



Improving The Efficiency Of Waste Heat Recovery Systems By Means Of A Combined Turbine-Peltier System

Ozan TEKİN¹, Ramin BARZEGAR^{2,*}, Mehmet Sait SÖYLEMEZ¹

¹ Gaziantep University, Engineering Faculty, Mechanical Engineering, Şahinbey, 27310, Gaziantep, Türkiye

² Hacettepe University, Faculty of Engineering, Mechanical Engineering, Çankaya, 06800, Ankara, Türkiye

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* Corresponding Author

e-mail:

raminbarzegar@hacettepe.edu.tr

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ORCID Numbers in author order:

0000-0001-9812-7752

0000-0003-2796-7126

0000-0001-8570-1321

ABSTRACT

A novel Combined Turbine-Peltier System (CTPS) was designed and built to harness both the thermal and kinetic energies of an engine's exhaust gas and convert it to electrical power. The turbine generator was connected to the turbocharger shaft and the Thermoelectric Generators (TEG) were assembled between the heat exchangers mounted on the exhaust pipe. The Computational Fluid Dynamics (CFD) approach and Taguchi optimization technique were employed in order to analyze and optimize the flow field and heat transfer characteristics of the system. Based on the optimized numerical results, an experimental setup was designed and manufactured, and the experiments were conducted on an engine operating at different load and speed conditions. The harvested maximum power output for a single TEG was 5.5 W with the thermal efficiency of 3.6% and the net output power of the CTPS was 190 W obtained at the engine speed of 3000 rpm. The maximum increase in the power of the ICE combined with the CTPS method was calculated as 1.6%. The obtained numerical results were compared with the experiments and showed a good accordance with the maximum deviation of 6%.

Kombine Türbin-Peltier Sistemi Aracılığıyla Atık Isı Geri Kazanım Sistemlerinin Verimliliğinin İyileştirilmesi

MAKALE BİLGİSİ

Anahtar Kelimeler:

Atık Isı Kazanımı
Termoelektrik Jeneratör
Isı Değiştirgeci
Türbin Jeneratör
İçten Yanmalı Motorlar

ÖZET

Yeni bir Kombine Türbin-Peltier Sistemi (CTPS), bir motorun egzoz gazının hem termal hem de kinetik enerjilerini kullanmak ve bunu elektrik gücüne dönüştürmek için tasarlandı ve üretildi. Türbin jeneratörü turboşarj miline bağlandı ve Termoelektrik Jeneratörler (TEG) egzoz borusuna monte edilen ısı değiştiriciler arasına monte edildi. Sistemin akış alanı ve ısı transferi özelliklerini analiz etmek ve optimize etmek için Hesaplamalı Akışkanlar Dinamiği (CFD) yaklaşımı ve Taguchi optimizasyon tekniği kullanıldı. Optimize edilmiş sayısal sonuçlara dayanarak, deneysel bir kurulum tasarlandı ve üretildi ve deneyler farklı yük ve hız koşullarında çalışan bir motorda gerçekleştirildi. Tek bir TEG için hasat edilen maksimum güç çıkışı %3,6 termal verimlilikle 5,5 W oldu ve CTPS'nin net çıkış gücü 3000 rpm motor hızında elde edildi ve 190 W oldu. ICE'nin gücündeki maksimum artış CTPS yöntemi ile birleştirildiğinde %1,6 olarak hesaplandı. Elde edilen sayısal sonuçlar deneylerle karşılaştırıldığında maksimum %6 sapmaya rağmen iyi bir uyum gösterdiği görülmüştür.

NOMENCLATURE

CTPS	Combined Turbine-Peltier System
HX	Heat Exchanger
ICE	Internal Combustion Engine
TEG	Thermoelectric Generator
WHRS	Waste Heat Recovery System
ORC	Organic Rankine Cycle
ZT	Figure of Merit
S/N	Signal to Noise
T	Temperature Difference [K]
T _c	Cold Surface Temperature [K]
T _H	Hot Surface Temperature [K]

T ₀	Dead State Temperature [K]
$\dot{W}$	Power [W]
α	Seebeck coefficient (V/K)
δ	Thermal Conductivity (W/m. K)
σ	Electrical Resistivity (S/m)
η	First Law Efficiency
μ	Second Law Efficiency
γ	Exergy or Availability [W]
s	Entropy of the State (kJ/kg. K)
h	Enthalpy of the State (kJ/kg. K)
$\dot{m}$	Mass Flow Rate (kg/sec)

INTRODUCTION

In Internal Combustion Engines (ICE), the stored chemical energy in the fuel is converted into thermal energy and then into mechanical energy. Due to thermodynamics laws and mechanical losses, this transformation can never be complete, and studies have shown that only about 38% of the chemical energy of the fuel is converted into mechanical energy (Orr et al., 2016). Figure 1 shows the total fuel energy distribution in a typical ICE in which the energy wasted through the exhaust system is about 33%. This energy is in the form of kinetic energy and enthalpy of exhaust gases. Waste heat recovery systems (WHRS) such as turbochargers, Organic Rankine Cycle (ORC), and Thermoelectric Generators (TEG) were introduced as an attempt to harness this energy (Stobart et al., 2017). Turbochargers only take advantage of the kinetic energy of the exhaust gases via a turbine-compressor combination to increase the intake air pressure of the ICE. On the other hand, ORC and TEG systems are driven by the enthalpy carried by the gases. Due to the fact that TEGs have fewer mechanical parts and maintenance needs, their implementation on the exhaust system of vehicles is more convenient and economical than ORC (Stobart et al., 2017).

