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Computational analysis and simulation of thermo-electric power generation from automotive exhaust gas

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Abstract—The growing demand for energy, increasing fuel price, and the environmental pollution issues associated with conventional energy generating technologies has increased interest in the use of Thermo electrics, which can directly convert thermal energy into electrical energy. There is a great possibility in utilizing thermoelectric devices for automobile exhaust to electricity conversion.Studies show that significant energy savings can be achieved by recovering even a small portion of this wasted energy. From the energy balance of combustion engine, it is found that approximately 40% of the fuel energy is lost in exhaust gas. Thermoelectricmodulescan operate over a wide range of transient temperature conditions. The electric power is generated as a result of the temperature difference based on the Seebeck effect.Computational fluid dynamics (CFD) is used to simulate the exhaust gas flow within the heat exchanger.The isothermal modeling approach is used in simulation.In this paper, two cases of heat exchangers such as rectangular type and square type were used in exhaust of internal combustion engine (ICE) are modeled numerically to recover the exhaust waste heat. The heat exchangerswere proposed and designed to recover energy from exhaust of apetrol engine. The simulations are done in five engine loads (idling, suburban driving, urban, cruisingand maximum power) and After the analysis, results of irreversibility, recovered heat, surface temperature, pressure drop, average power developed and current developed etc. are compared in different engine loads and speeds. The Rectangular heat exchanger made up of aluminium shows better performance among the two proposed designs. The analysis predicts optimum conditions forthermoelectric leg pairs that maximize the power extracted for any TEG modules, and the change in this optimum conditionmay degrade the system performance.

Keywords—Automotive exhaust, Isothermal approach, Seebeck effect, Thermoelectric , Heat exchangers,Design optimization, Convergence study, Power generation.

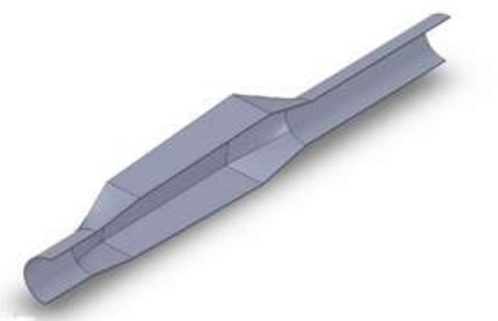
INTRODUCTION

In recent years there is a substantial interest in utilizing thermoelectric devices for automobile exhaust to electricity conversion. Studies show that significant energy savings can be achieved by recovering even a small portion of this wasted energy. From the energy balance of combustion engine, it is found that approximately 40% of the fuel energy is lost in exhaust gas and just12 to 25 % of the fuel energy converts to useful work, which results in the energy crisis and environment pollution[1]. Thermoelectric generation (TEG) technology have some distinct advantages such as, they are simple, have no moving parts, highly reliable, zero emission, low noise and are able to operate over a wide range of transient temperature conditions. The electric power is generated as a result of the temperature difference based on the Seebeckeffect[2].Computational fluid dynamics (CFD) is used to simulate the exhaust gas flow within the heat exchanger.The isothermal approach is followed throughout the research, thermal characteristics of heat exchangers with various heat transfer enhancement features are studied, such as; internal structure, material and surface area[3]. With the thermal energy of exhaust gas harvested by thermoelectric modules, a temperature gradient appears on the heat exchanger surface, so as the interior flow distribution of the heat exchanger.In this paper, two cases of heat exchangers such as rectangular type and square type were used in exhaust of internal combustion engine (ICE) are modeled numerically to recover the exhaust waste heat. The heat exchangers were proposed and designed to recover energy from exhaust of apetrol engine. The simulations are done in five engine loads (idling, suburban driving, urban, cruising and maximum power) and After the analysis, results of irreversibility, recovered heat, surface temperature, pressure drop, average power developed and current developed etc. are compared in different engine loads and speeds. The Rectangular heat exchanger made up of aluminium shows better performance among the two proposed designs. The limiting conditions, for the theoretical power generation used in previous studies are followed in this paper.

MODELLING

The computational fluid dynamics (CFD) is used to simulate the exhaust gas flow within the heat exchanger. The isothermal modelingapproach is followed during simulation[4]. Three dimensional views, wall thickness, internal structure and cross sectional view of the heat exchangers are shown in Fig.1

(a) (b)



(c) (d)

Fig.1 (a)three dimensional view of Rectangular heat exchanger (b) three dimensional view of Square heat exchanger (c) sectional view of Rectangular heat exchanger (d)sectional view of Square heat exchanger

The heat exchangers made up of aluminiumhave inlet and outlet manifold diameters as 80mm, the width of the wall at inlet as110mm and at the outlet 130mm. The cross sectional area of the square heat exchanger is 110mm x 110mm. The wall thickness is about 5mm. From inlet of the heat exchanger the cross section is found to be increasing gradually to the outlet, which provides smooth interaction of the exhaust gas with the exchanger walls. This causes the high temperature exhaust gas to diffuse in the entire lateral area rather than concentrating in the central region.

