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Investigations on Heat Pipe for Thermo-Electric Generator (TEG)

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Abstract

Thermal energy in the industry which is being dissipated as waste heat to the environment can be utilized further for power generation. Two promising technologies that were found to be useful for this purpose were Heat Pipe and Thermo-Electric Generator (TEG).In this study, the heat energy of high pressurized steam is used to increase the temperature of the hot side of TEG. A heat pipe of capacity 200 Watts is used on cold side of the TEG to decrease the temperature of cold side. The current drawn from the TEG can be used for other power consuming applications such as flow meter. The readings of output voltage for a pressure range of 1-10 gauge bar at constant interval of 1 gauge bar are taken experimentally. Further, the results of heat pipe are experimentally investigated with the fin assembly which was being used.It is found that voltage output and power out is increased by 26% and 43% respectively when Heat Pipe assembly is used at air velocity of 1.053 m/s and ambient temperature of 40°C.

Keywords:*Heat pipe, Thermo-Electric generator, Seebeck effect, Heat transfer, Thermoelectrics*

1. Introduction

A heat pipe is a device whicheffectively transports thermal energy from its one point to the other. Thermodynamic working fluid works as the medium to transfer heat from one point to another point. It utilizes the latent heat of the vaporization. (H. Jouhara et al., 2017)A heat pipe is a hollow cylinder enclosed by a wick structure.It mainly consists of three sections such as Evaporator, Condenser and Adiabatic section. Figure 1 shows the working of Heat Pipe. When heat is applied at the evaporator section, the liquid in the heat pipe heats and evaporates. As the evaporating fluid fills the heat pipe hollow center, it diffuses throughout its length. Condensation of the vapor occurs wherever the temperature is even slightly below that of the evaporation area. As it condenses, the vapor gives up the heat it acquired during evaporation. Then the condensate returns through the wick structure because of capillary action.

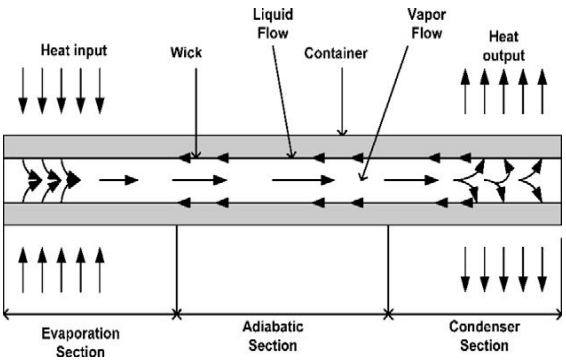

Fig. 1 Schematic of heat pipe operation (Nikhil S. Chougule et al., 2016)

Thermoelectric Generator (TEG)works on the Seebeck effect which states that the temperature difference between junctions of two dissimilar conductors or semiconductors produces a voltage difference between the two junctions. TEG is made up of many n-type and p-type semiconductors. These semiconductors are placed in such a way that they are thermally in parallel and electrically in series. The thermally conducting plates are joined at each side of the module. When there is some temperature difference across the TEG module, it converts it into electricity based on Seebeck effect. (Ju-Chan Jang et al., 2015)

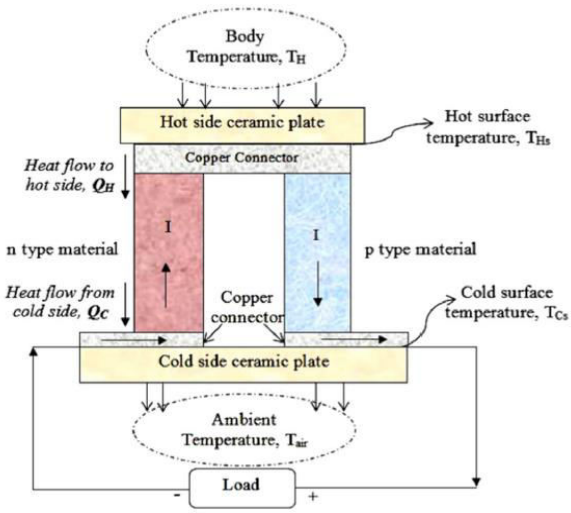

Fig. 2 Working principle of TEG (Abu Raihan Mohammad Siddique, 2017)

Heat pipe is used at the cold side of TEG as shown in the fig. 3 to maintain the temperature of the cold side as minimum as possible. The TEG is mounted on the

exhaust steam pipe of the boiler. The heat energy of high pressurized steam is used to increase the temperature of the hot side of TEG. A heat pipe as a heat transfer device is used on cold side of the TEG to decrease the temperature of cold side by improving the heat dissipation rate. The current drawn from the TEG is used for other power consuming applications such as flow meter.