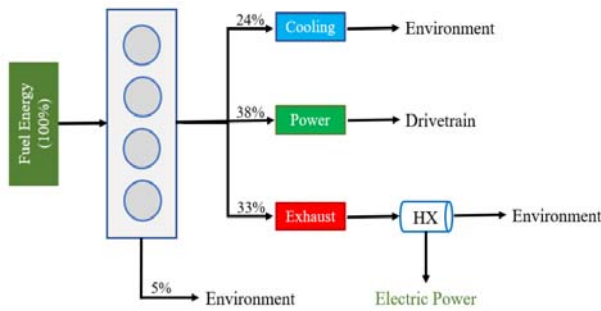


Figure 1. The distribution of the fuel energy in a typical ICE

TEGs are operating according to the Seebeck law in which electrical power is generated when there is a temperature difference between the semiconductor surfaces. The internal structure of TEGs includes p-type and n-type semiconductors for electricity generation, copper plates for electrical connection, and ceramic plates for insulation (Snyder & Ursell, 2003). The internal structure of TEGs can be seen in Figure 2.

The efficiency of the TEGs is calculated using Figure of Merit (ZT) which is defined as [6]:

$$ZT = \frac{\alpha^2 \cdot T}{\sigma \cdot \delta} \quad (1)$$

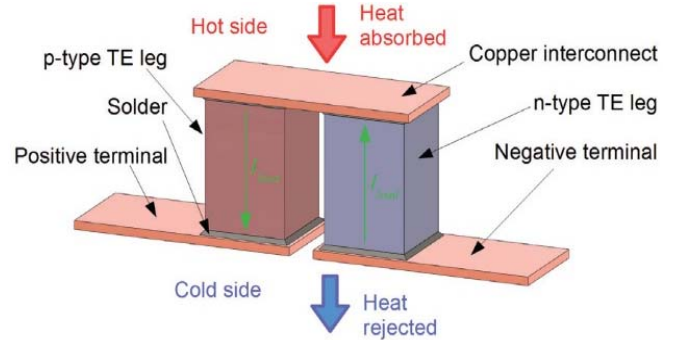


Figure 2. The internal structure of TEGs (Kempf & Zhang, 2016).

In which α is the Seebeck coefficient, T is the temperature difference between the TEG surfaces, σ is the electrical resistivity, and δ is the thermal conductivity. In order to calculate the efficiency of the TEG modules, the Carnot efficiency term was used considering the ZT value and the efficiency correlation is as follows:

$$\eta_{TEG} = \left(\frac{\sqrt{1 + ZT_{avg}} - 1}{\sqrt{1 + ZT_{avg}} + \frac{T_c}{T_H}} \right) \times \frac{(T_H - T_c)}{T_H} \quad (2)$$

Where, the ZT_{avg} indicates that the average ZT value at the average T . Also, T_c and T_H were the cold and hot surface temperatures of the TEG. Application of TEGs on the surfaces of the exhaust pipe is an effective method to produce electrical power from heat energy (Shen et al., 2019). Mastbergen et al. used both experimental and numerical methods to evaluate the efficiency of TEGs. The extended surfaces had been installed on TEG hot and cold surfaces to create a temperature difference. The working fluid for both heating and cooling purposes was air and the material of p-type and n-type semiconductors was Bi₂Te₃. The efficiency of TEG was calculated at around 1% during the test conditions. The deviation between the numerical results and experimental results has been reported at about 15% (Mastbergen et al., 2005).

Hsiao et al. have developed a mathematical model to evaluate the TEG application in automotive exhaust systems. The computer-aided evaluation method has been used to calculate the efficiency of TEGs. According to the results of this study, the maximum power density of TEGs has been measured as 51.13 mW/m³ (Hsiao et al., 2010).

The application of heat exchangers on the exhaust pipe of vehicles is a method used to enhance TEG performance. Karri et al. have manufactured a heat exchanger model that had finned surfaces assembled on the exhaust pipe. The effect of

TEG on the overall fuel economy of two different vehicles has been investigated. The obtained results have revealed that a fuel economy of about 4% could be achieved using the TEGs on the exhaust system (Karri et al., 2011a).

Evaluation of the heat exchanger performance has been conducted numerically by Wang and Liu in order to obtain a uniform temperature distribution on the surfaces of the heat exchangers. It has been suggested that the efficiency of the TEGs can be improved by the application of an alternative finned surface orientation in the heat exchangers mounted on the exhaust system (Wang & Liu, 2014a). Computational Fluid Dynamics (CFD) model was employed by Wang and Ma to study the increase in the efficiency of the heat exchanger with a spiral shape extended surface installed on the exhaust pipe. The efficiency of the heat exchangers had been increased by about 8.4% resulting in an increase in the efficiency of the TEGs by about 2% (Wang & Liu, 2014b).

In order to minimize the overall cost and complexities in the application of the TEG in vehicles, it has been suggested that the surface of the exhaust muffler of the automobiles be considered as heat exchanger (B. Huang & Shen, 2022). It has been shown that one of the immediate benefits of this method was that the increase in back pressure in the exhaust system of the vehicle has been canceled by applying the TEGs on the muffler surface (Hewawasam et al., 2020).