THERMAL SIMULATION

By applying fundamental formula of convective heat transfer Q = h A Δ T, heat convection can be greatly strengthened by the increase of the heat transfer area A. This target can be achieved by providing a sufficient conduction surface. Another approach is to increase the heat transfer coefficient h. According to the fluid dynamics theories, under the condition of Reynolds number Re >104, macro turbulent fluid flow is a significant factor for improving the heat transfer. Moreover, the greater the heat transfer coefficient h, the better the heat transfer quantity[5]. The thermal resistance of turbulent flow of convection mostly exists in the boundary layer. The better the synergy was between the temperature field and velocity field, the better the heat transfer[6].

According to both the theories mentioned above, the strengthening of the heat transfer can be approached by adding turbulence devices or altering the geometry to enhance the fluid disturbance and damage the boundary layer[7]. On the basis of the theories of thermal convection and turbulent flow as mentioned above, the three-dimensional models of heat exchangers such as rectangular type and square type are put forward by providing a new design. Among these internal structure, the geometrical model of the heat exchanger including rectangular internal structure, square internal structure are showed in Fig.1 a,b,c, d. The CFD simulation results including the temperature contour are showed in Fig. 3.a,b andFig. 4.a, b, however, the rectangular-shape design presents a better uniform temperature distribution than the square-shape. Considering the temperature distribution, the heat exchanger with rectangular-shape internal structure is more ideal for TEG.

BOUNDARY CONDITION

Simulation model is assured that the exhaust flow in the heat exchanger is fully turbulent and molecular viscosity can be neglected, so the standard κ–ε model is adopted in the CFD simulation. As Near wall area processing with standard wall function, the natural convection heat transfer coefficient and the environment temperature are set.

The mass flow rates for different conditions are as follows; idling: 1.2 g/sec, Urban: 5.7 g/sec, Suburban: 14.4 g/sec, Cruising: 24.3 g/sec, Maximum power: 80.1 g/sec. The Inlet temperature: 573.15 kand outlet: pressure outlet with gauge pressure 300 kPa.

As for the heat exchanger presenting an approximately axial symmetry in geometry, so the flow, pressure and temperature fields also show axisymmetric characteristics in the absence of ambient winds.

EXHAUST GAS PROPERTIES

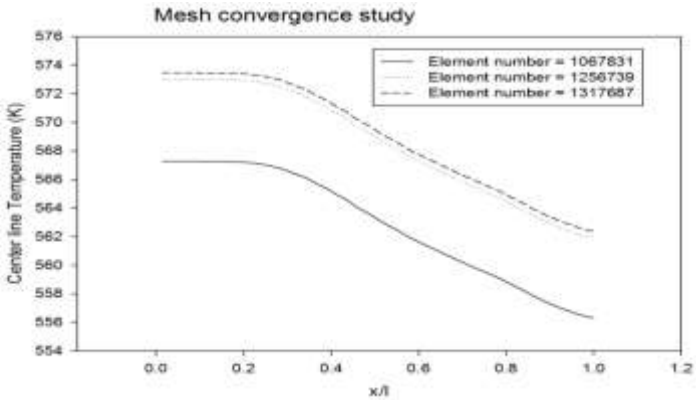
The major properties of the exhaust gas at any temperature were shown in table.1[8].

Table.1

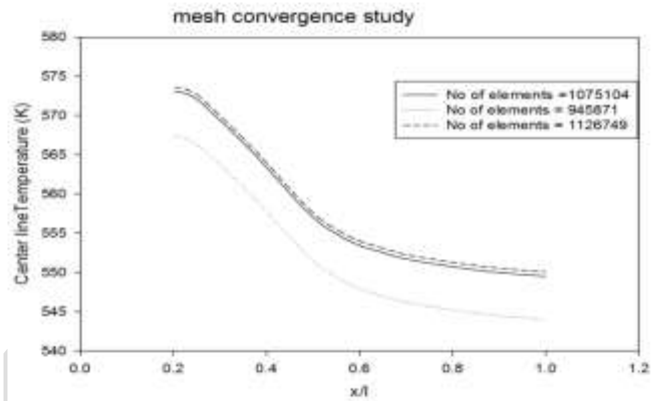
Properties	$A + B \times T + C \times T^2 + D \times T^3$			
	A	B	C	D
ρ (kg/m^3)	2.504012288761e + 00	-5.958486188418e - 03	5.578942358587e - 06	-1.772600918994e - 09
$C_{p\ (J/kg\ k)}$	1.015580935928e + 03	-1.512248401853e - 01	4.544870294058e - 04	-1.785063817137e - 07
μ (kg/ms)	1.325186910351e -06	6.740061370040e - 08	-3.749043579926e - 11	1.110074961972e - 14
K(w/mk)	-3.182421851331e - 03	1.185847825677e - 04	-7.706004236629e -08	2.939653967062e - 11

