Enhancing the Heat Transfer Rate in Conventional Parabolic through Type Solar Collector by Implementation of Helically Extruded-Cut Type Finned Pipe in CFD

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***Abstract* - The Absorber pipe is the heart of the heat transfer mechanism involved in a parabolic trough collector (PTC). The paper envisages the implementation of helically extruded-cut fin in the absorber pipe of a solar parabolic trough type collector and its influence in increasing the heat transfer coefficient, there by comparing each of the respective properties with the conventional plane pipe and solid finned pipe. A conventional plane pipe model has been used as a base for comparison of the two types of finned pipes and the various heat transfer properties that vary has been systematically presented using CFD. The thermal resistance property has been used as the standard of calculation. The HT domain of a PTC is designed with the fins and their contrasting heat transfer rate, hydrodynamic properties, thermal properties has been found and has been reported.**

Keywords: Heat transfer, Parabolic Trough Collector, Finned Pipe, Thermal Resistance, Helical Fins, Extruded-Cut Fins.

I. INTRODUCTION

The sun's total energy output is 3.8×10^{20} MW which is equal to 63 MW/m2 of the sun's surface. This energy radiates outwards in all directions. Only a tiny fraction, 1.7×10^{14} kW, of the total radiation emitted is intercepted by the earth. Solar energy collectors are special kinds of heat exchangers that transform solar radiation energy to internal energy of the transport medium. The major component of any solar thermal power system using solar radiation is the solar thermal collector. This is a device that absorbs the incoming solar radiation, converts it into heat, and transfers the heat to a fluid (usually air, water, or oil) flowing through the collector. The solar parabolic trough type collector usually has concave reflecting surfaces to intercept and focus the sun's beam radiation to a smaller receiving area, thereby increasing the radiation flux. Concentrating collectors are suitable for high-temperature applications. The main challenge in designing these systems is to select the correct operating temperature upto the super saturated temperature of (150ºC) above which heat losses from receiver pipe occurs. This is because the efficiency of the heat engine rises as its operating temperature rises, whereas the efficiency of the solar collector reduces as its operating temperature rises. Concentrating solar collectors are used exclusively for higher operating temperature of the heat transfer fluid required. Because the maximum operating temperature for flat-plate collectors is low relative to the desirable input temperature for heat engines, and therefore system efficiencies would be very low [1]

Solar thermal power plants produce electricity by converting the solar radiation into high temperature heat using mirrors and reflectors. The collectors are referred to as the solar-field. This energy is used to heat a working fluid and produce steam. Steam is then used to rotate a turbine or power an engine to drive a generator and produce electricity all solar thermal power plants are based on four basic essential systems which are collector, receiver (absorber), transport, storage and power conversion. Parabolic Trough, Solar towers, Parabolic Dishes and Linear Fresnel Reflectors are the four main technologies that are commercially available today. The parabolic shaped dish tracks the sun, through a two axis movement, onto a thermal receiver mounted at the focal point. The concentrated beam radiation is absorbed into a receiver to heat a fluid or gas to approximately 750°C. This fluid or gas is then used to generate electricity. The estimated output from the solar power plant depends on the design parameters [2]

The parabolic-trough solar concentrators are one of the basic elements of a concentrating solar power plant. This method of concentrated solar collection has the advantage of high efficiency and low cost, and can be used either for thermal energy collection, for generating electricity. Through solar trough collector temperature increase up to 100°C to 400°C or above 400°C [3]. A lot of remarkable simulations and experiments on the efficiency of heat collection of parabolic trough collector have been studied since 1980s. In order to reduce the number of receiver elements needed in the lines, the heat transfer rate from external heat flux to the inner absorber tube should be enhanced. The effect of inserting metal foams in absorber tube on thermal-hydraulic performance increases the Nusselt number about 5-10 times with the increase of friction factor 10-20 times. The thermal losses of the enhanced tubes can be reduced by 2.23%-13.62% compared to that of the smooth tubes [4].

