Effect of multiple jet impingement plate configurations on Reynolds Number in a pipe

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Abstract. Experimental investigations were carried out to study the effect of varying multiple jet impingement plate configurations on Reynolds Number (Re) in a closed conduit. Air was considered as the working fluid. There were six multiple impingement plates used for this experiment where each plate has a different hole configurations that include the hole diameter, hole orientation, pitch in x-direction and pitch in y-direction. Four sets of orifice plate with diameter of 0.02, 0.03, 0.04, and 0.05 m were used to get the mass flow rate in the pipe. Air was sucked through the impingement plate for five different settings of suction fan with an interval of 10Hz from 10 to 50Hz. By taking the data for constant suction fan setting at 50Hz, it was found that the impingement hole orientation for both in-line and staggered does not give any effect on the Re obtained since the differences was considerably small and fell within the accepted errors. Meanwhile, impingement hole diameter was found to be directly proportional with the Re obtained. It was also found that the different pitch in multiple hole impingement plate resulted in changes of Re. The results show that the Re was decreasing with higher pitch. The uncertainty analyses for the Re were also presented.

1. Introduction
In the field of pipes flow measurement, Reynolds number (Re) is one of the important and critical parameters to be determined. The effect of Re as one of the important parameters in jet impingement heat transfer field has been widely discussed and can be found in numerous literatures and books. The effect of Re on the development of flow structures in round jets has been explored by Dimotakis [1] who came out with the proposed critical value of Re = 10000, beyond which flow properties such as mixing become a weak dependence of Re.

Kwon and Seo [2] investigated the effect of Re on the behaviour of round jets using particle image velocimetry (PIV). Applying a range of low Re = 177–5142, they found out that the length of the zone of flow development decreased with increasing Re centerline velocity decayed more rapidly. Lee et al. [3] investigated the effects of jet-to-target plate distance and Re on the heat transfer from an array of jets impinging on a flat plate. They found out that the local and spatially averaged Nusselt numbers (Nu) showed strong dependence on the impingement jet Re for all situations examined. Goldstein et al. [4] investigated an injection of air perpendicular to the target surface. They dealt with high range of Re (60000 – 100000). They concluded that even with high jet Re, the dependency of Nu over jet-to-target distance did not change. They also gave the correlation of heat transfer as Nu/Re0.57 for all jet-to-target distance values. Meanwhile, Goldstein and Seol [5] concluded that the dependence of Nu on the jet Re could be approximated by Nu/Re0.7. Furthermore, Mohd Saiah [6] investigated single jet impingement heat transfer with Re ranging from 20000 to 30000. It was found that the optimal jet impingement Re...
for 5 mm hole diameter was at 20567 and any increase in Re would result in overdeveloped potential core, which lowered the heat transfer process [6]. The optimal jet impingement Re for 10 mm hole diameter was found to be at 28567 and lower Re would result in an underdeveloped potential core.

All the aforementioned literatures has proven that the effect of Re is very important to fluid flow experiments. Inspired by the work of Mohd Saiah [6], which involved a single hole jet impingement heat transfer, this study will investigate the effect of multiple hole impingement configurations on the pipe Re. There are three main configurations for impingement plate, which are the hole diameter, pitch and orientation. The effect of these three configurations will be investigated and examined.

2. Methodology

Re for internal flow in a circular pipe can be expressed as:

\[ Re = \frac{V D}{\nu} \]  

(1)

where \( V \) = mean velocity in m/s, \( D \) = characteristic length of the geometry (diameter in this case) in m and \( \nu = \mu/\rho \) = kinematic viscosity of the fluid in m\(^2\)/s.

There are three types of flow for Re, which are laminar, transition and turbulent flow. According to Yunus and Robert [7], the flow in a circular pipe is laminar for Re < 2300, turbulent for Re > 4000, and transitional in between.

- Re < 2300 laminar flow
- 2300 \( \leq \) Re \( \leq \) 4000 transitional flow
- Re > 4000 turbulent flow

The design of the experimental rig was based on Son et al. [8] and Mohd Saiah [6]. Although the objective of this study is not similar with the aforementioned researchers, design of their experimental set-up can still be used since it can determine the flow characteristics such as fluid mass flow rate, fluid velocity and Re.

