Abstract—In an internal combustion engine; efficiency of engine is around 30%, roughly 30% of the fuel energy is wasted in exhaust gases, and 30% in cooling water and 10% are unaccountable losses. Efforts are made to catch this 30% energy of exhaust gases. If this waste heat energy is tapped and converted into usable energy, the overall efficiency of an engine can be improved. Thermoelectric modules which are solid state devices that are used to convert thermal energy to electrical energy from a temperature gradient and it works on principle of Seebeck effect.

This paper demonstrates the potential of thermoelectric generation. The detailed experimental work was carried out to study the performance of thermoelectric generators under various engine speeds. A hot side heat exchanger as well as cold side heat sink was designed and tested on a 3 cylinder, 4 stroke, Maruti 800cc SI engine. Two Thermoelectric modules of bismuth telluride (Bi$_2$Te$_3$) were selected according to the temperature difference between exhaust gases side and the engine coolant side and placed (Just behind catalytic converter) A rectangular heat exchanger was fabricated and the thermo electric modules were placed on the heat exchanger for performance analysis. The study showed that energy can be tapped efficiently from the engine exhaust also in near future thermoelectric generators can reduce the size of the alternator or eliminate them in automobiles improving efficiency of engine.

Keywords— Thermoelectric generator, waste heat recovery, engine exhaust, hot side heat exchanger, cold side heat sink, insulation, load

I. INTRODUCTION

The "Energy Crisis" has become a major challenge in front of engineers across the globe due to rapidly increasing demands and consumption of energy. For almost two hundred years, the main energy resource has been fossil fuel and will continue to supply much of the energy for the next two and half decades. Worldwide oil consumption is expected to rise from 80 million barrels per day in 2003 to 98 million barrels per day in 2015 and then to 118 million barrels per day in 2030. [1]
The electric load of a vehicle is increasing due to improvements in driving performance and comfort. In order to satisfy the increasing demands of electricity in modern vehicles, bigger and heavy alternators are coupled to engines. These alternators which operate at an efficiency of 55 to 65% consume around 5% of the rated shaft power. And ultimately affects the fuel economy of vehicle. [6]

One potential solution is the usage of the exhaust waste heat of combustion engines. This is possible by the waste heat recovery using thermoelectric generator. A thermoelectric generator converts the temperature gradient into useful voltage that can used for providing power for auxiliary systems such as air conditioner and minor car electronics. Even it can reduce the size of the alternator that consumes shaft power. If approximately 6% of exhaust heat could be converted into electrical power, it will save approximately same quantity of driving energy. It will be possible to reduce fuel consumption around 10%; hence AETEG system can be profitable in the automobile industry.

II. PROPOSED SOLUTION

As shown in the figure 5, the proposed system consists of one hot side heat exchanger and one cold side heat exchanger. Between the two heat exchangers the thermoelectric modules (TEG) are placed. The exhaust gas from engine passes through hot side heat exchanger and cooling water from radiator passes through cold side heat sink. According to the principle of Seebeck effect, thermoelectric modules convert the heat into useful electricity.

III. TEG MATERIAL SELECTION

The driving principle behind thermoelectric generation is the known as the Seebeck effect. Whenever a temperature gradient is applied to a thermoelectric material, specifically metals or semiconductors, the heat passing through is conducted by the same particles that carry charge. The movement of charge produces a voltage. The junctions of the different conductors are kept at different temperatures which cause an open circuit electromotive force (e.m.f) to develop as follows:

\[ V_{OC} = \alpha (T_H - T_C) \]

Where \( \alpha \) is the difference in Seebeck coefficient of two leg materials and has the units of V/K, and \( T_H \) and \( T_C \) are the hot and cold side absolute temperatures both measured in Kelvin. A German Physicist, Thomas Johann Seebeck, discovered this effect in the early 1800s.

An important unit less constant to evaluate the performance of thermoelectric materials is the thermoelectric figure of merit, ZT. It describes the effectiveness of a specific thermoelectric material in terms of its electrical and thermal material properties. The figure of merit is expressed as:

\[ ZT = \frac{\alpha^2 \sigma}{\lambda} \]

ZT for materials has remained below 1 for decades, but in recent years, ZT of new materials has reached values greater than 2.