Convection is a process which involves mass movement of fluids. On getting heated, the fluid near the wall moves up due to the effect of buoyancy and this hot fluid is replaced by cold fluid moving towards the wall. There will always be a boundary layer adjacent to the wall, either in the natural convection or forced convection, where the temperature and velocity vary from plate to the free stream. Initially the velocity increase with increasing distance from the surface and reaches a maximum and then decrease to approach zero value. This is because of action of viscosity diminishes rapidly with distance from inner wall of the pipe, while density difference

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decreases more slowly. The thermal boundary layer is considered as the stationary fluid film which is responsible for heat conduction and then heat is transported by fluid motion. The used of heat transfer enhancement has become widespread during the last so many years. The need of heat transfer enhancement is to reduce the size and cost of the equipment. This goal can be achieving in two ways active and passive enhancement. The active enhancement is less common because it requires addition of external power (e.g., an electromagnetic field) to cause a desired flow modification. In the passive enhancement, it consists of alteration to the heat transfer surface or incorporation of a device whose presence results in a flow field modification. The most popular enhancement is the fin. Fins are the extended surfaces which are used to enhance the rate of heat transfer dissipation from heated surfaces to working fluid. The application of finned tubes to the design of parabolic trough collectors has some losses as the pressure losses, thermal losses and thermo-mechanical stress and thermal fatigue. The result shows an improvement potential in parabolic trough solar plants efficiency by the application of internal finned tubes [5].

II. THE PROCESS FLOWCHART:

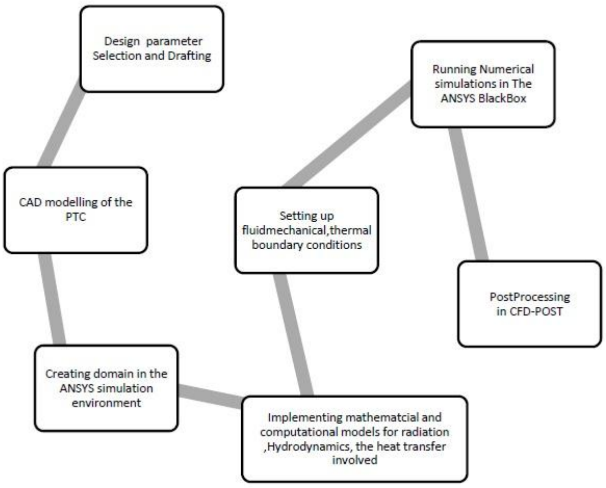


Fig I

III. DESIGN MODELLING:

The design parameters are calculated from geometric relation of parabolic collector, receiver pipe and internal fin as follows (Kalogirou, 2009; CFD Analysis of a Tube with Different Internal Fin Profile, I. Satyanarayana, 2015, Design support system for PTC, D.E Woldemichael).

A. Specifications of the Parabolic Collector:

Assumptions:

Focal Length (f) = 0.55 m
Rim Angle (Φ) = 90^0
Aperture Length (L_a) = 2.525 m

Aperture Width:

$$W_a = 4f \tan \Phi/2$$
$$= 2200 \text{ mm}$$

Aperture Area:

$$A_a = W_a \times L_a$$
$$= 5550 \text{ mm}^2$$

Fig III.1

A. Specifications of the receiver pipe:

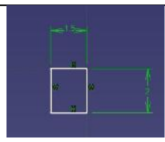
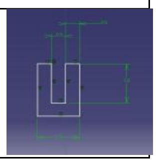
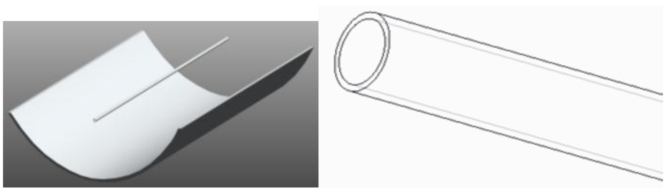
	Plane Pipe	Internally Solid Finned Pipe	Extruded Cut Finned Pipe
Inner Diameter (di)	25 mm	25 mm	25 mm
Outer Diameter (d_0)	30 mm	30 mm	30 mm
Pipe Thickness (t)	2.5 mm	2.5 mm	2.5 mm
Pipe Length (L)	2565 mm	2565 mm	2565 mm
Wetted Perimeter	39.25mm	43.25mm	43.25mm
Outer Surface Area (A_o)	0.241×10^{-6}	0.241×10^{-6}	0.254×10^{-6}
Inner Surface Area (A_i)	0.201×10^{-6}	0.261×10^{-6}	0.261×10^{-6}
Fin Profile and Path	-	Helically, internally wounded Rectangle	Helically Extruded-cut Rectangle
Fin Width (W_f)	-	2 mm	2 mm
Fin Thickness (t_f)	-	1.5 mm	1.5 mm
Fin Length (Lf)	-	3683.22 mm	3683.22 mm
Fin Area (A_f)	-	5524.83 mm^2	16573 mm^2
Fin Volume (V)	-	11049 mm^3	8286.75 mm^3
Pitch (p)	-	50 mm	50 mm
Number of Turns (n)	-	51	51
Cross Section			

