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Analytical and experimental investigation of double pipe heat exchanger for optimization of longitudinal fin profile

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ABSTRACT

In the present study the performance of the heat transfer process in a given heat exchanger is determined for longitudinal fin profiles (rectangular). The performance of a double pipe heat exchanger is analyzed in two parts. First part is optimization. In optimization for Numerical analysis a Matlab program was created .The theories of transient heat transfer in double-pipe heat exchangers were explained and followed by literature correlations. This program will serve to optimize the fin height fin so as to obtain maximum possible heat transfer without any wastage of material at a given length and inlet conditions. Also all the character like efficiency, pressure drop, effectiveness, heat transfer coefficient, outlet temp. of both fluid, overall heat transfer coefficient studied same time for all fin height because output of program is in tabular form. In second part Numerical analysis was carried out in a counter flow double pipe heat exchanger for optimized fin for varied mass flow conditions. Base width and height of the fin were kept constant .Exprimental results and analytical result from MATLAB program compared. They give good relation.

Keywords : **Fin height , optimization, double pipe heat exchanger ,longitudinal fin, effectiveness .**

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I.INTRODUCTION

Heat exchangers are used to transfer that energy from high temperature liquid to low temperature. Temperature of incoming and outgoing streams is important .These streams can either be gases or liquids. As per requirement heat exchangers raise or lower the temperature of these streams by transferring heat to or from the stream.

Double pipe heat exchangers are simplest devices in which fluid separated by cylindrical wall. Application of double pipe heat exchanger is in high temperature and high pressure .as compared other exchanger they required space but they are fairly cheap. Hence for the given design and length of the heat exchanger heat transfer enhancement in a double pipe heat exchanger is possibly achieved by several methods. These techniques are divided into active and passive techniques. Active methods involve internal parameter optimization. Another method is the passive method in which stimulation by external power such as surface coating, surface roughness and extended surfaces.

Several papers have studied and concluded that proper fin selection can help in obtaining substantial increase in the

values of heat transfer coefficients and effectiveness of a heat exchanger. This also demonstrates the fact that Fins provide a thermodynamic advantage. Thereby designing a heat exchanger at the optimum fin height can lead to reducing capital costs and increasing savings. Also providing cheap materials for the fin and expensive durable materials for thinner pipes can increase Heat Exchanger Life span and save capital costs as well.

II.LITERATURE REVIEW

Finned-tube heat exchangers are common devices; however, their performance Characteristics are complicated. Optimization causes lowering the fin mass by means of changing the shape of the fin also improving. Mass reduction, flow direction causes enhancement in the temperature changes on the fin contact surfaces. Hence it is important to pay attention on optimization. Hence a comparison study of heat transfer characteristics using different configurations of fins is very essential.

Wilson[1915]Experimental study on the air side performance of compact slit fin-and-tube heat exchangers was carried out. Performed an experimental work in which he developed a graphical Method of calculating the water-side heat transfer coefficient as a function of water velocity. Experimental results comparison shows that interrupted surface gives better result than plain surfaces. Wang et al. (1998) performed a experimental study of eight finned-tube heat exchangers. There is a systematic variation of parameters that define the heat exchangers studied. This study is similar to the variation of parameters in the present study. The louver height and major louver pitch are not known. Wang et al. concluded that the effect of fin pitch on heat transfer performance is negligible for four-row coils having Re > 1,000 and that for Re < 1,000 heat transfer performance is highly dependent on fin pitch.

Deepali (2012) in this study heat transfer enhances due to the twisted wire brush inserts . This technique also enhances pressure drop. Due to the twisted wire in tube turbulence created and swirl flow generated, the convective heat transfer obtained more than plain tube. Zukauskas and Ulinskas (1998) developed correlations for the pressure drop of a staggered bank of bare tubes (no fins) in cross flow. These correlations give pressure drop as a function of geometry over a range of Reynolds numbers. Geometric parameters included in the analysis are: tube diameter, transverse tube spacing, longitudinal tube spacing, and number of tube rows. Zukauskas and Ulinskas discuss several possible variations that influence the pressure drop, including

1. Wall to bulk viscosity.
2. Property variations through the bank of tubes.
3. Acceleration pressure drop arising from temperature rise.

