COMPARISON OF OVERALL HEAT TRANSFER COEFFICIENT IN PLAIN AND CORRUGATED PIPE AND IT'S CFD ANALYSIS

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ABSTRACT

Shell and tube heat exchangers are used for convective heat transfer between two fluids, where one fluid flows through the tubes and other flows in the shell. Smooth pipes are usually used in shell and tube heat exchangers. Corrugated pipe provides more heat transfer as it increases the surface area and turbulence of fluid and hence increases the effectiveness of heat exchanger. Experiments have been carried out on plain pipe and corrugated pipe to determine overall heat transfer coefficient. Experimental results of the experiments performed on a smooth pipe and corrugated pipe have been compared using CFD analysis for the same conditions. Numerical analysis on plain and corrugated pipes has been performed using ANSYS FLUENT.

NOMENCLATURE

D	-	Diameter of pipe, m
ρ	-	Density of fluid, Kg/m^3
q	-	Flow rate, m^3/s
v	-	Maximum velocity of object relative to fluid, m/s
L	-	Characteristic linear dimension, m
m	-	Mass flow rate, Kg/s
μ	-	Dynamic viscosity of fluid, Ns/m^2
k	-	Thermal conductivity, W/m-K
C_p	-	Specific heat capacity at constant pressure, J/kg-K
θ_1	-	Temperature difference at inlet, ^oC
θ_2	-	Temperature difference at outlet, ^oC
θ_m	-	LMTD, ^oC
h	-	Convective heat transfer coefficient, W/m^2-K
U	-	Overall heat transfer coefficient, W/m^2-K
G	-	Acceleration due to gravity, m/s^2
β	-	Coefficient of thermal expansion
T_∞	-	Outside water temperature, ^oC
T_b	-	Bulk mean temperature of cold fluid, ^oC
T_f	-	Film temperature at pipe wall, ^oC
ΔT	-	Wall temp (T_w) - Bulk temp (T_b)
U	-	The overall heat transfer coefficient, W/m^{2}K
A	-	Contact area for each fluid side, m^2
Pr	-	Prandtl number
Gr	-	Grashof number
Nu	-	Nusselt number
Re	-	Reynold's number
Ra	-	Rayleigh number
μ_b	-	Viscosity at bulk mean fluid temperature
μ_w	-	Viscosity at wall temperature

LIST OF ABBREVIATIONS

CFD	Computational Fluid Dynamics
AC	Alternating current

CAD	Computer Aided Design
LMTD	Log Mean Temperature Difference
LPM	Litres Per Minute
PVC	Polyvinyl Chloride

KEYWORDS: *Overall heat transfer coefficient; computational fluid dynamics; CFD; corrugated pipe; reynolds number; nusselt's number; prandlt number; LMTD; convergence*

I. INTRODUCTION

Shell and tube heat exchangers are used for convective heat transfer between two fluids, where one fluid flows through the tubes and other flows in the shell. The overall heat transfer coefficient, or U-value, refers to how well heat is conducted over a series of mediums. The overall heat transfer coefficient for a wall or heat exchanger can be calculated as:-

$$\frac{1}{U}=\frac{1}{h_i}+\frac{1}{h_o}$$

A corrugated pipe is a tube with a series of parallel ridges and grooves on its surface. Corrugated pipe provides more heat transfer as it increases the surface area and turbulence of fluid and hence increases the effectiveness of heat exchanger. A heat exchanger is a device which is used to transfer heat between one or more fluids and the fluids may be separated by a solid wall to prevent mixing or they may be in direct contact. They are widely used in space heating, refrigeration, air conditioning, power stations, chemical plants, petrochemical plants, petroleum refineries, natural-gas processing, and sewage treatment. The classic example of a heat exchanger is found in an internal combustion engine in which a circulating fluid known as engine coolant flows through radiator coils and air flows past the coils, which cools the coolant and heats the incoming air. Heat transfer augmentation techniques are broadly classified as passive techniques, active techniques and compound techniques. In passive techniques, any direct input of external power is not required instead; they use it from the system itself which ultimately leads to an increase in fluid pressure drop. They promote higher overall heat transfer coefficient by disturbing or altering the existing flow behavior and these techniques can be achieved by using corrugations. In general, passive augmentation techniques are easier to implement, and do not require significant changes to the setup. They are also much more economical as compared to active techniques. The augmentation technique used in this project is a passive augmentation type technique obtained by using corrugated pipe. The active techniques require external power which is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer by mechanical aids, surface vibration, and electrostatic fields, etc. The compound techniques involve complex design and hence have limited applications. When any two or more active and passive techniques are employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by either of them when used individually, is termed as compound enhancement. The simple design of a shell and tube heat exchanger makes it an ideal cooling solution for a wide variety of applications. One of the big advantages of using a shell and tube heat exchanger is that they are often easy to service and one of the most common applications is the cooling of hydraulic fluid and oil in engines, transmissions and hydraulic power packs. The plain tubes of traditional heat exchangers consisted of boundary layer formation due to which there is reduction in heat transfer rate, whereas the problem can be solved by corrugated pipes as there is induction of turbulence in the flow which results in the thorough mixing of the fluid within the pipe and this in turn increases the heat transfer rate. The present shell and tube heat exchangers occupy large space in order to increase heat transfer rate, this in turns increases the cost and space requirements. The corrugated pipe of equivalent length can increase the heat transfer rate by almost 80-100% and hence this can reduce the overall size of heat exchanger. Heat transfer by conduction is carried out through molecular motion which is quite a slow process in fluids. In solids, this process is much faster. Heat may be transferred by convection, which is the transport of heat by bulk motion of the fluid. Convection may be laminar or turbulent. When the bulk fluid motion is caused by differences in density against the gravity, it is called natural convection. Turbulent convection (or advection) is very effective due to the formation of eddies (localized rotating areas of fluid) that transport energy by local advection. The easiest way to think of it is that turbulence mixes the fluid, so in the case of hot surface and a cold fluid, turbulence in the boundary layer brings cold fluid from the bulk flow near to the wall and takes hot fluid from near the wall and transports it away into the bulk flow where it mixes with cold bulk fluid. Both these aspects of the mixing process promoted heat transfer. Conversely, in laminar flow, the flow is essentially parallel to the surface, so the heat transfer mechanism is just laminar conduction through the boundary layer and into the bulk. Thus, turbulent flow promotes better mixing of the fluid compared to laminar flow. This is what drives better heat or mass transfer. However, increase in turbulence causes a drop in pressure, unless there are external forces applied to the system for the purpose.

