Energy 65 (2014) 44–51

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Exergy recovery from the exhaust cooling in a DI diesel engine for BSFC reduction purposes

Mohsen Ghazikhani [a], Mohammad Hatami [b,*], Davood Domiri Ganji [b], Mofid Gorji-Bandpy [b], Ali Behravan [a], Gholamreza Shahi [a]

[a] Ferdowsi University of Mashhad, Department of Mechanical Engineering, Mashhad, Iran
[b] Babol University of Technology, Department of Mechanical Engineering, Babol, Iran

ARTICLE INFO

Article history:
Received 31 March 2013
Received in revised form
29 November 2013
Accepted 3 December 2013
Available online 25 December 2013

Keywords:
Diesel engine
Exhaust heat exchanger
Exergy recovery
Irreversibility
Second law efficiency

ABSTRACT

In this experimental research, the exergy recovery from a DI Diesel engine is investigated where a turbocharged OM314 DIMLER diesel engine was tested at various engine speeds (1200, 1400, 1600, 1800 and 2000 rpm) and torques (20, 40, 60, 80 and 100 N m). For this aim, a double pipe heat exchanger with counter current flow is used in the exhaust of engine. As an important outcome, by increasing the load and engine speed, the recovered exergy increased. Furthermore, the reduction of brake specific fuel consumption (bsfc) due to the use of recovered exergy from exhaust has also been studied in the current study. The results show that by using recovered exergy, bsfc decreased approximately 10%.

1. Introduction

The importance of using energy losses, because of energy sources reduction, is obvious. Exhaust has 25–30% losses from energy engine generate. However, the wasted energy cannot be fully recovered, but using heat exchanger can be a good method for heat recovery. Pandiyarajan et al. [1] experimentally studied heat recovery of finned shell and tube heat exchanger and storage of excess heat with using a thermal energy storage tank in an IC engine. They found that about 10–15% of fuel power is stored as heat in the combined storage system, which is available at higher temperature for suitable application. Kauranen et al. [2] studied diesel engine heating up with exhaust gas heat recovery at subzero temperatures where the average speed is low and the drive cycle includes much idling. In another experimental research, Yang et al. [3] studied heat recovery of heat pipe exhaust heat exchanger for heating up a large bus (HS663). Lee and Bae [4] designed an exhaust heat exchanger between the exhaust manifold and the inlet of catalytic converter to reduce the exhaust temperature with using a Design of Experiments (DOE) technique in an SI engine.

Heat exchangers also are used in diesel engines for reducing exhaust sound level as a muffler. Wonnacott [5] designed a better muffler with lower exhaust noise. Prasad and Crocker [6] studied acoustical performance model of a multi-cylinder engine exhaust muffler system to predict insertion loss and radiated sound pressure level of exhaust. Nakra et al. [7] experimentally studied reactive types of muffler to measure noise attenuation characteristics and the frequency spectra of attenuation levels were compared with corresponding theoretical predictions. Also, Alfredson and Davies [8] investigated performance of the muffler components and compared it with one-dimensional linearized theory. Muffler in the path of exhaust gas creates back pressure to the engine that causes an increase in the pumping friction in the engine [9]. The most important acoustic property of a muffler is its transmission loss which is defined as the difference between the output and input noise amplitude for a given frequency [10].

In this experimental work a muffler equipped with a heat exchanger with water as coolant fluid is used. For estimating the different kinds of irreversibility such as total irreversibility, irreversibility of heat transfer and internal irreversibility, exergy analysis in the muffler with and without heat exchanger at various engine operating conditions have been used. The results reveal that recovered exergy can decrease the amount of bsfc. The schematic of turbocharged OM314 DIMLER diesel engine is shown in Fig. 1.

* Corresponding author.
E-mail addresses: m.hatami2010@gmail.com, m.hatami@stu.nit.ac.ir (M. Hatami).

0360-5442/$ – see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.energy.2013.12.004

Nomenclature

BSFC	brake specific fuel consumption (g/kW h)
C_f	corrected brake power factor
C_p	specific heat in constant pressure (J/kg K)
DI	direct injection
h	specific enthalpy (J/kg)
HEX	heat exchanger
IC	internal combustion
$\dot{I}$	irreversibility (W)
K_e	kinetic energy (J)
$\dot{m}$	mass flow rate (kg/s)
N	engine speed (rpm)
P_b	brake power (W)
P_e	potential energy (J)
p	pressure (Pa)
$\dot{Q}$	heat transfer rate (W)
S	entropy (J/K)
SSSF	steady state-steady flow process
T_j	temperature of source j (K)
T	time (s)
W	work (J)

Greek symbols

ε	second law efficiency (%)
$\dot{\Phi}$	non-flow availability (W)
$\dot{\sigma}$	entropy generation rate (kJ/kg)
τ	torque (N m)
ψ	flow availability (J/kg)

Subscripts

0	restricted dead state
1	exhaust inlet to heat exchanger
2	exhaust outlet to heat exchanger
3	water inlet to heat exchanger
4	water outlet from heat exchanger
a	air
f	fuel
act	actual
cv	control volume
down	downstream or outlet flow
e	exit from control volume
exh	exhaust
i	inlet to control volume
m	measurement condition
Q	heat transfer parameter
s	standard condition
up	upstream or inlet flow

2. Exergy analysis

According to Fig. 2 and these assumptions:

a) Process is in steady state condition.
b) The work of control volume is zero.
c) Kinetic and potential energy change are negligible.
d) Specific heat of exhaust gases is temperature-dependent.
e) Combustion products are assumed as an ideal gas such as air.

