristic of the double pipe, but different in inner double pipe as the baffle arrange in the baffle layer for shake and the baffle characteristic as follow.



Figures 3.3 Baffle plate

<u>Inner pipe</u> *Dimensionless as follow*

α = Ratio between inner diameters of baffle per inside diameter pipe.

$$\alpha = \frac{d_o}{D_i} \tag{3.11}$$

β = Ratio between spacing baffle to inside diameter pipe.

$$\beta = \frac{S}{D_i} \tag{3.12}$$

Re_h = Hot flow Reynolds Number

$$\mathrm{Re}_h = \frac{\rho_i v_i D_i}{\mu_i} \tag{3.13}$$

Re_{osc} = Oscillatory flow Reynolds Number

$$\mathrm{Re}_{osc} = \frac{\rho_i 2\pi f x_o D_i}{\mu_i} \tag{3.14}$$

<u>Outer pipe</u> *Dimensionless as follow*

Re_c = Cool flow Reynolds Number

$$\mathrm{Re}_c = \frac{\rho_o v_o D_o}{\mu_o} \tag{3.15}$$

3.10 Pressure drop presentation

For most heat exchanger surfaces, the baffle pressure drop for a given heat transfer surface is determined experimentally (using a ‘small size’ heat exchanger) as a function of the fluid flow rate. Details on the experimental method are presented in thesis. These data may then be used for the design and analysis of heat exchanger of different physical size, operating at different temperatures and/or pressures, and operating with different fluids compared to those for the test exchanger, but using the same heat transfer surface. Hence, we need to present the test exchanger baffle Δp vs. $\dot{m}$ result in a universal form so that they can be used for operation conditions, physical sizes, and fluids beyond those for the test exchanger.

If the detail on geometry (such as the hydraulic diameter D_h, minimum free flow area A_0, etc.) of the heat exchanger surface is known, the best approach is to present the baffle pressure drop versus mass flow rate results in nondimensional form, such as an f vs. Re curve. If the detail on geometry information is not available, the measured baffle Δp vs. $\dot{m}$ result are presented in dimensional form. Both approaches are presented next.

3.11 Nondimensional presentation of pressure drop data

The baffle pressure drop in a dimensionless form generally presented in two alternative forms, in terms of the pressure loss coefficient K or Euler number Eu and the Fanning friction factor f defined by

$$\Delta p = K \frac{\rho u_m^2}{2g_c} = Eu \frac{\rho u_m^2}{2g_c} \tag{3.16}$$

$$\Delta p = f \frac{4L}{D_h} \frac{\rho u_m^2}{2g_c} \tag{3.17}$$

Thus, the Euler number is the same as the pressure loss coefficient K. If the pressure loss coefficient K is constant along the flow length, such as in tube banks, manifolds, bends, valves, and so on, as a result of turbulent flow, usually K or Eu is used to present the pressure drop.

The Fanning friction factor generally represents primarily the frictional component of the pressure drop and is used when the given heat transfer surface has approximately the same frictional pressure drop per unit length along the flow direction. Thus, use of the friction factor allows pressure drop prediction of different flow lengths of the heat exchanger surface. In fluid dynamics books, generally the Darcy friction factor f_D is used and is related to the Fanning friction factor f as

$$f_D = 4f = \frac{\Delta p}{(pu_m^2/2g_c)}\frac{D_h}{L} \qquad (3.18)$$

A comparison of Eq. (3.17) and (3.18) indicates that one needs to know the hydraulic diameter and flow length of the exchanger surface if the pressure drop is presented in terms of the friction factor. No such information is needed for Δp to be presented in terms of K or Eu.

The fluid flow rate is presented in dimensionless form as the Reynolds number defined as

$$\mathrm{Re} = \frac{\rho u_m D_h}{\mu} = \frac{GD_h}{\mu} = \frac{\dot{m}\, D_h}{A_0 \mu} = \frac{p\dot{V}\, D_h}{A_0 \mu} \qquad (3.19)$$

The greatest advantage of the nondimensional presentation is for a given Reynolds number, geometrically similar surfaces (regardless of the physical size) have the same friction factor with any fluid flowing through the surface. This means that when converted into terms of f vs. Re, the experimental data (Δp vs. $\dot{m}$) can be used for different operating conditions (temperature, pressure, etc.), different physical sizes (different D_h but geometrically similar surfaces), and different fluids from those used in the test conditions. Also, such f vs. Re plots allow one to compare Δp vs. $\dot{m}$ data taken on different surfaces with different fluids of operating conditions. So a heat exchanger design with minimum flow resistance can be selected for a given application.

3.12 Dimensional presentation of pressure drop data

In industry, it is a common practice to present the pressure drop data in a dimensional form for specified heat exchanger surfaces, since no geometry information is required for such a presentation. These results are presented in different forms by different industries, such as Δp vs. $\dot{m}$, Δp vs. $\dot{V}$, or Δp vs. G. To correct for an operating/design temperature being different from the test temperature, a density correction is usually applied to the pressure drop by plotting the pressure drop at some standard density. However, the outline of theory is for this pressure drop correction by matching the friction factor and Reynolds number between the actual and standard conditions for Δp vs. $\dot{m}$, $\dot{V}$, or G. The standard pressure and temperature are different for the different industries depending on their applications, fluids used, and other factors. For example, the standard conditions for air for some industrial heat exchangers could be I atm pressure and 20°C (68°F) temperature.

To obtain a standard flow rate (in terms of $\dot{m}$ or G) from the actual flow rate measured at operating conditions for a given exchanger surface, we need to match

$$\mathrm{Re}_{std} = \mathrm{Re}_{act} \Rightarrow \frac{\dot{m}_{std}\, D_h}{\mu_{std} A_0} = \frac{\dot{m}_{act}\, D_h}{\mu_{act} A_0} \tag{3.20}$$

Thus,

$$\dot{m}_{std} = \dot{m}_{act} \frac{\mu_{std}}{\mu_{act}} \tag{3.21}$$

Since $\dot{m} = \dot{V}\rho$, we get the following relationship between $\dot{V}_{std}$ and $\dot{V}_{act}$ using Eq.(3.21)

$$\dot{V}_{std} = \dot{V}_{act} \frac{\rho_{act}}{\rho_{std}} \frac{\mu_{std}}{\mu_{act}} \tag{3.22}$$

Since $G = \dot{m} / A_0$, we get from Eq. (3.21),

$$G_{std} = G_{act} \frac{\mu_{std}}{\mu_{act}} \tag{3.23}$$

To match friction factors, use Eq. (3.18) and $\dot{m} = p\mu_m A_o$ to get

$$f = \Delta p \frac{2g_c}{pu^2{}_m} \frac{D_h}{4L} = \Delta p \frac{2g_c \rho A^2{}_0}{\dot{m}^2} \frac{D_h}{4L} \tag{3.24}$$

Now let us match the standard and actual friction factors for a given geometry using Eq. (3.21)

$$f_{std} = f_{act} \Rightarrow \frac{\Delta p_{std} p_{std}}{\dot{m}^2_{std}} = \frac{\Delta p_{act} p_{act}}{\dot{m}^2_{act}} \tag{3.25}$$

Thus

$$\Delta p_{std} = \Delta p_{act} \frac{p_{act}}{P_{std}} \left(\frac{\dot{m}_{std}}{\dot{m}_{act}} \right)^2 \tag{3.26}$$

Substituting m_{std} from Eq. (3.17) into Eq. (3.26), we get

$$\Delta p_{std} = \Delta p_{act} \frac{\rho_{act}}{\rho_{std}} \left(\frac{\mu_{std}}{\mu_{act}} \right)^2 \tag{3.27}$$

This is the most general relationship between Δp_{std} and Δp_{act} that requires both density and viscosity correction, and should be used in all cases.

However, in industry the viscosity correction in Eq. 3.27 is usually neglected and only the density correction is applied. It can be rationalized as follows. When the friction factor is constant (independent of the Reynolds number for turbulent flow) as in a rough pipe or when f is not strongly variable for turbulent flow in a smooth pipe, we do not need to math Re_{std} and Re_{act} as in done Eq. 3.20. Hence, in this case,

$$\dot{m}_{std} = \dot{m}_{act} \tag{3.28}$$

And Eq. (3.26) simplifies to

$$\Delta p_{std} = \Delta p_{act} \frac{\rho_{act}}{\rho_{std}} \tag{3.29}$$

In most situations, whether or not f is constant is not known a priori, and therefore Eq. (3.27) is recommended for correcting $\Delta\rho_{act}$ to $\Delta\rho_{std}$ and Eq. (3.17), (3.18), and (3.19) for correcting $\dot{m}$, $\dot{V}$, and G , respectively, for a plot of $\Delta\rho$ vs. $\dot{m}$, $\dot{V}$, or G at standard conditions.

3.13 Pressure drop dependence on geometry and fluid property

Pressure drop in a heat exchanger baffle/matrix is dependent on some fluid properties and geometrical parameters. Since the frictional pressure drop is the dominating contribution to the baffle pressure drop, we use that item only for the analysis. Substituting $A_0 = D_h A / 4L$ from the definition of the hydraulic diameter, we get the expression for Δp from Eq. (3.3) as

$$\Delta p = \begin{cases} \dfrac{1}{D_h^3}\left[\dfrac{1}{2g_c}\dfrac{\mu}{\rho}\dfrac{(4L)^2}{A}\dot{m}(f.\mathrm{Re})\right] & \text{For laminar flow} \quad (3.30a) \\ \dfrac{1}{D_h^3}\left[\dfrac{0.046}{2gc}\dfrac{\mu^{0.2}}{\rho}\dfrac{(4L)^{2.8}}{A^{1.8}}\dot{m}^{1.8}\right] & \text{For turbulent flow} \quad (3.30b) \end{cases}$$

Thus the pressure drop is proportional to D_h^{-3} (for constant $\dot{m}$, *L*, *A*, and fluid properties). For a circular tube, since A = πD_h L, $\Delta\rho$ of Eq. (3.30) is proportional to D_h^{-4} and $D_h^{-4.8}$ in laminar and turbulent flows respectively. Hence, based on Eq. (3.30),

$$\Delta p \alpha \begin{cases} \dfrac{1}{D_h^3} to \dfrac{1}{D_h^4} & \text{For laminar flow} \quad (3.31a) \\ \dfrac{1}{D_h^3} to \dfrac{1}{D_h^{4.8}} & \text{For turbulent flow} \quad (3.31b) \end{cases}$$

Reviewing Eq. (3.3) we fine that Δp is proportional to *L* when D_h and A_0 are constant, and Δp is proportional to 1 / $A_0^{1.8}$ for laminar and turbulent flow respectively when D_h and *L* are constant. In these comparisons, we also keep m and fluid properties constant. Note that the surface area *A* is then not an independent variable since *A* = (4 A_0 *L*) / D_h

$$\Delta p \alpha \begin{cases} L, \dfrac{1}{A_0} & \text{For laminar flow} \quad (3.32a) \\ L, \dfrac{1}{A_0^{1.8}} & \text{For turbulent flow} \quad (3.32b) \end{cases}$$

From Eq. 3.30 we find that

$$\Delta p \alpha \begin{cases} \dfrac{1}{\rho}, \mu & \text{For laminar flow} \quad (3.33a) \\ \dfrac{1}{\rho}, \mu^{0.2} & \text{For turbulent flow} \quad (3.33b) \end{cases}$$

Thus, Δp is dependent on ρ and μ as shown, but is not directly dependent on c_p and k.

