1. INTRODUCTION

1.1 ORIGIN OF THE RESEARCH PROBLEM

One of the main obstacles that is encountered during the operation of photo voltaic panels (PV) is overheating due to excessive solar radiation and high ambient temperatures. Overheating reduces the efficiency of the panels dramatically. The maximum power output from the solar cells decreases as the cell temperature increases, the temperature coefficient of the PV panels is -0.5%/°C, which indicates that every 1°C of temperature rise corresponds to a drop in the efficiency by 0.5%. This indicates that heating of the PV panels can affect the output of the panels significantly. For a 250 W solar panel, the output power decreases from 217 W to 58 W in a given period due to increase in temperature. The efficiency of the PV solar panel decreases since output voltage decreases due to increase in temperature. The efficiency also decreases during the latter part of the day since output current decreases due to decrease in solar irradiance. Higher operating temperatures also threaten the long-term stability of cells.

Therefore, efficient and effective solar cell cooling is a key aspect of a solar cell design. For example, the amount of thermal energy available in a solar panel of area 1.675 m x 1.001 m and capable of producing 250W electrical output is 950 W, if the temperature is maintained at 35°C, which would have otherwise gone up to 75°C. This data gives a big inspiration to effectively draw out the heat from the solar panels which can be utilized for any heating applications and thus the solar cells can be operated at a stable temperature regime throughout its operation with constant energy conversion efficiency. This leads to twin benefits of higher power production and low cost utilization of heat energy for any applications.

2. LITERATURE SURVEY

2.1 INTRODUCTION:

The average solar power incident on earth is 1000 W/m^2. This power is far larger than the current world power consumption. A more efficient conversion (15% approx.) of solar energy directly to electrical power is provided by photovoltaic (PV) cells. The PV Cell itself is, in its most common form, made almost entirely from silicon, the second most abundant element in the earth's crust. It has no moving parts and can therefore in principle, if not yet in practice, operate for an indefinite period of time without wearing out.

2.2. PHOTOVOLTAIC EFFECT

PN junction under illumination generates carrier in space charge region and due to electric field electrons move toward N side and holes towards P side. The electron hole pair generated in quasi neutral region move randomly, and some of the generated minority carriers near the space charge region will cross the junction. In this way the minority electrons in P side will move toward N side and minority holes in N side will come to P side. This buildup of a positive and negative charge causes a potential difference to appear across the junction due to light falling on it. This generation of photo voltage is known as photovoltaic effect.

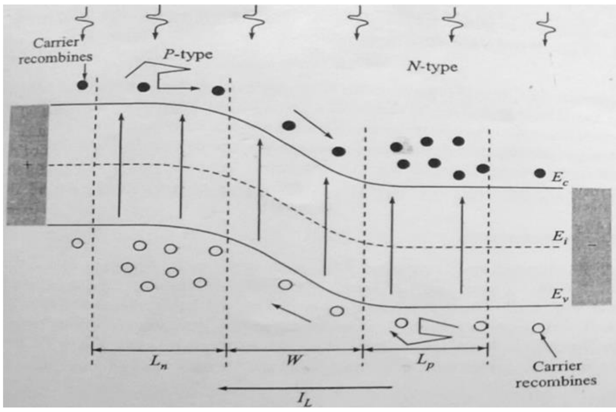


FIG 2.1 PHOTOVOLTAIC EFFECT

2.3 THERMOSIPHON

Thermosiphons have been utilized in applications as diverse as nuclear reactor cooling, internal transformer cooling, electric and gas-fired heaters with various hot and cold side orientations, and oil-filled radiators. These thermosiphons operate on the natural circulation of a fluid due to the temperature dependence of its density. The fluid loop is oriented vertically with respect to gravity. The fluid is heated on one side, reducing its density, and is cooled on the opposing side, increasing its density. The fluid rises as its density decreases through the heated section and then sinks as its density increases through the cooled section, resulting in a constant circulation of fluid.

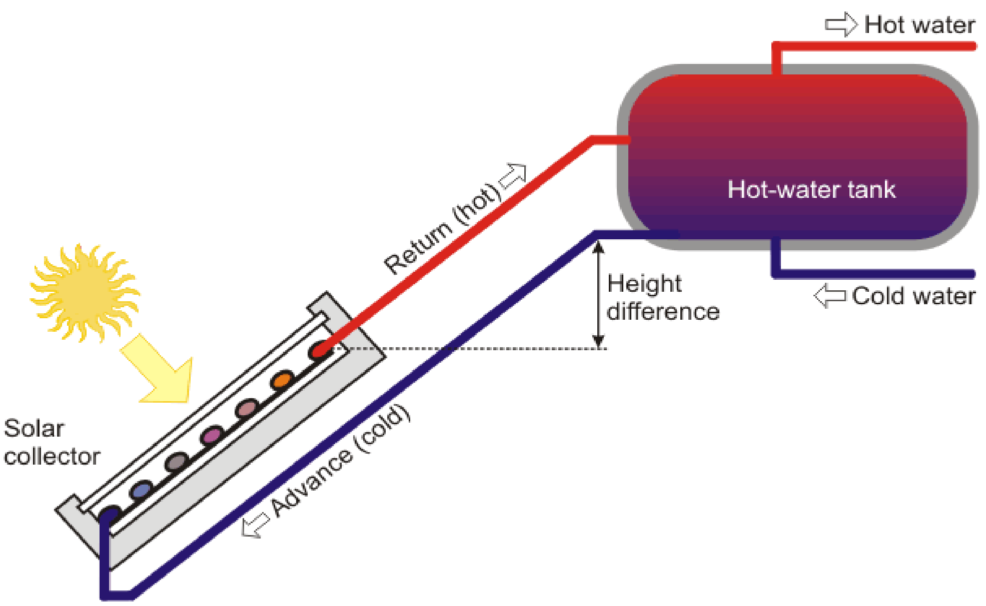


FIG 2.2 THERMOSIPHON

Passive thermal management utilizing single phase thermosiphon behavior is commonplace in the existing electrical grids. Traditional transformer windings are cooled via convection to surrounding oil in oil-filled transformer tanks. Natural convection within the tank carries heat to the walls of the tank, which have sufficient area to dissipate the required thermal load to ambient. General transformer cooling and design is a routine practice in industry, and standards have been developed for transformer loading and life ratings. Some transformers tanks include extended surfaces to increase heat transfer area. Others incorporate arrays of oil filled plates extending out from the cabinet as illustrated in Figure. These plates facilitate circulation of oil internally allowing thermosiphon operation.

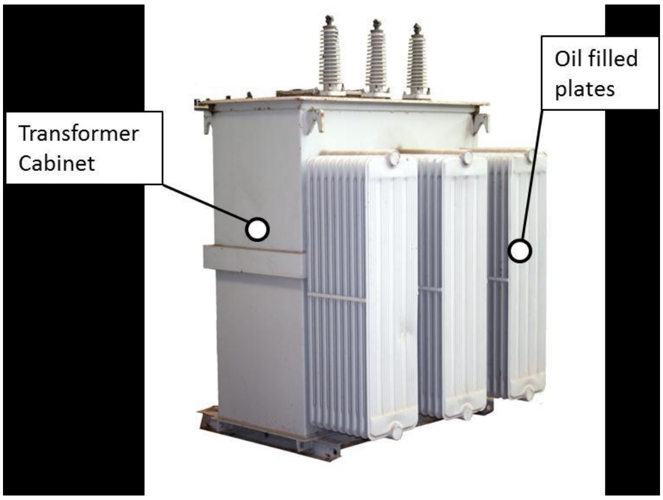


FIG 2.3 THERMOSIPHON COOLING OF TRANSFORMER CABINET

3. HEAT TRANSFER OVER THE PANEL

To calculate the total heat rejected by the solar module which is responsible for the heating of the module the difference between the net input and the output should be calculated. As the input is solely from the sun's radiation which is responsible for the generation of current in the solar cells the total irradiation should be considered. And not all of these radiation is absorbed by the silicon as it has an absorptivity of 0.7.

Out of this input only a certain percentage is converted into useful electrical energy which amounts to just 14-18%. The rest is rejected as heat from the panel. Normally some of this heat is taken away by the air moving over the panel through forced convection, some through radiation in to the atmosphere. The rest of the heat is stored in the panel which is responsible for the drop in power of the panel. If this heat is calculated the rise in temperature of the panel can be obtained at steady state. Since the power is a function of the temperature the energy loss can be found out through a course of time.

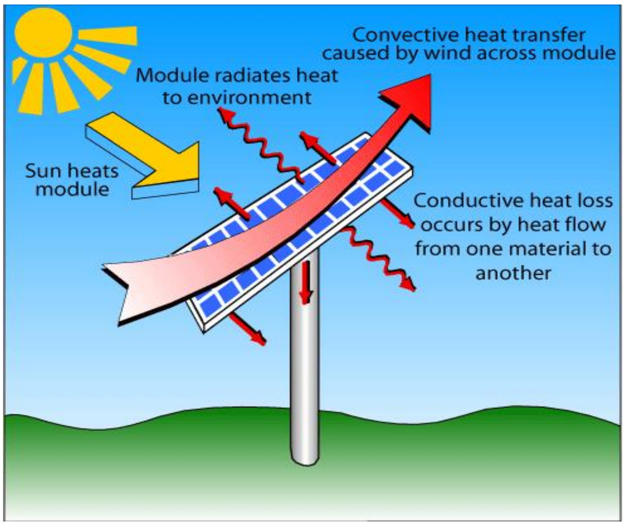


FIG 3.1 HEAT TRANSFER AT PV MODULE

3.1 SOLAR PV MODULE SPECIFICATION

Max Power	= 250 W
Short circuit current	= 37.2 V
Open circuit Voltage	= 8.39 A
Dimension	= 1675 x 1001 x 31 mm
Number of cells	= 60 in series/ 156 x 156 mm
Power Drop Coefficient	= 0.45%/°C