TABLE I

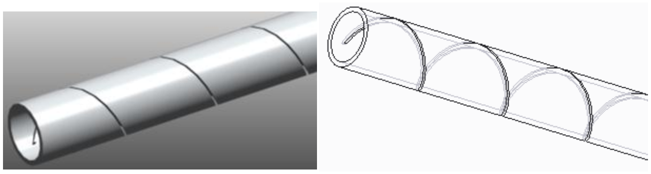
B. Design Rendering:

A. Plane Pipe:



B. Internally Solid Finned Pipe:

C. Extruded Cut Finned Pipe:



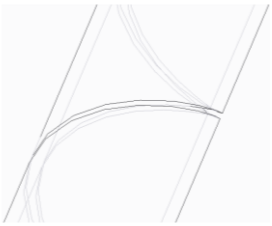


Fig III.2 The CAD rendering and the design Drafts.

B. PRE PROCESSING OF THE DOMAIN:

A. Input Parameters:

	Plane Pipe	Internally Solid Finned Pipe	Extruded Cut Finned Pipe
Inlet Temperature (K)	300	300	300
Radiation Intensity over Reflector (W/m^2)	600	600	600
Reflector Emissivity (ε)	0.9	0.9	0.9
Velocity Magnitude (m/s)	0.01	0.01	0.01
Heat transfer fluid	Water	Water	Water
Pipe Material	Aluminium	Aluminium	Aluminium
Reflector Material	Aluminized Mirror	Aluminized Mirror	Aluminized Mirror
Thermal Conductivity (W/mK)	200	200	200
Heat Transfer Rate of Air h_{air}, (W/m^{2}K)	50	50	50
Viscous Model	K-epsilon	K-epsilon	K-epsilon

Table II

The Radiation conditions were assumed to be same as that occurs at the Black Rock Desert Mojave, CA.

B. Meshing of CFD Domain:

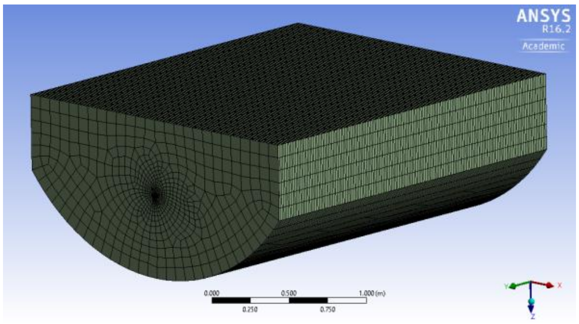


Fig IV.1 The PTC Domain

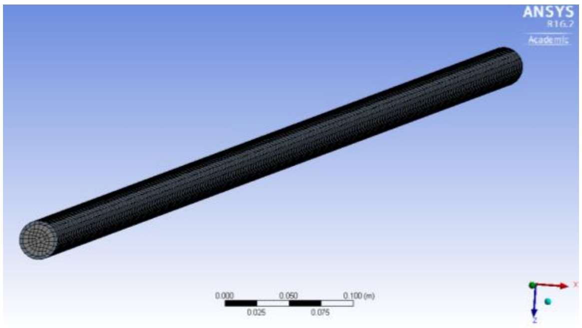


Fig IV.2 Concentric Pipe Domain

V. GOVERNING EQUATIONS

There are Three Equations Used for Flow Analysis they are:

1. Continuity Equation:

$$\frac{\delta\rho}{\delta t}+\frac{\delta\rho u}{\delta x}+\frac{\delta\rho v}{\delta y}+\frac{\delta\rho w}{\delta z}=0$$

2. Momentum Equation:

$$\frac{\partial(\rho u)}{\partial t}+\nabla\cdot(\rho u\mathbf{u})=\frac{\partial(-p+\tau_{xx})}{\partial x}+\frac{\partial\tau_{yx}}{\partial y}+\frac{\partial\tau_{zx}}{\partial z}+S_{Mx}$$
$$\frac{\partial(\rho v)}{\partial t}+\nabla\cdot(\rho v\mathbf{u})=\frac{\partial\tau_{xy}}{\partial x}+\frac{\partial(-p+\tau_{yy})}{\partial y}+\frac{\partial\tau_{zy}}{\partial z}+S_{My}$$
$$\frac{\partial(\rho w)}{\partial t}+\nabla\cdot(\rho w\mathbf{u})=\frac{\partial\tau_{xz}}{\partial x}+\frac{\partial\tau_{yz}}{\partial y}+\frac{\partial(-p+\tau_{zz})}{\partial z}+S_{Mz}$$

