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REPRESENTING A METHOD FOR COOLING THE AIR INTAKE TO THE RECIPROCATING SPARK- IGNITION ENGINE WITH HIGH COMPRESSION RATIO AND ITS EFFECT ON ENGINE PERFORMANCE BY MODELING

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ABSTRACT

The present paper have been studied the effect of air intake cooling to the reciprocating spark – ignition engine with high compression ratio on their energy consumption, power and output torque. Use of vehicle cooling system has been proposed for cooling the engine air intake. In the proposition, vehicle gas cooling system deriver will be a turbine connecting to exhaust that obtained its power from hot and high- pressure gases exited from exhaust by reducing a gear box, the power of turbine transfers to vehicle compressor of cooling system. Hereinafter, in the context of the paper, proposed component of turbine, gearbox and mentioned compressor is known as "Turbo- cooler". In simulation of ignition process, the Woschni heat transfer model with successive approximate method have been used and all thermodynamic properties containing intake air, type of fuel and ignition yield and mechanical properties containing ring and piston wall friction have been computed.

Key words: Modeling, Engine, Reciprocate, Spark Ignition, High Compression Ratio, Volumetric and Thermal Efficiency, Cooling System and Turbo-Cooler

INTRODUCTION

High energy consumption of reciprocating engines with crankshaft sensor (super charging) or high- pressure burned gases stimulus (turbo charging) is one of the common methods to increase volumetric efficiency of such engines **[1]**. On the other hand, desired ignition of gasoline fuel is limited to pressure and temperature of air intake. In high temperature and pressure, ignition will be occurred unusual and non-desirable **[1, 2, 6, 7]**. Therefore, simultaneous with super charging or turbo charging of the engines for reaching the desired performance of turbo charging, the compression ratio should be decreased in spark- ignition engine (SI); however, decrease of compression ratio is coincided with thermal efficiency decrease **[2]**. In addition, the problems arising from increase of noise in vehicle performance and necessary to strengthen the engine by increasing the mean effective pressure and expensive changes of super charging or turbo charging without the effect on decrease of fuel consumption has caused that the changes do not apply in public and urban vehicles **[1]**. The occurrence of undesirable knock ignition in SI engines significantly is decreased simultaneously with decrease of air intake temperature. For this reason, in engines having super charging or turbo charging is used for one or some inter cooler for cooling hot and high- pressure intake air to the engine. Although in SI engines simultaneous with increase of compression ratio, thermal efficiency is increased and burned exhaust gas pressure in the engines will be decreased, but yet, the significant percent of engine power with high compression ratio is lost through the exhaust gases **[8]**.

For increasing the volumetric efficiency and decreasing the pumping efficiency of spark ignition engine with high compression ratio, increase number of intake and smoke air valves per cylinder usually is used **[1]**. It leads to facilitate the entrance and exit of gases inside in cylinder. In addition, it causes the intake air pressure drop, as a result the pumping performance and volumetric efficiency of engine is improved. In this regard, some studies have been done for thermodynamic process simulation and modeling the internal friction and mechanical properties **[5]**. For significant and effective cooling of air intake of engine especially in the warm seasons, lightweight and profitable gas cooling system can be used. Since cooling the engine intake air by vehicle gas cooling system causes to high consumption energy that is obtained with engine fuel, it is not economical that its profits. One of the ways to reach the engine cold intake air without high consumption and high compression ratio is leading the wasted power of high- pressure exhaust gases to provide the requirement power of gas cooling system. In the present paper, computer modeling of gasoline engine air intake process without high consumption system with high compression ratio have been represented and the study will be conducted with forecast aim and estimation of operating performance **[10]**. In the simulation, it have been tried to consider the effective mechanical and thermal parameters in the study in order to be similar the result of the study with probable results. For achieving the aim, ignition process modeling and Woschni heat transfer model by successive approximation method have been used. In addition, all thermodynamically properties of air intake, type of fuel, combustion yield and time of ignition, cylinder wall temperature and heat losses from the cylinder walls and also mechanical properties containing the friction of cylinder wall with piston skirt and rings and physical and mechanical behaviors of slider and crank mechanism by considering the necessary masses of all reciprocated components have been considered in the simulation and have been concluded with the obtained results.