3.2 INPUT POWER

Irradiation, Ø = $1000\ \mathrm{W/m^2}$

Absorptivity of silicon, α = 0.7

Area, A = 60 x 0.156 x 0.156

Input Power $= \emptyset \times \alpha \times A$

$$= 1000 \times 0.7 \times 60 \times 0.156 \times 0.156$$

$$= 1022.112\ \mathrm{W}$$

3.3 OUTPUT POWER

Max Output Power at 25°C = 250 W

Power Drop Coefficient = 0.45%/°C

At any module temperature T_m,

Output power $= 250 - ((0.45/100) * 250 * (T_m - 298))$

$$= 250 - (1.125(T_m - 298))$$

$$= 250 - 1.125T_m + 335.25$$

$$= 585.25 - 1.125T_m$$

By maintaining solar cell at $40℃$, (313K)

Output power $= 585.25 - (1.125 * 313)$

$= 233.125\text{ W}$

3.4 HEAT TRANSFER DUE TO CONVECTION BY AIR

Air Temperature, T_f = 40°C

Wind speed, V = 1m/s

Length, L = 1.675m

Reynolds Number, Re = VL / (Kinematic Viscosity of air@40°C)

= 1 x 1.675 / (1.5036 x 10^{-6})

= 1113993.08

Prandtl Number, Pr = C_p x μ /k

= 0.7

$\text{Nusselt Number, Nu} = 0.664 \text{ x } \mathrm{Re}^{\frac{1}{2}} \text{ x } \mathrm{Pr}^{\frac{1}{3}}$

$= 0.664\text{x } 1113993.08^{\frac{1}{2}} \text{ x } 0.7^{\frac{1}{3}}$

= 622.264

Conductivity, k = 0.027155 W/m K

$\text{Heat transfer coefficient, h} = \text{Nu x } \frac{K}{L}$

= 10.088 $\mathrm{W/m^2K}$

Heat Loss by convection = h x A x ΔT

= 10.088 x 1.675 x 1.001x (T_m– 313)

= 15.93 T_m – 4988.69

3.5 HEAT LOSS DUE TO RADIATION

Stefan Boltzmann constant, σ $= 5.67 \text{ x } 10^{-8} \frac{\mathrm{W}}{\mathrm{m^2.K^4}}$

Total emissivity of silicon cell, ϵ = 0.9

Heat transfer by radiation $= \sigma \text{ x } \epsilon \text{ x } A(T_m^4 - T_{amb}^4)$

$= 5.67 \text{ x } 10^{-8} \text{ x } 0.9 \text{ x } 0.666 \text{ x } 0.464(T_m^4 - 308^4)$

$= 8.2145 \text{ x } 10^{-8}\ T_m^4 - 739.23$

Total Heat Loss $= \text{Heat loss by convection} + \text{Heat los by Radiation}$

$= 15.93\ T_m - 5727.92 + 8.2145 \text{ x } 10^{-8}\ T_m^4$

3.6 STEADY STATE

At steady state

Input power – Output power – Total Heat loss = 0

$$613.2672 - (596.5 - 1.125T_m) - (15.93\,T_m - 5727.92 + 8.2145 \text{ x } 10^{-8}\,T_m^4) = 0$$

$$8.2145 * 10^{-8}\,T_m^4 + 4.779T_m - 2779.93 = 0$$

On solving, we get $T_m = 72℃$

At 35℃ ambient temperature we find that the temperature of the panel can rise up to 72℃ . This will contribute to a significant loss in output power.

4. ENERGY LOSS PER DAY

Since the temperature of the day is always higher than the Standard testing condition temperature which is 25°C there will be energy loss from the panel as heat through the whole day. This heat loss can be obtained by calculating the heat lost at a certain initial time and then obtaining a characteristic equation. It can be iterated to the subsequent time by keeping the preceding temperature values as initial values for the forthcoming iterations. Using this we can plot a graph and integrating the curve over the whole time interval gives the total energy loss.

In order to know the energy lost per day due to increase in module temperature we should know temperature and input radiation data for a whole day. The input radiation can be measured by solar power meter at regular intervals and plotted against time to obtain a curve. The temperature which also varies with time can be obtained from weather data. They are as follows.

TABLE 4.1 TEMPERATURE AND INPUT RADIATION DATA

Hour	Time sec	Temperature °C	Input radiation W/m^2
9:00 AM	0	28	600
10:00 AM	3600	32	800
11:00 AM	7200	34	1000
12:00 PM	10800	34	1040
1:00 PM	14400	37	990
2:00 PM	18000	38	920
3:00 PM	21600	37	750
4:00 PM	25200	37	500

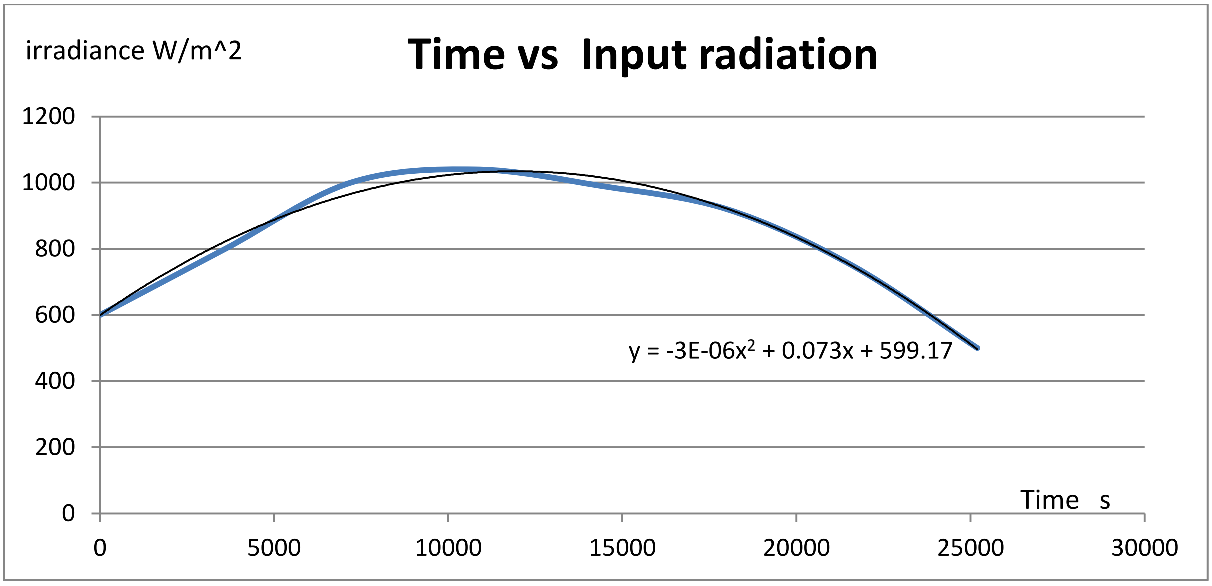


FIG 4.1 TIME VS INPUT RADIATION CURVE

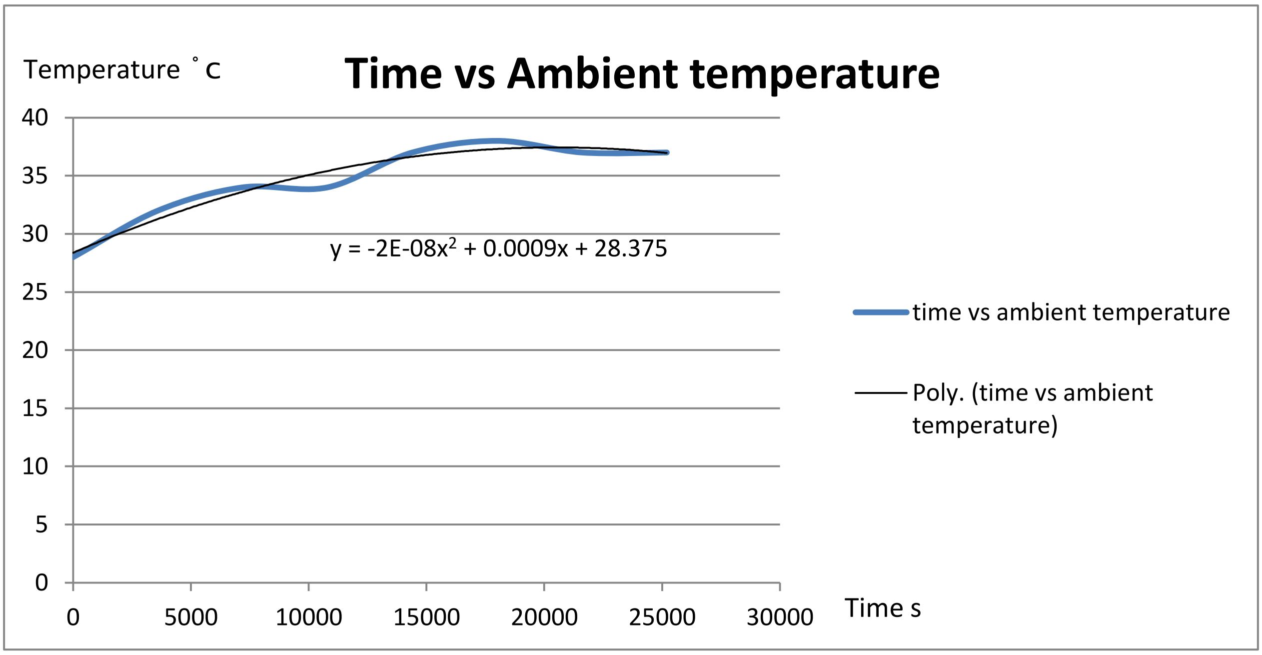


FIG 4.2 TIME VS AMBIENT TEMPERATURE CURVE

At time t = 0(9:00AM) ambient temperature is equal to 28°C whereas input radiation is equal to 600 W/m^2. At this condition module temperature is also assumed to be at ambient temperature at which the solar cell tends to operate.

