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EFFECT OF TUBE DIAMETER AND SURFACE ROUGHNESSON FLUID FLOW FRICTION FACTOR

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Abstract

Experiments have been performed to investigate the effect of channel roughness and diameter on fluid friction. Three different diameters and roughness of tubes were used to examine the friction factor. The first tube made of stainless steel with an inner diameter of 1.14 mm was investigated at Brunel University, whilst the others made of PVC with diameters of 17 mm and 15.5 mm rough were tested at Mataram University. The stainless steel was equipped with a 200 mm calming section and smooth one. The 15.5 mm diameter tube was coated internally with sand that had an average grain size of 0.5 mm so that the tube had a relative roughness of 0.032. The last tube with a diameter of 17 mm was smooth as explained in the H408 Fluid Friction Experimental Apparatus manual.

The results indicate that the flow in the stainless steel tube still obeys the theory and in the 17 mm tube shows a deviation in friction factor with the theory. However, this was due to no calming section installed in the test rig. Flow in the rough tube (15.5 mm diameter) demonstrates that the Reynolds number does not affect the friction factor in turbulent regimes and the experimental friction factors were reasonably in a good agreement with the theory or Moody diagram. Hence, the effect of decreasing in diameter of channels on friction factor is insignificant.

Keywords: friction factor, surface roughness, fluid friction theory.

1. Background

Friction factors are important parameters in a process that utilizes fluid flow. This can cause high pressure drops which further cause high demands of pumping energy. Processes that use small tubes/channels need carefully pressure drop calculations because in smaller tubes, the pressure drop becomes significantly higher than in bigger tubes. Even, in microchannels the pressure drop is one of the major problems that must be solved or eliminated correctly and still being a concerned problem in this field study. Researches on this field are still challenging the microchannel communities.

Still there are many contradictory conclusions on the definition of microchannels. Previous studies used flow behaviour as criteria to define microchannels, e.g. Brauner and Moalem-Maron [1], Kewand Cornwell [2], and Peng and Wang [3]. On the other hand, some studies used a dimension of channels to differ microchannel from macrochannel. Mehendale et al. [4] classified heat exchangers in general, in terms of D_h:

(a) Micro heat exchanger: D_h= 1 – 100 µm.
(b) Meso heat exchanger: D_h= 100 µm – 1 mm.
(c) Compact/macro heat exchanger: D_h= 1 – 6 mm.
(d) Conventional heat exchanger: D_h> 6m.

Finally, based on engineering practice and application areas, such as refrigeration industry in small tonnage units, compact evaporators, cryogenic industries, cooling elements of microelectronics and microelectromechanical systems (MEMS), Kandlikar [5] subdivided channels into three groups in terms of D_h, as follows:

(a) Conventional channels: D_h> 3 mm.
(b) Minichannels: D_h= 200 µm – 3 mm.
(c) Microchannels: D_h= 10 – 200 µm.

Research results published in the open literatures are seem different each other. Some previous studies stated that friction factors in large diameter tubes differed from those in small/micro tubes. Some indicated higher friction factors, e.g. Urbanek et al. [6], Papautsky et al. [7], Pfund et al. [8], Shen et al. [9]. Other studies showed no distinction between experimental friction factors and theory, e.g. Silverio and Moreira [11], Akbari et al. [12], Mirmanto et al. [13]. However, several studies presented friction factors that are lower than the theory, e.g. Jiang et al. [14].

