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Parametric Analysis of a Plain-Fin Compact Heat Exchanger for a Small-Scale Gas Turbine

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ABSTRACT

The influence of the thermal parameters of a Plain-Fin Compact Heat Exchanger on its performance is examined in this paper. The objective of this work is to analyse the effect of the different flow and geometric parameters on the output performance of a plain fin compact heat exchanger (PFCHE) designed to be used on a small-scale gas turbine and how these parameters can be used for the optimisation of PFCHEs.

In this work, we examined the effects of the variation of input parameters of a plain-fin compact heat exchanger (fin length, fin height, fin thickness, mass flow rate of air and turbine exhaust gas) on the output performance of the heat exchanger (Overall heat exchanger efficiency, fin heat transfer efficiency and the outlet temperatures). The analytical expressions for the outlet temperatures and heat exchanger efficiencies were derived and analysed. Then the derived model was designed and simulated using CFD codes. Also, from the derived expressions, the performance model of the heat exchanger was programmed for analysis. From the results, it shows that the effectiveness (e-value), fin length, fin height and mass flow rates of the gases influence performance of a plain-fin compact heat exchanger. A 50% reduction in fin height can cause as much as an 18% increase in the fin efficiency of the heat exchanger. While a 50% increase in the effectiveness value can cause as much as a 40% increase in the outlet temperature.

Keywords: Heat exchanger, compact heat exchanger, gas turbine, fin efficiency, overall efficiency, mass flow rate, PFCHE, fin length, fin height, fin thickness.

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1. INTRODUCTION

A gas turbine is a type of continuous internal combustion engine. Small scale gas turbines are gas turbines with an output of up to 500KW. These internal combustion engines unlike the traditional Otto and Diesel engines, work following the Brayton cycle. For these engines to achieve efficiencies values of about 50% [1], with an electrical efficiency of about 35%, they need to be run with a recuperated energy cycle of varying configurations. Due to the need for lightweight, space-saving, and economic factors, the Compact Heat Exchanger has been used in wide variety of applications of gas turbines. Typical among these are gas turbines used in automobiles, cryogenics, aircraft and spacecraft, ocean power plants, and small-scale or micro gas turbines. With the use of a compact heat exchanger, the turbine's overall efficiency can be improved up to 90%. [2–5].

Different studies have been carried out to investigate the effects of the different hydraulic and geometric features of a heat exchanger on its performance. For example, the effects of parameters like the heat transfer, effectiveness, heat transfer coefficient, etc. on the performance of heat exchanger and how they could be used to reduce the size of the heat exchanger have been carried out by Thakre et al, 2016 and Rahul et al 2018 [6,7]. But their studies were limited to CH₄ and rectangular offset strip fin compact heat exchangers. Also, they considered just liquid as working fluid. Experimental analysis of the cold inlet temperature on the thermal performance of a heat exchanger have also been carried out [8]. But their study was limited on a two mini channel flat-tube with a multi-louvered fin compact heat exchanger. However, the working fluid was a mixture of a liquid and a gas. Also, effects of the mass flow rate on the pressure drop and heat transfer and the heat exchanger effectiveness value have been studied [9,10]. But these effects were not studied against the outlet temperatures of the fluids. Also, the fluids used were liquids.

The effects of the fin length, fin height and fin thickness on the performance of compact heat

exchangers have not been given much attention and the available literature is limited to fin-and-tube heat exchangers. Furthermore, most of the studies are limited to the evaluation of one parameter at a time. There are no analyses on the variation of more than two input parameters. Lastly, most of the studies are limited to liquids as the working fluid or using a combination of a liquid through one path and a gas through the other. The purpose of this work is then to analyse and study how the fin length, height, thickness of a heat exchanger with gases as working medium affects its performance and how these features can be used to alter the size of the heat exchanger. Also, to study the effect of the mass flow rate and effectiveness value on the outlet temperatures with respect to the heat exchanger's performance. The type of heat exchanger used in this study is the plain-fin compact heat exchanger with rectangular fins and a crossflow arrangement as shown in figure 3.

2. MATHEMATICAL MODELLING OF THE COMPACT HEAT EXCHANGER (PFCHE)

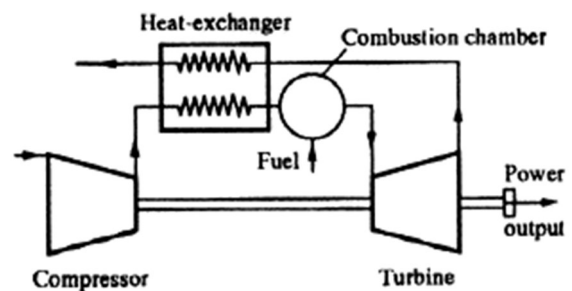


Figure 1 a. Gas turbine with heat exchanger

2.1. Thermodynamic Analysis of a Recuperated Heat Exchanger

Let's consider the figure 2 below [5] which describes the thermodynamic cycle of a gas turbine.

