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FIBERGLASS CIRCULAR TURBULATOR IN COUNTER FLOW DOUBLE PIPE HEAT EXCHANGER: A STUDY OF HEAT TRANSFER RATE AND PRESSURE DROP

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Abstract
This preliminary investigation studied the effect of circular turbulator vortex generator on heat transfer rate and pressure drop in a circular channel countercurrent double pipe heat exchanger with water working fluid. Increasing the number of circular turbulator yielded increasing heat transfer rate and pressure drop. The problem generated when increased pressure drop occurred in relation to more energy consumption of the water pumping system. Therefore, optimization in circular turbulator number is necessary to minimize the pressure drop about distance length between circular turbulator, tube diameter and thickness, type of material and crystal lattice, as well as the geometrical shape of fluid passage (circular or square). This study applied PVC outer tube and copper alloy inner tube, as well as fiberglass circular turbulator. The optimum results showed that seven parts of circular turbulator increasing heat transfer rate by 30% and pressure drop by 80% compared to that passage in the absence of circular turbulator at cool water debit of 7 L/min.

Keywords:
Circular Turbulator;
Heat Exchanger;
Heat Transfer Rate;
Pressure Drop;

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INTRODUCTION

The heat exchanger is a device commonly used in many industries, particularly for the production process. Heat energy is transferred from one fluid to other fluid by both convection and conduction through channel wall of a heat exchanger. Several efforts are generally known to enhance heat transfer rate, i.e. by elevating surface area, increasing temperature difference and escalating heat transfer coefficient. The heat transfer coefficient can be escalated applying high thermal conductivity fluid such as nanofluid or increasing Reynold Number either using higher velocity or fluid debit or vortex generator.

Several investigations on the important role of the heat exchanger are previously reported. Nanang [1][2] applied circular fins in varied tube cross sections to accelerate heat removal of refrigerant liquid in a cooling system. Singh et al. [3][4] used numerical Rung-Kutta method in alumina-water nanofluid in a heat transfer under magnetic field. They found that velocity distribution, temperature gradient and shear stress significantly affected the heat transfer rate. In the same year, Li et al. [5] applied wavy ribs in the fin-and-tube heat exchanger in order to get a balance between enhanced heat transfer rate and increased pressure loss. A year after Tang et al. [6][7] investigated smooth and micro-fine copper tubes heat exchanger applied in certain type refrigerants and found that all micro-fin tubes showed significant enhancement of heat transfer convection. However, the type of refrigerants showed little effect on the mechanism of heat transfer enhancement.

Furthermore, Fiebig [8] thoroughly studied compact heat exchangers particularly emphasized on wing-type vortex generators (WVG) and found many possibilities for heat transfer enhancement related to possible many geometrical parameters could incorporate and integrate with WVG type. Interestingly, Okten and Biyikoglu [9] investigated the effect of air bubble injection in a system integrated into a storage tank and found overall heat transfer by approximately 11%–14%. On other occasions, Kim [10] investigated heat transfer enhancement in wavy titanium tubes with three different diameter and thickness sizes and obtained a faster heat transfer rate with corrugated tubes rather than a smooth tube. Diaz and Guo [11] applied varied crystal planes of copper, nickel, platinum and silicon for heat transfer tubes coated by single layer graphene. They obtained much lower temperature gradient than that one of graphene free. Moreover, the different crystal planes gave effect on heat removal. In the latest report, Nalawade et al. [12] studied the characteristics of heat transfer applying a square channel heat exchanger inserted with wings vortex generator on two opposite walls to yield turbulent flow. Moreover, this comprehensive study investigated the effect of geometrical parameters, a number of wing vortex and the ratio of Nusselt number to friction factor compared to the smooth surface channel wall.

Since many types of heat exchangers have many practical uses in industries, this study attempted to investigate the role of circular turbulator on heat transfer rate and pressure loss in a counter current double pipe heat exchanger. This study applied copper alloy heat transfer tube coated by PVC synthetic polymer. The copper alloy is a good thermal conductor to increase heat transfer, and the PVC organic material is to minimize heat removal to the environment to get more efficient heat transfer. The fiberglass circular turbulator inserted in the counter flow heat exchanger tube as a vortex generator. The fiberglass was selected rather than a metallic element on account of economic reason and lighter material. The effect of the circular turbulator number on heat transfer rate and pressure drop has been investigated. In addition, the consideration of material applications is referred to the study of the double pipe heat exchanger as previously reported by [13][14], that both of them applied PVC outer pipe and copper inner pipe. The idea of fiberglass turbulator used in this study came from the study of heat transfer, applying fiberglass plate turbulator in fan flows [15]. Up to date, the application of fiberglass turbulator in double pipe heat exchanger has not ever reported.

