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COMPARING PERFORMANCE CHARACTERISTICS OF A GASOLINE AND CNG ENGINES AND INCREASING VOLUME EFFICIENCY AND POWER USING DESIGNED TURBOCHARGER

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ABSTRACT: In recent years, the utilization of cleaner alternative fuels such as natural gas, methanol reformed gas (MRG), and hydrogen, is much more common, as a solution for environmental problems including global warming effect and for the shortage of crude oil reserve in the world. Power reduction is one of the important problems of converting gasoline engine to CNG and using the turbocharger can be a solution for increasing the engine power. In this paper, a model of a 4-cylinder multi-point fuel injection engine was prepared using a one dimensional fluid dynamics code for both gasoline fuel and CNG. The accuracy of the model was verified using experimental results of the engine testing showing good agreement between the model and the real engine. As a result, predictions are obtained that provide a detailed picture of engine performance condition with different fuel. In addition, two different turbochargers has been added to intake system and the engine performance characteristics like efficiency, net power and fuel consumption compared to CNG and gasoline engines without turbocharger. As a result, according to engine results and compressor characteristics a turbocharger was select for the based CNG engine.
KEYWORDS: Turbocharger, CNG, Volume Efficiency, Power

INTRODUCTION

Comparing CNG spray engines with gasoline engines, their power can be reduced about 10 to 15 percent. The two main reasons for this decline when injected into a CNG gaseous state that some of the intake air space may be occupied to the engine and it reduces the volumetric efficiency. Another reason for high ratio of the air to the fuel in gas stiochiometric conditions is high ratio of the gasoline, the figure is about 17.2 to 1 for gas and it is 14.7 to 1 for gasoline. This factor is determined more necessary for CNG engine to the air than gasoline engine to the air. And the amount of additional air to the engine can be provided by overfeeding engine.

In other words, if we can increase the amount of air pressure inlet in the engine, so it can also increase the amount of inlet air mass to the engine. Mass density of air entering the engine will cause the combustion to be complete which improves the thermal efficiency and therefore to compensate for CNG engines seems to be necessary [1].

Until now, much research on using the models of turbocharged engine optimization is done. Andreas et al in 2001 replaced the engine with compressed natural gas instead of gasoline and its effect on engine performance and they analyzed CFD codes using laboratory on a real engine and he used experimental results as well [1]. Amseden et al used the code KIVA 3V to model the combustion process. [2, 3] During a study that was conducted in 1996 by Johnson, he found out these changes can be reduced by keeping the fuel consumption of the engine can be increased to 10% and reduce the amount of pollutants [4.] Kordiner et al (1999) have been analyzed the changes in the feeding engine and got similar answers [5]; another problem in the feeding engine is the occurrence of Nak phenomenon that Grandien and Ångström examined this phenomenon and the resistance of the engine against the NAK phenomenon in 1999 [7,6]. David Garni in 2001 with the creation of a one-dimensional model has analyzed the turbocharged engine performance and his model can be used for modeling non-uniform flow [8]. In the current study a one-dimensional model is used to model XU7 engine. The data required for the model include:

1. Geometric data and lateral friction engine components,
2. Engine characteristics such as diameter, stroke, compression ratio, valve timing and ignition.
3. The performance of the engine with duct wall temperature, connections and engine components.

Some of the parameters such as the geometric characteristics of the components are obtained as direct measures and while calibrating the model with the real state be corrected.

The simulation engine air inlet geometry for all runners, flow of exhaust gases to filter carefully measured.

Table 1: Specifications of Engine

Bore	83	mm
Stroke	81.4	Mm
Bore/Stroke	1.019	
Displacement	1761	cm^3
Compression Ratio	9.3	
Intake Valve Opening	32.5	BTDC
Intake Valve Closing	64.3	ABDC
Exhaust Valve Opening	61	BBDC
Exhaust Valve Closing	15	ATDC

After Gasoline and CNG engine model, engine dynamometer test results of the modeling results using both validation and performance characteristics of the engine can be compared with each other. Finally, with regard to the design parameters of the turbocharger, the proper feeding system was chosen for the selected engine.

MODELING

A one-dimensional dynamic gas-source code used to analyze the dynamic wave pressure, mass flow and energy loss in a variety of systems. Able to model fluid flow using a variety of volume compressibility, duct, tubing, orifices, and applying the boundary conditions of the environment. Different types of graphs of various parameters on the output of the engine to the code below uses a variety of models referred to in engine is able to calculate the performance parameters. Based on the analysis of motor runner compression wave function is used [9]. Unsteady compressible flow equation in order to achieve this should be resolved. This equation used by the method of characteristics (method of characteristic).

Extracted from the experimental data

One of the parameters using the test results is the values of the flow coefficient from air valve and smoke. These values were conducted using the flow test and by flow bench test that the scheme of the apparatus is shown in Figure 1.