the exergy equations in the muffler by using a heat exchanger can be driven as follows [11]:

$$\frac{d\Phi_{cv}}{dt} = \sum \dot{\Phi}_Q + \sum \dot{m}_i\psi_i - \sum \dot{m}_e\psi_e + \dot{W}_{act} - \dot{I}_{total} \tag{1}$$

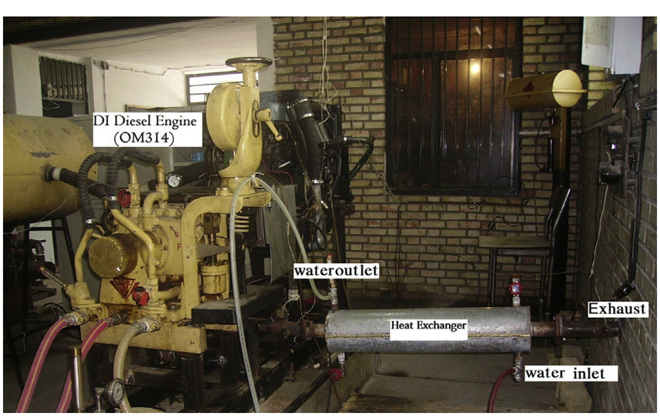

Fig. 1. DIMLER DI diesel engine (OM314).

In Eq. (1), $d\Phi_{cv}/dt$ is the non-flow exergy of the control volume that is zero in the steady state process.

$$\frac{d\Phi_{cv}}{dt} = 0 \tag{2}$$

Since $T_j = T_0$, therefore

$$\sum \dot{\Phi}_Q = \sum \dot{Q}\left(1 - \frac{T_0}{T_j}\right) = \sum \dot{Q}\left(1 - \frac{T_0}{T_0}\right) = 0 \tag{3}$$

The work of control volume is zero, so

$$\dot{W}_{act} = 0 \tag{4}$$

Substituting Eq. (2)–(4) into Eq. (1), the exergy equation will be as

$$\begin{aligned}\dot{I}_{total} &= \dot{m}_i\psi_i - \dot{m}_e\psi_e = \dot{m}_3\psi_3 + \dot{m}_1\psi_1 - \dot{m}_2\psi_2 - \dot{m}_4\psi_4 \\ &= \dot{m}_{water}(\psi_3 - \psi_4) + \dot{m}_{exh}(\psi_1 - \psi_2)\end{aligned} \tag{5}$$

where

$$\dot{m}_{exh} = \dot{m}_a + \dot{m}_f \tag{6}$$

Also, we have

$$\Delta\psi = \Delta h - T_0\Delta S + \Delta K_e + \Delta P_e \tag{7}$$

Kinetic and potential energy change are negligible and thus Eq. (7) becomes

$$\Delta\psi = \Delta h - T_0\Delta S \tag{8}$$

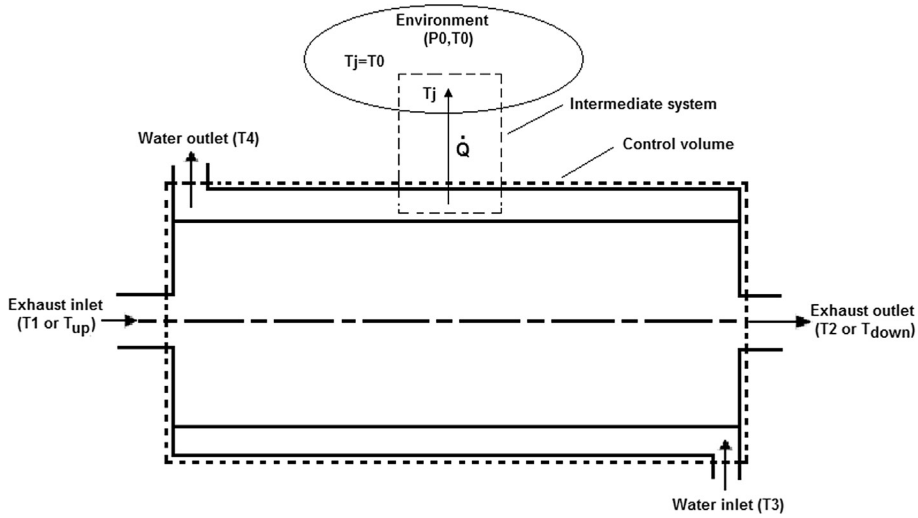

Fig. 2. Control volume of Muffler with heat exchanger.

$\Delta h_{\mathrm{water}}$ and $\Delta S_{\mathrm{water}}$ are produced from thermo dynamical tables with using the inlet and outlet temperatures and the pressure of water.

To calculate the enthalpy changes ($\Delta h_{\mathrm{exh}} = h_{\mathrm{e}} - h_{\mathrm{i}}$) between the two sides of the muffler, Eq. (9) can be employed:

$$\mathrm{d}h = C_p \mathrm{d}T \tag{9}$$

Integrating this equation gives enthalpy changes in the muffler. Also the specific heat variation should be considered. The specific heat of gas is

$$C_{pm} = \sum n_i \overline{C_{pi}} = C_{pm}(T) \tag{10}$$

Since the equivalence ratio in diesel engine is less than one, the combustion products can be assumed as an ideal gas (air). So, C_p can be given as a function of temperature [12]:

$$C_p = 28.11 + 0.1967 \times 10^{-2} T + 0.4802 \times 10^{-5} T^2 - 1.966 \times 10^{-9} T^3 \tag{11}$$