MESH CONVERGENCE

The Fig.2a, b shows the mesh convergence for the rectangular andsquare heat exchanger simulations. The convergence for rectangular heat exchanger was found to be at an element number1067831 corresponding to the centerline temperature of 555k. Similarly for the square heat exchanger, aboutan element number 1075104 and corresponding to the centerline temperature of 544k.



(a)

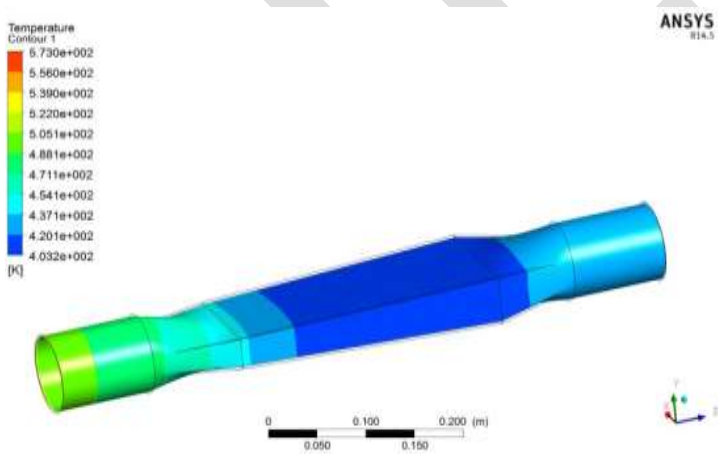


(b)

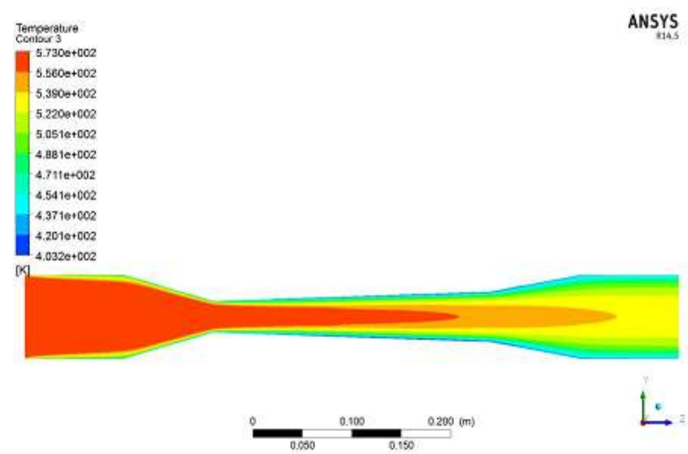
Fig.2 (a)convergence for Rectangular heat exchanger (b) convergence for square heat exchanger

TEMPERATURE FIELD ON HEAT EXCHANGERS

The temperature field on the heat exchanger plays an important role in thermoelectric energy conversion in three aspects such as: firstly, it determined the available thermoelectric material by maximum continues operating temperature; secondly, it affects the energy conversion efficiency of heat to electricity; thirdly, dominates the thermal stresses in device level and module level. A non uniform thermal stress may cause performance deterioration and permanent damage to the TEG modules[9].



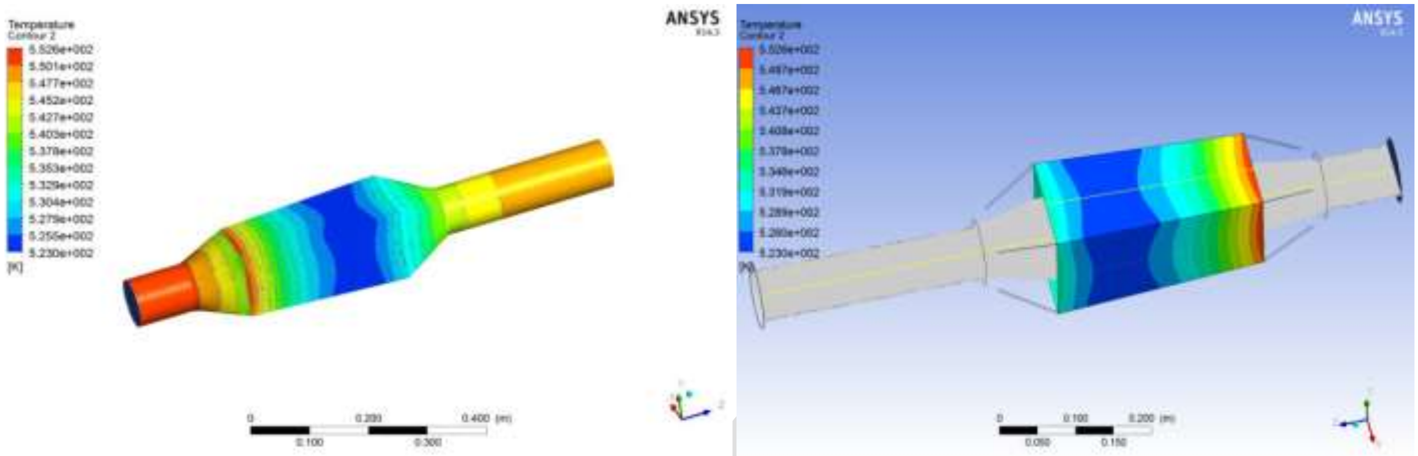
(a)