Enhancing the thermal efficiency of the TEGs directly relies on maintaining the uniform surface temperature on each side of the heat exchanger. To achieve this condition, Huang et al. have developed a new heat exchanger model that includes the phase change material (PCM) inside the hot and cold surfaces. Theoretical calculations resulted in a 15% increase in the thermal efficiency. However, in order to maintain a uniform temperature between the surfaces of the HX, a portion of the harvested heat energy from the exhaust was consumed by the PCM material. Consequently, the second law efficiency of the system decreased (K. Huang et al., 2021).

A comprehensive study of the literature showed that the commercialization of TEGs in automobiles could be possible by addressing issues such as non-uniform temperature distribution on the surfaces of heat exchangers, pressure drop effect, low thermal efficiency of TEGs, and the required cooling source. This study aims to tackle some of these issues by means of optimization techniques as well as numerical and experimental analysis. Additionally, due to the fact that the power produced by the turbine of the turbocharger is more than enough to just drive the compressor, a generator was mounted on the turbocharger shaft to harness the excess power. Therefore, a novel Combined Turbine-Peltier system (CTPS) was developed by combining a TEG system for waste heat recovery.

MATERIAL AND METHODS

Experimental Test Setup

The developed CTPS was a combination of two HXs, 20 pieces of Bi₂Te₃-based TEGs, a turbine-generator system, a water reservoir, and a cooling pump. The exhaust gas was passed through a heating channel acting as the hot reservoir and the water was circulated via a pump through the cooling channel and a water reservoir. The TEGs were sandwiched between the

heating and cooling channels. In order to obtain a comparable temperature distribution for TEGs, the crossflow orientation for the hot and cold sides was used. The turbine-generator was mounted immediately after the engine exhaust manifold to generate the electrical power using the kinetic energy of exhaust gas. Figure 3 indicates the overall system configuration.

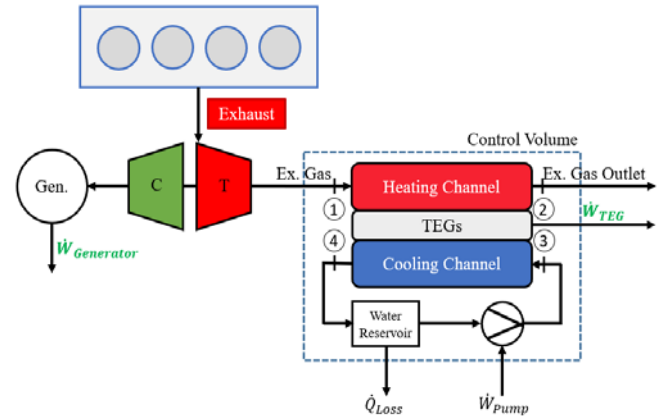


Figure 3. Configuration of the designed CTPS

The engine used was an 1150 cc Renault D4F spark ignition engine mounted on an AVL test bed shown in Figure 4. The engine speed and engine load were controlled with a dynamometer and the exhaust gas properties were measured by a set of thermocouples and mass flow controllers. The WHRS was connected immediately after the exhaust manifold to minimize the heat loss in the exhaust pipe. In the current study, the experimental data was collected after observing the steady temperature and mass flow rates of the exhaust gas at the test bench, and the delay time for the data gathering was measured at about 20 minutes. In order to check the validity of the experiments, more than ten measurements were recorded in the same environment and boundary conditions and the average value of the was presented in the Tables and Figures.



Figure 4. The experimental test setup

In the developed CTPS, the electrical power was produced by both WHRS and the turbine-generator system. The produced power from a turbine-generator was 3-phase due to the principles of the generators. Considering the regulation, all components located in the vehicle should operate with single-phase or DC. Thus, the produced 3-phase electrical power was converted to single-phase by means of an h-bridge constructed with six diodes. Figure 5 shows the turbine-generator circuit schematics and device in which the "K" signs the A2212 brushless DC motor, the "C" signs the multimeter, and the "L" signs the h-bridge.

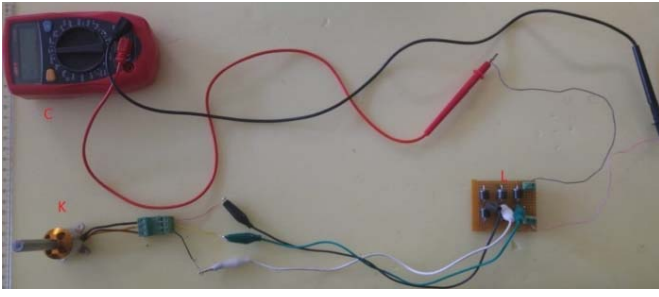
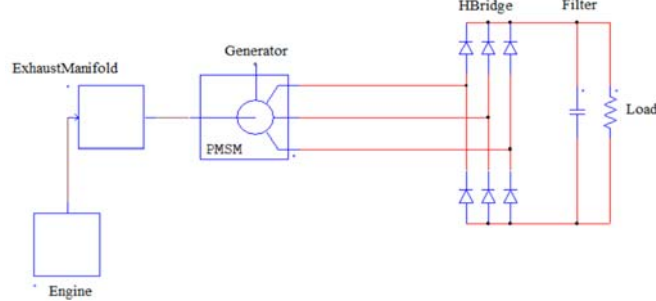