II. LITERATURE SURVEY

Giampitro F. 1 (2000) studied heat transfer in a channel composed of a smooth and a corrugated wall under laminar flow conditions. The velocity and temperature distributions were determined with the help of a finite element model. The heat transfer performance of the corrugated wall channel was compared with that of a smooth wall duct. The numerical model was utilized in a genetic algorithm to maximize the heat transfer by optimizing the corrugation profile, for given volume of the corrugated wall and pressure drop in the channel.

Fodemski T. R 2. Gorecki G. 2 and Jasinski P. 2 (2008) validated the Numerical simulations predictions of velocity field in corrugated geometry against the ones recorded by the PIV-method. It contained results for 2 different included angles between corrugations i.e. (i) zero (in this case it is a tube channel) and (ii) 90o. Different wall shapes were analyzed. The Bejan method was used to compare irreversible entropy generation from forced convection with heat transfer − for different corrugated geometry channels.

Azevedo H 3, Morales R. 3, Franco A. 3, Junqueira S. 3 and Erthal R. (2008) examined the influence of groove height and length in friction factor of turbulent flow through corrugated pipes. They used mass and momentum conservation equations. They assumed four geometric configurations involving variations on grooves length and height. A numerical analysis was carried on for Reynolds numbers from 6000 to 50000. It was found that the friction factor increases for a given Reynolds when the grooves length increases, and the friction factor also increases for a given cavity shape when the Reynolds number increases.

Popescu M 4, Johansen S4 and Shyy W 4. (2010) attempted to study the acoustics induced in a flexible corrugated pipe. They defined the phenomenon as “Singing” and large pressure fluctuation was the reason. It was proved that the reason for singing was vortex shedding due to large pressure fluctuations.

Jaiman R.5, Oakley O.5, Adkins5 (2010) presented Computational Fluid Dynamics (CFD) modeling of fully developed turbulent flow through a flexible corrugated pipe and the pressure drop reduction potential of liners was investigated. Significant 3D turbulence effects were found for the pipe geometry with circular corrugations suggested by both qualitative features and quantitative information.

Zachar A. 6 (2010) examined different geometrical parameters of helical corrugation on the outer surface of helically coiled-tube heat exchangers to improve the inside heat transfer rate. Several different inflow rates and temperatures were studied to test the impact of flow parameters for the efficiency of the heat exchanger. It was concluded that the heat transfer rate is almost independent from the inlet temperature and the outer surface temperature for this reason the further studies were conducted with Tin =20oC inlet and Tsurf=60oC surface temperature. It was also proved that the heat transfer rate could be increased by 80%-100% but consequently there was also an increase in pressure drop of about 10%-600%.

Kittur B.7 (2010) analyzed the impulsive motion of an incompressible viscous fluid in a longitudinally corrugated pipe, due to sudden application of the axial pressure gradient was done and the solution of the problem was obtained by using Laplace transform technique and perturbation on small amplitude of the corrugations. It was found that, by suitably corrugating the pipe, the momentum transfer may be augmented during the initial period of the flow development.

Linden B8, Tijsseling A8 (2011) investigated the friction factor in corrugated pipe with rough wall and made use of two models for their computation. They tested both corrugated with smooth and rough wall and proved that there is no significant gain in replacing rough fabric with a smooth one as the analysis was done by usage of (k epsilon) model such that the results obtained were much close to the ones measured.

Nyarko P9 (2012) investigated the effect of heat load on the fluid friction factor for laminar flow of a 2D axisymmetric straight corrugated pipe. He made some assumptions on the Navier-Stokes equations to derive analytic expressions for computing the friction factor for the flow in terms of average velocity, density of the fluid, pressure drop and Reynolds number. In this paper he showed that the heat load reduces the friction factor in corrugated pipes. The Moody Diagram shows a plot of the friction factor and Reynolds number at different corrugation heights without the effect of varying heat load. In this paper we show a diagram of the friction factor and Reynolds numbers at varying inlet and wall temperatures at a constant corrugation height. It can be proved useful for Engineers working in areas of high temperatures. They can use the plots given in this paper and the Darcy Weisbach equation to calculate frictional head loss for a given flow rate of a special fluid through a pipe with known diameter, length and roughness. Finally the increase in heat load on the pipe wall or the fluid changes the transition from laminar to turbulent flow drastically and the friction factor has been shown to decrease monotonically with increasing heat load.

Ahn H 10 and Uslu I 10 (2013) studied characteristics of pressure drop in corrugated pipes were experimentally in both straight and helically coiled configurations. The friction factors of the pipes remain constant over the range of Reynolds number from 4,000 to 50,000. It indicates that the flow in the pipe was fully turbulent. When the pipe was straight configured, the friction factors were measured to be 0.070, 0.075, 0.12 and 0.22 for the diameter of 20.4, 25.4, 34.5 and 40.5 mm, respectively. Thus they concluded that the friction factors increased with the increasing diameter of the pipe. They also found that the friction factors increased in the order: Straight configuration < Helix diameter of 0.64 m < Helix diameter of 0.43 m. Which means the smaller the helix diameter, the larger will be the friction factor.

Shejwalkar V. 11, Nadar M.D. 11 (2014) compared the overall heat transfer coefficient of pipe with internal thread of 4mm & 6mm pitch with plain pipe experimentally. It was proved that the overall heat transfer rate was more in case of pipe having internal threading of 4mm.

Prof. Ingole P.12, Prof. Kolhe S. 12 (2014) proved that the heat transfer rate was enhanced due to spiral corrugations on the wall of the tubes and also the heat transfer rate was increased by 80%-100% but consequently there was an increase in pressure drop of about 10%-600%.

Kareem Z.13, Mohd Jaafar M.N. 13, Lazim T. 13, Abdullah S13, Wahid A13 (2014) used two start spirally corrugated pipe to enhance the heat transfer rate from the fluid and numerical study of two start spirally corrugated pipe showed that there was an enhancement of about 21.68%– 60.54% along with the increase in friction factor which was 19.2–36.4% which was acceptable in comparison with increased heat transfer rate.

Nazri M. N 14, Lazim T. 14, Abdulla S14, Kaeem Z. 14, and Abdulwahd A. 14 (2015) presented the CFD analysis of heat transfer and fluid flow in a laminar flow regime in spirally corrugated tubes with horizontal orientation. They applied constant wall heat flux condition and used water as a working fluid. They examined the spirally corrugated tubes at Reynolds number in the range of 100-1300. They compared these results with standard smooth tube. It was seen that the heat transfer was increased by 18.4-36.

