

Effect of engine parameters on turbocharger second law efficiency in turbocharged diesel engines

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Abstract

In this paper a study to analyze the influence of engine parameters on second law turbocharger efficiency is investigated. The analysis has been performed by means of a four stroke, four cylinders turbocharged diesel engine and tests have been carried out in constant speed with different loads. In this work exhaust species were measured and exergy flow of exhaust and turbine power output has been calculated. Then turbocharger second law efficiency calculated. The results show that higher bmep, higher exhaust pressure and temperature, higher equivalence ratio and higher engine speed increase turbocharger second law efficiency. Higher second law efficiency of turbocharger means higher turbine work with a specific amount of exergy flow of exhaust. That means turbocharger feed back in transient conditions could be improved by higher engine speed, higher inlet turbine temperature and higher fuel-air ratio due to better available work transfer between engine cylinder and turbocharger.

Keywords: Turbocharger, Second law efficiency, Diesel engine.

Introduction

In order to increase the power output and to save the energy, the turbocharged diesel engines are usually used for automotive engines [1]. But turbocharger lag occurs in these engines during transient condition such as a rapid acceleration or a sudden large load application, and then result in a worse response performance than those of naturally aspirated engine [2].

By adopting the turbocharged diesel engine, the maximum thermal efficiency is increased, while turbocharged vehicle has a weak point of poor drivability under transient running conditions when the turbocharger does not work effectively, especially at low speed operation and rapid acceleration by the control of fuel-pump rack [2].

In general, the causes of the time delay in the transient operation of a turbocharged engine can be classified in three groups: mechanical, thermal and fluid-dynamic phenomena [3]. In the first group mechanical friction and especially the inertia of the turbocharger and other rotating elements of the engine are included. Relevant

thermo- and fluid dynamic causes are the mass and energy transfer processes from the exhaust valve up to the turbine and from the compressor outlet to the cylinder. Actually these thermo- and fluid dynamic parameters mainly depend on the inertia of the turbocharger and other rotating elements of the engine. So the specific characteristics of the energy transfer between the cylinder and the turbocharger are the main cause of the delay in the turbocharger acceleration. It is well understood that the energy transported by the exhaust gases arrives at the turbine very quickly during any transient operation [4]. Nevertheless, only a fraction of this energy is converted into mechanical work and transferred to the compressor, since the largest part of the thermal energy is employed in overcoming the turbocharger rotor inertia to accelerate it. The inertia of the turbocharger and other rotating elements of the engine is the only different between steady state and transient conditions in turbocharged diesel engine as J. Benajes et-al stress " it is well understood that the energy transported by the exhaust gases arrives at the turbine very quickly during any transient operation" [4]. In the point of turbocharged diesel engine, if the second law efficiency of turbocharger in steady state conditions is improved that means better energy transfer between engine cylinder and turbocharger results better overcoming to turbocharger inertia in transient condition. That is why in this research the higher steady state second law efficiency results are used to explain as a reason for better transient turbocharger behavior of diesel engine. Quick changes in rack position do not result in instantaneous response of the turbocharger, due to its inertia and compressibility of the exhaust gas with the engine. Thus the air-fuel ratio quickly decreases to a very low value and incomplete combustion occurs in the mixture; although much quantity of fuel can be rapidly injected into the cylinders, the turbocharger is slow to respond and provide a corresponding increase in air for the combustion [5]. Consequently turbocharger has a poor sensitivity under transient running condition and turbocharger does not work efficiently.