Introducing the turbo cooler and its performance

In a turbo cooler which is introduced with final aim of decrease the fuel consumption energy in public vehicles, the main component for installation on public vehicle engine is a turbine and planetary gearbox that is related to vehicle cooler compressor. For preparing the mentioned turbine, the used turbine in a turbo charging of SI engines used which is connected to a planetary gearbox with appropriate capacity and conversion ratio and its duty is transferring the wasted power of hot gases and high pressure exhaust gases engine to the compressor of vehicle cooling system. The components of a turbo cooler are as follows:

According the figure 1 in section 1, an impulse turbine among the applied type in the common turbo charging in SI engine have been proposed suitable with volumetric capacity of engine. In section 2, considering the speed of high axis of turbine, journal bearing will be used. Turbine axis to sun gear wheel in planetary gear box has been connected. Figure 3 is gear box output connected to Polly 4 which has been connected to a compressor of vehicle gas cooling system by a belt to the 5 clutch pulley. In addition, output pulley of gearbox directly is connected to the fan blower **[7]**.hence, secondary evaporator **[8]** is placed in the path of input filtered air which is requirement for the engine and then enters to the gearbox after passing the vehicle cabin.

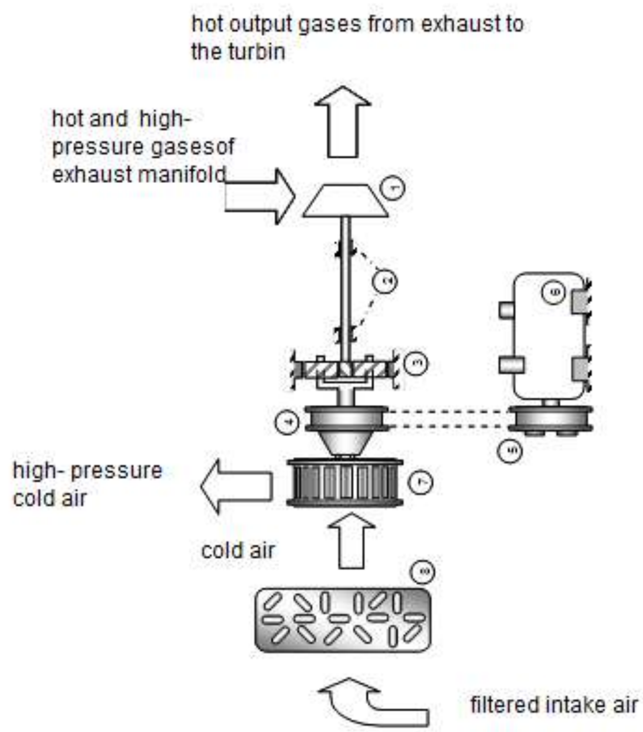

Figure 1: the component of a turbo cooler

3. Simulation of thermodynamic process

For simulating the thermodynamic process in a SI engine, the first law of thermodynamic can be written based on the losses and irreversibility concepts of energy and operation in an ignition engine (1,2,5):

$$(\delta Q_{in} - \delta Q_{loss}) - (PdV - \delta W_{irrev}) = dU \quad \dots (1)$$

The internal energy changes of an ideal gas with constant specific heat can be written as follow:

$$dU = mC_v dT_g \quad \dots (2)$$

By differentiating the equation of state, we have an ideal gas:

$$PdV + VdP = mR_g dT_g \quad \dots (3)$$

By combining the equation 2 and 3, the following results will be obtained:

$$dU = \frac{C_v}{R_g}(PdV + VdP) \quad \dots\dots (4)$$

By considering the effect of ring and skirt friction, for determing the properties of irreversible detail of work it can be written:

$$\delta W_{irrev} = \left(\mu \frac{du}{dy}\right)_{skirt} L_{skirt}\pi D dx + \left(\mu \frac{du}{dy}\right)_{ring} L_{ring}\pi D dx \quad .....(5)$$

In the mentioned equation, linear speed of piston can be determined using Kenematic analysis of slider- crank mechanism. If crank radius and piston rod length is represented with R and L, considering figure 2 we have:

$$\dot{x}_A = l\dot{\alpha}\cos\alpha - R\dot{\theta}\sin\theta \quad ...(6)$$

$$\dot{\alpha} = \frac{-R\dot{\theta}\cos\theta}{\left(l^2 - R^2\sin\theta\right)} \quad ......(7)$$

Considering that the linear speed of piston is $U_p(\theta) = \dot{x}_A(\theta)$,as a result the equation 8 is as follow (5):

$$\delta W_{irrev} = \mu\pi D dx U_p(\theta)\left(\frac{L_{skirt}}{C} + \frac{L_{ring}}{\varepsilon}\right) \quad ....(8)$$

In equitation 8, ε is oil film thickness between cylinder wall and ring and *C* represent oil film thickness between piston and cylinder wall and it is determined from experimental results.

