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Analysis of Longitudinal Finned Pipes in Cross-Flow Heat Exchanger

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ABSTRACT

In this case, the recovery of excess heat in the steel-making process becomes challenging to save energy effectively and efficiently. One way to help the process of heat recovery in the steel-making process is with a heat exchanger. This heat exchanger maximizes the production process work. However, in the application, the heat exchanger developed to date can still work optimally. Based on the literature study that has been done previously for the design of the heat exchanger, in this study, the addition of four longitudinal fins variations with the cross-flow flow was carried out. This research compares the heat exchanger performance finned pipes with pipes without fins design, where liquid and airflow rates are varied. This study evaluated whether the longitudinal fins configuration and the fluid flow rate affect the cross-flow heat exchanger performance. The results show that fins addition on the copper pipes increases the value of Nusselt number 2.56% in liquid fluid and 16.57% in air fluid and increases the NTU value of its effectiveness by 24.77%.

1. INTRODUCTION

In recent decades, several studies have focused on energy saving in industry and energy efficiency in various areas of the manufacturing sector. Different appropriate methods to optimize energy savings comprehensively and practically have also been used in today's industry [1]. Due to the current high energy price, the industry is constantly striving to develop more effective and efficient energy saving methods than ever before. The inefficiency of energy use has an impact on an industrial process. For example, the steel industry is one of the industries with the largest energy consumption, about 5% of the world's total energy consumption, with energy costs around 30% of the total production cost [2]. In this case, the recovery of excess heat in the steel-making process becomes challenging to save energy effectively and efficiently. One way to help the heat recovery process in the steel-making process is by using a heat exchanger.

In energy production and management, 90% of the heat energy used is distributed through various types of heat exchangers [3]. The heat exchanger is a device that utilizes the flow of thermal energy in two or more fluids with different temperatures, resulting in heat transfer [4, 5]. The heat exchanger is a device used to transfer heat efficiently from one fluid to another fluid separated by walls so that there is no direct mixing of fluids [6]. The importance of this heat exchanger is seen from the point of view of energy conservation, conversion, recovery, and successful implementation of new energy sources. Its essence is also increasing from environmental issues such as thermal, air, water, and waste heat disposal [7].

The primary key to heat transfer in the heat exchanger is heat conduction through the boundary layer on the tube [8]. The use of this heat exchanger is expected to help maximize the work of the production process in an industry. However, in the application, the heat exchanger has not been able to work optimally, especially in the heat transfer process that occurs in a production process, so it becomes less efficient. The use of this heat exchanger aims to maintain the temperature in production machines, coolers, air conditioning, space heating, electric generators, boilers, radiators, and chemical plants that are used continuously for a long time to remain stable and durable. Therefore, maximizing the work of a heat exchanger needs to be studied further to produce innovations in correcting the shortcomings of a heat exchanger.

One study compared the addition of star-shaped with annular variations in cross-flow heat exchangers. As a result, the addition of star-shaped fins on the heat exchanger can increase fluid flow turbulence. In addition, the addition of this star-shaped fins can increase the heat flow by 39.3% more effectively than the annular shape [9]. It can be influenced by the addition of fins on the copper pipes, which affects the heat transfer surface area of the heat exchanger. On the other hand, the heat transfer is also influenced by three things, namely the overall heat transfer coefficient (U), the heat transfer surface area (A), and the exact average temperature difference in the heat exchanger. Another study also applied an additional variation in the form of a perforated fin that maximizes heat transfer from the subsurface air side of a large pitch fin tube heat exchanger. By analyzing the ice mass, ice thickness, heat transfer rate, and heat transfer coefficient of the perforated fin heat exchanger tube, the results obtained that the heat transfer rate and heat transfer coefficient of the perforated fin increased by 38.9% and 31.8% [10].

The development of the current heat exchanger has penetrated the design of the heat exchanger. Various studies have used two different heuristics in its development, namely Genetic Algorithm (GA) and Particle Swarm Optimization

(PSO), to find the optimal heat exchanger design. However, the results obtained from both methods are almost the same. However, the PSO method has the advantage of a faster processing time [11]. The theory of optimization and performance improvement of this heat exchanger is based on the conventional design of the previous heat exchanger, in which fluid properties are often considered constant. Therefore, heat exchangers with cross-flow designs are more often used to analyze and discuss mechanisms for increasing and improving the effectiveness of heat exchangers for fluids with various variations. This aims to increase the heat load of the heat exchanger of new approach under conditions of a fixed heat transfer area to increase heat exchange rate with a drastic variation in properties [12].