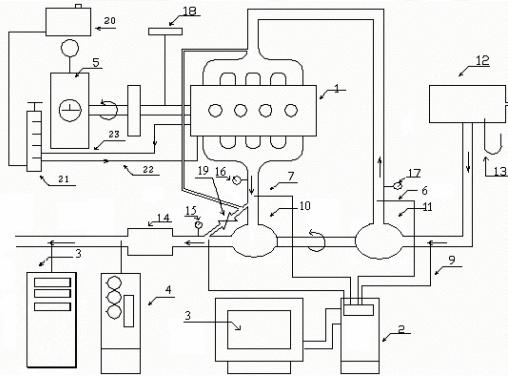
The aim of this work is to find the effect of engine parameters on turbocharger second law efficiency. Turbocharger exergy analysis is a powerful tool to find

the reasons of the rate of turbine work in turbocharged diesel engines. In the present work although the variation of engine parameters and turbocharger second law efficiency, in the steady state condition have been investigated. But the variations of steady state results are used to discuss on transient turbocharger behavior.

2. Experimental

2.1. Test rig

Figure 1 shows a schematic diagram of the test rig and measuring system. OM314 turbocharged diesel engine (1) was used in present work. Engine characteristics are given in table 1. Engine is turbocharged by a turbocharger (10), (11) equipped with a waste gate (19), supported by Borg Warner Company to convert aspirated engine to turbocharged diesel engine. The test rig is equipped with a 112 kW DXF Heenan & Froude hydraulic dynamometer (5). A Plint-RE205 gas analyzer (3) measures unburned hydrocarbon as C_6H_{14} , CO, CO₂ and O₂. The experimental apparatus is composed from AVL- 415 smoke meter (4), thermocouple type K (6),(7),(8),(9) interface to a PC(2), volumetric fuel meter(21), air meter (surge tank and orifice manometer) (12),(13) pressure gages (15),(16), (17) and electrical engine speed meter (18).



1_Engine 2_Computer and interface 3_Plint-RE205 gas analyzer 4_AVL_415 smoke meter 5_Hydraulic dynamometer 6_Thermocouple type(K) 7_Thermocouple type(K) 8_Thermocouple type(K) 9_Thermocouple type(K) 10_Turbocharger turbine 11_Turbocharger compressor 12_Surge tank 13_Manometer 14_Exhaust silencer 15_Pressure gage 16_Pressure gage 17_Pressure gage 18_Electrical speed meter 19_wastegate 20_Fuel meter 21_Fuel gage 22_Fuel inlet 23_Reverse fuel

Figure 1: Block diagram of the test rig

Table 1

Engine and turbocharger	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	17:1
Maximum power (h.p)	85
Maximum torque (N.m)	235
Maximum speed (rpm)	2800
Mean effective pressure (bar)	6.8@2800rpm
Turbocharger turbine	Radial type
Turbocharger compressor	Centrifugal type

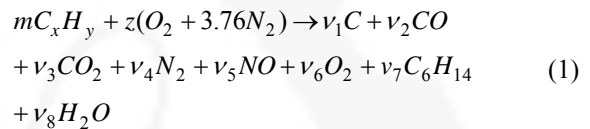
2.2. Test procedure

The tests were conducted with variable load at constant engine speeds of 1200, 1300, 1500, 1700 and 2000rpm. At each operating condition the dynamometer load, speed, fuel flow, air flow, soot density, CO, CO₂, UHC and O₂ were recorded after sufficient time for the engine to stabilize.

Also inlet and outlet temperature and pressure of turbine and compressor, ambient temperature and pressure were measured.

2.3. Exhaust emissions calculation

In the tests, CO, CO₂, UHC and O₂ were measured by Plint-RE205 gas analyzer. Also soot was measured by AVL-415 smoke meter in each test. Other species in exhaust gas was calculated with the following chemical equation:



The molecular weight of fuel and its molecular formula is considered as a heavy diesel fuel (200 kg/kmol , $C_nH_{1.7n}$), therefore the formula of fuel is $C_{14.59}H_{24.86}$ [1].