Irradiation, Ø = 600 W/m^2

Absorptivity of silicon, α = 0.7

Area = 60 x 0.156 x 0.156 m^2

Input Power = Ø x α x A

= 600 x 0.7 x 60 x 0.156 x 0.156

= 613.26 W

$$\text{Output power} = 250 - \left(\left(\frac{0.45}{100}\right) \text{x } 250 \text{ x } (T_m - 298)\right)$$

T_m is the module temperature which is equal to 28°C at t = 0

$$= 250 - (1.125(301 - 298))$$

= 246.625W

Air Temperature, T_{air} = 28°C

Module temperature, T_m = 28°C

Film temperature, T_f $= \frac{T_{air}+T_m}{2}$

= 28°C

Wind speed, V = 1m/s

Length, L = 1.675m

Reynolds Number, Re = VL / (Kinematic Viscosity of air at $T_f = 28$°C)

= 1 x 1.675 / (1.4529 x10^{-5})

= 115286.668 < 5 x 10^5 (Hence, Laminar)

Prandtl Number, Pr = 0.7

$$\text{Nusselt Number, Nu} = 0.664 \text{ x } Re^{\frac{1}{2}} \text{ x } Pr^{\frac{1}{3}}$$

$$= 0.664 \text{ x } 115286.668^{\frac{1}{2}} \text{ x } 0.7^{\frac{1}{3}}$$

= 200.2115

Conductivity, k = 0.0263 W/mK

$$\text{Heat Transfer Coefficient, h} = \text{Nu x } \frac{K}{L}$$

= 2.96 W/m^2K

Heat Loss by convection = h x A x ΔT

= 2.96 x 1.675 x 1.001 x ($\boldsymbol{T_m}$– 301)

= 2.96 x 1.675 x 1.001 x (301– 301)

= 0

Heat transfer by radiation = $\sigma.\epsilon.A.(T_m^4 - T_{amb}^4)$

$= 5.67 \text{ x } 10^{-8} \text{ x } 0.9 \text{ x } 1.626 \text{ x } 0.99(301^4 - 301^4)$

$= 0$

Total Heat Loss = 0

Heat Stored = Input power – Output power – Heat Loss

= 613.26 - 246.625 - 0

= 366.635 W

Heat stored = $C_v \text{ x } \left(\frac{dT}{dt}\right)$

= 366.635 W

Where C_v is called as the heat capacity of the module which is calculated as follows:

HEAT CAPACITY OF PV MODULE

Element of module	Density,þ $(\frac{kg}{m^3})$	Heat Capacity, C_e (J/kg K)	Depth,d (m)	Area x þ x C_e x d (J/K)
Polycrystalline silicon PV cell	2330	677	0.0003	761.765
Tedlar	1200	1250	0.002	4829.22
Glass Face	3000	500	0.003	7243.83
Total				12834.7

$C_v.\left(\frac{dT}{dt}\right)$ = 12834.7 x $\left(\frac{dT}{dt}\right)$

= 366.635

$\frac{dT}{dt}$ $= 0.0285$

The new module temperature after an interval of 30 seconds is calculated as follows:

T_{new} $= T_{mod} + (\frac{dT}{dt} \text{ x}30)$

= 28 +(0.0285x30) = 28.85°C

The above calculation was repeated for the solar panel with new value of module temperature (28.85°C). For the preceding calculation, the values of input radiation and ambient temperature can be found using the equation obtained from their respective curves since both the values change with time. By this we can find the output power and the energy lost at an interval of every 30 seconds in an 8-hour duration.

These iterations were performed using Excel which are as follows.

TABLE 4.3 ENERGY LOSS ITERATIONS

Time (s)	Module temperature (°C)	Output power (W)	Energy lost (W)	Temperature Gradient, dT/dt
0	301.28	227.13328	2.86672	0.030019
30	302.1805717	226.3461804	3.653819626	0.029345
60	303.0609241	225.5767524	4.423247649	0.028631
90	303.9198554	224.8260464	5.173953603	0.027934
120	304.7578635	224.0936273	5.90637266	0.027253
150	305.5754413	223.3790643	6.620935657	0.026588
180	306.3730764	222.6819312	7.318068774	0.025939
210	307.1512509	222.0018067	7.99819325	0.025306
330	307.9104406	221.3382749	8.661725113	0.024689
360	308.6511155	220.6909251	9.309074933	0.024087
390	309.3737387	220.0593524	9.9406476	0.023501
420	310.0787667	219.4431579	10.55684212	0.022929
450	310.7666492	218.8419486	11.15805142	0.022373
480	311.4378286	218.2553378	11.74466222	0.02183
510	312.0927401	217.6829452	12.31705483	0.021302
540	312.7318113	217.1243969	12.87560307	0.020788
570	313.3554624	216.5793258	13.42067415	0.020288
600	313.9641059	216.0473714	13.95262857	0.019801
630	314.5581465	215.52818	14.47182003	0.019328
660	315.137981	215.0214046	14.97859541	0.018867
690	315.7039985	214.5267053	15.47329465	0.018419
720	316.2565799	214.0437492	15.95625081	0.017984

750	316.7960984	213.57221	16.42778997	0.017561
780	317.3229191	213.1117687	16.88823127	0.017149
810	317.8373992	212.6621131	17.33788686	0.01675
840	318.3398879	212.222938	17.77706198	0.016361
870	318.8307265	211.7939451	18.20605495	0.015984
900	319.3102485	211.3748428	18.62515717	0.015618
930	319.7787794	210.9653468	19.03465322	0.015262
960	320.2366372	210.5651791	19.43482089	0.014916
990	320.6841318	210.1740688	19.82593121	0.014581
1020	321.1215658	209.7917515	20.20824854	0.014256
1050	321.5492342	209.4179693	20.58203066	0.01394
1080	321.9674242	209.0524712	20.94752878	0.013633
1110	322.3764161	208.6950123	21.3049877	0.013336
1140	322.7764826	208.3453542	21.65464583	0.013047
1170	323.1678894	208.0032647	21.99673533	0.012767
1200	323.5508949	207.6685178	22.33148218	0.012495
1230	323.9257509	207.3408937	22.65910627	0.012232
1260	324.292702	207.0201785	22.97982152	0.011976
1290	324.6519862	206.706164	23.29383597	0.011728
1320	325.0038351	206.3986481	23.60135191	0.011488
1350	325.3484736	206.0974341	23.90256592	0.011255
1380	325.6861202	205.8023309	24.19766908	0.011029
1410	326.0169874	205.513153	24.486847	0.01081
1440	326.3412814	205.2297201	24.77027995	0.010597
1470	326.6592025	204.951857	25.04814298	0.010391
1500	326.9709451	204.679394	25.32060605	0.010192

1530	327.2766981	204.4121659	25.5878341	0.009998
1560	327.5766444	204.1500128	25.84998719	0.009811
1590	327.8709618	203.8927794	26.10722062	0.009629
1620	328.1598227	203.640315	26.359685	0.009452
1650	328.4433941	203.3924736	26.60752642	0.009281
1680	328.7218381	203.1491135	26.8508865	0.009116
1710	328.9953118	202.9100975	27.08990255	0.008955
1740	329.2639675	202.6752924	27.32470764	0.0088
1770	329.5279528	202.4445693	27.55543073	0.008649
1800	329.7874105	202.2178032	27.78219678	0.008502
1830	330.0424792	201.9948732	28.00512682	0.00836
1860	330.293293	201.7756619	28.22433809	0.008223
1890	330.5399819	201.5600559	28.43994415	0.00809
1920	330.7826715	201.3479451	28.65205491	0.00796
1950	331.0214838	201.1392232	28.86077682	0.007835
1980	331.2565365	200.9337871	29.06621291	0.007714
2010	331.4879438	200.7315371	29.26846289	0.007596
2040	331.7158161	200.5323767	29.46762327	0.007481
2070	331.9402602	200.3362126	29.66378742	0.007371
2100	332.1613795	200.1429543	29.85704568	0.007263
2130	332.379274	199.9525146	30.04748544	0.007159
2160	332.5940403	199.7648088	30.23519123	0.007058
2190	332.8057721	199.5797552	30.4202448	0.00696
2220	333.0145597	199.3972748	30.60272521	0.006864
2250	333.2204907	199.2172911	30.7827089	0.006772
2280	333.4236496	199.0397302	30.96026976	0.006682

2310	333.6241181	198.8645207	31.13547926	0.006595
2340	333.8219753	198.6915936	31.30840645	0.006511
2370	334.0172976	198.5208819	31.47911807	0.006429
2400	334.2101586	198.3523214	31.64767865	0.006349
2430	334.4006299	198.1858495	31.81415051	0.006272
2460	334.5887802	198.0214061	31.97859391	0.006197
2490	334.7746762	197.858933	32.14106703	0.006124

Last few iterations:

Time	Module temperature °C	Output power W	Energy lost W	dT/dt
27840	322.5139773	208.5747838	21.42521617	-0.00598
27870	322.3346641	208.7315036	21.26849641	-0.006
27900	322.1547827	208.8887199	21.11128007	-0.00602
27930	321.9743316	209.0464342	20.95356578	-0.00603
27960	321.7933091	209.2046479	20.79535215	-0.00605
27990	321.6117137	209.3633622	20.63663778	-0.00607
28020	321.4295438	209.5225787	20.47742128	-0.00609
28050	321.2467978	209.6822988	20.31770123	-0.00611
28080	321.063474	209.8425238	20.15747623	-0.00613
28110	320.8795708	210.0032551	19.99674486	-0.00615
28140	320.6950866	210.1644943	19.83550568	-0.00617
28170	320.5100197	210.3262428	19.67375725	-0.00619
28200	320.3243686	210.4885019	19.51149813	-0.00621
28230	320.1381314	210.6512731	19.34872688	-0.00623
28260	319.9513067	210.814558	19.18544202	-0.00625
28290	319.7638926	210.9783579	19.02164209	-0.00627

28320	319.5758874	211.1426744	18.85732562	-0.00629
28350	319.3872896	211.3075089	18.69249112	-0.00631
28380	319.1980974	211.4728629	18.5271371	-0.00633
28410	319.008309	211.6387379	18.36126206	-0.00635
28440	318.8179228	211.8051355	18.19486449	-0.00637
28470	318.6269369	211.9720571	18.02794289	-0.00639
28500	318.4353498	212.1395043	17.86049571	-0.00641
28530	318.2431595	212.3074786	17.69252145	-0.00643
28560	318.0503645	212.4759815	17.52401854	-0.00645
28590	317.8569628	212.6450145	17.35498545	-0.00647
28620	317.6629527	212.8145794	17.18542062	-0.00649
28650	317.4683324	212.9846775	17.01532249	-0.00651
28680	317.2731001	213.1553105	16.84468947	-0.00653
28710	317.077254	213.32648	16.67352	-0.00655
28740	316.8807923	213.4981875	16.50181247	-0.00657
28770	316.6837132	213.6704347	16.3295653	-0.00659
28800	316.4860147	213.8432231	16.15677688	-0.00661

In order to find the total energy lost during the whole day, the energy lost is plotted against time and area under the curve is found.