Finally, the dependence of the pressure drop on the surface geometry is given as follows from the second equality of Eq. (3.3):

$$\Delta p \alpha \frac{L}{A_0^2 D_h} \tag{3.34}$$

For a specified fluid and mass flow rate. Since the friction factors and Colburn factors (or Nu) for enhanced heat transfer surfaces are higher than those for a plain surface, both heat transfer rate and pressure drop for an enhanced surface will be higher than those for a plain surface for a given fluid mass flow rate *m* and the same hear exchanger dimensions. However, it is possible to maintain high-hear-transfer performance with the same pressure drop (particularly in laminar flows) by properly choosing the heat exchanger dimensions and surface geometry. However, if the exchanger frontal area cannot be changed, one will end up with a larger pressure drop with enhanced surface compared to that for the plain surface for a given fluid flow rate. In that case (i.e., for a fixed frontal area), a plain underhanded surface will have a lower pressure drop but longer exchanger flow length (and hence larger volume and mass of the exchanger) to meet the hear transfer requirement specified.

3.14 Suitable function analyses

The built relation of equation for explains the data. In case, equations have unknown more than two, suppose to use static calculation. The method is Least Square Method as the follow.

The data from experiment (x_i,y_i) and calculate value from building equation G(x_i).The differ value is y_i-G(x_i). When finding different point of every point, the square so the value is plus. The summary of different 2 power of value of y_i (From experiment) and G(x_i) (from building equation) call the sum square of the residuals (SSE).

$$SSE = \sum_{i=1}^{n} (y_i - G(x_i))^2 \tag{3.35}$$

If mathematically model evaluation was made in suitable SSE, which should be at least.

When yi experiment value deduct from experiment data average ($\overline{y}$) , the result different between yi and $\overline{y}$ is call The sum of the square of the regression (SSR)

$$SSR = \sum_{i=1}^{n} (y_i - \bar{y})^2 \tag{3.36}$$

R^2 is show data and mathematically model equation is suitable and be accordance together or not by starch from the formula below

$$R^2 = 1 - \frac{SSE}{SSR} \tag{3.37}$$

R^2 will be at 0 to 1 if value from data and accumulate ion is same, which R^2 is near 1. The graph will be accordance and suitable with data. However, generally R^2 would be than 0.9.

CHAPTER IV

MATERIAL & METHODS

The experiment of heat exchanging oscillatory baffles heat exchanger in which study on affected of oscillation and alert. To reduce the thickness of heat resistance film, by phrasing orifice baffled installation and move up and down. There are tools and experimental as;

4.1 Instrument

The instrument in Heat Exchanging Machine consist of material and experiment tools

4.1.1 Material

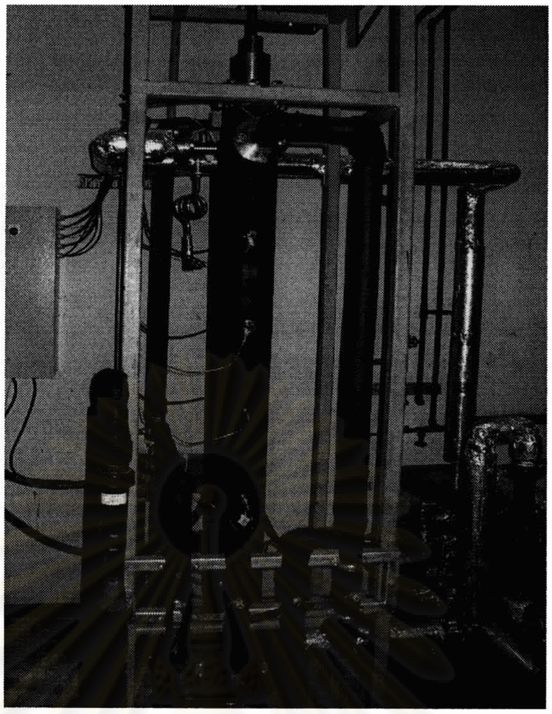
5.1.1.1 Paraffin oil or White oil NO.15, Grad A for hot line fluid.

5.1.1.2 Pipe water for cool line fluid

4.1.2 Experiment tools

4.1.2.1 Heat Exchanger Machine

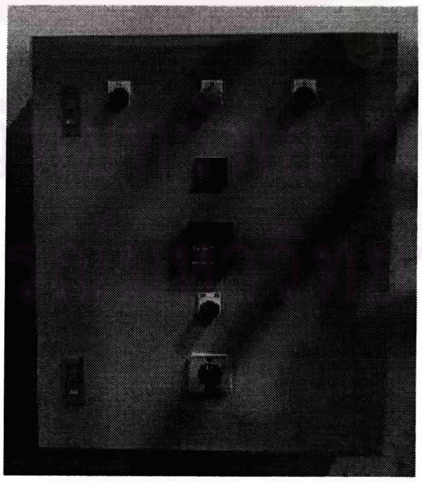
Composes of double pipe, Size of inside diameter pipe is 82 mm. and thickness around 3 mm., Size of outside diameter pipe is 125 mm. and length 150 cm. including with stainless steel pipe to cover by heat resistance plate and temperature indicator attached at input and output of line as figures 4.1



Figures 4.1 Heat Exchanger machine

4.1.2.2 Control and receiving system

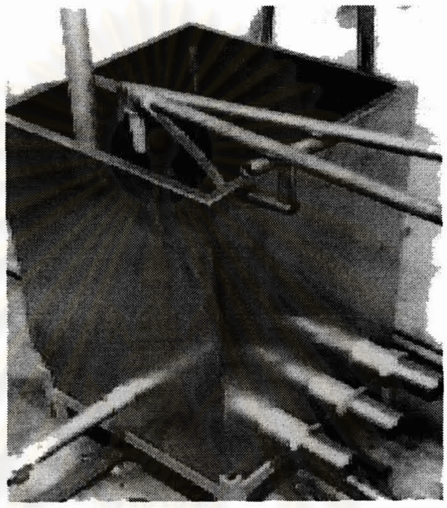
Link the receiver from Heat exchanger into computer and automatically record in every 10 seconds. In stable condition, get the digital record in excel file which will analyze into next step.



Figures 4.2 Temperature controller and signal receiver

4.1.2.3 Heat providing system

Using paraffin oil as hot line fluid in boilers tank and using heater 40 kW. by heat provider as figures 4.3. Temperature controlling by PID control at limited 80°C. which input on heat exchanger by Rotary meter as the flowing indicator.



Figures 4.3 Heater provider

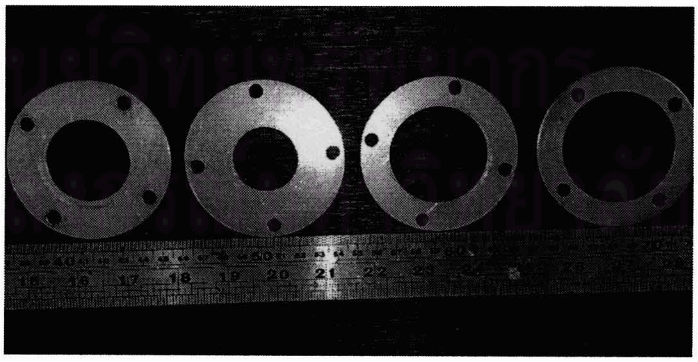
4.1.2.4 Cool providing system

Using water as cool line fluid which have Chiller as figures 4.4 as cool provider and limited temperature at 20 °C. To input by Rotary meter to heat exchanger as flowing indicator.

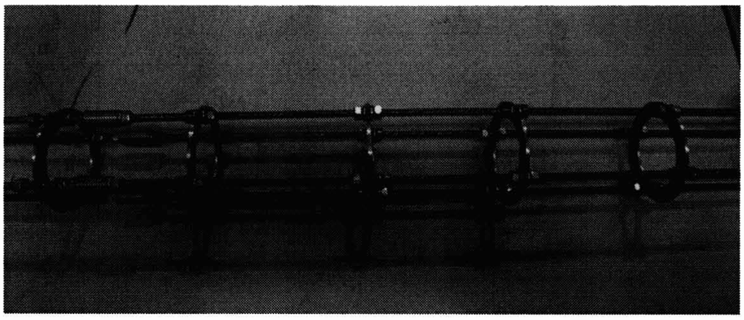
Figures 4.4 Chiller

4.1.2.5 Type and arrangement of baffles

For four types of orifice baffles as figures 4.5, which have inner hole compared with diameter of pipe as 0.4, 0.5, 0.6 and 0.7. And have core for baffles taping in difference phasing. Phasing between baffles compared with pipe diameter are 1, 1.5 and 2.5 as figures 4.6.



Figures 4.5 Type of baffles



Figures 4.6 The arrangement of baffles

4.2 Test method

This test method concerned about heat exchanger, which have the result as

4.2.1. Install baffle sheet inside baffle are $0.4D_i$, $0.5D_i$, $0.6D_i$ and 0.7 and the distance between baffle sheet are $1D_i$, $1.5D_i$, $2D_i$ and 2.5D.

4.2.2. Adjust the temperature of hot line fluid at 80 °C and cool line fluid at 20 °C.

4.2.3. Adjust flow ratio hot line and cool line required.

4.2.4. Adjust frequency and amplitude of baffle's vibration as required.

4.2.5. Wait until the temperature of fluid became steady-state and then record data.

4.2.6. After finished first test condition, change test condition and then do 4.2.1, 4.2.3 and 4.2.4 till target condition completed.

CHAPTER V

RESULTS AND DISCUSSION

5.1 Study of various parameters effected to pressure drop.

5.1.1 Experiment of various hot flow Reynolds number.

This experiment related with flowing condition at hot flow on ratio 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14 and 15 liters per minute. Cold flow rate have stability value on 20 liters per minute, oscillated amplitude at 0.75 cm, frequency 1 Hz.. As the Figure 5.1 the pressure is gradually increase on Reynolds number 200-700 and immediately decrease on 700-1,000 which mean the changing point. As the experiment can be concluded that the changing period is in lamina regime to turbulent regime. The pressure drop is various by Reynolds number or velocity square.