Based on previous research discussing heat exchangers with variations of star-shaped fins for cross-flow designs, four longitudinal fins variations with the cross-flow heat exchanger were added in this study. This is compared to its performance with copper pipes without fins, where the flow rate of liquid and air is one of the variations in this study. This study was conducted to evaluate whether fins position, different fins shapes, pipes number, and fluid flow rate affect the cross-flow heat exchanger performance. This study also focuses on varying fluid flow of the heat exchanger, which was developed to consider significant variables that are experiencing constraints.

2. METHOD

2.1 Determination of research variables and data analysis

This study is an experimental study that uses three variables, namely variations in liquid fluid temperatures of 50°C, 60°C, and 70°C, variations in water flow velocity from 0.06 l/s, 0.08 l/s, and 0.1 l/s, and variations of airflow from the speed of 1 m/s, 1.5 m/s, and 2 m/s. In comparison, the dependent variable is the heat transfer coefficient, LMTD, effectiveness-NTU, Reynolds number, Nusselt number, and Prandtl number. This study of the heat exchanger without fins and upright fins with this cross-fluid flow.

The heat transfer analysis carried out in this study is the LMTD method. This method is used with predetermined inlet and outlet fluid temperatures determined from the energy balance equation. Then the method is combined using the effectiveness-NTU (ε-NTU) method. In this case, the use of the LMTD method is suitable for designing heat exchangers, while the ε-NTU method is suitable for analyzing the performance of an existing one [5].

2.2 Cross-flow heat exchanger analysis

In analyzing the data obtained, it is necessary to know the various parameter values to get the experiments carried out. Where the mass flow rate of the water can be obtained from,

$$\dot{m} = \rho \, . \, v \tag{1}$$

Meanwhile, the rate of heat transfer of the liquid and gaseous fluids in the system can be determined by,

$$Q = Q_h = Q_c \tag{2}$$

$$Q = \dot{m}_h (h_1 - h_2) \tag{3}$$

It is assumed that the heat released by the hot water fluid is entirely absorbed by the air so that it can determine the magnitude of the mass flow rate of air which can be known as follows:

$$Q_c = \dot{m}_c \, . \, c_p \, . \, \Delta T \tag{4}$$

$$\dot{m}_c = \frac{Q_c}{c_{pc} \Delta T} \tag{5}$$

Then by looking for the values of P and R, as written in the equation below, can be determined the value of the correction factor,

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{h2}} \tag{6}$$

$$R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} \tag{7}$$

Based on the previously searched P and R values, the value of the correction factor is F = 0.995. Furthermore, the value of LMTD can be known from,

$$LMTD = \frac{(T_{ho} - T_{co}) - (T_{hi} - T_{ci})}{In \frac{\Delta T_1}{\Delta T_2}} \tag{8}$$

Then, in this case, the magnitude of the cross-sectional area can be divided into three parts. Where the overall surface area (A) is obtained by,

$$A = (Inner\ Pipe\ Area + Square\ Area) - Area\ of\ Outer\ Circle \tag{9}$$

In determining the value of the overall design heat transfer coefficient (U_d) it can be seen from,

$$U_d = \frac{q}{A \ x \ LMTD \ x \ F} \tag{10}$$

In this case, the calculation of the liquid fluid can include from the heat transfer surface area obtained from,

$$A = 2\pi rt \ x \ 25 \tag{11}$$

Then the mass flow velocity of the water can be determined by,

$$G = \frac{\dot{m}_h}{A_h} \tag{12}$$

where the Reynolds number value is determined based on,

$$Re_h = \frac{\rho . \upsilon . d}{\mu} \tag{13}$$

and the value of the convection heat transfer coefficient (h_i) can be obtained from the equation,

$$Nu = 0{,}023 \, . Re^{0{,}8} . Pr^{0{,}4} \tag{14}$$

So that,

$$h_i = \frac{Nu \, . K}{di} \tag{15}$$

In addition, the calculation of the air fluid includes the surface area of the air heat transfer, which can be known from,

