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EXPERIMENTAL INVESTIGATIONS OF IRREVERSIBILITY EFFECT ON NOISE LEVEL OF A DISSIPATIVE MUFFLER WITH HEAT EXCHANGER IN A DI DIESEL ENGINE

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ABSTRACT

In this research experimental work was carried out for measuring the silencer noise intensity of a turbocharged OM314 DIMLER diesel engine at various engine speeds, namely 1200, 1400, 1600, 1800 and 2000 rpm. At each engine speed, the torque varied from 20 to 100 N.m, in an interval of 20 N.m. In each test, the noise intensity level was measured at the exhaust of a dissipative muffler with and without a heat exchanger. The results were analyzed by utilizing the exergy method and irreversibility, based on second law of thermodynamics.

The results show that increasing the irreversibility of the dissipative muffler (silencer), when a comparison is made between the silencer with and without heat exchanger at similar operating conditions, causes a significant reduction in noise level.

KEYWORDS: Silencer, Diesel engine, Exergy, Irreversibility, Noise intensity.

INTRODUCTION

Noise pollution is increasing in the populated and industrial cities. This environmental pollution is of vital importance in human life, so use of suitable silencers in automobiles and factories can cause a substantial reduction in noise pollution. The car manufactures are confined to design instruments for decreasing the engines' noise level and other elements that cause noise pollution [1].

Silencers are widely used to reduce the exhaust discharge noise. There are two basic types of muffler or silencers: reactive and dissipative [2, 3, 4]. Dissipative silencers are widely used to attenuate broadband noise emanating from fluid moving devices such as internal combustion engines and fans. The silencers vary considerably in size and shape according to their applications, for example automotive exhaust silencers are relatively small whereas many Heat, Air, Ventilation and Cooling (HAVC) applications require much larger silencers [5, 6].

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The most important factors for generation and development of noise in an internal combustion engine are the unsteadiness and compressibility of flow. The emitted noise of air intake and exhaust system is caused by impulsive pressure and flow noise. Impulsive pressure is generated by periodicity of aspiration and discharge process and is emitted by open end of each manifold. This kind of noise is called impulsive or gas dynamic noise. The flow noise is caused by geometrical discontinuities which create turbulence and vortex in passing flow. This type of noise is called aerodynamic or gas flow noise [7, 8, 9].

Silencer in the path of gas exhaust creates back pressure to the engine that causes an increase in the pumping friction in the engine [10].

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 [11].

Exergy is a method for estimating the available work losses, using the second law of thermodynamics, which is known as availability analysis [12, 13, 14].

Exergy analysis is a combination of the first and second law of thermodynamics which allows finding the amount of available work output losses, location and its reason for the losses [15].

In order to estimate the system efficiency, considerations of processes in the system and its outside for calculating the work potential losses are necessary. Only in this condition the system efficiency can be calculated [16].

This research has tried to find a relationship between irreversibility and noise intensity in the silencer of an internal combustion engine.

In this experimental research work a silencer with a heat exchanger with water as coolant fluid is employed in order to estimate the different kinds of irreversibility such as total irreversibility, irreversibility of heat transfer and internal irreversibility, using exergy analyses in the silencer with or without heat exchanger at various engine operating conditions. In each state, the external noise intensity of silencer is measured.

1. EXERGY AND IRREVERSIBILITY IN THE SILENCER WITHOUT HEAT EXCHANGER

The exergy and irreversibility equations in the silencer without heat exchanger, using the control volume, can be driven according to Figure 1 [17]:

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

In this equation, $\frac{d\Phi_{cv}}{dt}$ is the nonflow exergy change of the control volume (C.V.), i.e. zero in the steady state process.

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

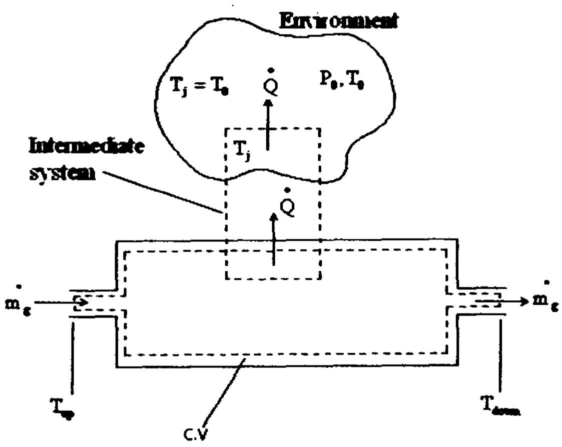

Figure1. Silencer without heat exchanger.