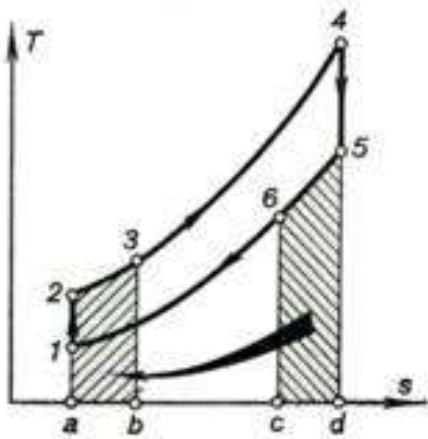


Figure 1 b. Constant pressure gas turbine with adiabatic air compression

The added heat is

$$q_1 = c_p(T_4 - T_3) \quad (1)$$

The rejected heat is

$$\begin{aligned} q_2 &= c_p(T_6 - T_1) \\ &= c_p(T_5 - T_1) \\ &\quad - c_p(T_5 - T_6) \end{aligned} \quad (2)$$

But

$$c_p(T_5 - T_6) = c_p(T_3 - T_2) \quad (3)$$

Which implies

$$q_2 = c_p(T_5 - T_1) - c_p(T_3 - T_2) \quad (4)$$

The thermal efficiency of the cycle will then be

$$\begin{aligned} \eta_{Brayton,regen} \\ = 1 - \frac{c_p(T_5 - T_1) - c_p(T_3 - T_2)}{c_p(T_4 - T_3)} \end{aligned} \quad (5)$$

The maximum possible degree of regeneration or regeneration fraction occurs at $T_3 = T_5$, i.e. at $\gamma_{max} = T_5/T_2$. In this case we have [5]

$$\eta_{Brayt,regen,max} = 1 - \frac{T_1}{T_5} \quad (6)$$

Thus, the thermal efficiency of a constant-pressure gas-turbine operating with maximum regeneration and adiabatic compression depends

only on the temperature of the gas at the end of adiabatic expansion, T_5 . So, to improve on the efficiency and performance of the gas turbine, there is a need to increase these temperatures while adjusting the other operating parameters of the recuperator.

2.2. Heat Transfer and Hydraulic Flow Analysis

Assuming the same heat capacity for both the hot and cold streams of air, we have [4], for the hot gas outlet temperature

$$T_{h,o} = T_{h,i} - \varepsilon(T_{h,i} - T_{c,i}) \quad (7)$$

And for the cold air we have,

$$T_{c,o} = T_{c,i} + \varepsilon \frac{\dot{m}_h}{\dot{m}_c} (T_{h,i} - T_{c,i}) \quad (8)$$

But taking into account the heat capacity of the gases, we have the corrected outlet temperature of the cold air stream as follows

$$T_{c,o} = T_{c,i} + \varepsilon \left(\frac{\dot{m}_h c_{p,h}}{\dot{m}_c c_{p,c}} \right) (T_{h,i} - T_{c,i}) \quad (9)$$

- Refined value of Cold stream AMT

$$T_{c,m} = \frac{T_{c,o} + T_{c,i}}{2} \quad (10)$$

From the above fluid property values, the NTU can be gotten.

- The Core mass velocities of the fluids (G)

The value of G is expressed as follows

$$G_i = \left[\frac{2g_c}{\left(\frac{1}{\rho}\right)_m} \frac{\eta_0 \Delta P}{P_r^{2/3} NTU} j/f \right]^{1/2} \quad (11)$$

Where, $i = h, c$ for the hot gas and cold air respectively.

- The Reynolds Number and the j and f factors

The Reynolds number is expressed as

$$Re_i = \left(\frac{GD_h}{\mu} \right)_i \quad (12)$$

For plane rectangular PFHE and laminar flow [11], we have

$$f_i = 12.892 Re_i^{-1.229} \left(\frac{b}{n} \right)_i^{0.452} \left(\frac{\delta_w}{n} \right)_i^{-0.198} \quad (13)$$

and

$$j_i = 0.454 Re_i^{-0.977} \left(\frac{b}{n} \right)_i^{0.455} \left(\frac{\delta_w}{n} \right)_i^{-0.277} \quad (14)$$

- The heat transfer coefficient (h)

The heat transfer coefficient can be calculated from the expression below

$$h_i = \left(\frac{jG C_p}{P_r^{2/3}} \right)_i \quad (15)$$

- Fin Efficiency (η_f)

The fin efficiency is expressed as follows

$$\eta_{f,i} = \left(\frac{\tanh(ml)}{ml} \right)_i \quad (16)$$

Where,

$$m_i = \left[\frac{2h}{k_f \delta} \left(1 + \frac{\delta}{l_s} \right) \right]_i^{1/2} \quad (17)$$

and

$$l_c = l_h = \frac{b}{2} - \delta \quad (18)$$

- Overall Surface Efficiency (η_o)

We have

$$\eta_o = 1 - (1 - \eta_f) \frac{A_f}{A} \quad (19)$$

- Overall Heat Transfer Coefficient

The overall heat transfer coefficient is evaluated as follows

$$\frac{1}{U} = \frac{1}{(\eta_o h)_h} + \frac{\alpha_h / \alpha_c}{(\eta_o h)_c} \quad (20)$$

Where,

$$\alpha_i = \frac{(b\beta)_i}{b_c + b_h + 2\delta_w} \quad (21)$$

And,

$$\frac{A_c}{A_h} = \frac{\alpha_c}{\alpha_h} = 1.0 \quad (22)$$

- Total Surface Area (A)