Figure 5. Turbine-Generator circuit schematic and device

In the analysis of thermal systems, whereas the general evaluation method calculates the thermal efficiency using the first law approaches, the exact performance indicator of the system is the second law efficiency since it gives a system whether effective or not. Therefore, to calculate the second law efficiency of the control volume that is presented in Figure 3, the availability energy of the exhaust gas and water was calculated using the Equation 3. Moreover, the overall second law efficiency of the CTPS was evaluated by employing the combined second law efficiencies of HX and TEGs as given in Equations 4 and 5, respectively. So, the overall second law efficiency of the system was calculated using the Equation 6 (Patowary & Baruah, 2018).

$$\gamma = \dot{m} \times [(h - h_0) - T_0 \times (s - s_0)] \quad (3)$$

$$\mu_{HX} = \frac{\dot{W}_{HX}}{\gamma_1 - \gamma_2} \quad (4)$$

$$\mu_{TEG} = \frac{\dot{W}_{TEG}}{(\gamma_2 - \gamma_1) - (\gamma_4 - \gamma_3)} \quad (5)$$

$$\mu_{II,overall} = \eta_{HX} \cdot \eta_{TEG} \quad (6)$$

Where $\dot{W}_{HX}$ is the power of the HX, $\dot{W}_{TEG}$ is the power of the TEG, η_{HX} and η_{TEG} are their efficiencies, respectively. In equation 6, the efficiency is based on the total produced power in the TEG system. Hence, to calculate the total power and net power values, Equation 7 was used as follows:

$$\dot{W}_{net} = \dot{W}_{TEG} + \dot{W}_{Generator} - \dot{W}_{Pump} \quad (7)$$

Numerical Study of CTPS

In the HX studies, the vane thickness, number of vanes, and gap size are directly related to the effectiveness of the HX, and these parameters should be selected considering the types of applied boundary conditions (Borcuch et al., 2017; Moon & Kim, 2014; Söylemez, 2003). In order to design an optimized HX model, the extended surface dimensions for the exhaust gas heating channel were selected using the Taguchi L9 (3 by 3) optimization method since the Taguchi model calculates the sensitivity of the design parameters on the output parameters by changing of the design points systematically. So, the

required analysis points could be reduced from 27 to 9. The selected vane heights were 10, 20, and 40 mm considering the hydraulic diameter of the exhaust pipe, and the gap between the vanes was selected as 5 and 10 mm. A gap size larger than 10 mm was not considered due to limitations in the heat exchanger manufacturing process. During the optimization studies, the cooling channel dimensions were kept constant according to the previous studies (Kara et al., 2019; Tekin et al., 2019). The cooling water flow rates of 0.1, 0.25, and 0.5 kg/sec were preferred in order to obtain minimum power consumption by cooling pump. The selected control factors and levels for Taguchi L9 are given in Table 1. The output parameters in the optimization were the pressure drop and the temperature difference between TEG surfaces which was chosen to be around 120°C to obtain maximum efficiency based on studies in the literature (Patil et al., 2018; Patowary & Baruah, 2018). The designed Taguchi L9 optimization matrix can be seen in Table 2.

Table 1. The Taguchi design parameters

Control Factors	Levels
Exhaust Channel Vane Height [mm]	10, 20, 40
Vane Gap [mm]	5, 10
Water Flow Rate [kg/sec]	0.1, 0.25, 0.5
Output Parameters	
Temperature Difference between the Surfaces of TEGs [K]	The desired value is 120
Pressure Drop [Pa]	The desired value is 0.1

Table 2. The Taguchi L9 design matrix

Experiment Number	Control Factors		
	Vane Height [mm]	Vane Gap [mm]	Water Flow Rate [kg/sec]
Exp. 1	10	5	0.1
Exp. 2	10	10	0.25
Exp. 3	10	5	0.5
Exp. 4	20	5	0.5
Exp. 5	20	10	0.1
Exp. 6	20	10	0.25
Exp. 7	40	5	0.25
Exp. 8	40	10	0.5
Exp. 9	40	5	0.1

In order to obtain the optimal HX model, CAD models were created for each of the experiments according to the Taguchi design table which can be seen in Figure 6. The numerical analyses were conducted utilizing the Ansys Fluent software.

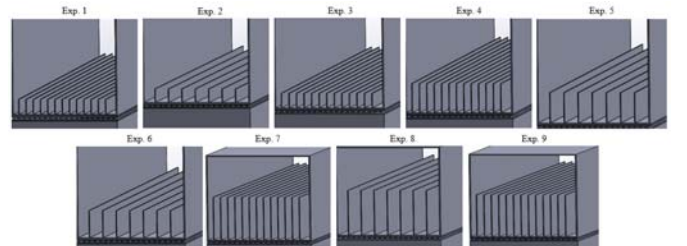


Figure 6. The heating channels of the HX for the numerical analysis

The obtained data from the performance test of the engine were applied as boundary conditions in the numerical analysis as shown in Table 3. In the numerical study, the air properties were used in the heating channel due to the fact that exhaust

gas properties are comparable to that of air and the working fluid in the cooling channel was water. In the computational fluid dynamics (CFD) analyses of the heat exchangers, the Reynold-Average Navier Stokes approach (RANS) was preferred by selecting the k-epsilon viscous model since this viscous model provides an appropriate near-wall modeling for the heat transfer solution in the conjugate heat and fluid problems. To estimate the actual heat flux and temperature distributions over the HX surfaces, the time-dependent analysis type was selected and the analysis duration was 3600 seconds based on the suggestions from the literature (Harun et al., 2018; Kutt & Lehtonen, 2015; Liu et al., 2014).