Substituting Eq. (11) into (9) and integrating in limits of the internal temperature and external temperature of muffler, the enthalpy changes of the muffler can be obtained as Eq. (12):

$$\begin{aligned}\Delta h_{\mathrm{exh}} &= h_{\mathrm{e}} - h_{\mathrm{i}} = (h_{\mathrm{down}} - h_{\mathrm{up}}) = \int_{T_{\mathrm{up}}}^{T_{\mathrm{down}}} \mathrm{d}h = \int_{T_{\mathrm{up}}}^{T_{\mathrm{down}}} C_p \mathrm{d}T \Rightarrow h_{\mathrm{down}} - h_{\mathrm{up}} \\ &= \frac{1}{28.9}\Big[28.11(T_{\mathrm{down}} - T_{\mathrm{up}}) + 0.9835 \times 10^{-3}\left(T_{\mathrm{down}}^{-2} - T_{\mathrm{up}}^{-2}\right) \\ &\quad + 0.16 \times 10^{-5}\left(T_{\mathrm{down}}^{-3} - T_{\mathrm{up}}^{-3}\right) - 0.49 \times 10^{-9}\left(T_{\mathrm{down}}^{-4} - T_{\mathrm{up}}^{-4}\right)\Big]\end{aligned} \tag{12}$$

For calculating entropy changes in the muffler (Fig. 2), the combustion products are considered with variable specific heat. Therefore:

$$\begin{aligned}\Delta S_{\mathrm{exh}} &= \int_{T_{\mathrm{up}}}^{T_{\mathrm{down}}} \mathrm{d}s = \int_{T_{\mathrm{up}}}^{T_{\mathrm{down}}} \frac{C_p \mathrm{d}T}{T} - R\,\mathrm{Ln}\left(\frac{p_{\mathrm{down}}}{p_{\mathrm{up}}}\right) \\ &= \frac{1}{28.9}\Big[28.11\,\mathrm{Ln}\left(\frac{T_{\mathrm{down}}}{T_{\mathrm{up}}}\right) + 0.1967 \\ &\quad \times 10^{-2}(T_{\mathrm{down}} - T_{\mathrm{up}}) + 0.2401 \times 10^{-5}\left(T_{\mathrm{down}}^2 - T_{\mathrm{up}}^2\right) \\ &\quad - 0.655 \times 10^{-9}\left(T_{\mathrm{down}}^3 - T_{\mathrm{up}}^3\right)\Big] - R\,\mathrm{Ln}\left(\frac{p_{\mathrm{down}}}{p_{\mathrm{up}}}\right)\end{aligned} \tag{13}$$

From Eq. (9)–(13), the total irreversibility can be obtained as

$$\begin{aligned}\dot{I}_{\mathrm{total}} &= \dot{m}_{\mathrm{exh}}[(h_{\mathrm{i}} - h_{\mathrm{e}}) - T_0(s_{\mathrm{i}} - s_{\mathrm{e}})] \\ &= -\dot{m}_{\mathrm{exh}}\Bigg[\left(\frac{1}{28.9}\Big[28.11(T_{\mathrm{down}} - T_{\mathrm{up}}) + 0.9835 \times 10^{-3}\left(T_{\mathrm{down}}^{-2} - T_{\mathrm{up}}^{-2}\right) + 0.16 \times 10^{-5}\left(T_{\mathrm{down}}^{-3} - T_{\mathrm{up}}^{-3}\right) - 0.49 \\ &\quad \times 10^{-9}\left(T_{\mathrm{down}}^{-4} - T_{\mathrm{up}}^{-4}\right)\Big]\right) - T_0\left(\frac{1}{28.9}\Big[28.11\mathrm{Ln}\left(\frac{T_{\mathrm{down}}}{T_{\mathrm{up}}}\right) + 0.1967 \times 10^{-2}(T_{\mathrm{down}} - T_{\mathrm{up}}) + 0.2401 \\ &\quad \times 10^{-5}\left(T_{\mathrm{down}}^2 - T_{\mathrm{up}}^2\right) - 0.655 \times 10^{-9}\left(T_{\mathrm{down}}^3 - T_{\mathrm{up}}^3\right)\Big] - R\mathrm{Ln}\left(\frac{p_{\mathrm{down}}}{p_{\mathrm{up}}}\right)\right)\Bigg]\end{aligned} \tag{14}$$

Table 1
Specifications of diesel engine OM314.

Engine	Specification
Engine type	4 stroke diesel engine
Number of cylinder	4
Combustion chamber	Direct injection
Bore × stroke (mm)	97 × 128
Piston displacement (cc)	3784
Compression ratio	17:01
Maximum power (hp)	85
Maximum torque (N m)	235
Maximum speed (rpm)	2800
Mean effective pressure (bar)	6.8 @ 2800 rpm

where the total irreversibility consists of two parts:

$$\dot{I}_{\text{total}} = \dot{I}_Q + \dot{I}_{CV} \tag{15}$$

The irreversibility due to heat transfer occurs in the region between ambient and control volume:

$$\dot{I}_Q = T_0\dot{\sigma}_Q \tag{16}$$

Also we have

$$\dot{\sigma}_Q = -\sum\frac{\dot{Q}_j}{T_j} \tag{17}$$

For calculating $\dot{Q}_j$, the heat transfer from muffler to environment should be obtained. We have

$$\dot{Q}_j = \int \delta\dot{Q}_j = \int \mathrm{d}\dot{h} = \int_{T_{\text{down}}}^{T_{\text{up}}} \dot{m}_{\text{exh}}C_p\mathrm{d}T \tag{18}$$