The British standard BS1042 [9] was chosen as a reference document since it provides the detailed geometry parameters for fabrication, instalment requirements and working conditions for primary devices. The conduit used in this study is a straight cylindrical passage of constant cross-sectional area and has a length that ensures the upstream fluid flow approaching the orifice plate is fully developed profile and free of swirl. This length is also known as hydrodynamic entry length, \( L_h \) that is commonly taken to be the distance from the pipe inlet to the point at which the velocity boundary layer merges at the centreline as shown in Figure 1 [7].

![Figure 1. The development of the velocity boundary layer in a pipe [7]](image)

The hydrodynamic entry length is different for both laminar and turbulent flow. According to Kays and Crawford [10], Shah and Bhatti [11] and also Zhi-qing [12], the approximate length of \( L_h \) for laminar and turbulent flow can be approximated as \( 115D_i \) and \( 10D_i \), respectively. However, the pipes
used in practice are commonly several times the length of the entrance region, so that the flow through the pipes is fully developed. In this study, the recommended length for upstream and downstream sections based on BS1042 were $20D_1$ and $6D_1$, respectively [9], where the $D_1$ for the pipe was equal to 0.1 m. Figure 2 shows a schematic diagram of the experimental rig set-up.

Figure 2. Schematic diagram of the experimental rig set-up

2.1. Intake nozzle
In this study, the chosen type for intake nozzle was ISA1932. The intake nozzle, as the name implies, directs the fluid into the test rig in an appropriate manner. The intake nozzle was situated at the most front section of the test rig. The earlier section of the test rig was a rectangular shape conduit, hence the cross-sectional diameter of the conduit at this section will be replaced by the hydraulic diameter, $D_H$ that Holman [13] defined as:

\[
D_H = \frac{4A}{P}
\]

(2)

\[
D_H = \frac{4(ab)}{2a + 2b}
\]

(3)

where in Equation (2), $A$ is the cross-sectional area of the conduit and $P$ is the wetted perimeter of the conduit. Equation (2) can also be rearranged to Equation (3), in which $a$ and $b$ are both the sides of a rectangular. The inner cross-sectional area for the intake nozzle was 100 mm x 100 mm.

2.2. Impingement plates
There are six multiple holes impingement plates that were used in this study. Each impingement plate had a different configuration in terms of hole diameter, hole orientation as well as pitch in $x$- and $y$-direction as listed in Table 1. Figure 3a and Figure 3b illustrate the overview of the impingement plate. Note that the pitch in $x$- and $y$-direction is the center-to-center spacing of holes.

2.3. Orifice plates
Orifice plate can be defined in various ways [9, 14]. The main important characteristic of this primary device is that it provides an abrupt change in the cross-sectional area of the conduit and thus results in a change of energy. Orifice plate had become the most preferable flow meter due to its versatility, cost effectiveness and simple construction.
Table 1. Six multiple holes impingement plates with different configurations

| Impingement Plates | Hole Diameter (mm) | Hole orientation | Pitch x-direction (mm) | Pitch y-direction (mm) | Number of holes |
|--------------------|-------------------|------------------|------------------------|------------------------|-----------------|
| A                  | 7.0               | in-line          | 21.0                   | 26.6                   | 15              |
| B                  | 7.0               | Staggered        | 21.0                   | 26.6                   | 17              |
| C                  | 5.0               | in-line          | 15.0                   | 20.0                   | 25              |
| D                  | 5.0               | Staggered        | 15.0                   | 20.0                   | 23              |
| E                  | 5.0               | in-line          | 20.0                   | 25.0                   | 15              |
| F                  | 5.0               | Staggered        | 20.0                   | 25.0                   | 17              |

Figure 3. (a) Multiple holes impingement plate with in-line orientation, (b) Multiple holes impingement plate with staggered orientation
There is a total of four orifice plates used in this study, which each of them has a diameter of 0.02, 0.03, 0.04 and 0.05 m, respectively. Pressure measurements on orifice plate can be divided into three different approaches [9]. For this study, one approach was selected due to its simplicity, which was the \(D_2/2\) and \(D_1\) pressure taps, where \(D_1\) is the pipe’s inner diameter and \(D_2\) is the orifice diameter as shown in Figure 4.

