Theoretical and Experimental Study of Pressure Prediction in Crevices Region between Piston Rings for Diesel Engine

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Abstract

An engine of Zeetor type Z – 6901 of three compression rings is employed in this investigation to accomplish the experimental work. The indicator diagram i.e. P=$f(\alpha)$ is recorded at different engine speeds. The present work elucidates the measurement and indication of gas pressure within the combustion chamber as a function of crank angle. The study contains the prediction of pressure in inter-ring volume pressure behind piston ring during engine operation.

For verification of a mathematical model a static test rig apparatus is designed in order to measure the inter-ring volume pressure within the range of compression pressure i.e. 1bar to 33.8bar. The comparison between mathematical and experimental results show good agreement.

Keywords: Diesel Engine, Piston Rings.

دراسة نظرية وعملية للتنبؤ بالضغط في منطقة الشقوق بين حلقات المكبس لمحرك ديزل

الخلاصة

تم إستخدام محرك نوع زيتور Z-6901 ذو ثلاث حلقات ضغط بأعمار مختلفة لإكمال متطلبات الجزء العملي من البحث و الحصول على مخططات الضغط كدالة لزوايا عمود المرفق بسرع مختلفة. يتضمن الجانب العملي قياس منحني الضغط المبين (indicator diagram) في غرفة الإحتراق كدالة لزاوية عمود المرفق. وكذلك تتضمن الدراسة التنبؤ بالضغوط بين حلقات المكبس و الضغوط خلف الحلقات أثناء أداء المحرك.

تم بناء منظومة متكاملة لقياس الضغوط بين حلقات المكبس لعدة سرع و عند أعمار مختلفة من الأداء و لغرض التحقق من صحة النتائج النظرية تم مقارنتها مع النتائج العملية. بنيت منظومة لقياس الضغوط بين حلقات المكبس لمدى ضغط يتراوح بين P=1 bar الى P=33.8 bar ضمن الضغط المتاح في إسطوانة الضغط والذي يوافق الضغط الحقيقي في شوط الضغط للمحرك أثناء أدائه، و قد تم الحصول على نتائج جيدة تبين أن هناك توافقاً جيداً بين النتائج العملية من الجهاز المستخدم والنتائج التي تم الحصول عليها من النموذج الرياضي.

Nomenclature

A: Flow cross section area perpendicular to flow direction m²
A_g: Area of orifice or area of ring end gap m²
C_1: Clearance between piston and cylinder wall m
C_d: Discharge coefficient
D_h: Hydraulic diameter m
f: Friction coefficient
g: Ring end gap length m
h: Ring side clearance m
$\dot{m}$: Mass flow rate $kg.s^{-1}$
$(\dot{m})_{ij}$: Mass flow rate through the ring gap or ring side clearance $kg.s^{-1}$

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m_{oi}: Initial mass in region i where i = 2,3,4,5, and 6 (evaluated at the initial pressure and temperature) kg

N: Engine speed rpm P_i: Pressure of gases in regions (i = 1,2,3,4,5,6,7) $N.m^{-2}$

P_j: Pressure in outlet of channel $N.m^{-2}$

P_{oi}: Initial pressure in region i where i = 2,3,4,5, and 6 (initial pressure in region i at the end of the intake stroke). $N.m^{-2}$

R: Specific gas constant $J.kg^{-1}.K^{-1}$

R_e: Reynolds number

T: Ring temperature K

z: Length of channel m

γ: specific heat ratio

μ_g: Dynamic viscosity of gases $kg.m^{-1}.s^{-1}$

η_G: Compressibility factor

Introduction

In internal combustion engine, piston rings play an important role in the performance and endurance of the engine, i.e., sealing combustion gas, controlling the lubricating oil as well as working as the passage of heat flow from the piston into the cylinder. It is required, at the same times, which piston rings have minimum friction and wear while rubbing the cylinder wall. Under a “desirable operation”, the purposes of rings are attained by an ingenious motion beyond all imagination. Nevertheless, if an abnormal phenomenon of rings occurs due to some reason, it caused that decrease in performance and endurance of the engine [1].