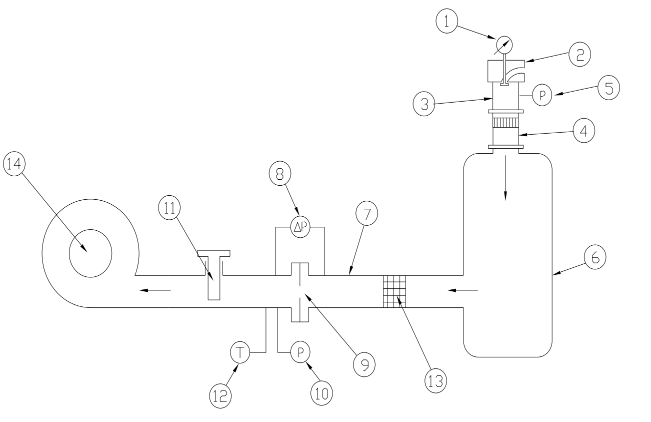

Figure 1: The schema of the table
1 - Time valves measuring openness 2 - closure 3 - Cylindrical Interfaces 4 - Meter turns 5 - 6 Gauge Pressure Tank - Relax 7 - rigid tube 8 - Pressure gauge 9 - Holes 10 - 11 gauge pressure - flow control valve 12 - Temperature gauge 13 - 14 honeycomb paths - blower / suction air

The coefficient of flow table test (flow bench) has been obtained. With this test, the pressure drop is measured between the valve and the flow rate. Values of upstream pressure and temperature, downstream static flow and flow rate are the necessary data for calculation the flow coefficients or drop coefficients. First, using the pressure ratio, the flow isotropic velocity is obtained as equation (1) shows.

$$V_{is} = \sqrt{2\frac{\lambda}{\lambda-1}RT_{\circ}\left[1-\left(\frac{P}{P_{\circ}}\right)^{\frac{\lambda}{\lambda-1}}\right]} \qquad (1)$$

R is the gas constant, λ is P and P_0 specific warmness that respectively is downstream and upstream pressure. The effective surface of valve is also calculated by using equation (2).

$$\frac{C_f}{2} = \frac{4}{Re_{\delta}} \qquad (2)$$

Discharge coefficients versus lift valves inlet and outlet in the figures (2) and (3) are shown.

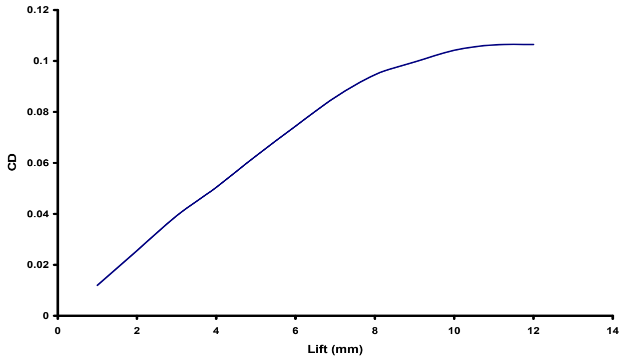

Figure 2. The discharge coefficients in the valve inlet

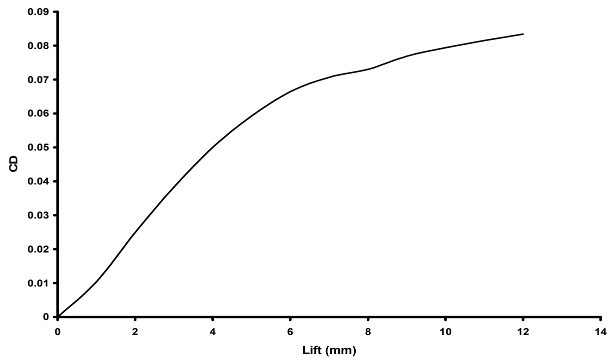

Figure 3.) The discharge coefficients in the valve outlet

Effect of openness valve is timing on volumetric efficiency of the engine's performance.

The parameters may influence directly on the cylinder intake air and combustion products withdrawal and it can control the rate of return flows to the ports. [10]

Flow Friction Model

Fluid and wall friction coefficients were calculated for the accomplished: [11]

A: First, the flow of Reynolds number is calculated $Re = \frac{\rho UD}{\mu}$. That U is an instantaneous velocity of the fluid; D is the pipe diameter, ρ is an instantaneous density and μ is the viscosity of the fluid.

B. The thickness of the boundary layer flow regime is dependent on is calculated from the following relationship:

For turbulent flow $\delta = .10D$

For laminar flow $\delta = .25D$

C: For high Reynolds numbers 4000, C_f friction coefficient is calculated by the following method.

$$\frac{C_f}{2} = .027\, Re_\delta^{-.025} \quad (3)$$

D. C_f friction coefficient for Reynolds numbers below 4000 would be as follows.

$$\frac{C_f}{2} = \frac{4}{Re_\delta} \quad (4)$$

Dynamic viscosity is strongly temperature dependent and significant changes will have to change it. Dynamic viscosity of air as a function of temperature is calculated by the following procedure.

$$\mu = 33 \times 10^{6} \times T^{7} \left[Kg / m / s \right] \quad (5)$$

In this regard, T is the temperature at any moment.

Model of heat transfer between fluid and runner

Heat transfer model is done by Celeborn Analogy Reynolds. Heat transfer and flow friction coefficients are combined during the analogy: [12]

$$h = (C_f / 2) \rho U C_p P_r^{\frac{-2}{3}} \quad (6)$$

which Cp is the specific heat capacity of the gas and Pr is the parental number.