METHOD

The equipment and its specification are presented in Table 1. The method of heat exchange system was adapted from previous work with some modifications [1,6,16].

Table 1 illustrated the equipment used in this study with its material specification. The cross section of fiberglass circular turbulator is shown in Figure 1 with size specification listed in Table 1.

The role of circular turbulator is to generate vortex in the fluid passage to enhance the heat transfer rate. This study selected the water debit in the range of 3–9 L/min on account of that range represented for all fluid patterns (laminar, transitional, and turbulent).

Table 1. Equipment with Its Specifications

Equipment	Specification
Thermocouple	K - type
Cooling water pump	Capacity100 L/min.; Head 23 m
Hot water pump	Capacity 900 L/h.; Head 0.9 m
Heater	Max temperature 50°C
Circular turbulator	Outer diameter 27 mm; Inner diameter 22 mm; Fiberglass
Inner tube	Outer diameter 9 mm; Inner diameter 7 mm; Copper alloy; Length 960 mm
Outer tube	Outer diameter 32 mm; Inner diameter 27 mm; PVC polymer; Length 960 mm
Manometer	U-type

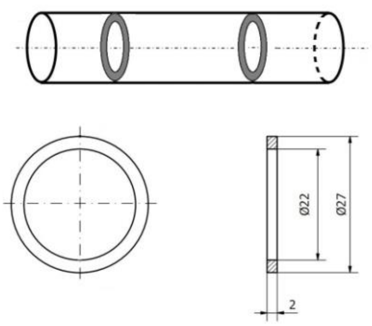

Figure 1. Design of Circular Turbulator Inserted in Heat Exchange Tube

According to Chengel et al. [17] there is a correlation between Reynold number and water debit as described in the following equation, i.e.

$$Re=(Wd/A).(D/\eta) \qquad (1)$$

Where Re = Reynold number; Wd = water debit; D and A are related to the diameter and cross section of outer and inner tubes; η = water kinematic viscosity at 23°C. By substitution the values of water debit (3 – 9 L/min) in this study to (1), we obtain data as shown in Table 2.

Table 2. Correlation between water debit in this study and flow pattern

Water debit (L/min)	Reynold number	Flow pattern
3	1888.2	laminar
5	3147.0	transitional
7	4405.9	turbulent
9	5664.7	turbulent

Figure 2 shows a virtual illustration of flow orientation classified by Reynold Number as laminar and turbulent flow.