3. Energy Equation:

$$\frac{\partial(\rho E)}{\partial t}+\nabla\cdot(\rho E\mathbf{u})=-\nabla\cdot(p\mathbf{u})+\left[\frac{\partial(u\tau_{xx})}{\partial x}+\frac{\partial(u\tau_{yx})}{\partial y}+\frac{\partial(u\tau_{zx})}{\partial z}+\frac{\partial(v\tau_{xy})}{\partial x}\right.$$
$$\left.+\frac{\partial(v\tau_{yy})}{\partial y}+\frac{\partial(v\tau_{yz})}{\partial z}+\frac{\partial(w\tau_{xz})}{\partial x}+\frac{\partial(w\tau_{yz})}{\partial y}+\frac{\partial(w\tau_{zz})}{\partial z}\right]$$
$$+\nabla\cdot(\lambda\nabla T)+S_E$$

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VI. ANALYSIS AND RESULTS.

A. Plane Pipe:

The results of the analysis of the collector tubes of Plane type:

A.1. Post processed results of the domain:

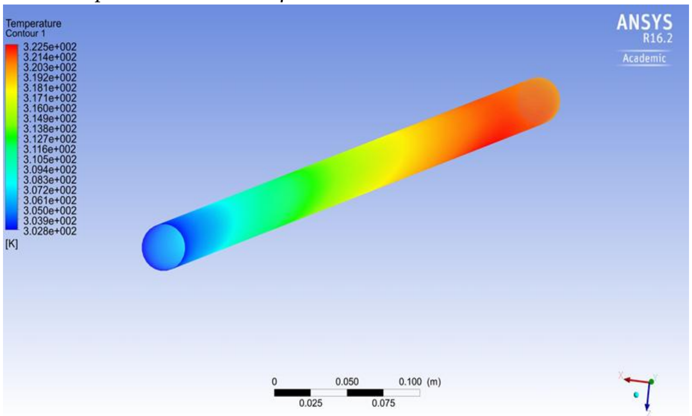

Fig VI.1 Temperature Contour

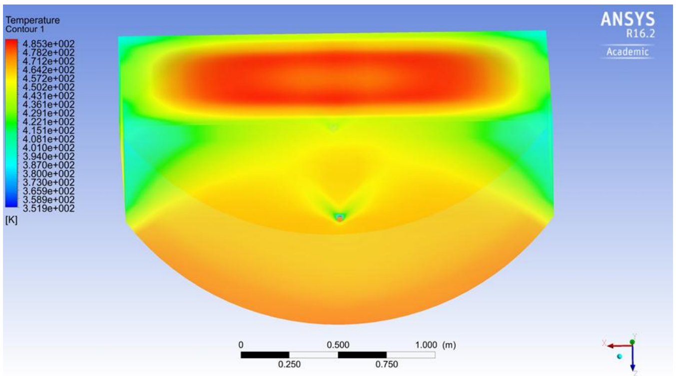

Fig VI. 2. Temperature contour of the PTC domain

B. Internally Solid Finned Pipe:

The results of the analysis of the collector tubes of Internally solid finned type:

B.1 Post processed results of the domain:

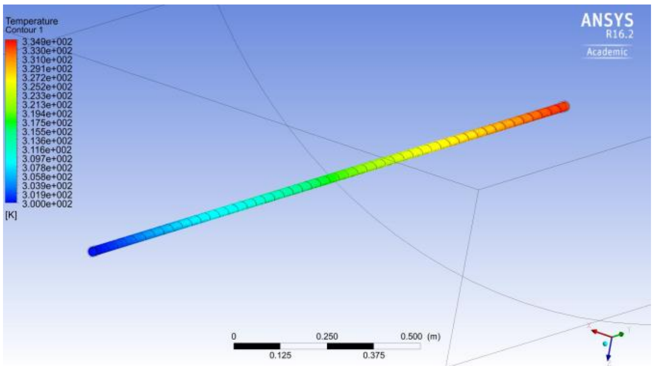

Fig VI.3 Temperature Contour

C. Helically Extruded-cut finned Pipe:

The results of the analysis of the collector tubes of extruded-cut type:

C.1. Post processed results of the domain:

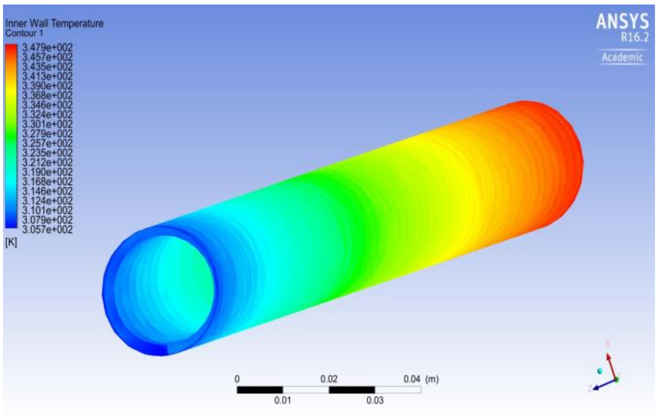

Fig VI.4 Temperature Contour

Wall heat transfer coefficient contour:

	Outlet Temperature (K)
Plane Pipe	321
Internally Solid Finned Pipe	337
Extruded Cut Finned Pipe	346

Table III

D. Corresponding Plots

D.1. Plane Pipe:

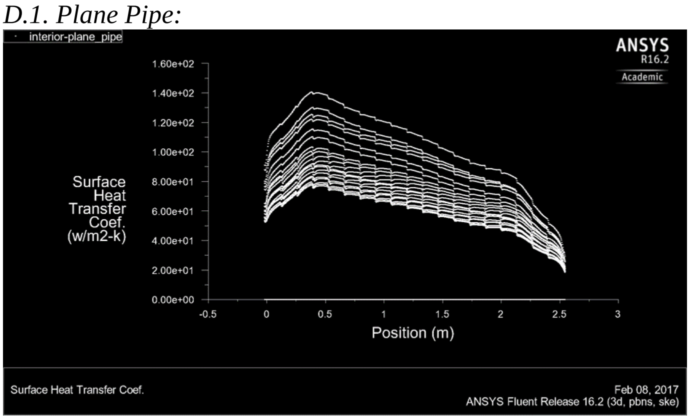

Fig VI.5 Surface heat transfer Coefficient

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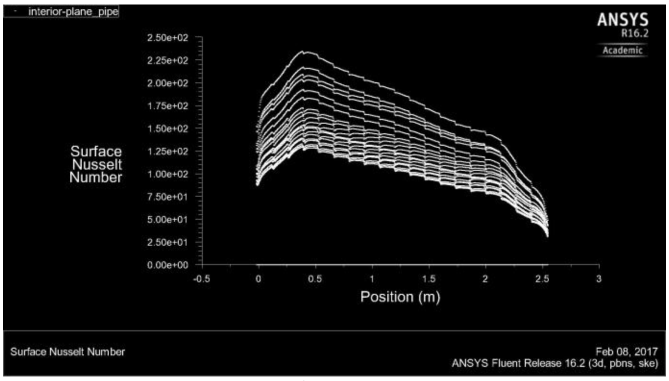

Fig VI.6 Surface Nusselt Number

D.2. Internally Solid Finned Pipe:

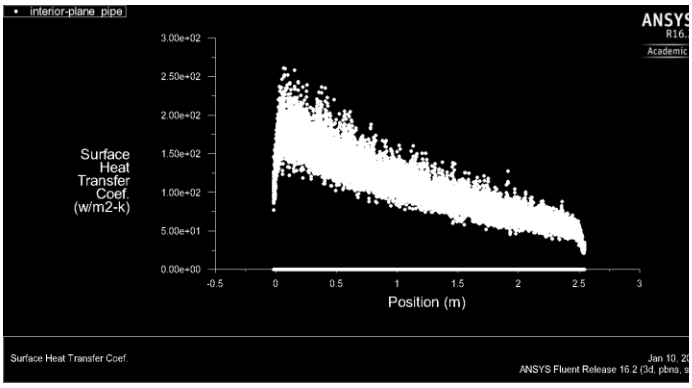

Fig VI.7 Surface Heat Transfer Coefficient

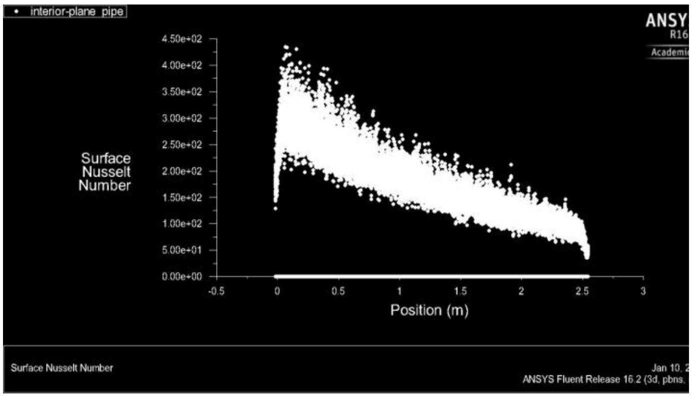

Fig VI.8 Surface Nusselt Number

D.3. Helically Extruded-cut finned Pipe:

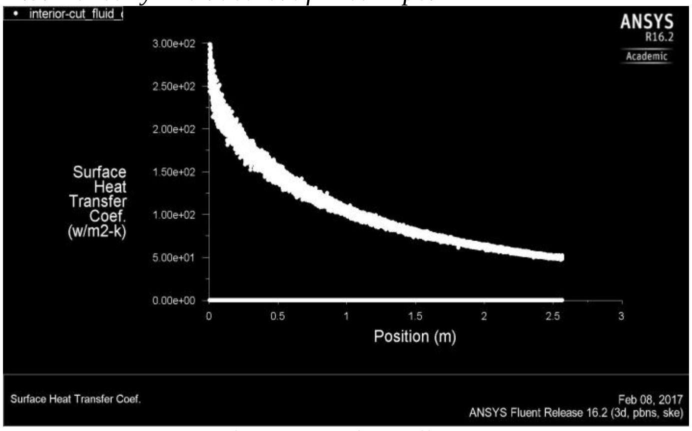

Fig VI.9 Surface heat transfer Coefficient

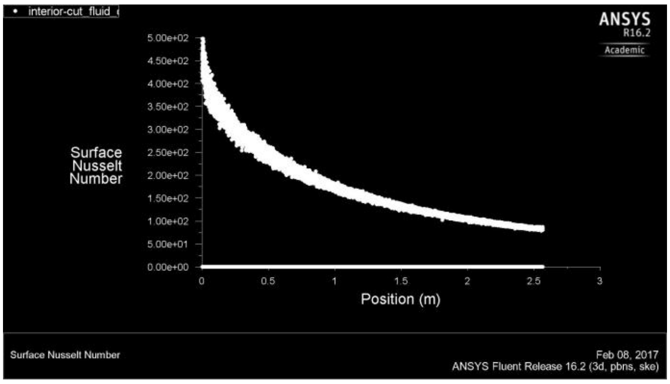

Fig VI.10 Surface Nusselt Number

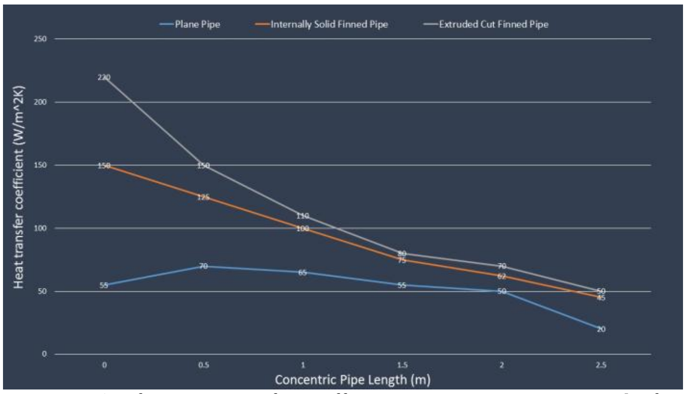

Fig VI.11 (Surface Heat Transfer Coefficient VS Concentric Pipe Length of Three Types of Pipe)

From the above plot (Fig VI.11) the mean surface heat transfer coefficient between concentric pipes are:

	Mean Heat Transfer Coefficient, h_{in} (W/m^{2}K)
Plane Pipe	52.5
Internally Solid Finned Pipe	92.83
Extruded Cut Finned Pipe	113.3

Table IV

So the (Table IV) shows the mean heat transfer coefficient of the Extruded Cut Finned Pipe is higher than the Plane Pipe and Internal Solid Finned Pipe due to higher outer surface area of the Extruded Cut Finned Pipe results in lower the convection and radiation resistance and also lower the volume of the Extruded Cut Fin comparing to Solid Fin will lower the conduction resistance through receiver pipe.