(b)

Fig.3 Temperature field: (a)on rectangular heat exchanger surface (b) Sectional view of Rectangular heat exchanger

The Fig.3. a,b and Fig.4. a,b illustrates the temperature distribution upon the heat exchanger surface. From the simulation it can be clearly seen that(Fig.3. a) the rectangular heat exchanger can give a uniform temperature distribution, closer to the isothermal model approach.



(a) (b)

Fig.4 Temperature field: (a)on square heat exchanger (b) on square heat exchanger surface alone

PRESSURE FIELD ON HEAT EXCHANGERS

The hot exhaust gas flow from engine, when interacts with the walls and transfer energy, experiences drop of pressure. As in the case of an I.C engine system, this drop of pressure is equivalent to the rise of atmospheric pressure and causes a drop in output power[10]. The pressure field in heat exchangers is showed in Fig.5. a, b. The pressure drop is dominant in rectangular heat exchanger than square heat exchanger.

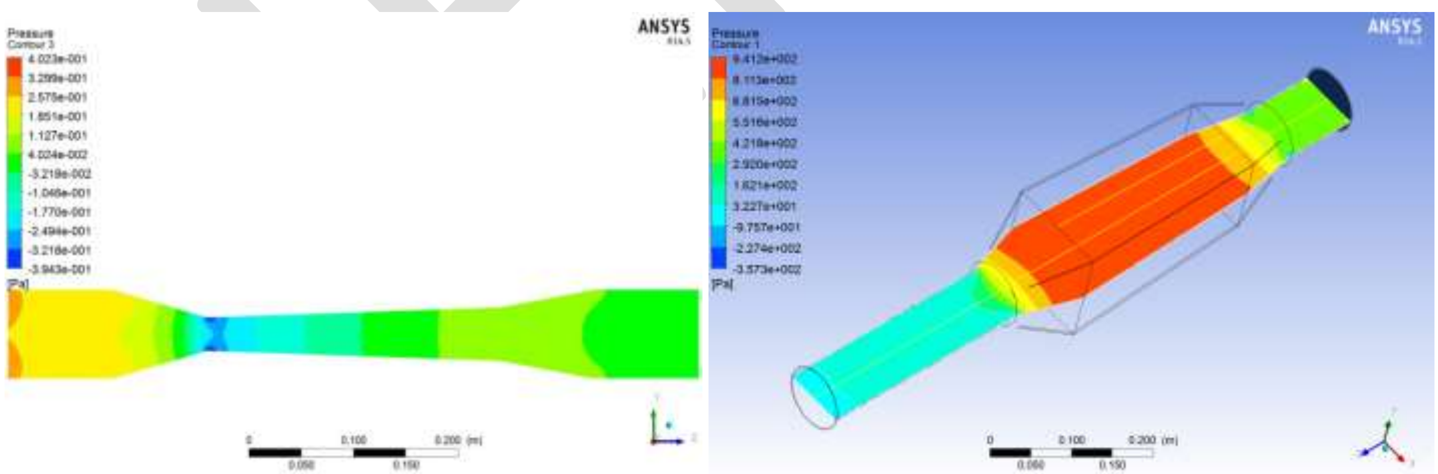


Fig.5 Pressure field: (a)on rectangular heat exchanger (b) on square heat exchanger

THERMAL SIMULATION RESULTS

The simulations are done in five engine loads such as; idling, suburban driving, urban, cruising and maximum power and the results were plotted below. The Fig.6 shows the simulation results for the surface temperatures in different conditions. Fig.6 (a) shows the rectangular heat exchanger surface temperature, which varies from 411k to 442.7k in idling to the range of 550.4k to 563.5k at maximum power. As in the case of square heat exchangerFig.6 (b) the surface temperature varies in the order of 392.7k to 406.4k at idling and from 534.5k to 552.7k at the maximum power.