![Diagram of orifice plate pressure taps](image)

**Figure 4.** Orifice plate pressure tapings

The principle of an orifice plate is based on the measurement of static pressure difference between the upstream and downstream sides. Based on ISO5167 [15] and ASME MFC-3M [16], the mass flow rate, \(q_m\) in pipe by using an orifice plate can be determined with the following equation:

\[
q_m = CA_2 \left( \frac{2\Delta p \rho_1}{1 - \beta^4} \right)^{1/2}
\]

where \(C\) is the discharge coefficient, \(A_2\) is the orifice area, \(\rho_1\) is the density at upstream side and \(\beta\) is the diameter ratio between pipe and orifice which is \(D_2/D_1\). Here the differential pressure, \(\Delta p\) can be measured by means of a digital manometer HHP 2021. Note that \(C\) is dependant of the diameter ratio, \(\beta\) and the pipe’s \(Re\). Equation (5) shows the governing relationship for discharge coefficient, \(C\) [9].

\[
C = 0.5959 + 0.0312 \beta^{2.1} - 0.1840 \beta^8 + 0.0029 \beta^{2.5} \left[ \frac{10^6}{Re} \right]^{0.75} + 0.09L_1 \beta^4 \left(1 - \beta^4 \right)^{-1} - 0.0337 L'_2 \beta^3
\]

in which \(L_1\) is a ratio of \(l_1/D_1\), \(L_2\) is a ratio of \(l'_2/D_1\) with \(l'_2\) is the distance of the downstream pressure tap from the downstream face of the orifice plate. Whenever \(L_1 \geq \frac{0.0390}{0.0900}\), 0.0390 will be used for the term \(\beta^4 \left(1 - \beta^4 \right)^{-1}\). In this study, \(L_1\) equals to 1, therefore the orifice discharge coefficient yields:

\[
C = 0.5959 + 0.0312 \beta^{2.1} - 0.1840 \beta^8 + 0.0029 \beta^{2.5} \left[ \frac{10^6}{Re} \right]^{0.75} + 0.0900L_1 - 0.0337 L'_2 \beta^3
\]

The calculation of \(C\) is an iterative process since \(Re\) is needed to calculate \(C\), but the velocity in the pipe needed for \(Re\) cannot be determined until \(C\) is known. However, this iterative process converges quite rapidly in just one or two iterations.
2.4. Uncertainty analysis

Predictions of errors are vital for researchers, especially for those who are dealing with experimental works. Experimentation is not just about taking the data. Each instrument has its own accuracy, hence the prediction of uncertainties is dominated by the accuracy errors that are also known as the random errors. The uncertainty analysis will provide the information for experimenters in knowing the scatter and how far the experimental data may deviate from the true value.

For this study, uncertainty analyses for $Re$ obtained have been done by using Taylor Series Method (TSM) for propagation of uncertainties. Equation (7) is given by Hugh and Glenn [17] that describes the propagation of overall uncertainties in measured variables into the overall uncertainty of the result.

$$U_r = \left[ \sum_{i=1}^{J} \left( \frac{\partial r}{\partial X_i} U_i \right)^2 \right]^{1/2} \tag{7}$$

where $r$ is the function of two or more variables and $U_i$ is the uncertainty for variable $X_i$.

Referring to Equation (8) for $Re$, there are only two variables that have been considered as sources of error, which are air velocity, $V$ and pipe diameter, $D$ while air density, $\rho$ and air viscosity, $\mu$ are assumed to be constant.

$$Re = \frac{\rho V D}{\mu} \tag{8}$$

The air velocity, $V$ can be defined from the mass flow rate, $q_m$ as shown in Equation (9).

$$V = \frac{q_m}{\rho A_1} \tag{9}$$

To get the uncertainty for $V$, the uncertainty for $q_m$ needs to be determined first. From the Equation (4), by using the general uncertainty analysis in Equation (7), the uncertainty for $q_m$ is determined as:

$$U_{q_m} = \left( \frac{\partial q_m}{\partial A_2} \right)^2 U_{A_2} \frac{2}{A_2} + \left( \frac{\partial q_m}{\partial \Delta p} \right)^2 U_{\Delta p} \frac{2}{\Delta p} + \left( \frac{\partial q_m}{\partial \beta} \right)^2 U_{\beta} \frac{2}{\beta} \tag{10}$$

By taking the derivatives of $q_m$ with respect to the variables, then substituting it into Equation (10) and dividing by $q_m^2$ yields:

$$\left( \frac{U_{q_m}}{q_m} \right)^2 = (1)^2 \left( \frac{U_{A_2}}{A_2} \right)^2 + \left( \frac{1}{2} \right)^2 \left( \frac{U_{\Delta p}}{\Delta p} \right)^2 + (2)^2 \left( \frac{U_{\beta}}{\beta} \right)^2 \tag{11}$$