Also the heat transfer from muffler to ambient is

$$\dot{Q}_j = \dot{Q}_0 = \int_{T_{\text{down}}}^{T_{\text{up}}} \dot{m}_{\text{exh}}c_p\mathrm{d}T \text{ and } T_j = T_0 \tag{19}$$

From Eq. (17) and (19):

$$\dot{\sigma}_Q = -\sum\frac{\dot{Q}_j}{T_j} = -\int_{T_{\text{down}}}^{T_{\text{up}}}\frac{\delta\dot{Q}}{T} + \frac{\dot{Q}_0}{T_0} = -\int_{T_{\text{down}}}^{T_{\text{up}}}\frac{\dot{m}_{\text{exh}}C_p\mathrm{d}T}{T} + \frac{1}{T_0}\int_{T_{\text{down}}}^{T_{\text{up}}}\dot{m}_{\text{exh}}C_p\mathrm{d}T \tag{20}$$

Calculating the above equation by using Eq. (16), the irreversibility due to the heat transfer is

$$\dot{I}_Q = T_0\dot{\sigma}_Q = -T_0\int_{T_{\text{down}}}^{T_{\text{up}}}\frac{\dot{m}_{\text{exh}}C_p\mathrm{d}T}{T} + \int_{T_{\text{down}}}^{T_{\text{up}}}\dot{m}_{\text{exh}}C_p\mathrm{d}T \tag{21}$$

The second law efficiency is defined as follows:

$$\varepsilon = \frac{\text{output exergy}}{\text{input exergy}} = \frac{\dot{m}_{\text{water}}(\psi_4-\psi_3)}{-\dot{m}_{\text{exh}}(\psi_2-\psi_1)} \tag{22}$$

The standard condition is defined as $P_a = 736$ mmHg, $P_v = 9.65$ mmHg and $\phi = 0.31$ at $T_a = 29.4$ °C, but the thermodynamical condition in experimental laboratory is different. So, the corrected factor is used to develop the real experimental condition to standard condition. For this reason, corrected brake power factor is defined as following equation:

$$C_f = \frac{P_{bs}}{P_{bm}} = \frac{\dot{m}_{as}}{\dot{m}_{am}} = \frac{p_{sd}}{p_m - p_{vm}}\sqrt{T_m/T_s} \tag{23}$$

With assumptions $P_{sd} = 736.6$ mmHg and $T_s = 29.4$ °C corrected brake power factor is calculated. Also, the amount of P_{bs} is defined as follows:

$$P_{bs} = C_f\cdot P_{bm} = C_f(2\pi N\tau) \tag{24}$$

Brake specific fuel consumption (bsfc) in internal combustion engines is defined as the ratio of fuel flow rate to engine brake power. Brake specific fuel consumption is calculated by the following calculation:

$$\text{bsfc} = \frac{\dot{m}_f}{P_b} = \frac{\dot{m}_f}{2\pi N\tau + \text{recovered exergy from water}} \tag{25}$$

1. Input air
2. Air intake manifold
3. Input Fuel
4. Diesel engine OM314
5. Magnetic tachometer
6. Torque meter
7. Dynamometer
8. Dynamometer loading lever
9. Inlet water path ofdynamometer
10. Outlet water path of dynamometer
11. Exhaust manifold
12. Thermocouple before thesilencer
13. Interface board
14. Computer and monitor
15. Thermocouple after the silencer
16. Pressure gauge
17. Inlet water path to heat exchanger
18. Mass flow rate control valve
19. Thermometer for heat exchanger inlet water
20. Heat exchanger
21. The silencer
22. Outlet engine coolant water thermometer
23. Outlet water path to heat exchanger
24. Exhaust manifold
25. Digital Noise meter

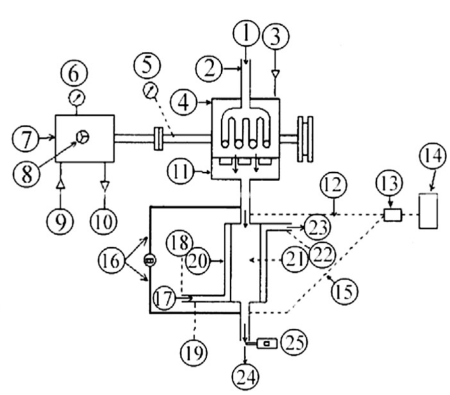

Fig. 3. Schematic of experimental apparatus.

Table 2
Sample experimental data for 1800 rpm in different engine torques (HEX: Heat exchanger).