Since $T_j = T_0$, therefore

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

The work is zero, so

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

We have

$$\dot{m}_g = \dot{m}_i = \dot{m}_e \tag{5}$$

Substituting the equations (2) and (5) into (1), the exergy equation becomes

$$\dot{I}_{total} = \dot{m}_i \psi_i - \dot{m}_e \psi_e = \dot{m}_g(\psi_i - \psi_e) \tag{6}$$

Also, the flow exergy equation is

$$\psi = h + \frac{v^2}{2} + gZ - T_0 s - (h_0 - T_0 s_0) \tag{7}$$

Substituting equation (7) into (6), the total irreversibility can be obtained as

$$\dot{I}_{total} = \dot{m}_g[h_i + \frac{v_i^2}{2} + gZ_i - T_0 s_i) - (h_e + \frac{v_e^2}{2} + gZ_e - T_0 s_e)] \tag{8}$$

Kinetic and potential energy change is negligible and thus equation (8) becomes

$$\dot{I}_{total} = \dot{m}_g[(h_i - h_e) - T_0(s_i - s_e)] \tag{9}$$

Also we have

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{10}$$

1.1. CALCULATION OF THE ENTHALPY CHANGES IN THE SILENCER

To calculate the enthalpy changes ($\Delta h = h_e - h_i$) between the two sides of the silencer, equation (11) can be employed

$$dh = C_p dT \tag{11}$$

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

$$C_{pm} = \sum n_i \overline{C_{pi}} = C_{pm}(T) \tag{12}$$

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 in term of T [16]

$$C_p = 28.11 + 0.1967\times10^{-2}T + 0.4802\times10^{-5}T^2 - 1.966\times10^{-9}T^3 \tag{13}$$

Substituting equation (13) in (11) and integrating in limits of the internal temperature and external temperature of silencer, the enthalpy changes of silencer can be obtained.

$$\int_{T_{up}}^{T_{down}} dh = \int_{T_{up}}^{T_{down}} C_p dT \Rightarrow$$

$$h_{down} - h_{up} = \frac{1}{28.9}[28.11(T_{down} - T_{up}) + 0.9835\times10^{-3}(T_{down}^{-2} - T_{up}^{-2}) + 0.16\times10^{-5}(T_{down}^{-3} - T_{up}^{-3}) - 0.49\times10^{-9}(T_{down}^{-4} - T_{up}^{-4})] \tag{14}$$

1.2. CALCULATION OF THE ENTROPY CHANGES AND THE TOTAL IRREVERSIBILITY IN THE SILENCER

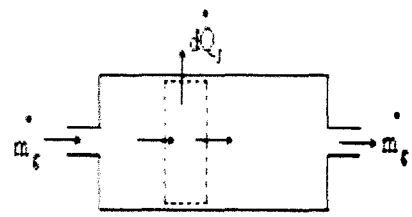

Figure2. Element of the silencer

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

$$\int_{T_{up}}^{T_{down}} ds = \int_{T_{up}}^{T_{down}} \frac{C_p dT}{T} - RLn\left(\frac{p_{down}}{p_{up}}\right) \tag{15}$$

From equations (9) to (15), the total irreversibility can be obtained as

$$\begin{aligned} \dot{I}_{total} &= \dot{m}_g[(h_i - h_e) - T_0(s_i - s_e)] = -\dot{m}_g[(\frac{1}{28.9}[28.11(T_{down} - T_{up}) + 0.9835\times 10^{-3}(T_{down}^{-2} - T_{up}^{-2}) \\ &+ 0.16\times 10^{-5}(T_{down}^{-3} - T_{up}^{-3}) - 0.49\times 10^{-9}(T_{down}^{-4} - T_{up}^{-4})]) - T_0(\frac{1}{28.9}[28.11Ln\left(\frac{T_{down}}{T_{up}}\right) + 0.1967\times 10^{-2}(T_{down} - T_{up}) \\ &+ 04802\times 10^{-5}(T_{down}^{2} - T_{up}^{2}) - 0.655\times 10^{-9}(T_{down}^{3} - T_{up}^{3})] - RLn(\frac{p_{down}}{p_{up}}))] \end{aligned} \tag{16}$$

where the total irreversibility consists of two parts

$$\dot{I}_{total} = \dot{I}_Q + \dot{I}_{cv} \tag{17}$$

1.3. CALCULATION OF IRREVERSIBILITY CAUSED BY HEAT TRANSFER IN THE SILENCER

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

$$\dot{I}_Q = T_0\, \dot{\sigma}_Q \tag{18}$$

Also we have

$$\dot{\sigma}_Q = -\sum \frac{\dot{Q}_j}{T_j} \tag{19}$$

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

$$\dot{Q}_j = \int \delta \dot{Q}_j = \int d\dot{h} = \int_{T_{down}}^{T_{up}} \dot{m}_g\, C_p dT \tag{20}$$