Webb and Gray (1986) find out correlations between heat transfer coefficient and fin friction factor from own experimental data as well as other sources. Sixteen heat exchanger configurations were used for experimentation. That experimental data used to develop the heat transfer coefficient correlation; the resulting RMS error is 7.3%. Similarly, data from 18 heat exchanger. configurations were used to develop the fin friction factor correlation; the resulting RMS error is 7.8%. A multiple regression technique was used with inputs being geometric quantities: transverse tube spacing, longitudinal tube spacing, tube diameter, number of tube rows, and fin spacing. Entrance and exit pressure drops were not included in the fin friction factor. The application of this correlation to compare with the coils in thepresent study stretches the limits of this correlation; the St/D parameter is 2.63 in the present study compared to the applicable 1.97 – 2.55 range. All other parameters are within their respective ranges.

The objective of present study is to develop heat transfer augmentation

For this optimization of fin height obtained from numerical analysis . In optimization for Numerical analysis a Matlab program was created .This program calculate all the parameter for different height within a fraction of second which

III.NUMERICAL SCHEME

The double pipe heat exchanger selected is a counterflow heat exchanger. Hence, the correlations for LMTD and $\in$ for counterflow heat exchangers derived holds true. As per the definition the mean temperature difference can be given by LMTD, thus

$$\Delta\ T_m = \Delta\ T_{lm} \quad (Eq.1)$$

Therefore the heat transfer equation reduces to $Q = U\,A\,\Delta\,T_{lm}$

In terms of the inlet and outlet temperatures the heat transfer Q can be written as (Eq.2)

Where suffixes h and c stand for hot and cold fluids and i and o indicate inlet and outlet conditions respectively. But main problem is the outlet conditions are unknown, so the above method cannot be directly applied. So we use the $\in$ –NTU method used here. In the £ –NTU method, the effectiveness is given by

$$\in = \frac{1 - [\exp[-NTU\,(1-R)]}{1 - R\exp\,[-NTU\,(1-R)]}$$

(Eq.3)

In recent times the P-R method has become very popular because it does not require specification of the fluid with minimum heat capacity a priori. But we decided to go with the usual $\in$–NTU method.

In the above section, the overall heat transfer coefficient U was used without any specific reference to its evaluation. Now we take up this task. So as per the definition of overall heat transfer coefficient we can write

$$\frac{1}{U_o A_o} = \frac{1}{h_o A_o} + \frac{R_o}{A_o} + \frac{RW}{A_i} + \frac{R_i}{A_i} + \frac{1}{h_i A_i}$$

(Eq.4)

$$Rw = \frac{di\ \ln\,({}^{do}/_{di})}{2\,k\,w}$$

Kw is the tube wall thermal conductivity .The tube resistance term can be determined for the case of steady state conduction through the walls of a symmetric cylinder. The overall heat transfer coefficient Uo is defined on the outside surface area of the (plain) pipe Ao. For double pipe configuration heat transfer area can be put as $A = \pi\, d\, L$

where L is the length of the tube surface and d is the corresponding diameter (do for outer and di for inner surface giving Ao and Ai respectively). This reduces the Eq.4

$$\frac{1}{Uo} = \frac{1}{ho} + Ro + \frac{di \ln(\frac{do}{di})}{2\,kw}\left(\frac{do}{di}\right) + \frac{Ri\, do}{di} + \frac{1}{hi}\left(\frac{do}{di}\right)$$

(Eq.5)

Thus, the task reduces to determination of film transfer coefficient hi and ho . For tube side coefficient well known correlation such as Dittus-Boelter equation can be used. However, a more frequently used correlation is the Sieder-Tate equation. But we used the former for simplicity in calculations. The important difficulty in using these correlations is the determination of fluid property because the equations mentioned above suggest to use the fluid properties at average film temperature which is the mean between bulk mean temperature of the fluid and the wall temperature. In

our problem the fluid temperature vary from inlet to outlet and the wall temperature is not known. Hence, two approaches are possible.