Combustion Modeling

In this model the energy release rate for spark ignition engines is described. The Web function (wiebe function) is used to calculate the energy release rate per crank angle. [11] The function is as follows:

$$W = 1 - \exp\left[- AWI \left(\frac{\Delta\theta}{BDUR} \right)^{(WEXP)} \right] \quad (7)$$

In which, W is the mass fraction of the burned mixture, $\Delta\theta$ is the elapsed time from ignition, BDUR is the elapsed time versus Crank Angle of 10 to 90% of the mass of the burned mixture, WEXP is the Web function power that this amount is usually between (1 to 5) and AWI is the Web coefficient function.

Modeling of heat transfer in the cylinder

The heat transfer rate is usually calculated using Woschni model. [13]

The calculation of the heat transfer coefficient is used by the Woschni function as follows:

$$h_g = .0128 D^{.2} P^{.8} T^{.53} V_c^{.8} C_{enht} \quad (8)$$

In this relationship, D is the cylinder diameter, P is the cylinder pressure at any given crank angle, T is the temperature of the cylinder at any given crank angle and Cent is the calibration coefficient that determined by the Model user.

V_c is a Velocity parameter that is a function of the velocity of the piston and the cylinder pressure without combustion.

$$V_c = C_1 . V_m + C_2 \frac{V_D T_r}{P_r V_2} (P - P_{mot}) \quad (9)$$

In the suction and discharge stage

$$C_1 = 6.18 + 0.417 \left(\frac{V_s}{V_m} \right)$$

In the compression and expansion stage

$$C_1 = 2.28 + 0.308 \left(\frac{V_s}{V_m} \right)$$

In the combustion stage

$$C_2 = 3.24 \times 10^{3}$$

In the other stages

$$C_2 = 0$$

V_m is an average speed of the piston, Imep is the mean effective pressure and V_r, P_r, T_r are the temperature, pressure and volume reference respectively.

CALIBRATION MODEL

After simulating the engine, it is necessary to verify the modeling and adaptation of the actual state of the model, with an adjusted set of parameters applied engineering are based on conjecture has been verified. For this purpose, parameters, volumetric efficiency, maximum cylinder pressure, torque and exhaust temperature were set and modeling results for the load conditions and the speed range 1500 rpm - 6000 rpm with data from the test to warm the engine dynamometer Iran Khodro Engine Research Center has been compared. For this purpose, a dynamometer is used for edicarent AFA-202/10 model specifications are shown in Table 2.

Table 2: Profile of engine dynamometer was used to measure functional parameters

dyno. tType	Max torque	Max speed	Max power	Inertia
	N.m.	Rpm	kW	kg/m^2
AFA-202/10	483	8000	120	0.351

If the geometric features are designed for precise motor and parameters in the model and test values are consistent with each other, so the volumetric efficiency of the model and test results will be consistent with each other too.

Volumetric efficiency

After calibration parameters affecting volumetric efficiency, the volumetric efficiency values obtained from model tests for both CNG and gasoline-powered engine as figures (4) and (5) are shown.

Combustion model of the engine

In the used combustion model, the combustion is modeled by using the Web function that is described the burning rate of the mixture using the thermodynamic calculations. As noted, the burning rate using 3 parameters is determined and control by the user in the model. Maximum cylinder pressure values in different periods for both CNG and gasoline hybrid engine are shown in figures (6) and (7) respectively.

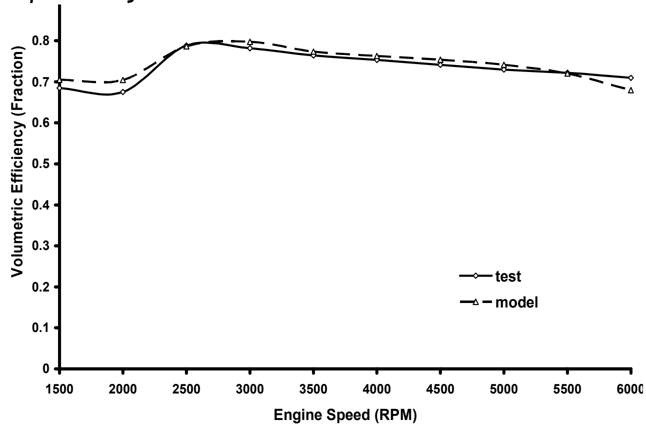

Figure 4. Values of volumetric efficiency in CNG engine in the engine at various speeds