For the hot air stream, we have

$$A_h = NTU \frac{C_h}{U_h} \quad (23)$$

- Heat Transfer between Fluids

The effectiveness ε is defined as the ratio of the actual heat transfer rate and the maximum heat transfer rate. This is expressed as follows [12]

$$\varepsilon = \frac{Q}{Q_{max}} \quad (24)$$

The maximum heat transfer rate, Q_{max} , between the fluids can be determined from Equation 2.7 [13] as expressed below

$$Q_{max} = U_h A (T_{h,i} - T_{h,o}) \quad (25)$$

Thus, the actual heat transfer between the fluids is

$$Q = \varepsilon Q_{max} \quad (26)$$

2.3. Numerical Application

Design specifications: Due to the operating fluids all being gases, low cost, low space requirements, manufacturability, and operating temperatures and pressures, a regenerator of Plate-Fin Compact Heat Exchanger (PFCHE) type with Rectangular fins and a crossflow arrangement was chosen.

- Required Effectiveness: $\varepsilon = 0.8381$
- Fin height: $b_h = b_c = 15mm$
- Fin thickness: $\delta_w = 1mm, \delta_h = \delta_c = \delta_w$
- Heat transfer surface area density: $\beta_c = \beta_h = 2000m^2/m^3$
- Fin area/total area ratio: $\left(\frac{A_f}{A}\right)_h = \left(\frac{A_f}{A}\right)_c = 0.785$
- Hydraulic diameter: $D_{h,h} = D_{h,c} = 1.6mm$
- Fluid mass flow rates: $\dot{m}_h = 1.66kg/s, \dot{m}_c = 2.0kg/s$
- Pressure drop : $\Delta P_h = 9.0kPa, \Delta P_c = 8.79kPa$
- Plate Thermal heat transfer (Both plates are made of steel): $k_w = 33W/mK$
- Inlet Temperatures of gases: $T_{h,i} = 798K, T_{c,i} = 317K$

Outlet temperatures $T_{h,o}$ and $T_{c,o}$

$$\begin{aligned} T_{h,o} &= 798 - 0.8381(798 \\ &\quad - 317) \\ &= 394.87 K \end{aligned} \quad (27)$$

$$\begin{aligned} T_{c,o} &= 317 \\ &+ 0.8381 \left(\frac{1.66}{2} \right) (798 - 317) \\ &= 651.59 K \end{aligned} \quad (28)$$

Corrected value of cold stream outlet temperature

$$\begin{aligned} T_{c,o} &= 317 \\ &+ 0.8381 \left(\frac{1.66 \times 1.051}{2 \times 1.030} \right) (798 \\ &\quad - 317) = 658.42 K \end{aligned} \quad (29)$$

Refined value of outlet temperatures

$$\begin{aligned} T_{h,o} &= \frac{798 + 394.87}{2} \\ &= 596.44 K \end{aligned} \quad (30)$$

$$\begin{aligned} T_{c,o} &= \frac{317 + 658.42}{2} \\ &= 487.71 K \end{aligned} \quad (31)$$

- The Core mass velocities of the fluids (G)

$$\begin{aligned} G_h &= \left(\frac{2 \times 1 \times 0.8 \times 9.05 \times 10^3 \times 0.25}{(1/0.5804) \times 0.777 \times 10.790} \right)^{1/2} \\ &= 6.3047 kg/m^2s \end{aligned} \quad (32)$$

$$\begin{aligned} G_c &= \left(\frac{2 \times 1 \times 0.8 \times 8.79 \times 10^3 \times 0.25}{(1/0.6964) \times 0.776 \times 9.139} \right)^{1/2} \\ &= 7.0153 kg/m^2s \end{aligned} \quad (33)$$

- The Reynolds Number and the j and f factors

$$Re_h = \frac{6.3047 \times 0.0016}{305.8 \times 10^{-7}} = 329 \quad (34)$$

$$Re_c = \frac{7.0153 \times 0.0016}{270.1 \times 10^{-7}} = 415.57 \quad (35)$$

- The heat transfer coefficient (h)

$$h_h = \frac{0.00358 \times 6.3047 \times 1.051 \times 10^3}{0.777} = 30.53 \text{ W/m}^2\text{K} \quad (36)$$

$$h_c = \frac{0.002856 \times 7.0153 \times 1.030 \times 10^3}{0.776} = 26.59 \text{ W/m}^2\text{K} \quad (37)$$

- Fin Efficiency (η_f)

$$\eta_{f,c} = \frac{\tanh(40.184 \times 0.0065)}{40.184 \times 0.0065} = 0.9779 \quad (38)$$

$$\eta_{f,h} = \frac{\tanh(43.008 \times 0.0065)}{43.008 \times 0.0065} = 0.9747 \quad (39)$$

- Overall Surface Efficiency (η_o)

$$\eta_{o,h} = 1 - (1 - 0.9747) \times 0.785 = 0.980 \quad (40)$$

$$\eta_{o,c} = 1 - (1 - 0.9779) \times 0.785 = 0.983 \quad (41)$$

- Overall Heat Transfer Coefficient

$$\frac{1}{U_h} = \frac{1}{0.98 \times 30.53} + \frac{1}{0.983 \times 26.59} \quad (42)$$

$$U = 13.95 \text{ W/m}^2\text{K} \quad (43)$$

- Total Surface Area (A)

$$A = \frac{5.395 \times 1.745 \times 10^3}{13.95} = 674.86 \text{ m}^2 \quad (44)$$