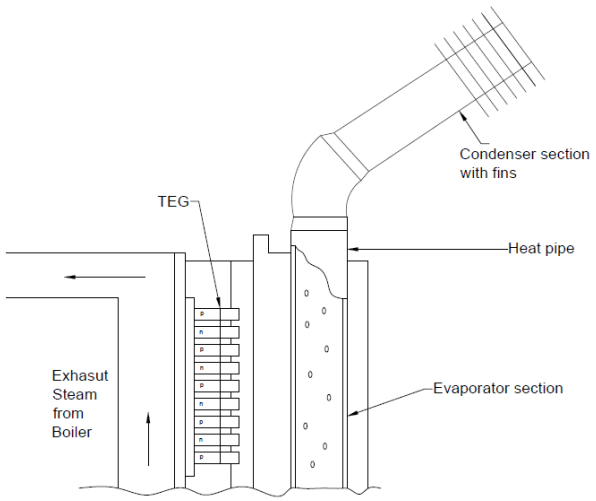

Fig. 3 Schematic of Heat pipe for TEG application

2. Literature review

M. F. Remeli et al. (2015) did experimental investigation of power generation by using the combination of heat pipes and thermo-electric generators. The experimental investigation was done on the model of power production by TEGs using heat pipes and simulated hot air. The counter flow air duct Heat Exchanger (HX) was used for experimental setup. The obtained results showedthat as the ratio of mass flow rate in upper duct to lower duct increased the overall system performance went on increasing. A higher mass flow rate ratio resulted in a higher amount of heat transfer and higher power output. The proposed system could be used for waste heat recovery from the industry where thermal energy is used in daily process. This research generated a novel concept of the power generation from waste heat using a combination of heat pipe and TEG with investigations into key performance parameters such as maximum power output and heat transfer rate.

A new method of recovering waste heat and electricity using a combination of heat pipes and TEGs was explored by Muhammad Fairuz Remeli et al. (2014). In this study, Bismuth-Telluride (Bi_2Te_3) based TEGs were sandwiched between two finned heat pipes to achieve a temperature difference across the TEG for thermoelectricity generation. To predict the performance of the system authors made a theoretical model and calculated results for different parametric conditions. The modelling results showed that the system had the capability of recovering 1.345 kW of waste heat and generating 10.39 W of electrical power using 8 installed TEGs. The proposed design was a new concept of a fully passive technique to produce power generation by using TEGs and Heat Pipes. In this study, it had numerically demonstrated that using passive devices such as heat pipe and TEGs for power generation and heat recovery is an effective method and the future of many energy systems.

Ju-Chan Jang et al. (2015) investigated the method of improving the power generation of a TEG system.In this study, a wickless loop heat pipe was used under the limited space of the exhaust gas pipeline. The concept of a loop-type heat pipe-assisted TEG was applied to hybrid vehicles. It showed a novel concept of producing the power generation by transferring heat from the source to the sinkusing a loop-type heat pipe. In this study, TEG system with a heat pipe couldgenerate a voltage output around 1.3 V in the case of 170°C hot exhaust gas. Two TEGs for a conductive block model and four Bi_2Te_3 TEGs with a heat pipe-assisted model were installed in the condenser section. Heat flows from the evaporator section to the condenser section of a loop-type heat pipe. This TEG system with a heat pipe can be placed in any location on an automobile.

F. P. Brito et al. (2013) explored the potential using TEG in combination with variable conductance heat pipes for transferring the exhaust heat to the generator with low thermal resistance and at a constant, prescribed temperature. This passive temperature control eliminated the need for bypass systems in the event of temperature overshoots. The present study assessed both theoretically and experimentally the influence of the HP temperature in the electric output of a thermoelectric generator. A small diesel engine and a generator were tested and it was found that a high HP operating temperature is only limitative for performance in the cases where low exhaust temperature and low engine power are present. In those cases it is possible to estimate an optimal HP temperature in order to maximize power output. The combined use of Seebeck modules and heat pipes was found to be highly advantageous in various ways.

Bin-Juine Huang et al. (2015) focused on the TEG consisting a loop heat pipe. The study also focused on the design match for maximum-power generation to eliminate the maximum power point tracking control. The results showed that the conversion efficiency of the TEG which was being used was low. It was found that 96W heat should be dissipated for a 4W power output from TEG. It had seen that the heat sink device with fins and fans wasconsuming about 2-3W to dissipate the heat and caused the net power generation of TEG greatly reduced. This study proposed a TEG design using loop heat pipe (LHP) to dissipate the heat to the environment because of the natural convection. The heat is transported to a large surface at a distance and dissipated without power consumption and noise. The experiments for a TEG with rated power 4W showed that the LHP performs very well with thermal resistance 0.35 K W-1, from the cold side of TEG module to the ambient.

3. Objectives

The main objectives of the current study are,

i. To design and develop the Heat Pipe of capacity 200 Watts for TEG module

ii. Experimental analysis of Heat Pipe for TEG application

Specifications of Exhaust steam, TEG module and Heat pipe are summarized in the table below.