• First, calculations are carried out with properties at the mean bulk temperature on each side where

$$T_{mean\ bulk} = \frac{T_{bulk,in} + T_{bulk,out}}{2}$$

On the basis of result obtained the wall temperature at each end can be calculated by temperature drop from the hot stream and temperature rise in the cold stream. At the hot fluid inlet wall temperature may be approximately calculated as

$$\frac{T_{wall} - T_{c,out}}{Rcold} = \frac{T_{h,in} - T_{wall}}{Rhot}$$

(Eq.6)

And at the cold fluid inlet the wall temperature is calculated as

$$\frac{T_{wall} - T_{c,in}}{Rcold} = \frac{T_{h,out} - T_{wall}}{Rhot}$$

(Eq.7)

Here the wall resistance is divided into two (may be equal) parts and added to film and fouling resistance of each side to get Rcold and Rhot. With these wall temperatures on

avoided complexicity of calculations. Program based on theories of transient heat transfer in double-pipe heat exchangers were explained and followed by literature correlations. This program will serve to optimized the fin height fin so as to obtain maximum possible heat transfer without any wastage of material at a given length and inlet conditions. Also all the character like efficiency, pressure drop ,effectiveness ,heat transfer coefficient, outlet temp. of both fluid, overall heat transfer coefficient studied same time for all fin height because output of program is in tabular form

In second part Numerical analysis was carried out in a counter flow double pipe heat exchanger for optimized fin for varied mass flow conditions. Base width and height of the fin were kept constant .

For shell side heat transfer coefficient the equation for heat transfer coefficient in annulus has to be used. For turbulent flow the same correlation can be used for tube flow only the diameter should be replaced by equivalent diameter, de. Here,

both the ends the mean wall temperature can be calculated and then the properties can evaluated at the mean film temperature given by

$$T_{f,mean} = \frac{T_{meanbulk} + T_{meanwall}}{2}$$

(Eq.8)

This has to be iterated and within two to three iterations and a good converged result can be obtained.

• If iteration is to be avoided we can assume that both U and ΔT vary linearly within the heat exchanger. This gives an average heat transfer coefficient Um as

$$Um = \frac{A\ (U2\,\Delta\ T1 - U1\ \ \Delta\ T2}{\ln\left(\frac{U2\,\Delta\,T1}{U1\,\Delta\,T2}\right)}$$

(Eq.9)Here the suffixes 1 and 2 refer to the ends of the heat exchanger.

one important distinction has to be made between thermal and hydraulic

Performance. The fluid friction for the annular space takes place at both the inner wall of the outer tube (shell) and outer wall of the inner tube, whereas, heat transfer takes place at the outer surface of the inner tube. Thus, for evaluating the heat transfer coefficient of the annular side, the equivalent diameter is calculated as

$$de = \frac{4\,x\ flow\ Area}{perimeter\ of\ heat\ trnsfer} = \frac{4\,x\,\frac{\pi}{4}(d_s^2 - d_o^2)}{\pi\, do} = \frac{(d_s^2 -}{d}$$

For fluid flow and the definition of the Reynolds Number, the hydraulic diameter should be used which is given by

$$d_h = \frac{4\,x\ flow\ area}{wetted\ perimeter} = \frac{4\,x\frac{\pi}{4}(d_s^2 - d_o^2)}{\pi\,(ds + do)} = d_s - d_o$$

$$Re = \frac{V\,dh}{\gamma}$$

$$N_{uo} = \frac{h_o \ d_e}{Ko}$$

(Eq.11)

where, o indicates outer surface of inner tube and ko is the thermal conductivity of the fluid in the annulus. However, the above quantities are for unfinned units only. For finned construction the
details are given in design section later.
us define the previously defined quantities in the light of finned construction.
Hydraulic Mean Diameter,