Then, by applying the Equation (7) again to the Equation (9), the uncertainty for $V$ was determined as:

$$\left( \frac{U_V}{V} \right)^2 = (1)^2 \left( \frac{U_{q_m}}{q_m} \right)^2 + (1)^2 \left( \frac{U_{A_1}}{A_1} \right)^2 \tag{12}$$

Thus, the uncertainty for $Re$ was determined as:

$$\frac{U_{Re}}{Re} = \sqrt{ \left( \frac{U_V}{V} \right)^2 + \left( \frac{U_{D_1}}{D_1} \right)^2 } \tag{13}$$
Table 2 shows the accuracy of the measuring devices that have been used in this study.

| Measuring Devices       | Accuracy                                                                 |
|-------------------------|---------------------------------------------------------------------------|
| Digital Manometer HHP 2021 | 0.1% reading +0.1 % full scale +1 digit (full scale of 13 kPa)           |
| Digital Caliper 500-752-10  | ± 0.001" / 0.01 mm                                                       |
| Ruler                   | ± 0.05 cm                                                                |

3. Results and discussions

Table 3 shows the results of $Re$ with their uncertainties for each setting of orifice diameter and suction fan without impingement plate being attached. Meanwhile, Table 4 to Table 9 show the results of $Re$ with their uncertainties for each orifice diameter and suction fan setting for every impingement plate. Since pressure gives the most contribution to uncertainties values, the uncertainties percentage values obtained for $Re$ were considerably high at smaller pressure difference, which was up to ±115%. This was due to the fact that the range of the digital manometer used was quite high for this experiment and it only has better accuracy for larger pressure difference measurements. For that reason, only $Re$ with uncertainties below 30% were selected and $Re$ with uncertainties more than 30% were neglected.

Table 3. Reynolds number with uncertainties for each orifice diameter and suction fan setting without impingement plate

| Suction Fan Setting (Hz) | Orifice Diameter (m) | 0.02       | 0.03       | 0.04       | 0.05       |
|--------------------------|----------------------|------------|------------|------------|------------|
|                          |                      | Reynolds Number, $Re$ |            |            |            |
| 10                       | 0.02                 | 918 ± 57.57 %       | 1760 ± 115.06 % | null       | null       |
| 20                       | 0.03                 | 1710 ± 16.55 %      | 3290 ± 23.1 %   | 4620 ± 38.41 % | 4350 ± 115.06 % |
| 30                       | 0.04                 | 2570 ± 7.41 %       | 5070 ± 9.76 %   | 7010 ± 16.55 % | 8540 ± 28.84 % |
| 40                       | 0.05                 | 3400 ± 4.44 %       | 6840 ± 5.5 %    | 9870 ± 8.41 %  | 11200 ± 16.55 % |
| 50                       | 4210 ± 3.15 %        | 8500 ± 3.76 %       | 12100 ± 5.74 %  | 14000 ± 10.62 %|

Table 4. Reynolds number with uncertainties for each orifice diameter and suction fan setting for impingement plate A

| Suction Fan Setting (Hz) | Orifice Diameter (m) | 0.02       | 0.03       | 0.04       | 0.05       |
|--------------------------|----------------------|------------|------------|------------|------------|
|                          |                      | Reynolds Number, $Re$ |            |            |            |
| 10                       | 0.02                 | 652 ± 115.06 %       | null       | null       | null       |
| 20                       | 0.03                 | 1450 ± 23.1 %       | 2550 ± 38.42 % | 2700 ± 115.06 % | null       |
| 30                       | 0.04                 | 2320 ± 9.03 %       | 3880 ± 16.55 % | 4620 ± 38.42 % | 4350 ± 115.06 % |
| 40                       | 0.05                 | 3080 ± 5.29 %       | 5270 ± 9.03 %  | 6500 ± 19.28 % | 7420 ± 38.42 % |
| 50                       | 3800 ± 3.69 %        | 6530 ± 6.01 %       | 7940 ± 12.92 % | 8540 ± 28.84%  |
Table 5. Reynolds number with uncertainties for each orifice diameter and suction fan setting for impingement plate B