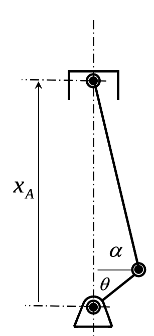

Figure 2: Slider and crank mechanism

In addition, in equation 1 for calculating δQ_{in}, total amount of heat released in mass of fuel consist of:

$$Q_{in} = m_f LHV \quad ...(9)$$

The amount of released energy is varied with crank angle which is obtained from the following equation:

$$\delta Q_{in}(\theta) = Q_{in} dx_b \quad .... (10)$$

Where, x as the function of fuel combustion rate consist of:

$$x_b = 1 - \exp\left(-5\left(\frac{\theta - \theta_s}{\Delta\theta}\right)^3\right) \quad .... (11)$$

Where, in the equation $\Delta\theta$ is total amount of ignition efficiency. The amount of waste heat by displacement equations proportional to volatile level of heat is as follows:

$$\delta Q_{loss} = \frac{h_{cg} A_h}{\dot{\theta}}(T_g - T_w)d\theta \quad .... (12)$$

By replacing the equation 4, 6, 10 and 12 in equation 1 and setting this equation based on dP and $d\theta$, we have:

$$\frac{dP}{d\theta} = \frac{k-1}{V}\left(Q_{in}\frac{dx_b}{d\theta} - \frac{h_{cg}A_h}{\dot{\theta}}(T_g - T_w)\right) - \frac{k}{V}P\frac{dV}{d\theta} + \frac{k-1}{V}\left(\mu\pi D U_p\left(\frac{L_{skirt}}{C} + \frac{L_{ring}}{\varepsilon}\right)\right)\frac{dx}{d\theta} \quad ..... (13)$$

Where, cylinder volume at angle θ consist of

$$V(\theta) = V_c + \frac{\pi D^2}{4}x(\theta) \quad .... (14)$$

The level of heat transfer consists of (5):

$$A_h(\theta) \cong \frac{\pi D^2}{2} + \pi D x(\theta) \quad \text{…. (15)}$$

In addition, considering heat capacity at constant volume, pressure and gases can be used for estimating the changes of heat capacity properties in temperature range from 300 to 3500 K and also the following equation is used per the component of 78.1 % nitrogen, 20.95% oxygen, 0.92 % argon and 0.03% dioxide carbon (4):

$$\begin{aligned} C_p &= 2.506 \times 10^{-11} T_g^2 + 1.454 \times 10^{-7} T_g^{1.5} \\ &- 4.246 \times 10^{-7} T_g + 3.162 \times 10^{-5} T_g^{0.5} \\ &+ 1.3303 - 1.512 \times 10^4 T_g^{-1.5} \\ &+ 3.063 \times 10^5 T_g^{-2} - 2.212 \times 10^7 T_g^{-3} \end{aligned} \quad \text{….(16)}$$

For determining the temperature changes of gases inside in the cylinder, we have the state equation:

$$T_g = \frac{P(\theta)V(\theta)}{mR_g} \quad \text{….. (17)}$$

The changes of thermal expansion coefficient based on the crank angle changes according the Woschni model is defined as follows:

$$h_{cg} = 3.26 D^{-0.2} P^{0.8} T_g^{-0.55} w^{0.8} \quad \text{…. (18)}$$

Where, *w* is combustion speed and it is calculated as follows:

$$w(\theta) = \left(C_1 (4R\omega) + C_2 \frac{V_d T_{gr}}{P_r V_r} \left(P(\theta) - P_m \right) \right) \quad \text{…. (19)}$$

Where, P_r, V_r and V_r respectively are pressure , volume and temperatures of gases inside in cylinder immediately after closing the intake valve. P_m is pressure inside cylinder per crank changes which is occurred in the condition without ignition. The terms of C_1 and C_2 are the constant ratios which are C_1= 2.28 and C_2= 0 during compression and without ignition and during ignition and expansion, they are C_1= 2.28 and C_2= 0.0324 (5).