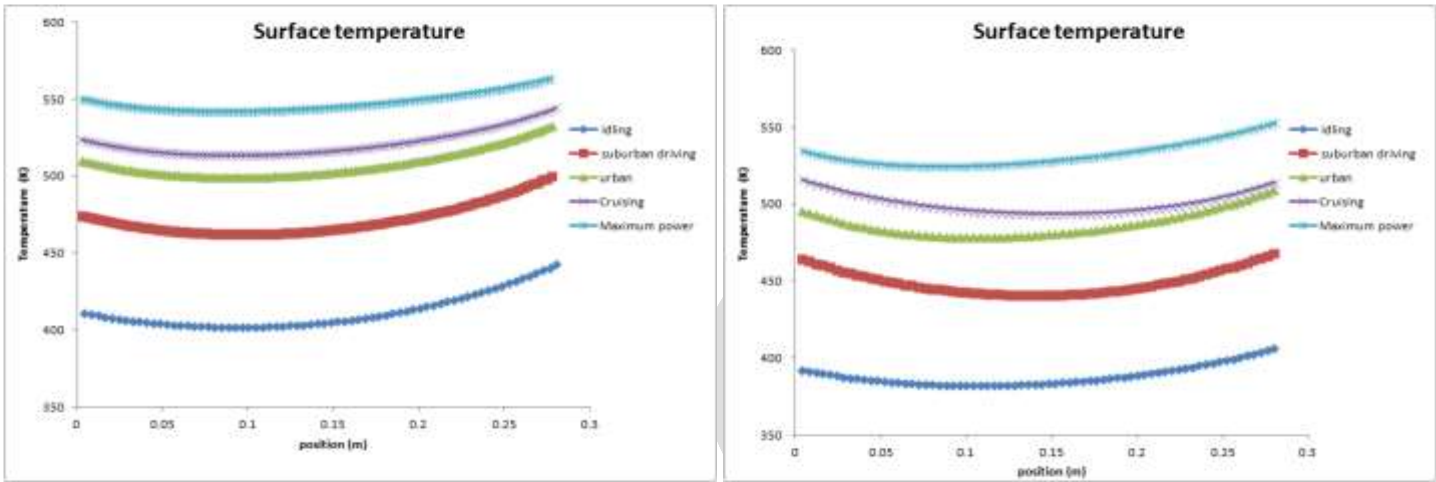


Fig.6 surface temperature: (a)on rectangular heat exchanger (b) on square heat exchanger

Fig.7 shows the variation of centerline velocity in both the heat exchangers under the given conditions. The centerline velocity of the rectangular heat exchangerFig.7 (a) varies from 1.618m/s to 1.029m/s at idling and from 83.7m/s to 55.3m/s at maximum power. The velocity variation in square heat exchanger Fig.7 (b) ranges from 0.936m/s to 1.026m/s at idling and 52.038m/s to 48.85m/s at maximum power.

