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Design and Performance Analysis of Heat Exchanger for Thermoelectric Power Generation Using Exhaust Waste-Heat Energy

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ABSTRACT

The efficiency in an internal combustion engine ranges from 25% to 35%. About 40% of the overall fuel energy losses in a combustion engine are waste heat which is blown out with the exhaust gases and 30% is cooled away by the vehicle's radiator. Even if a small fraction of the waste heat could be turned into useful energy again it would be a step in the right direction of improving fuel economy. Being one of the promising new device for an automotive waste heat recovery, thermoelectric generators will become one of the most important and outstanding device in the future. Thus this study involves generation of electrical energy with the help of thermoelectric power generator. Thermoelectric modules which are used as thermoelectric generators are solid state devices that are used to convert thermal energy from a temperature gradient to electrical energy and it works on basic principle of Seebeck effect. Hence the selection of thermoelectric materials plays very important role for energy conversion in thermoelectric applications. Thermoelectric modules will be selected according to the temperature difference between exhaust gases side and the engine coolant side. In order to achieve uniform temperature distribution and higher interface temperature, the thermal characteristics of heat exchangers with various heat transfer enhancement features such as internal structure, material and surface area is to be considered. After designing suitable heat exchanger the thermo electric modules will be incorporated on the heat exchanger for performance analysis. In order to observe the differential change in exhaust conditions due to the addition of thermoelectric generator to the exhaust system, experiments to be conducted on the test engine. Thus this paper aims to design and analyze the performance of heat exchanger for thermoelectric power generationusing waste-heat energy from internal combustion engine.

Keywords—Figure of Merit, Seebeck Effect, Thermoelectric Module, Thermoelectric Generator, Waste Heat Recovery.

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I. INTRODUCTION

Most engines operate with an efficiency rate of about 30%, with most of the wasted energy lost as heat. There is an increased need to identify alternative energy sources and enhance the efficiency of engines in order to reduce the consumption of fuel. The purpose of this project is to examine whether lost energy can be recovered in the form of electricity to power the electrical components of a vehicle. Thermoelectric Power generator will be analyzed as possible solutions to recover this lost energy in order to improve the overall engine efficiency.

A. Problem Statement

Study on automobiles gasoline powered internal combustion engine shows that only approximate 25% of the fuel energy is used to drive the engine, whereas 40% of the fuel energy is wasted in exhaust gas, 30% in engine coolant and 5% in friction and parasitic losses. For example, a full tank of vehicle capacity is 100 litre fuels, but only 25 litre of fuel is turn into useful mechanical energy to power vehicle, the remaining of 75 litre of fuel is dissipate as heat energy. This is not logic and non-economical but this is what vehicle does every day. Therefore many studies had carried out to recover the waste heat dissipated by vehicle.

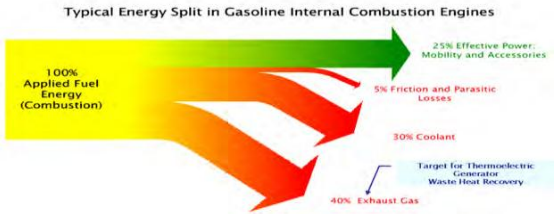

Fig.1 Energy Split in IC Engines

If the waste heat can recover, not only the every Ringgit spends for fuel is become more valuable, but also can reduce the fuel consumption due to less fuel require to generate electric for vehicle. As a conclusion, the increasing of oil prices in the world market and low utilization of gasoline powered engine makes it necessary to generate new sustainable sources of electric power in modern automobiles. Furthermore, vehicle nowadays requires more and more electricity energy in order to maintain the communication, navigation, engine control, and safety systems of the vehicle. Therefore TEG is the best solution to recover waste heat through converts the heat energy into electricity. The focus of this project is to design a TEG system which can integrate into the automobile vehicle to generate electricity.