The unknown coefficients are v_1, v_4, v_5, v_8, z, m . Exhaust density is calculated with Eq.(2). Fuel and air flow also is measured respectively with Eq.(3) and Eq.(4) by orifice manometer and fuel gage measurement and characteristics. Then soot flow rate is calculated with Eq.(5) and Eq.(6). The unknown coefficients are calculated with use of equations (7),(8),(9),(10),(11) and (12). Equation (13) is a extra checking equation.

$$\rho_e = \frac{P_e}{(287 \times (T_e + 273))} \quad (2)$$

In Eq.(2), T_e is the turbine outlet temperature and P_e is turbine outlet pressure. The turbine outlet pressure is assumed equal to ambient pressure ($P_e \approx P_0$).

$$\dot{m}_{air} = c_d \times A_o \times \sqrt{2 \times 9.81 \times \rho_1 \times \Delta h_{orifice} \rho_{air}} \quad (3)$$

$$\dot{m}_f = \frac{50 \times 10^{-6}}{t_f} \times \rho_f \quad (4)$$

$$\dot{V}_e = \frac{\dot{m}_a + \dot{m}_f}{\rho_e} \quad (5)$$

$$\dot{m}_{soot} = \rho_{soot} \times \dot{V}_e \quad (6)$$

$$\frac{12v_1}{z * 4.76 * 29} = \frac{\dot{m}_{soot}}{\dot{m}_{air}} \quad (7)$$

$$\frac{200m}{z * 4.76 * 29} = \frac{\dot{m}_f}{\dot{m}_{air}} \quad (8)$$

$$mx = v_1 + v_2 + v_3 + 6v_7 \quad (9)$$

$$my = 14v_7 + 2v_8 \quad (10)$$

$$2z = v_2 + 2v_3 + v_5 + 2v_6 \quad (11)$$

$$2 * 3.76z = 2v_4 + v_5 \quad (12)$$

Plint-Re205 measures emissions in a mixture of CO, CO₂, N₂, NO, O₂, C₆H₁₄ and H₂O so:

$$v_2 + v_3 + v_4 + v_5 + v_6 + v_7 + v_8 = 100 \quad (13)$$

3. Exergy and reversible work

Exergy is a thermodynamic concept that measures the maximum amount of useful work that can be extracted when a system is brought into thermodynamic equilibrium reversibly with the environment conditions [9].

In this study, the following assumptions were made:

- Eight exhaust's species were considered (CO, CO₂, N₂, NO, O₂, H₂O, C and C₆H₁₄) in calculation of exhaust exergy.
- All species comprised carbon were considered as ideal gas.
- Mole fraction of species were not changed throughout the turbocharger.
- The fuel used in the analysis was considered as C_{14.59}H_{24.86}.
- Exhaust species were assumed to be in chemical equilibrium.

Since we are interested only in turbine, the process occurred in steady state with one inlet and outlet steady flow condition. The first and second law of thermodynamics can be written as, [6]

$$\dot{m}[h_2 - h_1 + \frac{1}{2}(V_2^2 - V_1^2) + g(z_2 - z_1)] = \sum_{k=1} \dot{Q}_k \quad (14)$$

$$+ \dot{Q}_0 + \dot{W}$$

$$\dot{m}(s_2 - s_1) - \frac{\dot{Q}_0}{T_0} - \sum_{k=1} \frac{\dot{Q}_k}{T_k} = \dot{S}_{gen} \quad (15)$$

Eliminating \dot{Q}_0 between the two equations yields the following exergy balance equation.

$$\dot{W} = \sum_{k=1} (1 - \frac{T_0}{T_k}) \dot{Q}_k + \dot{m}(\psi_1 - \psi_2) - T_0 \dot{S}_{gen} \quad (16)$$

In Eq.(16) ψ is the specific flow exergy, it can be calculated by Eq.(17)

$$\psi = h - h_0 - T_0(s - s_0) + \frac{1}{2}V^2 + gz \quad (17)$$

Since the total entropy production rate, $\dot{S}_{gen} \quad (total)$ is strictly positive. The power becomes maximal for vanishing entropy production i.e. for reversible process