*Bismuth Telluride (Bi\textsubscript{2}Te\textsubscript{3})*:
The maximum value of figure of merit, 

\[ Z_{max} = 3 \times 10^{-3} \text{ K}^{-1} \]

The optimum value of the resistance ratio, 

\[ M_0 = \sqrt{1 + Z (\frac{T_1 + T_0}{2})} \]

\( T_1 \) = temperature of the source (K)

\( T_0 \) = temperature of the sink (K)

\( T_1 = 400 \text{ K} \quad T_0 = 315 \text{ K} \)

\( M_0 = 1.4396 \)

The maximum thermal efficiency is given by,

\[ \eta_{th,max} = \sqrt{\left(\frac{T_1 - T_0}{T_1}\right) \times (M_0 - 1)} \]

\[ \eta_{th,max} = 0.083130 = 8.313\% \]
Bi-Te is one of the available materials with highest value of $\alpha$. Also the position of TEG system is just behind the catalyst converter ($^{1}$) (400$^\circ$C and 200$^\circ$C) All the TEGs designed to be mounted in this position are based on bismuth telluride alloys, specifically Hi-Z commercial modules. It minimizes the amount of heat transfer surface required. This decreases the pressure drop across the generator and results in a lower back pressure. Hence we have selected Bismuth Telluride as TEG material.

IV. DESIGN OF HOT SIDE HEAT EXCHANGER

Sizing up the heat exchanger is based on the size, orientation, and number of modules. Because the modules are assumed to be square as they often are, length and width differ by the number of modules defined by flow orientation. $N_{mod,par}$ (no. of modules in parallel) and $N_{mod,ser}$ (no. of modules in series) exist to aid in developing the orientation of the modules in a zone $w_{mod,zone}$ is the width of all the modules in a zone if they were side by side. $L_{mod,zone}$ is the length of all of the modules in a zone if they were directly adjacent to each other. [2]

Module size: $40\times40\times3$ mm

\[ L_{mod,zone} = w_{mod} \times N_{mod,ser} \]
\[ = 40 \times 2 = 80 \text{ mm} \]

\[ W_{mod,zone} = w_{mod} \times N_{mod,par} \]
\[ = 40 \times 1 = 40 \text{ mm} \]

$A_{mod,zone}$ is the surface area of all the modules in the zone and $\beta_{lw}$ is the ratio of the length of all the modules in the zone to the width of all the modules in the zone.

\[ A_{mod,zone} = L_{mod,zone} \times W_{mod,zone} \]
\[ = 80 \times 40 = 3200 \text{ sq.mm} \]

\[ \beta_{lw} = \frac{L_{mod,zone}}{w_{mod,zone}} = \frac{80}{40} = 2 \]

$A_{zone}$ is the surface area of a zone and $\gamma$ is the user defined ratio for zone area to modules in a zone area. $\gamma$ is always greater than or equal to one by its definition. We take it as 2. $A_{ins}$ is the area of the insulation.

\[ A_{zone} = \gamma \times A_{mod,zone} \]
\[ = 2 \times 3200 = 6400 \text{ sq.mm} \]

\[ A_{ins} = A_{zone} - A_{mod,zone} \]
\[ = 6400 - 3200 = 3200 \text{ sq.mm} \]

$L_z$ is the length of an entire zone and $w_z$ is the width of a zone.

\[ L_z = \sqrt{B_{lw} \times A_{zone}} = 120 \text{ mm} \]

\[ W_z = \sqrt{\frac{1}{B_{lw}} \times A_{zone}} = 60 \text{ mm} \]

Therefore final dimension of Hot side heat sink = $120 \times 60$ mm

A. Design of Rectangular Straight Fins

Rectangular straight fins are very common fin geometry because of their simplicity to manufacturer. Parameters include the number of fins ($N_f$), the thickness of an individual fin ($t_f$), the length an individual fin protrudes from its base ($L_f$), and the thickness of the base ($t_b$), the conductive coefficient ($K_{fin}$). [2]

\[ N_f = 8 \]
\[ N_{ch} = N_f - 1 = 7 \]

Thickness of an individual fin ($t_f$) = 2 mm

The length of an individual fin protrudes from its base ($L_f$) = 26 mm

Thickness of the base ($t_b$) = 7 mm

Calculations:

1) Pitch of Fin ($P_f$):

\[ P_f = \frac{W_z - T_f}{N_{ch}} = \frac{60 - 2}{7} = 8.28 \text{ mm} \]

2) Spacing between fins ($S_f$)

\[ S_f = P_f - t_f = 8.28 - 2 = 6.28 \text{ mm} \]

3) Wetted Perimeter, ($P_{wet}$):

\[ P_{wet} \] is the perimeter of a flow path or one channel created by the fins.