The reasons of being higher than the theory are usually due to entrance effect, roughness effect, dimension errors and flow measurements. Qu et al. [15] demonstrated relatively high friction factors from conventional theory when they measured pressure drops for water flowing in trapezoidal silicon microchannels with hydraulic diameters ranging from 51 µm to 169 µm. Their friction factors were about 8% to 38% higher than theory. However, they justified the deviation as being the result of the high relative roughness (3.5% to 5.7%). Jiang et al. [16], who used a microchannel with a hydraulic diameter of 300 µm, and Kandlikar et al. [17], who employed diameters of 1.06 mm and 0.62 mm tubes elucidated that because of the roughness their friction factors were higher than theory. However, Kandlikar et al. [17] specified that the effect of surface roughness (relative roughness of 0.36%) was significant only for the smallest diameter test section (0.62 mm). Shen et al. [9] studied flow and heat transfer for deionized water in rough-walled copper microchannels assembled in a 26-channel array. The rectangular channels were 300 µm wide and 800 µm deep. They applied three different inlet temperatures; 30^{o}C, 50^{o}C and 70^{o}C and their Reynolds numbers varied from 162 to 1257. They found that the effect of surface roughness (relative roughness 4 - 6%) on laminar flow was significant and that higher inlet temperatures decreased the pressure drop. However, they did not explain the effect of fluid temperature on the friction factor.

In this study, experimental friction factors obtained from flow in different diameters of the tube are compared each other and also compared with theory. The aims are to see if there are any differences of investigated friction factors found in each tube at the same Reynolds numbers. Furthermore, as the test sections used in this research were not equipped with calming sections, pressure drop predictions in the inlet and outlet plenums are analyzed and presented in the forms of graph.

2. Experimental Facility and Test Sections

Experiments using the 1.14 mm diameter tube were performed using the test rig at Brunel University, United Kingdom, see Fig. 1 in Mirmanto et al. [13]. The test rig used in this work consisted of a reservoir made of SS316, micropump (model Micropump GA-T23, PFSB), Coriolis flowmeter (model Micromotion Elite CMF010), preheaters and test sections. Deionized water was used as the working fluid that was set at 30°C. The water temperature was measured using K type thermocouple with an uncertainty of ±0.2 K calibrated against the Platinum Precision Thermometer with an accuracy of ±0.025 K. To obtain pressure drops, two pressure transducers model Honeywell 26PCCD with an uncertainty of ±0.2 kPa were installed in the inlet and outlet and were calibrated against the deadweight tester for high pressures and the water manometer for low pressures. The experiments using 15.5 mm and 17 mm tubes were conducted at Mataram University, Indonesia. The test rig used was H408 Fluid Friction Apparatus (see Fig. 1) and the test section were fabricated by TecQuipment Ltd, [10].

The lengths of the test sections were 200 mm for the 15.5 mm tube diameter (tapping 30 and 31) and 912 mm for the 17 mm tube diameter (tapping 7 and 8). The 15.5 mm tube was roughen using sand grain with an average size of 0.5 mm giving its relative roughness of 0.032, see Fig. 2. The pressure drop was measured using closed manometer with a resolution of 1 mm.

Three test sections employed were (i) a stainless steel tube with an inner diameter of 1.14 mm (measured using a TSER microscope with an accuracy of ±1 µm, see Fig. 3), 200 mm long, equipped with 200 mm long calming sections placed before and after the test section, (ii) a PVC tube with an inner diameter of 17 mm and a length of 912 mm without a calming section, (iii) a PVC rough tube with an effective diameter of 15.5 mm and a length of 200 mm without a calming section.