[$\dot{S}_{gen} \quad (total) = 0$]. Then the reversible work equal:

$$\dot{W}_{rev} = \sum_{k=1} (1 - \frac{T_0}{T_k}) \dot{Q}_k + \dot{m}(\psi_1 - \psi_2) \quad (18)$$

In turbocharger $T_k = T_o$ so:

$$\dot{W}_{rev} = -\dot{m} \Delta \psi = \dot{m}(\psi_1 - \psi_2) = \dot{m}[h_2 - h_1 - T_0(s_2 - s_1)] \quad (19)$$

For turbine with inlet condition (T_1, P_1) and outlet condition (T_2, P_2) and the mole fraction of components in exhaust gas are y_i , the reversible work gives maximum turbine work output as:

$$\dot{W}_{rev,t} = \sum y_i (\bar{\psi}_1 - \bar{\psi}_2) = \sum y_i [\bar{h}_1 - \bar{h}_2 - T_0(\bar{s}_1 - \bar{s}_2)] \quad (20)$$

$$\dot{W}_{rev,t} = \dot{n}_e \cdot \bar{w}_{rev,t}$$

Since the temperature difference of compressor and ambient is small, the compressor is assumed adiabatic and changes in kinetic and potential energies are negligible. Consequently first law of thermodynamics can be written as :

$$\dot{m}_{air}(h_4 - h_3) = -\dot{W}_c \quad (21)$$

In a turbocharger, the turbine is mechanically linked to the compressor, so:

$$\dot{W}_T = \frac{-\dot{W}_c}{\eta_m} \quad (22)$$

In Eq.(22), η_m is turbine mechanical efficiency. The mechanical losses are mainly bearing friction losses [7]. In this paper, the mechanical efficiency is assumed equal to 97%.

Consequently turbocharger second law efficiency can be written as,

$$\eta_{II} = \frac{\dot{W}_t}{\dot{W}_{rev,t}} = \frac{(\frac{-\dot{W}_c}{\eta_m})}{\dot{n}_e \sum y_i (\bar{\psi}_1 - \bar{\psi}_2)} \quad (23)$$

$$= \frac{\dot{m}_{air}(h_4 - h_3)}{\eta_m \dot{n}_e \sum y_i (\bar{\psi}_1 - \bar{\psi}_2)}$$

4. Experimental results and discussion

In order to investigate the influence of engine parameters on second law efficiency of turbine in turbocharged diesel engine, the following engine parameters presented against the turbocharger second law efficiency.

4.1. Effect of bmep on turbocharger second law efficiency

Figure 2 shows the influence of bmep on turbocharger second law efficiency at different constant speeds. As illustrated in the figure, turbocharger second law efficiency is increased with the increase of bmep.

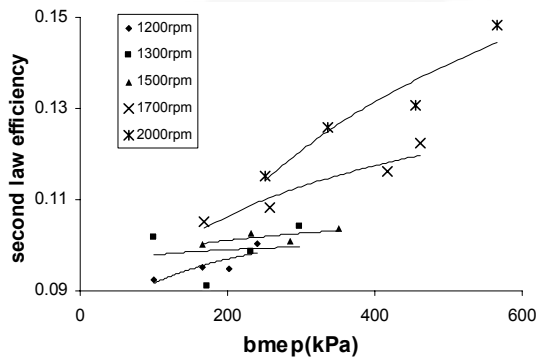


Figure 2: Effects of bmep on second law efficiency

It is obvious that as engine speed is increased, slope of the second law efficiency against bmep is promoted. Also with increase of bmep in constant speed, work of turbine and exhaust exergy drop in turbine is increased, but percentage of increase of work is higher than percentage of increase of exhaust exergy drop as shown in figure 3 for speed of 1200rpm. Also for other speeds percentage of increase of work is higher than percentage of increase of exhaust exergy drop as shown in figures 4,5,6. This causes that the turbocharger second law efficiency is increased.