3.2. Thermodynamic effect of gas cooling system on air

For considering the thermodynamic process of cooling system, the general equations dominant on the systems have been used and then, based on the density valve of cooling system types and the results related to each data in the next chapter, the calculations will be represented. For this aim, performance coefficient of a cooling system consists of:

$$COP = \frac{\dot{Q}_c}{\dot{W}_{in}} \quad \text{….(20)}$$

Where, W_{in} is gas consumption rate for producing cooling (consumption power) and $\dot{Q}_c$ is heat energy rate obtained from cooling space (cooling power0 and consist of :

$$\dot{Q}_c = \dot{m}_a (h_{2a} - h_{1a}) \quad \text{….(21)}$$

It is important to note that in the above equation, m_a is mass rate of air intakes.in addition, h_{1a} and h_{2a} are enthalpies of the air intakes (warm) and outlets (cool) from turbo cooler system.

For determining the enthalpy changes of air intakes can be used existence gases in air and polynomial equations used for determining the properties of environmental air (8). The combination of ambient air with combination percent (molar ratio) have been shown in the followings.

$$y_1 = y_{CO_2} = 8.45\%,\ \ y_2 = y_{H_2O} = 14.69\%$$
$$y_3 = y_{N_2} = 69.46\%,$$
$$y_4 = y_{O_2} = 1.4232e-5\%$$
$$y_5 = y_{CO} = 5.34\%,\ \ y_6 = y_{H_2} = 2.05\%$$
$$y_7 = y_H = 4.2357e-3\%,$$
$$y_8 = y_O = 4.5753e-6\%$$
$$y_9 = y_{OH} = 3.0183e-3\%,$$
$$y_{10} = y_{NO} = 6.2883e-6\% \quad \text{....(22)}$$

Considering the molar mass of each gas in term of kg/kmol consist of (8):

$$M_1 = M_{CO_2} = 44.01,$$
$$M_2 = M_{H_2O} = 18.015,$$
$$M_3 = M_{N_2} = 28.013,$$
$$M_4 = M_{O_2} = 31.999;$$
$$M_5 = M_{CO} = 28.01,\ M_6 = M_{H2} = 2.016;$$
$$M_7 = M_H = 1.008,\ M_8 = M_O = 16.00;$$
$$M_9 = M_{OH} = 17.007;$$
$$M_{10} = M_{NO} = 30.006; \quad \text{.... (23)}$$

By using the equations 22 ad 23, molar mass of air with above components consists of:

$$M_0 = \sum_{i=1}^{10} y_i M_i \quad \text{.....(24)}$$

For obtaining the air enthalpy in the specific temperature rage 300≤T≤1000 k, the reference processing results has been used, then, we have for enthalpy of each gas:

$$h_i(T) = R(a_{i1}T + \frac{a_{i2}}{2}T^2 + \frac{a_{i3}}{3}T^3 + \frac{a_{i4}}{4}T^4 + \frac{a_{i5}}{5}T^5 + a_{i6}) \quad i = 1,...,10 \quad \text{.....(25)}$$

Where a_{ij} ($i = 1,...,10$ و $j = 1,...,6$) coefficients is in reference [8]; as a results, by mentioned equation and using equation 24, air enthalpy of desired temperature will be:

$$h_a(T) = \frac{1}{M_0}\sum_{i=1}^{10} y_i h_i(T) \quad \text{....(26)}$$

By using the equtation 20 and 21, and mentioned results based on the comnsumption power of cooling system and thermodynamic properties of air intake to the engine, it can be possible to study the effect of tubo cooler on decreasing temperature of the intake air.