Table 1 Specifications of Exhaust steam, TEG module and Heat pipe

Specifications	
Exhaust steam	Steam quality: Saturated steam Pressure range of steam: 1-10 gauge bar
TEG module	Module: TEG127-1.4-1.0 Dimensions: 40×40 mm Seebeck coefficient: 0.05818 V/K
Heat pipe	Capacity: 200 Watts

4. Design methodology

4.1 Selection of working fluid, wick material and heat pipe material

Distilled water is used as working fluid since the operating range of distilled water is 20°C to 210°C.
Also, composite wick structure is selected since it has better operating limitations than homogeneous wick structure. Phosphor bronzeis selected as a wick material since it is compatible with water as well as it has high thermal conductivity.
Further, Copper is selected as heat pipe material since it has high thermal conductivity as well as it is compatible with water.

4.2 Calculations for Operating Limits

The sample calculations are shown for heat pipe operating temperature = 45˚C.

At 45˚C, properties of water are,

$\rho l = 990.18\ kg/m^3$
$\rho v = 0.07\ kg/m^3$
$\mu l = 0.00059\ Ns/m^2$
$\mu v = 1.068 \times 10^{-5}\ Ns/m^2$
$\sigma = 0.0688\ N/m$
$\lambda = 2394952\ J/kg$

Configuration of Heat Pipe assembly is given below,

$L_{adiaatic} = 0.08\ m$
$L_{evaporator} = 0.07\ m$
$L_{condenser} = 0.1\ m$

O. D. of heat pipe = 0.0095 m
I. D. of heat pipe = 0.0008 m
No. of heat pipes = 4

Table 2 Wick configuration

Wick No.	Parameters
Wick 1 (adjacent to vapor side)	7087 mesh per m (180 mesh per inch), two layers, wire diameter = 0.000051 m
Wick 2 (adjacent to pipe wall)	2362 mesh per m (60 mesh per inch), two layers, wire diameter = 0.000122 m

1. Input Parameters:

N_1 = 7087 mesh per m and N_2 = 2362 mesh per m
$d_i = 8\ mm = 0.008\ m$

$d_{w1} = 0.051\ mm = 0.000051\ m$ for 180 mesh per inch
$d_{w2} = 0.122\ mm = 0.000122\ m$ for 60 mesh per inch

$$d_v = d_i - 2(t_w \times \text{Number of layers of wire mesh})$$
$$= 0.008 - 2(2 \times 0.000051 \times 2 + 2 \times 0.000122 \times 2)$$
$$= 0.006616\ m$$

$$r_{hv} = \frac{d_v}{2} = 0.003308\ m$$

$$A_v = \frac{\pi d_v^2}{4} = 3.438E-05\ m^2$$

$$L_{eff} = L_a + \frac{L_e + L_c}{2} = 0.165\ m$$

Assuming S = 1.05

$$\varepsilon_w = 1 - \frac{\pi S N d_w}{4} = 1 - \frac{\pi \times 1.05 \times 2362 \times 0.000122}{4}$$
$$= 0.7623$$

$$K = \frac{d_w^2 \varepsilon_w{}^3}{122(1-\varepsilon_w)^2} = \frac{(0.000122)^2 \times (0.7623)^3}{122 \times (1-0.7623)^2}$$
$$= 9.57E-10\ m^2$$

$$A_w = \frac{\pi(d_i^2 - d_v^2)}{4} = 1.58E-05\ m^2$$

$$r_{hw} = \frac{A_w}{\pi(d_v + d_i)} = 3.46E-04\ m$$

$$k_{eff} = \frac{k_l[(k_l + k_w) - (1-\varepsilon_w)(k_l - k_w)]}{[(k_l + k_w) + (1-\varepsilon_w)(k_l - k_w)]}$$
$$= 1.194\ W/m^oC$$

$$\therefore r_c = \frac{1}{2 \times 7087} = 7.05517E-05\ m$$

2. Output Parameters:

1) Capillary limit:-

$$\Delta P_{c,m} = \frac{2 \times \sigma}{r_c} = \frac{2 \times 0.0688}{7.05517E-05} = 1950.33\ N/m^2$$

$\Delta P_+ = \rho l \times 9.81 \times d_v \times \cos 0 = 990.18 \times 9.81 \times 0.006616 \times \cos 0 = 64.26\ N/m^2$

$\Delta P_{||} = 0\ N/m^2$

$$\Delta P_v = \frac{16\mu_v L_{eff} q}{2r_{hv}^2 A_v \rho_v \lambda} = 0.2367q \text{ N/m}^2$$

$$\Delta P_l = \frac{\mu_l L_{eff} q}{K A_w \rho_l \lambda} = 2.7171q \text{ N/m}^2$$

$$\Delta P_{cm} = \Delta P_+ + \Delta P_{||} + \Delta P_v + \Delta P_l \quad (1)$$
$$\therefore 1950.33 = 64.26 + 0 + 0.2367\,q + 2.7171\,q$$
$$\therefore q = 638.50 \text{ W}$$

2) Sonic limit:-

$$q_s = 0.474\lambda A_v (\rho_v P_v)^{1/2} = 980.87 \text{ W} \quad (2)$$

3) Entrainment limit:-

$$q_{ent} = A_v \lambda \left(\frac{\sigma \rho_v}{2r_{hw}}\right)^{1/2} = 211.03 \text{ W} \quad (3)$$