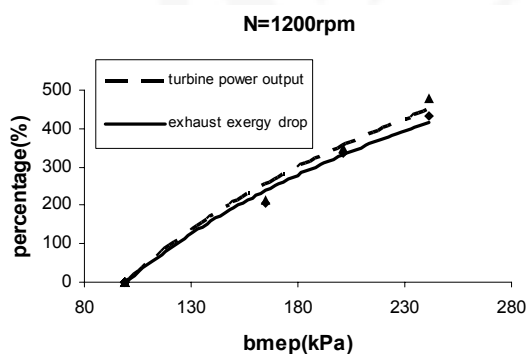


Figure 3: Percentage of increase of turbine power output and exhaust exergy drop against bmep

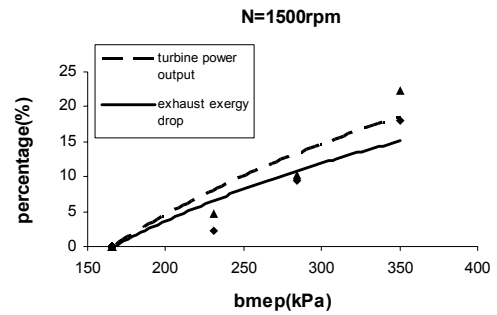


Figure 4: Percentage of increase of turbine power output and exhaust exergy drop against bmep

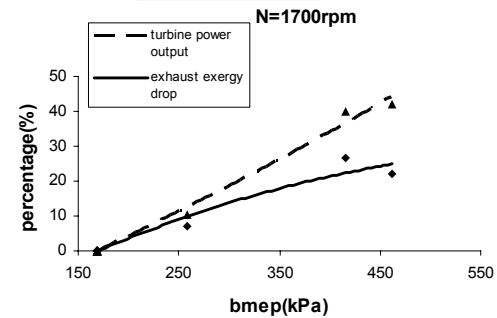


Figure 5: Percentage of increase of turbine power output and exhaust exergy drop against bmep

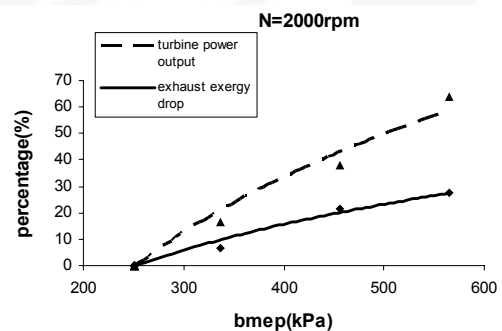


Figure 6: Percentage of increase of turbine power output and exhaust exergy drop against bmep

4.2. Effect of equivalence ratio (ϕ) on turbocharger second law efficiency

Figure 7 shows the influence of equivalence ratio on turbocharger second law efficiency. It is clear from this figure, as ϕ is increased the turbocharger second law efficiency is increased.

As ϕ is increased the inlet turbine temperature is increased as shown in figure 8, consequently second law efficiency of turbine is increased as H. Struchtrup et.al, showed [8].

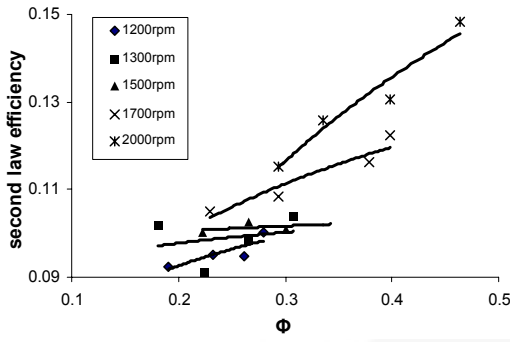


Figure 7: Effects of ϕ on second law efficiency

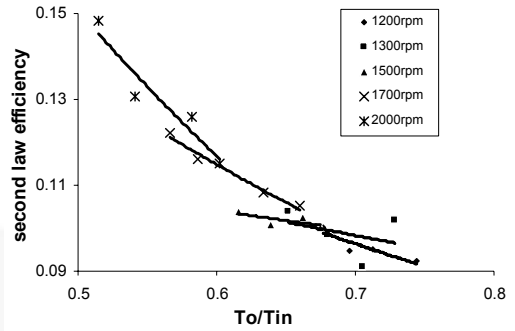


Figure 10: Effects of $\frac{T_o}{T_{in}}$ on second law efficiency [8]

