ON THE ESTIMATION OF POWER FROM A DIESEL ENGINE CONVERTED FOR GAS OPERATION – A SIMPLE ANALYSIS

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ABSTRACT

In this paper, major factors influencing the power output from a gas engine are utilized to predict the power output from a diesel engine converted to gas operation using various gaseous fuels. The primary factors chosen are the gas calorific value, compression ratio, and molar changes due to combustion; leading to changes in peak pressures, peak temperature, turbo charger pressure and finally the quality of combustion for a given cylinder geometry. Analysis is carried out using empirical relationships available from the literature. Case studies are presented for different fuels to understand these effects on the engine output.

1. INTRODUCTION

Increasing availability of gaseous fuel and the demand to use them for power generation has led to manufacturing of the gas engines. Most of the engines are modified from diesel engines to run on gas by introducing the ignition, gas governing, and carburetion systems along with change in compression ratio and in some cases change in the combustion chamber. One major point that is generally considered related to the performance, is the power delivered by the engine compared with the liquid fuel engines of comparable cylinder capacity. The choice of the above mentioned hardware depends on the gaseous fuel to be used. Each of these system components plays an important role in the overall performance of the engine. It is assumed here, that the effect of ignition time, ignition quality and the mixture ratio control for a given a combustion chamber design, are chosen in such a way that they are the best

2. THE ANALYSIS

Over the last three decades research and development on the engine has been addressing the use of technology for better combustion within a cylinder volume, by improving amongst other aspects, the spray characteristics of fuel and mixing with air. Recent development in the injectors and combustion chamber designs have led to very compact engines with the primary aim to reduce the weight, while improving the overall conversion efficiency. In the bargain, the excess air factors that were in the range of 25 – 30 % have been restricted to about 15 - 20% in most of the engines.

The present designs of gas engines adopt most of the hardware related to diesel engines. Thus, a diesel engine is considered as a benchmark for the comparison of the power output of the gas engine. In the analysis, only 4 stroke engine designs are considered.

Effect of compression ratio on power output and efficiency

The power delivered from an engine is directly proportional to the mean effective pressure (mep) developed in the engine cylinder. The fuel conversion efficiency, i.e., sum of thermal and mechanical efficiency, increases with increase in compression ratio, implying the power output increases with compression ratio under a given set of operating conditions.

From the relationships [1] on engine performance we have:

$$\text{Power} = \frac{\eta_f\ \eta_v N\, V_d\, Q_{h\vartheta}\ \rho_a\ \lambda}{2} \qquad (1)$$

Mean effective pressure

$$= \eta_f\ \eta_v N\, V_d\, Q_{h\vartheta}\ \rho_a\ \lambda \qquad (2)$$

Otto Cycle Efficiency

$$= 1 - \frac{1}{r_c^{\gamma-1}} \qquad (3)$$

where, η_f, η_v, V_d $Q_{h\vartheta}$, ρ_a and λ are the fuel conversion efficiency, volumetric efficiency, displacement volume, lower heating value of the fuel and air density, and fuel to air ratio respectively. *N* is the engine speed (rpm). *rc*, is the compression ratio and g the ratio of specific heat.

From the above expressions (1) – (3), power output is related to the compression ratio through the mean effective pressure and the cycle efficiency. Heywood [1] indicates that only few studies have focused on the effect of compression ratio on engine performance and efficiency over a wide range of compression ratios in the case of a spark ignited engines. The basic limitation in using higher compression ratio for spark-ignited engines arises out of the properties of the fuel used. The range of operating compression ratio found the literature is between 8 and 14. For spark-ignited engines with the compression ratio less than 12, for a unit change of compression ratio, the output changes by about 3 % [1].

In order to establish the effect of compression ratio on the power, data on Cummins gas engines is used from published data [2]. Using the data, the engine power output is estimated for different derating factors; 1 – 3 % in power output per unit change in compression ratio. Figure 1 shows the plot of percentage error for various models at 1, 2 and 3 % power change for unit change in compression ratio. Details related to the error estimation carried out on different engine models are as shown in Appendix 1.

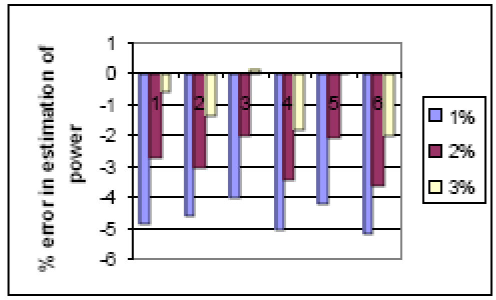

Fig. 1 Plot of % error in estimation of power for different engine model at for 1, 2 and 3 % loss of power per unit change in compression ratio

Error is calculated comparing the estimated power with the rated power indicated by the manufacturer for various compression ratios. For the various engine models, the calculated error is minimum at 3 % change in power for a unit change in compression ratio. The negative values indicate the over estimation of power. Using this information, conversion factors for the power output from commercial gas engines can be established. Based on these estimates, a 3 % change in power occurs for a unit change in compression ratio is chosen for further analysis in the paper.