Also the transferring heat from silencer to ambient is

$$\dot{Q}_j = \dot{Q}_0 = \int_{T_{down}}^{T_{up}} \dot{m}_g c_p dT \quad and \quad T_j = T_0 \tag{21}$$

From equations (19) and (21)

$$\dot{\sigma}_Q = -\sum \frac{\dot{Q}_j}{T_j} = -\int_{T_{down}}^{T_{up}} \frac{\delta \dot{Q}}{T} + \frac{\dot{Q}_0}{T_0} = -\int_{T_{down}}^{T_{up}} \frac{\dot{m}_g C_p dT}{T} + \frac{1}{T_0}\int_{T_{down}}^{T_{up}} \dot{m}_g C_p dT \tag{22}$$

$$\int_{T_{up}}^{T_{down}} ds = \int_{T_{up}}^{T_{down}} \frac{C_p dT}{T} - RLn(\frac{p_{down}}{p_{up}}) \tag{15}$$

From equations (9) to (15), the total irreversibility can be obtained as

$$\begin{aligned} \dot{I}_{total} &= \dot{m}_g[(h_i - h_e) - T_0(s_i - s_e)] = -\dot{m}_g[(\frac{1}{28.9}[28.11(T_{down} - T_{up}) + 0.9835\times10^{-3}(T_{down}^{-2} - T_{up}^{-2}) \\ &+ 0.16\times10^{-5}(T_{down}^{-3} - T_{up}^{-3}) - 0.49\times10^{-9}(T_{down}^{-4} - T_{up}^{-4})]) - T_0(\frac{1}{28.9}[28.11Ln\left(\frac{T_{down}}{T_{up}}\right) + 0.1967\times10^{-2}(T_{down} - T_{up}) \\ &+ 04802\times10^{-5}(T_{down}^{2} - T_{up}^{2}) - 0.655\times10^{-9}(T_{down}^{3} - T_{up}^{3})] - RLn(\frac{p_{down}}{p_{up}}))] \end{aligned} \tag{16}$$

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Also the transferring heat from silencer to ambient is

$$\dot{Q}_j = \dot{Q}_0 = \int_{T_{down}}^{T_{up}} \dot{m}_g c_p dT \quad and \quad T_j = T_0 \tag{21}$$

From equations (19) and (21)

$$\dot{\sigma}_Q = -\sum \frac{\dot{Q}_j}{T_j} = -\int_{T_{down}}^{T_{up}} \frac{\delta\dot{Q}}{T} + \frac{\dot{Q}_0}{T_0} = -\int_{T_{down}}^{T_{up}} \frac{\dot{m}_g C_p dT}{T} + \frac{1}{T_0}\int_{T_{down}}^{T_{up}} \dot{m}_g C_p dT \tag{22}$$

Calculating the above equation by using equation (18), the irreversibility due to the heat transfer is

$$\dot{I}_Q = T_0\,\dot{\sigma}_Q = -T_0 \int_{T_{down}}^{T_{up}} \frac{\dot{m}_g\,C_p\,dT}{T} + \int_{T_{down}}^{T_{up}} \dot{m}_g\,C_p\,dT \tag{23}$$

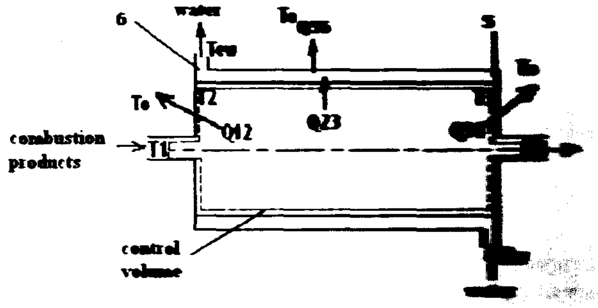

Figure 3. First control volume of silencer

1.4. THE EXERGY AND THE IRREVERSIBILITY EQUATIONS IN THE SILENCER WITH HEAT EXCHANGER

Consider two control volumes. First the equations for initial control volume are derived (Figure 3) which include only the silencer. Using the equations (1) and (2) when the heat transfer is among the control volume, the ambient and the heat exchanger, we obtain