$$d_h = \frac{4 \ x \ NFA}{Wp}$$

Thermal Equivalent Diameter, $d_e = \frac{4 \ x \ NFA}{wp - \pi \ ds}$

where, NFA is the Net Flow Area for fluid flow in the annulus and Wp is the Wetted Perimeter. These parameters are given by $NFA = \frac{\pi \, ds^2}{4} - [\ \frac{\pi \, d \, o^2}{4} + Nf \ Hf \ \delta \ f]$

(Eq.12)

$$Wp = \pi \, ds + \pi \, do + 2 \, Nf \ Hf - Nf\delta f$$

A typical finned tube with the geometrical parameters is shown in following fig.

Tube Outside Dia. (mm)	No. Of. Fin
25.4	20
48.3	36
60.3	40
73	48

Fig.1: - Thermal Deign Data Table

With the assumption of absence of contact resistance between the tube and the fins, a constant heat transfer coefficient over the entire finned length and fin Biot Number along thickness small enough to consider it one dimentional, the fin efficiency can be calculated as

$$\eta_f = \frac{\tanh mHf}{mHf}$$

the total area Atot is given by $A_{total} = A_f + A_b$

Where , A_f = total fin surface area = $2 \, N_f \, L_f \, H_f$

A_b = unfinned bare tube area= $L_f \ (\pi \, do - N_f \ \delta_f)$

The total fin efficiency of the finned surface neglecting heat transfer from the fin tip is given by

DESIGN OF LONGITUDINALLY FINNED DOUBLE PIPE HEAT EXCHANGERS :

In the last section description of analysis of simple unfinned tube heat exchanger for understanding the process of heat transfer and important issues such as evaluation of properties and heat transfer in annulus. This section describes design approach used in the case of a finned construction. Now let

$$\eta_f = \frac{A_f \ \eta_f + A_b}{A_{total}} = \eta_f \left(\frac{A_f}{A_{total}} \right) + (1 - \frac{Af}{A_{total}} \ )$$

Since the total efficiency affects the fin surface as well as the fouling surface, the fouled surface heat transfer coefficient can be given by

$$\frac{1}{U_{fs}} = R_o + \frac{1}{ho}$$

The fin perimeter Pf can be approximated as 2Lf since Lf>>δf. This reduces the value of m is

$$m = \sqrt{\frac{2 \ U \ f_s}{k_w \ \delta_f}}$$

Thus, the total finned surface efficiency acts as a correlation factor to Ufs based on the total area. Thus, the overall heat transfer coefficient can be calculated as

$$\frac{1}{Uo} = \frac{1}{U_{fs} \, \eta_f} + Rw \left(\frac{A_{total}}{A_o} \right) + \left(Ri + \frac{1}{hi} \right) \frac{Atot}{Ai}$$

$$Rw = \frac{do \ln ({}^{do}/_{di})}{2 \, k \, w}$$

Now the design of a double pipe unit can be carried out based on the above equations .The only
difference between the finned and unfinned construction being, in case of unfinned Atot = Ao and η =1.
The design we did was only for the rating or performance evaluation.
For rating purpose the information available are

- The pipe dimensions (tube and shell)
- The fin geometry and numbers (Fin Height was varied from 0 mm to maximum possible for the given shell size)
- The material data for pipes and fins
- The inlet temperatures and property data tables for both the fluids

All the thermal data were determined at unfinned condition as well as at all possible fin heights in 1mm steps. In
computer program, developed using Matlab to perform this design .