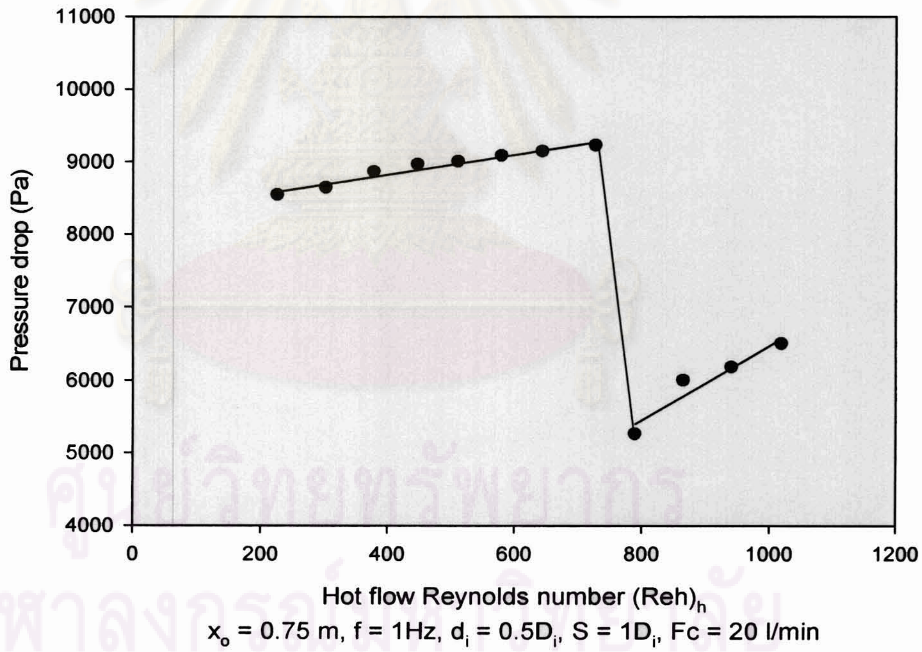


Figure 5.1 Result of pressure drop various hot flow Reynolds number.

5.1.2 Experiment of various oscillatory and no oscillatory

This experiment related with flowing condition at heat flow on ratio 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14 and15 liters per minute. Cold flow rate have stability value on 20 liters per minute, oscillated amplitude at 0 and 1 cm. frequency 1 Hz.. As the Figure 5.2 the pressure of oscillatory is gradually increase on Reynolds number 200-700 and immediately decrease on 700-900 which mean the changing point. In addition, the pressure of no oscillatory is gradually increase on Reynolds number 200-600 and immediately decreases on 600-900 which mean the changing point. When compare the experiment between oscillatory and no oscillatory, the result is no oscillatory rapidly pressure drop more than oscillatory. The result is no oscillatory rapidly decrease pressure drop more than oscillatory because the oscillatory probably have some liquid be trapped in the movement of baffled which have to get speed up for changing the fluid flow from laminar regime to turbulent regime. In contrast, the oscillatory have to gain more pressure because there is some lost energy from surrounding; such as baffled plate and pipe surface which made fluid more attack with pipe surface and baffled plate.

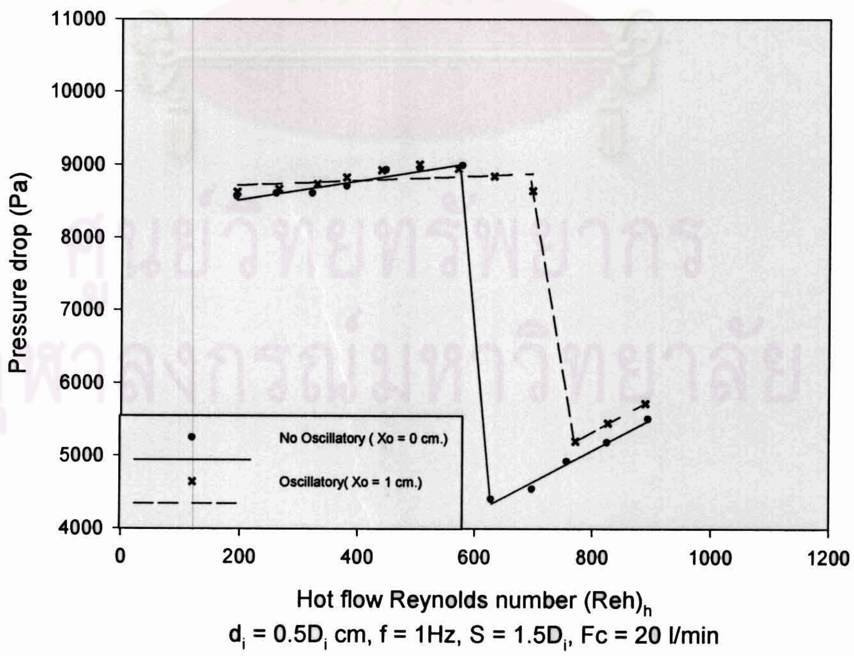


Figure 5.2 Result of pressure drop between oscillatory and no oscillatory

5.1.3 Experiment of various spacing per inside pipe diameter : S/D_i

This experiment related with flowing condition at the ratio inside diameter of baffle plate per diameter of pipe is 0.5, frequency 1 Hz., oscillated amplitude at 0.75 cm, cold flow rate have stability value on 20 liters per minute related with flowing condition at hot flow on ratio 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14 and 15 liters per minute. As the Figure 5.3a the pressure of oscillatory is gradually increase on Reynolds number 200-700 and immediately decrease on 700-1,000 because of flow change from lamina regime to turbulent regime. As the figure 5.3b, the ratio of spacing per inside pipe diameter is increasing (the number of baffled plate is decreasing), the pressure drop decreasing cause of the attacked area of fluid flowing decreasing which is lost of energy. And the distain between baffle plate at S = 2.0D is decrease pressure drop less than other.

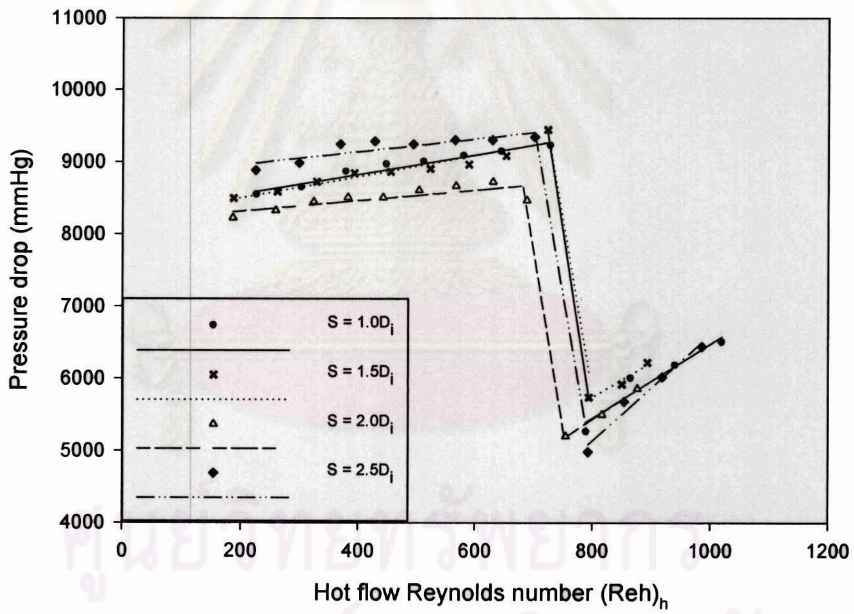


Figure 5.3a. Results of pressure drop vary the ratio of spacing per inside pipe diameter. S = $1.0D_i$, $1.5D_i$, $2.0D_i$ and $2.5D_i$ at $d_i = 0.5D_i$, $x_o = 0.75$ cm, f = 1 Hz, Cool Flow = 20 l/min

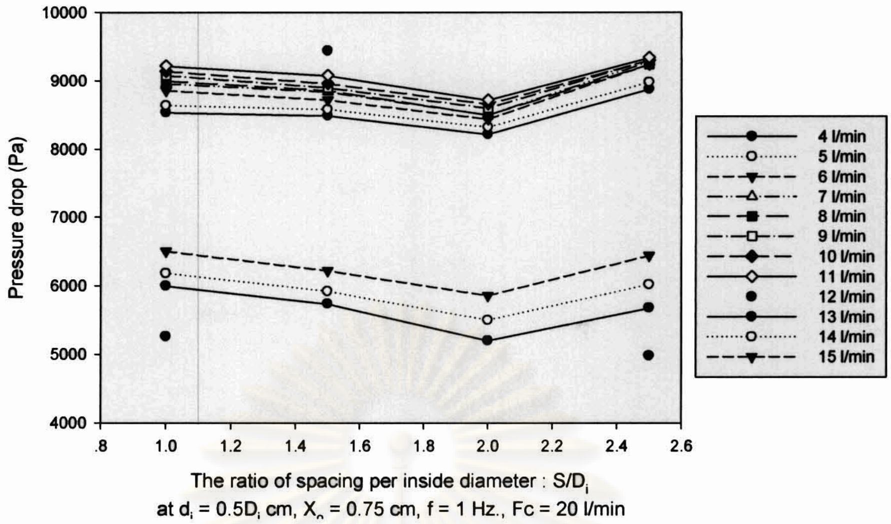


Figure 5.3b. Results of pressure drop vary the ratio of spacing per inside pipe diameter. S = $1.0D_i$, $1.5D_i$, $2.0D_i$ and $2.5D_i$ at $d_i = 0.5D_i$ x_o = 0.75 cm, f = 1 Hz, Cool Flow = 20 l/min

5.1.4 Experiment of various inside baffle plate diameter per inside pipe diameter: d_i/D_i

This experiment related with flowing condition at the ratio of spacing per inside diameter of pipe is 1.5, frequency 1 Hz., oscillated amplitude at 0.25 cm, cold flow rate have stability value on 20 liters per minute related with flowing condition at hot flow on ratio 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14 and 15 liters per minute. As the Figure 5.4a the pressure of oscillatory is gradually increase on Reynolds number 200-600 and immediately decrease on 600-800 because of flow change from lamina regime to turbulent regime. As the figure 5.4b, the ratio of inside baffle plate diameter per inside pipe diameter increasing), the pressure drop decreasing cause of the attacked area of fluid flowing decreasing which is lost of energy. And the inside diameter of baffle plate per inside diameter tube at do = 0.7D decrease pressure drop less than other.