From the simple analysis a fairly accurate estimation the power output from engines is possible, if the compression ratios of all the engines and the output of one of the engines are known. Thus the scaling law for correcting the compression ratio appears reasonably well established.

Engine efficiency also gets affected due to the change in compression to similar extent [1]. Depending upon the cylinder sizes and the operating conditions, Heywood indicates that, for a unit change in the compression ratio in the range of 9 to 11, the relative change in efficiency is between 1 and 3 percent. Further in a detailed work carried out the by Kerley and Thruston [3], the effective change in efficiency is found to be in the range of 1 – 1.4 % per unit change in compression ratio. It is also found from the work of Sridhar et al [4], that the factor is about 1.3 per unit change in compression ratio with different operating conditions.

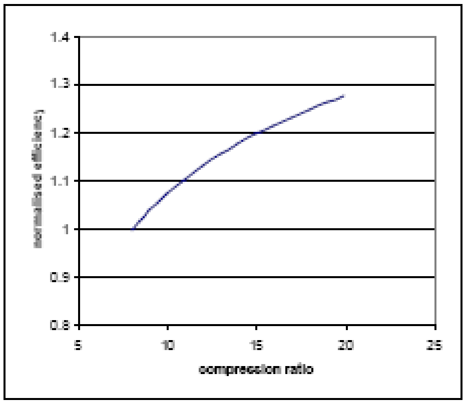

Fig.2. VARIATION OF EFFICIENCY WITH COMPRESSION RATIO USING (EQ. 3) FROM [1]

Fig.2 shows the variation of normalized efficiency with compression ratio using the ideal efficiency relation (eq. 3) for γ = 1.4. Efficiency obtained with compression ratio of 8 is used to normalize the efficiency calculated using other compression ratios. From the experimental analysis Heywood presents the details of an 8-cylinder engine with wide open throttle having similar features as in Figure 2. The change in efficiency is about 3 % per unit change in compression ratio in the range of r_c < 12 and is about 1.8 to 2.4 in the range of compression ratio between 12 and 17. Based on the results available from other sources presented here and the nonavailability of any commercial data in this regard, it is assumed a change of a 1.5 % in efficiency for unit change of compression ratio for the gas engines under consideration.

Effect of fuel calorific value on power

The energy content in the air-fuel mixture has an effect on the output of the engine. Depending upon the calorific value and the stoichiometric air requirement, the energy content in the engine cylinder, which is denoted by the energy density (MJ/m^3), determines the power developed in that cylinder. In order to compare the performance of the engine using different fuel, the energy densities for various fuel is obtained as indicated below.

Diesel

Based on the stoichiometric requirements, the A/F for diesel ($C_{14.4}H_{24.4}$) is 15 and with an excess air
factor of 15 %, the airflow per unit weight of the diesel used is 18. This is about the range of designed air availability for most of the high - speed engines operating at nominal ratings. In the cylinder, the energy density at the nominal rating of the engine would be about 2.83 MJ/m^3 of the mixture (see Appendix 1 for calculations).

Natural gas

At stoichiometry, the A/F requirement for natural gas is about 17. Excess air factor in the range of 5 % for gaseous fuel combustion results in an air requirement of say 18 kg per kg of natural gas. With lower calorific value of 45 MJ/kg the energy density inside the cylinder volume would be about 3.0 MJ/m^3.

Producer gas

Using the similar logic as that of natural gas, evaluating producer gas fuel in engines, we have the following; The calorific value of the gas taken at 5.2 MJ/kg would result in an energy density of 2.4 MJ/m^3. This value is lower than that of diesel and about 75 % the value of natural gas.

Biogas

The calorific value of biogas with 75 % methane and 25 % carbon dioxide is about 23.6 MJ/kg. At stoichiometry, A/F is about 10.5 and with 5 % excess air, the energy density is 2.3 MJ/m^3. This value is lower by about 5 % in comparison to producer gas.

Table 2 shows the energy densities various fuels. It is clear that the energy density of natural gas is higher than other gaseous fuels.