IV.OPTIMZATION USING MATLAB PROGRAM

Based on correlation of previous section a computer program is written in MATLAB R2010. The program is able to perform the performance evaluation of any longitudinally finned double pipe heat exchanger if the required geometrical data and fluid properties at inlet are provided The Uo, Heat Transfer, NTU, effectiveness and outlet temperatures of hot and cold fluids at unfinned stage, and at all possible fin heights are displayed as a table. From the table we can see

that the overall heat transfer coefficient is continuously decreasing since the surface area is increasing even though there is an increase in the heat transfer coefficient. But the effectiveness and heat transfer values are found increasing. This is because the small drop in Uo is compensated by a large increase in area, Atot. The effectiveness will not stop increasing within our size limits. So to find an optimum fin height, we cant just take the point of maximum effectiveness.. To solve this problem a graph is plotted with the fin height on the x-axis and effectiveness on the y-axis.

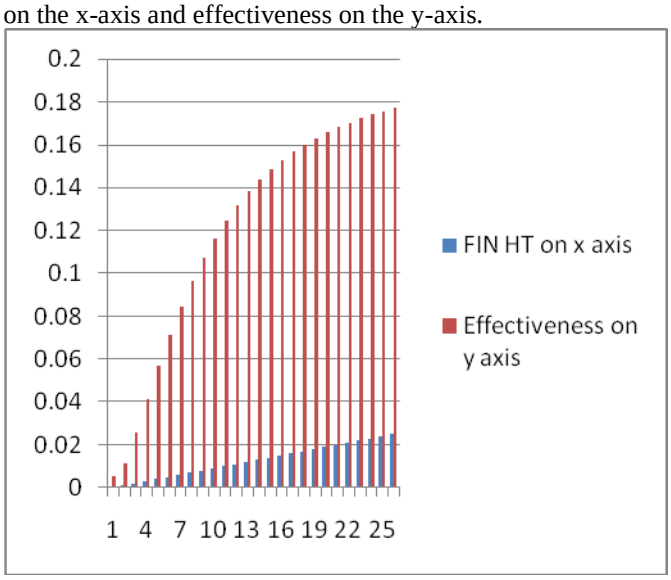

Fig 2: Variation of effectiveness for different fin ht

Beyond particular point graph become almost horizontal i.e. optimum height get at 15 mm. To confirm this again graph is plotted fin height against heat transfer

The graph shows increase in heat transfer falls to almost zero at a particular fin height .So beyond this point increasing fin height results more in wastage of material and thus more cost than in increase of heat transfer. Therefore this point fixed as the optimum fin height.

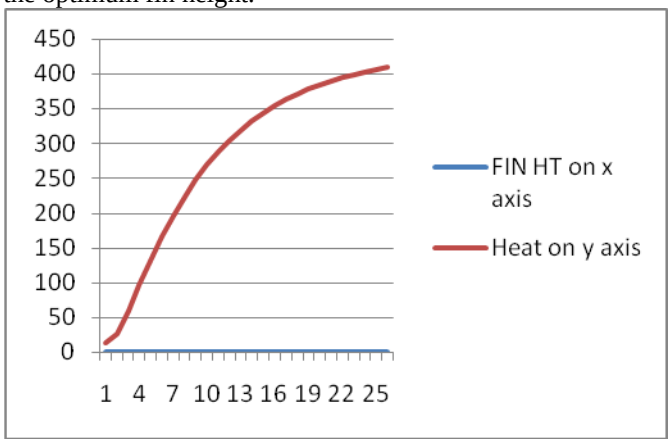

Fig 3: Variation of heat transfer for different fin ht

Other parameter also verified as well as studied by using computer program

A double pipe heat exchanger with rectangular longitudinal fins having Base width of the fin was 1mm (18 degrees) kept constant throughout the study. Analysis was done using fin heights from 0mm to 25mm were placed circumferentially around the thickness of the inner tube which remained constant throughout the study. Experimentation was carried out for various mass flow rates temperature distribution at the outlet for fin height = 15mm heat exchanger is shown in Fig .