$$\sum \dot{\Phi}_Q = \int_1^4 (1-\frac{T_0}{T_j})\,\delta\dot{Q}_{14} = \int_1^2 (1-\frac{T_0}{T_0})\,\delta\dot{Q}_{12} + \int_2^3 (1-\frac{T_0}{T_w})\,\delta\dot{Q}_{23} + \int_3^4 (1-\frac{T_0}{T_0})\,\delta\dot{Q}_{34} = \int_2^3 (1-\frac{T_0}{T_w})\,\delta\dot{Q}_{23} \tag{24}$$

From equations (4) to (9) and the entropy equation, the total irreversibility equation is as followed

$$\dot{I}_{total,1} = \int_2^3 (1-\frac{T_0}{T_w})\,\delta\dot{Q}_{23} + \dot{m}_g[(h_{ig}-h_{eg}) - T_0(s_{ig}-s_{eg})] \tag{25}$$

The total entropy generation due to heat transfer in the first control volume equals to the entropy generation due to heat transfer from T_1 to T_2 plus the entropy generation due to the heat transfer from T_2 to T_3 plus the entropy generation due to the heat transfer from T_3 to T_4.

$$\dot{\sigma}_{Q,1} = \dot{\sigma}_1 + \dot{\sigma}_2 + \dot{\sigma}_3 = \int_{T1}^{T4} \frac{\dot{m}_g\,c_{p,g}\,dT_g}{T_g} + \int_2^3 (\frac{\delta\dot{Q}_{23}}{T_w}) + \frac{\dot{Q}_{12}+\dot{Q}_{34}}{T_0} \tag{26}$$

Now, the equations for second control volume (Figure 4) which contains the water stream and the external wall of the heat exchanger are done as

$$\sum \dot{\Phi}_Q = \sum \dot{Q}(1-\frac{T_0}{T_j}) = -\int_2^3 (1-\frac{T_0}{T_w})\delta\dot{Q}_{23} + \int_5^6 \frac{\delta\dot{Q}_0}{T_0} \tag{27}$$

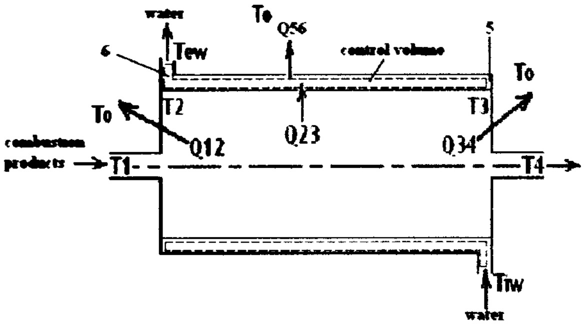

Figure 4. Second control volume of silencer

Note that the heat transfer in the second control volume is between the exhaust gases and the ambient.

Having the above equations and external work which is zero, we obtain

$$\begin{aligned} \dot{I}_{total,2} &= \sum \dot{\Phi}_Q + \sum \dot{m}_i\psi_i - \sum \dot{m}_e\psi_e = -\int_2^3 (1-\frac{T_0}{T_w})\delta\dot{Q}_{23} + \int_5^6 \delta\dot{Q}_0(1-\frac{T_0}{T_0}) + \sum \dot{m}_i\psi_i - \sum \dot{m}_e\psi_e \\ &= -\int_2^3 (1-\frac{T_0}{T_w})\delta\dot{Q}_{23} + \dot{m}_w[(h_{iw}-h_{ew}) - T_0(s_{iw}-s_{ew})] \end{aligned} \tag{28}$$

In order to calculate the entropy generation due to heat transfer in this system, only the system that intermediates the heat transfer with the ambient is considered and the entropy generation due to heat transfer in the system that intermediates the heat transfer with the silencer is calculated as

$$\dot{\sigma}_{Q,2} = \int_5^6 \frac{\delta\dot{Q}_{56}}{T_0} - \int_5^6 \frac{\delta\dot{Q}_{56}}{T_w} = \frac{\dot{Q}_{56}}{T_0} - \int_5^b \frac{\delta\dot{Q}_{56}}{T_0} \tag{29}$$