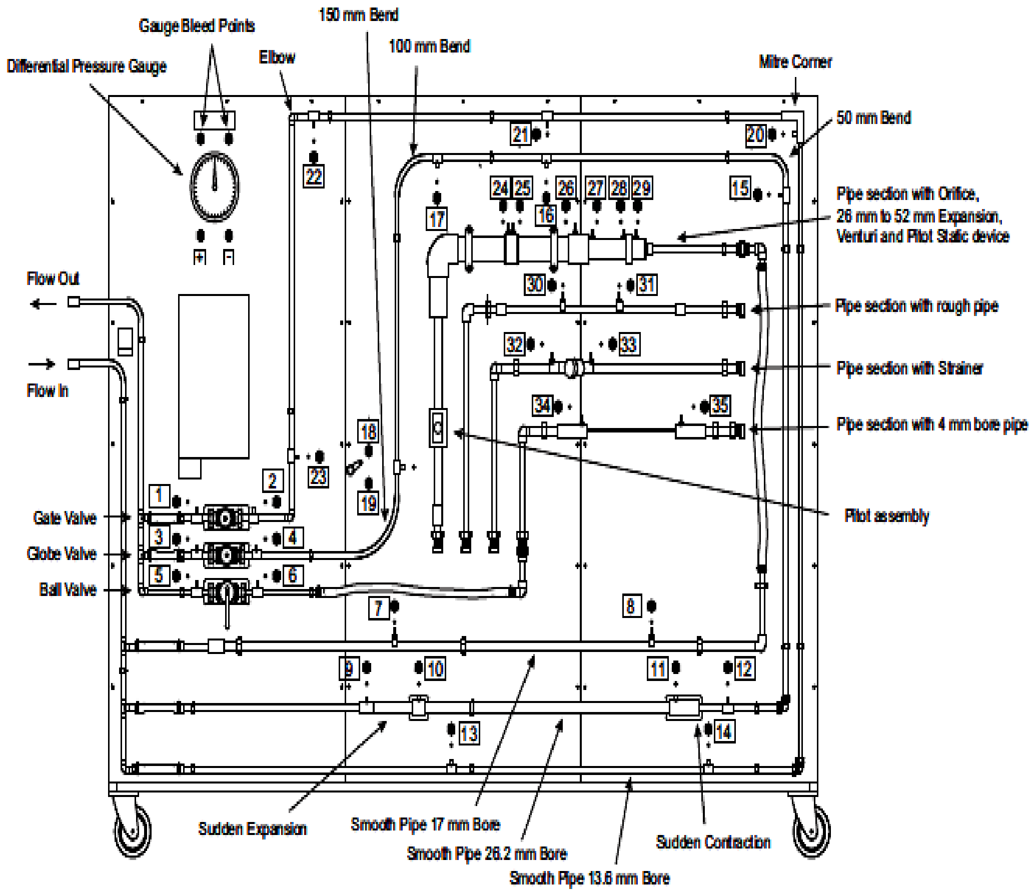

Figure 1. Experimental schematic diagram [10]

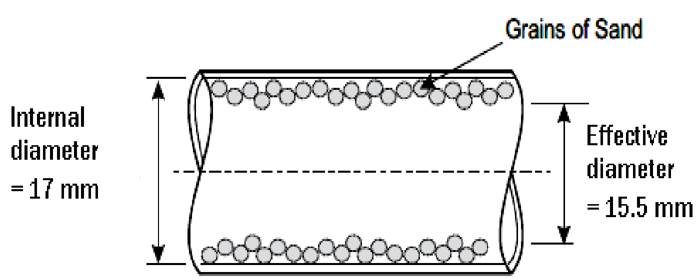

Figure 2. Rough pipe test section

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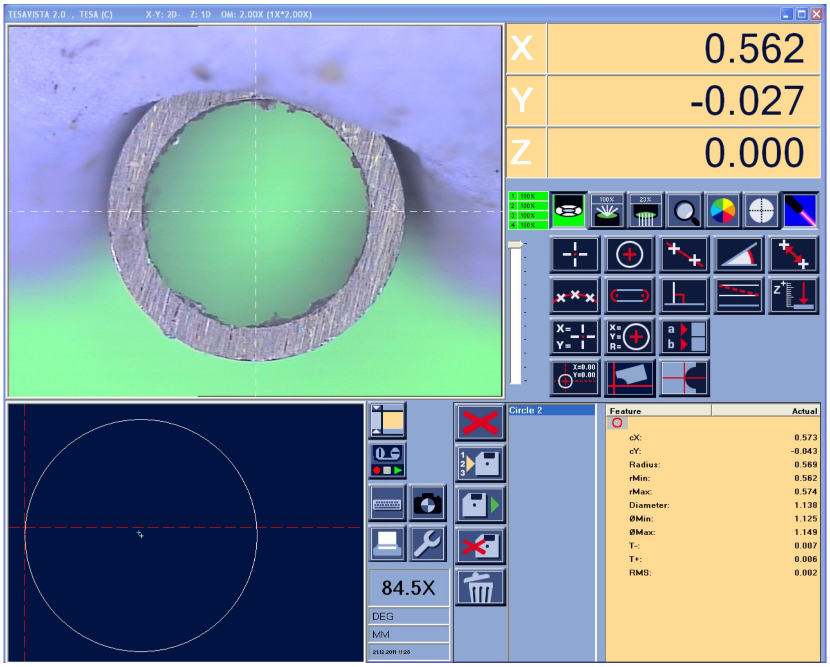