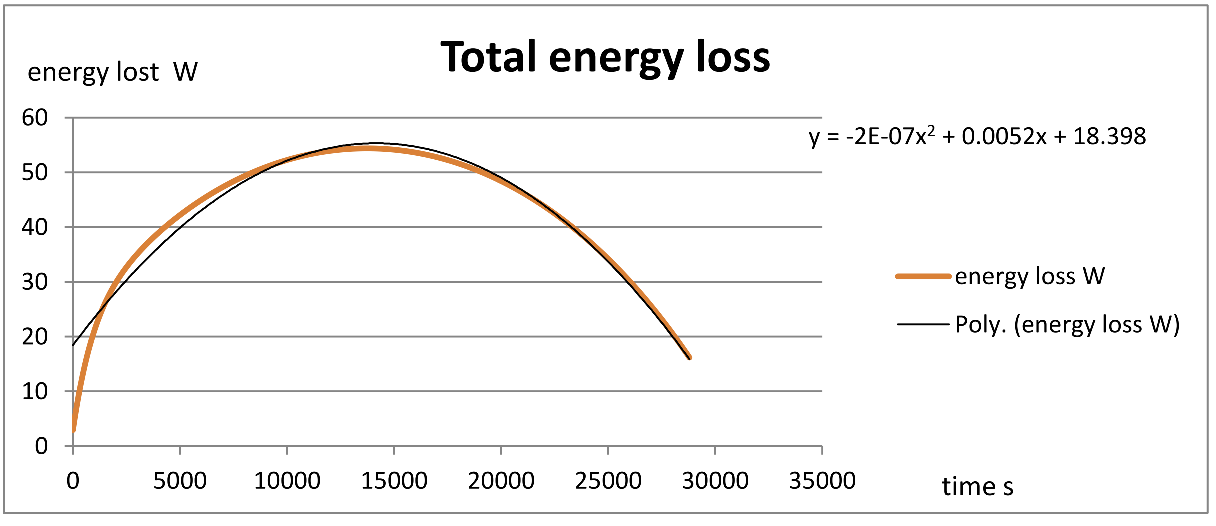


FIG 4.3 ENERGY LOSS CURVE

Total energy loss $= \int_0^{28800} (-2E - 07\, t^2 + 0.0052t + 18.398)\ dt$

$= 1.0938 \text{ x } 10^6 \text{ Joules}$

Thus, a total of $1.0938 \text{ x } 10^6$ Joules is lost through the course of a whole day, which amounts to 37.97 watts of power from the panel.

5. DESIGN OF COOLING SYSTEM

A closed loop thermosiphon model is considered as the base of the design where the hot side is the panel and the cooling on the cold side is done by parallel rectangular channels. The fluid is heated over the panel where it flows in a rectangular channel enclosed by the panel and a insulated glass layer above. Buoyancy drives the fluid due to change in density between the upper and lower layers of the fluid. This layer of fluid passing over the panel should be able to effectively remove the heat from the surface of the panel. And the parallel plates on the cold side should effectively cool the fluid to its inlet temperature before it enters the panel again.

5.1 MODEL

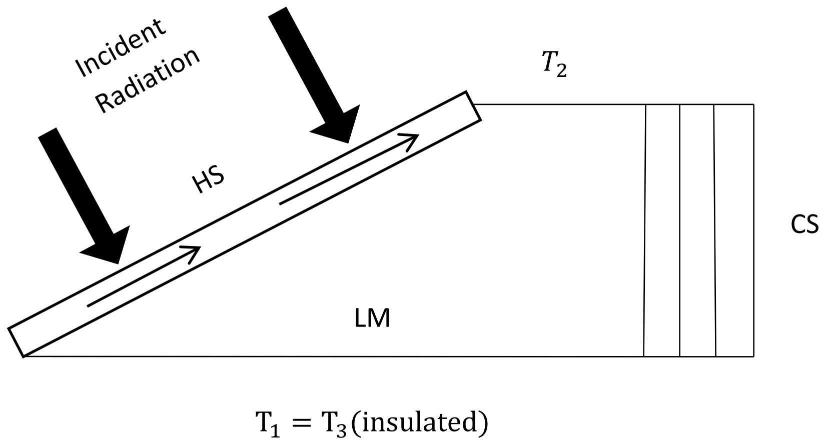


FIG 5.1 CLOSED LOOP THERMOSIPHON OPERATION

CS – Cold side rectangular channel

LM – Insulated Lower manifold

HS – Hot side panel

$\boldsymbol{T_1}$ - Inlet temperature of fluid to the panel

$\boldsymbol{T_2}$ - Outlet temperature of fluid from the panel

$\boldsymbol{T_3}$ - Outlet temperature of fluid from the cold side channel

5.2 CHARACTERISTIC EQUATIONS

The first law of thermodynamics is used to determine the temperature rise($\Delta \boldsymbol{T}$) across the hot and cold sides using the energy balance relations. This is used in the equations (a) and (b).

$$Q_{in} = \dot{m}C_p(T_2 - T_1) \quad \text{(a)}$$

$$Q_{CS} = \dot{m}C_p(T_2 - T_3) \quad \text{(b)}$$

On the cold side the heat should be removed from the fluid through a series of stages by conduction, convection and radiation. The total resistance should be determined in order to find the dimensions.

$$Q_{CS} = \frac{T_2 - T_{amb}}{R_{cs}} \quad \text{(c)}$$

$$R_{cs} = \frac{t}{K.A} + \frac{1}{h_{air}.A} + \frac{1}{h_f.A} + \frac{1}{h_{RAD}.A} \quad \text{(d)}$$

Where R_{CS} is the total thermal resistance on the cold side, $\frac{t}{k.A}$ is the conduction resistance provided by the cold side plates, $\frac{1}{h_f.A}$ is the convection resistance between the fluid and the cold side plate, $\frac{1}{h_{air}.A}$ is the convection resistance between the cold side plate and air.

The next step is to determine the net mass flow rate of the fluid in the system as the convection heat transfer depends on the velocity of the fluid which is crucial in determining the dimensions of the cold side system. For this the net pressure drop in the system can be equated as that is the driving force. It is created by the buoyant forces which drive the fluid and the viscous forces which offer resistance to the flow

Equating the buoyant forces and net viscous forces to find the mass flow rate, is given in the following equations

$$\Delta P_{HS,\ Buoyancy} = (\Delta P_{LM} + \Delta P_{CS} + \Delta P_{HS})_{Viscous}$$

$$g * \beta * \Delta T * L = \left(c * \mu * L * \frac{\dot{m}}{2D_h^2 * \rho * A}\right)_{LM} + \left(c * \mu * L * \frac{\dot{m} * n}{2D_h^2 * \rho * A}\right)_{CS} + \left(c * \mu * L * \frac{\dot{m}}{2D_h^2 * \rho * A}\right)_{HS}$$

n is the number of plates required to remove the heat carried by the fluid

C = 96 for rectangular plate, which is substituted in the above equation in the cold side and hot side viscous force relations and C = 64 for circular manifolds is substituted in the above equation for lower manifold viscous force relation

[Ref: Munsun Young, Fundamentals of fluid mechanics]

5.3 ASSUMPTIONS AND ANALYTICAL CALCULATION

The first step is to determine the thickness of the fluid film required to take out the required heat loss from the panel. The system is taken as flow between inclined channels with one side flux flow.

5.3.4 FINDING INLET TEMPERATURE AND THICKNESS OF FLUID FILM ON PANEL

At steady state

$\text{Input power} - \text{output power} - \text{heat loss} = 0$

$\text{Heat loss} = \text{Input power} - \text{Output power}$

$= 788.987 \text{ W}$

$788.987 = \text{h x A x } \Delta\text{T}$

Where h is the heat transfer coefficient

$\Delta\text{T} = \text{Ts} - \text{T1}$

$\text{Ts} = \text{Solar cell temperature}$

$\text{T1} = \text{Inlet temperature of fluid}$

$\text{Area} = 1.675 \text{ x } 1.001$

$\text{h} = 788.987/(1.675 \text{ x } 1.001 \text{ x } \Delta\text{T})$

$= 470.566/\Delta\text{T}$

$\text{Nusselt number, Nu} = \text{h x D/k}$

$= (470.566 * \text{D})/(\Delta\text{T} * \text{k})$

Where D is the thickness of fluid film

Properties of fluid at 40°C,

$\text{Viscosity, } \mu = 1.96 \text{ mPa.s}$

$\text{Conductivity, k} = 0.5599\text{W/mK}$

$\text{Specific Heat, C}_\text{p} = 3.051 \text{ KJ/kgK}$

$\text{Density, } \rho = 1281.383 \text{ kg/m3}$

$\text{Thermal diffusivity, } \alpha = 1.42599 \text{ x } 10^{-7}$

$\text{Volumetric expansion coefficient, } \beta = 0.000429/\text{K}$

$\text{Nu} = (470.566 \text{ x D})/(\Delta\text{T x } 0.5599)$

$\text{Nu} = (840.446 \text{ x D})/\Delta\text{T}$

$$\mathrm{Ra}\left(\frac{D}{L}\right) = (g \times \beta \times D^4 \times \Delta T)/(\alpha \times \nu \times L)$$

$$= \frac{9.8 \times 4.24 \times 10^{-4} \times D^4 \times \Delta T}{1.3628 \times 10^{-7} \times 1.5 \times 10^{-6} \times 1.675}$$