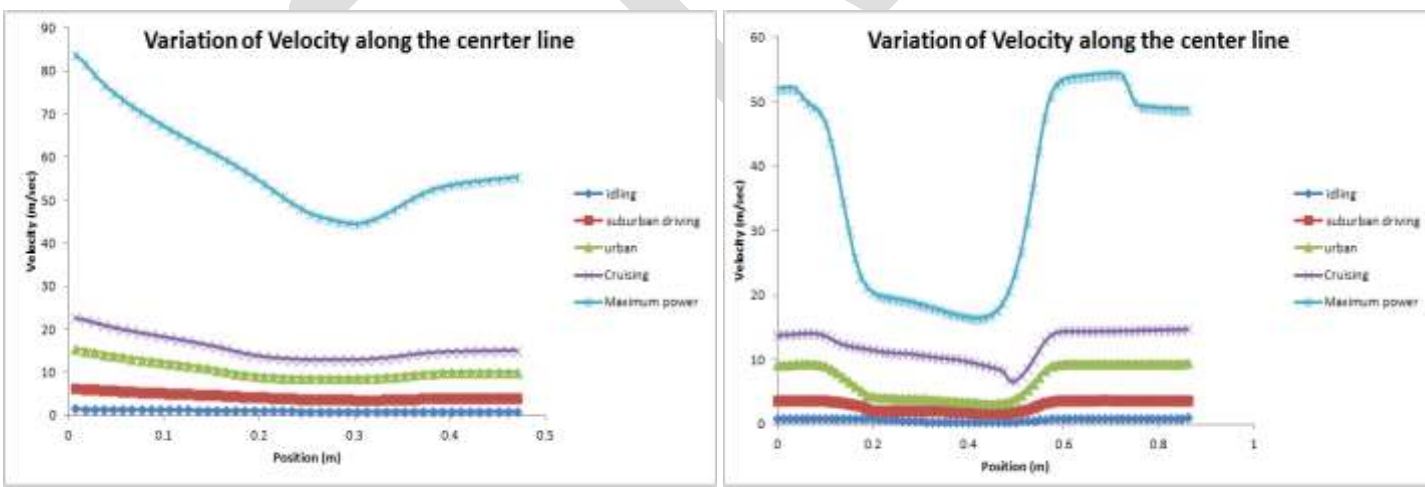


Fig.7 velocity along centerline: (a)rectangular heat exchanger (b) square heat exchanger

The pressure variation of the heat exchangers along the centerline is shown in Fig.8. The pressure variation shows less deviation as in the case of rectangular heat exchanger (Fig.8.a). For the rectangular heat exchanger a maximum deviation occurs during cruising and it varies between 300.04Pa to 299.94Pa. Similarly for the square heat exchanger (Fig.8.b), the pressure variation occurs at cruising and ranges between 300.06Pa to 300Pa.

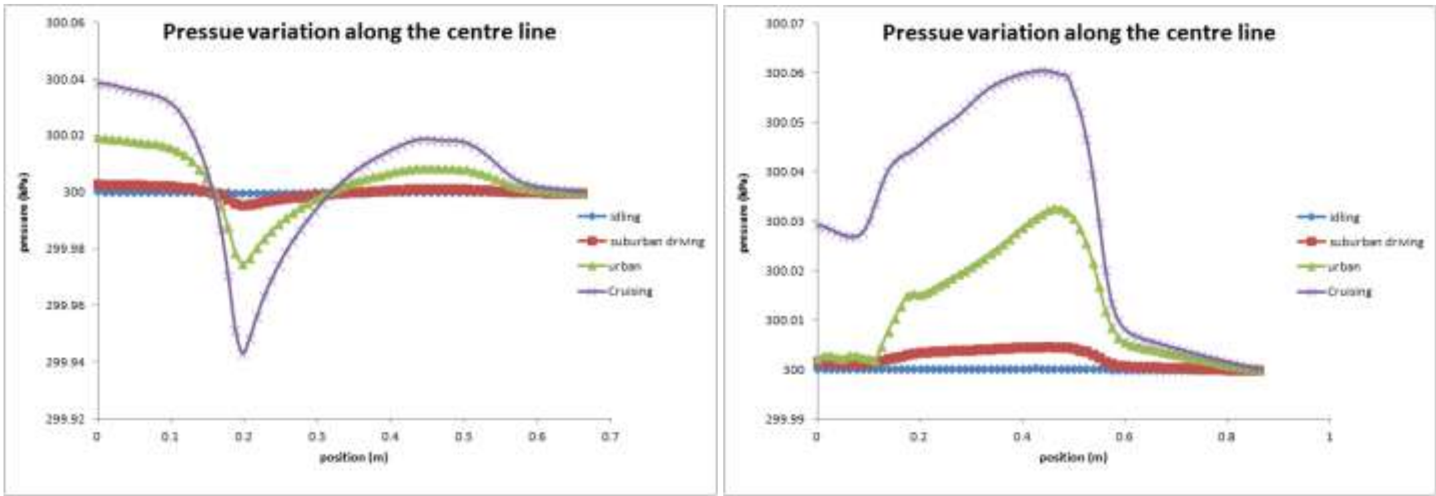


Fig.8 Pressure variation: (a)rectangular heat exchanger (b) square heat exchanger

A clear comparison of the pressure drop as well as heat flux of both the heat exchangers was shown in Fig.9 a, b below. From the chart shown below, it can be seen that the pressure drop was slightly predominant in the case of rectangular heat exchanger especially during the maximum power. But the heat flux available in the rectangular heat exchanger is higher in all conditions.