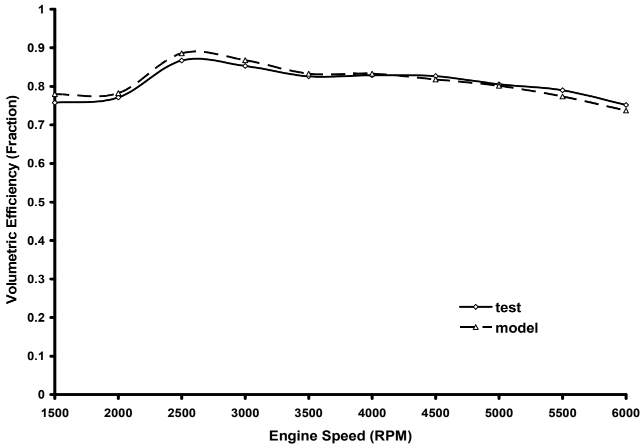

Figure 5. Values of volumetric efficiency in gasoline engine at various speeds

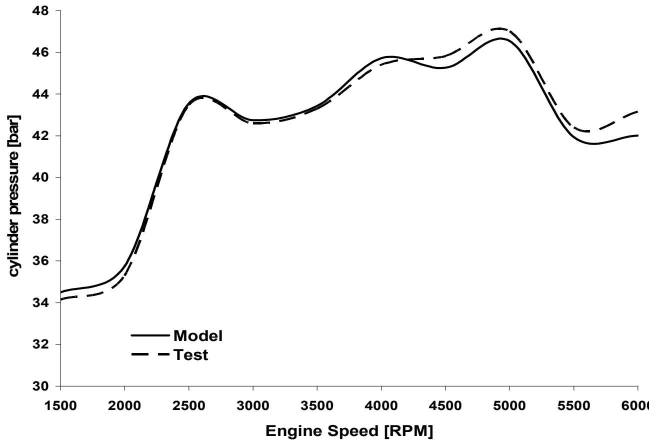

Figure 6. Values for maximum pressure cylinder CNG engine at various speeds

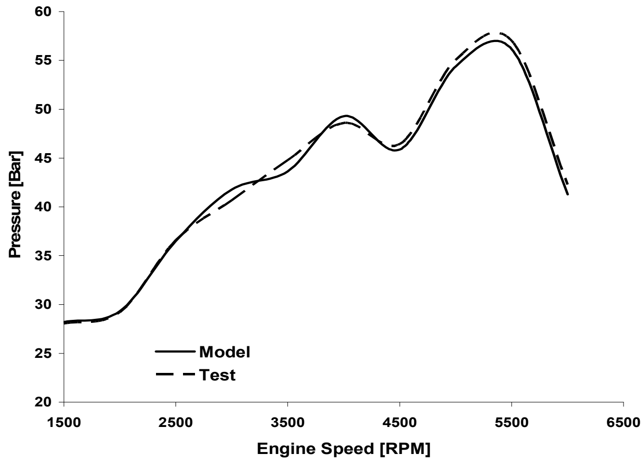

Figure 7. Values for maximum pressure cylinder gasoline engine at various speeds

Heat transfer model in cylinder

Woschni model is used to calculate the heat transfer rate from the cylinder. With the energy release, some of the air-fuel mixture into energy used and the amount of additional heat is wasted. To ensure the accuracy of calculating the heat transfer rate and the strain energy release rate must be calculated at and this was done with great care in the previous section. Therefore, the pressure inside the cylinder to model and test moments, mean effective pressure for the test model must be equal. Figures (8) and (9) for both CNG and gasoline hybrid engine have indicated them.