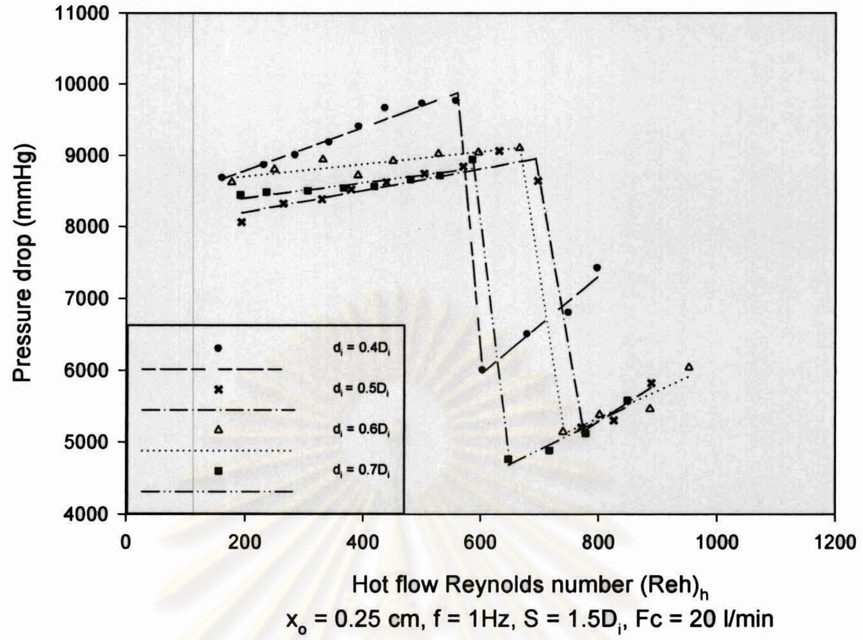


Figure 5.4a. Results of pressure drop vary the ratio of inside baffle plate diameter per inside pipe diameter $d_i = 0.4D_i$, $0.5D_i$, $0.6D_i$ and $0.7D_i$ at $S = 1.5D_i$, $x_o = 0.25$ cm, f = 1 Hz, Cool Flow = 20 l/min

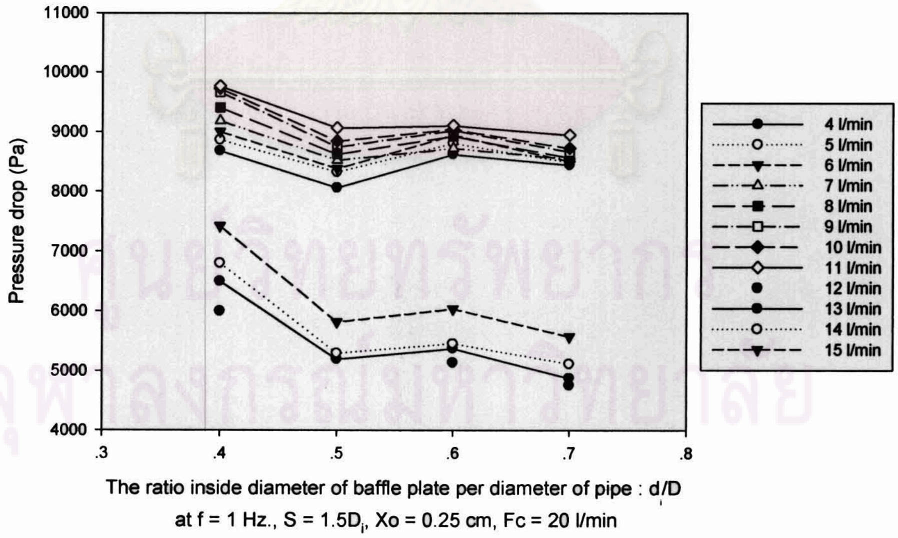


Figure 5.4b. Results of pressure drop vary the ratio of inside baffle plate diameter per inside pipe diameter $d_i = 0.4D_i$, $0.5D_i$, $0.6D_i$ and $0.7D_i$ at $S = 1.5D_i$, $x_o = 0.25$ cm, f = 1 Hz, Cool Flow = 20 l/min

5.1.5 Experiment of various Amplitude; x_o

This experiment related with flowing condition at the ratio inside diameter of baffle plate per diameter of pipe is 0.5, distain between baffle plate per diameter of pipe is 1.5, frequency 1 Hz., cold flow rate have stability value on 20 liters per minute related with flowing condition at hot flow on ratio 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14 and 15 liters per minute. As the Figure 5.5a the pressure of oscillatory is gradually increase on Reynolds number 200-700 and immediately decrease on 700-1,000 because of flow change from lamina regime to turbulent regime. As the figure 5.5b, the increasing of amplitude could made pressure drop increasing also. Because the lost of energy during attacking of fluid flow is increasing. And the amplitude at 1.0 m is decrease pressure less than other.

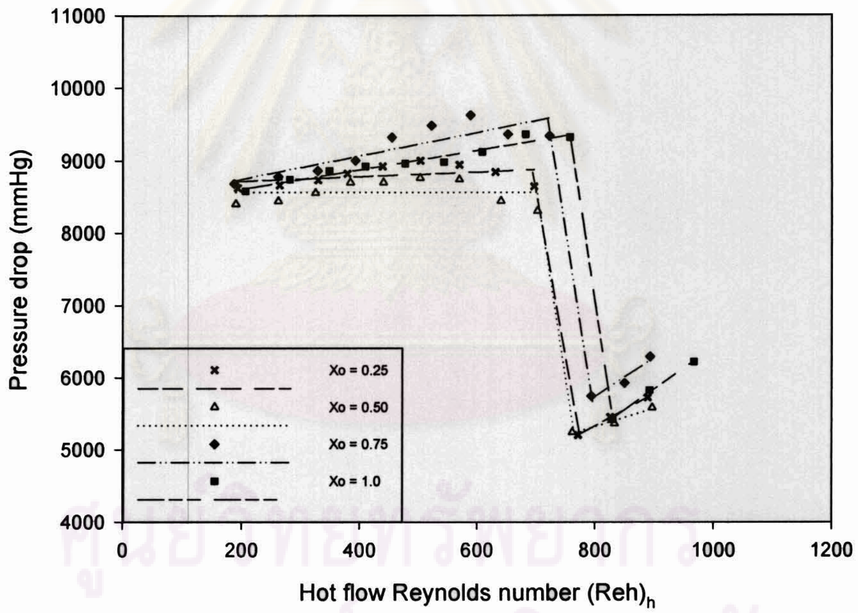


Figure 5.5a. Results of pressure drop vary amplitude (x_o) = 0.25, 0.50, 0.75 and 1.00 m. at d_i = 0.5D_i, S = 1.5D_i, f = 1 Hz, Cool Flow = 20 l/min

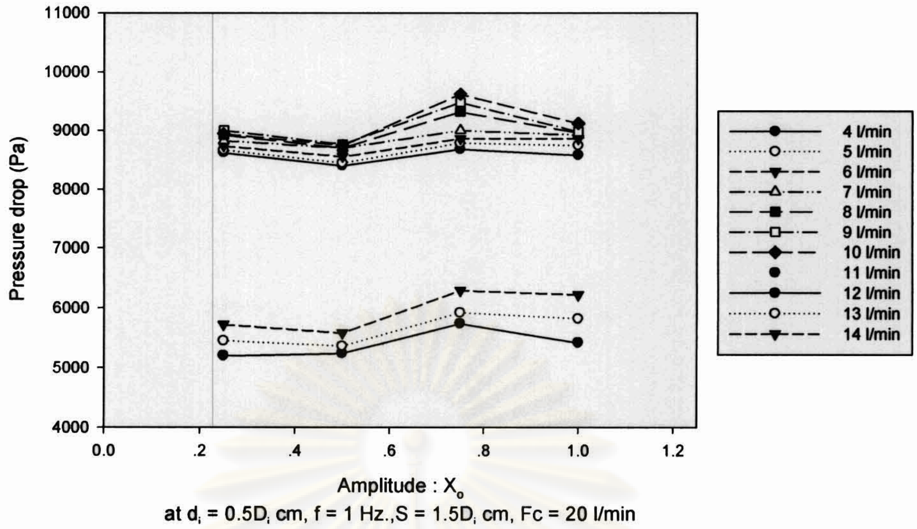


Figure 5.5b. Results of pressure drop vary amplitude (x_o) = 0.25, 0.50, 0.75 and 1.00 m.

at d_i = 0.5D_i, S = 1.5D_i, f = 1 Hz, Cool Flow = 20 l/min

5.1.6 Experiment of various Frequency; f

This experiment related with flowing condition at the ratio inside diameter of baffle plate per diameter of pipe is 0.5, distain between baffle plate per diameter of pipe is 1.5, oscillated amplitude 0.5 cm. cold flow rate have stability value on 20 liters per minute related with flowing condition at hot flow on ratio 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14 and15 liters per minute. As the Figure 5.6 the pressure of oscillatory is gradually increase on Reynolds number 200-600 and immediately decrease on 600-1,000 because of flow change from lamina regime to turbulent regime. As the figure 5.6b, the increasing of frequency could made pressure drop increasing also. Because the lost of energy during attacking of fluid flow is increasing. And the frequency at 1 Hz. is decrease pressure less than other.