III. CFD

CFD Analysis

The third phase of the results after experimental and theoretical results, involves Computational Fluid Dynamics. The CFD analysis has been done using the software module ANSYS Fluent 15.0. The CFD results involve comparison of obtained temperature contours with experimentally observed temperatures.

Geometric Modeling

The models of pipe and corrugated pipe are built using the Creo Parametric software and imported to ANSYS Design Modeler. All models are to scale and consistent with respect to the manufactures specimens. The geometric models are given below.

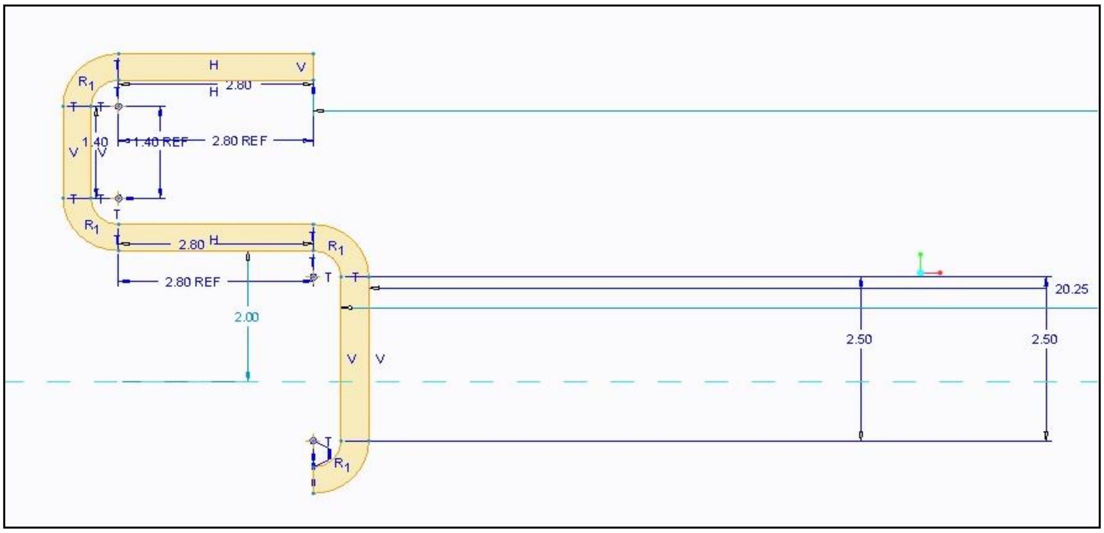


Fig.1 (a) 3D modeling steps of corrugated pipe on CREO

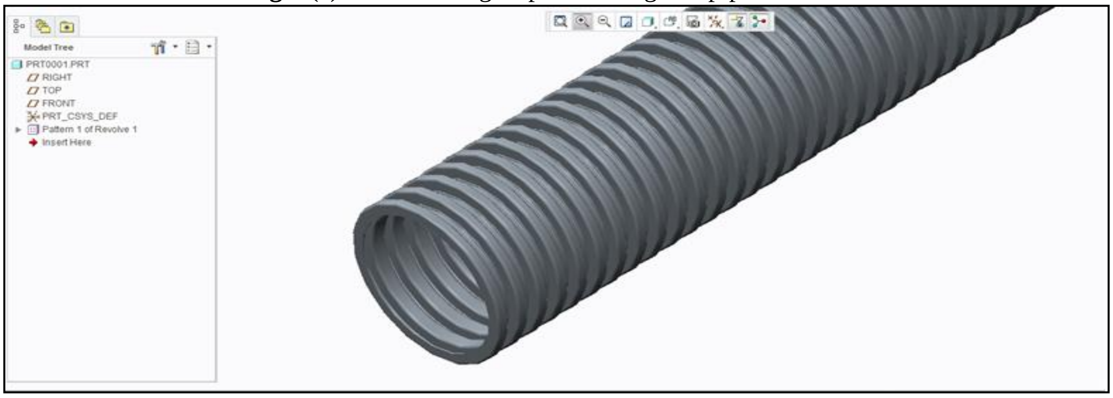


Fig.1 (b) 3D modeling steps of corrugated pipe on CREO

Meshing

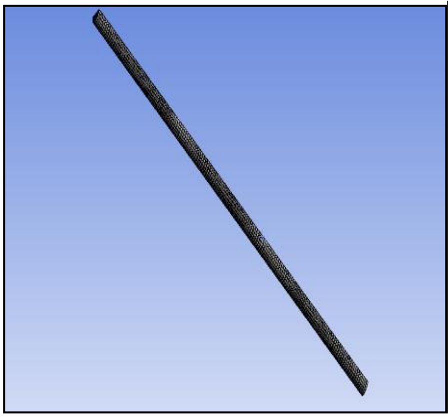
The software package used for meshing and CFD analysis is ANSYS Fluent 15.0. Nature of meshes generated is 'fine mesh'. The mesh details give the following information.

Plain pipe:

Relevance centre: Fine mesh
No. of nodes: 8666
No. of elements: 4989

Corrugated pipe:

Relevance centre: Fine mesh
No. of nodes: 975452
No. of elements: 4486381



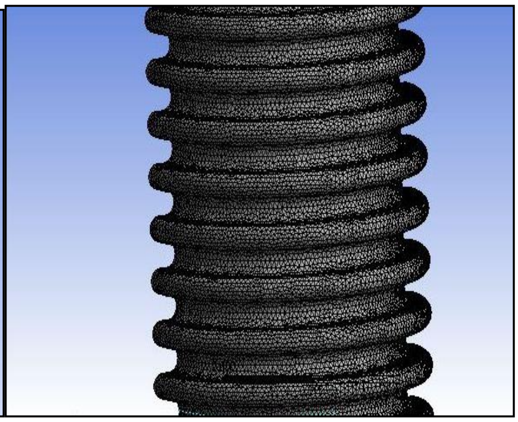
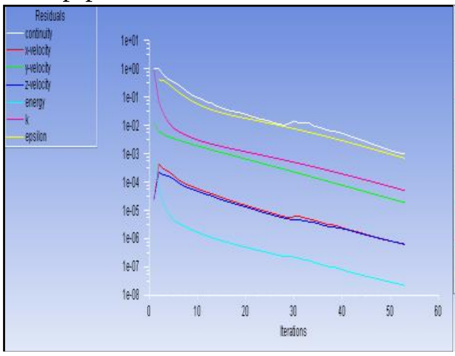


Fig.2 Mesh representation of plain pipe **Fig.3** Mesh representation of corrugated pipe

Named selection
There are only two materials involved in the entire analysis, namely water (fluid) and Stainless Steel (solid). The different surfaces are named as per relevance, the important ones being the pipe inlet and outlet as well as the pipe wall.

