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RESEARCH

Determination of Optimum Compression Ratio: A Tribological Aspect

L. Yüksek [a], O. Özener [a], H. Kaleli [a]

[a] *Yıldız Technical University, Mechanical Engineering Department, Beşiktaş, İstanbul, Turkey.*

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Corresponding author:

Levent YÜKSEK
Yıldız Technical University,
Mechanical Engineering
Department, Beşiktaş,
İstanbul, Turkey
E-mail: lyuksek@yildiz.edu.tr

A B S T R A C T

Internal combustion engines are the primary energy conversion machines both in industry and transportation. Modern technologies are being implemented to engines to fulfil today's low fuel consumption demand. Friction energy consumed by the rubbing parts of the engines are becoming an important parameter for higher fuel efficiency. Rate of friction loss is primarily affected by sliding speed and the load acting upon rubbing surfaces. Compression ratio is the main parameter that increases the peak cylinder pressure and hence normal load on components.

Aim of this study is to investigate the effect of compression ratio on total friction loss of a diesel engine. A variable compression ratio Diesel engine was operated at four different compression ratios which were "12.96", "15.59", "18.03", "20.17". Brake power and speed was kept constant at predefined value while measuring the in-cylinder pressure. Friction mean effective pressure (FMEP) data were obtained from the in cylinder pressure curves for each compression ratio. Ratio of friction power to indicated power of the engine was increased from 22.83 % to 37.06 % with varying compression ratio from 12.96 to 20.17. Considering the thermal efficiency, FMEP and maximum in-cylinder pressure optimum compression ratio interval of the test engine was determined as 18.8÷19.6.

1. INTRODUCTION

There has been an increasing demand for reducing fuel consumption of the internal combustion engines (ICE). On the other hand, tail-pipe emission regulations force engine developers to dramatically cut toxic pollutants [1]. A proper engine calibration for balancing the fuel consumption and exhaust emission has to be maintained [2]. An optimization is required from the very beginning of the design which comprises the selection of materials, exact determination of main design parameters, proper component selection and computer based modelling process [3-6]. Developments both in spark ignition (SI) and compression ignition engines lead to better fuel economy but the limits of further improvements are drawn by the rules of thermodynamic and the science of material [7,8]. Effective heat of fuel is relatively small, unused energy is transferred to coolant as heat or expelled with the combustion products to atmosphere. Improving the conversion efficiency of fuel by optimizing the engine operation and design parameters is a substantial issue [9,10]. Compression ratio is an important structural parameter which significantly affects thermal efficiency [11-14]. High compression ratios are favourable for SI engines at part load conditions while the cold start improvement is proved for diesel engines [13,15,16]. Literature offers significant amount of studies on compression ratio and effects on engine parameters [17-22]. On the other hand, friction is one of the main contributors of the energy loss in ICE where the heat generated from the rubbing surfaces is transferred to the engine lubricant and coolant [23]. Reduction of engine friction has already have a great potential for reducing fuel consumption where the improvements made on combustion science and thermodynamics have almost been reached to limits [5]. Friction force is a function of normal load and lubrication regime. In order to reduce friction torque of the ICE without modifying the lubrication system, normal load acted on rubbing surfaces has to be decreased. Normal load acting upon bearings, piston and rings is created by the in-cylinder pressure and linearly related with the compression ratio of the engine. In actual engine conditions, the increase of normal load affects the lubrication regime related to Stribeck curve as shown in Fig. 1 [24,25].

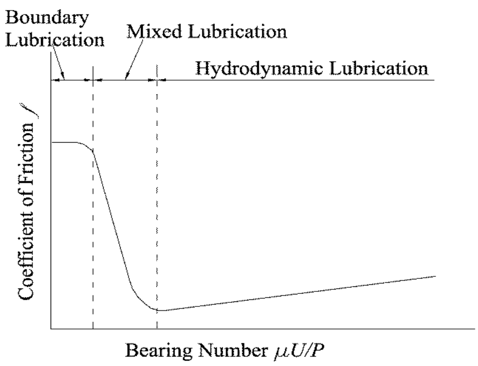


Fig. 1. Definition of various lubrication regimes with Stribeck curve.