So in order to calculate the total entropy generation in the silencer with heat exchanger the following equations can be employed

$$\dot{I}_{total} = \dot{I}_{total,1} + \dot{I}_{total,2} = \dot{m}_g[(h_{ig}-h_{eg}) - T_0(s_{ig}-s_{eg})] + \dot{m}_w[(h_{iw}-h_{ew}) - T_0(s_{iw}-s_{ew})] \tag{30}$$

where the total entropy due to heat transfer is

$$\dot{\sigma}_Q = \dot{\sigma}_{Q1} + \dot{\sigma}_{Q,2} = \int_{T1}^{T4} \frac{\dot{m}_g c_{p,g} dT_g}{T_g} + \int_2^3 (\frac{\delta\dot{Q}_{23} - \delta\dot{Q}_{56}}{T_w}) + \frac{\dot{Q}_{12} + \dot{Q}_{34} + \dot{Q}_{56}}{T_0} \tag{31}$$

If the first law of thermodynamics is applied to the control volume in the Figure (5), then

$$\delta \dot{Q}_{23} - \delta \dot{Q}_{56} = \dot{m}_w\, c_{p,w} dT_w \tag{32}$$

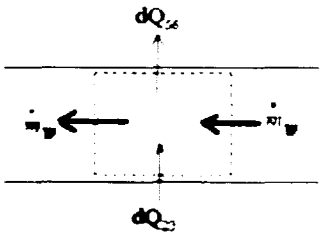

Figure 5. Element in the heat exchanger

Considering the first law of thermodynamics applied to the control volume which contains the silencer and the heat exchanger (Figure 6), we get

$$\sum \dot{Q}_j = \sum \dot{h} + \sum \dot{w} \tag{33}$$

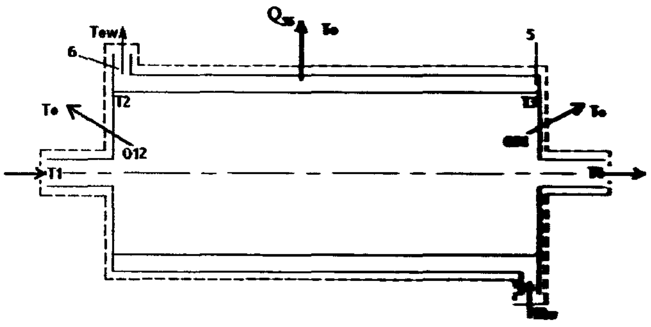

Figure 6. The control volume of silencer and heat exchanger

$$\sum \dot{w} = 0 \Rightarrow \sum \dot{Q}_j = \sum \dot{h}_j \Rightarrow \dot{Q}_{12} + \dot{Q}_{34} + \dot{Q}_{56} = \int_{T_4}^{T_1} \dot{m}_g\, c_{p,g} dT_g + \dot{m}_w\, c_{p,w} (T_{iw} - T_{ew}) \tag{34}$$

Considering the equations (31) to (34)

$$\dot{\sigma}_Q = \int_{T1}^{T4} \frac{\dot{m}_g\, c_{p,g} dT_g}{T_g} + \int_5^6 \left(\frac{\dot{m}_w\, c_{p,w} dT_w}{T_w}\right) + \frac{1}{T_0}\left[\int_{T_4}^{T_1} \dot{m}_g\, c_{p,g} dT_g + \dot{m}_w\, c_{p,w} (T_{iw} - T_{ew})\right] \tag{35}$$

2. EXPERIMENTAL APPARATUS AND TEST PROCEDURE

In order to measure irreversibility in the silencer and its relation to the exhaust noise intensity, the following experimental apparatus were set up (Figure 7).

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

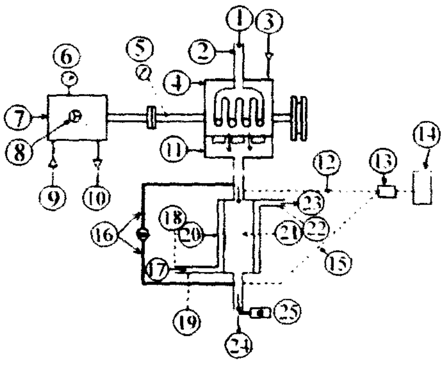

Figure7. Experimental apparatus

2.1. DIESEL ENGINE

The turbocharged OM314 DIMLER diesel engine was employed for the test procedure. The engine's characteristics are shown in Table 1.