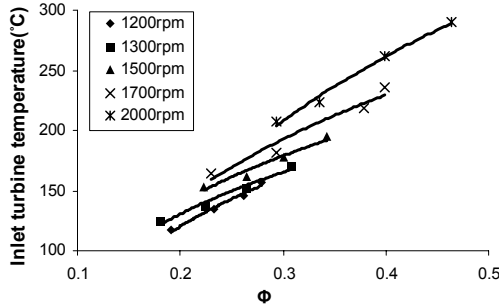


Figure 8: Inlet turbine temperature against equivalence ratio

4.3. Effect of inlet turbine temperature on turbocharger second law efficiency

As illustrated in figure 9 higher inlet turbine temperature causes the higher second law efficiency.

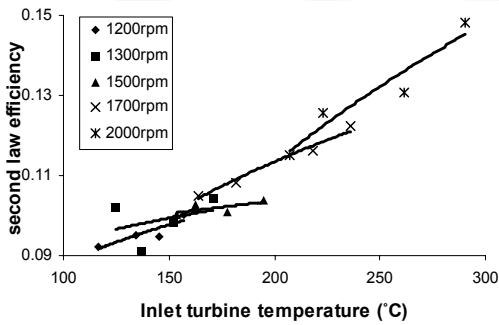


Figure 9: Effects of inlet turbine temperature on second law efficiency

As torque is increased, the inlet turbine temperature is increased because with increase of the torque, the charge of engine is promoted and this causes the higher work with constant compression ratio and the higher temperature and pressure in cylinders consequently the temperature of exhaust gas is increased. Also in higher speeds the slope of efficiency against inlet turbine temperature is greater.

H. Struchtrup et.al, showed with increasing $\frac{T_o}{T_{in}}$ (reduction of inlet turbine temperature), second law efficiency of turbocharger is decreased. This is similar to our results [8] (figure 10).

With additional inlet turbine temperature, the work of turbine is increased as enthalpy drop in turbine is increased. But with attention to Eq.(20), entropy drop makes the increase of work of turbine greater than the increase of exergy drop. This causes increasing the turbocharger second law efficiency (figures 11,12,13, 14).

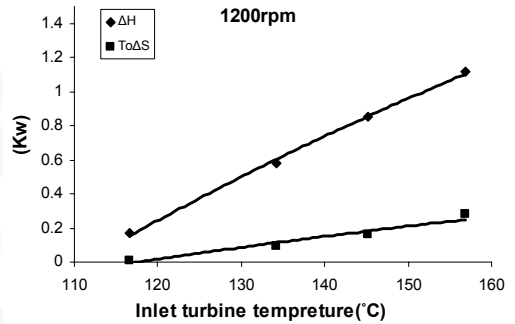


Figure 11: Variation of ΔH and $T_o \Delta S$ with inlet turbine temperature

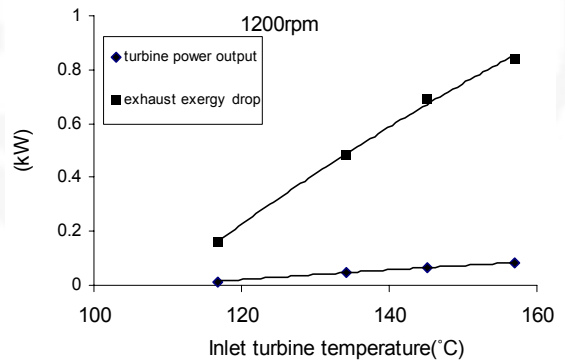


Figure 12: Variation of turbine power output and exhaust exergy drop with inlet turbine temperature