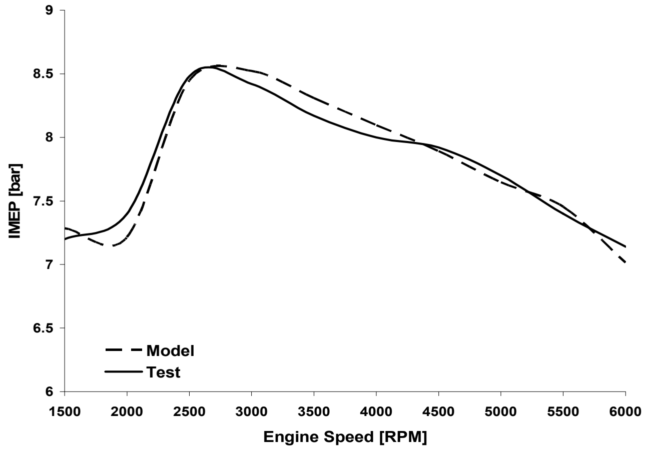

Figure 8. Indicated mean effective pressure for CNG engine at various speeds

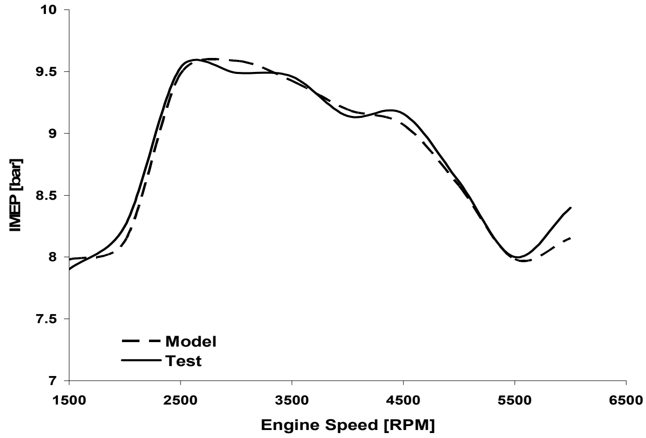

Figure 9. The mean effective pressure for the gasoline engine at various speeds

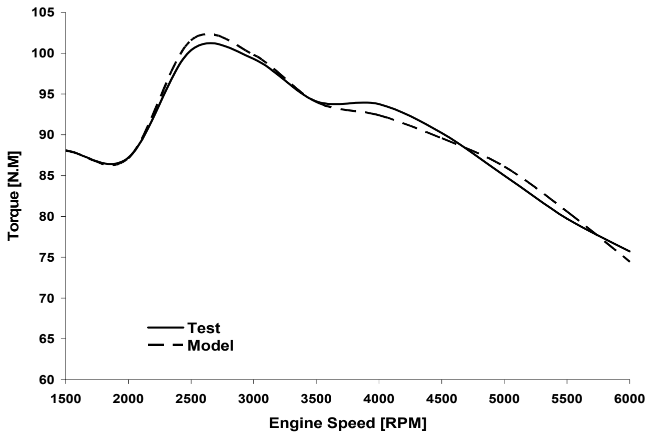

Figure 10. CNG engine brake torque at various speeds

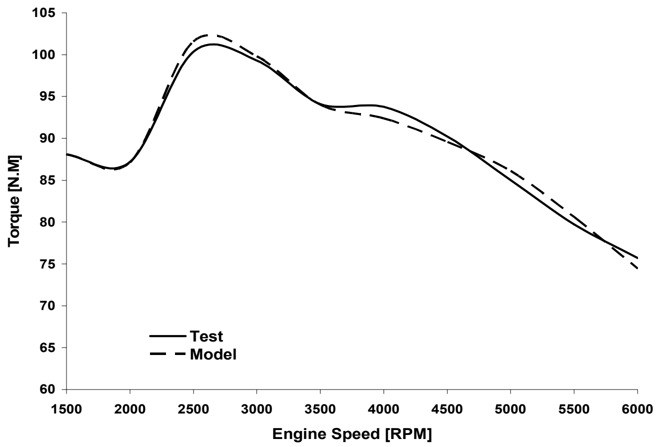

Figure 11. Gasoline engine brake torque at various speeds

After ensuring the same indicatedr mean effective pressure for test and model, if the mechanical efficiency of the test and model are consistent with each other, then be concluded that the heat transfer rate is modeled correctly. Moreover, figures (10) and (11) are shown the brake torque model and test for both CNG and gasoline engines and fuel-burning natural gas.

Select the appropriate turbocharger

After calibrating the exact model, in the figures (4) and (5) is seen to be the volumetric efficiency of the CNG engine is lower that gasoline engine and therefore we should compensate the loss by a technique and also increase the volumetric efficiency as possible and even for this reason, the turbocharger is used to compensate this loss and increasing the volumetric efficiency. Most of overfeeding is done in the engines made and the purpose overfeeding is to increase the power and torque versus speed range. The problem of the overfeeding engine is to achieve the maximum output power, and in combination with maximum strength, improving the low-speed torque is reduced to postpone Turbo Lag [8]. To achieve this goal, careful planning and consideration of turbine engine components are required harmonized. We should change some of the design parameters to converting XU-7 natural breathing engine to an overfeeding engine, of course some parameters such as engine geometry and runner do not changed and should enter the codeand no optimization on any of these parameters. These changes will be discussed. In general, the following points should be noted for the choice of a turbocharged engine:

***A**) The selected turbocharger engine torque for different speeds increased to the desired value.*

***B**) The turbocharger is chosen as small as possible.*

As turbocharger is smaller so its launching is faster, and increasing or decreasing its speed will be faster with high or low load , as a result the delay will be lower in the performance turbocharger (turbo lag). Inertia turbocharger is much smaller than it would be there and a further delay in performance will be resolved.

***C**) much less of a turbocharger to the combustion gases is used to increase the torque, the better. However, the turbine control valve (waste gate) will open more and more gases are discharged to the atmosphere. Or in other words, as the combustion gases flow through the turbine is less, so turbocharger will have a higher response rate.*

***D**) the turbocharger engine must be carefully selected that the selected compressor turbocharger and turbine will work with high efficiency and all functional parts and engine turbine and compressor are working in the area. This points do not reach to the surge line.*

***E**) the maximum turbocharger shall not exceed the maximum speed as the manufacturer made. [15]*

Some tips for choosing the turbocharger to the engine XU-7 has been taken into consideration.

Two turbocharger engines XU-7 is considered to be there and the analysis results will be shown that which of the turbocharger is suitable for the engine and the turbochargers are mady by the Company IHIthat the characteristics of the turbine and its compressor are presented in Table 3.

Table 3: turbine and compressor for both turbochargers

	Turbine	Compressor
turbocharger b	RHF4HTTW14P12	f4c109a
turbocharger a	RHF4HTTW14P12	f4c93f

Some engine design parameters such as timing of inputs and outputs valves, valve lift curve, and ignition timing and etc have been changed in conversion of a normal breathing engine to an overfeeding engine. Thus, the first of these changes and then overfeeding is appropriate.The optimum choice of two turbochargers must be found that would allow the engine to increase torque. The results for both turbocharger models (b) and (a) is shown in figure (12).