$$= 1.246 \times 10^{10} \times D^4 \times \Delta T$$

Nusselt number correlation for free convection between inclined channels is given as follows. The below equation is valid for symmetric isothermal plates and isothermal insulated plates considered for $0 \leq \theta \leq 45$ and $\mathrm{Ra}.(\frac{D}{L}) > 200$

$$\mathrm{Nu} = \left(\mathrm{Ra}\frac{D}{L}\right)^{\frac{1}{4}} * 0.645$$

[Ref: Fundamental of Heat and Mass Tranfer by Frank P.Incropera and David P.Dewitt]

$$\frac{840.446 * D}{\Delta T} = (1.246 * 10^{10} * D^4 * \Delta T)^{\frac{1}{4}} * 0.645$$

$$\frac{840.446 * D}{\Delta T} = 334.10 * D * \Delta T^{\frac{1}{4}} * 0.645$$

$$3.9005 = \Delta T^{\frac{5}{4}}$$

$$\Delta T = 2.970$$

$$T_s - T_1 = 2.970$$

$$T1 = 40 - 2.970$$

$$= 37.03℃$$

$$\text{Assume } \left(\mathrm{Ra}\frac{D}{L}\right) = 200 \text{ (critical value)}$$

$$200 = 1.246 * 10^{10} * D^4 * \Delta T$$

$$200 = 1.246 * 10^{10} * D^4 * 2.970$$

$$D = 8.574 * 10^{-3}\mathrm{m}$$

$$D \approx 8.574 \text{ mm}$$

Next the flowrate is found out using the pressure drop equation by equating the buoyancy and viscous forces.

5.3.5 FINDING OUTLET TEMPERATURE OF FLUID AND MASS FLOWRATE

On the hot side, (panel side)

$\text{Breadth} = 1.001\text{m}$, $\text{width} = 8.574 \text{ mm}$, $\text{Area A} = 1.001 \times 0.00857$

$$\text{Hydraulic diameter, } D_h = \frac{2.b.w}{b+w}$$

$$= 0.0173 \text{ m}$$

On the cold side,

$$\text{Breadth, b} = 1.6 \text{ m}, \text{Width, w} = 0.3 \text{ mm}, \qquad \text{Area, A} = 1.6 \text{ x } 0.3$$

$$\text{Hydraulic diameter, } D_c = \frac{2 \text{ x b x w}}{b+w} = 0.00594\text{m}$$

The lower manifold is assumed a circular tube of diameter 25 mm, which is the hydraulic diameter in its case.

For 22 channels, on the cold side to remove the heat carried by the fluid, and substituting the values for density ρ, thermal expansion coefficient β, dynamic viscosity μ at $40°\text{C}$ in equation (f), we get the mass flow rate as $\dot{m} = 0.00509 \text{ Kg/s}$.

This value of mass flow rate is substituted in equation (a) to find outlet temperature of the fluid which is given as follows:

$$Q_{in} = \dot{m}C_p(T_2 - T_1)$$

$$788.987 = 0.002019 \text{ x } 2981.6 \text{ x } (T_2 - 37.08)$$

$$T_2 = 103.02°\text{C}$$

5.3.6 TOTAL THERMAL RESISTANCE

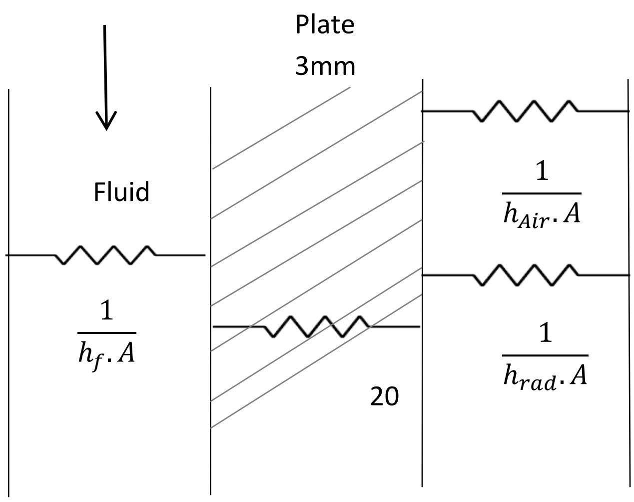
Assuming the lower manifold to be insulated $(Q_{LM} = 0)$,and considering negligible temperature change we get $T_1 = T_3$, and $Q_{in} = Q_{CS}$. We also assume ambient temperature to be $35°\text{C}$.

$$Q_{CS} = \frac{T_2 - T_{amb}}{R_{cs}}$$

$$788.987 = \frac{103.2 - 35}{R_{cs}}$$

$$R_{CS} = 0.0865 \frac{K}{W} \quad \text{(g)}$$

Hence total thermal resistance on the cold side is given as 0.0865 K/W



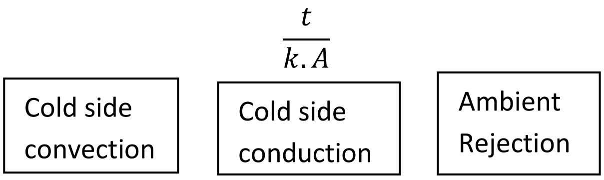


FIG 5.2 COLD SIDE THERMAL CIRCUIT

5.3.6.1 CONDUCTION RESISTANCE

Assumptions:

Thickness $\mathrm{t} = 3\mathrm{mm}$

Aluminium material for plates having thermal conductivity of $180\ \frac{\mathrm{W}}{\mathrm{m.K}}$

Area of plate $= 22 \text{ x } 1.6 \text{ x } 0.5 \text{ x } 0.333 = 5.86\ \mathrm{m}^2$

$$\mathrm{R_{CONDUCTION}} = \frac{\mathrm{t}}{\mathrm{k.A}}$$

$$= \frac{37}{117200}\ \frac{\mathrm{K}}{\mathrm{W}}$$

5.3.6.2 CONVECTION RESISTANCE BETWEEN FLUID AND PLATE

$\mathrm{T_2} = 103.02\ ℃$

$\mathrm{T_3} = \mathrm{T_1} = 37.190℃$

Considering the properties at mean temperature $\mathrm{T_f}$, which is given as $\frac{\mathrm{T_2+T_3}}{2}$

$\mathrm{T_f} = 70.105\ ℃$

Properties of fluid at 70.105 ℃ is given as follows

dynamic viscosity $\mu = 0.00102\ \mathrm{mPa.s}$

thermal conductivity $\mathrm{k} = 0.5697\,\frac{\mathrm{W}}{\mathrm{mK}}$

specific heat $\mathrm{C_p} = 3073.61\,\frac{\mathrm{J}}{\mathrm{KgK}}$

Density $\rho = 1281.49\,\frac{\mathrm{Kg}}{\mathrm{m^3}}$

Kinematic viscosity $= 8.002\text{x } 10^{-7}\ \frac{\mathrm{m^2}}{\mathrm{s}}$

Thermal diffusivity α = 1.44 $\text{x}10^{-7}$ $\left(\frac{m^2}{s}\right)$

Flow rate for a single channel is given as total flow rate divided by 22, hence

$$\rho \text{ x A x V} = \frac{0.002019}{22}$$

Where A is the cross-sectional area of cold side plate given as $1 \text{ x } 0.003(3 \text{ x } 10^{-3} m^2)$ and V is the velocity of fluid which is required to find the Reynolds number in turn leads to finding of Nusselt number

From the above equation by substituting values for Area and density we find velocity of fluid to be $1.75 \text{ x } 10^{-5} \frac{m}{s}$.

$$\text{Reynold number} \qquad = \frac{\rho \text{ x V x } D_h}{\mu} = \frac{1281.49 \text{ x } 1.45 \text{ x } 10^{-5} \text{ x } 0.010713}{0.001025} = 0.194$$

$$\text{Grashof number} = \frac{g\beta(T_f - T_{amb})w^3}{\vartheta^2}$$

Where w is the width of the channel which is equal to 3mm. on substituting the known property values, $T_f - T_{amb}$ =40.007, we get

$$\text{Grashof number Gr} = 8016.9$$

The following are the conditions for predicting the type of convection heat transfer occurring between the fluid and the plate. They are

If $\frac{Gr}{Re^2} \gg 1$, free convection heat transfer

$$\frac{Gr}{Re^2} = 1, \text{Mixed Convection heat transfer}$$

$$\frac{Gr}{Re^2} \ll 1, \text{Forced convection heat transfer}$$

In our case $\frac{Gr}{Re^2} = 63956.326$ which is much greater than one, hence heat transfer between fluid and plate is considered to be free convection.

$$\text{Rayleigh Number} = \frac{g\beta\Delta T w^3}{\alpha\nu} = 44357.9$$

where, $\Delta T = T_f - T_{amb}$

Nusselt number correlation for free convection heat transfer between parallel plates is given as:

$$Nu = \left(\frac{48}{Ra\left(\frac{W}{L}\right)} + \frac{2.51}{\left(Ra\left(\frac{W}{L}\right)\right)^{\frac{2}{5}}} \right)^{-\frac{1}{2}}$$

[Ref: Fundamentals of Heat and Mass Transfer by Frank P.Incropera, David P.Dewitt]

On substituting known values for width, Length L(0.5m), Rayleigh number Ra, we get the dimensionless form of Heat transfer coefficient as 1.17 i.e, Nu = 1.17

$$\text{Hence Nu} = \frac{\text{h x w}}{\text{k}}$$

$$\text{h} = \frac{1.17 \text{ x } 0.552}{0.003}$$

heat tranfer coefficient between fluid and plate is given as $h_{fluid} = 321.59 \frac{W}{m^2.K}$

Convection resistance between fluid and the plate is given as

$$R_{CONVECTION,\ FLUID} = \frac{1}{h_{fluid}.A}$$

Where A is area involved in convection between plate and the fluid. Since there are 22 channels, the number of sides in convection are 44(22x2). Hence the total area is equal to m^2(44 x Breadth x Height)

$$\frac{1}{h_{fluid}.A} = \frac{1}{321.59 \text{ x } 19.98} = \frac{1}{6430.08} \quad \frac{K}{W}$$

5.3.6.3 CONVECTION RESISTANCE BETWEEN PLATE AND AIR

There are two parts of convection resistance between plate and air, i.e, the convection resistance $h_{AIR,1}$ for free convection heat transfer between vertical channels and the convection resistance $h_{AIR,2}$ for free convection heat transfer between end plate and the surrounding air.