4) Boiling limit:-

Assuming the value of $r_n = 2.54 \times 10^{-7}$m.

$$q_b = \frac{2\pi L_e k_{eff} T_v}{\lambda \rho_v \ln\left(\frac{r_i}{r_v}\right)} \left(\frac{2\sigma}{r_n} - \Delta P_{cm}\right) = 3000.04 \text{ W} \quad (4)$$

5) Viscous limit:-

$$q_v = d_v^2 \lambda A_v \left(\frac{P_v \rho_v}{64\mu_v L_e}\right) = 47596.9 \text{ W} \quad (5)$$

Table 3 Operating limits at operating temp. 45˚C

Limit	Value
Capillary	638.50 W
Sonic	980.87 W
Entrainment	211.03 W
Boiling	3000.04 W
Viscous	47596.9 W

4.3 Calculations for Effectiveness of HP assembly

The sample calculations are shown for temperature of air at condenser section = 40˚C and Gauge Pressure in steam exhaust pipe = 1 bar

The theoretical effectiveness of Heat Pipe is calculated by Effectiveness-NTU as available in open literature. Since in this study heat exchanger is present only at the condenser section of heat pipe, effectiveness of condenser section will be equal to effectiveness of Heat Pipe assembly.
Since the vapor inside a heat pipe is almost at constant temperature, its capacity rate,C_l, becomes by definition, equal to infinity and as a result $\frac{C_c}{C_l} = 0$.

Therefore the effectiveness of single row condenser section is given by,

$$\varepsilon_{c_1} = 1 - \exp^{(-NTU)_c} \quad (6)$$

Since $C_c < C_l$, therefore

$$NTU_c = \frac{(UA)_c}{C_c} \quad (7)$$

Where,

$$UA = \frac{1}{R_{total}} \quad (8)$$

$$R_{total} = R_{conv_0} + R_{sca} + R_{cond_p} + R_{cond_w}$$

Where 'R_{total}' is the total resistance, 'R_{conv}' is the convection resistance, 'R_{sca}' is resistance due to scaling and 'R_{cond}' is the conduction resistance.
The inside resistance due to boiling and condensation are neglected as their values are comparatively very small due to larger values of heat transfer coefficient. In addition, absence of fouling resistance and perfect contact between pipe and fin are assumed while determining the total resistance.

Fin details are given in the table below,

Table 4 Details of Fins

FPM	δ_f, m	l_f, m	k_f, W/mK
310	0.0004	0.000625	177

Properties of air is given below,

$\rho = 1.1274$ kg/m^3
$\mu = 1.9346 \times 10^{-5}$ Ns/m^2
$C_p = 1006.07\ J/kgK$
$k = 0.0272\ W/mK$
$Pr = 0.7154$

Calculations for resistances are given below,

1) Convection resistance:-

Total area can be calculated as,
$A = A_f + A_p$
Where,
A_f is the fin area and A_p is the pipe area.
$A_p = \pi d_o l[1 - (\delta_f \times FPM)]N_t$
$A_p = 0.0104$ m^2
$A_f = L \times FPM \times 2 \times [H \times S_l \times N_r - (N_t \times N_r \times \pi \times r_o^2)]$
$A_f = 0.0661$ m^2
So, $A = 0.0765$ m^2

Reynolds number is calculated by,
$$Re = \frac{\rho V D_h}{\mu}$$

Where, $D_h = 2s$
Where, sis the spacing between adjacent fins and calculated as,
$$s = \left(\frac{1}{FPM}\right) - \delta_f = 0.0028 \text{ m}$$

So, $D_h = 5.65E - 03$ m

$Re = 343.54$

The Colburn factor 'j' is determined by,

$$j = 0.0014 + 0.2618Re^{-0.4}\left(\frac{A}{N_t N_r \pi d_o H}\right)^{-0.15}$$

So, $j = 0.0202$

Also, $j = St \times Pr^{\frac{2}{3}}$

We get, $St = 0.0253$

Now, $Nu = St \times Re \times Pr$

$Nu = 6.2264$

Also,

$$Nu = \frac{hD_h}{k}$$

So, $h = 29.96\ W/m^2K$

Now, the fin parameter 'm' is given by,

$$m = \sqrt{\frac{2h}{k_f \delta_f}} = \sqrt{\frac{2 \times 29.96}{177 \times 0.0004}}$$

So, $m = 29.09$

The fin efficiency 'η_f' is given by,

$$\eta_f = \frac{\tanh(ml_f)}{ml_f} = \frac{\tanh(29.09 \times 0.000625)}{29.09 \times 0.000625}$$

$\eta_f = 0.9998$

The overall efficiency 'η_o' is given by,

$$\eta_o = 1 - \frac{A_f}{A}(1 - \eta_f) = 0.9999$$

Now, convection resistance is given by,

$$R_{conv_o} = \frac{1}{\eta_o hA} = \frac{1}{0.9999 \times 29.96 \times 0.0765} \quad (9)$$