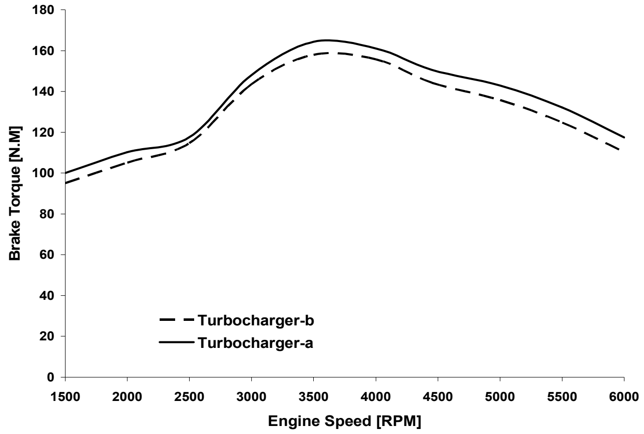

Figure 12. Torque values of the overfeeding engine with two turbochargers at full load state

As the two diagrams show that the turbocharger (a) can be a limit engine torque; although in the diagram (28), the turbocharger (b) can not provide the desired torque.

So the first reason for being better turbocharger (a) of the turbocharger (b) to provide the desired torque. Generally due to the exhaust of combustion energy is low at low speeds thus, selecting the proper turbocharger and engine torque at low speeds, in order to increase the torque is very important; compressor efficiency model has been calculated in both turbochargers (a) and (b).

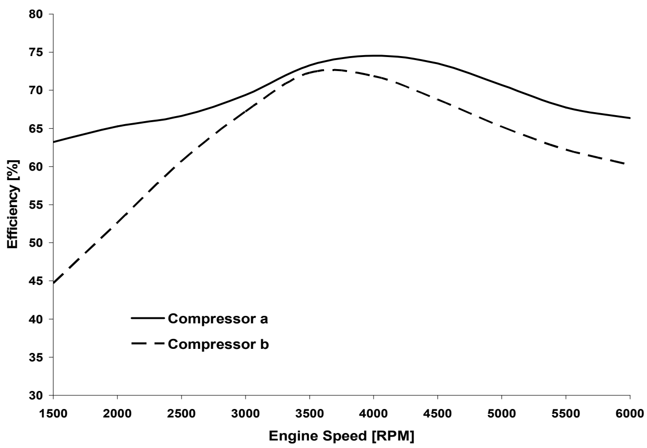

Figure 13. Comparing the compressor efficiency for two turbochargers at various speeds and at full load state

The efficiency for both compressors at different speeds in is shown in figure (13). As the digaram showed the turbocharger compressor efficiency (a) is *much that turbocharger compressor efficiency (b). Turbocharger compressor performance (a) is better than the performance of the turbocharger compressor (b).*

In general it can be said that in all rounds of turbocharger compressor (a) will work with a higher efficiency than turbocharger compressor (b). It can be concluded from this figure that the turbocharger (a) for the engine is betther than turbocharger (b), because the turbocharger will be appropriate in a region with high efficiency operation.

Because turbine for both turbochargers (b) and (a) are the same and also shown in Figure (14), the mass flow through the turbine control valves for both turbines are the same, so it is expected that the work produced by the turbine for different modes is the same in both of them as diagram (14) shows it.

Figure (15) shows two turbochargers (b) and (a) at different engine speeds at full load state. As figure shows the turbocharge round (b) has the maximum turbocharger speed from turbocharge (a). Therfore, dynamic forces and vibrations of the turbocharger (a) is far less than the turbocharger (b) and the turbocharger (a) is better than turbocharge (b).

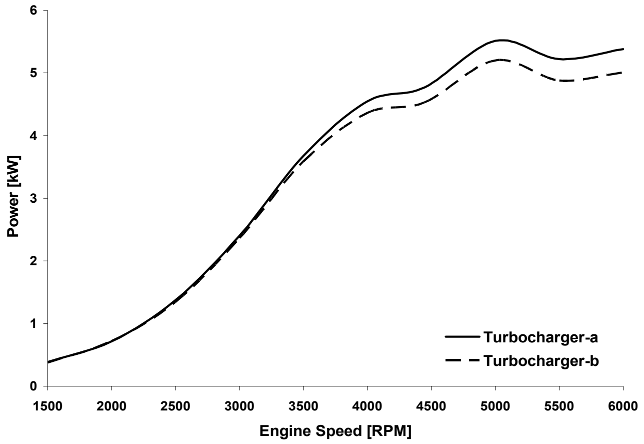

Figure 14. Produced power by both turbocharger turbines at full load state

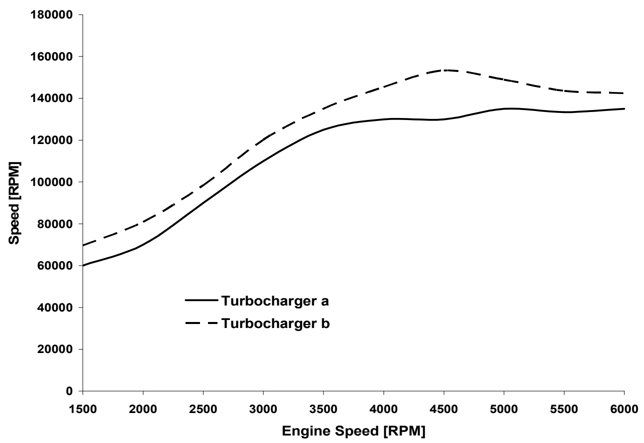

Figure 15. The turbocharge round across two engines at full load state

The reasons for turbocharger (a) is suitable for XU-7 engine. So in modeling , turbocharge (a) is used; the engine volumetric efficiency with normal breathing and forced breathing were compared after engine modeling in gas mode. The results for both turbocharge is shown in figure (16).