- Heat Transfer between Fluids (\dot{q})

$$\dot{q} = 0.8381 \times 3785.2 \times 10^3 = 3180.8 \text{ kW} \quad (45)$$

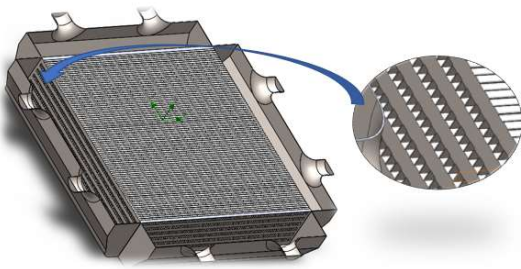
Tables 1 and 2 present a summary of the coefficients and geometric properties of the HE

Table 1. Heat Exchanger Coefficients and Fluid Properties

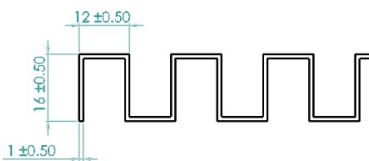
Coefficient/Property	Value	
	Cold Air Stream	Hot Gas Stream
Outlet Temperature $T_{i,o}$	685.42 K	394.87 K
NTU	9.139	9.139
Core Velocity G_i	7.0153 kg/m ² s	6.3047 kg/m ² s
Reynolds Number Re_i	415.57	329.84
j-factor j_i	0.002856	0.00358
f-factor f_i	0.014777	0.01963
Heat transfer Coefficient h_i	30.53 W/m ² K	26.59 W/m ² K
Fin Efficiency $\eta_{f,i}$	0.9779	0.9747
Overall Surface Efficiency η_o	0.983	0.980
Overall Heat Transfer Coefficient U	13.95	-
Maximum Heat transfer between Fluids Q_{max}	3795.2 kW	-
Heat transfer between Fluids Q	3180.8 kW	-

Table 2. Heat Exchanger Geometric features

Feature	Calculated Value		Rounded Value
	Cold Air Stream	Hot Air Stream	
Total Surface Area A (m^2)	674.86	674.86	-
Minimum free flow area A_o (m^2)	0.2851	0.2635	-
Frontal Area A_{fr} (m^2)	0.6740	0.6229	-
Air Flow Length (m)			
L_c	0.947	-	1.0
L_h	-	1.024	1.0
L_3 (Height)	0.648		0.5



(a)



(b)

Figure 2. (a) Cross section of plain-fin compact heat exchanger. (b) Fin geometry (rectangular fins)

RESULTS AND DISCUSSIONS

2.4. Boundary Conditions

The boundary conditions for the CFD analysis were as specified above in input data. The CFD analyses were carried with the assumptions of ideal operating states. A full adaptive and automatic meshing structure was used in the simulation. The figure 3a shows the meshed structure. In the final optimised structure (figure 3a), we used a minimum length and an adaptive mesh structure. This was appropriate for the simulation of our heat exchanger since it permits for an increased computational and storage savings, it accommodates the complexity of our structure and it is proper for our physical system since it makes the simulation independent of the mesh size.

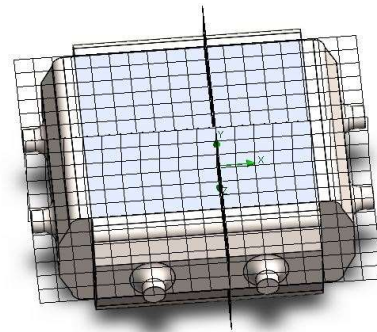


Figure 3a: Meshed Heat exchanger

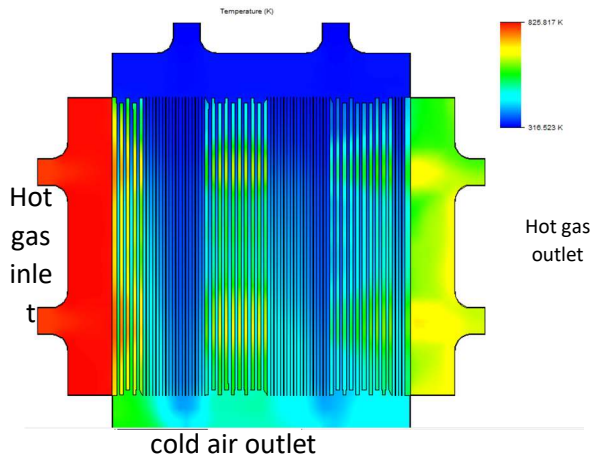
A fluid flow simulation was performed using SOLIDWORKS Flow Simulation. During the simulations we applied a finite volume method (FVM) for simulation algorithm. The results of the CFD study were as follows.