$$A = [2(\, sxs\,) + 4\,(\,pxl\,)] - 2\pi rt\ x\ 25 \tag{16}$$

Then the mass flow velocity of air can be obtained by the equation,

$$Gc = \frac{\dot{\mathrm{m}}_c}{A} \tag{17}$$

where, the Reynolds number value can be determined by the hydraulic diameter *(Dh)* with,

$$Dh = \frac{4ab}{2(a+b)} \tag{18}$$

So,

$$Re = \frac{D_h \, . \, G_c}{\mu} \tag{19}$$

and the value of the convection heat transfer coefficient (h_o) can be obtained from the equation,

$$Nu = 0{,}023 \, . Re^{0{,}8} . Pr^{0{,}4} \tag{20}$$

So that,

$$h_o = \frac{Nu \, . K}{di} \tag{21}$$

Then for the overall heat transfer coefficient value is determined based on the equation,

$$U_c = \frac{1}{\frac{1}{h_i} + \frac{In(\frac{r_o}{r_i})}{2\pi lk} + \frac{1}{h_o}} \tag{22}$$

where, the magnitude of the heat capacity rate (*C*) for both air and water are as follows.

Air heat capacity rate:

$$C_c = m_c \, . Cp_c \tag{23}$$

Water heat capacity rate:

$$C_h = m_h \, . Cp_h \tag{24}$$

Based on the rate of heat capacity obtained, $C_c > C_h$, then $C_{max} = C_h$ and $C_{min} = C_c$, so that the maximum heat transfer rate can be determined by,

$$Q_{maks} = C_{min}(\, T_{hi} - T_{ci}) \tag{25}$$

Then for the effectiveness value can be determined from,

$$\varepsilon = \frac{Q_{aktual}}{Q_{maks}} \times \, 100\% \tag{26}$$

and its NTU value can be obtained based on,

$$NTU = \frac{UA}{C_{min}} \tag{27}$$

2.3 Experimental setup

This experimental study using copper pipes without fins and with variations of four upright fins on each pipe. The copper pipes used is 400 mm long and 12.7 mm in diameter. This heat exchanger uses liquid fluid and air as the test material. Air fluid that acts as cooling flows from the blower to the heat exchanger. While the liquid fluid as a cooled fluid is pumped with various variations of the flow rate and the initial temperature determined. Tables 1 and 2 describe the various measuring instruments and equipment used in this study, along with their detailed specifications. It aims to provide detailed information related to the experimental setup carried out.

Table 1. Experimental measurement tools

Device	Measurement Range	Error
SEA Water Flowmeter Sensor YF-S402	0.3 to 6 liters	2%
Krisbow Digital Thermometer KW0600283	-20 to 1000°C	3%
Digital Temperature Controler XH- W3001	-50 to 110°C	0.1%
Thermocouple type-K	-50 to 350°C	0.75%
Krisbow Digital Anemometer SKU10176567	0.4 to 20 m/s	3.5%

Table 2. Experimental set-up specifications

Component	Type	Specifications
Heater	Electric	220V/50Hz, 500 watt, 16cm x 3.5cm x 1.5cm
Bypass faucet	AW	¾ inch
Water pump	AA-1800	50/60 Hz, 30 watt, max flow 1,800 liter/h, max head 1.8m
Blower	Conch Blower	220-230 V, 370 watt, max rpm 3,000-3,600 rpm, 3 inch, 27 cm x 27 cm x 29.5 cm
Arduino Microcontroller	ATmega328P	5V, Digital I/O pins: 14 (of which 6 provide PWM output)
Heat Exchanger Pipe	Copper Pipe	Pipe d = 12.7 mm, l = 400 mm, and fins l = 360 mm

Figure 1 is a cross-flow heat exchanger design along with a description of the device parts. Furthermore, the pipes variation used is shown in Figure 2, where the pipes used are made of copper without variations in fins and with variations of four longitudinal fins on each pipe.

This research procedure was carried out by heating the water in the reservoir using an electric hanging water heater until it reached the specified temperature, namely 50°C, 60°C, and 70°C. Then turn on water pump and adjust it according to the predetermined fluid flow rate, namely 0.06 l/s, 0.08 l/s, and 0.1 l/s. This fluid flows into the heat exchanger through copper pipes. Next is to turn on the blower with airspeeds of 1 m/s, 1.5 m/s, and 2 m/s. It is related to research flow, further shown in Figure 3.