Table 2: Properties of various fuels [1]

Fuel*	Lower heating value MJ/kg	A/F	Energy density MJ/ m^3
Diesel	42.5	18	2.83
Natural Gas	45.0	18	3.00
Producer Gas	5.2	1.2	2.40
Biogas	23.6	11	2.30

* Please see Appendix 1 for composition

Estimation of power from an engine with gaseous fuel

Consider first, the diesel engine being converted to operate on natural gas as the fuel. From the above estimates it is clear that the energy density in a given cylinder volume is higher by about 6 %in the case of natural gas compared to diesel. Assuming same compression ratio and conversion efficiencies for diesel and gas operation, the output from a natural gas engine should be about 6 % higher than of diesel engine arising from energy density. Due to combustion characteristics of natural gas, the compression ratio is reduced to about 12 to eliminate the knocking. With reduction in compression ratio from say from 15 to about 12, there is a 3 points drop, resulting in a maximum reduction in power by about 9 %. Also the efficiency gets affected due to the reduction in compression by about 4.5 %. These lead to following effects,

a. Reduction in efficiency would result, reduction in the energy content per unit volume by about 4.5 %; thus 2.87 MJ/m^3 is available in the engine cylinder for useful work.
b. Further, there is a reduction in power due compression ratio amounting to about 9 %,

Thus the effect of changes in compression ratio on diesel engine converted to operate on natural gas results in reduction of power by about 8 %.

Effect of change in moles between the reactants and products

Apart from the above factors related to the fuel combustion properties, the changes between the moles of products and reactants have to be accounted for, as they affect the peak pressure in the engine cylinder. For example, between diesel and natural gas the reactants and the products have the same number of molecules, indicating no change in the moles, hence no change in the pressure. For producer gas there is a reduction from 3.05 to 2.63 and 12.9 to 11.9 in the case of biogas. The other parameter that could affect the performance is the adiabatic flame temperature. These two additional effects have an influence on the peak pressures inside the engine cylinder.

Table 3: Adiabatic flame temperature and mole change factor for various fuels

Fuel*	Mole change factor	T_{adi} (theory) K	Temperature factor
Diesel	1.0	2290	1
Natural Gas	1.0	2225	1
Producer Gas	0.86	1925	0.87
Biogas	0.91	2160	0.97

Table 3 gives the details of the change in moles between the reactant and the products for various fuels along with the adiabatic flame temperature. Adiabatic flame temperature is calculated using [5]. The change in flame temperature affects the peak pressure inside the engine cylinder. These are obtained using the ideal gas law.

Effects of turbo charging

In a typical naturally aspirated diesel engine the peak pressure inside the engine cylinder is about 88 atm at the rated conditions (Greaves, 1991) [6]. As indicated above, the mole change factor and the peak temperature inside the cylinder could affect the peak pressure. Correcting for these we have a peak pressure that could be achieved as 66 atm (88 * 0.86 * 0.87), which is important while turbo charging. This value of 66 atm is closer with the experimental results (69 atm) at a compression ratio of 17: 1 [4].

In the case of turbo charged engines, the performance is affected by the change in the inlet conditions to the turbine. Since the peak pressure gets affected due to gas operation, the performance of the turbo charged gets linked to this. Further assuming that the inlet pressure to the turbine has a proportional effect on the compressor output, one can allow for the reduction in the pressure ratio of the compressor by the same factors. In the case of turbo charged operation, the inlet super charger pressure is affected by the change in moles between the product and reactant and the temperature change. This has a multiplying factor of 0.75 (resulting from 0.86 x 0.87) for producer gas operation and by a factor 0.88 (resulting from 0.91 x 0.97) for biogas operation on the compressor pressure ratio.

Thus, from the above analysis, power from a gas engine is related to change in compression ratio, the energy density, changes in the moles between product and reactant and temperature. Further, some of these factors have influence on the peak pressures attained in the engine cylinder, which affects the turbo charger performance.

Summarizing the above empirical relationships we have;

Power output = $f\,(\eta_f, P_f, E_f, M_f, T_f)$

where, η_f *and* P_f are the factors due to change in compression ratio affecting efficiency and power output, E_f is the energy density factor resulting from the difference in the energy density of the fuels, M_f, the factor due to change in the mole factor between the reactant and products and finally T_f the temperature effect on pressure due the change in peak tem perature in the engine cylinder.

Case study 1: Chatel St. Denis - Liebherr engine

The 6 cylinder Liebherr engine with 1.66 litres capacity running at 1500 rpm has a total cylinder capacity of 10 lts. The naturally aspirated engine with a compression ratio of 12:1 has been tested in Switzerland, at EPFL (Rothlisberger 1998) [7]. The system has delivered about 96 kWe at nearly stoichiometric condition.