2.5. Heat distribution profile within the Heat Exchanger

During the simulation, the heat distribution profile within the HE was recorded. Figure 3b below shows the temperature variation within the HE. After 116 iterations, the HE attained a

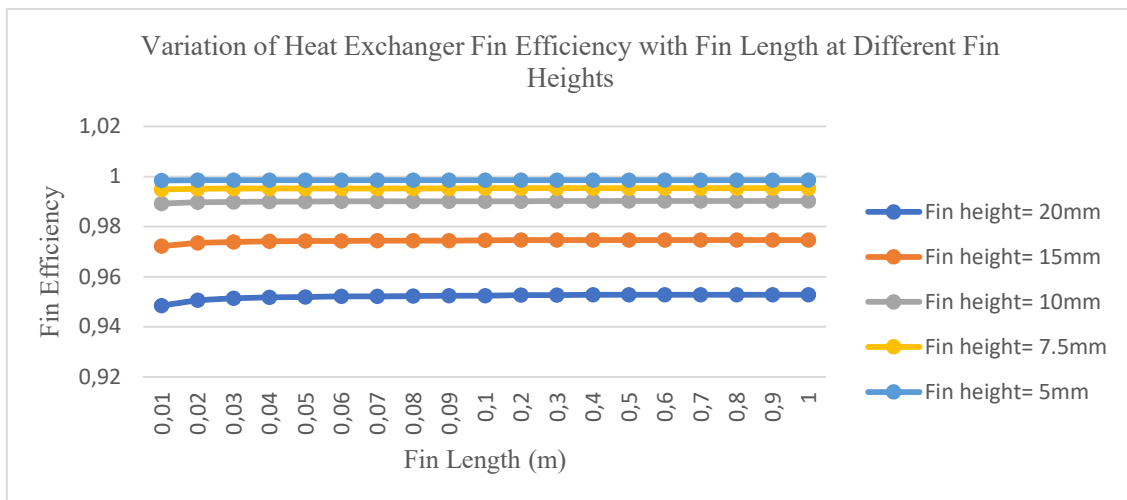
convergence point and reached maximum operating state. cold air inlet

2.6. Effects of fin length

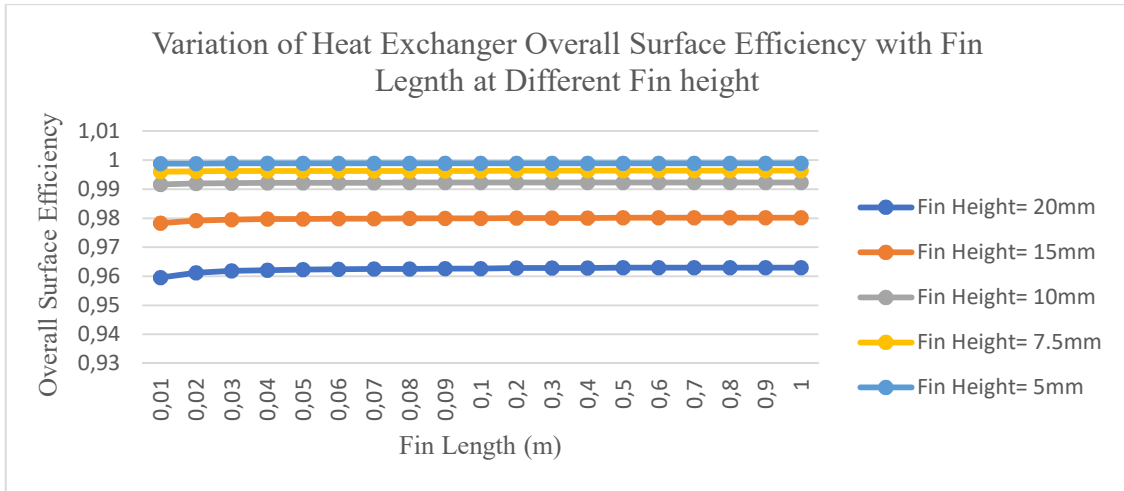


The figure 4 presents the relation between the heat exchanger fin efficiency versus the fin length at different values of fin height. It can be seen that the fin length has no effect on the fin efficiency. But as the fin height increases, there is an increase in the Fin Efficiency. As shown by figure 4, fin length does not have any effect on the Overall Surface Efficiency of the heat exchanger. An increase in fin height causes an increase on the Overall Surface Efficiency.

Figure 3b. Heat distribution in Heat Exchanger



(a) Fin Efficiency



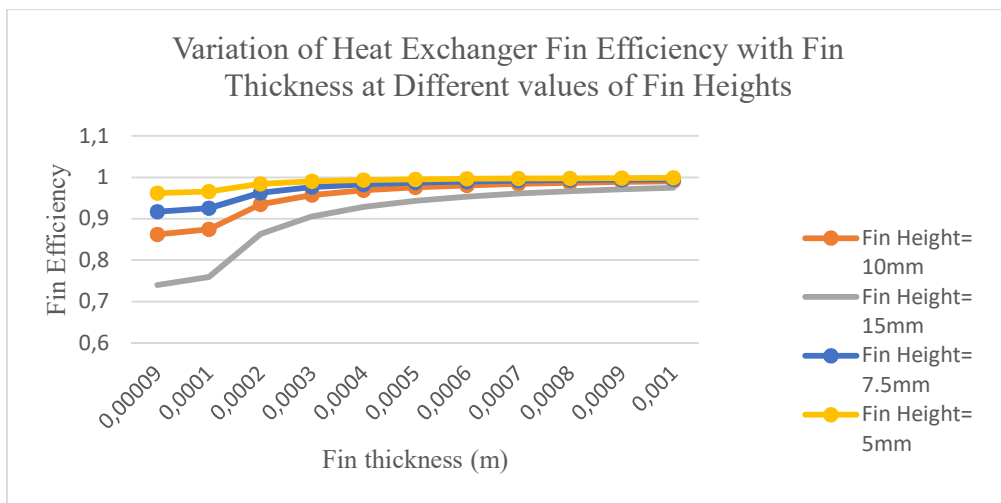
(b) Overall Surface Efficiency

Figure 4. Variation of Heat Exchanger Fin Efficiency and Overall Surface Efficiency versus Fin Length at different values of fin height