Properties of air at 80.007ºC

Thermal Expansion Coefficient, β = 2.975×10^{-3}/K

Thermal Diffusivity, α = $28.77 \times 10^{-6} m^2/s$

Kinematic Viscosity, v = $1.933 \times 10^{-5} m^2/s$

Thermal conductivity, k = 0.030w/mK

Prandtl number, Pr = 0.69

$$\text{Rayleigh Number} \qquad = \frac{g\beta\Delta TS^3}{\alpha\nu}$$

$\Delta T = Tf - Tamb = 80.007 - 35 = 45.007^0C$

S = Spacing between the channels (unknown)

On substituting the known values in above equation, we get

$$\text{Rayleigh Number} = 1.848 \text{ x } 10^9 \text{ x } S^3$$

Nusselt Number Correlation for free convection between vertical channels is given as

$$Nu = \left(\frac{48}{Ra\left(\frac{S}{L}\right)} + \frac{2.51}{\left(Ra\left(\frac{S}{L}\right)\right)^{\frac{2}{5}}} \right)^{-\frac{1}{2}}$$

[Ref: Fundamental of Heat and Mass Tranfer by Frank P.Incropera and David P.Dewitt]

Where L = Length of channel = 0.5m

$$Ra = 1.848 \text{ x } 10^9 \text{ x } S^3$$

Hence

$$Nu = \left(\frac{1.298 \text{ x } 10^{-8}}{S^4} + \frac{3.73 \text{ x } 10^{-4}}{S^{\frac{8}{5}}} \right)^{-\frac{1}{2}}$$

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

$$h_{AIR,1} = Nu\frac{K}{S}$$

$$= \left(\frac{(1.298 \text{ x } 10^{-8})}{S^4} + \frac{3.73 \text{ x } 10^{-4}}{S^{\frac{8}{5}}} \right)^{-\frac{1}{2}} * \frac{0.029}{S}$$

$$\frac{1}{h_{AIR,1}.A} = \left(\frac{(1.298 \text{ x } 10^{-8})}{S^4} + \frac{3.73 \text{ x } 10^{-4}}{S^{\frac{8}{5}}} \right)^{\frac{1}{2}} \text{x} \frac{S}{0.029 \text{ x } 42 \text{ x } 1 \text{ x } 0.333}$$

Where A is the area required for convection between vertical channel and ambient air which is equal to 44 x 1.6 x 0.5 [number of sides - 44, Length – 0.5m, Breadth of the plate – 1.6m]

The second part of convection resistance is the resistance between end plates and ambient air, which is calculated as follows:

The Rayleigh number for heat transfer between vertical plate and air is given as

$$\text{Rayleigh Number } Ra_L = \frac{g\beta\Delta TL^3}{\alpha\nu}$$

Where L is the length of the vertical plate which is equal to 0.5m

$\Delta T = T_f - T_{amb} = 45.07°C$

Also substituting the other known property values for air, we get

$$\text{Rayleigh number } Ra_L = 231067033.7$$

Nusselt number correlation for heat transfer between vertical plate and air is given as

$$Nu = 0.68 + \frac{0.67\, Ra_L^{\frac{1}{4}}}{\left(1 + \left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right)^{\frac{4}{9}}}$$

On substitution of known values, we get

Nu = 50.19

$$h_{AIR,2} = Nu.\frac{k}{L}$$

Where, thermal conductivity of air, k = 0.029, length of the plate, l = 0.5m

$$\text{Hence } h_{AIR,2} = 4.56 \frac{W}{m^2.K}$$

$$\text{Convection resistance between end plates and air } = \frac{1}{h_{AIR,2}.A}$$

A is the area of the end plates which is given 2 x length of the plate x breadth of the plate. i.e, 2 x 1.6 x 0.5

$$\frac{1}{h_{AIR,2}.A} = \frac{1}{4.56}\frac{K}{W}$$

5.3.6.4 RADIATION RESISTANCE BETWEEN PLATE AND AIR

Net heat transfer by radiation between vertical channels is zero as the channels are at same temperature. Hence the only heat transfer by radiation is between end plates of the channel and air.

Radiation heat transfer coefficient between plate and air is given as $4\sigma\varepsilon T_m^4$. This equation is valid only if the temperature difference between plate and air is less than 100℃.

$$h_{RAD} = 4\sigma\varepsilon T_m^4$$

Where σ, stefan BoltZmann constant $= 5.67 \text{ x } 10^{-8} \frac{W}{m^2.K}$

ε, total hemispherical emissivity of black paint $= 0.98$

$$T_m, \text{Average temperature between plate and air} = \frac{T_f + T_{amb}}{2} = \frac{63.007 + 35}{2} = 49.004℃$$

$$T_m = 322.004 \text{ K}$$

On substitution, we get $$h_{RAD} = 8.024 \frac{W}{m^2.K}$$

Radiation Resistance between end plate and air = $\frac{1}{h_{RAD}.A}$

Where A is the end plates area given as 2x1.6x0.5

$$\frac{1}{h_{RAD}.A} = \frac{1}{8.024}\frac{K}{W}$$

5.3.7 SPACING BETWEEN THE CHANNELS

We know that the convection resistance between end plates and air, and radiation resistance between end plates and air are in parallel. Hence equivalent resistance is obtained using following equation

$$\frac{1}{R_{EQ,\ PAR}} = \frac{1}{\frac{1}{h_{AIR,2}.A}} + \frac{1}{\frac{1}{h_{RAD}.A}}$$

= 8.4033

$$R_{EQ,\ PAR} = 0.119 \frac{K}{W}$$

The other resistances like convection resistance between fluid and plate, conduction resistance by plate, convection resistance between vertical channel and air are in series. The equivalent resistance in this case is given as

$$R_{EQ,\ SERIES} = \frac{t}{k.A} + \frac{1}{h_f.A} + \frac{1}{h_{AIR,1}.A}$$

$$= \frac{37}{117200} + \frac{1}{6430.34} + \left(\frac{(1.298 \text{ x } 10^{-8})}{S^4} + \frac{3.73 \text{ x } 10^{-4}}{S^{\frac{8}{5}}}\right)^{\frac{1}{2}} \text{x } \frac{S}{0.029 \text{ x } 44 \text{ x } 1.6 \text{ x } 0.5}$$

Total resistance $= R_{EQ,\ SERIES} + R_{EQ,\ PAR}$

$$= \frac{1}{4184630} + \frac{1}{6430.34} + \left(\frac{(1.298 \text{ x } 10^{-8})}{S^4} + \frac{3.73 \text{ x } 10^{-4}}{S^{\frac{8}{5}}}\right)^{\frac{1}{2}} \text{x } \frac{S}{0.029 \text{ x } 44 \text{ x } 1.6 \text{ x } 0.5} + 0.056$$

We know that for taking the heat of 788.987 W from the cold side we need a total resistance value of about 0.068 K/W.

Hence, $0.161 = \frac{1}{4184630} + \frac{1}{6430.34} + \left(\frac{(1.298 \text{ x } 10^{-8})}{S^4} + \frac{3.73 \text{ x } 10^{-4}}{S^{\frac{8}{5}}}\right)^{\frac{1}{2}} \text{x } \frac{S}{0.029 \text{ x } 44 \text{ x } 1.6 \text{ x } 0.5} + 0.056$

On solving the above equation, we get

$\text{Spacing between the channel} = 7.37\ \text{mm}$ (Valid solution)