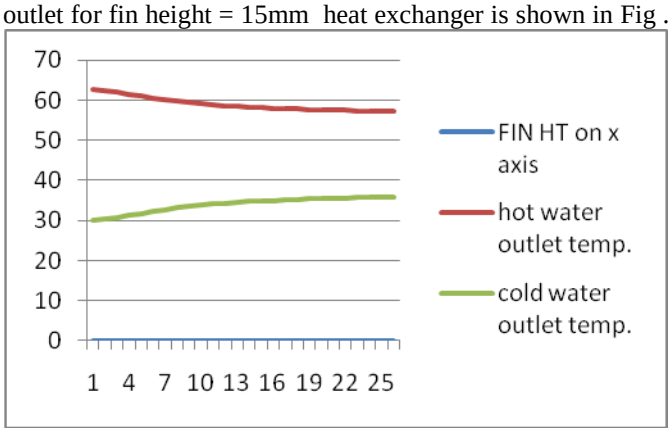

Fig: variation of temp. difference for different fin height

V.EXPERIMENTAL SET UP AND EXPERIMENTATION

Fig .5 shows the experimental set up of the concentric tube double pipe heat exchanger. It consists of inner tube with rectangular fin made of copper where in hot water flows from a geyser attached to it. Cold water flows in the annulus which can be admitted at any one of the ends enabling the heat exchanger to run as a counter flow exchanger. This can be done by operating the valves provided. Specifications of the heat exchanger are mentioned in Tab. 1. Temperatures of the fluid can be measured using thermocouples with digital display. Flow rates of hot and cold water can be measured by rotometers connected to the pipes.. The inlet temperature of the hot fluid was maintained at 63ºc and the cold fluid at 30 ºc. Experiments were conducted for counter flow arrangement at various mass flow rates of hot water (mch) ranging from 0.0168kg/s to 0.0126 kg. Outlet temperatures of the hot water and cold water were noted each times. This experimental process was repeated These experimental results were then compared with the inlet and outlet temperatures found in the theoretical analysis of the problem. This obtained from MATLAB program.

Fig 6 : Exprimental set up of double pipe heat exchanger

S. No.	Specification	Dimension (mm)

1	Inner diameter	12.5
2	Thickness of the inner tube	1
3	Inner diameter of the outer tube	40
4	Length of the heat exchanger	700
5	Fin Height	15
6	Fin thickness	1

Fig 6 : Table of specification of double pipe heat exchanger

S. No.	Massflow rate (kg /s)	Exprimental measured hot outlet Water temp.	Analytical measured hot outlet Water temp.	Exprimental measured cold outlet Water temp.	Analytical measured cold outlet Water temp.	Qactual	Qanalytical
1	0.0168	54	57.96	37	35.04	491.568	354.38
2	0.0155	53	57.86	36	35.14	388.74	333.44
3	0.0144	53	57.77	35	35.23	300.96	315.28
4	0.0134	52	57.68	34	35.12	224.048	299.08
5	0.0126	50	57.60	32	35.40	105.336	284.6

Fig 7 :Table : Comparison of experimental and computer program result for optimum fin height 15 mm

VI. RESULT AND DISCUSSION

The results from the experimental and analytical were compared in above table. The total heat transfer from heat exchanger were compared with There is a difference between the heat transfer from the model and the experimental data. It is not known what has caused this discrepancy; it could be due the oversimplification of the model, unconvergence, experimental error and/or many other possible factors.

The results obtained from experiments and analytical on heat transfer and fluid outlet temp. compared

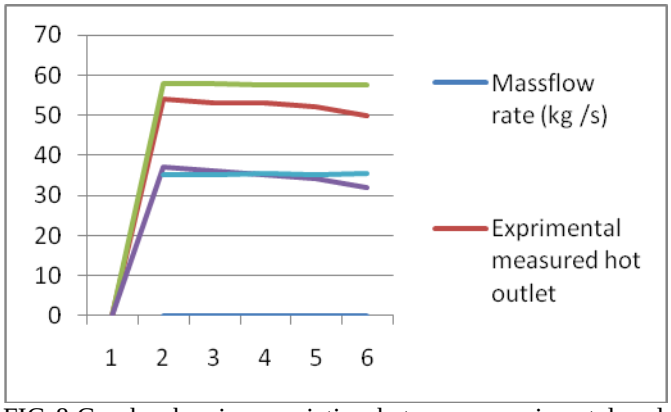

FIG 8:Graph showing variation between experimental and analytical value of outlet temp