Solution
The solution is obtained in 3D mode under double precision, with serial type processing. After the mesh is checked and its quality is obtained, the energy equation is turned ON and viscous model is selected as k-epsilon (2 eqn.) and enhanced wall treatment is selected. The analysis type is set to Temperature type and velocity formulation is absolute and time is steady state. The CFD simulations are run for the median flow rate of 2.74 lpm i.e. a mass flow rate of 0.045355 kg/s, for plain pipe as well as for corrugated pipe. In order to get proper convergence for the simulation, residual is set to 10-3. The solution method is selected as Green-Gauss node based and initialization method is hybrid initialization.

Scaled Residuals
The number of iterations is set to 500 and the calculation is run. The scaled residuals and temperature contours are obtained, and are given below. The temperature contours show the variation in temperature at the inlet, outlet and also along the length of the pipes. The temperature to be observed for CFD analysis is taken as the temperature at the exit of the pipe. This temperature is then compared with the observed temperature from the experimental procedure. There are three different contours for each pipe.



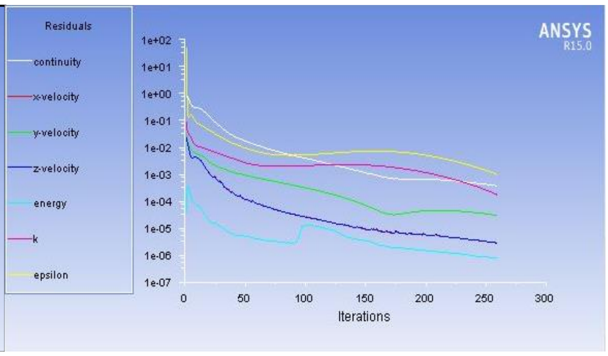


Fig.4 Scaled residues for plain pipe **Fig.5** Scaled residues for corrugated pipe

IV. FABRICATION AND TESTING OF EXPERIMENTAL SETUP

Experimental Setup
The experimental setup consists of corrugated pipe on the left and plain pipe on the right. Thermocouples are placed on the pipe at equal distance throughout the pipe. Inlets to both these pipes are through a pump of 1HP situated at the base. Both these pipes are fitted in a tank which consists of heaters placed at its bottom. These heaters are used to heat the water present in the tank.

Table 1: Specifications of pipes

Sr. No.	Specification	Corrugated Pipe	Plain Pipe
1	Outer diameter	49.60mm	45.80mm
2	Inner diameter	40.50mm	45.00mm
3	Thickness	0.40mm	0.40mm
4	Mean diameter	45.05mm	45.00mm
5	Length	1500mm	1500mm
6	Material	Stainless Steel(304)	Stainless Steel(304)


Fig.6 Experimental Setup **Fig.7** Thermocouples on Pipes

The setup includes:-

- Water tank
- Corrugated pipe and plain pipe
- Heaters placed in the tank
- Eight thermocouples (K-type) placed on each pipe
- Two thermometers placed at inlet and outlet of each pipe
- Manometer
- Flow control valve at outlet of each pipe.
- Pump
- Connecting pipes
- Rotameter

Bill of materials

Table 2: Bill of materials

Sr. No	Name of component	Specifications	Quantity (nos.)
1	Plain pipe	SS, length 1.5m	1
2	Corrugated pipe	SS, length 1.5m	1
3	Fabrication of setup	PVC pipes, metal strap, flexible plastic pipes, reducers	
4	Thermocouples	K type	18

Working of setup and Experimental procedure

All thermocouples have been calibrated between operating temperatures i.e. 00 C to 800C. The calibration was performed using a simple mercury thermometer at the testing laboratory under normal conditions, by checking the temperature for the rise of every 5 degrees.
Both tanks are filled with water and water heaters attached to the hot water tank are switched on.
When hot water reaches desired temperature, the pump is switched on to start the flow of cold water.
Flow rate is adjusted using bypass valve of rotameter to achieve desired flow water.
When steady state is achieved i.e. cold water outlet temperature reaches a constant value,
All temperatures from the thermocouples and pressures from both the pressure gauges are noted. This time required was later observed to be 15-20 minutes.
Flow rate is adjusted again to subsequent flow rates and above procedure is repeated.
The pump was switched off and hot water tank temperature is brought to next desired value and pump was switched on.
After switching on the pump, the experiment is repeated for the same flow rates as before.
Theoretical heat transfer coefficient for all flow rates and desired temperature levels is calculated using empirical relations for Nusselt number at bulk mean temperature of water obtained from experimental readings.

V. EMPIRICAL CORRELATIONS

Typically, for free convection, the average Nusselt number is expressed as a function of the Rayleigh number and the Prandtl number, written as:-

Nu = f (Ra, Pr)

Otherwise, for forced convection, the Nusselt number is generally a function of the Reynolds number and the Prandtl number, written as:-

Nu = f (Re, Pr)

Natural Convection Heat Transfer from a Horizontal Cylinder:-

$$Nu = \left\{ 0.60 + \frac{0.387\ Ra^{\frac{1}{6}}}{\left[1 + \left(\frac{0.559}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}} \right\}^2$$

Where $Gr = \frac{D^3 \rho^2 g\ \Delta T\ \beta}{\mu^2}$

$$Nu = \frac{h\ D}{k}$$

$$Pr = \frac{\mu\ C_p}{k}$$

For Ra < 1012

Where Ra = Gr x Pr

Correlations for Laminar Flow inside a Circular Pipe:-

Thermal or Thermal/Hydrodynamic Entrance Region,

Constant Wall Temperature, Pr $\geq$ 5

Laminar Pipe Flow, Re < 2300

$$Nu_o = 3.66 + \frac{0.0668\ Re\ Pr(\frac{D}{L})}{1 + 0.04\ [\ Re\mathrm{Pr}\ (\frac{D}{L})]^{\frac{2}{3}}}$$

Thermal and Hydrodynamic Entrance Region

Constant Wall Temperature – Laminar Flow

(Re < 2300)0.6 < Pr < 5 $\quad 0.0044 < \frac{\mu_b}{\mu_w} < 9.75$

$$Nu_o = 1.86 \left(\frac{Re\ Pr}{L/D}\right)^{\frac{1}{3}} \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