II. METHODOLOGY

A. Thermoelectric Power Generator:

The basic theory and operation of thermoelectric based systems have been developed for many years. Thermoelectric power generation is based on a phenomenon called “Seebeck effect” discovered by Thomas Seebeck in 1821. When a temperature difference is established between the hot and cold junctions of two dissimilar materials (metals or semiconductors) a voltage is generated, i.e., Seebeck voltage. In fact, this phenomenon is applied to thermocouples that are extensively used for temperature measurements. Based on this Seebeck effect, thermoelectric devices can act as electrical power generators.

Fig.2 shows a schematic diagram illustrating components and arrangement of a conventional thermoelectric power generator. As shown in figure, it is composed of two ceramic plates (substrates) that serve as a foundation, providing mechanical integrity, and electrical insulation for n-type (heavily doped to create excess electrons) and p-type (heavily doped to create excess holes) semiconductor thermoelements. In thermoelectric materials, electrons and holes operate as both charge carriers and energy carriers. The ceramic plates are commonly made from alumina (Al_2O_3), but when large lateral heat transfer is required, materials with higher thermal conductivity (e.g. beryllium and aluminium nitride) are desired. The semiconductor thermoelements (e.g. silicon-germanium SiGe, lead-telluride PbTe based alloys) that are sandwiched between the ceramic plates are connected thermally in parallel and electrically in series to form a thermoelectric device (module). More than one pair of semiconductors are normally assembled together to form a thermoelectric module and within the module a pair of thermoelements is called a thermocouple. The junctions connecting the thermoelements between the hot and cold plates are interconnected using highly conducting metal (e.g. copper) strips as shown in figure.

The potential of a material for thermoelectric applications is determined in large part to a measure of the material’s dimensionless figure of merit (ZT). Semiconductors have been primarily the materials of choice for thermoelectric applications. There are challenges in choosing suitable materials with sufficiently higher ZT for the applications.

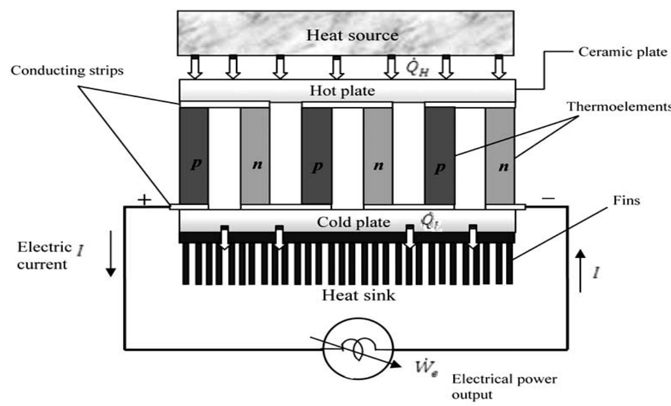

Fig.2 Components and arrangement of a typical single-stage thermoelectric power generator. [2]

B. Figure of merit:

The performance of thermoelectric materials can be expressed as

$$Z = \frac{\alpha^2}{kR} \qquad (1)$$

Z is the thermoelectric material figure-of-merit,

α is the Seebeck coefficient given by,

$$\alpha = \frac{\Delta V}{\Delta T} \qquad (2)$$

R is the electric resistivity (inverse of electric conductivity) and k is the total thermal conductivity.

The figure of merit depends on the properties of thermoelectric material used. A high value of Z is obtained by using materials of large seebeck coefficient, small thermal conductivity and small electrical resistivity. This figure-of-merit may be made dimensionless by multiplying by $\overline{T}$ (average absolute temperature of hot and cold plates of the thermoelectric module, K), i.e,

$$Z\overline{T} = \frac{\alpha^2 \overline{T}}{kR} \tag{3}$$

$$\overline{T} = \frac{T_H + T_L}{2} \tag{4}$$

The term $\frac{\alpha^2}{kR}$ is referred to as the electrical power factor. In general, a thermoelectric power generator exhibits low efficiency due to the relatively small dimensionless figure-of-merit ($Z\overline{T} \le 1$) of currently available thermoelectric materials. The conversion efficiency of a thermoelectric power generator defined as the ratio of power delivered to the heat input at the hot junction of the thermoelectric device, is given by,

$$n = \frac{W_e}{Q_H} \tag{5}$$

The maximum conversion efficiency of an irreversible thermoelectric power generator can be estimated using,

$$n_{\text{ideal}} = \left(1 - \frac{T_L}{T_H}\right)\left[\frac{M-1}{M+\frac{T_L}{T_H}}\right] \tag{6}$$

Where,

$$M = \left[1 + \frac{Z}{2}(T_H + T_L)\quad\right]^{1/2} \tag{7}$$

The value of the figure-of-merit is usually proportional to the conversion efficiency. The dimensionless term $Z\overline{T}$ is therefore a very convenient figure for comparing the potential conversion efficiency of modules using different thermoelectric materials.