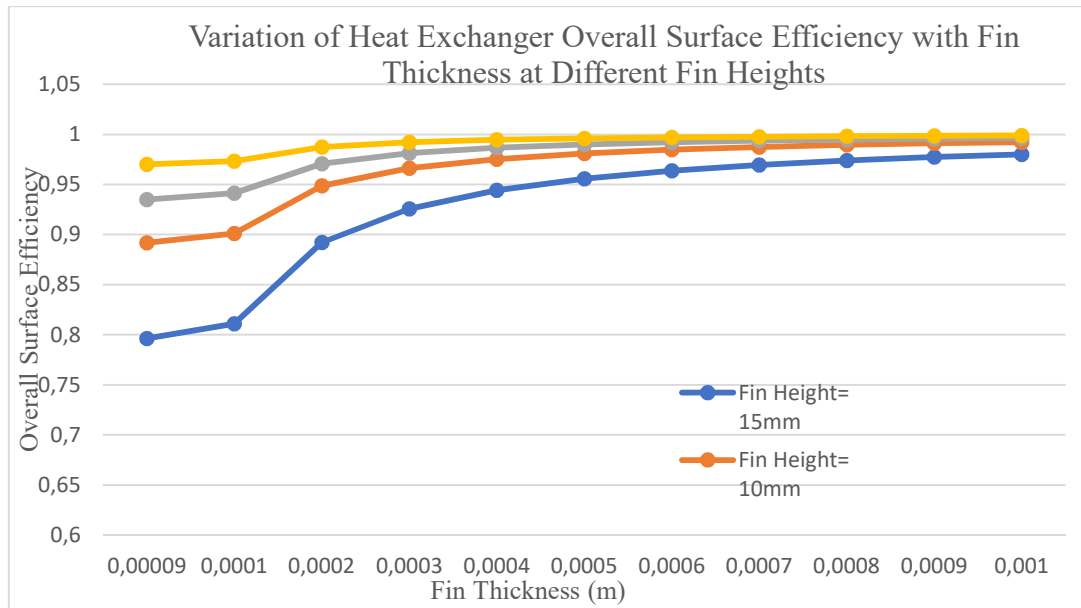
2.7. Effects of fin thickness

As presented in figure 5, the relationship between the heat exchanger fin efficiency and overall surface efficiency versus the fin thickness can be seen. The figure 5 also shows the effects of the fin height on these relations. As we can see on figure 5a, the fin thickness has a significant effect on the fin efficiency. As the fin thickness increases, the

fin efficiency increases as well. This continues until it reaches a peak value after which it stays constant. Thus, there is an optimum fin thickness where the fin efficiency is maximum and an increase in fin thickness causes no change on the efficiency. Also, as the fin height increases, this optimum fin thickness also shifts to the right. But it can be seen that, the increase in fin height has a negative effect on the fin efficiency. Furthermore, at lower values of fin thickness, a change in fin height has more effect on the heat exchanger efficiency.



(a) Fin efficiency



(b) Overall surface Efficiency

Figure 5. Variation of Heat Exchanger Fin Efficiency and Overall surface Efficiency with Fin Thickness at different fin heights (other parameters are constant as stated in the design specifications)

The figure 5b, presents the relationship between the overall surface efficiency versus the fin thickness. It can be observed that the overall surface efficiency has a positive relationship with the fin thickness. This positive relation continues until a maximum value is reached. As the fin height increases, there is a decrease in the overall surface efficiency. Finally, we observe that at lower values of fin thickness, an increase in fin height has more effect on the efficiency of the heat exchanger. The Same observations were made by Kourosch et al, 2018 in their experimental study of the effects of fin height, fin-tube contact thickness compact of a heat exchanger [14].

2.8. Effects of hot gas stream mass flow rate

Figure 6 presents the relation between the outlet temperatures of the two streams versus the gas mass flow and this study was done based on ideal conditions. It is observed that the gas mass flow rate has no effect on the hot gas stream. But it has a positive effect on the cold air stream.

The figure 7 presents the relationship between the hot gas stream mass flow rate and the cold air stream outlet temperature (the air from the compressor). This relationship is examined at different values of the Effectiveness Value, the cold air stream inlet temperature, and the hot gas stream inlet temperature. It can be observed that the hot gas flow rate has a positive

effect on the cold air stream outlet temperature. The graphs show an increasing linear relationship between the two variables. From figure 7a and 7b, an increase in effectiveness value positively affects the cold air stream outlet temperature. These observations were also made by Panthee, 2017 and Thakre et al, 2016 in their experimental observations of the effect of the mass flow rate over the effectiveness value [6,10]. Also, at lower mass flow rates, there little effect of effectiveness value over the outlet temperature. But as the mass flow rate increases, the effect becomes more important.