Table 3. Boundary conditions of the numerical study

Parameters	Cooling Channel	Heating Channel
Fluid Type	Liquid Water	Air
Inlet Temperature	303 [K]	753 [K]
Dynamic Viscosity	0.7978 [mPa.s]	3.48e-05 [kg/m.s]
Channel Material	Aluminum	Aluminum
Viscous Type	k-epsilon Realizable	k-epsilon Realizable
Y+ Value	30	30

RESULTS AND DISCUSSION

In order to estimate the system efficiency and optimization of the HX model that was designed for cooling and heating purposes, initially, the engine was tested without the CTPS. After benchmarking the engine in the test bed, the engine exhaust gas properties were measured considering the literature recommendations (Champier, 2017; Yang, 2005; Zhou et al., 2014). In order to obtain the engine exhaust gas flow rate and temperature values, the AVL test bed was operated using the speed/alpha mode. In this mode, to measure the exhaust gas and engine properties, the butterfly angle was increased continuously by about 5% until the torque value of 50 N-m was achieved. The preferred torque value was obtained in full throttle at 2000 rpm, and in 70% and 50% of butterfly angles at 2500 and 3000 rpm, respectively. Table 4 indicates the engine exhaust gas properties at different engine speeds.

Table 4. The engine output results at different speeds

Parameters	2000 [rpm]	2500 [rpm]	3000 [rpm]
Exhaust Gas Flow Rate [kg/sec]	0.0075	0.0088	0.0103
Exhaust Gas Temperature [K]	753	789	818
Engine Torque [N-m]	50	50	50
Brake Power [kW]	11.7	13.1	15.7

In the Taguchi analysis method, the effects of the control factors on the output parameters are assessed by changing the control factor levels. In this way, whether the selected control and output factors are suitable or not could be checked. The selected output control parameters were the temperature difference between the surfaces of the HX and the pressure drop in the exhaust channel. For the CFD analysis of the Taguchi L9 array, the 2000 rpm engine speed was chosen as a boundary condition because not only it was expected that the minimum temperature difference and exhaust gas mass flow rate would occur at this speed but also to optimize and increase the effectiveness of the CTPS system considering the minimum conditions.

The CFD analysis was carried out for all the experiments based on the control factors suggested by the Taguchi optimization method in order to obtain the temperature difference and pressure drop values for each of the models. Initially, the mesh independency tests were carried out with four different element numbers ranging from 0.8 up to 6.5 million cells. As can be seen in Figure 7a, the mesh size of about 5.3 million cells provided sufficient accuracy for the numerical model. The created mesh structure is also presented in Figure 7b.

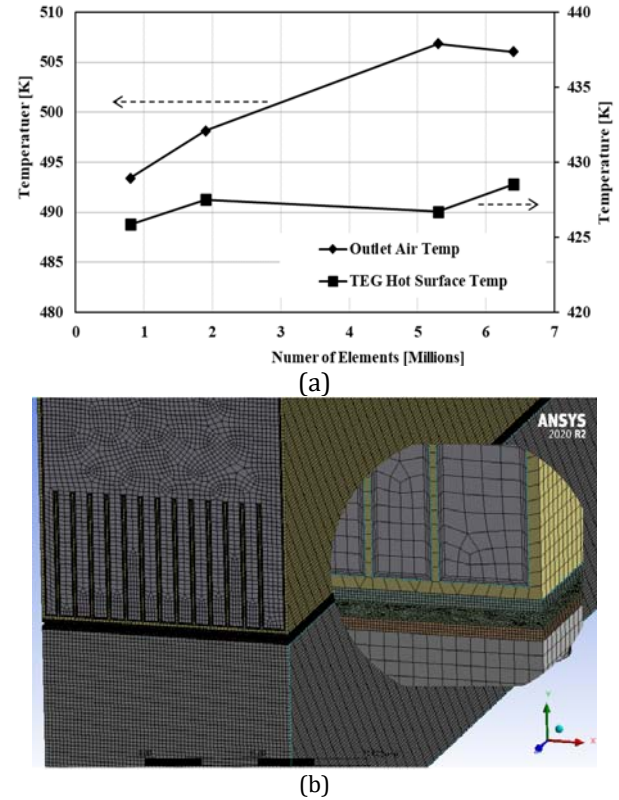


Figure 7. The mesh independence test results (a) and the constructed grid (b)

Figure 8 shows the temperature distribution on the hot and cold surfaces of TEGs. It was seen that while the temperature over the hot surface decreased from the inlet to the outlet, the cold surface temperature increased from the inlet to the outlet since the cross-flow method was selected. The surface temperature difference on the TEGs was in the range of about 85 to 190 K with an average value of around 113 K which is in the optimum range of TEG operational conditions (Patowary & Baruah, 2018). The results for the average temperature difference on the TEG surface as well as the pressure drop for the experiments are presented in Table 5. These results were used to assess the optimum dimensions and cooling flow rate in the Taguchi optimization method.