Correlations for Turbulent Flow inside a Circular Pipe:-

The Dittus- Boelter equation:-

Nuo = 0.023 Re0.8 Pr0.4, for 'heating' (temperature of wall > temperature of fluid), and

Subject to: 0.7 < Pr < 120; 10,000 < Re < 160,000; L/D > 50

A second relation is given as,

$$Nu_o = \frac{\left(\frac{f}{8}\right)(Re - 1000)Pr}{1 + 12.7\left(\frac{f}{8}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)}$$

Where $f = (0.790 \ln Re - 1.64)_{-2}$

For: 0.5 < Pr < 2000 and 3000 < Re < 5 x 106

Initial Data and Formulae for Experimental Calculations

Diameter of pipe, D = 45 mm = 0.045m

Length of pipe, L = 1500 mm = 1.5 m

Contact area between fluids, $A = \pi DL$ = 0.2121 m2

Mass flow rate, $m = (\rho\ x\ q)$ kg/s

Temperature difference at inlet, θ1 = Th – Ti

Temperature difference at outlet, θ2 = Th – To

LMTD, θm = $\frac{(\theta 1 - \theta 2)}{ln(\theta 1/\theta 2)}$

Q= m x Cp x (To-Ti) = U x A x θmU= $\frac{Q}{A * \theta m}$

Sample experimental Calculations

The sample calculation has been taken for reading no 1 i.e. plain pipe under a flow rate of 2.74 lpm. This procedure applies to all flow rates as well as to conditions involving corrugated pipe. Assumptions made in all calculations are as follows:

Effect of fouling elements is neglected.

Fluid temperature is assumed to be constant throughout the vertical axis, so that it has uniform viscosity i.e. µb (bulk viscosity) = µw (wall viscosity)

Mass flow rate:

$m = (\rho\ x\ q)$ = 994.03 x 4.56 x 10-5 = 0.045355 kg/s.

Hot water temperature:

Th = 65^{o}C

Temperature difference:

At inlet, θ1 = (Th – Ti) = 65 – 30 = 35^{o}C

At outlet, θ2 = (Th – To) = 65 – 37 = 28^{o}C

LMTD

θm = $\frac{(\theta 1 - \theta 2)}{ln(\theta 1/\theta 2)}$ = (35 - 28) / ln (35 / 28)

= 31.37oC

Heat transfer rate:

Q = m x Cp x (To-Ti) = 0.045355 x 4177 x (37-30) = 1320 W

Overall heat transfer coefficient:

Q = U x A x θm

U = 1320 / (0.2121 x 31.37) = 198 W/ m2K

There are no changes to the above procedure for any other readings. All quantities are in MKS system. Volumetric flow rate needs to be converted from lpm to kg/s before initiating calculation of mass flow rate.

Observations for Plain Pipe

The first set of observations is for plain pipe and it was observed that pressure drop in plain pipe is very small and almost negligible in some cases. The highest observed drop in pressure was 0.133 kPa whereas the highest observed rise in temperature of cold fluid was 7^{o}C.

Table 3: Observations for plain pipe

Hot Water Temperature(^{o}C)	65		60		55	
Velocity(m/s)	Surface	Outlet	Surface	Outlet	Surface	Outlet
0.028689	56.58	37.00	52.52	35.50	48.61	35.00
0.040565	55.98	35.5	52.08	35.00	48.14	33.80
0.049968	54.87	34.00	51.13	33.80	47.42	33.50
0.061342	53.87	33.80	50.25	33.60	46.70	33.10
0.071858	53.03	33.50	49.54	33.40	46.02	32.80
0.083835	52.18	33.20	48.80	33.10	45.38	32.50

Observations for Corrugated pipe.

After the readings for plain pipe were taken, the pump was switched off, and the inlet was connected to corrugated pipe and pump was switched on. It was noted that the pressure drop in corrugated pipe was found to be almost four times than that in plain pipe for higher flow rates. Highest pressure drop obtained was 0.533 kPa and the highest observed rise in temperature of cold fluid was 14°C.

Table 4: Observations for corrugated pipe

Hot Water Temperature(°C)	65		60		55	
Velocity(m/s)	Surface	Outlet	Surface	Outlet	Surface	Outlet
0.028689	56.58	44.00	52.52	42.00	48.61	40.00
0.040565	55.98	42.00	52.08	40.00	48.14	38.00
0.049968	54.87	40.50	51.13	39.00	47.42	37.00
0.061342	53.87	39.00	50.25	38.00	46.70	36.00
0.071858	53.03	38.40	49.54	37.00	46.02	35.50
0.083835	52.18	38.00	48.80	36.50	45.38	35.00

Overall heat transfer coefficient

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o}$$

Forced Convection Calculations

Bulk mean temperature

Tb = $\frac{(Ti + To)}{2}$ = (30 + 37)/ 2 = 33.5°C

Prandtl number (Pr), dynamic viscosity(μ), density(ρ), specific heat(Cp), thermal conductivity(k), fluid thermal expansion coefficient(β) are obtained from thermo physical properties table.

Velocity of flow

V = q/A = (4.56 x 10-5)/ (.00159) = 0.028689 m/s

Reynolds number

Re = $\frac{\rho v D}{\mu}$ = (994.03 x 0.028689 x 0.045) / (0.0007197)

= 1783.1 (Re < 2300. Hence, flow is laminar)

Nusselt number

$$Nu_o = 3.66 + \frac{0.0668\ Re\ Pr (\frac{D}{L})}{1 + 0.04\,[\,Re \Pr\ (\frac{D}{L})]^{\frac{2}{3}}}$$

= 20.89

Theoretical heat transfer coefficient

hi = Nu x (k/D) = 20.89 x 0.6233 / 0.045

= 289.405 W/m2k

Natural Convection Calculations

Film temperature, Tf = (T∞ + Tw) / 2 = (65 +56.58) / 2

= 60.79°C

Grashof number, $Gr = \frac{D^3 \rho^2 g\ \Delta T\ \beta}{\mu^2}$ = 1.749 x 107

Rayleigh number, Ra = Gr x Pr = 5.218 x 107

Nusselt-number, $$Nu = \left\{0.60 + \frac{0.387\ Ra^{\frac{1}{6}}}{\left[1 + \left(\frac{0.559}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}2$$

= 54.547

Theoretical heat transfer coefficient, ho = Nu x (k / D) = 54.547 x (0.6544/ 0.045)
= 793.24 W/m2k