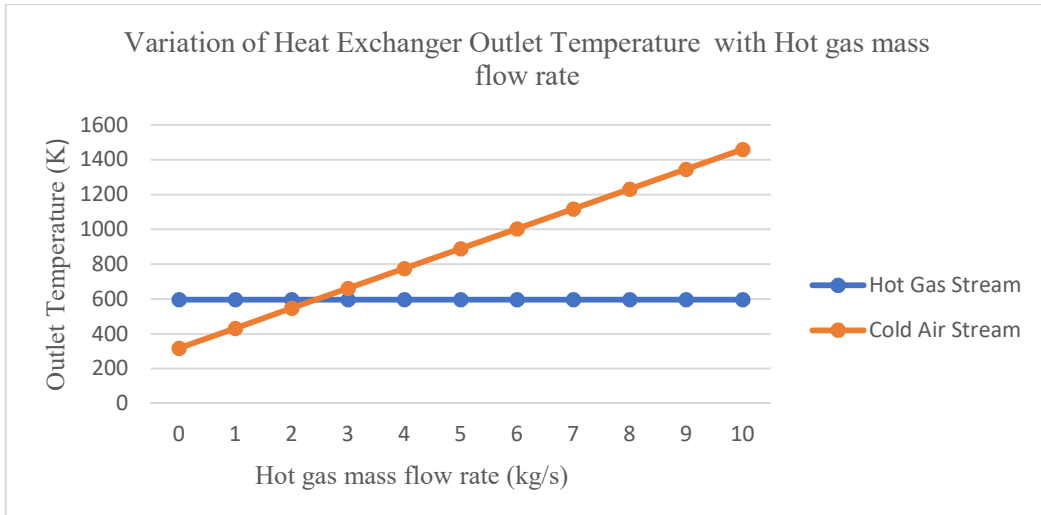
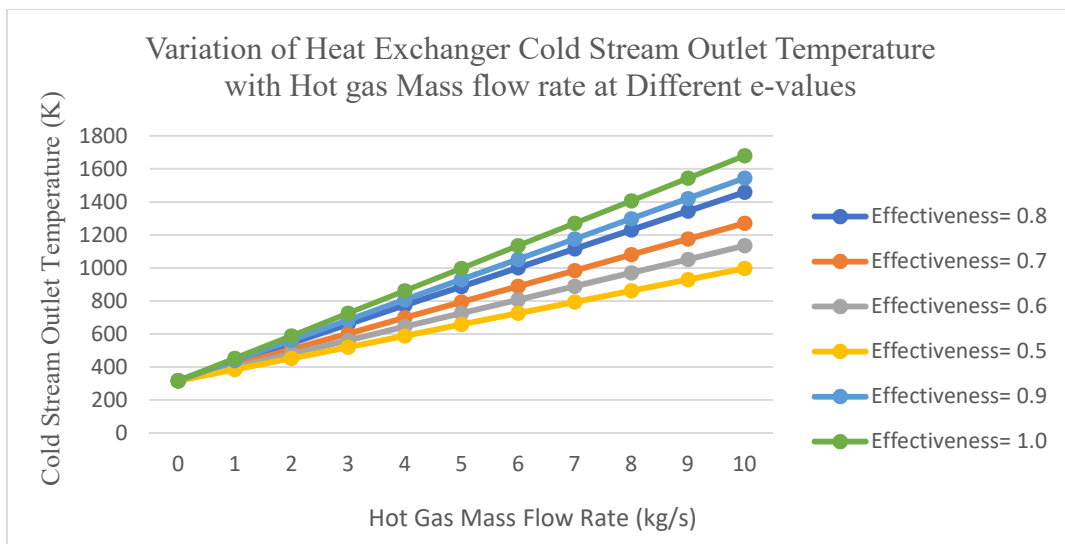


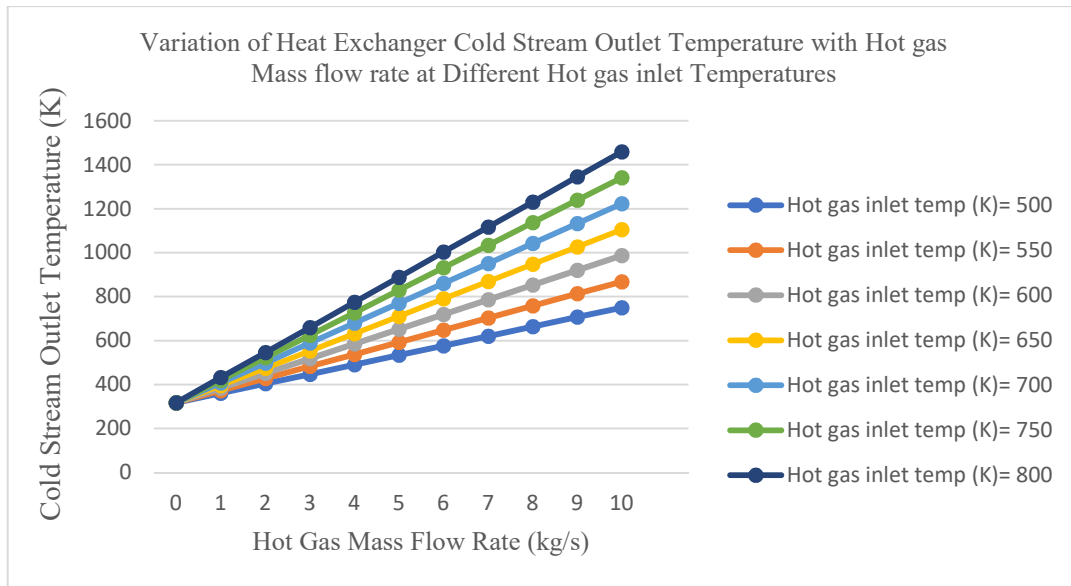
Figure 6. Cold air stream and Hot gas stream outlet temperatures versus hot gas mass flow rate