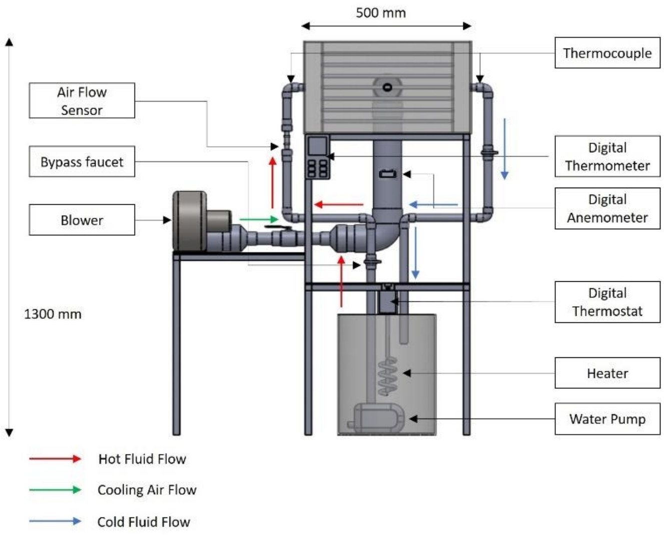

Figure 1. Cross-flow heat exchanger experimental setup

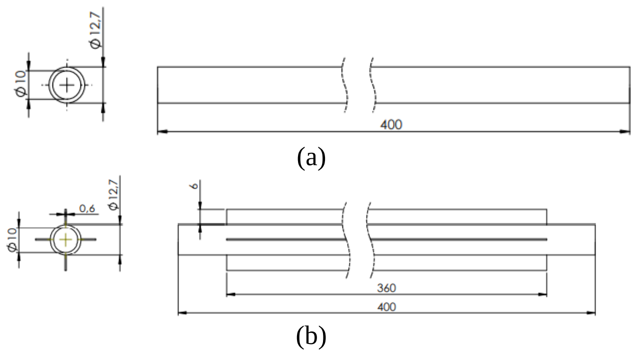

Figure 2. (a) Copper pipe without fins and (b) Copper pipe with fins

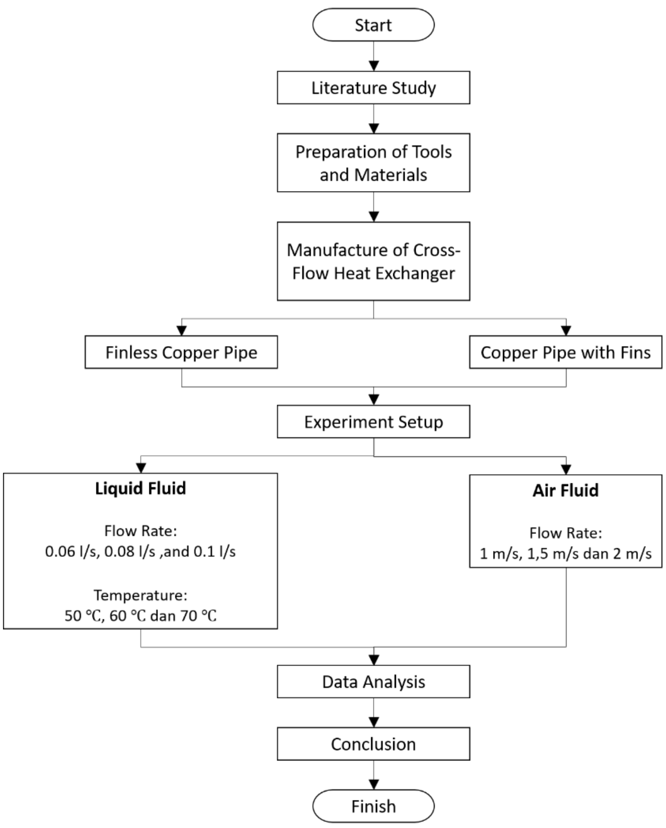

Figure 3. Research flow

3. RESULT AND DISCUSSION

3.1 Overall heat transfer coefficient

The heat transfer coefficient is influenced by the movement of fluids, both water, and air, resulting in forced convection in the heat exchange [13]. From Tables 3 and 4, the heat transfer coefficient increases and decreases due to differences in heat, fluid velocity, and gas fluid velocity. Where the smallest pipes without fins heat transfer coefficient is 0.48784 W/m^2.°C and the largest is 0.92129 W/m^2.°C. The coefficient of heat transfer with the addition of fins is the smallest 0.56959 W/ m^2.°C, and the largest is 1.08141 W/m^2.°C.