	Engine torque, T (N m)	Ambient temp., T_0 (°C)	Ambient pressure, P_0 (mmHg)	Engine speed, N (rpm)	Air inlet temp., T_i (°C)	Orifice pressure difference, ΔP (mm H_2O)	Time for 50 cc fuel consumption, t_f (s)	Inlet temp. to HEX, T_{up} (°C)	Out temp. to HEX, T_{down} (°C)	Inlet pressure of HEX, P_{up} (mmHg)	Outlet pressure of HEX, P_{down} (mmHg)	Inlet water temp., $T_{w,i}$ (°C)	Outlet water temp., $T_{w,e}$ (°C)	Water volume rate, $\dot{m}_w$ (ml/s)	Water pressure, P_W (bar)	Relative humidity (%)
With HEX	20	16	671	1800	21.05	61.5	50.64	129.7	98.35	5.5	0	12	22	100.36	1.5	**68**
Without HEX	20	16.5	671	1800	23.4	62	51.63	130.13	116.7	5.8	0	–	–	–	1.5	**67**
With HEX	40	18	671	1800	24.08	65	42.73	148.3	112.6	6.4	0	13	22	76.27	1.6	**63**
Without HEX	40	18	671	1800	23.8	66	41.88	149.11	132.6	7	0	–	–	–	1.6	**62**
With HEX	60	18.5	671	1800	24.8	67	35.41	167.72	127.3	7	0	14	23	73.32	1.6	**61**
Without HEX	60	19	671	1800	25.45	67.2	35.33	169.55	150.5	7.3	0	–	–	–	1.6	**58**
With HEX	80	19	671	1800	25.7	68	30.57	187.7	142.5	8	0	15	26	69.04	1.6	**55**
Without HEX	80	19	671	1800	26.44	68.5	30.58	188.3	166.22	8.3	0	–	–	–	1.6	**53**
With HEX	100	19.5	671	1800	26.7	70.5	27.23	206.6	156.88	8.5	0	16	28	66.01	1.6	**50**
Without HEX	100	19.5	671	1800	26.6	70.5	27.23	208.1	184.3	9	0	–	–	–	1.6	**49**

3. Experimental techniques and apparatus

For calculating the irreversibility and exergy recovery from exhaust, the following experimental apparatus was set up. The picture of turbocharged OM314 DIMLER diesel engine is shown in Fig. 1 and the specifications of this research engine are shown in Table 1. Fuel used in this engine is gasoil with density 841 kg/m^3 at 15 °C and normal paraffin 18.7% and maximum sulfur 1% in weight. A double pipe heat exchanger is used in the exhaust because it's the simplest exchangers used in industries. On one hand, these heat exchangers are cheap for both design and maintenance, making them a good choice for small applications. The flow is considered to be counter current flow for better heat transfer from exhaust gases to cooling water. A DXF Heenan-Froude dynamometer with capacity 112 kW in 7500 rpm is used for applying a specific torque on engine. All temperature sensors are connected to a personal computer (PC) which shows their value. In this experiment for calculating the air mass flow rate an orifice and air tank is used, and fuel mass flow rate is obtained by a measuring pipet and digital stopwatch. All of the setup components are shown in Fig. 3 schematically.

All experiments carried out for five speeds (1200, 1400, 1600, 1800, and 2000 rpm) and for each speed, five steps were taken. The First step in each speed started with the torque of $T = 20$ N m and continued with the torques of 40, 60, 80 and 100 N m. The torque applied to the engine by a dynamometer as follows:

1. The engine started, after 10 min the water temperature reached to 70 °C.
2. By using a crowbar, the engine speed was fixed and the torque is applied by a dynamometer. Torque started from 20 N m and reached to 100 N m after five steps.

In each step (speed and torque were constant), first, the water flowed in the heat exchanger for 10 min until the Steady State-Steady Flow process (SSSF) occurred. Then, these data were measured (the temperature, the pressure and the relative humidity of the ambient, the engine speed, engine coolant water temperature, the oil pressure, the engine torque, the inlet and outlet temperatures of the muffler, inlet and outlet pressure of muffler, inlet and outlet temperatures of water in the heat exchanger, mass flow rate of water in the heat exchanger). Finally, the water flow was blocked and drained from the water jacket until the muffler reached SSSF and the previous parameters were measured. For verifying the recorded

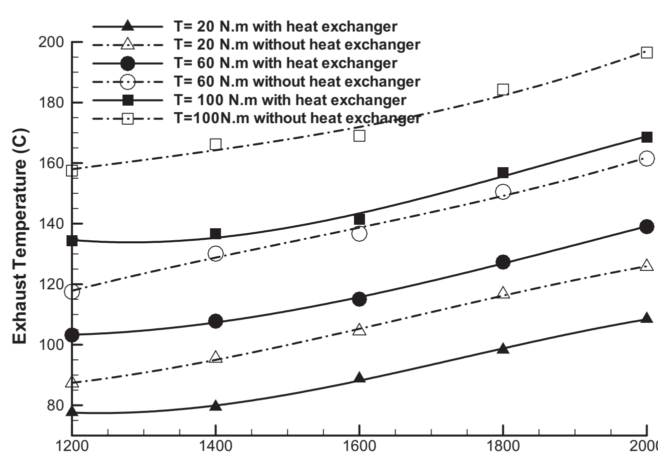

Fig. 4. Exhaust outlet temperature versus engine speed with and without heat exchanger.

Table 3
Enthalpy and entropy for exhaust gases in different loads and speeds.

Engine speed (rpm)	Engine torque (N M)	Exhaust mass flow rate (kg/s)	Exhaust inlet temp. to HEX (K)	Exhaust outlet temp. to HEX (K)	Δh (kJ/kg)	Δs (kJ/kg K)
1200	20	0.0365	378.17	350.85	−27.76429956	−0.075988721
	60	0.0376	394.53	376.35	−18.53800315	−0.047676668
	100	0.0381	460.59	407.45	−54.62606868	−0.124939284
1400	20	0.042106	383.75	352.65	−31.62446627	−0.084024907
	60	0.042962	420.95	380.95	−40.89177959	−0.099899652
	100	0.043057	463.35	409.73	−55.14302855	−0.123918285
1600	20	0.048236	392.55	361.95	−31.1613305	−0.081152979
	60	0.049023	429.05	388.2	−41.81378464	−0.100276882
	100	0.049781	466.55	414.65	−53.41089636	−0.118162776
1800	20	0.053076	402.85	371.5	−31.97652752	−0.080288815
	60	0.055497	440.87	400.45	−41.45671014	−0.095643576
	100	0.057218	479.75	430.03	−51.2923821	−0.109248451
2000	20	0.060032	413.17	381.65	−32.20381706	−0.077461204
	60	0.06135	452.65	412.15	−41.62087828	−0.091864434
	100	0.063326	493.03	441.65	−53.11874847	−0.108469726

data, they were checked three times in each step on running engine. Sample experiment data are presented in Table 2.