Engine type	4 stroke diesel engine
Number of cylinders	4
Combustion chamber	Direct injection
Bore × stroke (mm)	97 × 128
Piston displacement (cc)	3784
Comprettion ratio	17:1
Maximum power (hp)	85
Maximum torque (N.m)	235
Maximum speed (rpm)	2800
Mean effective pressure (bar)	6.8@2800 rpm

Table 1. Specifications of Diesel engine OM314

2.2. TEST PROCEDURE

A heat exchanger was used in order to increase the irreversibility due to heat transfer. These experiments were 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, 100 N.m. The torque applied to the engine by a dynamometer was as follows:

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

In each step (speed and torque were constant) the water flowed in the heat exchanger for ten minutes until the Steady State Steady Flow (SSSF) process occurred. Data like temperature, pressure and relative humidity of the ambient, engine speed, engine coolant water temperature, oil pressure, engine torque, inlet and outlet temperature of the silencer, inlet and outlet pressure of silencer, inlet and outlet temperature of water in the heat exchanger, mass flow rate of water in the heat exchanger were measured. Then, the water flow was blocked and drained from the water jacket until the silencer reached SSSF and the previous parameters were measured.

3. RESULTS AND DISCUSSION

From the results shown in Figures 8 to 12, it is clear that when the heat exchanger is employed the irreversibility increases, but noise intensity level decreases.

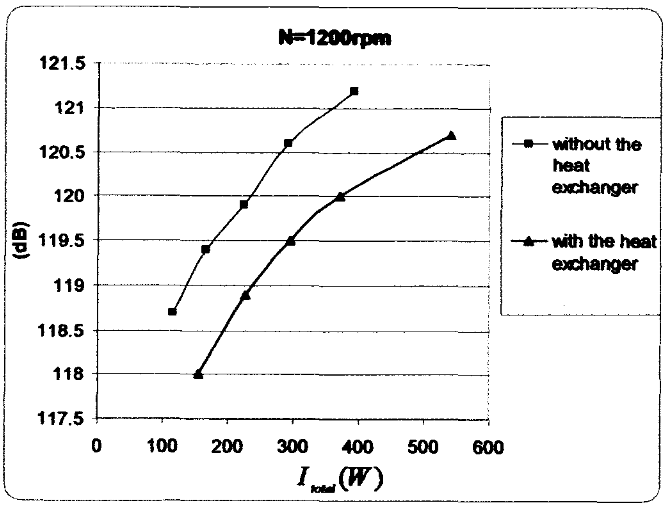

Figure 8. Noise intensity verses the total irreversibility in two states of with and without the heat exchanger in the constant speed (1200rpm) and different torque

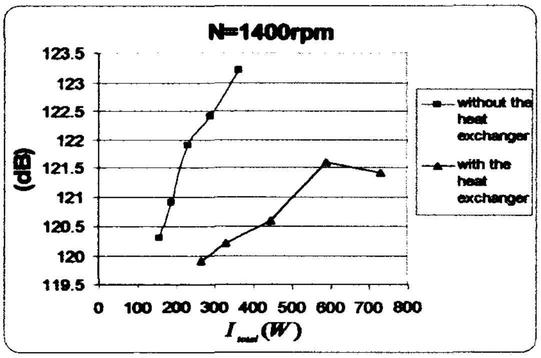

Figure 9. Noise intensity verses the total irreversibility in two states of with and without the heat exchanger in the constant speed (1400rpm) and different torque

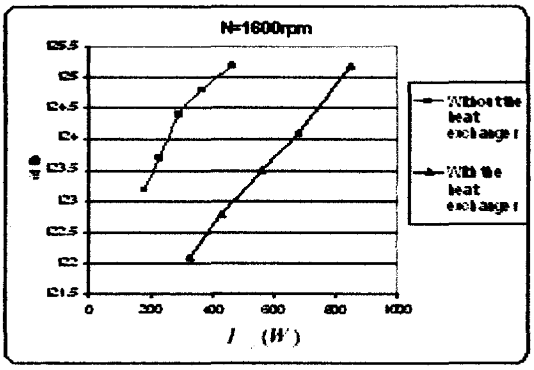

Figure 10. Noise intensity verses the total irreversibility in two states of with and without the heat exchanger in the constant speed (1600rpm) and different torque