On the other hand, liquid and air have different thermal conductivity. Liquid fluids have a greater value than gas fluids. In addition, the heat transfer coefficient is also influenced by the velocity of the liquid fluid and air, where the slower the fluid flow, the smaller the heat transfer coefficient. Meanwhile, the greater the airflow velocity, the greater the heat transfer coefficient. It is also supported by previous research, which explains that the mass flow rate can increase from 0.3 kg/s to 0.6 kg/s, and the overall heat transfer coefficient increases from 36.88 W/m^2.K to 67.12 W/m^2.K [14].

The condensation heat transfer coefficient increases with increasing heat flow and the average vapor quality [15]. In this case, the addition of fins on the copper pipe increases the heat transfer cross-sectional area and increases the turbulence of the gas fluid flow. A heat exchanger with upright fins increases the value of the heat transfer coefficient by 16.57%. As explained in the previous study, the pin fin heat exchanger showed a heat transfer coefficient that was consistently higher than the plate-fin heat transfer coefficient. The higher heat transfer coefficient value for the pin fins is due to the increased turbulence and the separation of the boundary layer between each pin fin [16].

3.2 Reynolds number and Nusselt number for fluid flow

The Reynolds number for water or nanofluids is calculated based on the channel's density, viscosity, average velocity, and diameter [17]. The calculation results of these four things significantly affect the value of the Reynolds number or the heat flow in the heat exchanger. The higher the temperature of the fluid, the average velocity of the fluid will increase the Reynolds number and Nusselt number. In addition, the flow rate of liquid and gas fluids also affects the Reynolds number and Nusselt number. The faster the fluid flow causes the Reynolds number and Nusselt number to increase [18].

In Figure 4, the Reynolds number represents the heat flow, and the Nusselt number represents the heat transfer. The heat flow in the pipe without fins is slightly greater than the pipe with longitudinal fins. The addition of fins affects the pipe's cross-sectional area so that the cross-section is wider and results in a faster heat flow that is equal to 30.783 or 2.5%. Heat transfer in the heat exchanger with the addition of longitudinal fins is better than without fins because the pipe's cross-sectional area is wider. The addition of longitudinal fins increases the heat transfer by 0.2528 or 2.5%. The results of this study were confirmed by previous studies, which also explained that a higher Reynolds number causes a higher heat transfer rate [19]. In this cross-flow heat exchanger research, the result is that the faster the rate of heat transfer for each variation, the greater the Reynolds number.

Table 3. Overall heat transfer coefficient of pipes without fins heat exchanger

Liquid Fluid Temperature (°C)	Overall Heat Transfer Coefficient (W/m² °C) Water Flow Rate (L /s) 0.06 Airflow Rate (m/s)			0.08 Airflow Rate (m/s)			0.1 Airflow Rate (m/s)		
	1	1.5	2	1	1.5	2	1	1.5	2
50	0.51493	0.58245	0.59354	0.68274	0.69270	0.71091	0.80465	0.83922	0.92129
60	0.78437	0.72552	0.76822	0.78057	0.81426	0.86555	0.87692	0.88807	0.91637
70	0.48784	0.54817	0.57846	0.67015	0.79429	0.84596	0.75038	0.80148	0.90934

Table 4. Overall heat transfer coefficient of heat exchanger with longitudinal fins

Liquid Fluid Temperature (°C)	Overall Heat Transfer Coefficient (W/m² °C) Water Flow Rate (L /s) 0.06 Airflow Rate (m/s)			0.08 Airflow Rate (m/s)			0.06 Airflow Rate (m/s)		
	1	1.5	2	1	1.5	2	1	1.5	2
50	0.66827	0.83422	0.89645	0.81875	0.76681	0.98375	1.08141	0.98239	1.03226
60	0.65619	0.56959	0.60506	0.62806	0.76575	0.65631	0.66980	0.69207	0.95454
70	0.63820	0.60303	0.57887	0.63661	0.61292	0.73724	0.78968	0.72835	0.80801