The higher the compression ratio the more the energy converted to effective work, both in SI and diesel engines theoretically. However in actual engine conditions it is rather different than theory due to the detonation phenomena in SI engines and the increase of friction and wear in diesel engines. With considering the up-coming EU-6 legislative, in-cylinder peak temperature of modern diesel engines has to be reduced due to lower nitrogen oxides demand [26]. One possible way to obtain this is to decrease structural compression ratio of the engine, which also leads to a reduction of friction torque but possible penalty of thermal efficiency has to be considered. Main aim of this study is to investigate the effect of compression ratio on engine friction power at various compression ratios. Secondary objective of this work is to optimize compression ratio with considering the brake power, friction loss and thermal efficiency. A single cylinder research engine with variable compression ratio system was equipped, engine friction, brake power and specific fuel consumption was measured.

2. EXPERIMENTAL SETUP

The test bed consisted of a research engine and a direct-current dynamometer; the test engine can operate as an SI or compression ignition principles, thanks to its variable compression ratio apparatus. Detailed information about the test bed is listed in Table 1.

Table 1. Technical details of test bed and engine.

Engine Manufacturer	Ferryman 4-stroke CFR engine
Aspiration	Naturally aspirated
Number of cylinders	1
Bore x Stroke (mm)	90x120
Cylinder volume (cm^3)	799
Compression ratio adjustability range	9.5 : 23.5
Speed range min-max (rpm)	600-2000
Number of intake & exhaust valves	1&1
Cooling	Water cooled
Dyno type & power (kW)	DC&7.5

Engine clearance volume can be adjusted by the piston which assembled into cylinder head. Piston movement is driven by a screw type bushing assembly as can be seen in Fig. 2.

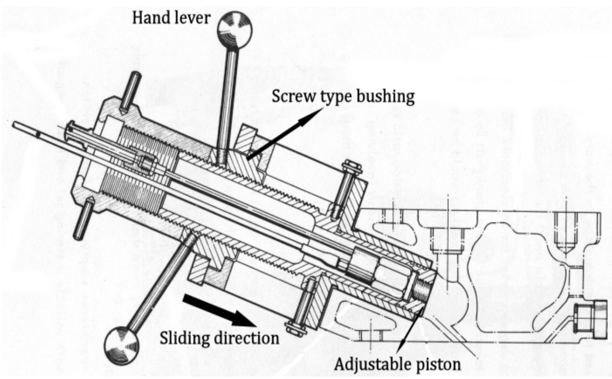
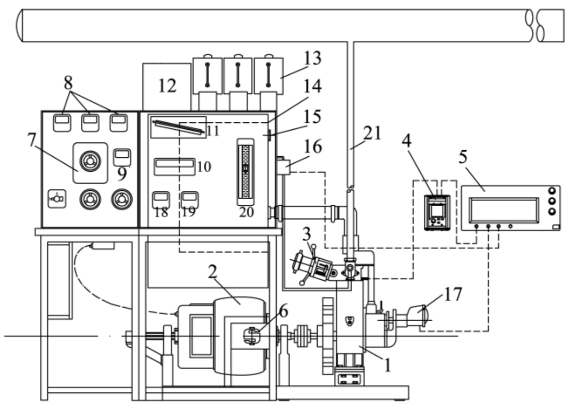


Fig. 2. Variable compression ratio apparatus.

Engine brake torque was measured via a load cell. The fuel consumption was measured with a gear-type flow meter and the engine coolant temperature was measured with K-type thermocouples. An incremental encoder with 0.1 degree crank angle resolution used for monitoring the engine speed and top dead center (TDC) pick up. Kistler 6052 B piezoelectric pressure transducer, Kistler 5011 B charge amplifier and LeCroy Wave surfer 24Xs 4 channel digital oscilloscope were utilized for measurement and acquisition of the in-cylinder pressure data. Schematic of the test system is illustrated in Fig. 3.



1-Internal combustion engine	11-Inclined manometer
2-Dynamometer	12-Cooling Water tank
3-Variable Comp. ratio apparatus	13-Fuel tanks
4-Amplifier	14-Air fluctuation absorption tank
5-ECU & Data logger	15-Air flow rate measurement orifice
6-Load cell	16-Fuel flow meter
7-Loading rheostat	17-Incremental encoder
8-Rheostat voltage indicators	18&19-Water temperature indicators
9-Rheostat current indicator	20-Cooling water flow meter
10-Engine Speed indicator	21-Exhaust line

Fig. 3. Test system.

Engine compression ratio was varied for each test condition. In order to obtain comparable results, engine speed and brake power were kept constant for all compression ratios. A tabulated data of test conditions are listed in Table 2.

Table 2. Test conditions.