Using the same logic as that of natural gas, let us evaluate the performance of the engine system with producer gas.

Using an energy density of 2.4 MJ/m^3, we have about 80 % energy density compared with that of natural gas; thus one would expect about 80 % output of that of a natural gas engine. Taking into account the moles of products and reactants for both the fuels, we have a same number in the case of methane, while in producer gas there is a reduction from 3.00 to 2.63. The adiabatic flame temperature that could be achieved in the producer gas operation is lower by about 300 K (2225 – 1925). These two additional effects have an influence on the peak pressures inside the engine cylinder.

From the above facts we have the following features to account in the producer gas operation;

a. Reduction of power output to 80 % of natural gas because of energy effects
b. Reduction in the peak pressure by 0.86 due to reduction in the moles
c. Reduction in the peak temperature amount to about 13 % reduction in pressure.

From the above equation for no change in compression ratio, we have a power derating as $f\,(E_f, M_f, T_f)$ as 0.80 * 0.86 * 0.87 = 0.60

Using this in the Liebherr engine, we can expect about 96 * 0.60, say 58 kWe from the engine. Detailed engine testing carried out by coupling the IISc gasifier system to the Liebherr engine (Goirdano, 1998) [8] has resulted in a peak power of 56 kWe. From the gas analysis it is found that the mole fraction H_2 = 0.17, CO 0.19 and CH_4 = 0.022 amounting to a calorific value of 4.7 MJ/kg.

This has an energy density of 2.24, resulting in a factor, E_f, of 0.75 instead of 0.87. The derating factor is 0.56, which amounts to about 54 kWe.

If we use the same scaling laws, we can expect about 0.60 times power from the producer gas engine using a modified diesel engine.

Case study 2: Indian Institute of Science, Bangalore; 25 kVA Kirloskar engine.

Extensive study on a RB 33 model Kirloskar make engine has been made with producer gas. The 3-cylinder engine with a volumetric capacity of 3.3 lts, 1500 rpm has a nominal output of 28 kW. The engine has a compression ratio of 17 :1. In a recent study carried out by Sridhar et al (2001), the engine performance on the producer gas operation is presented. Experiments have been conducted at various compression ratios to obtain the peak power from the engine.

Using similar logic indicated above, the expected output has an effect of energy density at the same compression ratio. The expected output is about 28 x 0.8 = 22.4 kW. Further, we have the mole correction factor and the peak pressure, which will result in 22.4 x 0.86 x 0.87 = 16.8. The peak output obtained from this study [4] is 17.4 kW.

Case study 3: Ugar sugars and KCP sugars - Greaves

Consider the 12 cylinder Greaves engine with a cylinder capacity of 21.6 litres, rate for 444 kW at
1500 rpm and the compression ratio being 15:1. The engine is fitted with a turbo charger and an after cooler. The turbo charger has a pressure ratio of 2.2.

Based on the scaling laws we could expect;

a. Natural gas operation with TA

1. No change in mole fraction between the products and reactants
2. Small difference in the temperature effects
3. The combined compression ratio and calorific value changes 8 % derating in the engine output (as above)

Thus one would expect nearly 408 kW output from natural gas operation, except for the charge intake into the turbo charger.

b. Biogas operation with TA

If we consider biogas as a fuel with a gas composition of 75 % CH_4 and 25 % CO_2, the energy density factor is 0.64, molecules reduce by a factor 0.91 and the temperature effect is about 3 %. Taking into account the change in compression and other gas properties, output power would be

$$f\,(\eta_f, P_f, E_f, M_f, T_f);$$

0.955*0.91*0.81 x 0.91 x 0.97 x 444 = 062 x 444 = 276 kW

Taking note on the turbocharger de-rating due the peak pressure being lower, one can expect about a pressure ratio of 1.94 (0.91 x 0.97) as against 2.2; resulting power of about 240 kW. Any variation in the calorific value would also change the output.

Testing of these engines (Subbukrishna 2000) [9] for has been done stationary application with the engine connected to an alternator at M/s Ugar sugars, Karnataka (4 numbers) and KCP sugars, Andhra Pradesh (4 numbers), using biogas - obtained by treating distillery effluent. The peak output measured at both the locations in the range of 225±5 kW. This may be considered consistent with the predictions from the simple analysis.

c. Producer gas operation with TA

On similar grounds we can estimate the power from the engine operated on producer gas.
Expected output = 0.955 x 0.91 x 0.81 0.74 x 0.86 x 0.87 x 444 = 230 kW

Also the turbocharger de-rating needs to be accounted. On similar arguments indicated above, the pressure ratio could be around 1.76 as against 2.2. Assuming that power would be proportional to the turbocharger derating, we can expect about 0.8 x 230 = 203 kW on this engine. Thus a de-rating of about 55 % compared with the diesel engine.