Figure 3. Stainless steel test section measured using a TSER microscope

3. Data Reduction

In this study, the inlet and outlet pressures were measured directly, therefore, the pressure drop is obtained by subtracting the outlet pressure from the inlet pressure and called as total pressure drop, taken from [18]:

$$\Delta p_t = p_i - p_o \quad (1)$$

wherep_i is the measured inlet pressure and $p_o$is the measured outlet pressure. The channel pressure drop is then determined using Eq. (3), however, for the test section that is equipped with a calming section, the channel pressure drop is the same as the total pressure drop, Eq. (2), taken from [12]:

$$\Delta p_{ch} = \Delta p_t \quad (2)$$

$$\Delta p_{ch} = \Delta p_t - \Delta p_i - \Delta p_o \quad (3)$$

the inlet and outlet pressure drops are due to the differences between inlet and outlet tube diameters and channel diameter. Meanwhile, in the outlet, there is a pressure recovery due to the deceleration of the fluid. The inlet and outlet pressure drops can be estimated as follows:

$$\Delta p_i = \Delta p_l + \Delta p_a = \rho k_l V_{ch}^2 / 2 + \rho V_{ch}^2 \left(1 - \left(A_{ch} / A_i\right)^2\right) / 2 \quad (4)$$

$$\Delta p_o = \Delta p_l - \Delta p_d = \rho k_l \left(V_{ch} - V_o\right)^2 / 2 - \rho V_{ch}^2 \left(1 - \left(A_{ch} / A_o\right)^2\right) / 2 \quad (5)$$

where $\Delta p_l$is the static pressure drop, $\Delta p_a$is the pressure drop due to fluid acceleration and $\Delta p_d$is the pressure recovery due to fluid deceleration. $A_{ch}$is the channel cross sectional area,A_i is the inlet plenum cross sectional area and $A_o$is the outlet plenum cross sectional area. $V_{ch}$is the average channel fluid velocity and $V_o$is the average outlet plenum fluid velocity. $k_l$is the loss coefficient that is equal to 1 for Eq. (5) and dependent on the ratio of channel diameter and inlet plenum diameter for Eq. (4), see Table 1, whilst ρ is the fluid density.

The experimental friction factor, f, is calculated using Eq. (6), which is given by (taken from [21]).

$$f = \frac{2 \Delta p_{ch} D_{ch}}{\rho L V_{ch}^2} \quad (6)$$

where L is the channel length and $D_{ch}$is the channel diameter. The friction factor theory in this study is the Darcy-Weisbach equation for laminar which is given by, taken from [21].

$$f = \frac{64}{\mathrm{Re}} \quad (7)$$

and for turbulent flow in the smooth pipe, the friction factor equation used is Blasius equation which is expressed as, taken from [21].

$$f = 0.316\,\mathrm{Re}^{-0.25} \quad (8)$$

whilst for turbulent flow in a rough channel, the friction factor formula selected is Colebrook-White equation or Moody diagram which is written as [18]:

$$\frac{1}{\sqrt{f}} = 1.74 - 0.869\ln\left(\frac{2k}{D_{ch}} + \frac{18.7}{\mathrm{Re}\sqrt{f}}\right) \quad (9)$$

where k is the channel absolute roughness and Re is the Reynolds number.