Overall Heat transfer coefficient Calculations

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o}$$

$$= \frac{1}{289.405} + \frac{1}{793.24}$$

Therefore, U = 212.04 W/m2k

VI. Results and Analysis

Experimental Results

The apparatus was operated successfully, and experimental readings were obtained for six different flow rates. Heat transfer rate was calculated using the classical formula, and was equated with the LMTD formula to find the overall heat transfer coefficient. The increase in overall heat transfer coefficient due to corrugation was also calculated for each flow rate. The gauge pressures were measured, and the drop in pressure was calculated for each flow rate using the manometers.
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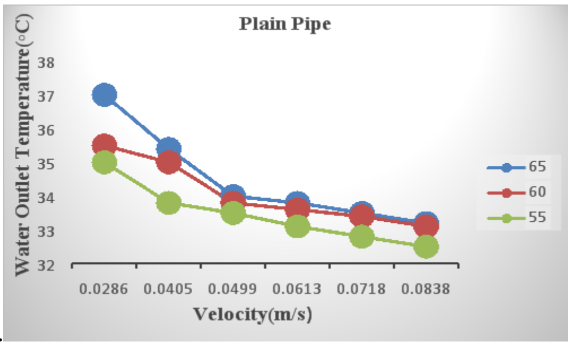

Fig.8 Outlet temperature variation of plain pipe

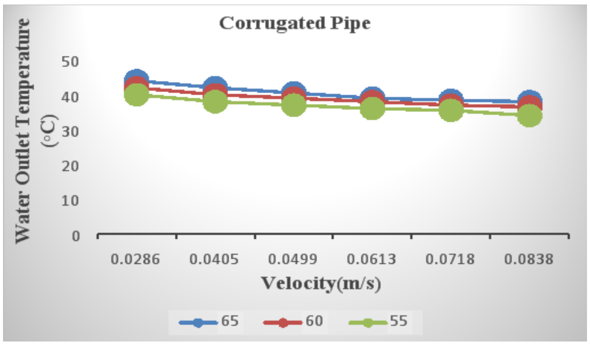

Fig.9 Outlet temperature variation of corrugated pipe

Above figures are the plots for outlet temperature variation of plain pipe and corrugated pipe. The graphs are plotted for three different hot fluid temperatures viz. 65°C, 60°C, 55°C. For plain pipe, there is more variation in outlet temperature at lower flow rates but as the flow rate increases the outlet temperature almost becomes constant. For corrugated pipes, outlet temperature linearly decreases for increasing flow rates. This is due to the extent of the “thermal entrance region” in which the transverse temperature distribution becomes “fully developed.” This region is relatively short in turbulent flow (precisely because of the intense turbulent transverse transport of energy), whereas it tends to be long in laminar flow which generally occur in plain pipes. In the case of laminar flow, the heat exchanger is designed to take advantage of relatively high heat transfer rates that are achievable in the thermal entrance region. In the case of turbulent flow, the thermal entrance region is short and typically heat transfer occurs mostly in the “fully developed” region. Therefore, for lower flow rates, plain pipes have higher heat transfer and there is decrease in heat transfer rate for higher flow rates. In corrugated pipes, there is slight decrease in heat transfer rate for increased flow rates but for a given flow rate and surrounding hot fluid temperature, inlet cold fluid has a greater heat transfer gain along the length of the corrugated pipe than plain pipe . .

Table 5: Overall heat transfer coefficient at 65ºC outside temperature for plain pipe

Velocity (m/s)	Reynolds number	Utheoretical (W/m2k)	Uexperimental (W/m2k)	Deviation (%)
0.028689	1783	212	198	06.60
0.040565	2521	226	215	04.87
0.049968	3106	257	189	26.56
0.061342	3813	290	219	24.48
0.071858	4466	318	235	26.10
0.083835	5211	346	250	27.74

Table 6: Overall heat transfer coefficient at 60ºC outside temperature for plain pipe

Velocity (m/s)	Reynolds number	Utheoretical (W/m2k)	Uexperimental (W/m2k)	Deviation (%)
0.028689	1783	210	205	02.38
0.040565	2521	224	252	-11.11
0.049968	3106	254	232	08.66
0.061342	3813	286	264	07.69
0.071858	4466	313	287	08.31
0.083835	5211	340	312	08.23

Table 7: Overall heat transfer coefficient at 55oC outside temperature for plain pipe

Velocity (m/s)	Reynolds number	Utheoretical (W/m2k)	Uexperimental (W/m2k)	Deviation (%)
0.028689	1783	207	178	14.01
0.040565	2521	221	185	16.29
0.049968	3106	250	205	18.00
0.061342	3813	281	219	22.06
0.071858	4466	307	230	25.08
0.083835	5211	334	237	29.04

Table 8: Overall heat transfer coefficient

Velocity, v (m/s)	Condition	Overall heat transfer coefficient, U (W/m2K) At 65ºC	Overall heat transfer coefficient, U (W/m2K) At 60ºC	Overall heat transfer coefficient, U (W/m2K) At 55ºC	Increase in Overall heat transfer coefficient (%)
0.028689	Plain	198	205	178	53.10
	Corrugated	447	447	357	
0.040565	Plain	215	252	185	53.74
	Corrugated	522	505	389	
0.049968	Plain	189	232	205	57.74
	Corrugated	549	549	410	
0.061342	Plain	219	264	219	54.85
	Corrugated	564	588	424	
0.071858	Plain	235	287	230	54.00
	Corrugated	610	591	452	
0.083835	Plain	250	312	237	54.59
	Corrugated	674	634	475	

The overall heat transfer coefficient is influenced by various parameters such as flow rate and hot water temperature and there is an average increase in overall heat transfer coefficient of 54.67 %, for taken flow rates. We can see that the percentage increase in overall heat transfer coefficient is almost independent of flow rate as it remains in the range of 50-60%.