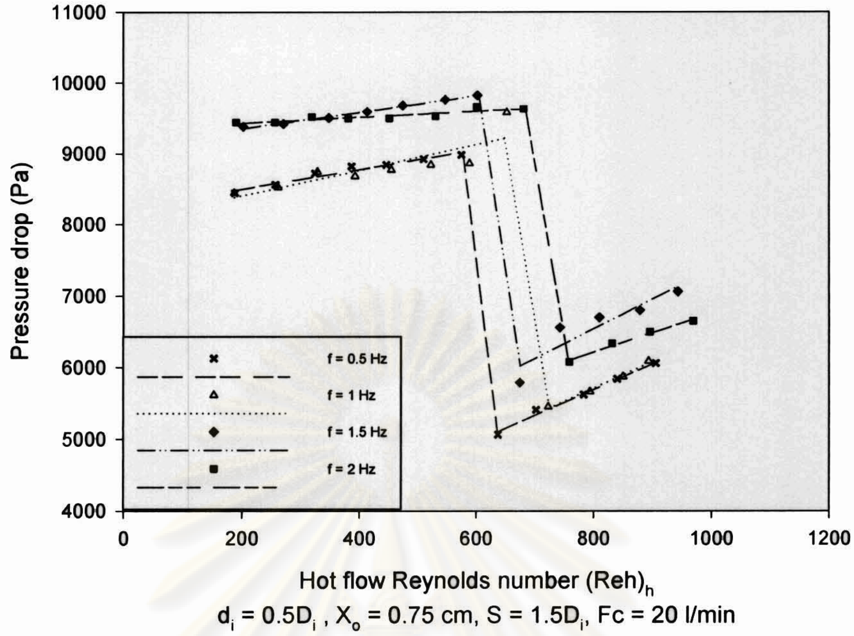


Figure 5.6a. Results of pressure drop vary frequency (f) = 0.5, 1.0, 1.5 and 2 Hz. at d_i = $0.5D_i$, S = $1.5D_i$, X_o = 0.75 cm., Cool Flow = 20 l/min

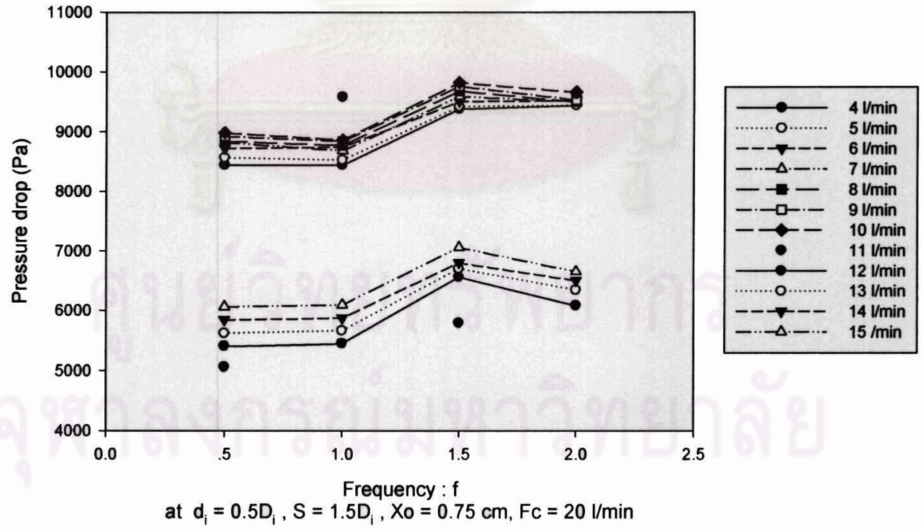


Figure 5.6b. Results of pressure drop vary frequency (f) = 0.5, 1.0, 1.5 and 2 Hz. at d_i = $0.5D_i$, S = $1.5D_i$, X_o = 0.75 cm., Cool Flow = 20 l/min

5.2 Study of pressure drop equation.

The relationships between pressure drop of inner heat exchanger can be shown as functions of oscillatory flow Reynolds number in rang of 0-2,000, hot flow Reynolds number in range of 100-1,000, the ratio of inner hole diameter and tube diameter in range of 0.4-0.7 and the ratio of the space between each baffles and tube diameter in range of 1-2.5 are

<u>$200 < Reh < 600$</u>

$$\frac{\Delta p}{\frac{1}{2}\rho v^{2}} = \frac{1.19066x10^{11}[\mathrm{Re}\,o]^{0.14758}}{[\mathrm{Re}\,h]^{1.62023}} \tag{5.1}$$

<u>$700 < Reh < 1200$</u>

$$\frac{\Delta p}{\frac{1}{2}\rho v^{2}} = \frac{0.15039x10^{10}[\mathrm{Re}\,o]^{0.17981}}{[\mathrm{Re}\,h]^{1.11808}[\alpha]^{0.30682}[\beta]^{0.14299}} \tag{5.2}$$

The standard deviation result in $200<Reh<600$ is 0.968 and $700<Reh<1200$ is 0.912 which can make the prediction of equation nearly the truth as figure 5.7.

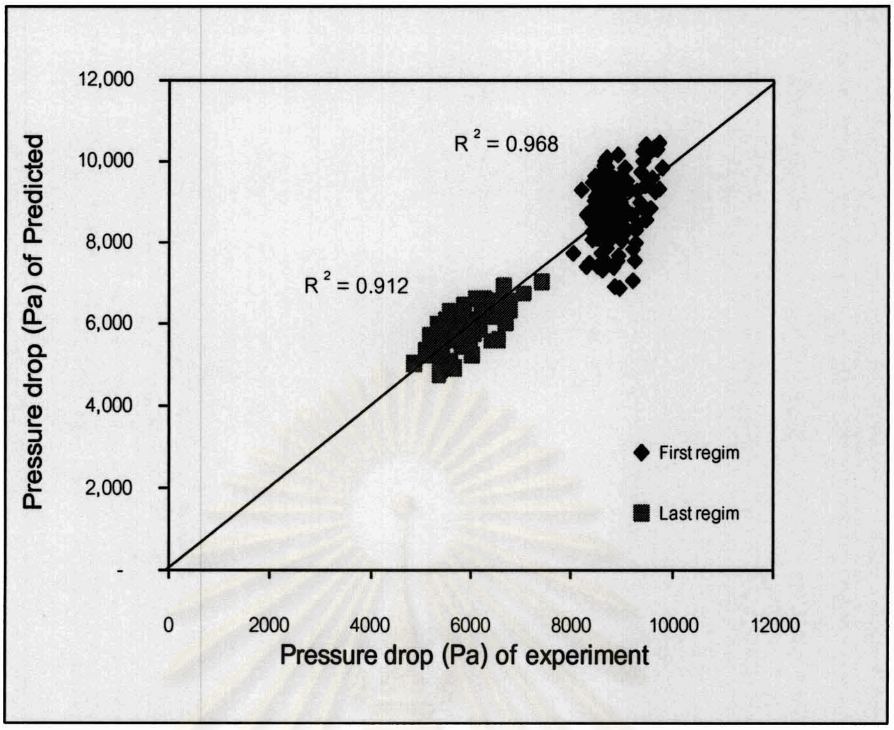


Figure 5.7 Results of pressure drop of experiment compare predicted.

CHAPTER VI

CONCLUSION AND RECOMMENDATION

From the experiment, pressure is immediately decrease at Reynolds number of hot flow around 600-700 due to changing of Lamina regime to Turbulence regime. Moving of baffle plates in the heat exchanger is a cause of turbulence flow at the lower Reynolds number. The variables have effected to the pressure drop such as the ratio of the hole in the baffled plate, the ratio of distance between the baffled plate, hot flow, frequency and amplitude.

As the result of experiment, the increasing ratio of distance between the baffled plates per inside pipe diameter tube and ratio of the hole in the baffled plate per inside pipe diameter could make pressure drop decreasing cause of lost energy in fluid flowing. Because the attacked area which is the loss of energy is less. The increasing amplitude and frequency is increasing could make pressure drop increasing. That is from the increasing of fluid flowing attacked, which could lose the energy in. And the ratio of distance between the baffled plates per inside pipe diameter tube is 2. The ratio of the hole in the baffled plate per inside pipe diameter is 0.7. Amplitude is 0.5 cm.. Frequency is 1 Hz.. Those parameter are gave the least pressure drop.

The relationships between pressure drop of inner heat exchanger can be shown as functions of oscillatory flow Reynolds number in rang of 0-2,000, hot flow Reynolds number in range of 100-1,000, the ratio of inner hole diameter and tube diameter in range of 0.4-0.7, the ratio of the space between each baffles and tube diameter in range of 1-2.5, frequency is 0.5 to 2 Hertz and amplitude 0.25 to 1 centimeter are

200 < Reh < 600

$$\frac{\Delta p}{\frac{1}{2}\rho v^2} = \frac{1.19066x10^{11}[\mathrm{Re}\,o]^{0.14758}}{[\mathrm{Re}\,h]^{1.62023}}$$ and

700 < Reh < 1200

$$\frac{\Delta p}{\frac{1}{2}\rho v^2} = \frac{0.15039x10^{10}[\mathrm{Re}\,o]^{0.17981}}{[\mathrm{Re}\,h]^{1.11808}[\alpha]^{0.30682}[\beta]^{0.14299}}$$, respectively.

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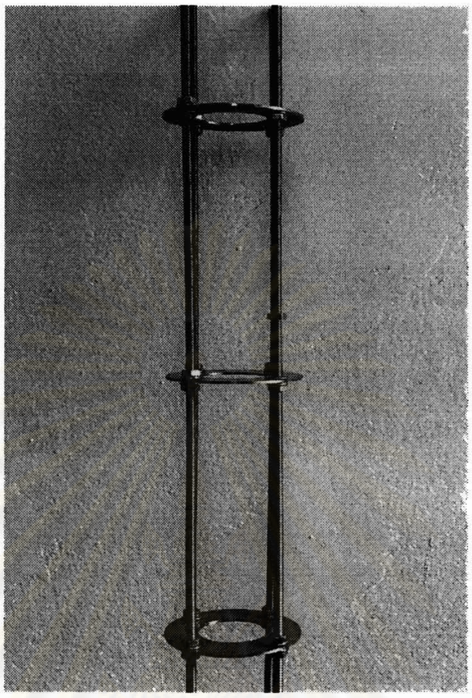
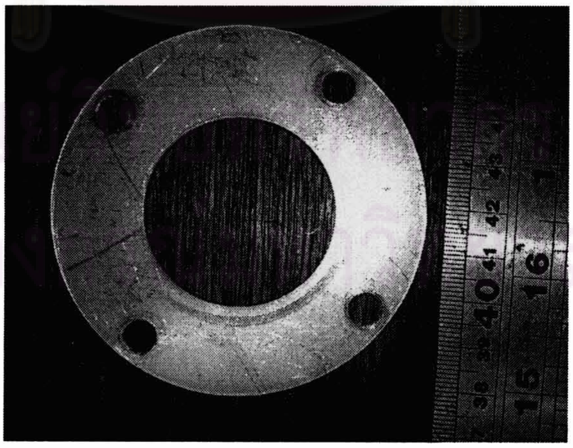
APPENDICES