VII. THERMAL RESISTANCE:

Thermal Resistance is a property of heat transfer and the measurement of amount of resistance an object offers to the heat flow. Thermal Resistance of a medium depends on the geometry and the thermal properties of the medium.

Conduction Resistance can be reduced by:

- Reducing the thickness of the material and increasing the surface area of the material. So in Helically Extruded-Cut Fin the thickness of the material is reduced and the surface area of the material is increased, while comparing with Helical Solid Fin.

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Convection and Radiation Resistance can be reduced by:

- Increasing the surface area of the material. So the overall outer surface area of the Helically Extruded-Cut Finned Pipe is maximum than the Helical Solid Finned Pipe and Plane Pipe.

VIII. RESISTANCE CALCULATIONS

A. *Plane Pipe:*

Conduction Resistance:

$$R_{cond} = [1/2\pi KL] \times \ln [r_o / r_i]$$
$$= 5.659 \times 10^{-5} \text{ K/W}$$

Convection Resistance:

$$R_{conv\ outer} = [1/\ 2\pi L] \times [1/h_a \times r_o]$$
$$= 0.08277 \text{ K/W}$$

$$R_{conv\ inner} = [1/2\pi L] \times [1/h_{in} \times r_i]$$
$$= 0.0945 \text{ K/W}$$

Radiation Resistance:

$$R_{rad\ pipe} = [1-\mathcal{E}]/\ [A_0 \times \mathcal{E}]$$
$$= 0.461 \text{ K/W}$$

Surface Area of Reflector (A_{ref}) = 5.77 m^2

$$R_{rad\ reflector} = [1-\mathcal{E}]/\ [A_{ref} \times \mathcal{E}]$$
$$= 0.019 \text{ K/W}$$

$$R_{overall\ rad} = 0.019 + 0.461$$
$$= 0.48 \text{ K/W}$$

Overall Resistance of Plane Pipe:

$$R_{overall} = 1 / [\ [R_{overall\ rad} + R_{conv\ outer}]/[R_{overall\ rad} \times R_{conv\ outer}]] + R_{cond} + R_{conv\ inner}$$
$$= 0.1651 \text{ K/W}$$

B. *Internally Solid finned Pipe:*

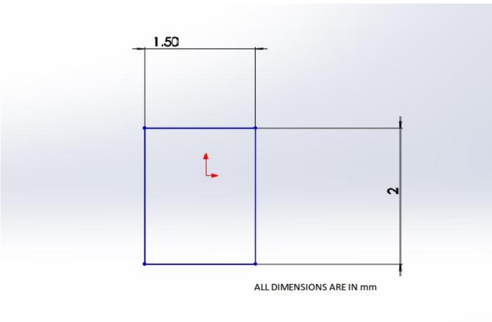

Fig VII.1

Conduction Resistance:

1. Resistance through pipe:

$$R_{cond1} = [1/2\pi KL] \times \ln [r_o / r_i]$$
$$= 5.659 \times 10^{-5} \text{ K/W}$$

2. Resistance through helical inner solid fin:

$$R_{cond2} = L/[k \times A_f]$$
$$= 0.00181 \text{ K/W}$$

Convection Resistance:

$$R_{conv\ outer} = [1/\ 2\pi L] \times [1/h_a \times r_o]$$
$$= 0.08277 \text{ K/W}$$

$$R_{conv\ inner} = 1/[h_{in} \times A_i]$$
$$= 0.0498 \text{ K/W}$$

Radiation Resistance:

$$R_{rad\ pipe} = [1-\mathcal{E}]/\ [A_0 \times \mathcal{E}]$$
$$= 0.461 \text{ K/W}$$

Surface Area of Reflector (A_{ref}) = 5.77 m^2

$$R_{rad\ reflector} = [1-\mathcal{E}]/\ [A_{ref} \times \mathcal{E}]$$
$$= 0.019 \text{ K/W}$$

$$R_{overall\ rad} = 0.019 + 0.461$$
$$= 0.48 \text{ K/W}$$

Overall Resistance of Plane Pipe:

$$R_{overall} = 1 / [[R_{overall\ rad} + R_{conv\ outer}]/\ [R_{overall\ rad} \times R_{conv\ outer}]] + R_{cond1} + R_{conv\ inner} + R_{cond2}$$
$$= 0.1222 \text{ K/W}$$