4- Engine Brake Torque Analysis

The equitation 6 and 7 has been used for piston acceleration:

$$\ddot{x}_A = l\ddot{\alpha}\cos\alpha - l\dot{\alpha}^2\sin\alpha - R\ddot{\theta}\sin\theta - R\dot{\theta}^2\cos\theta \quad \text{.... (27)}$$

$$\ddot{\alpha} = \frac{\left(R\dot{\theta}^2\sin\theta - R\ddot{\theta}\cos\theta\right)\left(l^2 - R^2\sin^2\theta\right)}{\left(l^2 - R^2\sin^2\theta\right)^{\frac{3}{2}}} - \frac{\left(R^3\dot{\theta}^2\sin\theta\cos^2\theta\right)}{\left(l^2 - R^2\sin^2\theta\right)^{\frac{3}{2}}} \quad \text{.....(28)}$$

As the piston mass depicted with m_p and by assuming the connecting-rod as a member of two concentrate masses (by ignoring the effects of inertia of mass center) and also the effect of ring friction and piston skirt is F_f in the opposite direction of piston movement, we have the following equitation:

$$F_f = \frac{\delta W_{irrev}}{dx_A} \quad .....(29)$$

$$F' = -\frac{m_p a_p + F \pm F_f}{\sin\alpha} \quad ....(30)$$

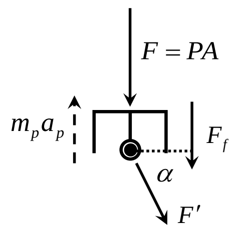

Figure 3: approximation of incoming forces on a piston

Finally the torque transferred in to crankshaft or brake torque consists of:

$$T = F'R\cos(\theta - \alpha) \quad .... (31)$$

It should be noted that torque T is the engine output torque and it should be considered in the solutions of the calculations.

RESULTS AND DISCUSSION

5.1. Combustion analysis

Considering the used method in reference (5), solving the equitation 13 in -180≤θ≤180 and by selecting the suitable changes step, it can be obtained the pressure changes inside the cylinder. Considering *P*, *k*, h_{cg} and T_g variables are related to each other in solving the above equitation, at the first step it should be considered the gas pressure in TDC. For assuming the pressure changes per Δθ, the following equation is obtained:

$$\frac{dP}{d\theta} = \frac{P(\theta) - P(\theta - \Delta\theta)}{\Delta\theta} \quad ... (32)$$

In equation 13 and solving it simultaneous with its equations which consists of *P*, *k*, h_{cg} and T_g and by successive approximation method, it can be obtained the *p(θ)* value based on the determined accuracy of solving, however, for the next steps after the more accurate approximation, the equation 32 have been used and the pressure distribution *p(θ)* have been obtained.

$$\frac{dP}{d\theta} = \frac{3P(\theta) - 4P(\theta - \Delta\theta) + P(\theta - 2\Delta\theta)}{2\Delta\theta} \quad ....(33)$$

In addition, the parameter Q_{in} is entered after the spark plugs in θ_s and it is continued until during the combustion θ_d. The time of spark plug may be set based on the most output operation. Then, the indicated thermal efficiency and mechanical thermal efficiency by applying the friction effects, the following equations are obtained:

$$w_i = \oint p dV \quad .... (34)$$

$$\eta_i = \frac{w_i}{Q_{in}} \quad ....(35)$$

$$\eta = \frac{w_i - w_{irre} - w_{pump}}{Q_{in}} \quad ...(36)$$

Where in equation 36, the term of W_{pump} is the operation of pump engine. Considering the aim of the present paper which is the comparison between engine performances with or without turbo cooler the pumping work have been approximated as $W_{pump} = (P(180^{\circ}) - P_r)V_d$ by difference of intake and outlet air pressure in both situations.

Finally, the domain relations on brake average torque, brake specific fuel consumption, output power for a four – stroke spark - ignition engine consists of:

$$T_{ave} = \frac{1}{4\pi}\int_0^{2\pi} T d\theta \quad \text{....(37)}$$

$$BSFC = \frac{m_i}{w_i - w_{irre} - w_{pump}} \quad \text{... (38)}$$

$$H = T_{ave}.\omega \quad \text{.... (39)}$$

For studying the turbo cooler effects, the following examples are represented and the change of some parameters is investigated during it. In addition, crank and slider components with features of table 1 have been considered.

For better analysis of intake air cooling on the performance of engine performance, in the section, the torque, power , fuel and engine thermal efficiency diaphragms have been simulated with or without a turbo cooler. Furthermore, the effect of turbo cooler as decrease of temperature and partial increase of pressure due to blower connected to a turbo cooler as separately or simultaneously is seen in diagrams.