$R_{conv_o} = 0.4357\ K/W$

2) Conduction resistance of pipe:

$$R_{cond_p} = \frac{\ln\left({}^{r_o}/_{r_i}\right)}{2\pi l k_p N_t} \quad (10)$$

$R_{cond_p} = 0.00018\ K/W$

3) Conduction resistance of wick structure:

$$R_{cond_w} = \frac{\ln\left({}^{r_i}/_{r_v}\right)}{2\pi l k_{eff} N_t} \quad (11)$$

$R_{cond_w} = 0.0633\ K/W$

So,

$$R_{total} = R_{conv_o} + R_{sca} + R_{cond_p} + R_{cond_w} \quad (12)$$

$R_{total} = 0.4992\ K/W$

Again, from eq. (8)

$$UA = \frac{1}{R_{total}} = \frac{1}{0.4992}$$

$UA = 2.0029\ W/K$

Here, velocity of air (V = 1.043 m/s) is determined from cfm of the fan. So, from eq. (7) we can calculate

$$NTU_c = \frac{(UA)_c}{C_c}$$

$$= \frac{2.0029}{(1.1274 \times 0.1 \times 0.09 \times 1.043 \times 1006.07)}$$

$NTU_c = 0.1881$

Effectiveness of condenser is calculated from eq. (6),

$\varepsilon_c = 1 - \exp^{(-NTU)_c}$

So, $\varepsilon_c = 0.1714$

4.3 Calculations for Voltage output

Now, voltage output of TEG is calculated from effectiveness of condenser section of HPHX.

Effectiveness can be given by,

$$\varepsilon_c = 0.1714 = \frac{(\Delta T)act}{(\Delta T)max} \quad (13)$$

$(\Delta T)\ max = Tfin - Tair$

At 1 bar, Tfin = 58.42 °C

$(\Delta T)max = 58.42 - 40 = 18.42$ °C

So, $(\Delta T)act = 3.1587$ °C

Again heat gained by the air at condenser section is given by,

$$Q_{air} = Q_c = m \times Cp \times (\Delta T)act \quad (14)$$

$Q_{air} = Q_c = 33.63\ W$

Since, $Q_c = Q_e$

$Q_e = 33.63\ W$

Heat gained by water at evaporator section can be given by,

$$Q_e = \frac{(\Delta T)_{TEG}}{R_{TEG}} \quad (15)$$

Since, $R_{TEG} = 1.0576\ K/W$

So, $(\Delta T)_{TEG} = 35.56$ °C

Since, Seebeck coefficient (α) of TEG = 0.05818 V/K

Voltage output from TEG is given by,

$$V = \alpha \times (\Delta T)_{TEG} \quad (16)$$

$V = 0.05818 \times 35.56$

$V = 2.0692$ Volts

5. Experimental setup

Heat pipe assembly with 4 number of heat pipes is shown in the Fig. 4. Heat pipe is made up of Copper whereas fins and block are made up of Aluminium. The dimensions of block is 0.09m×0.04m×0.012m.

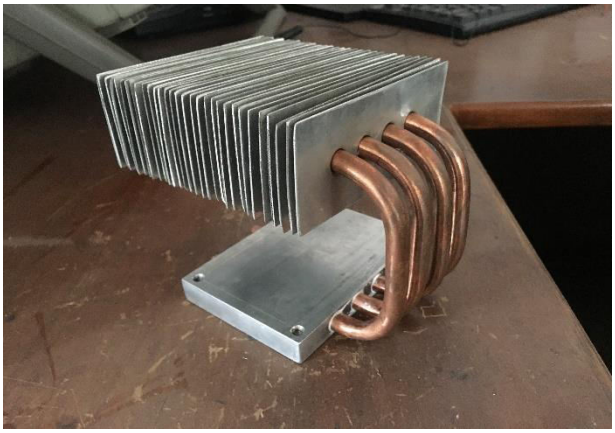

Fig. 4 Heat Pipe assembly (Experimental setup)

TEG is mounted on the exhaust steam pipe of the boiler with the help of metal block. Heat Pipe assembly is attached to the cold side of the TEG. 3-D drawing of the above assembly is shown in the Fig. 5. Figure 6 shows the actual test rig with Heat Pipe assembly attached on the cold side of TEG.