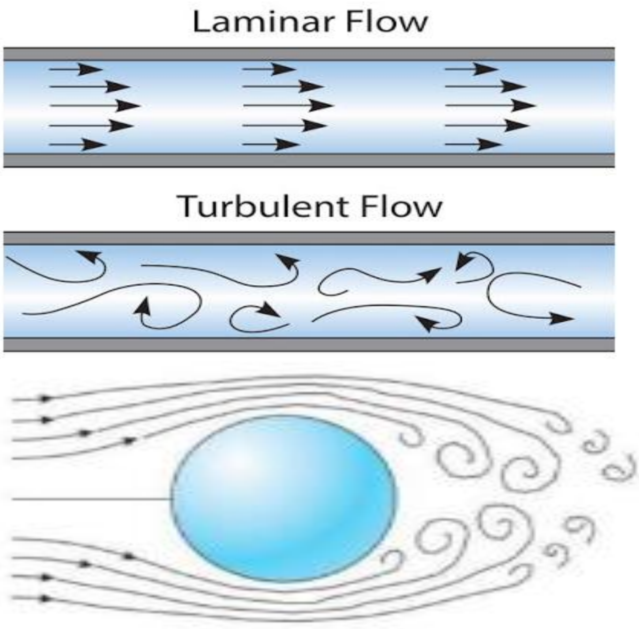

Figure 2. Virtual Illustration of Fluid Flow Orientation

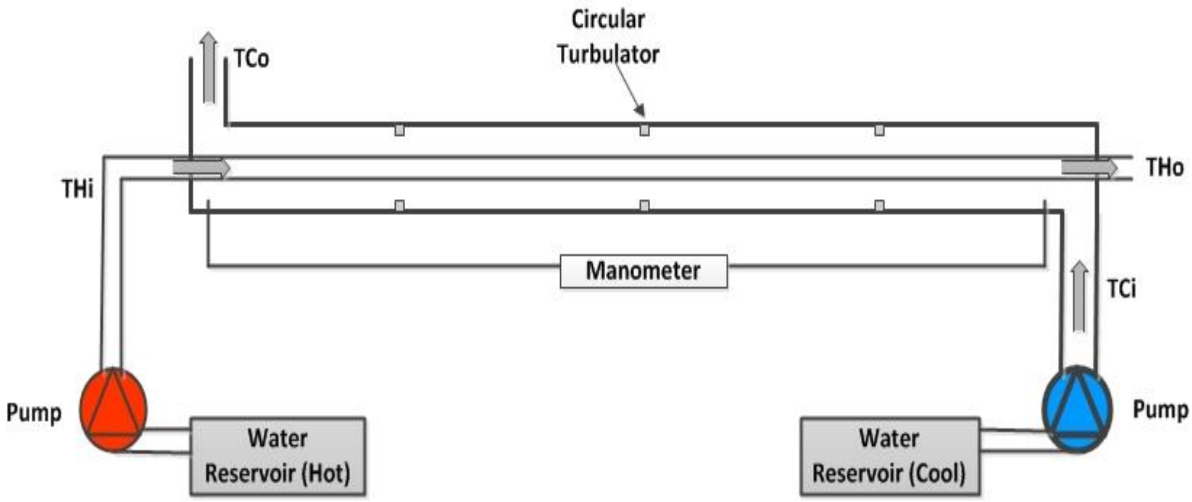

Figure 3. The mechanism of counter current double pipe heat exchanger inserted with fiberglass circular turbulator. PVC outer channel filled with cool water (inlet 23°C). Copper alloy inner channel filled with hot water (inlet 50°C)

The basic mathematical concept of heat exchange follows the classical rule concerning ratio of convection (*h*) and conduction (*k*) of heat transfer rate due to temperature difference (*ΔT*) for a given tube diameter (δ) expressed as Nusselt Number (*Nu*) (2).

$$\frac{\dot{q}_{conv}}{\dot{q}_{conv}} = \frac{h\nabla T}{k\nabla T/\delta} = \frac{h\delta}{k} = Nu \qquad (2)$$

The Nusselt Number is a multiplication of Reynold Number (*Re*) and Prandtl (*Pr*) Number with a constant (*C*) defined as Eq.3.

$$Nu = C.\ Re^{m}.\ Pr^{n} \qquad (3)$$

Rearrangement of (2) and (3) yields a new correlation described as (4).

$$\frac{h\delta}{k} = C.Re^{m}.Pr^{n} \qquad (4)$$

The mechanism of heat exchange in counter current heat exchanger with water working fluid used in this study is illustrated in Figure 3. This study applied an inner tube made of copper alloy filled with hot water and the outer tube made of PVC filled with cool water, as shown in Figure 3. The energy of heat transfer described in (5).

$$q = m \,.\, Cp.\, \Delta T \qquad (5)$$

Where q is the heat transfer rate (J/s), m the water mass rate (kg/s), Cp the water specific caloric (J/kg.K) and ΔT as temperature difference in Kelvin (K).

The friction factor between fluid and channel wall yielded pressure drop. Increased pressure drops needed higher pumping power of the compressor. The pressure drops (Pa) was measured by two pieces of manometer (U-type), which connected to cool water inlet and outlet.

RESULTS AND DISCUSSION

Effect of Circular Turbulator on Heat Transfer Rate

The measurements based on cool water pumping (L/min). The heat transfer rate (J/s) monitored by a thermocouple and the calculation used (5). The results show graphically in Figure 4.