The goodness of data is analyzed using error analysis proposed by Coleman and Steele [19]. In this study the errors consist of bias/systematic and random errors. Systematic errors can be minimized with a calibration whilst random errors cannot. Following Coleman and Steele [19], the random uncertainty of a measured variable, X, is estimated as the standard deviation, S_x, of a sample of N measurements of the variable, X, calculated as follows, taken from [19]:

$$S_X = \sqrt{\frac{1}{N-1}\sum_{i=1}^{N}(X_i - \bar{X})^2} \quad (10)$$

$$\bar{X} = \frac{1}{N}\sum_{i=1}^{N} X_i \quad (11)$$

where $\bar{X}$ is the mean value of the sample population. By contrast, the systematic uncertainty of a measured variable, X, is calculated as the root sum (RSS) given by Eq. (12), taken from [19].

$$B_X = \sqrt{\sum_{k=1}^{M}\left(B_X\right)_k^2} \quad (12)$$

Where $(B_x)_j$ is the j^{th} of the elemental systematic uncertainties $(B_x)_1$, $(B_x)_2$, $(B_x)_3$,…… $(B_x)_M$, estimated from, for example calibration data and instrument specifications given by the manufacturers. The combined error u_c is then given by Eq. (13) and the propagated error can be estimated using Eq. (14), taken from [19].

$$u_C = \sqrt{B_x^2 + S_x^2} \quad (13)$$

$$U_r = \sqrt{\left(\frac{\partial r}{\partial X_1}\right)^2 U_{X_1}^2 + \left(\frac{\partial r}{\partial X_2}\right)^2 U_{X_2}^2 + \ldots\ldots\ldots\ldots + \left(\frac{\partial r}{\partial X_j}\right)^2 U_{X_j}^2} \quad (14)$$

Equation (14) gives the absolute uncertainty, U_r, in the result.

Table 1. k_l, loss coefficient for the sudden contraction, taken from [10]

D_{ch} / D_i	k_l
0	0.50
0.2	0.45
0.4	0.38
0.6	0.28
0.8	0.14
1.0	0

4. Results and Discussions

The inlet and outlet pressures have been measured and the total pressure drop obtained from the three test section is presented in Fig. 4. The test section #1 is the smooth test section with a diameter of 1.14 mm equipped with calming sections installed before and after the test section, therefore, the total pressure drop is the same as the channel pressure drop and the associated flow is fully developed flow. The test section #2 is the smooth test section with a diameter of 17 mm and without a calming section and the test #3 is the rough test section with an average diameter of 15.5 mm (the actual diameter is 17 mm and the effective diameter is 15.5 mm) and without a calming section. From Fig. 4, it is clear that decreasing in diameter increases the pressure drop significantly. For example, the decrease in diameter from 17 mm to 15.5 mm, the pressure drop deviates of about 400% of the pressure drop obtained in the 17 mm tube diameter at the same Reynolds number of 20000, whilst from 17 mm to 1.14 mm the pressure drop deviates of approximately 166567% of the pressure drop gained in the 17 mm tube diameter at the same Reynolds number of 3000. This is becoming a serious problem in the use of small to micro channels.

In Fig. 4, the pressure drops were measured in the test section with diameters of 15.5 mm and 17 mm and without a calming section, unless for the 1.14 mm tube diameter. Therefore, to obtain the channel pressure drop Eq. (3) to (5) should be employed. The cross sectional channel area for the 15.5 mm tube diameter is equal to 0.000154 m², whilst for the 17 mm tube diameter is 0.000227 m², therefore, flow velocities in the channel can be calculated as, taken from [17]:

$$V_{ch} = \frac{\dot{m}}{\rho A_{ch}} \qquad (15)$$

where $\dot{m}$ is the mass flow rate. For the 17 mm tube diameter, Eq. (4) and (5) result in zero pressure drop because the inlet and outlet plenum diameters were the same as the channel diameter, whilst for the 15.5 mm tube diameter were not. The loss coefficients for the 15.5 mm tube diameter are approximately 0.14 for Eq. (4) and 1 for Eq. (5). Pressure drops obtained in the 15.5 mm tube diameter are higher than those in the 17 mm tube diameter. This was obviously due to the surface roughness or rough channel wall and channel diameter.