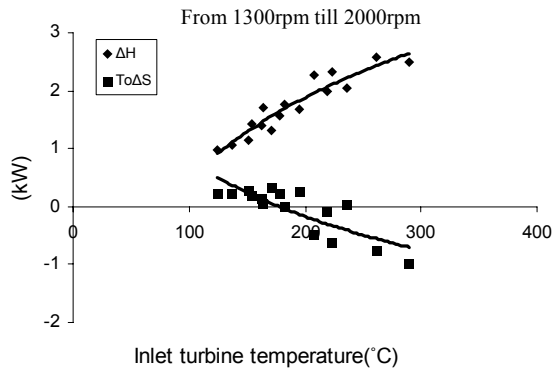


Figure 13: Variation of ΔH and $T_0\Delta S$ with inlet turbine temperature

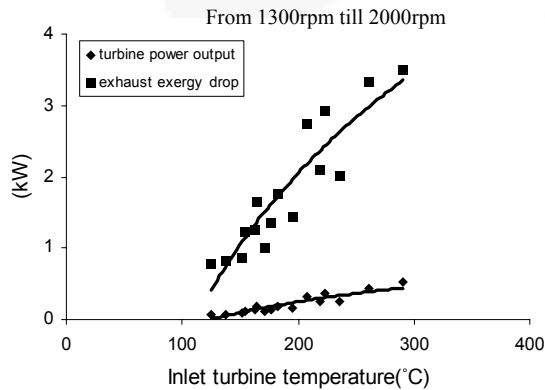


Figure 14: Variation of turbine power output and exhaust exergy drop with inlet turbine temperature

increasing turbocharger second law efficiency (figures 16, 17, 18, 19).

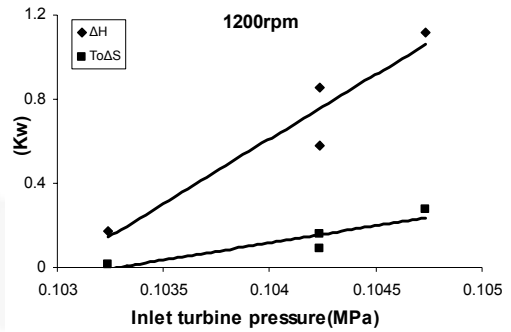


Figure 16: Variation of ΔH and $T_0\Delta S$ with inlet turbine pressure

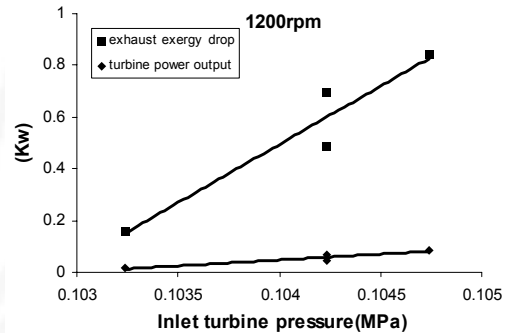


Figure 17: Variation of turbine power output and exhaust exergy drop with inlet turbine pressure

4.4. Effect of turbine inlet pressure on turbocharger second law efficiency

Figure 15 shows the influence of inlet turbine pressure on turbocharger second law efficiency. As shown in figure, with increase of inlet turbine pressure the turbocharger second law efficiency is increased. It is also obvious that the inlet pressure turbine is greatly affected by the engine speed, as engine speed is increased; the turbocharger second law efficiency is also increased.