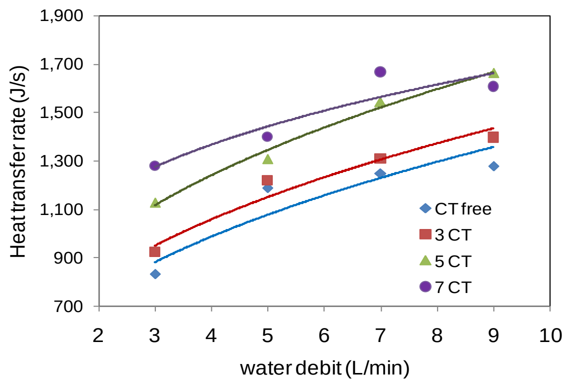

Figure 4. Effect of Circular turbulator on Heat Transfer Rate. Counter Flow Heat Exchanger. Cool Water Debit. CT = circular turbulator

Figure 4 shows an increasing tendency of heat transfer rate with an increased number of circular turbulator (CT) compared to that without CT. It is reasonable since CT yielded a vortex in water fluid, producing more turbulent stream. It looks there is a level of saturation after 7 L/min water debit. The heat rate transfer increased by about 30 % by applying the 5-CT or 7-CT compared to that without CT. As seen in Figure 4, with regard to the number of CT, either the 5-CT or 7-CT yielded no significant change in the value of heat transfer rate. Therefore, the saturation level achieved either using the 5-CT or 7-CT and after cool water debit 7 L/min. Further study on Figure 4 about 5-CT and 7-CT can be explained based on distance values between CTs for 3-CT, 5-CT and 7-CT, as shown in Table 3.

Table 3. Distance between CTs in this study

Number of CT	Distance between CTs (m)
3	0.24
5	0.16
7	0.12

Then the distance value of 5-CT and 7-CT are being related to the concept of dynamic fluid defined as "fully developed" stated in hydrodynamic entry length (Lh) shown in Figure 5.

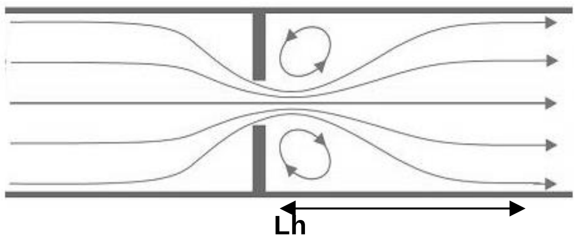

Figure 5. The Hydrodynamic Entry Length (Lh) in the Concept of Dynamic Fluid

According to Chengel et al. [17], for turbulent flow, Lh ≈ 10 D. From Table 3, the distance values of 5-CT (0.16 m) and 7-CT (0.12 m) are not significantly different. The hydrodynamic entry length (Lh) requires debit rate data (7 and 9 L/min) referred to turbulent flow, and the equation Lh ≈ 10 D, the result is shown in Table 4.

Table 4. Data of hydrodynamic entry length (Lh)

Debit rate (L/min)	Reynold	D (m)	Lh (m)
7	4405.90	0.018	0.18
9	5664.73	0.018	0.18

Note: D = difference between inner diameter of the outer tube and outer diameter of the inner tube is 0.018 m.

Since the Lh data (0.18 m) in Table 4 is larger than the distance values of 5-CT (0.16 m) and 7-CT (0.12 m), at this condition the concept of "fully developed" has never been achieved, or in other words, the heat transfer rates for 5-CT and 7-CT are not significantly different. Concerning this concept, this study describes the 7-CT as the maximum heat transfer rate ever achieved.

Figure 6 shows Bunker and Osgood [16] applied varied angles of lean turbulator about positions of (-45° – +45°) and found that in the position of normal to surface yielded the best overall performance. The study applied turbulator at varied leaning position and rectangle cross section channel instead of the number of CTs and circular channel used in this study [16].

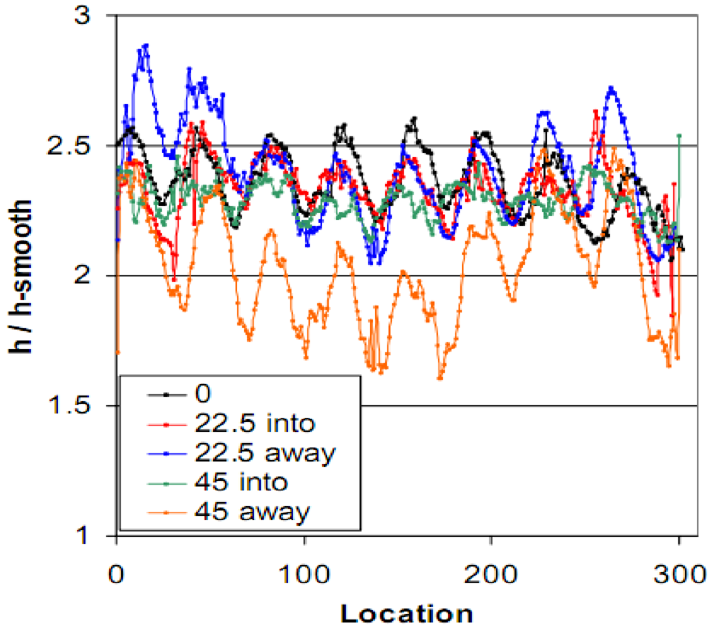

Figure 6. Leaning turbulator at varied angles (- 45º-+45º) in a square channel heat exchanger. h = lean turbulator. h-smooth = turbulator free. *Re* 50 000 [16]