Table 9: Pressure drop

Velocity (m/s)	Plain pipe (mm of Hg)	Corrugated pipe (mm of Hg)
0.028689	0	1
0.040565	1	1
0.049968	1	1
0.061342	1	2
0.071858	1	2
0.083835	1	2
0.086725	1	3
0.094314	1	3
0.103358	1	3
0.110957	1	4
0.114320	1	4

Darcy friction factor, f = $\frac{\Delta p .2gd}{4Lv^2}$

Table 10: Darcy friction factor

Velocity (m/s)	Δp(mm of Hg)	f
0.028689	1	2.430
0.083835	2	0.569
0.103358	3	0.562
0.114320	4	0.612

VII. CFD RESULTS

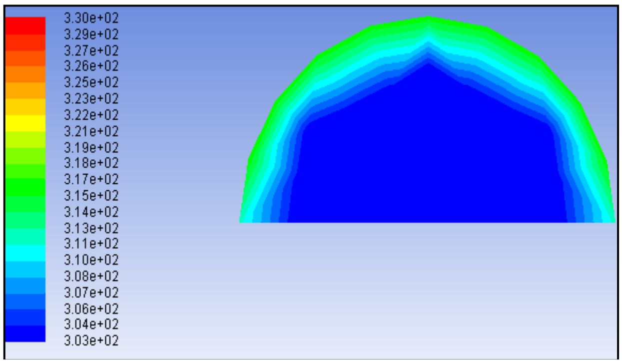

Fig.10 Temp contour at inlet of plain pipe

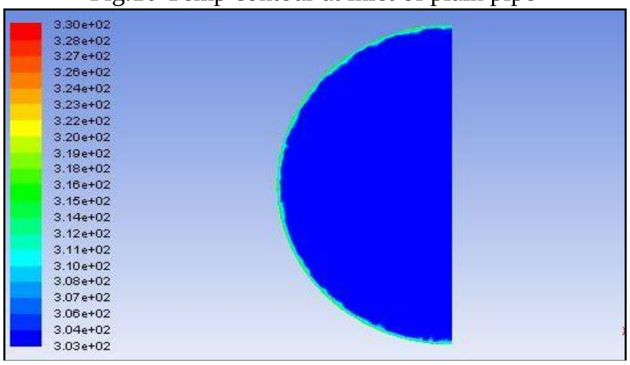

Fig.11 Temp contour at inlet of corrugated pipe

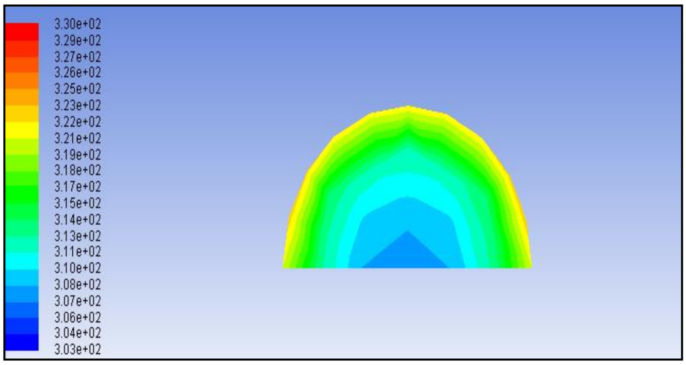

Fig. 12 Temp contour at outlet of plain pipe

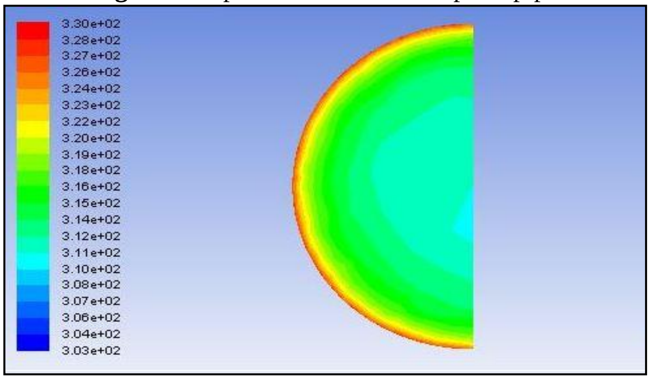

Fig.13 Temp contour at outlet of corrugated pipe

Above figure shows the contours of temperature at the outlet of plain and corrugated pipe respectively. The increment of temperature, from innermost layer to outermost layer, in corrugated pipe can be seen to be at higher temperature range than that in the plain pipe.