Table 3 gives the details of the output expect from an engine of 21.6 its capacity with a turbo charger of pressure ratio 2.2. Variations in the output are expected with change in gas composition as it is going to affect the energy density and the mole fraction factor.

Table 4: Output of an engine for various fuels (Engine model: TB232, make Greaves)

Diesel	Natural gas	Bio Gas	Producer Gas
444	408	220	203

Conclusions

A simple procedure to estimate the power from a diesel engine converted to run on gas is brought out. The parameters chosen for the analysis are related to the properties of the fuel and the only other parameter from the engine is the compression ratio. The influence of the fuel properties on the peak temperature and pressure has been used for the analysis. From the results it is clear that the simple analysis using empirical relations seems to predict the power output of a diesel engine modified to operate on gas quite satisfactorily. The effects of these have been discussed and case studies are also presented. The effect of using a turbocharger is also presented.

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Appendix 1

Estimation of energy density for various fuels used in the engine

Density of air 1.2 kg/m^3
Density of diesel is 850 kg/m^3,
Density of natural gas is 0.72 kg/m^3,
Density of producer gas is 1.0 kg/m^3,
Density of biogas 0.82 kg/m3

Chemical composition of the fuels:

Diesel: $C_{14.4}H_{24.4}$
Natural gas: CH_4
Producer gas: CO =0.2, H_2 = 0.2,
CH_4 = 0.025, CO_2 = 0.1
N_2 = 0.4
Biogas: CH_4 = 0.75, CO_2 = 0.25

Total volume in the cylinder = volume occupied by (air + fuel)

Fuel	Lower Calorific Value [1] MJ/kg	Total cylinder Volume m^3	Energy density MJ/ m^3
Diesel	42.5	15.0	2.83
Natural Gas	45.0	16.4	3.00
Producer Gas	5.2	2.0	2.60
Biogas	23.6	10.4	2.2

Tables A1a-c presents the estimated power output and the factor to account for the change is compression ratio (CR factor). Three different engine models have been chosen for this exercise. The rated output in BHP, is the claimed output by the manufacturer under standard conditions. Using the data, the engine power output is estimated for different derating factors; 1 – 3 % in power output per unit change in compression ratio. The estimated power is derived from the fact that the output of an engine changes with compression [1]. Error is calculated comparing the estimated power with the rated power indicated by the manufacturer for various compression ratios. In the remarks column the reference compression ratio is indicated. The negative value of the error indicates the over estimation of the power.

Table A1a: Engine model G-495 -G; 1500 rpm, naturally aspirated

Compression Ratio	Out put BHP	Estimated power BHP			% error in estimating the power out put			Remarks
		1%	2%	3%	1%	2%	3%	
12 : 1	92							
10 :1	86	90.2	88.3	86.5	-4.9	-2.7	-0.6	Output compared with 12:1
8.5 : 1	81	84.7	83.4	82.1	-4.6	-3.0	-1.3	Output compared with 12:1
8.5 : 1	81	88.9	85.6	82.3	-9.8	-5.7	-1.6	Output compared with 12:1

Table A1b: Engine model G-743-G; 1500 rpm, naturally aspirated

Compression Ratio	Out put BHP	Estimated power BHP			% error in estimating the power out put			Remarks
		1%	2%	3%	1%	2%	3%	
12 : 1	137							
10 :1	129	134.3	131.5	128.8	-4.1	-1.9	-0.1	Output compared with 12:1
8.5 : 1	121	127.1	125.1	123.0	-5.0	-3.4	-1.6	Output compared with 12:1
8.5 : 1	121	132.0	127.4	122.6	-9.0	-5.3	-1.3	Output compared with 12:1

Table A1c: Engine model G-1710-G; 1500 rpm, naturally aspirated

Compression Ratio	Out put BHP	Estimated power BHP			% error in estimating the power out put			Remarks
		1%	2%	3%	1%	2%	3%	
12 : 1	333							
10 :1	313	326.3	319.7	313.0	-4.2	-2.1	0.0	Output compared with 12:1
8.5 : 1	293	308.3	303.6	298.9	-5.2	-3.6	-2.0	Output compared with 12:1
8.5 : 1	293	302.0	309.7	298.0	-3.1	-5.7	-1.7	Output compared with 12:1