4. Results and analysis

Fig. 4 shows the variation of exhaust outlet temperature against engine speeds respectively. As Fig. 4 shows, cooling system makes a significant decrease in the exhaust outlet temperature. Also, it can be seen that by increasing the engine load and speed, exhaust temperatures are increased due to increase in exhaust mass flow rate.

After exergy analysis, the irreversibility and recovered exergy and so second law efficiency is calculated. Tables 3 and 4 show some calculated parameters such as enthalpy, entropy, irreversibility and etc. for exhaust and cooling water. Fig. 5 shows the effect of using cooling system on irreversibility. As seen by using cooling system, total irreversibility increased in most torque and speeds due to increase in irreversibility of heat transfer mechanism in heat exchanger. The recovered exergy from exhaust cooling is shown in Fig. 6. As Fig. 6 shows, the exergy recovery is increased at higher engine speeds (i.e., 1600–2000 rpm). This is due to higher muffler temperature (Fig. 4) causing higher exergy recovery. At lower engine speeds (i.e., 1200–1600 rpm), small reduction seems in exergy recovery. This is due to increase of the engine speed corresponding to insignificant changes in muffler temperature (Fig. 4) which creates small heat transfer and also providing reduction in exergy recovery. Furthermore, recovered exergy can be increased by increasing the engine torque due to increase in exhaust temperature which was observed in Fig. 4. Second law efficiency from Eq. (22) is calculated and outcome is presented in Fig. 7. In an ideal condition, it can be assumed that all of the exhaust exergy from the muffler will be recovered. So, the maximum amount of exergy recovery (or ideal recovered exergy) will be happened in this case which its result can be seen in Fig. 8.

Table 4
Exhaust and water exergy and mass flow rates and total irreversibility in different engine loads and speeds.

Engine speed (rpm)	Engine torque (N m)	$\Delta\psi_{exh}$ (kJ/s)	$\Delta\psi_{water}$ (J/s)	Exhaust flow rate (kg/s)	Water flow rate (kg/s)	Total irreversibility (J/s)
1200	20	−0.207253494	16.59296613	0.0365	0.0504	190.6605283
	40	−0.28132427	45.26971125	0.037	0.0285	236.3242701
	60	−0.175997318	65.68915235	0.0376	0.0202	110.9973179
	80	−0.498413689	91.7789978	0.0375	0.0203	406.6346908
	100	−0.697704945	109.1427845	0.0381	0.0301	588.5621609
1400	20	−0.303273817	14.60575561	0.042106	0.048864	288.6680611
	40	−0.384814415	26.66042902	0.042445	0.047566	358.1539862
	60	−0.509355147	38.27007888	0.042962	0.0531	471.0850678
	80	−0.661122081	48.0991543	0.043055	0.101272	613.0229271
	100	−0.823515889	69.74208623	0.043057	0.107285	753.7738029
1600	20	−0.365349941	15.65448426	0.048236	0.050588	349.6954566
	40	−0.483504033	26.10704127	0.048595	0.057249	457.3969913
	60	−0.621038512	30.5723231	0.049023	0.087755	590.4661893
	80	−0.759739548	46.0652401	0.049363	0.086397	713.6743078
	100	−0.949168631	58.38679049	0.049781	0.116468	890.7818408
1800	20	−0.458607605	35.40150925	0.053076	0.042846	423.2060954
	40	−0.599848951	45.60709611	0.054562	0.056379	554.2418552
	60	−0.757972737	54.27880881	0.055497	0.058647	703.6939285
	80	−0.932483768	88.17504332	0.056095	0.062283	844.308725
	100	−1.118000801	117.437847	0.057218	0.065142	1000.562954
2000	20	−0.581693167	85.23970739	0.060032	0.042604	496.4534591
	40	−0.736072786	87.82562406	0.06043	0.058575	648.2471615
	60	−0.915371482	125.0119353	0.06135	0.055341	790.3595463
	80	−1.128389687	162.1967812	0.062643	0.066574	966.1929058
	100	−1.367336423	201.0973129	0.063326	0.07277	1166.23911

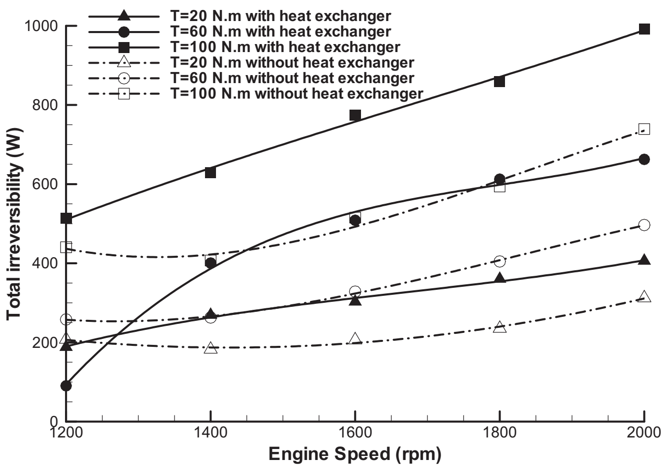

Fig. 5. Total irreversibility in muffler versus engine speed.