The increase in cold air stream inlet temperature has a converging effect on the cold air stream outlet temperature at lower mass flow rates (figure 7c). This happens until it reaches the convergence point. From this point, as the cold air stream inlet temperature increases, the cold air

outlet temperature decreases. This point of convergence is the when the cold air stream inlet temperatures equals the hot gas stream inlet temperatures. So, after this point, the direction of heat flow interchanges.



(a) Effectiveness value (e)



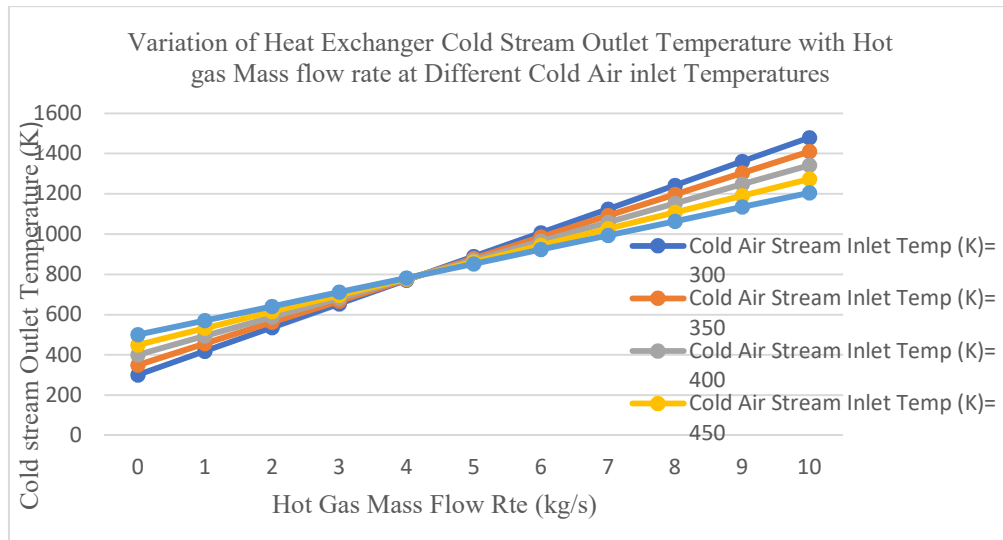
(b) Hot gas stream inlet temperature

2.9. Discussions

The fin efficiency, overall surface efficiency and outlet temperatures of a PFCHE depends on the operating conditions (mass flow rates and inlet temperatures of fluids), and the heat exchanger geometric features (fin length, fin height, fin thickness and the effectiveness value).

From figures 4 and 5, it can be seen that; using thinner and shorter fins will produce efficient and more compact plain fin compact heat exchangers. A 50% reduction in fin height can cause as much as an 18% increase in the fin efficiency of the heat exchanger. This can be considered when there is a problem of space. But this will also require advanced manufacturing techniques and thus incurring more cost.

From the figures 6 and 7, it can be seen that plain fin compact heat exchangers with higher effectiveness values attain higher outlet temperatures. For example, a 50% increase in the effectiveness value can cause as much as a 40% increase in the outlet temperature. Furthermore, as the fluid flow rate increases, this effect of the e-value on the outlet temperature also increases (with values of 15% at 1 kg/s, 34% at 5 kg/s and 40.1% at 10 kg/s). Also, to obtain higher outlet temperatures on the cold stream, the hot gas flow rate (flow velocity) can be increased. These results were obtained based on ideal conditions but experimental works done by other researchers obtained similar results. For example in Panthee's work, for plate heat exchangers he registered a 20 % increase in e-value and a 9 % increase in outlet temperature for a 50% change in mass flow rate of hot gas [10].



(c) Cold air stream inlet temperature

Figure 7. Variation of Heat Exchanger Outlet Temperatures with Hot gas Mass flow rate (other parameters are constant as stated in the design specifications)

3. CONCLUSIONS

In this study the parametric analysis of the PFCHE was carried out. The different flow and geometric parameters of the heat exchanger were studied to evaluate their influence on the performance of the heat exchanger. The results of the simulation from the modelling of the influence of heat exchanger parameters showed that the hot gas mass flow rate, effectiveness value, fin height, fin thickness and the inlet temperatures of the fluids have effect on the performance of the PFCHE. The summary of these results are as follows:

- Increasing the fin thickness and fin height reduces the fin efficiency and thus the overall efficiency of PFCHEs
- There is an optimum fin thickness after which increasing the fin thickness no longer influences the fin and overall surface efficiencies.
- The fin length has little effect on the fin and overall surface efficiencies of the PFCHE.

- Increasing the effectiveness value and the hot gas flow rate also increases the cold stream outlet temperature of the PFCHE.
- After the convergence point, the performance of the heat exchanger inverses due to change in heat flow direction.

4. REFERENCES

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