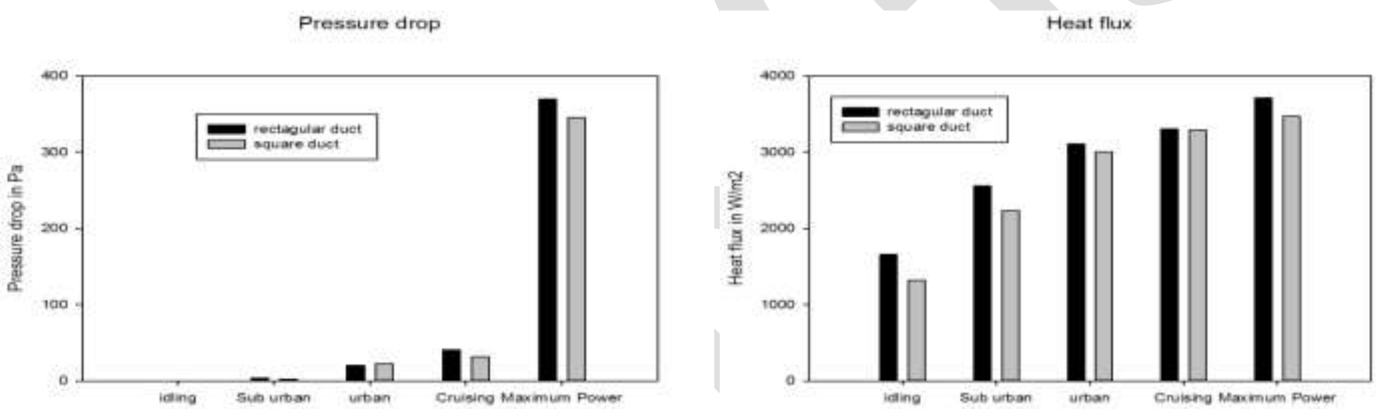


Fig.9 (a)Pressure drop(b) heat flux

From the previous studies and literature conducted on thermoelectric modules, the theoretical limit for the thermoelectric heat exchangers can be calculated. This may help us to predict the available current and power for the proposed design. For this, we need to predict and assign thermoelectric properties for the heat exchanger along with its thermoelectric materials[11]. The assumptions for p-type and n- type materials were shown in Table.2[12]. The total rate of energy transferred to all thermoelectric elementsin the form of heat is shown by equation (1).Thetotal rate of energy transferred from the thermoelectric elementson the coolant side of the system is represented by equation (2). The electrical power (p) generated by the entire system is the difference in the two energy rates as shown in equation (3)[13]. Where,Q_h heat transferred at hot side, Qc Heat transferred to the cold fluid, m mass flow rates, $c_p$specific heat capacity, $T_{h,i}$ temperature at hot side inlet, $T_{h,o}$ temperature at hot side outlet, $T_{c,\,i}$ temperature at cold side inlet and $T_{c,\,o}$temperature at cold side outlet.

$$Q_h = (m\,c_p)_h(T_{h,i}-T_{h,o}) \qquad (1)$$

$$Qc = (mc_p)_c(T_{c,o}-T_{c,i}) \qquad (2)$$

$$P = Q_h-Q_c \qquad (3)$$

We then assume that, the length of any leg pair is small comparedwith the totallength of the system, i.e., b/L<<1, whichallows us to view the system as a single continuous module alongwhich the temperatures and current can vary. Consistent with thisapproximation, we define a local current flux (i.e., current perarea), j(x)[14].

$$I = w.\, b.j(x). \qquad (4)$$

$$J(x) = \alpha\,(T_1 - T_2) / R\ddot{c}(1+m) \qquad (5)$$

$$m = (1+ Z\overline{T})^{0.5} \qquad (6)$$

$$Z = \alpha^2 / \ddot{K}\, R\ddot{c} \qquad (7)$$

$$\overline{T} = (T_1 + T_2) / 2 \qquad (8)$$

Here the current is now interpreted as being continuous.

Where,
I is the current generated, w width of the leg, b breadth of the leg, j(x) is the current flux, αseebeck coefficient, k thermal conductivity, $\overline{T}$ average temperature and R contact resistance of the conductor.

Table.2

Parameters	p type	n type
Seebeck coefficient, α	200×10^{-6} v/k	-150×10^{-6} v/k
Electrical resistivity, ρ	1.5×10^{-3} Ω cm	2.0×10^{-3} Ω cm
Thermal conductivity, k	1.1 w/m k	1.3 w/m k
Leg length, l	1mm	1mm
Leg area, A	2.25mm^2	2.25mm^2
Electrical contact resistance, $R\ddot{c}$	0 Ω m^2	0 Ω m^2