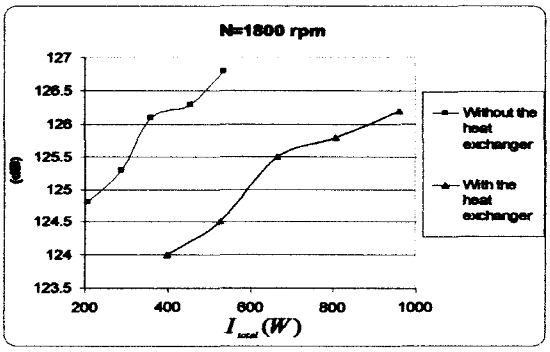

Figure 11. Noise intensity verses the total irreversibility in two states of with and without the heat exchanger in the constant speed (1800rpm) and different torque

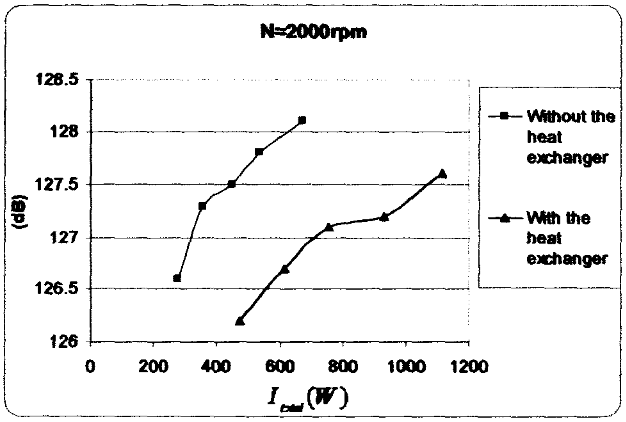

Figure12. Noise intensity verses the total irreversibility in two states of with and without the heat exchanger in the constant speed (2000rpm) and different

At constant speed, as shown in Figures 13 and 14, increasing the torque causes a substantial increase in noise level while at constant torque, the increase of speed causes a considerable increase in noise intensity.

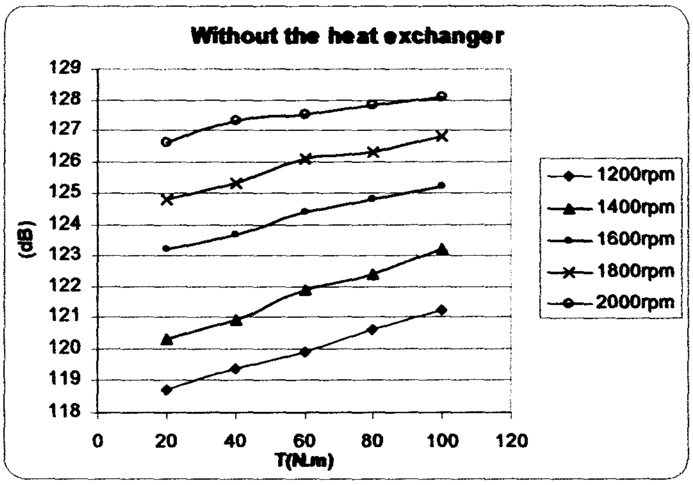

Figure13. Noise intensity verses torque in the different speed and in the state without heat exchanger

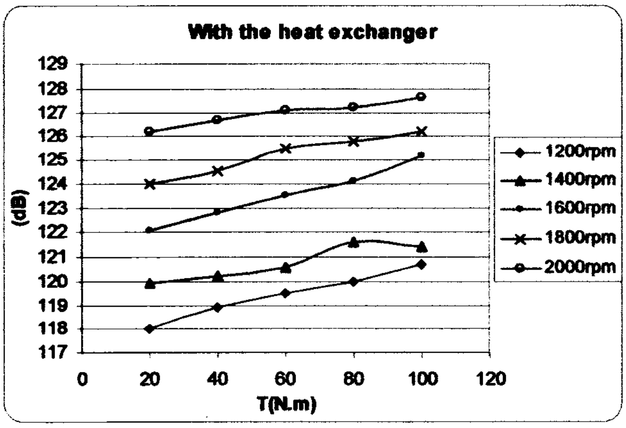

Figure 14. Noise intensity verses torque in the different speed and in the state with heat exchanger

Figures 15 and 16 also show the results at constant speed. By increasing the external temperature of the silencer, noise intensity increases, while at constant temperature by increasing the speed, noise intensity will increase.

At constant torque (Figures 17 and 18), increase in mass flow rate causes the noise intensity level to increase.