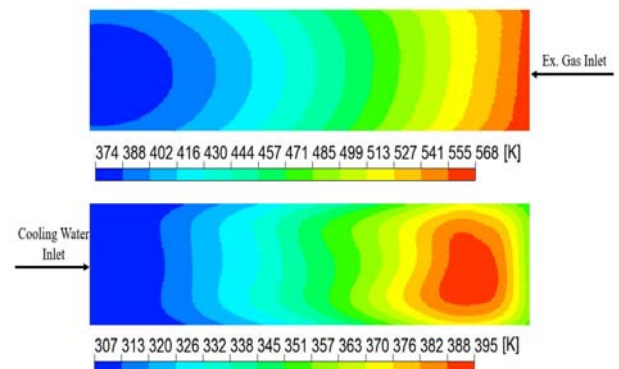


Figure 8. The temperature distribution on the hot and cold surfaces of TEGs

In order to assess the effect of control factors on temperature difference between TEG surfaces and pressure drop in the heating channel, the signal-to-noise (S/N) analysis was performed, and the results are presented in Figure 9. Considering the obtained data from S/N analysis for the temperature difference, it was seen that the effect of vane height was more significant than the other two factors. Increasing the vane height to 40 mm resulted in an increase in temperature difference by about 17% above the mean values. However, the effect of change in the vane distance and cooling flow rate was less than 5%. The same results were also noticed for pressure drop values as well. It can be concluded that the most important control factor on the HX was the vane height.

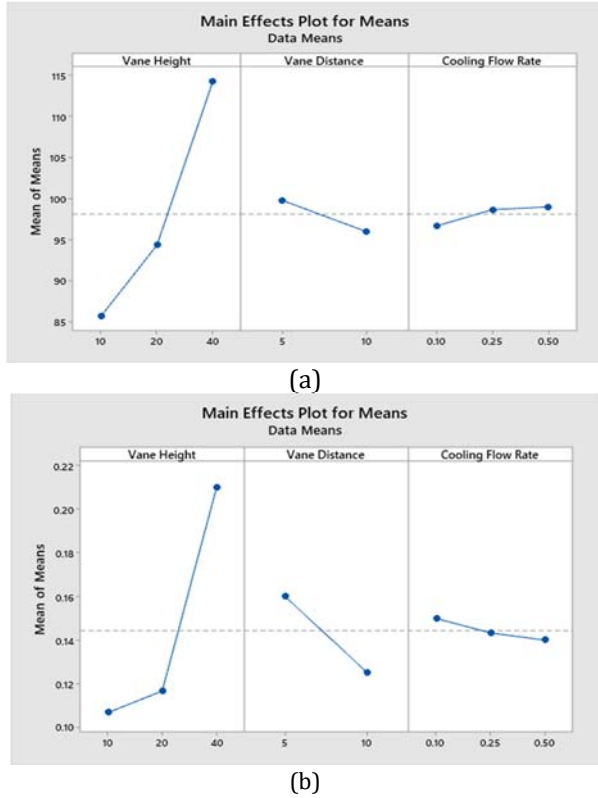


Figure 9. The S/N curves for (a) temperature difference and (b) pressure drop

Table 5 shows the temperature difference and pressure drop results for the different control factors. It was seen that the highest temperature difference was obtained at Exp. 7 at about 117 K and the lowest pressure drop was measured at Exp. 2 at about 0.10 Pa. Reducing the cooling flow rate from 0.25 kg/sec in Exp. 7 to 0.1 kg/sec in Exp. 9 resulted in a temperature difference decrease by about 4 K, however, the required pumping power for cooling water was also decreased significantly. When the obtained results from the numerical analysis were investigated together as given in Table 5 and Figure 8, to obtain the maximum temperature differences between the TEG surfaces, the hot channel vane height should be 40 mm. Moreover, the vane distance changes strongly affect the pressure drop more than the temperature differences, the vane distance was selected as 5 mm since the pressure drop in the channel was negligible considering the environmental pressure was about 101,3 kPa. When the sensitivity of the cooling flow rate on the temperature differences and pressure drop were examined, the results showed that the cooling flow rate was insensitive to the output parameters. So, to reduce the required power for the circulation of the cooling fluid in the cold channel, the cooling flow rate was selected as 0.1 kg/sec. Considering the results, Exp. 9 with a vane height of 40 mm, vane distance of 5 mm, and cooling flow rate of 0.1 kg/sec was preferred as optimal geometry for the experimental analysis and manufacturing. Moreover, when the vane height, hot surface temperature, and pressure drop results were investigated together, the increase rate of the vane height was directly proportional to the increase rate of the temperature and pressured drop due to the flow area being reduced and the high turbulence intensity being occurred.

After obtaining the optimum HX model from optimization studies, the CFD studies of the CTPS were carried out by defining the engine test results given in Table 4 as boundary conditions. During the analysis, three exhaust gas flow rates and temperatures obtained under three engine speeds measured at constant torque were used.

Considering the results, when the exhaust gas flow rate increased at higher engine speeds, the temperature difference and pressure drop increased by about 12% and 25%. The numerical analysis results of the CTPS are listed in Table 6.