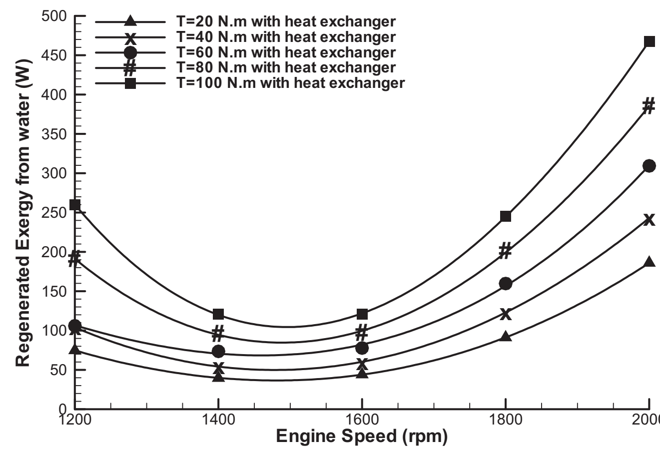

Fig. 8. Ideal recovered exergy from water with using heat exchanger.

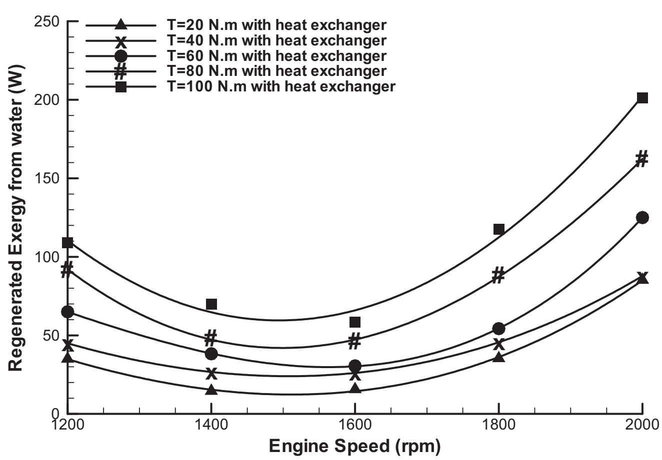

Fig. 6. Actual recovered exergy from water with using heat exchanger.

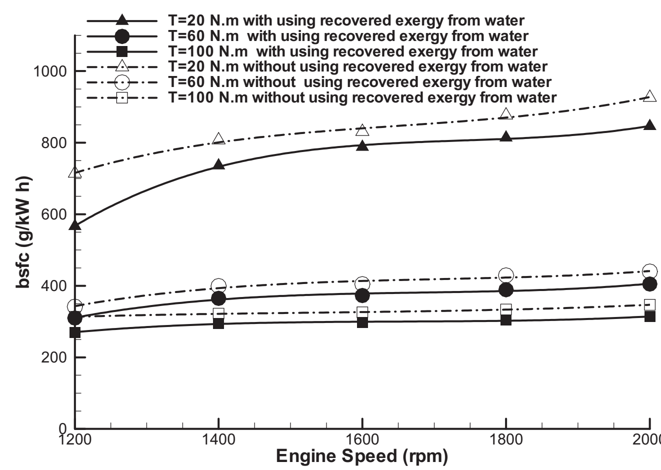

Fig. 9. bsfc versus engine speed with and without actual muffler cooling.

Figs. 9–12 show the bsfc and the percentage of bsfc reduction versus the engine speed for actual and ideal muffler cooling. In this work, for calculating the bsfc, exergy recovery from muffler cooling is added to the engine power as an input power. As the figures show, the bsfc is reduced with using muffler cooling. This reduction is in the range of 5–14% for actual cooling process while for the ideal one, these changes are in 6–15% range. The maximum reduction of bsfc happened at 1200 rpm and 20 N m with the amount of 24.62%. The average reduction is about 10% and this is reasonable and appreciable. As seen in this figures, generally in speeds more than 1400 rpm, by increasing the engine speed and load, percentages of bsfc reduction increases. This fact is due to

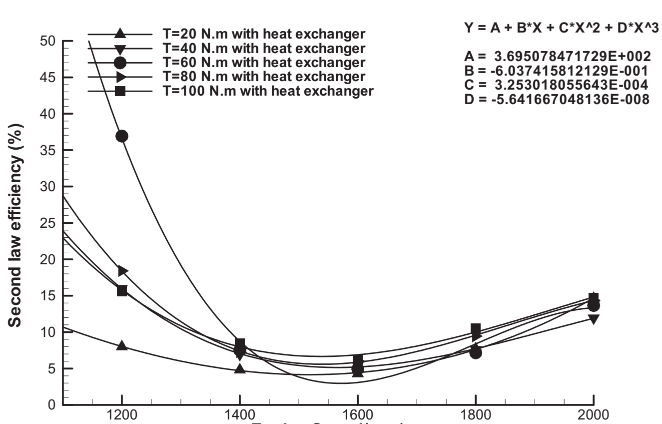

Fig. 7. Second law efficiency of the muffler cooling.