In addition, Bunker and Osgood [16] studied the effect of turbular leaning angle on heat transfer coefficient while this study examined the effect CT number on heat transfer rate. The heat transfer coefficient is linearly proportional to the heat transfer rate. On other occasions, Okten and Biyikoglu [9] applied air bubble system and found an enhancement by 11 – 14%, while Li et al. [5] used wavy fins tube achieving an increase overall performance by 40%. Diaz and Guo [11] applied to copper, and platinum tubes yielded increased performance of 14% and 9%, respectively.

Therefore, many modifications with heat exchanger have been made to enhance performance such as tube material, type of crystals, wavy surfaces, varied diameters and thickness, geometrical of tubes (circular or square) and many other combinations.

Effect of Circular Turbulator on Pressure Drop

On one side, many efforts have done to enhance heat transfer performance, however, pressure drop due to friction factor and increased turbulent character cannot avoid. Figure 7 shows a different profile compared to that of Figure 4. Figure 4 shows a saturation degree with regard to heat transfer rate, while Figure 7 shows an increasing trend for pressure drop at a given range of water debit. The condition is reasonable since higher water debit yielded increased turbulent wave at a high number of CT. This phenomenon can be explained from (6) described the correlation between increased pressure drop and average fluid rate. The water debit rate is linearly proportional, with an average fluid rate [14].

$$\Delta P_L = f.(L/D).(\rho V^2/2) \quad (6)$$

From (6), the water debit rate (3 L/min) yielded low fluid rate related to low increased in pressure drop (ΔP_L). As the water debit getting higher (9 L/min), the higher fluid rate (V) in quadratic form yielded significantly increased of pressure drop.

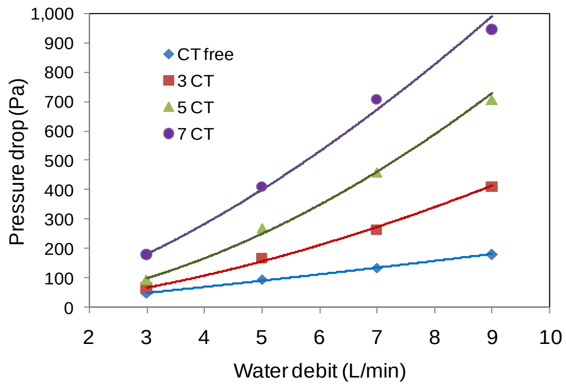

Figure 7. Effect of circular turbulator on pressure drop. Counter flow heat exchanger. Cool water debit. CT = circular turbulator

This reason can explain that at water debit of 3 L/min, it yielded insignificant effect on pressure drop for all CTs and CT free, while at high water debit of 9 L/min, it gave significant effect on pressure drop.

From experimental data, the number of CT on heat transfer rate caused an increase of 30%, on the other side, the same number of CT on pressure drop caused an increase of 80% compared to that without CT at a given range of condition.

The significant imbalance between heat transfer rate and pressure drop requires a more comprehensive examination for tube material, tube diameters, length distance between CTs, wavy ribs/fins, bubbling effect, and another parameter. This investigation applied distance between CTs is 238 mm (3CTs), 158 mm (5CTs) and 118 mm (7CTs), respectively.

The report of Bunker and Osgood [16] shows the imbalance condition as illustrated in Figure 8 that the friction factor increased for all positions of lean turbulators (varied lean angels) with increasing turbulent flow. As already explained as depicted in Figure 6, the imbalance condition in this study is due to the application of a number of CTs while Bunker and Osgood [16] reported turbulator at varied leaning position (varied angles). In addition, Bunker and Osgood [16] studied the effect of turbulator leaning angle on heat transfer coefficient while this study examined the effect of CT numbers on heat transfer rate. The heat transfer coefficient is linearly proportional to the heat transfer rate.