APPENDIX A

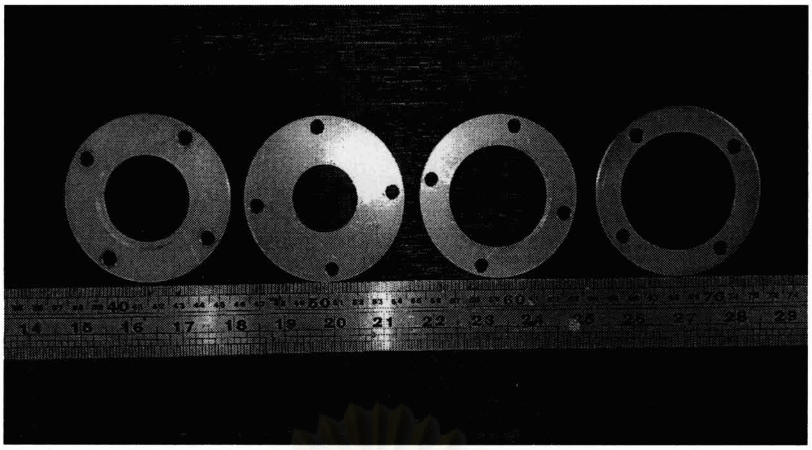
HEAT EXCHANGER



Figures A-1 The arrangement of baffles



Figures A-2 Baffles

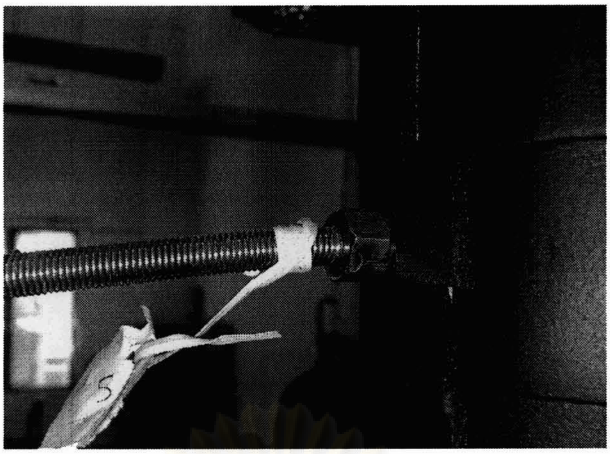


Figures A-3 The different type of baffles

Figures A-4 Cooling Tower

Figures A-5 Chiller

Figures A-6 Rotary meter

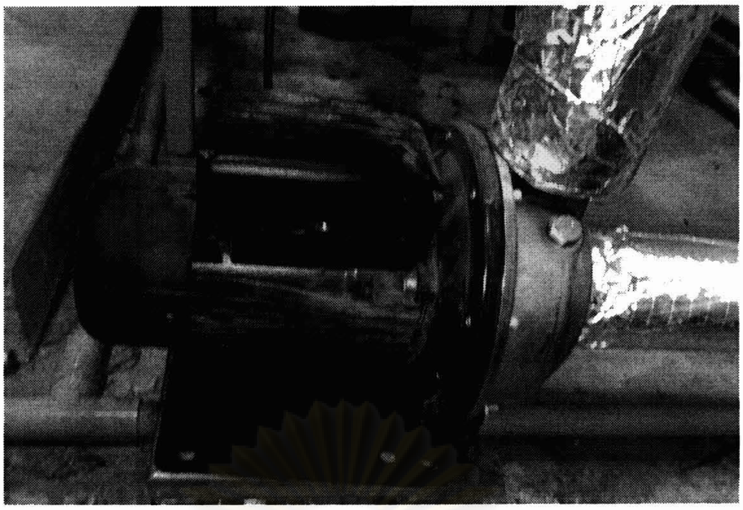


Figures A-7 Thermo couple

Figures A-8 Data Loggor

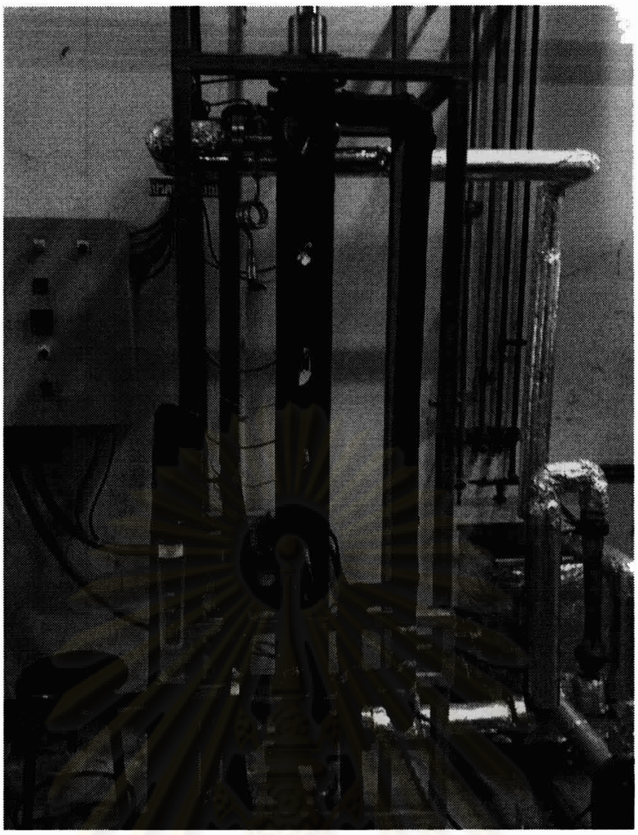
Figures A-9 Heater

Figures A-10 Heater box



Figures A-11 Pump

Figures A-12 Pressure transducer



Figures A-13 Heat exchanger

Figures A-14 Moter

APPENDIX B

EXAMPLE CALCULATE

As experiment first in Appendix D

<u>The property of baffle plat as below</u>

- The ratio inside diameter per inside diameter.

$$\alpha = \frac{d_i}{D_i} = 0.4$$

When d_i is inside diameter of baffle plate.

D_i is inside diameter of pipe.

- The ratio of spacing per inside diameter.

$$\beta = \frac{S}{D_i} = 1$$

When S is distance between baffle plate.

- Amplitude.

x_o = 0.0025 m

- Frequency

f = 0 Hz

<u>Hot flow rate</u>

F_h = 4 l/min

Cool flow rate

F_c = 5 l/min

Result of experiment

As experiment 200 (Appendix D) that show temperature ($^{\circ}$C) flowing .

$T_{h,i}$ = 78.7 $^{\circ}$C

$T_{h,o}$ = 49.2 $^{\circ}$C

$T_{c,o}$ = 25.9 $^{\circ}$C

$T_{c,i}$ = 21.5 $^{\circ}$C

Calculate.

Average temperature of hot flow.

$$T_{h,av} = \frac{T_{h,i} + T_{h,o}}{2} = 63.95$$

Average temperature of cool flow.

$$T_{c,av} = \frac{T_{c,i} + T_{c,o}}{2} = 23.7$$

Calculate of dimensionless.

At hot flow @ 63.95 °C

The property of paraffin oil as below

$$Cp_i = 2.1466(\text{kJ/kg}\cdot°C)$$

$$v_i = 5.4324505\text{x}10^{-6}(\text{m}^2/\text{s})$$

$$\rho_i = 824.965(\text{kg/m}^3)$$

$$k_i = 0.1325(\text{W/m}\cdot\text{K})$$

$$\mu_i = 4.481x10^{-3}(\text{kg/m}\cdot s)$$

Hot flow Reynolds Number

$$\text{Re}_h = \frac{\rho_i v_i D_i}{\mu_i} = 190.64$$

when $v_i = \dfrac{F_h(l/\min)}{A(m^2)} x \left|\dfrac{1m^3}{1000l}\right| x \left|\dfrac{1\min}{60s}\right| = 0.0126(\text{m/s})$

Oscillatory flow Reynolds Number

$$\text{Re}_{osc} = \frac{\rho_i 2\pi f x_o D_i}{\mu_i} = 0$$

From the first experiment $f = 0(Hz)$, $\therefore \ \mathrm{Re}_{osc} = 0$

At cool flow @ 23.7 °C

<u>The property of water as below</u>

$$Cp_o = 4.1779(\mathrm{kJ/kg}\cdot °C)$$

$$v_o = 9.7973\mathrm{x}10^{-7}(\mathrm{m^2/s})$$

$$\rho_o = 996.805(\mathrm{kg/m^3})$$

$$k_o = 0.61(\mathrm{W/m\cdot K})$$

$$\mu_o = 9.766x10^{-4}(\mathrm{kg/m}\cdot s)$$

<u>Cool flow Reynolds Number</u>

$$\mathrm{Re}_c = \frac{\rho_o v_o D_h}{\mu_o} = 515.92$$

when $D_h = \dfrac{4V_{wet}}{A_{wet}} = \dfrac{\pi(D_s^2 - D_i^2)L}{\pi(D_s + D_i)L} = 0.04$

$$v_o = \frac{F_h(l/\min)}{A(m^2)} x \left|\frac{1m^3}{1000l}\right| x \left|\frac{1\min}{60s}\right| = 0.01263(\mathrm{m/s}),\ A = \frac{\pi(D_s^2 - D_o^2)}{4} = 0.00659(m^2)$$

APPENDIX C

PROPERTY

1. Characteristic of heat exchanger

Inside diameter (inner pipe)

D_i = 0.082 m

Outside diameter (inner pipe)

D_o = 0.085 m

Inside diameter (outter pipe)

D_s = 0.125 m

Long

L = 1.25 m

Cross-section area inside diameter (hot flow)

$$A^i = \frac{\pi D_i^2}{4} = 0.00527 \quad m^2$$

Cross-section area outside diameter (cool flow)

$$A^o = \frac{\pi (D_s^2 - D_o^2)}{4} = 0.006594 \quad m^2$$

Wall area inside pipe

$$A_i = \pi D_i L = 0.3210 \quad m^2$$

Wall area outside pipe

$$A_o = \pi D_o L = 0.3336 \quad m^2$$

Volume inside pipe (hot flow)

$$V_i = \frac{\pi D_i^2 L}{4} = 0.0065875 \quad m^2$$

Volume outside pipe (cool flow)

$$V_o = \frac{\pi (D_s^2 - D_o^2) L}{4} = 0.008243 \quad m^2$$

Frequency

$$f = 0 - 2 \quad Hz$$

Amplitude

$$x_o = 0 - 0.01 \quad m$$

Spacing/Inner Diameter

$S/D_i = 1 - 2.5$

Orifice Diameter/Inner Diameter

$d_i/D_i = 0.4 - 0.7$

2. physical property of heat exchanger solute (hot flow)

Paraffin oil or white oil

Temperature inlet

$T_{h,in}$ = 80 °C

Temperature outlet

$T_{h,out}$ = 50 - 70 °C

Mass flow rate of paraffin oil

F_p = 4 – 15 liter/min

Velocity of paraffin oil

v_p = F_p/A_i = 0.01 – 0.06 m/s

Viscosity (20 - 100 °C)

μ = 3.5 – 35 cP

Density (20 - 100 °C)

ρ = 851 – 814 kg/m^3

Heat capacity (20 - 100 °C)