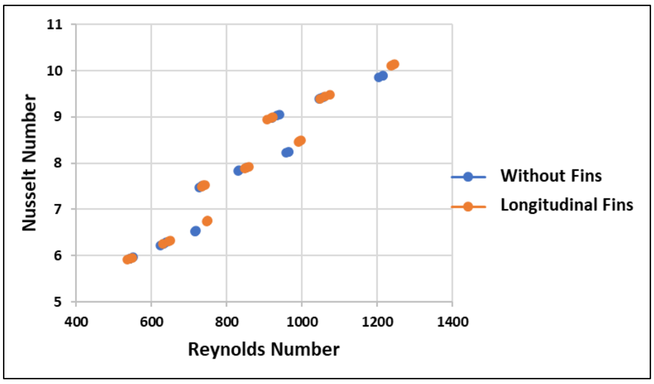

Figure 4. Comparison of the effect of pipes without fins and longitudinal finned heat exchanger on heat transfer

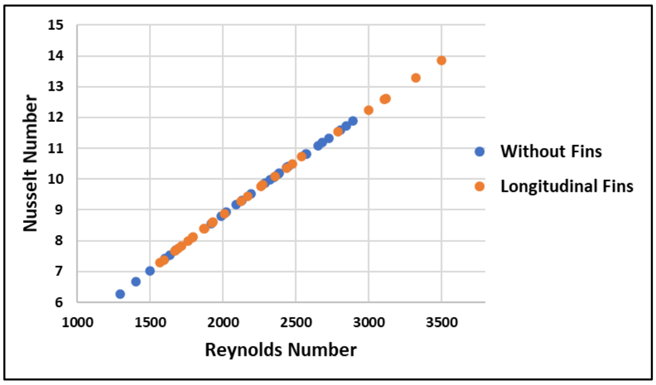

Figure 5. Comparison of the effect of heat transfer on without fins and longitudinal finned heat exchangers

3.3 Reynolds and Nusselt numbers in air-fluid flow

Figure 5 shows that the Reynolds number represents heat flow, and the Nusselt number represents heat transfer. The heat flow in the pipe without fins shows a smaller value than the pipe with the addition of longitudinal fins. The heat flow in the pipe without fins shows a smaller value than the pipe with the addition of longitudinal fins, the gain is equal 612.2463 or 21.182% compare pipe without fins. The heat transfer that occurs in the heat exchanger with the addition of fins is also faster than without fins because the pipe's cross-sectional area is wider, so the heat transfer can be faster by 1.9695 or 16.57%. Previous research explained that the backflow near the wake becomes stronger as the Reynolds number increases. As the Reynolds number increases, the Nusselt number reaches a maximum at the rear surface, appearing near the back saddle. The cliff body formation contributes to a larger and stronger wake eddy that increases heat transfer in this region [20].

The ratio of Reynolds and Nusselt numbers in the longitudinal finned pipes in this study was greater than that of the finned pipes. It is due to the difference in the area of heat transfer in the heat exchanger. The addition of fins accelerates heat transfer which is indicated by an increase in Reynolds number of 21.182% and Nusselt number of 16.57%. As a previous study mentioned, the rate of heat transfer, the Nusselt number of finned tubes, was relatively higher than that of ordinary tubes without fins [21].

Previous studies have shown that backflow near the wake becomes stronger with increasing Reynolds number. As the Reynolds number increases, the Nusselt number reaches its maximum at the rear surface. It appears near the rear saddle, where the cliff body formation contributes to a larger and more powerful eddy that increases heat transfer in this region [20]. A high Reynolds number will overcome the effect of fluid viscosity. Thus, the fluid velocity in the plate cavity increases while the velocity gradient decreases. Higher fluid velocity also means less contact time of fluid molecules to the plate surface [22]. Previous research described the variation of Nusselt number with Reynolds number for pipe surface roughness with different fins. As the Reynolds number increases, the Nusselt number increases. The increase in the Nusselt number is due to the increase in the intensity of the turbulence near the absorber plate [23]. The research results presented in Figure 5 explain that the greater the Reynolds number, the higher the Nusselt number.

The experimental results show that forced convective heat transfer is characterized by a nearly linear relationship between the Nusselt and Reynolds numbers, which agrees with the correlations available for fully developed turbulent convective heat transfer for tubes with uniform heat flux. Further investigation of the effect of the fin geometry can be carried out further by numerical methods. Previous studies also explained that shorter fin channels, in which the flow develops thermally and hydrodynamically, are more effective in heat transfer than longer channels, where the flow approaches or reaches full development. The heat transfer rate per channel decreases linearly with the increasing channel length but remains constant with an expanding number of fins [24].