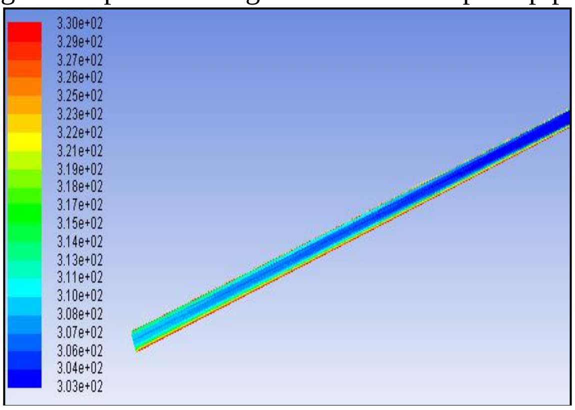

Fig. 14 Symmetrical temp contour plain

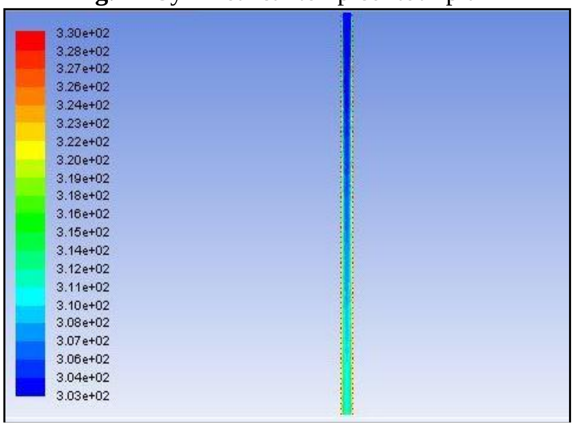

Fig. 15 Symmetrical temp contour corrugated pipe

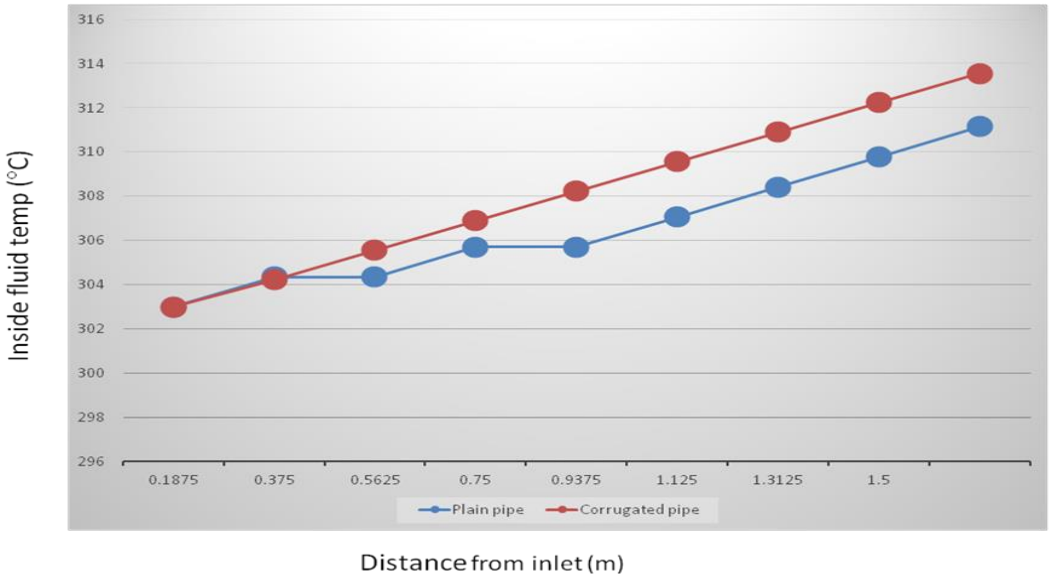

Fig. 16 Temperature along the length of the pipes

In above figure, temperature plot of plain and corrugated pipe, the slope shows how the water gets heated as it moves towards the outlet of the pipe. For corrugated pipe, due to turbulence and increased surface area, there is augmentation of heat transfer due to which the temperature at the outlet of corrugated pipe is more than that of plain pipe. The length of the plain and corrugated pipe is divided into eight parts and using probe along the symmetrical section, the temperature was determined. The inside fluid temperature in corrugated pipe varies linearly along the length of the pipe. In plain pipe it is uneven at the start due to formation of boundary layer and varies linearly along the length of the pipe when the flow is fully developed.

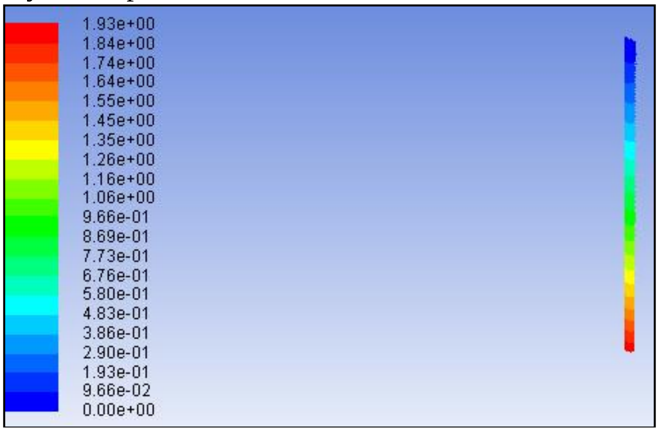

Fig.17 Symmetrical pressure contour of plain pipe

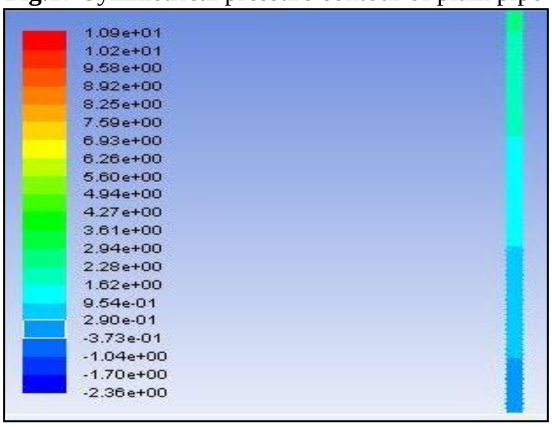

Fig.18 Symmetrical pressure contour of corrugated pipe

VIII. CONCLUSION

The temperature contours from the CFD analysis conducted on ANSYS Fluent 15.0, when compared with experimental results, were found to be within a reasonable margin of average error. The error may be attributed to scaling, the human error in measurement of flow rate and instrumental error of rotameter, manometer and thermocouples. The overall heat transfer coefficient is influenced by various parameters such as flow rate and hot water temperature. It is observed that there is an average increase in overall heat transfer coefficient of 54.67%. We can see that percentage increase in overall heat transfer coefficient almost becomes independent of flow rate, as it remains in the range of 50-60%. From the graphs corresponding to overall heat transfer coefficient vs. velocity at the hot fluid temperatures of 65oC, 60oC and 55oC it is noticed that the overall heat transfer coefficient for plain pipe increases slightly with increase in flow velocity of water whereas overall heat transfer coefficient for corrugated pipe increases significantly with increase in flow velocity of water. The inside fluid temperature for corrugated pipe varies linearly along the length of the pipe whereas for plain pipe it is uneven at the start due to formation of boundary layer and then varies linearly when the flow is fully developed. The equivalent size of corrugated pipe for same flow rate of 0.0969 kg/s is 0.407 m for length of plain pipe as 1.5 m to achieve same outlet temperature. Hence there is a decrease of almost 70% in length of corrugated pipe. Due to increase in pressure drop in corrugated pipe, pumping power required is more than plain pipe for same flow rate. Since pressure drop for corrugated pipe is twice than that of plain pipe for mass flow rate of 0.0969 kg/s, hence pumping power required is double than that of plain pipe. Hence, there is an increase in 50% of pumping power for corrugated pipe.

Future Scope

Corrugated pipe heat transfer is a very promising area of research and experimentation and more advanced modifications, such as spiral and different types of corrugations could be produced to achieve much higher levels of heat transfer. The pitch can be varied to see the variation in results. By altering certain parameters such as corrugation size, heating capacity and pumping power, this setup can also be used for industrial applications. The working fluid can be taken as non-Newtonian fluids and Nano fluids. The properties of these are different from that of water and hence may exhibit different results for heat transfer. The friction factor can be reduced by changing design of corrugation. By introducing liner materials the friction factor can be reduced by 80%.

IX. FUTURE SCOPE

Corrugated pipe heat transfer is a very promising area of research and experimentation and more advanced modifications, such as spiral and different types of corrugations could be produced to achieve much higher levels of heat transfer. The pitch can be varied to see the variation in results. By altering certain parameters such as corrugation size, heating capacity and pumping power, this setup can also be used for industrial applications. The working fluid can be taken as non-Newtonian fluids and Nano fluids. The properties of these are different from that of water and hence may exhibit different results for heat transfer. The friction factor can be reduced by changing design of corrugation. By introducing liner materials the friction factor can be reduced by 80%.

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