| Suction Fan Setting (Hz) | Orifice Diameter (m) | 0.02 | 0.03 | 0.04 | 0.05 |
|--------------------------|----------------------|------|------|------|------|
|                          | Reynolds Number, Re   | Re   | Re   | Re   | Re   |
|                          | 10                   | 652 ± 115.06 % | 1760 ± 115.06 % | null  | null |
|                          | 20                   | 1580 ± 19.28 % | 2550 ± 38.41 % | 2700 ± 115.06 % | null |
|                          | 30                   | 2410 ± 8.41 %  | 4150 ± 14.51 % | 5320 ± 28.84 % | 4350 ± 115.06 % |
|                          | 40                   | 3210 ± 4.91 %  | 5470 ± 8.41 %  | 7010 ± 16.55 % | 7420 ± 38.41 % |
|                          | 50                   | 3960 ± 3.46 %  | 6840 ± 5.5 %   | 8760 ± 10.62 % | 9530 ± 23.10 % |

Table 6. Reynolds number with uncertainties for each orifice diameter and suction fan setting for impingement plate C

| Suction Fan Setting (Hz) | Orifice Diameter (m) | 0.02 | 0.03 | 0.04 | 0.05 |
|--------------------------|----------------------|------|------|------|------|
|                          | Reynolds Number, Re   | Re   | Re   | Re   | Re   |
|                          | 10                   | 652 ± 115.06 % | null | null | null |
|                          | 20                   | 1450 ± 23.1 %  | 2550 ± 38.41 % | 2700 ± 115.06 % | null |
|                          | 30                   | 2320 ± 9.03 %  | 3600 ± 19.28 % | 4620 ± 38.41 % | 4350 ± 115.06 % |
|                          | 40                   | 3020 ± 5.51 %  | 4850 ± 10.62 % | 5940 ± 23.10 % | 6090 ± 57.57 % |
|                          | 50                   | 3750 ± 3.78 %  | 6200 ± 6.62 %  | 7490 ± 14.51 % | 7420 ± 38.41 % |

Table 7. Reynolds number with uncertainties for each orifice diameter and suction fan setting for impingement plate D

| Suction Fan Setting (Hz) | Orifice Diameter (m) | 0.02 | 0.03 | 0.04 | 0.05 |
|--------------------------|----------------------|------|------|------|------|
|                          | Reynolds Number, Re   | Re   | Re   | Re   | Re   |
|                          | 10                   | 652 ± 115.06 % | null | null | null |
|                          | 20                   | 1450 ± 23.1 %  | 2090 ± 57.57 % | 2700 ± 115.06 % | null |
|                          | 30                   | 2230 ± 9.76 %  | 3600 ± 19.28 % | 3790 ± 57.57 % | 4350 ± 1.53 % |
|                          | 40                   | 3020 ± 5.51 %  | 4850 ± 10.62 % | 5320 ± 28.84 % | 6090 ± 57.57 % |
|                          | 50                   | 3750 ± 3.78 %  | 6020 ± 6.99 %  | 7010 ± 16.55 % | 7420 ± 38.41 % |

Table 8. Reynolds number with uncertainties for each orifice diameter and suction fan setting for impingement plate E

| Suction Fan Setting (Hz) | Orifice Diameter (m) | 0.02 | 0.03 | 0.04 | 0.05 |
|--------------------------|----------------------|------|------|------|------|
|                          | Reynolds Number, Re   | Re   | Re   | Re   | Re   |
|                          | 10                   | 652 ± 115.06 % | null | null | null |
|                          | 20                   | 1290 ± 28.84 % | 1490 ± 115.06 % | null | null |
|                          | 30                   | 2040 ± 11.6 %  | 2940 ± 28.84 % | 2700 ± 115.06 % | null |
|                          | 40                   | 2730 ± 6.63 %  | 3880 ± 16.55 % | 3790 ± 57.57 % | 4350 ± 115.06 % |
|                          | 50                   | 3400 ± 4.44 %  | 4630 ± 11.66 % | 5320 ± 28.84 % | 4350 ± 115.06 % |
Table 9. Reynolds number with uncertainties for each orifice diameter and suction fan setting for impingement plate F

| Suction Fan Setting (Hz) | Orifice Diameter (m) | Reynolds Number, Re |
|-------------------------|----------------------|---------------------|
|                         | 0.02                 | 0.03                | 0.04                | 0.05                |
| 10                      | 652 ± 115.06 %       | null                | null                | null                |
| 20                      | 1290 ± 28.84 %       | 2090 ± 57.57 %      | 2700 ± 115.06 %     | null                |
| 30                      | 2140 ± 10.62 %       | 2940 ± 28.84 %      | 3790 ± 57.57 %      | 4350 ± 115.06 %     |
| 40                      | 2800 ± 6.3 %         | 4150 ± 14.51 %      | 4620 ± 38.41 %      | 4350 ± 115.06 %     |
| 50                      | 3520 ± 4.19 %        | 5070 ± 9.76 %       | 5940 ± 23.10 %      | 6090 ± 57.57 %      |

Figure 5 shows the Re at 50 Hz without impingement plate being attached. These values of Re can be used as a benchmark to other Re values obtained with impingement plates.