Table 5. The obtained numerical results for the temperature difference and pressure drop

Experiment Number	Control Factors			Result Factors	
	Fin Height [mm]	Fin Distance [mm]	Cold Flow Rate [kg/sec]	Temp. Diff. [K]	Pressure Drop [Pa]
Exp. 1	10	5	0.1	85	0.11
Exp. 2	10	10	0.25	85	0.10
Exp. 3	10	5	0.5	87	0.11
Exp. 4	20	5	0.5	97	0.13
Exp. 5	20	10	0.1	92	0.11
Exp. 6	20	10	0.25	94	0.11
Exp. 7	40	5	0.25	117	0.22
Exp. 8	40	10	0.5	113	0.18
Exp. 9	40	5	0.1	113	0.23

Table 6. The numerical results of the optimum HX model

Parameters	2000 [rpm]	2500 [rpm]	3000 [rpm]
Temperature Difference [K]	108.8	123.3	147.5
Pressure Drop of Heating Channel [Pa]	0.18	0.24	0.35
Outlet Air Temperature [K]	528.3	561.6	584
Outlet Water Temperature [K]	305.3	305.7	305.9
Inlet Water Temperature [K]	303	303	303
Inlet Air Temperature [K]	753	798	818
Water Flow Rate [kg/sec]	0.1	0.1	0.1

The obtained results from the experimental study as given in Table 6 were used to calculate the first and second law efficiencies of the TEG system by using Equations 3 to 6 and the calculated results are presented in Table 7. The exergy (γ) was calculated considering the environmental condition and the produced energy was collected from the experiment. It was seen that the first and second-law efficiencies were increased with the engine speed due to the increase in the mass flow rate and converted energy from the exhaust gas. Moreover, the selected control volume for the exergy analysis is given in Figure 3.

Table 7. The first and second law efficiencies of the TEG system

Parameters	2000 [rpm]	2500 [rpm]	3000 [rpm]
Calculated Power Output of TEGs [W]	68.5	101.7	148.9
Exergy of Inlet Air [W]	1411.4	1866.9	2395.9
Exergy of Outlet Air [W]	468.9	680.9	911.5
Reversible Energy of Air [W]	942.5	1186.0	1484.4
Reversible Energy of Water [W]	61.0	102.3	113.2
Second Law Efficiency of TEGs [%]	7.27	8.57	10.03
Theoretical First Law Efficiency of TEGs [%]	4.86	5.41	6.36
Calculated First Law Efficiency of TEGs [%]	2.6	3.0	3.6

In order to reveal the performance increases with the CTPS in the WHRS, the already shared similar studies were compared with the current study as given in Table 8. The measured or calculated thermal efficiencies gradually increase in the thermoelectric generator studies by year because of the optimized utilization and obtaining the smooth temperature gradient over the HX surfaces. The comparison parameters for the WHRS studies that are

The maximum thermal efficiency of the TEGs was 3.6% and the maximum second law efficiency was about 10% obtained at 3000 rpm. The calculated maximum theoretical efficiency of TEGs based on Equation 1 was 6.36% which was about 43% higher than the obtained thermal efficiency. This was because in theoretical calculations, the temperature distribution on the TEG surfaces was considered to be uniform, and the Seebeck coefficient and thermal conductivity of the materials were assumed to be identical.

related to TEGs were the produced power quantity from the single TEG and the thermal efficiency of the TEGs. Therefore, the singular TEG power quantity and thermal efficiency were used in Table 8. Considering the obtained results, the thermal efficiency was higher than the study conducted in 2022 by about 32.2%. Additionally, the produced power quantity from the single TEG was greater than the study conducted in 2011 at around 27.2%.

Table 8. Comparison of the results for the designed TEG with the results in the literature

Reference	Year Published	TEG Material	Method	Output Power per a TEG [W]	Thermal Efficiency [%]
(Ikoma et al., 1998)	1998	Si-Ge	Experimental	2.8	0.9
(Mastbergen et al., 2005)	2005	Bi_2Te_3	Experimental	2.1	1.0
(Espinosa et al., 2010)	2010	$\text{Mg}_2\text{Si}/\text{Zn}_4\text{Sb}_3$ - Bi_2Te_3	Simulation	1.2	0.8
(Hsiao et al., 2010)	2010	Bi_2Te_3	Experimental	1.24	0.4
(Karri et al., 2011b)	2011	Bi_2Te_3	Experimental	4.0	1.26
(Liu et al., 2014)	2014	Bi_2Te_3	Simulation and Experimental	1.0	1.4
(B. Huang & Shen, 2022)	2022	PbSn	Simulation	0.4	2.44
Current Study	-	Bi_2Te_3	Simulation and Experimental	5.5	3.6

After conducting the numerical analysis, the HX was manufactured and assembled on the experimental test setup. During the experimental studies, the engine was operated at three different speeds and four different engine torques. All TEGs were connected in series and the total voltage and current were measured utilizing a rheostat as a load resistor. The same method was used for turbine-generator output power, as well. The consumed power of the cooling pump was at a constant value of 9.7W with a mass flow rate of 0.1 kg/sec. The obtained results showed that the power output value of the turbine generator and TEGs were directly proportional to the increment of the engine speed and torque values. Under the no-load condition, the power generated by TEGs was significantly higher than the turbine-generator system. However, by increasing the engine torque, the rate of increase in power generated by the turbine compressor was much higher than that of TEGs. The maximum net power output values measured at 2000, 2500, and 3000 rpm engine speeds were 90.6, 132.7, and 190.4 W, respectively.