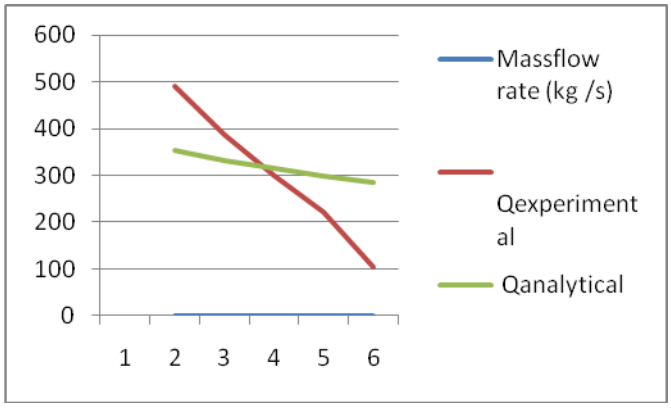

FIG 9:Graph showing variation between experimental and analytical value of heat transfer

VII. CONCLUSION

From above discussion Optimization Program and experimentation result hold small difference but it hold appropriate good relation . This small model validate the MATLAB program so it can used for other double heat pipe exchanger by changing data performance of heat exchanger at different height can analyzed eaily ,also all the parameter studied simultaneously without complication. Hence complexity of calculation avoided .e.g.

FIN HT	Re. No.	Nu.No	h	Total Area	Uo	Δp	Effectiv.	Heat	Tho	Tco
0.000	456.460	2.690	14.290	0.028	14.010	560.002	0.005	202.700	62.818	30.181
0.001	407.186	2.456	19.995	0.042	19.184	502.473	0.011	26.108	62.629	30.372
0.002	334.877	2.101	29.102	0.070	26.421	263.037	0.026	59.226	62.158	30.843
0.003	284.377	1.843	36.460	0.098	30.948	178.147	0.041	95.674	61.640	31.362
0.004	247.112	1.647	42.719	0.126	33.657	134.681	0.057	131.695	61.128	31.875
0.005	218.482	1.493	48.251	0.154	35.127	108.266	0.071	165.468	60.648	32.356
0.006	195.797	1.367	53.284	0.182	35.749	90.513	0.085	196.231	60.211	32.794
0.007	177.380	1.264	57.968	0.210	35.792	77.762	0.097	223.793	59.819	33.187
0.008	162.129	1.176	62.406	0.238	35.440	68.160	0.107	248.248	59.471	33.535
0.009	149.294	1.101	66.675	0.266	34.824	60.669	0.116	269.828	59.164	33.842
0.010	138.341	1.036	70.831	0.294	34.036	54.661	0.125	288.614	58.895	34.113
0.015	101.215	0.807	91.250	0.434	29.226	36.560	0.153	354.382	57.963	35.046

With the help of the Optimization Program (results shown in above table) and experimentation it can be conclude that we have been able to optimized Fin height to increases the rate of increase. There is an optimum value of fin height above which further increase in height does not aid the heat transfer process considerably. With the help of the MATLAB R2010 program we have been able to successfully determine this value of optimum fin height for particular input conditions and fin thickness. We also obtained substantial increase in the values of heat transfer coefficients and effectiveness of a heat exchanger when fins were provided. This also demonstrates the fact that Fins provide a thermodynamic advantage. Thereby designing a heat exchanger at the optimum fin height can lead to reducing capital costs and increasing savings. Also providing cheap materials for the fin and expensive durable materials for thinner pipes can increase Heat Exchanger Life span and save capital costs as well. If outlet conditions are provided and fins are also created then by virtue of the fins we can decrease the length of the heat exchanger thus save material.

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