III.LITERATURE REVIEW

Crane D. et al. [1] described a design concept that maximizes the performance for thermoelectric power generation systems in which the thermal power to be recovered is from a fluid stream (e.g., exhaust gas) subject to varying temperatures and a broad range of exhaust flow rates. The device is constructed in several parts, with each part optimized for a specific range of operating conditions. The thermoelectric system characteristics, inlet mass flow rates and fluid temperatures, and load and internal electrical resistances are monitored and generator operation is controlled to maximize performance. With this design, the system operates near optimal efficiency for a much wider range of operating conditions. Application of the design concept to an automobile is used to show the benefits to overall system performance.

Basel I.I. et al. [2] presented a background on the basic concepts of thermoelectric power generation and recent patents of thermoelectric power generation with their important and relevantapplications to waste-heat energy are reviewed and discussed.

C.Ramesh Kumar et al. [3] experimentally studied the performance of thermoelectric generators under various engine operating conditions. A heat exchanger with 18 thermoelectric generator modules was designed and tested in the engine test rig. Various designs of the heat exchangers were modelled using computer aided design and analysis was done using a computational fluid dynamics code. From the simulated results it was found that rectangular shaped heat exchanger met our requirements and also satisfied the space and weight constraint. Hence fabricated and the thermoelectric modules were incorporated on the heat exchanger for performance analysis. Also revealed that energy can be tapped efficiently from the engine exhaust and in near future thermoelectric generators can reduce the size of the alternator or eliminate them in automobiles.

Hsu C. et al. [4] constructed a system to recover waste heat comprised 24 thermoelectric generators. Simulations and experiments for the thermoelectric module in this system are undertaken to assess the feasibility of these applications. A slopping block is designed on the basis of simulation results to uniform the interior thermal field that improves the performance of TEG modules. Besides simulations, the system is designed and assembled. Measurements followed the connection of the system to the middle of an exhaust pipe. Open circuit voltage and maximum power output of the system are characterized as a function of temperature difference. Through simulations and experiments, the power generated with a commercial TEG module is presented.

IV.PROPOSED WORK

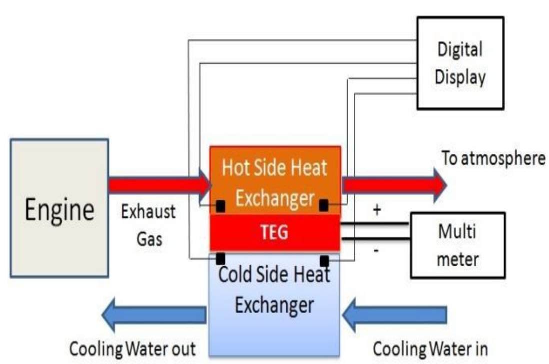

Fig.3 TEG System Layout

As shown in the Fig.3, 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.

Thermoelectric power generator consists of three main components, they are:

i. Thermoelectric materials
ii. Hot-side heat exchanger
iii. Cold-side heat exchanger
iv.

IV. EXPERIMENTAL WORK

A. Selection of Suitable Thermoelectric Material for TEM:

Among the vast number of materials known to date, only a relatively few are identified as thermoelectric materials. Thermoelectric materials can be categorized into established (conventional) and new (novel) materials. Today's most thermoelectric materials, such as Bismuth Telluride (Bi_2Te_3)-based alloys and PbTe-based alloys, have a $Z\overline{T}$value of around unity (at room temperature for Bi_2Te_3 and 500-700K for PbTe). However, at a $Z\overline{T}$of 2-3 range, thermoelectric power generators would become competitive with other power generation systems. Effective thermoelectric materials should have a low thermal conductivity but a high electrical conductivity. A large amount of research in thermoelectric materials has focused on increasing the Seebeck coefficient and reducing the thermal conductivity, especially by manipulating the nanostructure of the thermoelectric materials. Because the thermal and electrical conductivity correlate with the charge carriers, new means must be introduced in order to conciliate the contradiction between high electrical conductivity and low thermal Conductivity.