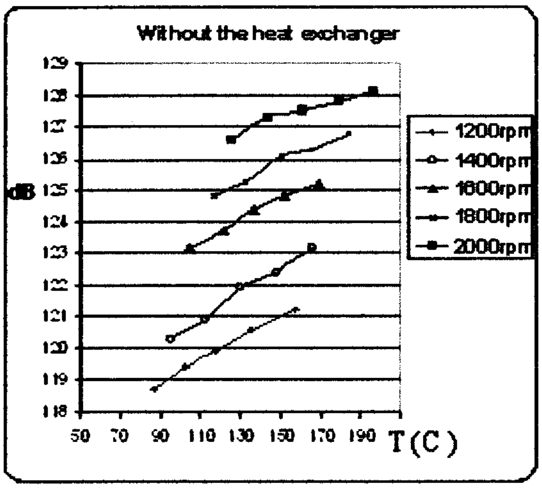

Figure15. Noise intensity verses the external temperature in the different speed and without the heat exchanger

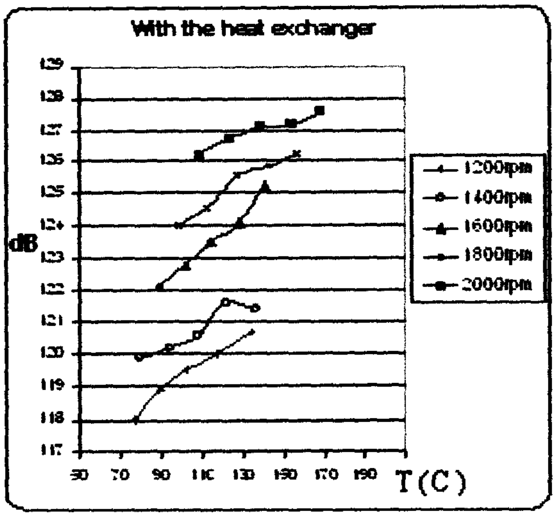

Figure16. Noise intensity verses the external temperature in the different speed and with the heat exchanger

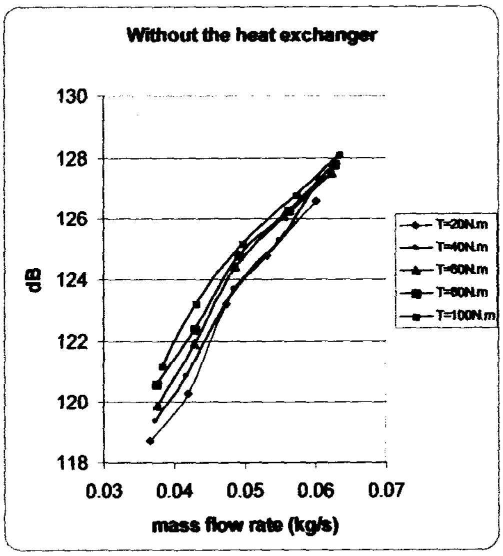

Figure17. Noise intensity verses exhaust mass flow rate in the different speed and without heat exchanger

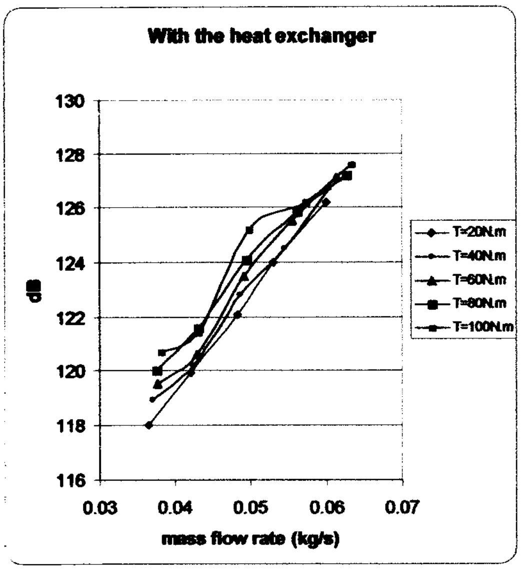

Figure18. Noise intensity verses exhaust mass flow rate in the different speed and with heat exchanger

CONCLUSIONS

Experiments on a silencer connected to a diesel engine indicate the effect of different parameters such as speed, torque, mass flow rate, temperature and also irreversibility due to heat transfer on variation on noise intensity level. An increase in torque, speed, and mass flow rate causes a considerable increase in the irreversibility and noise level.

It is also concluded that noise level is a function of speed, torque, mass flow rate, silencer outlet and the temperature as well as irreversibility due to heat transfer of exhaust gases.

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