Compression Ratio	Brake power (Generated) (kW)	Engine speed (rpm)	Brake mean effective pressure (Bars)
12.96	2.00	1065	3
15.59	2.00	1065	3
18.03	2.00	1065	3
20.17	2.00	1065	3

Friction losses of the engine were determined by using in-cylinder pressure data. Indicated power of the engine was calculated in post-mortem analysis. Indicated work of the system was calculated with using equation 1 where the differentiation of cylinder volume and pressure are inputs. Instantaneous cylinder volume of the engine was determined with incremental encoder.

$$W_{\text{ind}}=\oint p dV \tag{1}$$

Equation 2 was used to calculate indicated power of the test engine.

$$P_{ind}=\frac{W_{ind}N}{{}^{n}R} \tag{2}$$

$$P_{\text{fricion}}=P_{\text{ind}} - P_{\text{brake}} \tag{3}$$

$$FMEP=\frac{P_{friction}\,{}^{n}R}{V_{d}N} \tag{4}$$

The difference between indicated power and the measured power is the friction energy consumed by the rubbing parts. Relative friction performance of the engine can easily be compared with the literature by the means of friction mean effective pressure definition (FMEP) which stated in equation 4. Auxiliary power consumers on the engine are cooling water pump and fuel pump, which are not affected by the compression ratio and hence it was not necessary to make a change.

Steady-state test conditions were facilitated for the measurements. Fifty consecutive cycles were logged and then averaged to minimize the effect of cyclic variations.

3. RESULTS AND DISCUSSIONS

Compression ratio is a geometrical design parameter which affects several parameters including construction dimensions, maximum in-cylinder pressure, thermal efficiency, tail-pipe emissions and effective work. Increasing compression ratio results higher peak in-cylinder pressure and hence higher bulk temperature of the charge. Due to stricter nitrogen oxides legislations, lower temperature peaks are required. Additionally, higher end of compression stroke pressure leads to higher thermal efficiency theoretically. In case of engine tribology, compression ratio induced higher in-cylinder pressure lead to an increase of the ring-liner contact pressure which results higher friction and wear. This phenomenon is one of the main reason that limiting the increase of compression ratio in compression ignition engines.

Figure 4 shows the indicated power, brake power generated and the friction power consumed by the test engine. According to pre-determined test conditions brake power was kept constant and hence the effect of compression ratio itself exhibited clearly.

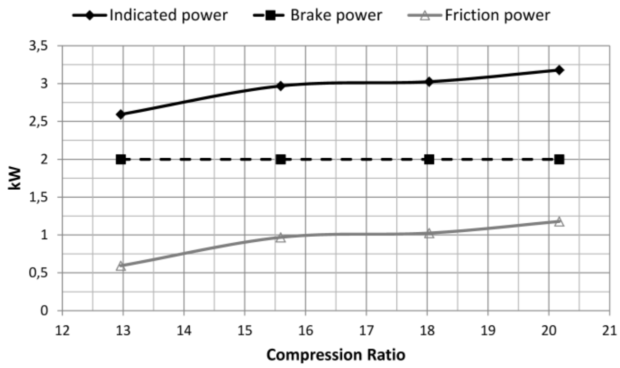


Fig. 4. Variation of brake power with respect to compression ratio.

Almost linear trend was observed both in indicated power and friction power. Ratio of friction power to indicated power of the engine was increased from 22.83 % to 37.06 % with varying compression ratio from 12.96 to 20.17. Additionally thermal efficiency of the test engine was also increased with the increase of compression ratio as illustrated in Fig. 5.

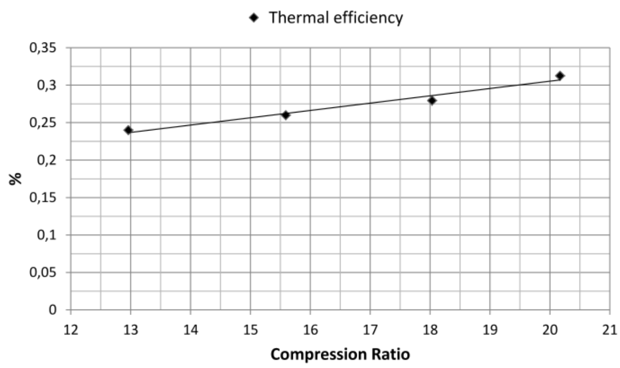


Fig. 5. Variation of engine thermal efficiency with respect to compression ratio.

Considering the Figs. 4 and 5, it can be concluded as the results are in harmony with published literature [11,27]. On the other hand, determining for an optimum compression ratio range with only considering the thermal efficiency would not be so effective. Also, the range has to be investigated from the view of tribological perspective.