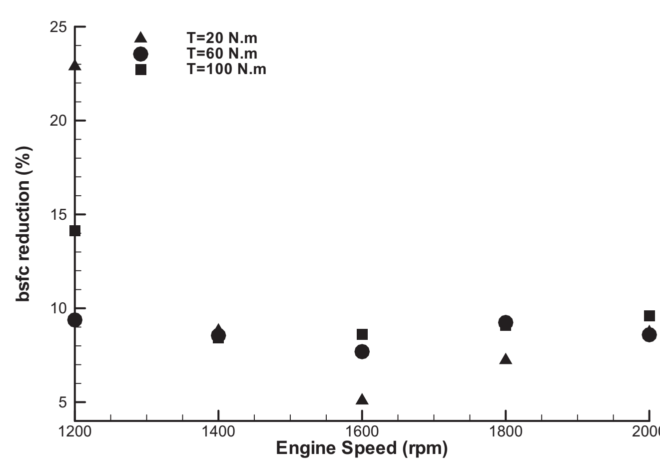

Fig. 10. Percentage of bsfc reduction with using actual recovered exergy.

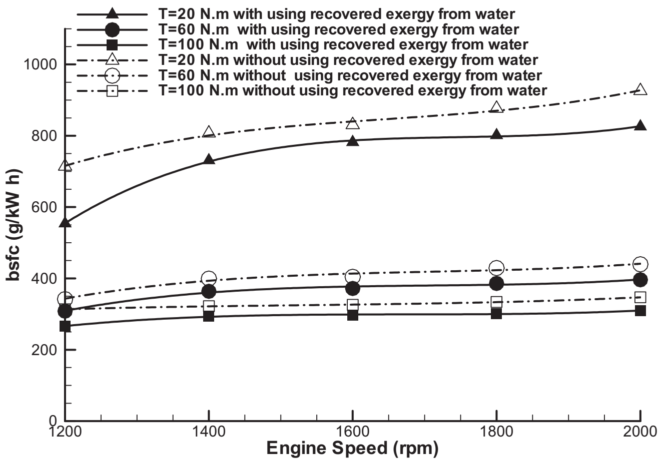

Fig. 11. bsfc versus engine speed with and without using ideal recovered exergy.

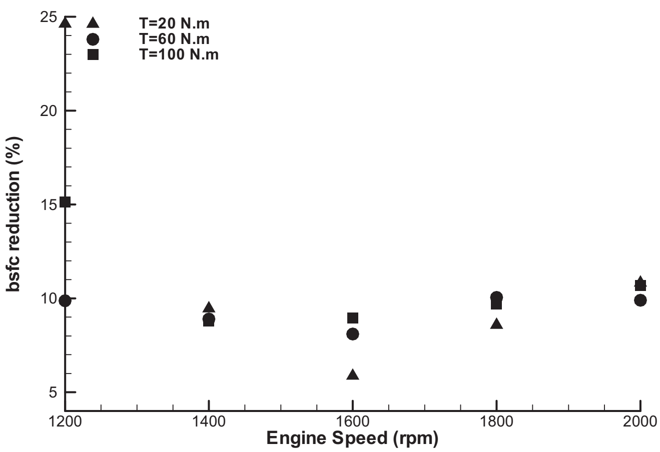

Fig. 12. Percentage of bsfc reduction versus engine speed with using ideal recovered exergy.

higher recovered exergy in these speed and loads which previously were discussed on Fig. 8. So, using the recovered exergy from exhaust by using a heat exchanger can reduce the fuel consumption and help to improve the energy crisis.

5. Conclusion

In this research, an experimental exhaust cooling in a DI diesel engine with the aim of exergy recovery was investigated. The results reveal that amount of the recovered exergy from the exhaust is affected by engine load and speeds, so the second law efficiency for cooling system is presented in wide range of load and engine speeds. It is shown that exergy recovery from muffler cooling can properly be assumed as a part of engine power. In this case, the bsfc is reduced approximately 5–15%.

References

[1] Pandiyarajan V, Chinna Pandian M, Malan E, Velraj R, Seeniraj RV. Experimental investigation on heat recovery from diesel engine exhaust using finned shell and tube heat exchanger and thermal storage system. Appl Energy 2011;88:77–87.
[2] Kauranen P, Elonen T, Wikström L, Heikkinen J, Laurikko J. Temperature optimisation of a diesel engine using exhaust gas heat recovery and thermal energy storage diesel engine with thermal energy storage. Appl Therm Eng 2010;30:631–8.
[3] Yang F, Yuan X, Lin G. Waste heat recovery using heat pipe heat exchanger for heating automobile using exhaust gas. Appl Therm Eng 2003;23:367–72.
[4] Lee S, Bae C. Design of a heat exchanger to reduce the exhaust temperature in a spark-ignition engine. Int J Therm Sci 2008;47:468–78.
[5] Wonnacott EJ. Lower exhaust noise from better silencer design techniques. J Sound Vib 1974;37:17–26.
[6] Prasad MG, Crocker MJ. Studies of acoustical performance of a multi-cylinder engine exhaust muffler system. J Sound Vib 1983;90:491–508.
[7] Nakra BC, Sa'id WK, Nassir A. Investigations on mufflers for internal combustion engines. Appl Acoust 1981;14:135–45.
[8] Alfredson RJ, Davies POAL. Performance of exhaust silencer components. J Sound Vib 1971;15:175–96.
[9] Hyeon-Don J, Shi-Bok L, Weui-Bong J, Byung-Hoon L. Design of an acoustic enclosure with duct mufflers for the heavy duty diesel engine generator set. Appl Acoust April 2004:441–55.
[10] Kirby R, Lawrie JB. A point collocation approach to modeling large dissipative mufflers. J Sound Vib 2005;286:313–39.
[11] Rakopoulos CD, Giakoumis EG. Second law analysis applied to internal combustion engines operation. Prog Energy Combust Sci 2006;32:2–47.
[12] Rosen MA. Clarifying thermodynamic efficiencies and losses via exergy. Exergy Int J 2002;2:3–5.