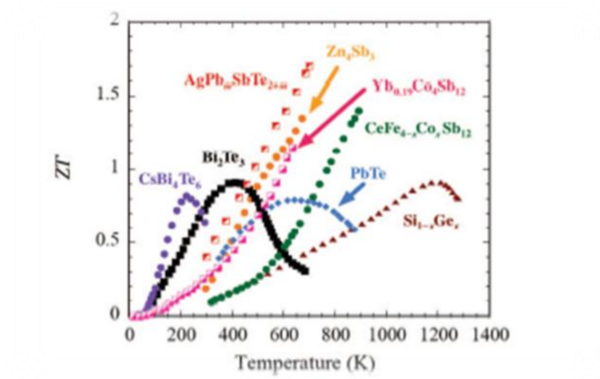

Fig.4 Figure of merit ZT shown as a function of temperature for several bulk thermoelectric materials [2]

Thermoelectric materials (those which are employed in commercial applications) can be conveniently divided into three groupings based on the temperature range of operation, as shown in table 1. Although the above mentioned materials still remain the cornerstone for commercial and practical applications in thermoelectric power generation, significant advances have been made in synthesising new materials and fabricating material structures with improved thermoelectric performance. However the proportion of heat supplied that is converted into electrical energy is only about 5 to 7 percent. Efforts have focused primarily on improving the material's figure-of-merit, and hence the conversion efficiency.

TABLE I
TEMPERATURE RANGE FOR THERMOELECTRIC MATERIALS

Temperature range	**Thermoelectric material**
Up to around 450K.	Alloys based on Bismuth (Bi) in combinations with Antimony (Sn), Tellurium (Te) or Selenium (Se)
Up to around 850K	Based on alloys of Lead (Pb)
Up to 1300K.	SiGe alloys

B. Performance analysis of selected thermoelectric material

Bismuth Telluride (Bi_2Te_3):

The maximum value of figure of merit,

Zmax = $3 \times 10^{-3} K^{-1}$

The optimum value of the resistance ratio,

$$M = \left[1 + \frac{Z}{2}(T_H + T_L)\right]^{1/2}$$

Where,

T_H = temperature of the source (K)

T_L = temperature of the sink (K)

T_H = 400 K T_L= 315 K

By putting above values in equation,

$$M = \left[1 + \frac{3 \times 10^{-3}}{2}(400 + 315)\right]^{1/2}$$

We get, M= 1.4396

We know that, the maximum or ideal thermal efficiency of a thermoelectric convertor is given by,

$$n_{\mathrm{th\ max}} = \left(1 - \frac{T_L}{T_H}\right)\left[\frac{M-1}{M + \frac{T_L}{T_H}}\right]$$

By putting above values in given equation,

$$n_{\mathrm{th\ max}} = \left(1 - \frac{315}{400}\right)\left[\frac{1.4396-1}{1.4396 + \frac{315}{400}}\right]$$

We get, maximum thermal efficiency is

$n_{\mathrm{th\ max}}$= 0.083130= 8.313%

Bi-Te is one of the easy available materials with highest value of α, low cost. Also the position of TEG system is just behind the catalyst converter. All the TEGs designed to be mounted in this position are based on bismuth telluride alloys. 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.