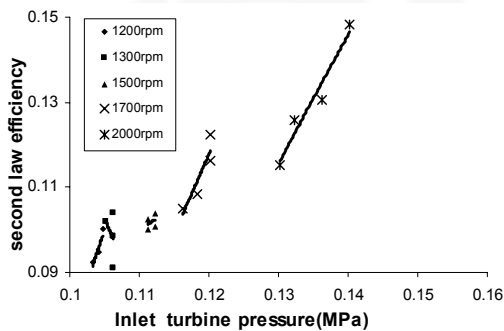


Figure 15: Effects of inlet turbine pressure on second law efficiency

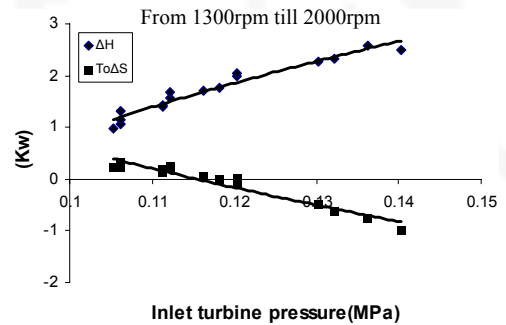


Figure 18: Variation of ΔH and $T_0\Delta S$ with inlet turbine pressure

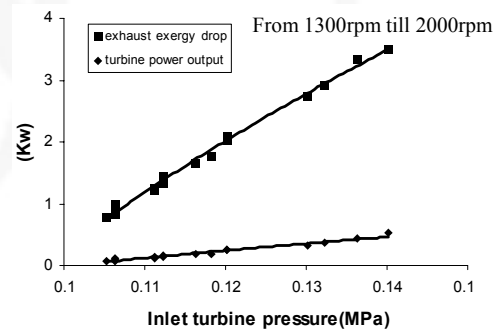


Figure 19: Variation of turbine power output and exhaust exergy drop with inlet turbine pressure

With increase of inlet turbine pressure, the work of turbine is increased as enthalpy drop in turbine is increased. But entropy drop makes increase of turbine greater than the increase of exergy drop. This causes

5. Conclusion

An experimental study to investigate the effect of engine parameters on second law efficiency of turbocharger was carried. The following results were achieved:

1- Inlet turbine temperature:

As illustrated in this paper, with increase of bmep, exhaust gas temperature or in other words inlet turbine temperature is increased. This causes that the turbocharger second law efficiency is increased.

2- Speed of engine:

As speed of engine is increased, the inlet turbine temperature is increased consequently the turbocharger second law efficiency is increased.

3- Exhaust pressure:

As described in this paper with increase of inlet turbine pressure, the turbocharger second law efficiency is increased.

4- Effect of turbocharger second law efficiency on transient operation in turbocharged diesel engines:

With additional turbocharger second law efficiency, the turbocharger feed back in transient running conditions could be improved. This is because of better available work transfer between engine cylinder and turbocharger. So by higher engine speed, higher inlet turbine temperature and pressure and higher fuel-air ratio, the transient operation of turbocharged diesel engine may be improved.

List of Symbols

A_o	orifice area (m ²)
Bmep	brake mean effective pressure(kPa)
C_d	discharge coefficient of orifice
DI	direct injection
g	gravitational acceleration (m/s ²)
h	enthalpy (kJ/kmole)
\dot{m}	mass flow rate (kg/s)
\dot{n}	mole flow rate (kmole/s)
P	pressure (kPa)
\dot{Q}	heat transfer (kW)
$S_{gen})_{total}$	total entropy generation (kJ/K)
s	entropy (kJ/kmole K)
T	temperature(K)
t_f	time for fuel consumption of 50cc (s)
V	velocity (m/s)
\dot{V}	volume flow rate(m ³ /s)
w	specific work (kJ/kmole)
\dot{W}	power (kW)
y	mole fraction of species
z	height (m)
$\Delta h_{orifice}$	monometer pressure drop

ψ	specific flow exergy (kJ/kmole)
η_m	mechanical efficiency (%)
η_{II}	second law efficiency (%)
ρ	density (kg/m ³)

Subscripts

c	compressor
e	exhaust
f	fuel
i	species
k	external reservoir
l	Liquid used in manometer
rev	Reversible
o	ambient
t	turbine
1	turbine inlet
2	turbine outlet
3	compressor inlet
4	compressor outlet

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