\[ P_{wet} = 2L_f + 2S_f \]

\[ = (2 \times 26) + (2 \times 6.28) = 64.57 \text{ mm} \]

4) Hydraulic diameter, ($D_h$):

It is an artificial diameter representing the channel in which flow travels through.

\[ D_h = \frac{4S_f \times P_f}{P_{wet}} \]
An entrance area, \( A_{\text{ent}} \):
\[
A_{\text{ent}} = S_f \times L_f \times N_{ch}
\]
\[
= 26 \times 6.28 \times 7 = 1142.96 \text{mm}^2
\]

6) The characteristic length of the fin, \( L_{f,\text{char}} \):
\[
L_{f,\text{char}} = \frac{L_f + \frac{T_f}{2}}{2}
\]
\[
= 26 + \frac{27}{2} = 27 \text{ mm}
\]

7) The perimeter of the face of the fin, \( P_{\text{face}} \):
\[
P_{\text{face}} = 2T_f + 2L_x
\]
\[
= (2 \times 2) + (2 \times 120) = 244 \text{ mm}
\]

8) The cross sectional area of the fin, \( A_c \):
\[
A_c = T_f \times L_x = 240 \text{ mm}^2
\]

9) The total surface area of all the fins, \( A_{f,\text{surf}} \):
\[
A_{f,\text{surf}} = 2AXch \times L_{f,\text{char}} \times L_x
\]
\[
= 2 \times 7 \times 27 \times 120 = 45360 \text{ mm}^2
\]

10) The total area of the base, \( A_{b,\text{surf}} \):
\[
A_{b,\text{surf}} = A_{\text{zone}} - (A_c \times N_f)
\]
\[
= (6400) - (240 \times 8) = 4480 \text{ mm}^2
\]

11) Total effective surface area, \( A_{\text{tot,surf}} \):

Total effective surface area, \( A_{\text{tot,surf}} \), is the area which fluid flow occurs and convective heat transfer is present
\[
A_{\text{tot,surf}} = A_{f,\text{surf}} + A_{b,\text{surf}}
\]
\[
45360 + 4480 = 49840 \text{ mm}^2
\]

B. Sample Calculations

1) Fuel intake
\[
(m_f) = \frac{\text{fuel consumed in m}^3 \times \text{density of fuel}}{\text{time required (t)}}
\]
\[
= \frac{100 \times 10^{-6} \times 740}{49}
\]
\[
= 1.51 \times 10^{-3} \text{ kg/s}
\]

2) Mass flow rate of air \( (m_a) \):
\[
= C_d \times A \times \sqrt{2g \times H_w} \times (g_w / g_s) = 0.6 \times \left( \frac{R}{M} \times 0.035 \right) \times \\
\sqrt{2 \times 9.81 \times 0.07 \times (1000 / 1.16)} = 19.86 \times 10^{-3} \text{ kg/sec}
\]

3) Exhaust mass flow rate \( (m_{ex}) \):
\[
= m_a + m_f
\]
\[
= 1.51 \times 10^{-3} + 19.86 \times 10^{-3} \text{ kg/s}
\]
\[
= 21.37 \times 10^{-3} \text{ kg/s}
\]

Now that the fins have been designed, their performance needs to be evaluated. Reynolds number is the ratio of inertial forces to viscous forces.

4) Reynolds Number \( Re \):
\[
Re = \frac{(4 \times m_{ex})}{(\mu \times P_{wet} \times L_f)}
\]
\[
= \frac{(4 \times 21.37 \times 10^{-3})}{(0.362 \times 10^{-4} \times 64.57 \times 10^{-3} \times 7)}
\]
\[
= 5224.28
\]

5) Nusselt Number \( Nu \):

Prandtl number, \( Pr \), is the ratio of the momentum and thermal diffusivities. Nusselt number, \( Nu \), is the ratio of convection to pure conduction heat transfer.

\[
Nu = 0.664 \times (Re)^{1/2} \times (Pr)^{1/3}
\]
\[
= 0.664 \times (5224.28)^{1/2} \times (0.7055)^{1/3} \text{ (Pr from properties of exhaust gas at 502 deg C [8])}
\]
\[
= 42.77
\]

Having solved the Nusselt number for either laminar or turbulent flow depending on the conditions, it is now possible to determine the convective coefficient,

6) \( h \) = \[ \frac{NuK}{L} \]
\[
= \frac{(42.77 \times 56.31 \times 10^{-3})}{8.09 \times 10^{-3}} = 238.25 \text{ W/m}^2\text{K}
\]