V. DESIGN OF HOT SIDE HEAT EXCHANGER

The overall heat exchanger model is in need of an increase in heat transfer effectiveness. To enable more heat to be channelled into the thermoelectric modules, finned heat exchangers are needed. The increase in surface area increases convection which in turn increases the heat transferred to or from the fluid depending on which side of the heat exchanger the fins are on. An appropriately selected material with high thermal conductivity is needed to maintain an effective level of conduction. When more heat is transferred to the TEG more power is removed from the system and thus, generated by the heat exchanger. Many fin options are available and selection of a configuration is dependent on several reasons. The overall intent of the finned heat exchangers is to increase surface area which will increase heat transfer. However, the increase in surface area will increase the pressure drop of the flow of fluid passing over the heat exchangers. This substitution needs to be balanced as part of the design of the fin assemblies. The design also needs to reflect material and manufacturing costs. Before the extended surfaces can be modelled, the size of the heat exchanger unit must be determined.

Sizing up the heat exchanger is based on the size, orientation, and number of modules. Instead of finding the whole heat exchanger size, the size of an individual zone is to be found and then extended to represent the entire heat exchanger. First, the length and width of all the modules combined within a zone is to be determined. 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 help 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.

Module size: 40 x 40 x 3.4 mm

$$L_{\mathrm{mod,zone}} = W_{\mathrm{mod}} \times N_{\mathrm{mod,ser}}$$
$$L_{\mathrm{mod,zone}} = 40 \times 2 = 80\ \mathrm{mm}$$
$$W_{\mathrm{mod,zone}} = W_{\mathrm{mod}} \times N_{\mathrm{mod,par}}$$
$$W_{\mathrm{mod,zone}} = 40 \times 1 = 40\ \mathrm{mm}$$

These parameters can be used to develop two more equations and important values. One of these values is the surface area of the modules in the zone. The other value is a ratio of length to width. Each of these can be used to determine the dimensions of the zone which are of more value and a necessity in determining performance of the finned heat exchangers. $A_{mod,zone}$is the surface area of all the modules in the zone and β_{lw} is the ratio of the length of all the modules in the zone to the width of all the modules in the zone. These two parameters, in addition to a user input, can now be used to determine the dimensions of the zone. It is required that the user define the ratio of the total zone surface area to $A_{mod,zone}$.

$$A_{\mathrm{mod,zone}} = L_{\mathrm{mod,zone}} \times W_{\mathrm{mod,zone}}$$
$$A_{\mathrm{mod,zone}} = 80 \times 40 = 3200\ \mathrm{mm}^2$$
$$\beta_{\mathrm{lw}} = \frac{L_{\mathrm{mod,zone}}}{W_{\mathrm{mod,zone}}} = \frac{80}{40} = 2$$

$A_{zone}$is the surface area of a zone and γ is the user defined ratio for zone area to modules in a zone area.If it is defined as one, then there is no insulation surrounding the modules within a zone. If it is greater than one, there is considered to be insulation surrounding the modules and the insulation is adjacent to the modules on all sides. It fills in the lengths and widths not defined as module length or width. The exact length and width of each stretch of insulation is not needed to be known. γ is always greater than or equal to one by its definition. We take it as 2. $A_{ins}$is the area of the insulation.

$$A_{\mathrm{zone}} = \gamma \times A_{\mathrm{mod,zone}} = 2 \times 3200 = 6400\ \mathrm{mm}^2$$
$$A_{\mathrm{ins}} = A_{\mathrm{zone}} - A_{\mathrm{mod,zone}} = 6400 - 3200 = 3200\ \mathrm{mm}^2$$

Two more important calculations that define zone dimensions are needed. They are length and width of the zone itself, insulation included. Now that area of a zone is known and the total module length to total module width ratio is known, it is rather simple to calculate zone length and zone width.

$$L_z = \sqrt{\beta_{lw} \times A_{zone}} = 120 \text{ mm}$$
$$W_z = \sqrt{\frac{1}{\beta_{lw}} \times A_{zone}} = 60 \text{ mm}$$

L_z is the length of an entire zone and W_z is the width of a zone. These are used throughout the calculations of the finned heat exchangers and are necessary for defining various fin dimensions. Therefore final dimension of Hot side heat sink = 120 x 60 mm.