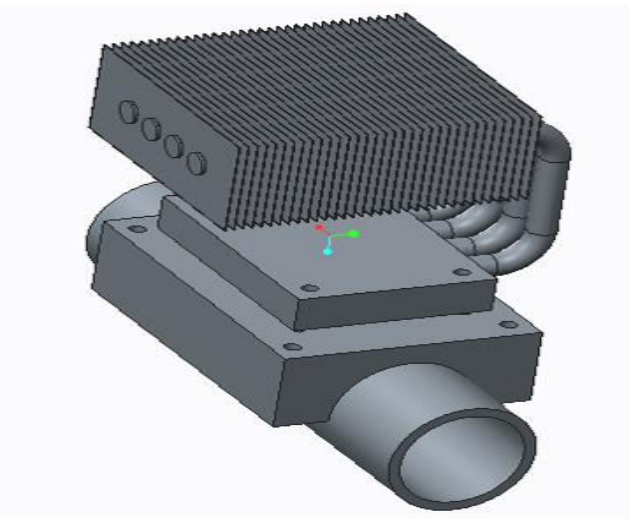

Fig. 5 3-D module of Heat Pipe assembly for TEG

Saturated exhaust steam flows though the steam pipe. Metal block is attached on steam pipe as shown in Fig. 6. Readings are taken at the interval of 1 gauge bar in the range of 1-10 gauge bar. Pressure is controlled by pressure relief valve (PRV). As the pressure is increased the temperature difference across TEG increases and so the voltage output. The voltage output is measured by multimeter.

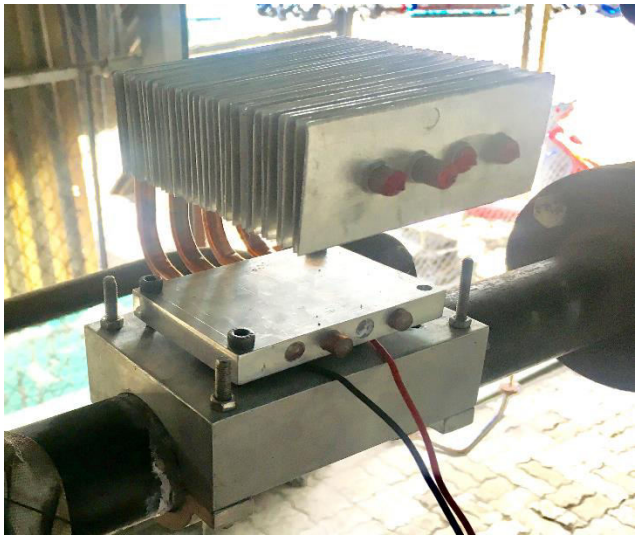

Fig. 6 Test rig with experimental setup

6. Results and discussion

Readings are taken for the open circuit voltage and closed circuit voltage. In case of the closed circuit voltage, resistance of 24 Ω is attached as a load. Two sets of readings are taken as the first for Heat Pipe heat sink with fan and the second is same heat sink without fan (i.e. forced convection and natural convection). The fan with specifications of 22.1 CFM, 12 V DC, 0.18 A is used for the experiment.
Readings for both sets are summarized in the tables below.

Table 5 Readings for Heat Pipe heat sink with fan

Pressure (bar)	V_{oc} (Volts)	V_{load} (Volts)	Power o/p for R = 24 Ω	
			For V_{oc}	For V_{load}
1	1.958	1.861	0.15974	0.144305
2	2.291	2.075	0.218695	0.179401
3	2.504	2.263	0.261251	0.213382
4	2.796	2.59	0.325734	0.279504
5	2.895	2.638	0.349209	0.28996
6	3.01	2.687	0.377504	0.300832
7	3.18	2.93	0.42135	0.357704
8	3.391	2.998	0.47912	0.3745
9	3.51	3.101	0.513338	0.400675
10	3.657	3.13	0.557235	0.408204

Readings at the interval of 1 bar for Heat Pipe heat sink without fan is given in the Table 6.

Table 6 Readings for Heat Pipe heat sink without fan

Pressure (bar)	V_{oc} (Volts)	V_{load} (Volts)	Power o/p for R = 24 Ω	
			For V_{oc}	For V_{load}
1	1.696	1.413	0.11985	0.08319
2	1.882	1.678	0.14758	0.11732
3	1.907	1.702	0.151527	0.120700
4	2.108	1.925	0.185152	0.154401
5	2.288	2.016	0.218122	0.169344
6	2.317	2.108	0.223687	0.185152
7	2.405	2.193	0.241001	0.200385
8	2.576	2.23	0.27649	0.207204
9	2.601	2.298	0.281883	0.220033

10	2.734	2.417	0.311448	0.243412

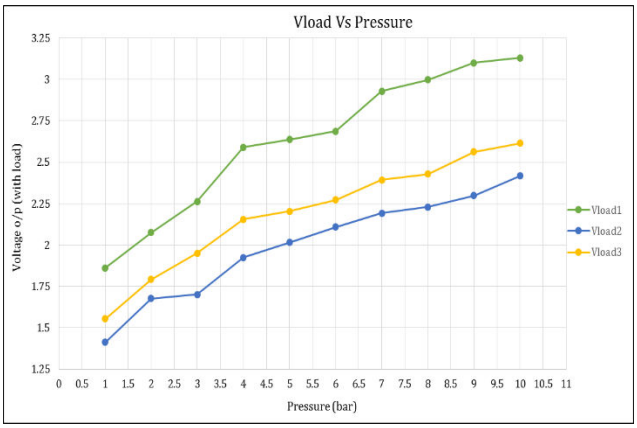

Fig. 7 Comparison between Old heat sink and Heat Pipe heat sink based on closed circuit Voltage o/p