B. *Helically Extruded-cut finned Pipe:*

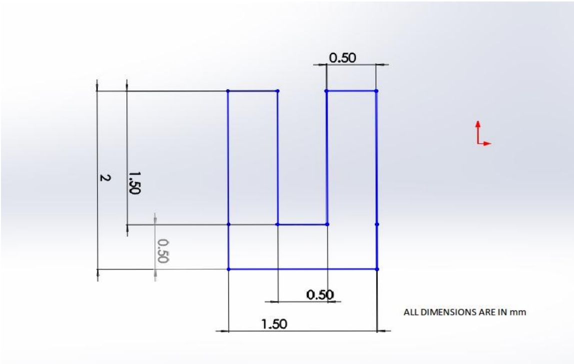

Fig VIII.2 CS of Extruded-Cut Fin

Length of tube with helically extruded-cut (L_c) = 2.496 m

Conduction Resistance:

1. Resistance through the pipe with helically extruded-cut:

$$R_{cond1} = [1/2\pi KL_c] \times \ln [r_o / r_i]$$
$$= 5.8152 \times 10^{-5} \text{ K/W}$$

2. Resistance through the Helically Extruded-cut fin:

$$R_{cond2} = L/[k \times A_f]$$
$$= 0.001415 \text{ K/W}$$

Convectional Resistance:

$$R_{conv\ outer} = 1/[h_a \times A_o]$$
$$= 0.0787 \text{ K/W}$$

$$R_{conv\ inner} = 1/[h_{in} \times A_i]$$
$$= 0.04163 \text{ K/W}$$

Radiation Resistance:

$$R_{rad\ pipe} = [1-\mathcal{E}]/\ [A_0 \times \mathcal{E}]$$
$$= 0.4362 \text{ K/W}$$

Surface Area of Reflector (A_{ref}) = 5.77 m^2

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$$R_{rad\ reflector} = [1-\mathcal{E}]/[A_{ref}\times\mathcal{E}] = 0.019\ K/W$$

$$R_{overall\ rad} = 0.019 + 0.4362 = 0.4552\ K/W$$

Overall Resistance of Plane Pipe:

$$R_{overall} = 1/[[R_{overall\ rad} + R_{conv\ outer}]/[R_{overall\ rad}\times R_{conv\ outer}]] + R_{cond} + R_{conv\ inner} = 0.1093\ K/W$$

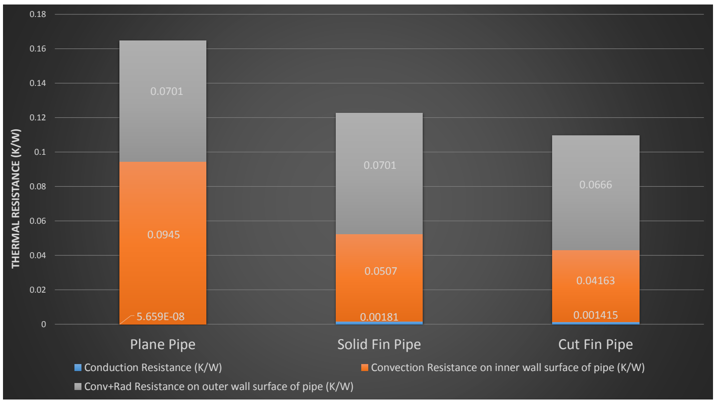

TABLE VII.1 Comparison of thermal resistance for three concentric pipe types

III CONCLUSION

Thus the Thermal performance of all the three types of collector tubes of the PTC were analyzed and it has been arrived that implementation of the extruded cut finned type collector tube shows minimum thermal resistance and maximum overall heat transfer coefficient .This is attributed due to the geometrical exposure of the high heat transfer area of the extruded-cut finned type collector tube to the radiation from the trough which considerably gets heated more than the other types. This increase directly affects the Nusselt number which is the index of thermal conductivity and the convective efficiency happening in real-time. As far as the hydro-dynamical aspect of heat transfer is considered, in the extruded-cut type finned tube it is similar to the internally solid finned tubes as both of the tubes causes turbulence dissipation and increases the kinetic energy and the velocity of the working fluid that flows through it. So future holds a key for PTC technology in the application of extruded-cut type finned collector tubes.

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