7) \( m = \frac{(6 \times P_{\text{face}})^{1/2}}{(K_{\text{fin}} \times A_c)} \]
\[
= \frac{(238.24 \times 244 \times 10^{-3})^{1/2}}{(200 \times 240 \times 10^{-6})}
\]
\[
= 34.80
\]

8) Efficiency of fin:

\( N_f \) is the efficiency of one fin with the previously calculated fin parameters and provided inputs.
\[
N_f = \frac{\text{tanh}(m \times L_{f,\text{char}})}{(m \times L_{f,\text{char}})}
\]
\[
= \frac{\text{tanh}(34.80 \times 27)}{34.80 \times 27}
\]
\[
= 0.7822 \times 78.22\%
9) Overall efficiency of fin:

The overall surface efficiency, $\eta_0$, is the efficiency of the array of fins as well as the base surface to which the fins are attached

$$\eta_0 = 1 - \frac{A_{surf}}{A_{otsurf}}(1 - \eta_f)$$

$$= 1 - \frac{4.5360}{49840}(1 - 0.7822) = 0.8018$$

$\approx 80.18\%$

![Figure 6: Actual View of Hot Side Heat Exchanger](image)

C. Calculation Table

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Engine Speed RPM</th>
<th>Air intake ($m_a$) $\times 10^{-3}$</th>
<th>Fuel intake ($m_f$) $\times 10^{-3}$</th>
<th>$m_{exhaust}$ $\times 10^{-3}$</th>
</tr>
</thead>
<tbody>
<tr>
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<td>2250</td>
<td>15.01</td>
<td>1.1212</td>
<td>16.136</td>
</tr>
<tr>
<td>2</td>
<td>2850</td>
<td>18.54</td>
<td>1.3962</td>
<td>19.938</td>
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<tr>
<td>3</td>
<td>3200</td>
<td>20.69</td>
<td>1.5744</td>
<td>22.271</td>
</tr>
<tr>
<td>4</td>
<td>3606</td>
<td>23.01</td>
<td>1.7209</td>
<td>24.738</td>
</tr>
<tr>
<td>5</td>
<td>3970</td>
<td>24.78</td>
<td>1.9473</td>
<td>26.733</td>
</tr>
</tbody>
</table>

$m_{exhaust} = m_a + m_f$

![Figure 7: Front View of Double Stacked Type Heat Sink For Cold Side](image)

V. DESIGN OF COLD SIDE HEAT SINK

The basic requirement of cold side heat sink was

1. Heat sink should flow with full of water i.e. no air gap should get created.
2. Length of cold side heat sink should be larger than hot side heat exchanger as cooling should be effective.

From various permutations and combinations, we selected stacked type heat sink for cold side.

A. Design of Cold Side Heat Sink Stacks

Number of fins ($N_f$) = 11
Number of channels ($N_{ch}$) = $N_f$ - 1 = 10 Thickness of an individual fin ($t_f$) = 2mm

The length of heat sink = $K \times$ (length of hot side heat exchanger) = 1.5 $\times$ 120 = 180mm

The width of heat sink = 62mm

The length of an individual fin protrudes from its base ($l_f$) = 14mm

Thickness of the base ($t_b$) = 5mm

Number of stacks = 2

Calculations:

1) Pitch of Fin ($P_f$):

$$P_f = \frac{W_x - T_f}{N_{ch}} = \frac{62 - 2}{10} = 6 \text{mm}$$

2) Spacing between fins ($S_f$)

$$S_f = P_f - t_f = 6 - 2 = 4 \text{mm}$$

3) Wetted Perimeter, $P_{wet}$:

$$P_{wet} = 2L_f + 2S_f$$

$= (2 \times 14) + (2 \times 4) = 36 \text{mm}$

4) Hydraulic diameter, $D_h$:

$$D_h = \frac{4 \times L_f \times S_f}{4 \times L_f \times S_f + \frac{4 \times L_f \times S_f}{36}} = 6.22 \text{mm}$$