The obtained results are compared with the Old heat sink which was being used for the same application. Figure 7 gives the comparison between Old heat sink and Heat Pipe heat sink based on closed circuit voltage output. V_{load1}, V_{load2} and V_{load3} indicate the closed circuit voltage output for Heat Pipe heat sink with fan, without fan and Old heat sink respectively. From the graph, it can be stated that the performance of Heat Pipe heat sink with fan is better than that of without fan and Old heat sink. When Heat Pipe heat sink with fan is compared with the Old heat sink then the overall increase in voltage output is around 20%. The maximum increase in voltage is at 8 gauge bar and is around 24%. Whereas, when it is compared with Heat Pipe heat sink without fan, the decrease in voltage output is around 8%.

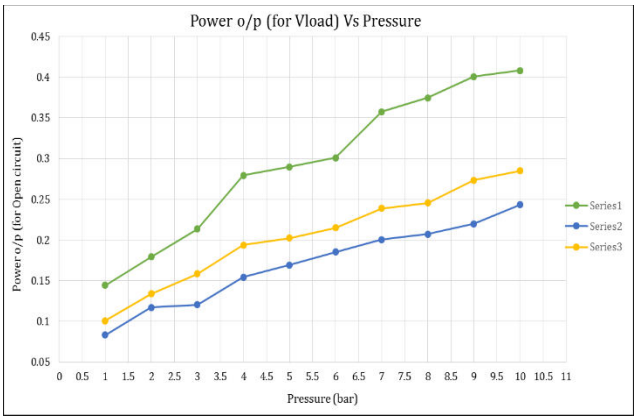

Fig. 8 Comparison between old heat sink and Heat Pipe heat sink based on Power o/p for closed circuit voltage

In another graph comparison is done on the basis of Power output for closed circuit voltage. Series1, Series2 and Series3 indicate Power output for closed circuit voltage for Heat Pipe heat sink with fan, without fan and Old heat sink respectively. When the Heat Pipe heat sink with fan is used then the overall increase in the power output is around 43%. The maximum increase can be seen at 8 bar and is equal to 53%. Whereas if same heat sink is used without fan then the overall decrement in the Power output is equal to 17%.

Table 7 Values of ΔT across TEG for Old heat sink and Heat Pipe heat sink

Pressure (bar)	$\Delta T1_{TEG}$(°C)	$\Delta T2_{TEG}$(°C)	$\Delta T3_{TEG}$(°C)
1	33.65418	29.15091	29.21966
2	39.37779	32.34789	33.99794
3	43.03884	32.77759	36.43864
4	48.05775	36.23238	40.70127
5	49.75937	39.32623	41.42317
6	51.73599	39.82468	43.57167
7	54.65796	41.33723	45.37642
8	58.28463	44.27638	46.5452
9	60.33001	44.70608	49.03747
10	62.85665	46.99209	50.10313

Another set of reading is taken for the theoretical ΔT across TEG for open circuit voltage for both heat sinks. Values of ΔT are calculated over the pressure range 0f 1-10 gauge bar at the constant time interval of 1 gauge bar. ΔT1, ΔT2 and ΔT3 denote ΔT across TEG for Heat Pipe heat sink with fan, without fan and Old heat sink respectively. From the graph it is calculated that the overall increase in the ΔT across the TEG for Heat Pipe heat sink with fan is around 26%. Whereas when fan is not used with same heat sink then the performance is decreased. The overall decrement in ΔT across the TEG is 6% which is very less as compared to the increase in the ΔT when the fan is used. Also maximum increase can be seen at 10 bar and is approximately equal to 20%.

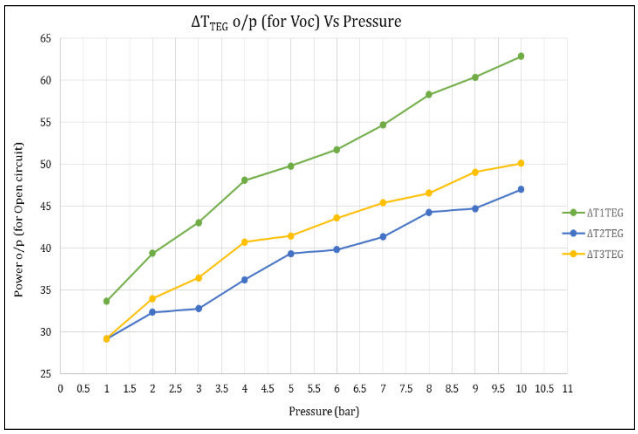

Fig. 9 Comparison between old heat sink and Heat Pipe heat sink based on ΔT across TEG

Conclusions

This study investigates the performance of Heat Pipe assembly for Thermo-Electric Generation application. The results are plotted for Heat Pipe assembly with fan, without fan and Old heat sink.

1) Voltage output is directly proportional to the temperature difference across TEG. It is increased by 26% in the case of Heat Pipe assembly with fan.