Considering the moisture in the air and icing phenomenon in the evaporator of gas cooler, decrease of temperature of intake air with turbo cooler simulation is limit to the constant temperature. In addition, blower turbo cooler will be provided the necessary power for supplying input constant pressure (P_{m}). With regard to the limit of cooling system capacity and also blower of turbo cooler system, as the situation of output power dissipation of engine performance is more than the requirement power in the turbo cooler system, the additional pressure of exhaust manifold is transferred to the environment.

By mentioned assumptions and temperature changes of engine intake air in first step and also the changes of input pressure in second step, the diagrams of brake power changes, brake torque and brake specific fuel consumption have ben represented in the **Figures 4-6**.

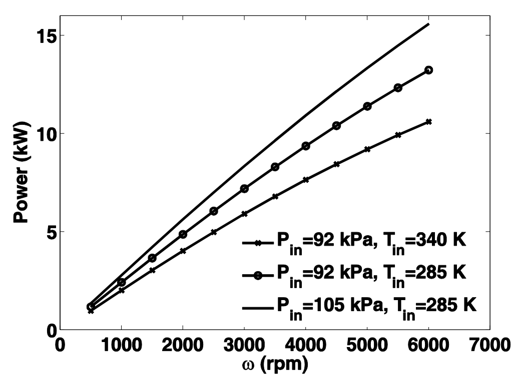

Figure 4: power changes than angular output speed of engine in the situation of different temperatures and intake air pressure

Engine intake air temperature intentionally may assumed maximum (T_r= 340k) in order to show the turbo cooler performance exchanges on the engine parameters.

For comparing and concluding the effect of turbo cooler on spark ignition engine fuel consumption, it is fair that the effect of it in the situation of engine output torque is considered for two states of using turbo cooler and no- using the turbo cooler. In this situation, the present paper concluded the figure 7 with changing the equivalence ratio or injected fuel to the engine and reaching the equal output torque in two states.

In addition, brake thermal efficiency with the same assumption of constant brake torque in two different situations under the effect of turbo cooler is obtained as the following and the effect of fuel consumption improvement and brake thermal efficiency will be seen in the figures 7 and 8.

Based on the obtained results from the simulation, the **Figure of 4 and 5** have represented 38 percent of brake torque growth arising from additional pressure and reducing the intake air temperature due to the effect of use in the turbo cooler. However, figure 6 simultaneously represents the specific fuel consumption decrease (about 2 kg decrease pre each Kw/h).

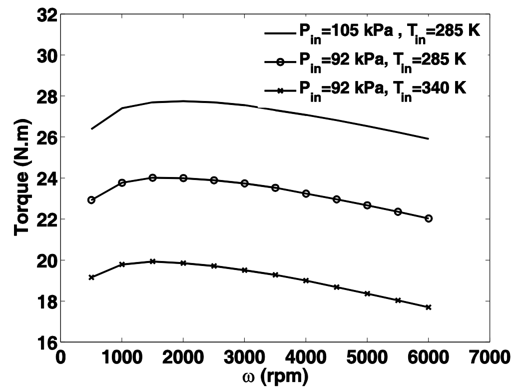

Figure 5: torque changes than the angular output speed of the engine in different situation of intake air temperature and pressure

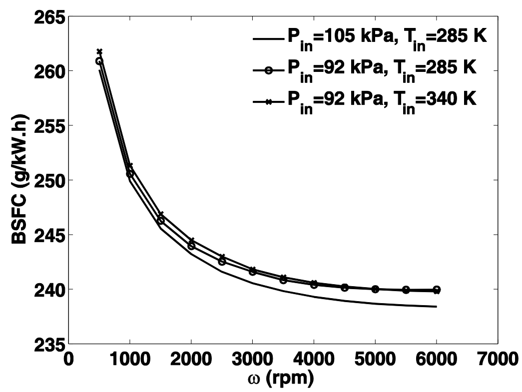

Figure 6: brake fuel specific consumption changes than the angular output speed of the engine in different situation of intake air pressure and temperature in an equivalence ratio of specific fuel (φ= 0.9).