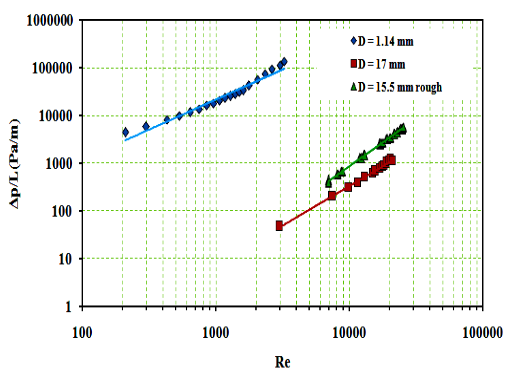

Figure 4. Total pressure drops per meter obtained from flow in the three test sections

As there are many contradictory results in experiments that use small to micro channels, in this occasion, experimental friction factors are presented in Fig. 5 and discussed. The theories used for comparison with the experimental data are Eq. (6) trough Eq. (9). In addition, a correlation proposed by Mirmanto [20] is also used here to see the accuracy of the correlation. The correlation of friction factor in [20] was proposed for flow in rectangular channels with diameters ranging from 0.438 mm to 0.635 mm and without a calming section or it is for developing flow. the correlation is expressed as

$$f = 12.55\,\mathrm{Re}^{-0.92} \qquad (16)$$

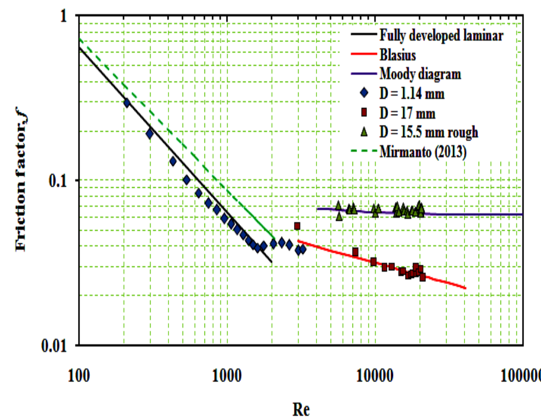

Figure 5. Experimental friction factors

In the Fig. 5, the experimental friction factors are in a good agreement with the theory. The friction factor for the 15.5 mm tube diameter obeys the Moody diagram at *k/D* = 0.032 after all minor losses are subtracted from the total pressure drop. The deviation between total pressure drop and channel pressure drop is approximately 16.4%. For the 17 mm tube diameter, the friction factor agrees well with Blasius [21] correlation for smooth pipe. However, for the 15.5 mm and 17 mm tube diameters, laminar experiments could not be performed due to their lengths. if the lengths were quite long, then in laminar flow, the pressure difference could be read. Additionally, the friction factor obtained in the 15.5 mm tube diameter is higher than that obtained in the 17 mm. This is not due to the tube diameter but the roughness. Meanwhile, the correlation proposed by Mirmanto [20] over predicts the data, because the correlation was created for developing flow condition and rectangular channels. Hence, it evidences that the friction factor is not affected by the diameter but by several factors such as entrance condition, minor losses and developing flow condition.