A. Design of Rectangular Straight Fins [10]:

Rectangular straight fins are very common fin geometry because of their simplicity to manufacturer. They are commonly available in different sizes and can be mounted to other structures fairly easily. Their simplicity also makes for a good starting place in describing their relation to the methods used in TEG. The initial set up of the fins and their geometry requires a few input parameters. 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}).Equally important as the number of fins is the number of channels, N_{ch}. This is because the number of channels provides the area that allows fluid flow to pass through the heat exchanger. The number of channels is simply equal to the number fins minus one. This is because two fins are considered the outer wall of the shell of the heat exchanger.

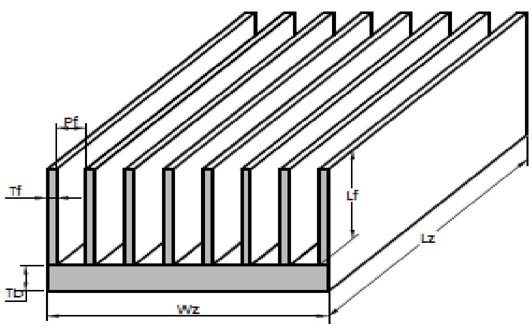

Fig.5 Rectangular Hot Side Heat Exchanger Geometry

Number of fins (N_f) =8
Number of channels (N_{ch}) =N_f-1 =7
Thickness of an individual fin (T_f)=2mm
The length of an individual fin protrudes from its base
(L_f) =26mm
Thickness of the base (T_b) =7mm
Calculations:
1) Pitch of Fin (P_f):
The pitch of a fin is needed to be known to help determine the spacing between fins. The pitch helps to keep the fins constrained to the size of the heat exchanger.

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

2) Spacing between fins (S_f):
The spacing between fins is used for several calculations because it represents part of the dimensioning of the flow path.

$$S_f = P_f - T_f = 8.28 - 2 = 6.28 \text{ mm}$$

3) Wetted Perimeter, (P_{wet}):
P_{wet} , is the perimeter of a flow path or one channel created by the fins.

$$P_f = 2(L_f + S_f) = 2(26 + 6.28) = 64.57 \text{ mm}$$

4) Hydraulic diameter (D_h):
The wetted perimeter of the fin is known, it is no possible to find the hydraulic diameter, D_h. It is an artificial diameter representing the channel in which flow travels through.

$$D_h = \frac{4 \times L_f \times S_f}{P_{wet}} = \frac{4 \times 26 \times 6.28}{64.57} = 10.11\text{mm}$$

5) An entrance area, (A_{ent}):
An entrance area needs to be considered for the flow paths through the rectangular straight fins.

$$A_{ent} = S_f \times L_f \times N_{ch} = 6.28 \times 26 \times 7 = 1142.96\text{mm}^2$$

6) The characteristic length of the fin, ($L_{f,char}$) :
This is needed for fin efficiency calculations.

$$L_{f,char} = L_f + \frac{T_f}{2} = 26 + \frac{2}{2} = 27 \text{ mm}$$

7) The perimeter of the face of the fin, (P_{face}):
It is needed to support in finding the efficiency of the proposed fin.

$$P_f = 2(T_f + L_z) = 2(2 + 120) = 244 \text{ mm}$$

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

$$A_c = T_f \times L_z = 2 \times 120 = 240 \text{ mm}^2$$

This value is also used in calculating the efficiency of the designed fin.
9) The total surface area of all the fins, ($A_{f,surf}$) :
The total surface area of all the fins is needed for finding the total surface area that is affected by convection. The combined fin surface area is equal to two times the number of channels to represent each of the fin surfaces within the flow path.

$$A_{f,surf} = 2 \times N_{ch} \times L_{f,char} \times L_z = 2 \times 7 \times 27 \times 120 = 45360\text{mm}^2$$

10) The total area of the base, ($A_{b,surf}$) :
The total area of the base, $A_{b,surf,}$ is also needed to help find the total effective surface area.

$$A_{b,surf} = A_{zone} - (A_C \times N_f) = 6400 - (240 \times 7) = 4480\text{mm}^2$$

11) Total effective surface area, $A_{tot,surf}$:
Total effective surface area,$A_{tot,surf}$, is the area which fluid flow occurs and convective heat transfer is present.

$$A_{tot,surf} = A_{f,surf} + A_{b,surf} = 45360 + 4480 = 49840\text{mm}^2$$

Now that the fins have been designed, their performance needs to be evaluated.