3.1. Effect of impingement plate hole orientations

Figure 6 (a), (b) and (c) show the comparison of Re values for impingement plates that have a similar hole diameter and pitch but different holes orientation. All three graphs obviously give a quite similar trend. At suction fan of 50 Hz, the Re for impingement plate A and B, C and D, and E and F increased similarly with an increase of orifice diameter regardless of the hole orientation. Though there were some slight differences between Re values at orifice diameter 0.04 m, they were considerably small (not more than 10%) and fell within the uncertainties. The small differences were due to higher holes number in the impingement plate B, which resulted in higher open area and fluid mass flow rate.

3.2. Effect of impingement plate hole pitch

Figure 7 (a) and (b) show the comparison of Re values for impingement plates that have similar hole diameter and orientation but different pitch. All two graphs obviously give a quite similar trend. At suction fan of 50 Hz, Re for impingement plate C and E, and D and F increased in different increments with an increase of orifice diameter. This shows that different pitch on the impingement plates resulted in different Re. This was due to the fact that by having higher hole pitch, it will reduce the number of hole in the fixed area of impingement plate. This will result in a lower open area in the impingement plate and thus the fluid mass flow rate through the impingement plate will be reduced.
Figure 6. (a) Comparison of $Re$ between in-line and staggered impingement plates A and B for 7mm hole diameter

Figure 6. (b) Comparison of $Re$ between in-line and staggered impingement plates C and D for 5mm hole diameter with 15mm pitch-x and 20mm pitch-y
Figure 6. (c) Comparison of \( Re \) between in-line and staggered impingement plates E and F for 5mm hole diameter with 20mm pitch-x and 25mm pitch-y

Figure 7. (a) Comparison of \( Re \) between 15mm pitch-x, 20mm pitch-y and 20mm pitch-x, 25mm pitch-y impingement plates C and E for in-line orientation
3.3. Effect of impingement plate hole diameter

Figure 8 (a) and (b) show the comparison of $Re$ values for impingement plates that have similar hole pitch and orientation but different diameter. Though the hole pitch between impingement plate A and E, and B and F were not exactly the same, their differences were small and would not give significant effect to the overall $Re$ since the numbers of hole were the same. Thus the effect of hole diameter can be investigated by using these impingement plates. All two graphs obviously show a rather similar trend. At suction fan of 50Hz, $Re$ for impingement plate A and E, and B and F increased in different increments with increase of orifice diameter. This shows different hole diameter on the impingement plates resulted in different $Re$. This was due to the fact that by having larger hole diameter will result in a higher open area of the impingement plate and thus higher fluid mass flow rate through the hole. Consequently, higher $Re$ will be obtained.

4. Conclusion

The overall goal of this experimental study was to examine the effect of having different configuration of multiple impingement plate on $Re$ in a pipe. The configurations include the holes diameter, hole pitch as well as hole orientation that were in-line and staggered. Based on obtained data for constant suction fan setting at 50Hz and for orifice diameters of 0.02m, 0.03m and 0.04m, it was found that for the fixed area of impingement plates, orientation of the impingement plate does not give different $Re$ as long as the number of impingement hole remains the same. Meanwhile, $Re$ was found to be lower when the holes pitch is higher. It shows that for fixed area of impingement plate, the holes pitch was inversely proportional with the $Re$. Also found in this study was that the impingement hole diameter gives a significant effect on $Re$ in a pipe. There was a directly proportional relationship between holes diameter with $Re$ obtained. The larger the holes diameter, the higher the $Re$ would be.
**Figure 8.** (a) Comparison of $Re$ between hole diameter of 7mm and 5mm impingement plates A and E for in-line orientation

**Figure 8.** (b) Comparison of $Re$ between hole diameter of 7mm and 5mm impingement plates B and F for staggered orientation
Acknowledgement
The first author likes to thank the Ministry of Higher Education Malaysia for provision of MyBrain15 scholarship.

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