The Ratio of Heat Transfer Rate to Pressure Drop

The effect of CT number compared to that without CT on heat transfer is viewed from the standpoint based on ratio of heat transfer rate to pressure drop, as shown in Figure 9.

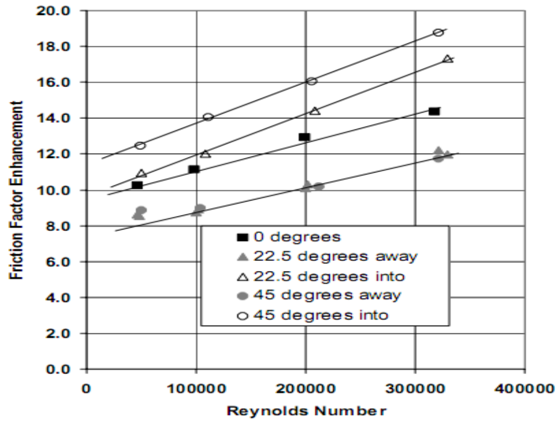

Figure 8. The friction coefficient of high turbulent flow. Leaning turbulator at varied angles (- 45° - +45°) in square channel heat exchanger [16]

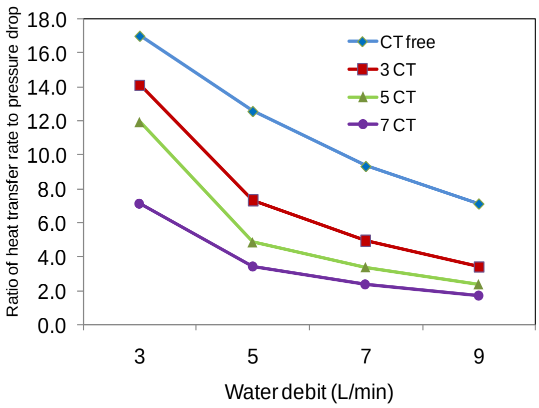

Figure 9. Effect of CT number on heat transfer based on ratio of heat transfer and pressure drop. Cool water debit. CT = circular turbulator

The general trend, as seen in Figure 9 tends to illustrate exponential decreasing with increasing water consumption from cooling pumping unit at given CT numbers and without CT. Furthermore, the curve slope gets steeper with an increasing number of CT compared to that without CT. From this experimental illustration (Figure 9), the 7-CT tends to show a more stable response with increasing cool water consumption.

A better explanation for Figure 9 will be given based on pressure drop values as listed in Table 5.

Table 5. The ratio of heat transfer to pressure drop (Circular Turbulator free)

Water debit (L/min)	Heat transfer (W)	Pressure drop (Pa)	Ratio (W/Pa)
3	831.738	48.917	17.0
5	1,188.622	94.573	12.6
7	1,247.482	133.708	9.3
9	1,277.184	179.364	7.1

Table 5 shows that increasing heat transfer is likely proportional with increasing pressure drop, so the ratio values in descending mode have a tendency of linearity. At CT free, the increasing pressure drop is only caused by a major loss of fluid flow related to friction between fluid flow and pipe wall. At the existence of CT(s) in the pipe, besides the mayor loss due to friction, there is another minor loss of fluid flow due to the presence of obstacle material such as turbulator, valve, turning point, etc.

Table 6. Increasing fluid velocity in the presence of 3 – 7 CTs

Water debit (L/min)	Fluid velocity (V) m/s	V^2	Increasing V^2
3	0.0983	0.0097	
5	0.1638	0.0268	178 %
7	0.2294	0.0526	96 %
9	0.2949	0.0870	65 %

Concerning Figure 9, in the existence of 3 – 7 CTs the fluid velocity in quadratic form is remarkable increased, i.e. 178% (3-5 L/min.) as shown in Table 6, and followed by lower increasing quadratic fluid velocity only by 96% and 65% (5-9 L/min). From (6), the quadratic velocity is linearly proportional with increasing pressure drop. Since the pressure drop is the denominator in the Ratio of heat transfer to pressure drop, therefore, the results show a sharp negative slope for 3-5 L/min followed by a slow and little negative slope for 5-9 L/min.

CONCLUSION

Although this preliminary study has not yet shown optimum results, the investigation shows a brief outlook that high heat exchange coefficient requires a more elaborate design engineering system considering factors maximizing heat transfer and minimizing pressure loss. Moreover, this study is very valuable for several industrial and engineering applications, especially for specific application addressing to miniaturization on consideration of savings in energy and cost yielding augmentation of heat transfer.

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