Cp = 1.994 – 2.21 kJ/kg-K

Thermal Conductivity (20 - 100 °C)

k = 0.136 – 0.131 W/m-K

TableC-1 Characteristic of paraffin oil when temperature changes

Temperature		Viscosity	Density		Heat Capacity		Thermal Conductivity	
ºF	ºC	cP	lb/ft3	kg/m3	BTU/lb-ºF	kJ/kg-K	BTU/ft-hr-ºF	W/m-K
50	10.0	75.697	53.55	857.81	0.467	1.956	0.0791	0.1369
60	15.6	50.182	53.33	854.41	0.472	1.975	0.0788	0.1364
65	18.3	41.886	53.23	852.72	0.474	1.985	0.0787	0.1362
70	21.1	35.431	53.13	851.12	0.477	1.994	0.0785	0.1359
80	26.7	26.191	52.92	847.72	0.481	2.013	0.0783	0.1355
90	32.2	20.055	52.71	844.42	0.486	2.033	0.078	0.135
100	37.8	15.489	52.50	841.03	0.490	2.051	0.0778	0.1347
110	43.3	12.087	52.29	837.63	0.497	2.078	0.0775	0.1342
120	48.9	9.734	52.08	834.33	0.501	2.097	0.0773	0.1338
130	54.4	7.973	51.87	830.93	0.506	2.116	0.077	0.1333
140	60.0	6.625	51.66	827.54	0.510	2.134	0.0768	0.1329
150	65.6	5.574	51.44	824.14	0.515	2.153	0.0765	0.1324
160	71.1	4.742	51.24	820.84	0.519	2.172	0.0762	0.1320
170	76.7	4.072	51.03	817.44	0.524	2.191	0.076	0.1315
180	82.2	3.526	50.81	814.05	0.528	2.210	0.0757	0.1311
190	87.8	3.077	50.60	810.65	0.533	2.229	0.0755	0.1306
200	93.3	2.703	50.39	807.25	0.537	2.247	0.0752	0.1302
210	98.9	2.432	50.18	803.95	0.542	2.266	0.0749	0.1297
220	104.4	2.191	49.97	800.56	0.546	2.285	0.0747	0.1293
230	110.0	1.983	49.76	797.16	0.551	2.304	0.0744	0.1288
240	115.6	1.801	49.55	793.76	0.555	2.323	0.0742	0.1284
250	121.1	1.642	49.34	790.47	0.560	2.342	0.0739	0.1279
260	126.7	1.503	49.13	787.07	0.564	2.36	0.0736	0.1275
270	132.2	1.382	49.02	785.37	0.569	2.379	0.0734	0.1270
280	137.8	1.270	48.71	780.27	0.573	2.398	0.0731	0.1266

290	143.3	1.172	48.49	776.88	0.578	2.417	0.0729	0.1261
300	148.9	1.085	48.29	773.58	0.582	2.436	0.0726	0.1257
310	154.4	1.010	48.08	770.18	0.587	2.455	0.0723	0.1252
320	160.0	0.945	47.86	766.78	0.591	2.473	0.0721	0.1248
330	165.6	0.885	47.65	763.39	0.596	2.492	0.0718	0.1243
340	171.1	0.832	47.45	760.09	0.600	2.511	0.0716	0.1239
350	176.7	0.783	47.23	756.69	0.605	2.530	0.0713	0.1234
360	182.2	0.737	47.02	753.30	0.609	2.549	0.0710	0.1230
370	187.8	0.696	46.81	749.90	0.614	2.568	0.0708	0.1225
380	193.3	0.658	46.60	746.60	0.618	2.586	0.0705	0.1221
390	198.9	0.623	46.39	743.20	0.623	2.605	0.0703	0.1216
400	204.4	0.590	46.18	739.81	0.627	2.624	0.0700	0.1212
410	210.0	0.559	45.97	736.41	0.632	2.643	0.0697	0.1207
420	215.6	0.532	45.76	733.01	0.636	2.662	0.0695	0.1203
430	221.1	0.507	45.55	729.71	0.641	2.681	0.0692	0.1198
440	226.7	0.483	45.34	726.32	0.645	2.699	0.0690	0.1194
450	232.2	0.460	45.13	722.92	0.650	2.718	0.0687	0.1189
460	237.8	0.439	44.91	719.52	0.654	2.737	0.0684	0.1185
470	243.3	0.420	44.71	716.23	0.659	2.756	0.0682	0.1180
480	248.9	0.402	44.50	712.83	0.663	2.775	0.0679	0.1176

3. Physical property of heat exchanger solute (cool flow)

Water

Temperature inlet

$T_{h,in}$ = 20 °C

Temperature outlet

$T_{h,out}$ = 20 – 40 °C

Mass flow rate of water

F_w = 5 – 25 liter/min

Velocity of water

v_p = F_p/A_i = 0.01 – 0.20 m/s

Viscosity (0 - 60 °C)

μ = 1.50 – 0.45 cP

Density (0 - 60 °C)

ρ = 1,000 – 980 kg/m^3

Heat capacity (0 - 60 °C)

Cp = 4.2 – 4.18 kJ/kg-K

Thermal Conductivity (0 - 60 °C)

k = 0.566 – 0.660 W/m-K

TableC-2 Characteristic of paraffin oil when temperature changes

Temperature		Viscosity	Density	Heat Capacity	Thermal Conductivity
ºF	ºC	Kg/m-s	kg/m3	kJ/kg-C	W/m-K
32	0	0.001790	999.8	4.225	0.566
40	4.4	0.001550	999.8	4.208	0.575
50	10.0	0.001310	999.2	4.195	0.585
60	15.5	0.001120	998.6	4.186	0.595
70	21.1	0.000980	887.4	4.179	0.604
80	26.6	0.000860	995.8	4.179	0.614
90	32.2	0.000765	994.9	4.174	0.623
100	37.7	0.000682	993.0	4.174	0.630
110	43.3	0.000616	990.6	4.174	0.637
120	48.8	0.000562	988.8	4.174	0.644
130	54.4	0.000513	985.7	4.179	0.649
140	60.0	0.000471	983.3	4.179	0.654
150	65.5	0.000430	980.3	4.183	0.659
160	71.1	0.000401	977.3	4.186	0.665
170	76.6	0.000372	973.7	4.191	0.668
180	82.2	0.000347	970.2	4.195	0.673
190	87.7	0.000327	966.7	4.199	0.675
200	93.3	0.000306	963.2	4.204	0.678
220	104.4	0.000267	955.1	4.216	0.684
240	115.6	0.000244	946.7	4.229	0.685
260	126.7	0.000219	937.2	4.250	0.685
280	137.8	0.000198	928.1	4.271	0.685
300	148.9	0.000186	918.0	4.296	0.684
350	176.7	0.000157	890.4	4.371	0.677
400	204.4	0.000136	859.4	4.467	0.665
450	232.2	0.000120	825.7	4.585	0.646

APPENDIX D

RESULT

The result parameter effect pressure in heat exchanger machine

Table D-1 The pressure drop at different Reynolds number: Vary the ratio of spacing per inside pipe diameter. $(S/D_i) = 1.0$, $d_i = 0.5D_i$, $x_o = 0.75$ cm, f = 1 Hz, Cool Flow = 20 l/min

Experiment 1	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8540	845.672	226.773	2205.902
2	8640	904.073	303.042	2217.956
3	8860	941.561	378.730	2198.680
4	8960	954.272	447.817	2196.275
5	9000	954.272	511.791	2193.871
6	9080	961.580	580.174	2181.863
7	9140	961.580	644.638	2189.065
8	9220	987.396	728.139	2191.467
9	5260	981.835	789.860	2208.311
10	6000	992.973	865.389	2193.871
11	6180	1002.300	940.710	2208.311
12	6500	1013.541	1019.209	2232.449

Table D-2 The pressure drop at different Reynolds number: Vary the ratio of spacing per inside pipe diameter. $(S/D_i) = 1.5$, $d_i = 0.5D_i$, $x_o = 0.75$ cm, f = 1 Hz, Cool Flow = 20 l/min

Experiment 2	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8490	699.113	187.472	2143.142
2	8580	781.315	261.894	2157.927
3	8720	816.983	328.620	2180.904
4	8840	836.678	392.633	2208.311
5	8860	848.352	454.984	2211.203
6	8900	866.452	522.778	2201.568
7	8960	878.284	588.797	2222.784
8	9080	884.853	652.521	2174.671
9	9440	898.801	723.062	2217.956
10	5740	911.135	794.066	2234.868
11	5920	905.835	850.173	2220.370
12	6220	888.325	893.292	2246.970

Table D-3 The pressure drop at different Reynolds number: Vary the ratio of spacing per inside pipe diameter (S/D_i) = 2.0, $d_i = 0.5D_i$, x_o = 0.75 cm, f = 1 Hz, Cool Flow = 20 l/min

Experiment 3	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8220	695.491	186.501	2181.863
2	8320	771.390	258.568	2182.343
3	8440	804.079	323.430	2180.424
4	8500	813.417	381.717	2202.049
5	8500	824.158	442.009	2179.945
6	8600	834.360	503.415	2181.384
7	8660	845.003	566.485	2198.199
8	8720	854.407	630.069	2216.509
9	8460	855.082	687.891	2178.985
10	5200	865.940	754.678	2184.743
11	5500	869.697	816.257	2177.547
12	5860	871.410	876.282	2215.544

Table D-4 The pressure drop at different Reynolds number: Vary the ratio of spacing per inside pipe diameter (S/D_i) = 2.5, $d_i = 0.5D_i$, x_o = 0.75 cm, f = 1 Hz, Cool Flow = 20 l/min

Experiment 4	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8880	837.342	224.539	2165.096
2	8980	890.065	298.347	2165.096
3	9240	916.456	368.632	2198.680
4	9280	911.135	427.574	2196.275
5	9240	920.016	493.419	2225.199
6	9300	936.145	564.828	2186.663
7	9300	937.948	628.795	2189.065
8	9340	950.630	701.027	2220.370
9	4980	985.541	792.841	2251.817
10	5680	979.984	854.069	2246.970
11	6020	978.136	918.031	2222.784
12	6440	979.984	985.464	2254.241

Table D-5 The pressure drop at different Reynolds number: Vary the ratio of inside baffle plate diameter per inside pipe diameter (d_i/D_i) = 0.4, S = $1.5D_i$, x_o = 0.75 cm, f = 1 Hz, Cool Flow = 20 l/min

Experiment 5	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8680	200.171	161.032	2191.467
2	8860	229.986	231.272	2179.465
3	9000	235.566	284.260	2210.721
4	9180	242.739	341.735	2205.902
5	9400	243.712	392.120	2234.868
6	9660	241.287	436.745	2246.97
7	9720	248.637	500.054	2254.241
8	9760	252.142	557.815	2232.449
9	6000	250.133	603.677	2193.871
10	6500	259.815	679.297	2239.706
11	6800	266.111	749.277	2217.956
12	7420	264.524	798.009	2254.241

Table D-6 The pressure drop at different Reynolds number: Vary the ratio of inside baffle plate diameter per inside pipe diameter (d_i/D_i) = 0.5, S = 1.5D_i, xo = 0.75 cm, f = 1 Hz, Cool Flow = 20 l/min