VII. DESIGN OF COLD SIDE HEAT EXCHANGER

The basic requirement of cold side heat sink was

- Heat sink should flow with full of water i.e. no air gap should get created.
- 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 [10]:

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 × (length of hot side heat exchanger) = 1.5 × 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

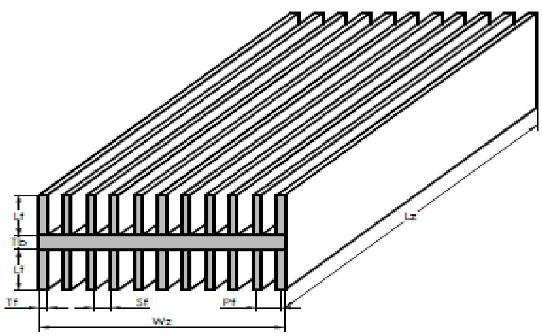

Fig.6 Double Stacked Type Heat Sink for ColdSide

Calculations:

1) Pitch of Fin (P_f):

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

2) Spacing between fins (S_f):

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

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

$$P_f = 2(L_f + S_f) = 2(14+4) = 36\text{ mm}$$

4) Hydraulic diameter (D_h):

$$D_h = \frac{4 \times L_f \times S_f}{P_{wet}} = \frac{4 \times 14 \times 4}{36} = 6.22\text{mm}$$

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

$$A_{ent} = S_f \times L_f \times N_{ch} = 4 \times 14 \times 10 = 560\text{mm}^2$$

6) The characteristic length of the fin, ($L_{f,char}$) :

$$L_{f,char} = L_f + \frac{T_f}{2} = 14 + \frac{2}{2} = 15\text{ mm}$$

7) The perimeter of the face of the fin, (P_{face}):

$$P_f = 2(T_f + L_z) = 2(2+180) = 364\text{mm}$$

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

$$A_c = T_f \times L_z = 2 \times 180 = 360\text{ mm}^2$$

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

$$A_{f,surf} = 2 \times N_{ch} \times L_{f,char} \times L_z = 2 \times 10 \times 15 \times 180 = 54000\text{mm}^2$$

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

$$A_{b,surf} = A_{zone} - (A_C \times N_f) = 11160 - (360 \times 11) = 7200\text{mm}^2$$

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

$$A_{tot,surf} = A_{f,surf} + A_{b,surf} = 54000 + 7200 = 61200\text{mm}^2$$

As we have designed cold side heat exchanger using two stacks, the capacity of it gets doubled.

I. ASSEMBLY

The heat exchangers are assembled with the sandwich arrangement of TEG modules between them as shown in fig.7. Before assembly the thermal grease is applied on both the surfaces of TEG modules to enhance the heat transfer. Counter flow type arrangement is made for the heat exchanger. After successfully assembly, performance of TEG will be investigated.

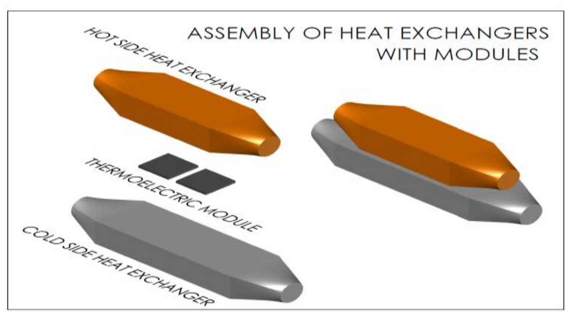

Fig.7 3-D Drawing of Assembly of Heat Exchangers with Modules

II. CONCLUSION

Thermoelectric power generation offers a promising technology in the direct conversion of waste-heat energy, into electrical power. In this paper, a background on the basic concepts of thermoelectric power generation is presented. Thermoelectric modules which are used as thermoelectric generators are solid state devices and it works on basic principle of Seebeck effect. Hence the selection of thermoelectric materials for module plays very important role. Bi-Te is one of the available materials with highest value of α. 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.

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.

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