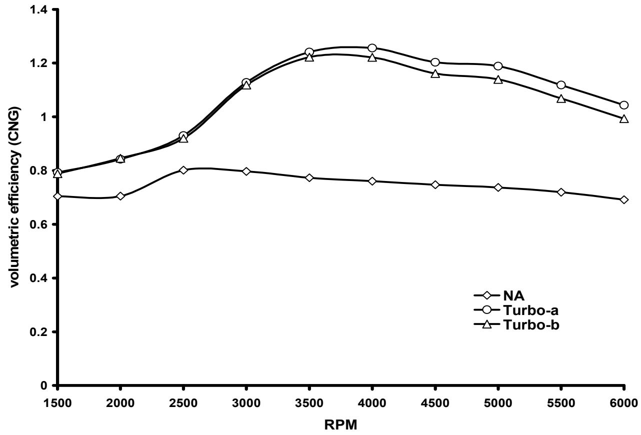

Figure 16. Comparing the volumetric efficiency of the engine to normal breathing and forced breathing engine with two turbochargers

Comparing the results of CNG engine natural breathing and at overfeeding

One of the important tips on choosing a turbocharger engine is that engine performance parts at different rounds lie in the functional area of the turbine and compressor and possibly functional areas are in a region that turbine efficiency and compressor is high in this area. Engine performance parts should not be close to the surge line and if the points of the line are far better. Because turbine efficiency and compressor is very low in the area and if the motor functional areas are close to the surge line or in the surge area, so the engine torque will be loss there; This phenomenon also causes severe fluctuations in volumetric efficiency and engine torque. In this area turbocharger increasingly works unstable and flow through the compressor can greatly vary. This is a very unpleasant and cause severe vibration and turbocharger engine. The functional parts of engine XU-7 on the compressor map is shown in figure (17).

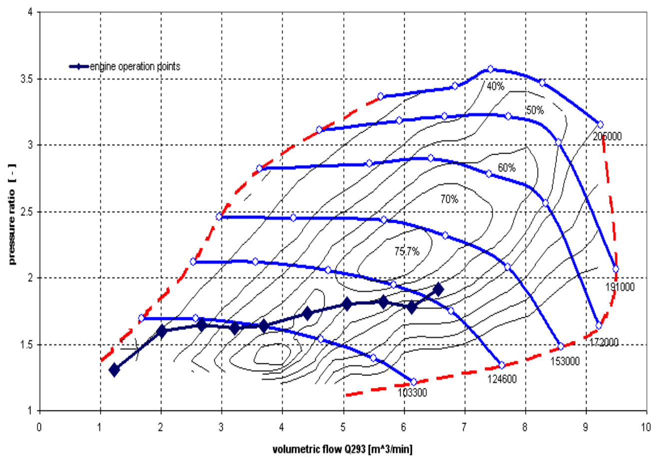

Figure 17. The functional parts of the engine (around 1500 to 6000 rpm) on compressor performance curve

As the figure (17) shows, we close on surge area at around 1500 RPM and one of the reasons why this model is around 1500 RPM is not desirable. Another reason for the lack of proper for feeding engine model at round 1500 RPM is to ignore the effects of oscillating pressure resulting from combustion gases (pulse) and the unstable nature of the flow. According to the functional map of a turbocharger compressor and turbine (map) with the steady flow test that should be done.

The engine speed is greater than the effects of pulse and flow goes to quasi-steady.. Another improper reason model at around 1500 RPM ignores the effects of three-dimensional flow. Because at low speeds due to the inertia of the fluid is lower than the low point, Viscosity forces against the forces of inertia can not be ignored. At low speeds (almost below 2000 RPM) because the combustion gases are down so the engine torque will not increase as desire. Using Overfeeding , inlet air pressure to the engine by the turbocharger compressor will increase; This will increase more air into the cylinder and engine volumetric efficiency. Figure (18) shows the values of volumetric efficiency for XU-7 engine with natural breathing (NA) and overfeeding (TC), as is clear from this diagram, the volumetric efficiency at high speeds is almost more; this is due to the high energy of combustion gases and the high mass flow rate of combustion gases at high speeds.

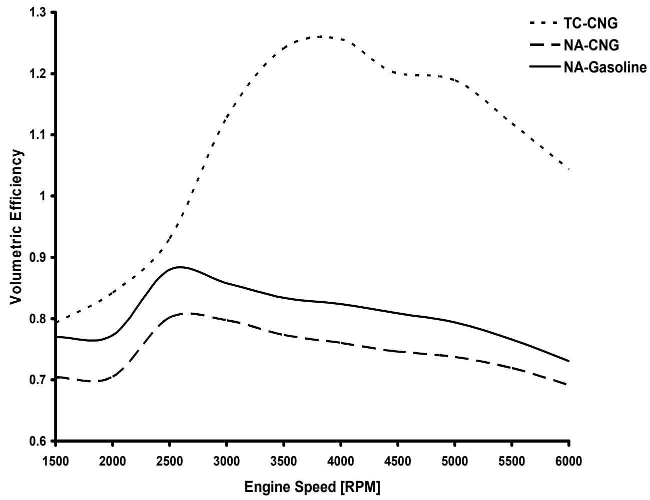

Figure 18. XU-7 engine volumetric efficiency versus engine speed for both normal breathing and overfeeding

In general it can be said that overfeeding engine causes more air-fuel mixture into the cylinder; also this causes more energy to be released and the amount of torque and maximum cylinder pressure is higher than normal breathing engine. Diagrams (19) and (20) show the amount of torque and maximum cylinder pressure at different periods for both normal breathing and overfeeding engines XU-7.