5) An entrance area, $A_{ent}$:

$$A_{ent} = S_f \times L_f \times N_{ch} = 14 \times 4 \times 10$$

$= 560 \text{mm}^2$
6) The characteristic length of the fin, $L_{f,\text{char}}$:

$$L_{f,\text{char}} = L_f \times \frac{h}{2} = 14 + \frac{z}{2}$$

= 15 mm

7) The perimeter of the face of the fin, $P_{\text{face}}$:

$$P_{\text{face}} = 2T_f + 2L_z$$

= (2x2) + (2x180) = 364 mm

8) The cross sectional area of the fin, $A_c$:

$$A_c = T_f \times L_z = 360 \text{ mm}^2$$

9) The total surface area of all the fins, $A_{f,\text{surf}}$:

$$A_{f,\text{surf}} = 2\times N_{ch} \times L_{f,\text{char}} \times L_z$$

= 2x10 x15 x 180

= 54000 mm$^2$

10) The total area of the base, $A_{b,\text{surf}}$:

$$A_{b,\text{surf}} = A_{\text{zone}} - (A_c \times N_f)$$

= (11160) – (360 x11)

= 7200 mm$^2$

11) Total effective surface area, $A_{\text{tot,\text{surf}}}$:

$$A_{\text{tot,\text{surf}}} = A_{f,\text{surf}} + A_{b,\text{surf}}$$

= 54000 + 7200 = 61200 mm$^2$

12) Reynolds Number $Re$:

$$Re = \frac{(4\times m_w)}{(\mu \times \rho \times L)}$$

(m$_w$= 200 LPH = 56x10$^{-3}$ kg/sec.)

= $$\frac{(4\times 56\times 10^{-3})}{(0.891\times 10^{-3}\times 36\times 10^{-3}\times 10)}$$

= 698.34

13) Nusselt Number $Nu$:

$$Nu = 0.332 \times (Re)^{1/2} \times (Pr)^{1/3}$$

= 0.332x (698.34)$^{1/2}$x (6.14)$^{1/3}$ (Pr from Properties of saturated water) = 16.068

14) $h = \frac{Nu \times K}{Dh}$

$$h = \frac{(16.068 \times 0.607)}{6.22 \times 10^{-3}}$$

= 1567.80 W/m$^2$K

15) $m = \left(\frac{h \times P_{\text{face}}}{K_{\text{fin}} \times A_c}\right)^{1/2}$

(K$_{\text{fin}}$=200 W/m$^2$K for aluminium fins)

= $$\frac{(1567.80 \times 364 \times 10^{-3})}{(200 \times 360 \times 10^{-6})}^{1/2}$$

= 89.03

16) Efficiency of fin:

$$\eta_f = \frac{[\tan h(m \times L_{f,\text{char}})]}{(m \times L_{f,\text{char}})}$$

= $$\frac{[\tan h(89.02 \times 15)]}{(89.02 \times 15)}$$

= 0.6519 = 65.19%

17) Overall efficiency of fin:

$$\eta_o = 1 - \left[\frac{A_{\text{surf}}}{A_{\text{tot,\text{surf}}}}\right] (1 - \eta_f)$$

= 1 - $$\frac{54000}{61200}$$ (1-0.65)

= 0.6928 = 69.28%

As we are using two stacks here, the capacity gets doubled.

![Figure 8: Actual View of the Cold Side Heat Sink](image-url)
Area Not Covered by TEG Modules = Area of insulation
= Azone - Amod_zone
= 6400 – 3200
= 3200 mm²

Selection of Material for Insulation:
Ceramic Pads
Temperature range: above 500 deg. C
Thermal Conductivity: k = 0.15 W/mK

Hence ceramic pads are selected for highest temperature range, low thermal conductivity. Also it can be cut in to required size and shape easily.

VII. MANUFACTURING & ASSEMBLY

A. Aluminium Welding
Heat exchangers were rectangular in shape. Our requirement was to flow fluid through it without any kind of leakage. Hence we selected aluminium TIG welding of convergent zone to the ends of both heat exchangers. After welding, heat exchangers look like-

Figure 9: Aluminium Welding of Heat Exchangers.

B. Assembly
The heat exchangers are assembled with the sandwich arrangement of TEG modules between them as shown in fig.9. Before assembly the thermal grease is applied on both the surfaces of TEG modules to enhance the heat transfer. Ceramic pads are inserted between the exchangers for insulation for the area not covered by modules as shown in fig.

Counter flow type arrangement is made for the heat exchangers.

Two ‘C’ clamps are used for clamping of heat exchangers. Thermocouples (K-Type) are connected along with the display for temperature measurement.