To ensure that the experimental data agree with the theory, an excel analysis and error analysis have been performed. Examples of excel analysis and error analysis are described here. Consider the experimental data obtained in the 17 mm tube diameter, using excel analysis (ANOVA regression), the significance *F* is small to be compared with *F* values. It means that there is no deviation between the experimental results and the theory, see Table 2. Additionally, an error analysis following Coleman and Steele [19] is presented in Fig. 6 for the data obtained for the 17 mm tube diameter. The first point with big error bars indicating the furthers point, still covers the Blasius [21] correlation, hence, the data are in a good agreement with the theory. The error bars for the experimental friction factor shown in Fig. 6 are given by, using equation proposed by [19]:

$$\frac{U_f}{f}=\left[\left(\frac{U_{\Delta p_{ch}}}{\Delta p_{ch}}\right)^2+\left(\frac{U_{D_{ch}}}{D_{ch}}\right)^2+\left(\frac{U_L}{L}\right)^2+\left(\frac{2U_{V_{ch}}}{V_{ch}}\right)^2\right]^{1/2} \quad (16)$$

The uncertainty of the pressure drop, diameter, tube length and fluid velocity are known from the experiments, hence, the relative error of friction factors can be found and shown in Fig. 6.

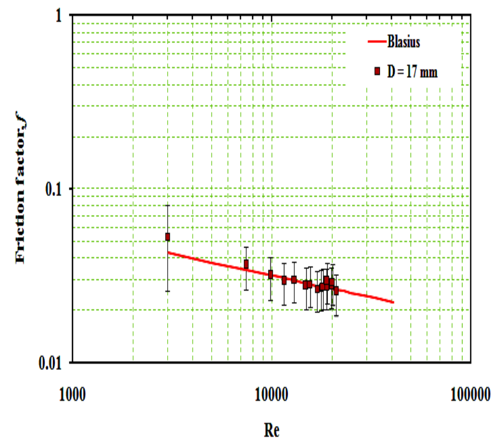

Figure 6. Error analysis for data obtained for the 17 mm tube diameter

Table 2. Excel analysis using ANOVA regression

SUMMARY OUTPUT

Regression Statistics	
Multiple R	0.964487317
R Square	0.930235785
Adjusted R Square	0.924869307
Standard Error	0.000293979
Observations	15

ANOVA

	df	*SS*	*MS*	*F*	*Significance F*
Regression	1	1.49809E-05	1.498E-05	173.34195	6.82013E-09
Residual	13	1.12351E-06	8.642E-08		
Total	14	1.61044E-05			

	Coefficients	*Standard Error*	*t Stat*	*P-value*	*Lower 95%*	*Upper 95%*	*Lower 95.0%*	*Upper 95.0%*
Intercept	0.002699278	0.000361599	7.4648378	4.732E-06	0.001918091	0.003480465	0.001918091	0.003480465
X Variable 1	0.152781938	0.011604333	13.165939	6.82E-09	0.127712302	0.177851575	0.127712302	0.177851575

5. Conclusion

Experiments to see the difference of friction factors obtained for flow in several different diameter tubes have been performed. For the 1.14 tube diameter, the experiments were conducted at the Brunel University, whilst for other test sections were performed at the Mataram University. The results show that, for both experiments conducted in different locations, the experimental friction factors are not influenced by the size of the tubes but by the condition of the entrance and surface roughness, and they obey the theory very well. However, as the size of the tubes reduces, the pressure drop increases drastically.

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Nomenclature

A	Cross sectional area [m^2]	p	Measured pressure [Pa]
B_x	Systematic uncertainty	R	Regression
D	Diameter	Re	Reynolds number
df	Degree of freedom	S_x	Standard deviation
f	Friction factor	u_c	Combined error
j	Number of element systematic uncertainty	U_r	Propagated error of r function
k	Absolute roughness [m]	V	Channel average velocity [m/s]
k_l	Losses coefficient	X	Measured variable
L	Tube length [m]	$\overline{X}$	Average of measured variable
M	Maximum number of element systematic uncertainty		
$\dot{m}$	Mass flow rate [kg/s]		
N	Number of measurements		

Greek symbol

Δp	Pressure drop [Pa]
ρ	Fluid density [kg/m^3]

Subscript

ch	Channel
i	Inlet
o	Outlet
t	Total

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