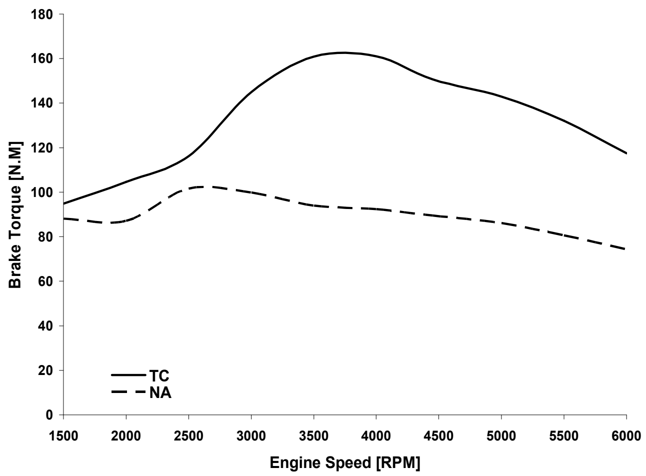

Figure 19. Values of brake cracking versus engine speed for engine XU-7 in normal breathing and overfeeding

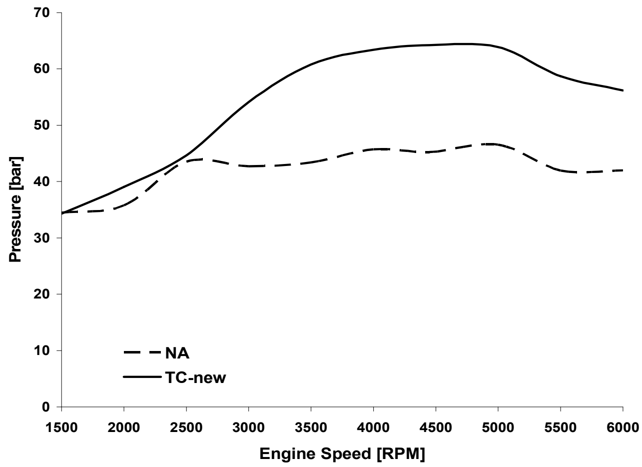

Figure 20. Values of maximum cylinder pressure at different engine speeds for both normal breathing and overfeeding

By increasing the amount of realeased energy and maximum cylinder pressure at feeding effect, the temperature of the combustion gases in the engine TC will be higher than engine NA. Figure (21) shows the temperature of the combustion gases for both NA and TC engines.

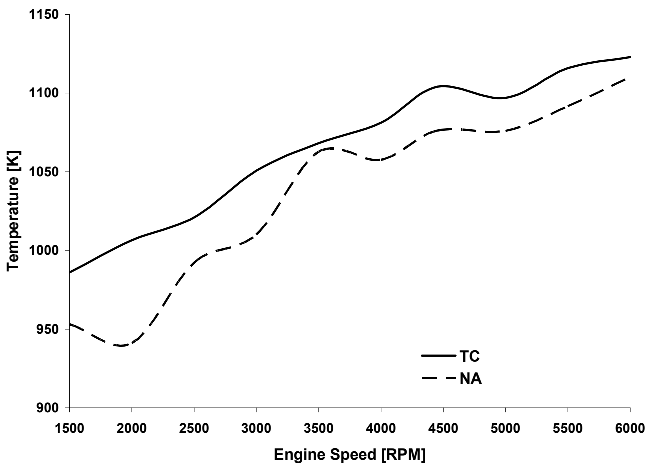

Figure 21. The tempreture of combustion gases in exhaust manifold at engine XU-7 for both natural breathing and overfeeding

So according to the above mentioned, one-dimensional modeling is a useful tool for predecting the engine performance. Using the modeling, all the engine performance parameters can be predicted. It can also effect changes in the design parameters on the performance of the engine. As modeling shows, one of the turbochargers have high engine power for other performance parameters of the engine.

CONCLUSIONS

In the present study, a one-dimensional model was used to model the XU-7 engine.

After Gasoline and CNG engine model, the results of modeling was valid by using the results of engine dynamometer test and functional characteristics were compared between the two engines, and finally given the design parameters of the turbocharger, the engine system was selected for overfeeding. The results were presented in a CNG engine with a significant power reduction compared to gasoline engine. So two systems were modeled on CNG engine for solving this problem.

The results were presented that using the designed turbocharger, the volumetric efficiency, braking torque and pressure inside the combustion chamber for overfeeding CNG engine have high level than natural breathing.

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