The obtained results of **Figure 7 and 8** represents the significant effect of turbo cooler in the brake output torque for the spark ignition engine with high compression ratio. Based on the results of thermal efficiency of the system, nearly 2.8 percent have been improved in all angular speed of engine and also specific fuel consumption have been decrease about 8 g/kW.h. The main reason of the significant decrease can be found in appearance of lean burn combustion simultaneous with use of turbo cooler which will be increased the thermal efficiency.

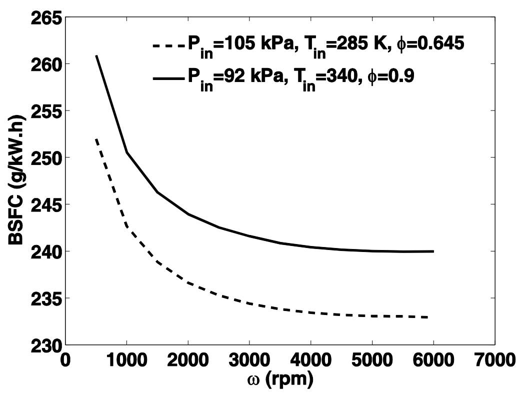

Figure 7: brake specific fuel consumption than the angular output speed of the engine in the equal output engine torque

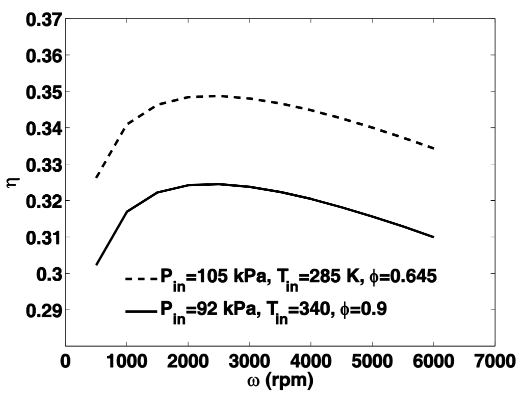

Figure 8: brake thermal efficiency changes than the angular output speed in the equal output engine torque

CONCLUSION

In the recent paper which have been set with the aim of the inhibition of wasted energy by the engine exhaust emission to decrease the energy consumption in SI engine with high compression ratio have been introduced a tool called turbo-cooler. The instrument can improve the mechanical and thermal performance of spark ignition engine with the high compression ratio by cooling the intake air and increasing its pressure. The source of turbo cooler energy is high pressure and warm exhaust emission of the engine.

Based on the simulation, it generally can be concluded:

1- Considering the obtained results from figure 4 and 5, cooling the intake air and simultaneous increasing its pressure have the significant effect to increase the brake torque and output power of the engine

2- From the obtained figures it can be concluded hat cooling the intake air and simultaneous increasing its pressure can decrease the brake specific fuel consumption, however, in a specific engine if the main aim only is decreasing the engine energy consumption, not only the use of system such as the turbo cooler is necessary, but also the setting the central programs of control of fuel injection should be revised. In the situation, without the significant changes in the torque and engine output power, the specific fuel consumption will be decreases such as the figure 7. However, without revision of fuel controlling system, specific fuel consumption will significantly not be decreased as the figure 6 shows an only the engine output torque will be significantly is increased.

3- Due to the temperature effect and partial increase of intake air pressssure to the engine based on the obtained results in the figure 8, the thermal waste of the engine will be decrease that helps to decrease the global warming and especially the urban environment and urban traffic environment. In the quantified results obtained from the figures of simulation, it should be considered that there are the quantities which have been ignored in the paper as the follows:

1- Due to the use of turbine in the hot exhaust gas pass of the engine based on the requirement power of the turbo cooler, the pumping losses of the engine will be increase which has been ignored in the calculations.

2- Due to the use of turbo cooler, some of the moisture of the engine inlet air by cooling system absorbed which will improve the output parameters of the engine and it is not considered in the calculations (3).

3- The temperature situation of the environment have been assumed as the worse state which usually appears in the tropics, summer season, the traffic within the city and automobile stopping situations and the issue should be considered in the total conclusion.

Generally, it should be considered that different simple hypotheses have been used that can keep away the obtained results from the real results. However, exhoust high-pressure gas losses in the engine without turbo-charger is an important issue. In addition, controlling this lost power can be useful in order to improve the engine volumetric efficiency by increasing the pressure and decreasing the temperature or both of them in intake air quality.

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