2) The power output for closed circuit voltage is increased by 43% when Heat Pipe assembly is used with fan.
3) It is concluded from the experimental investigations that by using the fan for heat dissipation at condenser section improves performance of Heat pipe assembly.
4) It is recommended to use this Heat Pipe assembly where the air velocity is better than 1 m/s at condenser section and for wide range of ambient temperature.

Nomenclature

A_f — Area of Fin, m^2
A_p —Area of Pipe, m^2
Av – Vapor area, m^2
A —Total Area of pipe, m^2
C_c — Heat capactiy of fluid of condenser side, W/K
D_h – Hydraulic diameter, m
FPM – Fins per Meter
L_a – Length of adiabatic section, m
L_c – Length of condenser section, m
L_e–Length of evaporator, m
N_1 – Mesh number of wick 1, mesh per m
N_2 – Mesh number of wick 2, mesh per m
N_r – Number of rows
N_t – Number of tubes
NTU – Number of Units Transfer
Nu – Nusselt number
Pr – Prandtl number
Re — Reynold Number
S_l – Longitudinal distance between heat pipes, m
S_t – Stanton number
Q_{air} – Heat absorbed by air, W
d_i– Inner diameter of heat pipe, m
d_o – Outer diameter of heat pipe, m
d_{w1} – Diameter of wick 1, m
d_{w2} – Diameter of wick 2, m
d_v – Diameter of vapor area, m
h – Heat transfer coefficient, W/m^2K
j — Colburn factor
k_{eff} – Effective thermal conductivity of liquid-saturated wick structure, W/mK
k_f – Thermal conductivity of fin, W/mK
kp – Thermal conductivity of pipe, W/mK
k_w – Thermal conductivity of wick, W/mK
$\dot{m}$ — Mass flow rate of Air, kg/sec
r_{hv} – Hydraulic radius for vapor area, m
r_i — Inner radius of pipe, m
r_o — Outter radius of pipe, m
ρ_l – Liquid density, kg/m^3
ρ_v – Vapor density, kg/m^3
μ_v – Dynamic viscosity of vapor, Ns/m^2
μ_l – Dynamic viscosity of liquid, Ns/m^2
δ_f – Thickness of fin, m
ε_c — Effectiveness of condenser side of Heat Pipe

References

H. Jouhara, A. Chauhan, T. Nannou, S. Almahmoud, B. Delpech, L. C. Wrobel, (2017), Heat Pipe based systems – Advances and applications, *Energy*, 17

Nikhil S. Chougule, Tushar S. Jadhav, Mandar M. Lele, (2016), A review on Heat Pipe for Air Conditioning applications, *International Journal of Current Engineering and Technology,* 4, 204-207

Ju-Chan Jang, Ri-Guang Chi, Seok-Ho Rhi, Kye-Bock Lee, Hyun-Chang Hwang, Ji-Su Lee, Wook-Hyun Lee, (2015), Heat Pipe-assisted thermoelectric power generation technology for waste heat recovery, *Journal of Electronics materials*

Abu Raihan Mohammad Siddique, Shohel Mahmud, Bill Van Heyst, (2017), A review of the state of the science on wearable thermoelectric power generators (TEGs) and their existing challenges, *Renewable and Sustainable Energy Reviews,* 73, 730-744

M. F. Remeli, L. Kiatbodin, B. Singh, K. Verojporn, A. Date, A. Akbarzadeh, (2015), Power generation from waste heat using Heat Pipe and Thermoelectric Generator, *The 7thInternational Conference on Applied Energy – ICAE*, 75, 645-650

Muhammad Fairuz Remeli, Lippong Tan, Abhijit Date, Baljit Singh, Aliakbar Akbarzadeh, (2014), Simultaneous power generation and heat recovery using a heat pipe assisted thermoelectric generator system, *Energy Conversion and Management,* , 91, 110-119

F. P. Brito, J. Martins, L M Goncalves, D Souza, N Autunes, (2013), Influence of heat pipe operating temperature on exhaust heat thermoelectric generation, *SAE International Journal of Passenger Cars – Mechanical Systems,* 2, 652-664

Bin-Juine Huang, Po-Chien Hsu, Rung-Je Tsai, Muhammad Mustafa Hussain, (2015), A thermoelectric generator using loop heat pipe and design match for maximum power generation, *Applied Thermal Energy,* 15

Steven P. Benn, Leonard M. Poplaski, Amir Faghri, Theodore L. Bergman, (2016), Analysis of thermosyphon/heat pipe integration for feasibility of dry cooling for thermoelectric power generation, *Applied Thermal Engineering,* 16

Muhammad Fairuz Remeli, Lippong Tan, Abhijit Date, Baljit Singh, Aliakbar Akbarzadeh, (2014), Power generation and Heat recovery using heat pipe thermoelectric generator (TEG),*Australian Solar Energy Society Conference,* 52

Jorge Martins, Francisco P. Brito, L. M. Goncalves, Joaquim Antunes, (2011), Thermoelectric exhaust energy recovery with temperature control through heat pipes, *SAE International Journal*