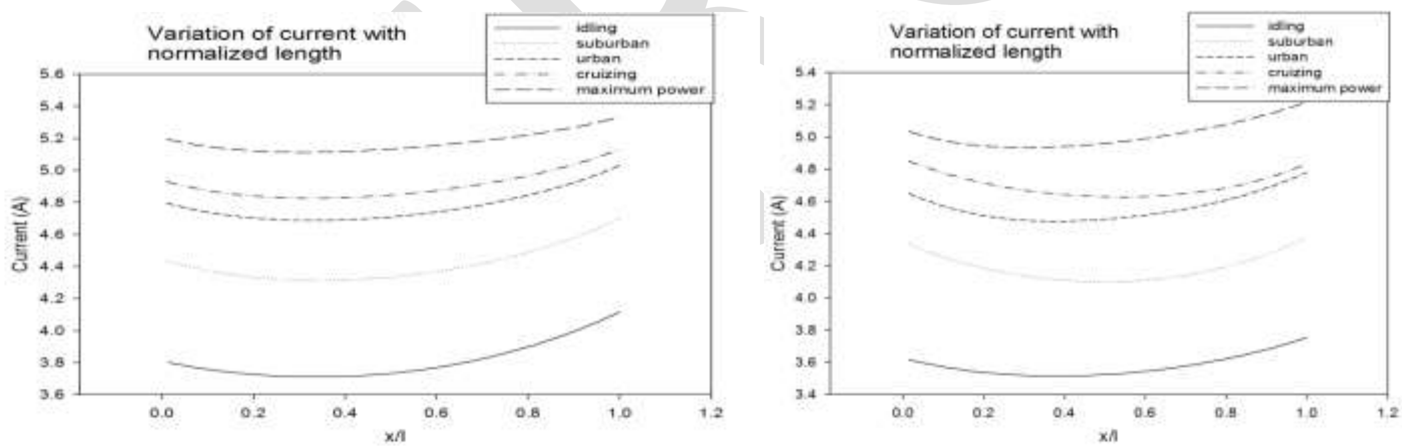


Fig.10 current: (a)rectangular heat exchanger (b) square heat exchanger

The Fig.10.a,b shows the current generated in heat exchangers, at the given set of conditions. It can be seen that maximum current was produced during the maximum power condition. The rectangular heat exchanger can generate more current compared to the square heat exchanger. The current ranges from 3.8A to 5.4A, as in the case of rectangular heat exchanger and from 3.6A to 5.2A in the case of square heat exchanger.

The average power produced by the heat exchangers is shown in Fig.11.a, b below. It can be seen that the rectangular heat exchanger gives better power output and it ranges between 260W to 590W. But in the case of square heat exchanger it ranges between 100W to 270W.

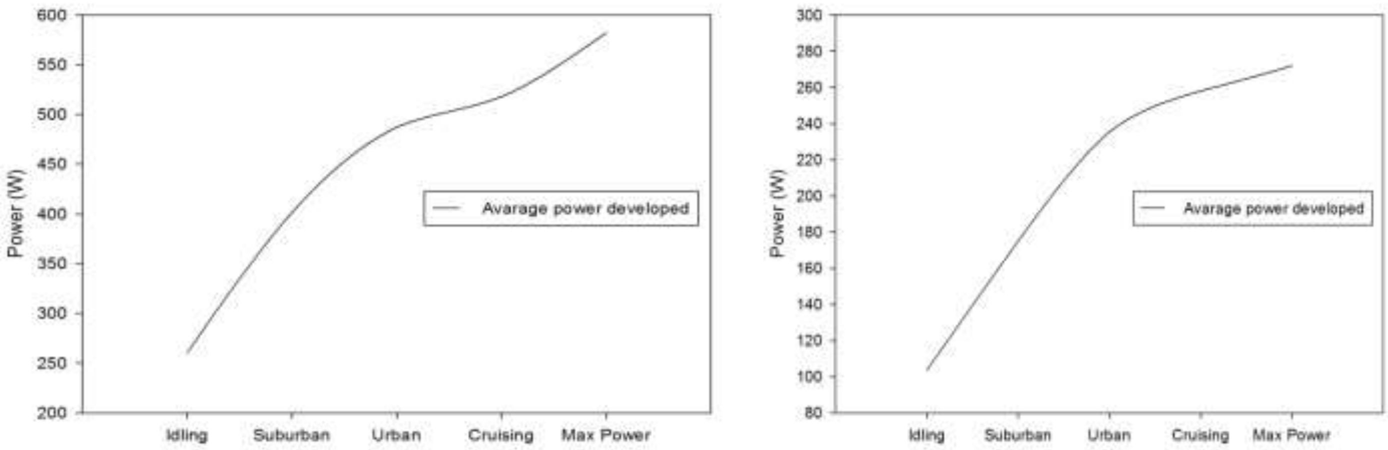


Fig.11 power: (a)rectangular heat exchanger (b) square heat exchanger

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CONCLUSION

In this work, two heat exchangers of rectangular as well as square cross section were analyzed. In which the rectangular heat exchanger shows better surface temperature, an ideal temperature uniformity which improves TEG performance. Also the rectangular heat exchanger shows comparatively less pressure drop as well as better heat flux. Further, an improved current and power generation can be observed in rectangular heat exchanger model.

In future study, the method of simulation modeling with infrared experimental verification needs to be combined with heat transfer theory and material science to serve for further structural design and optimization of thermoelectric modules and TEG, so as to improve the overall exhaust heat utilization and enhance the power generation.

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