5.4 THREE DIMENSIONAL MODEL

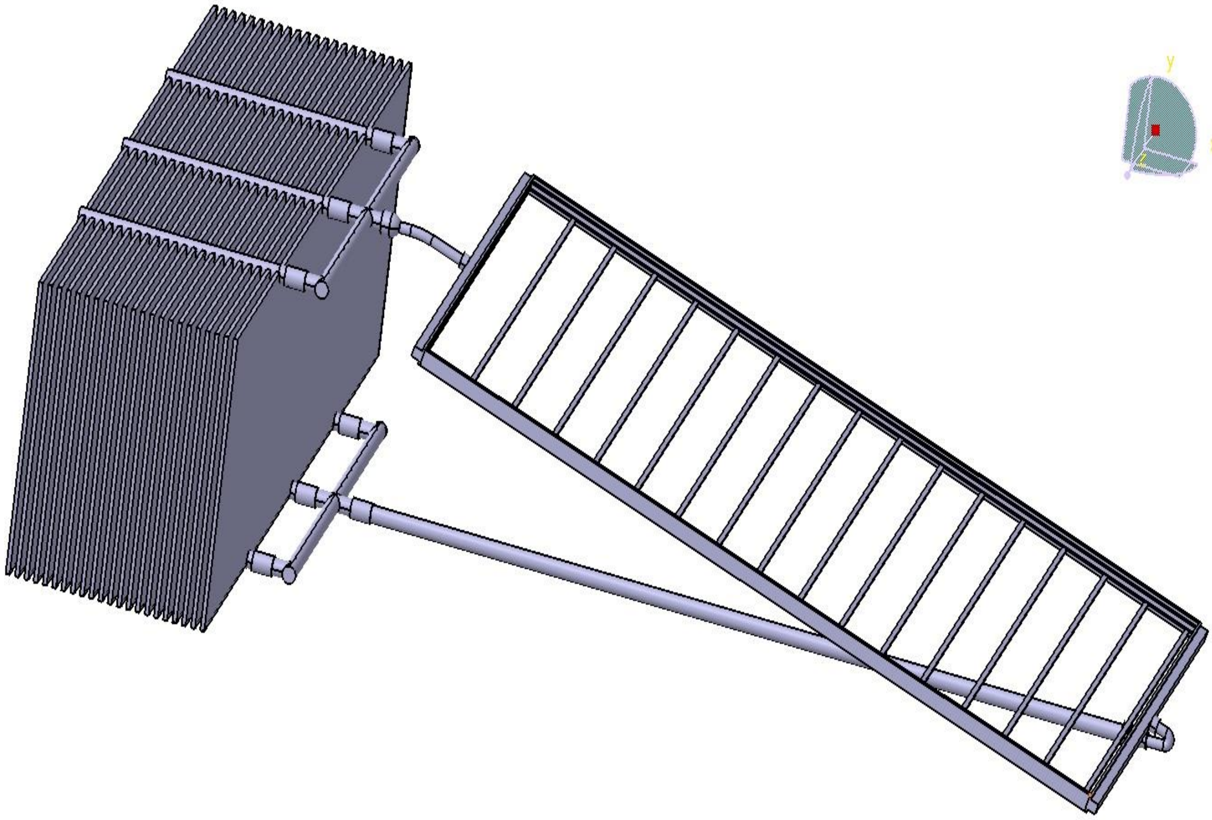


Fig 5.3 THREE DIMENSIONAL MODEL OF PARALLEL PLATE HEAT EXCHANGER

A three dimensional model of the system was generated using Pro engineer software to visualize and then manufacture the required system. It consists of a slotted frame to fix the panel and the glass covering between which the fluid passes through which is closed on either sides with a hub to channel the fluid through the manifolds. As the plates are long the manifolds are branched out for even distribution of the fluid inside the plates. The plates are connected to the manifolds by cutting out slots in them. Thus the plate array is slotted and welded to the upper and lower manifolds. The material considered is aluminum as it is easy to machine and it has very high conductivity which will help in heat transfer.

5.5 WEIGHT CALCULATION:

Density of aluminium = 2700 kg/m^3

Number of channels = 22

Volume of cold channel= Area of single channel x thickness of aluminium sheet x 22

= 0.5 x 1.6 x 0.003 x 22 x 2

$= 0.1056\ m^3$

Weight of heat exchanger = Volume x density

= 0.1056 x 2700

= 285.12 kg

Since the weight of heat exchanger is too high it is not practically possible to place it on roofs. So, we have planned to go for fin based heat exchanger design.

6. DESIGN OF FIN TYPE HEAT EXCHANGER

6.1 FINS:

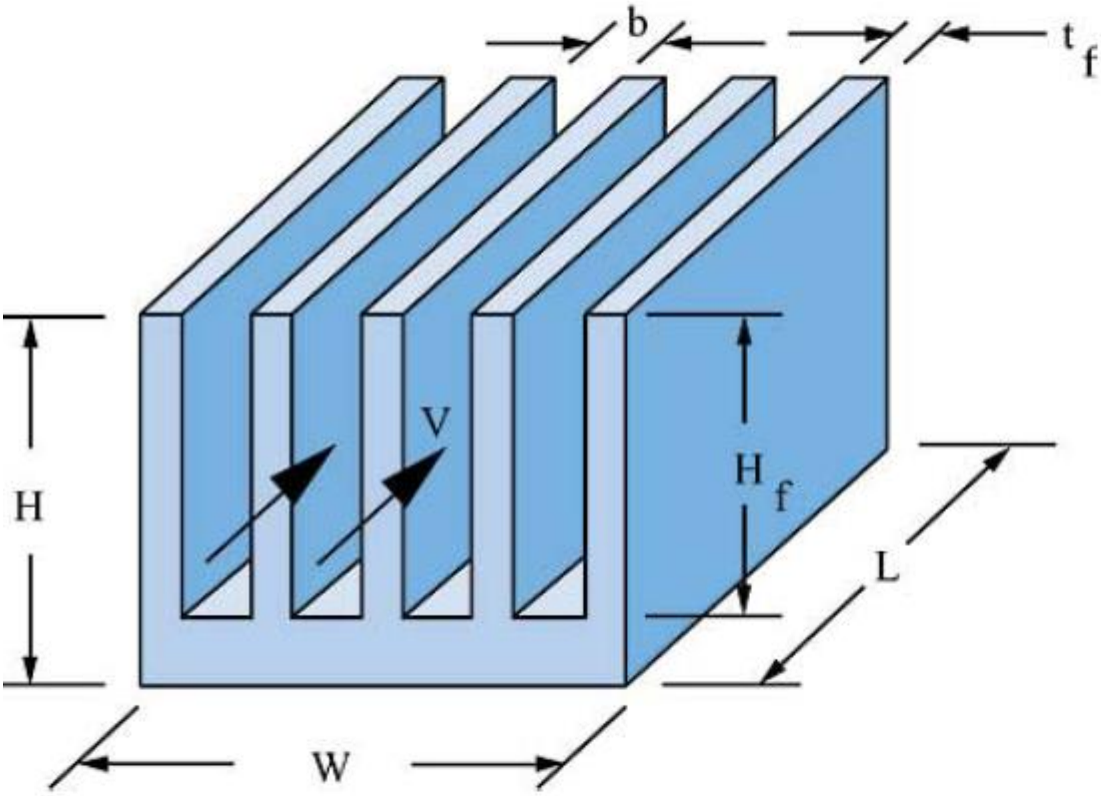


Fig 6.1 Parallel plate fin heat sink

Thickness of fin = 0.003 m.

Air gap is given by

$$b = \frac{W - N_{fin}\ x\ t_{fin}}{N_{fin} - 1}$$

Exposed base surface area is given by

$$A_{base} = (N_{fin} - 1) \text{ x b x L}$$

Heat transfer area per fin is given by

$$A_{fin} = 2 \text{ x } H_f \text{ x L}$$

Volumetric flow rate is given by

$$V = \frac{G}{N_{fin} \text{ x b x } H_f}$$

To determine the heat transfer coefficient acting upon the fins, an equation developed by Teertstra relating Nusselt number, Nu, to Reynolds number, Re, and Pr number, Pr, may be employed. The equation is

$$Nu_b = \left[\frac{1}{\left[\frac{\text{Re x Pr}}{2}\right]^3} + \frac{1}{\left[0.664 \text{ x } \sqrt{Re} \text{ x } Pr^{0.33} \text{ x } \sqrt{1 + \frac{3.65}{\sqrt{Re}}}\right]^3} \right]^{-0.33}$$

Prandtl number is given by

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

Modified channel Reynolds number defined as

$$Re = \frac{\rho \text{ x V x b}}{\mu} \text{ x } \frac{b}{L}$$

Heat transfer coefficient is given by

$$h = Nu_b \text{ x } \frac{k_{fluid}}{b}$$

Where k_{fluid} is the thermal conductivity of the cooling fluid (i.e. air).

The efficiency of the fins may be calculated using

$$\eta_{fin} = \frac{\tanh(m \text{ x } H_f)}{m \text{ x } H_f}$$

Where m is given by

$$m = \sqrt{\frac{2 \text{ x h}}{k_{fin} \text{ x } t_{fin}}}$$

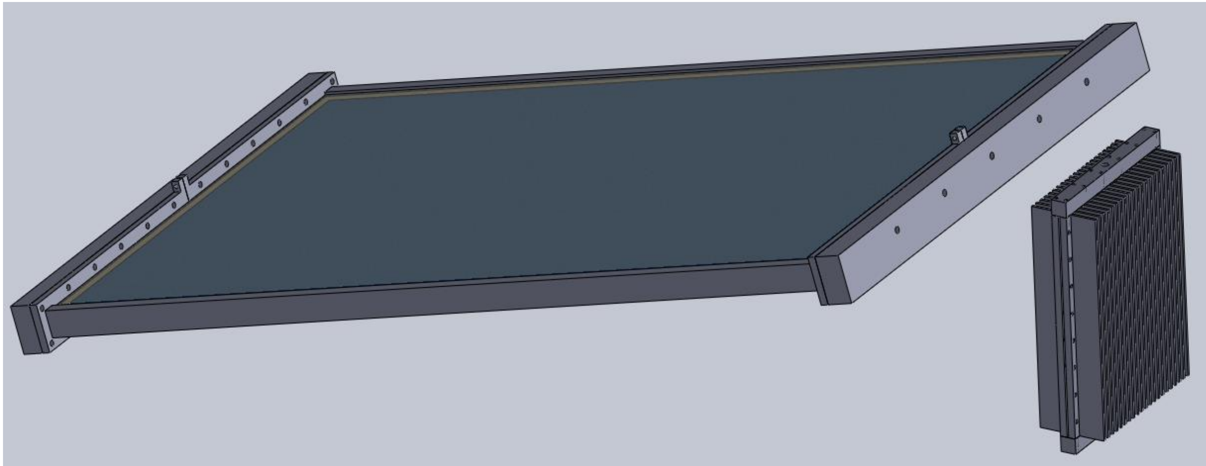
Table 6.1 Fin Calculations

Rectangular Fins

Parameter	Value	Unit
Fin Length, L	0.05	m
Number of Fins, n	**21**	
Thickness of fin, t	0.005	m
Air gap, b	0.00975	m
Area of base, A_{base}	0.084536595	m^2
Area of fins, A_{fin}	0.0433521	m^2
Volumetric flow rate, G	0.01	m^3/s
Velocity of air flow, V	0.976800977	m/s
Prandtl number	0.67410032	
Reynolds number	11.65780743	
Nusselt number	2.564965918	
Heat transfer coefficient, h	**7.774997067**	**W/m^2K**
Efficiency of fins	0.985846351	
Total area of fin plate	0.982045296	m^2
Required heat transfer coefficient, h_{req}	**7.589573995**	**W/m^2K**