Experiment 6	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8060	240.612	193.566	2143.142
2	8320	262.421	263.889	2157.927
3	8380	272.870	329.275	2180.904
4	8520	269.311	379.143	2208.311
5	8620	272.870	439.033	2211.203
6	8740	278.120	503.416	2201.568
7	8840	283.344	569.856	2222.784
8	9060	285.253	631.065	2174.671
9	8640	288.647	696.626	2217.956
10	5200	295.067	771.463	2234.868
11	5300	293.337	825.935	2220.37
12	5820	294.720	889.104	2246.97

Table D-7 The pressure drop at different Reynolds number: Vary the ratio of inside baffle plate diameter per inside pipe diameter (d_i/D_i) = 0.6, S = 1.5D, x_o = 0.75 cm, f = 1 Hz, Cool Flow = 20 l/min

Experiment 7	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8620	219.680	176.727	2131.727
2	8800	247.149	248.531	2174.671
3	8940	274.174	330.849	2210.721
4	8720	277.459	390.615	2217.956
5	8920	279.668	449.970	2210.721
6	9020	291.499	527.633	2191.467
7	9040	296.109	595.529	2239.706
8	9100	300.771	665.397	2201.087
9	5140	306.672	740.128	2198.68
10	5380	306.672	801.806	2222.784
11	5460	315.061	887.103	2227.615
12	6040	315.665	952.292	2251.817

Table D-8 The pressure drop at different Reynolds number: Vary the ratio of inside baffle plate diameter per inside pipe diameter (d_i/D_i) = 0.7, S = 1.5D_i, x_o = 0.75 cm, f = 1 Hz, Cool Flow = 20 l/min

Experiment 8	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8440	237.933	191.411	2143.618
2	8480	233.691	234.997	2169.882
3	8500	252.647	304.872	2186.663
4	8540	260.334	366.506	2208.311
5	8560	259.815	418.029	2213.132
6	8660	265.052	479.761	2210.721
7	8720	263.471	529.888	2217.956
8	8940	264.524	585.207	2165.096
9	4760	268.240	647.376	2215.544
10	4880	274.174	716.840	2225.199
11	5120	276.361	778.136	2179.465
12	5580	281.334	848.720	2230.032

Table D-9 The pressure drop at different Reynolds number: Vary amplitude (x_o) = 0 cm, d_i/D_i = 0.5, S = 1.5D_i, f = 1 Hz, Cool Flow = 20 l/min

Experiment 9	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8560	0	193.334	2166.053
2	8600	0	260.434	2145.523
3	8600	0	321.631	2177.547
4	8700	0	380.125	2203.494
5	8920	0	446.948	2181.863
6	8940	0	504.614	2210.721
7	8980	0	577.328	2217.956
8	4400	0	628.826	2193.871
9	4540	0	698.273	2221.335
10	4920	0	757.058	2187.624
11	5180	0	825.287	2188.104
12	5500	0	894.342	2182.823

Table D-10 The pressure drop at different Reynolds number: Vary amplitude (x_o) = 0.25 cm, di/D_i = 0.5, S = 1.5D_i, f = 1 Hz, Cool Flow = 20 l/min

Experiment 10	**Pressure drop**	**Reynolds Number**		
	Pascal (Pa)	**Oscillatory (Re_{os})**	**Hot flow (Re_h)**	**Cool flow (Re_c)**
1	8620	240.612	193.566	2145.047
2	8660	262.421	263.888	2196.275
3	8730	272.869	329.274	2156.972
4	8820	269.310	379.143	2194.832
5	8920	272.869	439.033	2209.275
6	8997	278.120	503.415	2199.162
7	8940	283.343	569.856	2200.605
8	8840	285.252	631.065	2179.945
9	8640	288.646	696.626	2215.544
10	5200	295.066	771.463	2177.068
11	5450	293.336	825.935	2198.68
12	5720	294.720	889.104	2235.835

Table D-11 The pressure drop at different Reynolds number: Vary amplitude (x_o) = 0.50 cm, d_i/D_i = 0.5, S = 1.5D_i, f = 1 Hz, Cool Flow = 20 l/min

Experiment 11	Overall heat transfer coefficient	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8400	472.264	189.962	2175.150
2	8440	520.668	261.790	2150.290
3	8553	538.192	324.721	2177.068
4	8700	545.739	384.154	2176.588
5	8700	547.260	440.256	2161.271
6	8760	555.799	503.016	2181.863
7	8740	566.239	569.405	2212.167
8	8440	578.886	640.336	2210.721
9	8300	582.083	702.405	2228.582
10	5240	582.769	761.837	2187.143
11	5360	591.753	833.087	2236.319
12	5580	594.306	896.443	2229.065

Table D-12 The pressure drop at different Reynolds number: Vary amplitude (x_o) = 0.75 cm, d_i/D_i = 0.5, S = 1.5D_i, f = 1 Hz, Cool Flow = 20 l/min

Experiment 12	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8680	699.113	187.472	2143.142
2	8780	781.315	261.894	2157.927
3	8860	816.983	328.620	2180.904
4	9000	836.678	392.633	2208.311
5	9320	848.352	454.984	2211.203
6	9480	866.452	522.778	2201.568
7	9620	878.284	588.797	2222.784
8	9360	884.853	652.521	2174.671
9	9340	898.801	723.062	2217.956
10	5740	911.135	794.066	2234.868
11	5920	905.835	850.173	2220.370
12	6290	888.325	893.292	2246.970

Table D-13 The pressure drop at different Reynolds number: Vary amplitude (x_o) = 1 cm, d_f/D_i = 0.5, S = 1.5D_i, f = 1 Hz, Cool Flow = 20 l/min

Experiment 13	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8580	1024.828	206.1118	2205.902
2	8740	1118.670	281.2315	2234.868
3	8860	1154.587	348.3131	2201.087
4	8920	1163.709	409.5756	2249.393
5	8960	1186.754	477.356	2210.721
6	8980	1200.742	543.3553	2208.311
7	9120	1210.132	608.4497	2263.944
8	9360	1233.83	682.4009	2215.544
9	9320	1255.415	757.4611	2256.666
10	5420	1269.934	830.0728	2271.228
11	5820	1267.507	892.2164	2276.087
12	6220	1282.107	966.9576	2256.666

Table D-14 The pressure drop at different Reynolds number: frequency (f) = 0 Hz., d_i/D_i = 0.5, S = 1.5D_i, x_o = 1.5 cm., Cool Flow = 20 l/min

Experiment 14	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8560	0	193.334	2166.053
2	8580	0	260.434	2145.523
3	8600	0	321.631	2177.547
4	8700	0	380.125	2203.494
5	8920	0	446.948	2181.863
6	8930	0	504.614	2210.721
7	8940	0	577.328	2217.956
8	4400	0	628.826	2193.871
9	4540	0	698.273	2221.335
10	4920	0	757.058	2187.624
11	5180	0	825.287	2188.104
12	5500	0	894.342	2182.823

Table D-15 The pressure drop at different Reynolds number: frequency (f) = 0.5 Hz., d_i/D_i = 0.5, S = $1.5D_i$, x_o = 1.5 cm., Cool Flow = 20 l/min

Experiment 15	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8440	351.237	188.374	2131.727
2	8560	383.543	257.125	2160.315
3	8720	403.162	324.333	2184.263
4	8820	412.079	386.758	2157.927
5	8840	416.188	446.417	2174.671
6	8920	422.836	510.240	2184.263
7	8980	428.724	574.828	2193.871
8	5060	432.118	637.317	2222.784
9	5405	436.390	702.129	2225.199
10	5620	449.400	783.317	2213.132
11	5840	447.649	840.285	2222.784
12	6060	450.278	905.592	2239.706

Table D-16 The pressure drop at different Reynolds number: frequency (f) = 1.0 Hz., d_i/D_i = 0.5, S = 1.5D_i, x_o = 1.5 cm., Cool Flow = 20 l/min

Experiment 16	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	8440	699.113	187.472	2143.142
2	8523	781.315	261.894	2157.927
3	8740	816.983	328.620	2180.904
4	8680	836.678	392.633	2208.311
5	8770	848.352	454.984	2211.203
6	8840	866.452	522.778	2201.568
7	8860	878.284	588.797	2222.784
8	9580	884.853	652.521	2174.671
9	5450	898.801	723.062	2217.956
10	5660	911.135	794.066	2234.868
11	5870	905.835	850.173	2220.370
12	6090	888.325	893.292	2246.970

Table D-17 The pressure drop at different Reynolds number: frequency (f) = 1.5 Hz., d_i/D_i = 0.5, S = 1.5D_i, x_o = 1.5 cm., Cool Flow = 20 l/min

Experiment 17	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	9380	1132.373	202.436	2205.902
2	9420	1211.898	270.816	2181.863
3	9506	1298.911	348.313	2210.721
4	9590	1322.087	413.616	2191.467
5	9680	1327.28	474.560	2232.449
6	9760	1361.399	547.604	2225.199
7	9820	1348.203	602.551	2203.494
8	5790	1372.021	674.516	2242.127
9	6560	1385.377	742.999	2198.68
10	6700	1393.431	809.595	2227.615
11	6800	1404.218	878.621	2186.663
12	7060	1406.923	943.193	2210.721

Table D-18 The pressure drop at different Reynolds number: frequency (f) = 2.0 Hz., d_i/D_i = 0.5, S = 1.5D_i, x_o = 1.5 cm., Cool Flow = 20 l/min

Experiment 18	Pressure drop	Reynolds Number		
	Pascal (Pa)	Oscillatory (Re_{os})	Hot flow (Re_h)	Cool flow (Re_c)
1	9440	1421.900	190.646	2181.863
2	9440	1531.110	256.611	2193.871
3	9520	1587.143	319.203	2201.087
4	9498	1625.540	381.413	2215.544
5	9500	1684.664	451.755	2193.871
6	9530	1759.329	530.750	2189.065
7	9660	1794.099	601.377	2196.275
8	9630	1847.169	681.082	2205.902
9	6080	1886.742	758.916	2203.494
10	6340	1908.545	831.660	2227.615
11	6500	1908.545	895.634	2208.311
12	6650	1926.825	968.799	2196.275

BIOGRAPHY

Miss Sareya sornviboonsak was born on 2nd April, 1983 in Bangkok. She finished her secondary course from Thanyarat School in March, 2001. After that, she studied in the major of Industrial Chemistry in Faculty of Science at King Mongkut's Institute of Technology North Bangkok. She continued her further study for master's degree in Chemical Engineering at Chulalongkorn University. She participated in the Control and systems Engineering Research Group and achieved her Master's degree in April, 2010.