C. Connection of thermoelectric modules
When thermoelectric modules are connected in series they operate under the condition of increasing voltage. From Seebeck Equation,

\[ V_1 = \alpha_1 (T_H - T_C) \]
\[ V_2 = \alpha_2 (T_H - T_C) \]
Adding 1&2,

\[ V_1 + V_2 = V_{\text{Series}} \]
So, \[ V_{\text{Series}} = \Delta T (\alpha_1 + \alpha_2) \]

VIII. RESULT ANALYSIS
After successful assembly, sets of trials are taken on the AETEG System retrofitted on a 4 stroke, 3 cylinder, MARUTI 800 SI Engine at different RPMs. As a load on system, LED load bank is used. Using the thermocouples; temperatures at 4 sections are measured on Digital temperature indicator. Then voltage & current at various engine speeds are measured on Digital multimeter.

Figure 10: Assembly of heat exchangers with modules and insulation

Figure 11: Maruti 800 Engine for AETEG System Trials
In Table II,

- $T_1$: Hot side inlet temperature
- $T_2$: Hot side outlet temperature
- $T_3$: Cold side inlet temperature
- $T_4$: Cold side outlet temperature
- $T_{in}$: Exhaust gas temperature at AETEG inlet
- $T_{ex}$: Exhaust gas temperature at AETEG exit

**A. Experimental Observation Table & Result Table:**

**Table II
Experimental Observation Table**

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>Engine Speed RPM</th>
<th>$T_1$ °C</th>
<th>$T_2$ °C</th>
<th>$T_3$ °C</th>
<th>$T_4$ °C</th>
<th>Tin °C</th>
<th>Tex °C</th>
<th>Voltage V (V)</th>
<th>Current I A</th>
<th>Power P = V*I (W)</th>
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</thead>
<tbody>
<tr>
<td>1</td>
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<td>102</td>
<td>42</td>
<td>43</td>
<td>296</td>
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<td>104</td>
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</table>

**Table III
Result Table**

<table>
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<tr>
<th>Sr.No</th>
<th>Engine Speed (RPM)</th>
<th>Temp. Drop $(T_{ex} - T_{in}) = \Delta T$ °C</th>
<th>Voltage V (V)</th>
<th>Current I (A)</th>
<th>$P_{in} = m_{ex}<em>C_p</em>\Delta T$ (W)</th>
<th>$P_{out}$ (W)</th>
<th>AETEG Overall Efficiency $\eta = P_{out}/P_{in}$ (%)</th>
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<td>15.12</td>
<td>5.0708</td>
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</table>
B. Characteristics Graphs

In order to evolve performance characteristics of the system following graphs were plotted and studied-

1. Voltage Vs Engine Speed

![Voltage Vs Engine Speed](image1)

The graph shows that as the engine speed increases voltage generated also increases. Hence voltage is proportional to engine speed. With the engine speed of 3970 RPM, voltage generated was 10.5 V.

2. Current Vs Engine Speed

![Current Vs Engine Speed](image2)

The graph explains that the current increases with the engine speed. It first increases gradually up to 2850 RPM then rapidly beyond that speed. At the speed of 3970 RPM the current was 1.45 A.

3. Power output Vs Engine Speed

![Power output Vs Engine Speed](image3)

The graph shows that the power output is function of engine speed. At the speed of 3970 RPM, the power developed by TEG was 15.225 W.

4. AETEG Overall Efficiency Vs Engine Speed

![Efficiency Vs Engine Speed](image4)

The graph explains the relation between the overall efficiency of the system and engine speed. At 3970 RPM the efficiency obtained was 5.078%.

IX. CONCLUSION

1. An Automobile Exhaust Thermoelectric System was designed and developed for the waste heat recovery of an automobile engine.

2. The system was retrofitted to the exhaust line of a 4 stroke, 3 cylinder Maruti 800cc SI engine and measurements were taken to study the performance of this system.
3. It was found that to get improved efficiency of this system, thermal management is very important. Double stacked type cold side heat sink gives better temperature gradient across the TEG. Counter flow type arrangement enhances the effective heat transfer. Also insulation used for the area not covered by TEG modules avoids the heat losses.

4. At high vehicle speeds, the total power that could be extracted was increased. More power could also be extracted by improving the exhaust gas heat exchanger. However with the current design the hot junction temperatures at or above 250°C were allowed for the given material of TEG (Bi-Te) and results were obtained.

5. Results show that voltage, current, power developed and efficiency of the system increase with the increase in engine speed. At the engine speed of 3970 RPM, the power generated was 15.12W and efficiency of the system was 5.0708%.

6. Hence the AE-TEG system traps the waste heat of exhaust gases from engine & generates useful power which can be used to charge the car battery, to power auxiliary systems and minor car electronics.

7. As AE-TEG reduces the wastage of energy, it improves the overall efficiency of vehicle. AE-TEG system can be profitable in the automobile industry.

REFERENCES