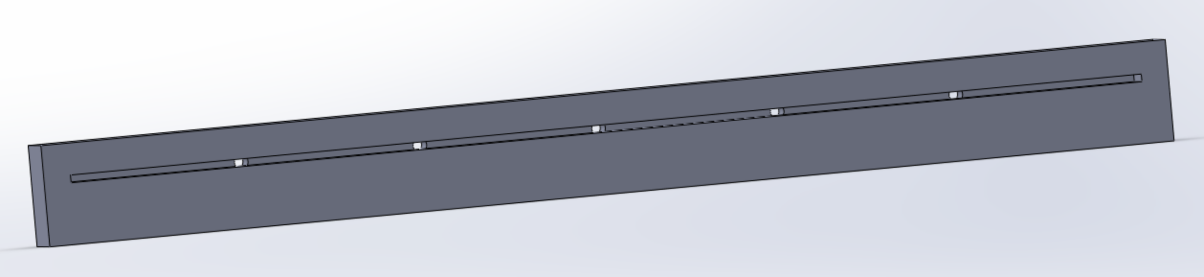
6.2 THREE DIMENSIONAL MODEL

7.4 DESIGN OF THE SETUP

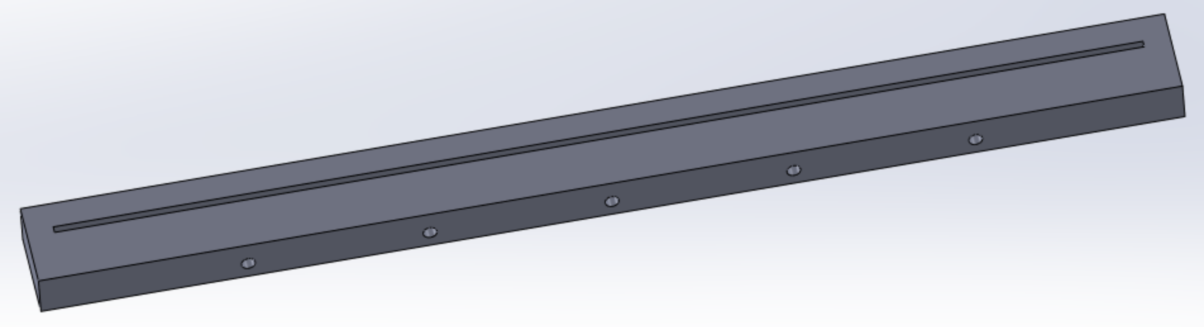


COMPONENTS REQUIRED

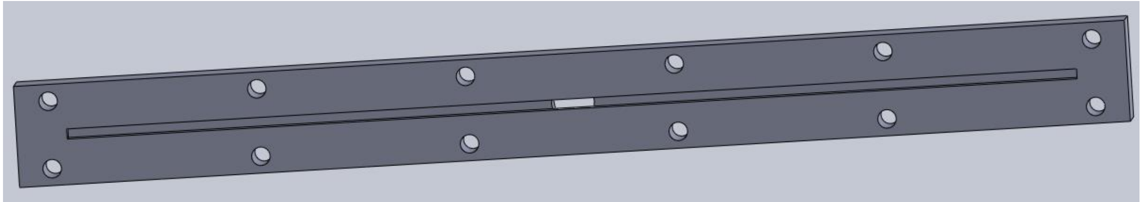
i. Manifold (Panel Side)



(Top Manifold)

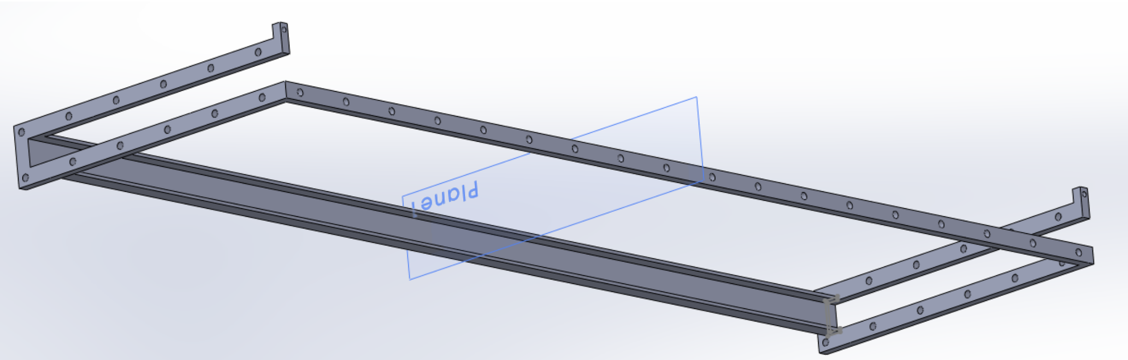


(Bottom Manifold)

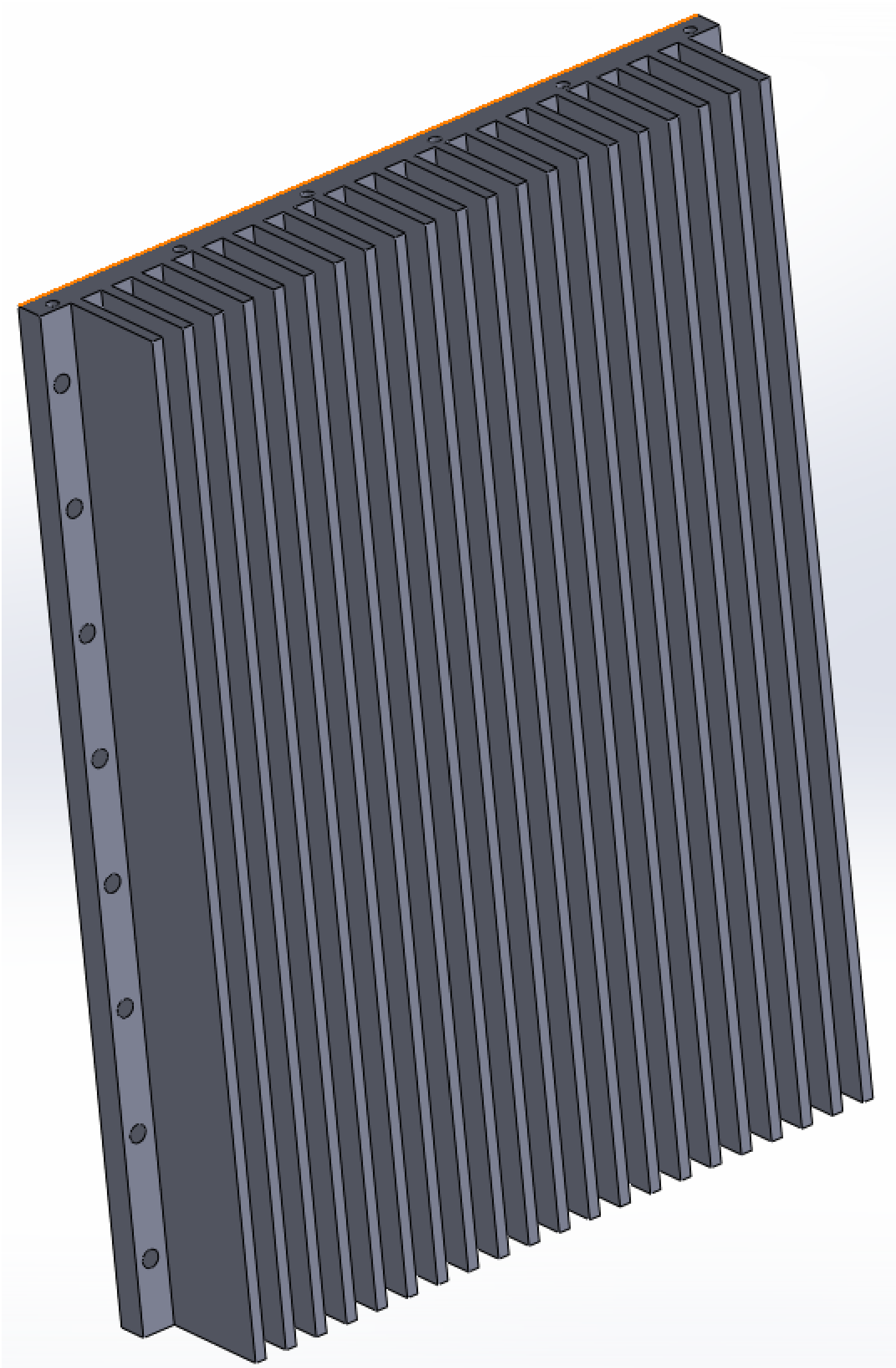


ii. Heat Exchanger Manifold

iii. Panel Side Support



iv. Heat Exchanger



v. Tubing
vi. Anti-Reflective glass

vii. Dielectric Fluid (Propylene Glycol Solution)
viii. Thermocouple
ix. Flowmeter

6.4 SAVE IN ENERGY

Maximum power output = 250 W

Power drop coefficient = 0.45 %/°C

Panel Surface Temperature = 40 °C

Output Power = 233.125 W

Energy generated per day = 233.125 x 28800
= 6714000 J

At standard conditions (25 °C),
Energy generated per day = 250 x 28800
= 7200000 J

Energy lost with cooling = 7200000 - 6714000
486000 J

Energy lost without cooling = 1.0938×10^6 J

Save in Energy = $1.0938 \times 10^6 - 486000$
= 607800 J
Instantaneous Save in Energy = 21.1 W

7. CONCLUSION

Probing into solar photovoltaics with its basic equations the rise in temperature of the panel leads to a drop in power which is linear with respect to each other. Using experimental data it is verified that power of the solar drops with increase in temperature of the panel. This was experimentally obtained by using the Ecosense solar apparatus and comparing it with the calculated theoretical data .The calculation of the surface temperature at steady state at an ambient temperature was found to be 85°C which will lead to a significant loss of power. The temperature and radiation with respect to time is found for a period of 8hours. From the observed values the total energy loss calculated from the panel is about 676213 joules ie about 23.47 watts through a whole day. Using the cooling system designed we can maintain the temperature of the panel at 42°C. At this temperature the output power of the panel is a constant 39.28 W. This leads to saving of about 17.76W per day. Since this value is for a single panel it may seem significant when we consider it for a whole solar power grid. The proposed design is a single time investment. The cost incurred is from the materials used which is aluminum and the manufacturing expenses. The cost of the cooling fluid is should also be considered. In the future the design has to be manufactured and tested. Owing to the accuracy of the design the testing results are expected to be close to the design outputs.

8. REFERENCES

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