3.4 The effectiveness-NTU

The effectiveness-NTU is the ratio of the actual heat transfer to the maximum possible heat transfer that a heat exchanger can carry out. The difference influences the effectiveness in the temperature of the fluid in and out of the heat exchanger. The addition of longitudinal fins increases the cross-sectional area so that it affects the amount of heat transfer. In addition, the fin material affects the thermal conductivity value of the heat exchanger, which ultimately affects the efficiency value [8].

The effectiveness of the hot fluid has the lowest heat capacity, which is higher than the effectiveness of the cold fluid when it has the lowest heat capacity [25]. The data processing results show that the lowest heat capacity in the hot fluid is higher than the heat capacity in the cold fluid. The comparison of the effectiveness NTU values between cross-flow heat exchangers without fins and upright fins are shown in Figure 6. The cross-flow heat exchangers without fins and with fins show the same results. With lower air mass flow velocity, the NTU value is greater than faster air mass flow. The addition of fins on the heat exchanger increases the effectiveness by 24.77%. Previous research on a higher average air velocity also resulted in a smaller NTU value. The greater NTU value, the greater effectiveness, and the curve became flatter as the NTU increased [26].

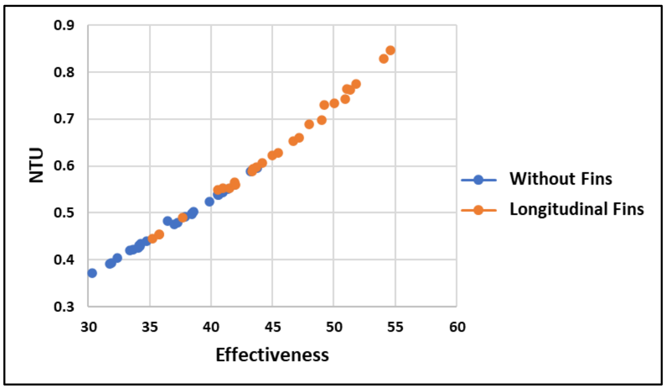

Figure 6. Comparison of NTU effectiveness on without fins pipes and longitudinal finned pipes heat exchanger

4. CONCLUSION

Based on the results of data analysis that has been carried out, the following conclusions can be drawn which is the addition of fins on the copper pipes increases the effectiveness of the heat transfer coefficient is increased by 17.3% with increased in Reynolds number 2.53% in liquid fluids, and 21.182% in air fluids. The addition of fins also increases the value of Nusselt number 2.56% in liquid fluid and 16.57% in air fluid. A cross-flow heat exchanger has also been shown to increase the LMTD value, heat transfer rate, and NTU value, and Its effectiveness is 24.77%.

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NOMENCLATURE

$\dot{m}$	Mass Flow Rate
ρ	Fluid Density
v	Flow Rate Speed
Q	Fluid Heat Transfer Rate
h	Heat Transfer Coefficient
c_p	Specific Heat
ΔT	Temperature Gradient
P	Effectiveness of Temperature on the Cold Fluid Side
R	Energy Capacity Rate Ratio
T	Temperature
F	Correction Factor ΔT_{LMTD}
A	Cross-sectional Area
U	Overall Heat Transfer Coefficient
q	Heat Flow Rate
π	Pi
r	Radius
t	Height
G	Air Mass Flow Rate
Re	Reynolds Number
d	Outer Pipe Diameter
μ	Kinematic Viscosity
Nu	Nusselt Number
Pr	Prandtl Number
K	Thermal Conductivity
di	Inner Pipe Diameter
s	Heat Exchanger Side
p	Heat Exchanger Length
l	Heat Exchanger Width
Dh	Hydraulic Diameter
a	Thermal Diffusivity
b	Heat Exchanger Side Width

ε	Effectiveness

Subscripts

h	Hot
c	Cold
1	Inlet
2	Outlet
0	Initial Condition
i	Under Certain Conditions
d	Design
n	The direction of Heat Flow (X, Y, Z)
$actual$	Value in Actual Condition
max	Maximum
min	Minimum

Abbreviation

PSO	Particle Swarm Optimization
GA	Genetic Algorithm
LMTD	Log Mean Temperature Difference
NTU	Number of Transfer Unit