The CTPS power ratio defined as the percentage of increase in the engine brake power was also calculated. Even though the highest power output of 190.4 W was measured under 75 N-m load at 3000 rpm, the maximum CTPS power ratio was evaluated as 1.6% under 25 N-m load at 3000 rpm. The reason

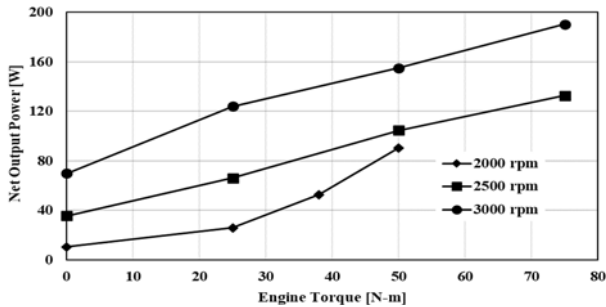
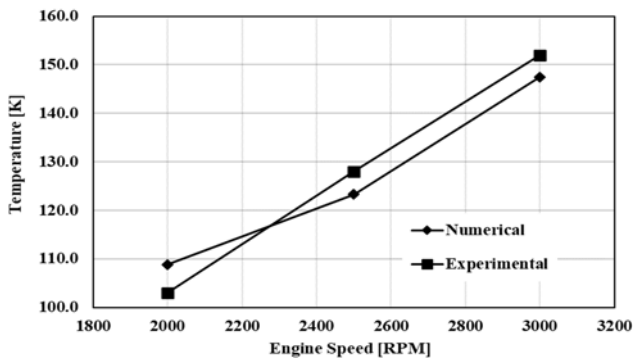
for this condition can be clarified with the ZT value of the TEGs because the efficiency of the CTPS system is directly proportional to TEGs efficiency. The measured experimental results are presented in detail in Table 9.

In order to illustrate the relationship between the waste heat recovery systems and the engine speed, the net power output values of the CTPS are given in Figure 10. It can be seen that the net power output of the system had a linear correlation with the engine torque, especially at higher engine speeds. This linear trend was not followed at 2000 rpm due to the fact that at this speed the maximum torque of 50 N-m was achieved at fully open throttle conditions and the exhaust gas temperature was at its maximum.

In order to validate the numerical model, the deviation between the simulation results and experimental data was compared. The numerical results were conducted by applying the exhaust gas flow rate and temperature values which were obtained for engine speeds of 2000 to 3000 rpm at 50 N-m engine load. The maximum temperature deviation between the experimental data and numerical results was around 6%. Figure 11 shows the comparison between the experiments and the numerical results at different engine speeds.

Table 9. The experimental test results of the CTPS

Engine Speed [rpm]	Engine Torque [N-m]	Engine Brake Power [kW]	Temp. Difference [K]	TEGs Ratings [V-A]	TEGs Power Output [W]	Turbine-Generator Power [W]	Cooling System Power [W]	Net Power Output [W]	CTPS Power Ratio [%]
2000	0	1.9	62	47.7-0.3	14.3	6	9.7	10.6	0.6
2000	25	5.2	85	56.6-0.42	23.7	12	9.7	26.0	0.5
2000	38	10.5	103	68.2-0.55	37.5	25	9.7	52.8	0.5
2000	50	15.7	118	80.4-0.65	52.3	48	9.7	90.6	0.6
2500	0	3.1	95	62.8-0.53	33.2	12	9.7	35.5	1.1
2500	25	6.5	112	77.4-0.62	47.9	28	9.7	66.2	1.0
2500	50	13.1	128	92-0.7	64.4	50	9.7	104.7	0.8
2500	75	19.6	141	99.2-0.78	77.4	65	9.7	132.7	0.7
3000	0	4.7	115	78.5-0.63	49.5	30	9.7	69.8	1.5
3000	25	7.9	142	99.6-0.79	78.7	55	9.7	124.0	1.6
3000	50	15.7	152	112-0.82	91.8	73	9.7	155.1	1.0
3000	75	23.6	168	117-0.95	111.1	89	9.7	190.4	0.8

**Figure 10.** Illustration of the net power output of the CTPS**Figure 11.** Comparison of the temperature between the experimental and numerical results

CONCLUSION

In this study, the optimization methods, analysis, and experimental approaches were employed to design and manufacture a novel Combined Turbine-Peltier System (CTPS) to harness the energy of the exhaust gases from an ICE operating under different conditions. The obtained results can be summarized as follows:

- ❖ In the design of HX, the effect of vane height was more significant than the vane distance and the cooling flow rate. Increasing the vane height to 40 mm resulted in an increase in temperature difference by about 17% above the mean values.
- ❖ By increasing the engine speed from 2000 rpm to 2500 rpm and then to 3000 rpm, the exhaust gas flow rate increased and resulted in a higher temperature difference between TEG surfaces by about 12% and 25%.
- ❖ The first and second law efficiencies were directly proportional to the engine speed and the maximum thermal efficiency of the TEGs was 3.6% and the maximum second law efficiency was about 10% obtained at 3000 rpm. Also, the maximum obtained power from a single TEG was 5.5W.

- ❖ Under the no-load condition, the power generated by TEGs was significantly higher than the turbine-generator system. However, by increasing the engine torque, the rate of increase in power generated by the turbine-compressor was much higher than that of TEGs. The maximum net power output values measured at 2000, 2500, and 3000 rpm engine speeds were 90.6, 132.7, and 190.4W, respectively.
- ❖ The highest CTPS power ratio was evaluated as 1.6% under 25 N-m load at 3000 rpm.
- ❖ The maximum temperature deviation between the experimental data and numerical results was around 6%.

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