Figure 6 shows the in-cylinder pressure curves depending on the test points. In spite of equal brake power output, higher component load is distinct. In terms of durability, ring-pack and journal bearing load which directly affected by the in-cylinder pressure has to be taken into consideration. Friction power curve in figure 4 indicates significant increase when compression ratio varied from 12.96 to 15.59 which can be related with the lubrication regimes of mentioned components. Also, fluid film thickness was not measured.

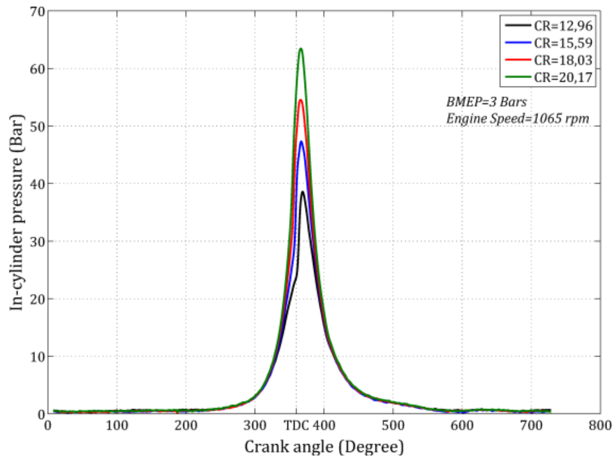


Fig. 6. In-cylinder pressure curves of test points.

In order to evaluate the frictional loss of the engine, normalized FMEP is calculated with using in-cylinder pressure data. Optimum compression ratio determination is a multi-variable investigation which must comprises thermal efficiency and the component load as input data. Normalized thermal efficiency, FMEP and load acting upon tribological systems are concerned together in Fig. 7.

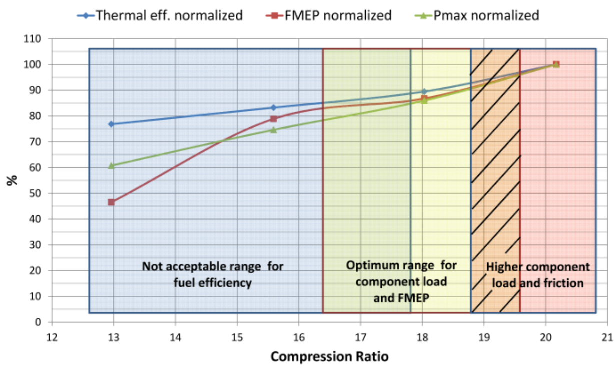


Fig. 7. Parameters acting upon the determination process of optimum compression ratio.

Thermal efficiency is the key factor for determining optimum compression ratio range. Blue and green regions in Fig. 7 representing an interval where the reduction in thermal efficiency is higher than 10 % which cannot be considered as acceptable. Remaining yellow region represents the ideal operation condition for the components while red region indicates the possibility of higher wear. A compromise between high component load and the reduction in thermal efficiency is required, and hence the dashed region in Fig. 7 can be chosen for the range of optimum compression ratio. Primary criteria while selecting this area is the 10 % friction reduction corresponds to 7 % thermal efficiency loss which is a good balance point for the initialization of the range. Further increase of compression ratio leads 5 % component load reduction for 4 % thermal efficiency loss, beyond this point, equal reduction in mentioned quantities are observed and hence further increase of compression ratio is not feasible.

From this point of view, compression ratio of a newly designed engine can be adjusted according to the data obtained from the in-cylinder pressure. This approach can also be used to limit the peak combustion pressure of new generation supercharged engines.

4. CONCLUSIONS

The aim of this study is to investigate an optimum compression ratio by considering the friction loss and thermal efficiency of an engine. A single cylinder research engine with variable compression ratio was equipped for the tests. Engine brake power and speed was kept constant for all compression ratio experiments. According to test results, with increasing the compression ratio, indicated power and friction power of the test engine was increased linearly. On the other hand, with considering the thermal efficiency, FMEP and maximum in-cylinder pressure optimum compression ratio interval of the test engine was determined as 18.8÷19.6.

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NOMENCLATURE

f	coefficient of friction
μ	viscosity of the lubricant (Pa.s)
n_R	number of crank revolutions per stroke
N	engine speed (revolution per minute)
P	journal load (Pa)
P_{ind}	indicated power (kW)
P_{brake}	brake power (kW)
$P_{friction}$	friction power (kW)
U	circumferential velocity of the journal (1/s)
V_d	displaced cylinder volume (m^3)
W_{ind}	indicated work per cycle (kJ)