The following is a summary of the most significant published work involved in the important function of piston rings in internal combustion engines: H. Lindgren [2], has derived an approximated formula for calculation the pressure distribution in the contact between a hydraulic cylinder piston and scraper ring. V. Wong and A. Brown [3], studied the mechanisms of oil consumption. This involves evaporation from the lubricating system and leakage various seals piston rings. P. Reipert and M. Voigt [4], studied the dynamic behavior of piston ring and cylinder of large bore diesel engine. Sutaria B.M. and Bhatt D.V. [5], studied a basic tribological parameters that influences performance of an internal combustion engine. Richard Mittler and Albin Mierbach [6], presented a physical description of the real ring and gas forces, as well as the effects of the moments during the engine cycle.

Now, in the present work the dynamic behavior of piston ring during engine operation of a diesel engine is presented. This can be achieved by employing the following method:

1. The dynamic method for calculation and prediction of pressure in the inter-ring volume and crevices region of piston.
2. The static test system is constructed to measure the value of pressure in the inter-ring crevices volume in order to verify the experimental and theoretical results.

Theoretical Analysis

The engine combustion chamber is connected to several small volumes usually called crevices because of their narrow entrances. Gas flows in to and out of these volumes during the engine operation cycle as the cylinder pressure changes. The largest crevices are the volumes between the piston,

piston rings and cylinder wall. Some gas flows out of these regions in to the crankcase; it is called blow-by.

The volumes between the piston, piston rings, and cylinder wall are shown schematically in Fig. (1). These crevices consist of a series volumes (numbered 1 to 7) connected by flow restriction such as the ringside clearance and ring gap. The geometry changes as each rings moves up and down in its ring groove, sealing either at the top or bottom ring surface. The gas flow, pressure distribution, and ring motion are therefore coupled [11, 12].

Gas flow passages

Fig. (1) Show a schematic diagram of a piston with three compression rings, which represents the current diesel engine. Numbers 1 to 7 identify the different crevice region. Region 7 is located just above the oil ring and is assumed to be at crankcase pressure. Since each ring is split, any two neighboring regions are always connected by the ring end gap. Depending on the position of the ring in the groove, neighboring regions may also be connected by ring side clearance. For example, consider region 1 (top land crevice) and region 2 (crevice behind the top ring) during the compression stroke, the top ring sits on the lower surface of the groove. Therefore, there is a flow over the ring from 1 to 2; also, region 2 and 3 are connected only through the gap. However, as the direction of the force acting on the ring changes the ring lifts from the lower surface of the groove. Region 2 and 3 as well as region 1 and 2 are then connected by the side clearance until the ring approaches the other side of the groove and blocks that passage.

Gas flow equations

To model the flow, we follow the flow from the combustion chamber plane to the crankcase (plane f). The flow in the piston top and crevice (region 1) is a fully developed laminar flow in a channel [14]. Hence, the pressure in region 1 can be assumed to be uniform and equal to cylinder pressure. Regions 2,3,4,5 and 6 can be assumed to have a uniform pressure.

The flow of gases behind and between rings of 2,3,4,5 and 6 is isothermal. This assumption is justified by our estimate of the characteristic time it takes for the gas to reach the wall temperature. With the above assumption, the continuity equations for the regions 2,3,4,5 and 6 can be written [10]:

$$\left(\frac{m_{o2}}{P_{o2}}\right)*\left(\frac{dP_2}{dt}\right)=\dot{m}_{12}-\dot{m}_{23} \quad \text{............(1)}$$

$$\left(\frac{m_{o3}}{P_{o3}}\right)*\left(\frac{dP_3}{dt}\right)=\dot{m}_{13}+\dot{m}_{23}-\dot{m}_{34}-\dot{m}_{35} \quad \text{........(2)}$$

$$\left(\frac{m_{o4}}{P_{o4}}\right)*\left(\frac{dP_4}{dt}\right)=\dot{m}_{34}-\dot{m}_{45} \quad \text{.......(3)}$$

$$\left(\frac{m_{o5}}{P_{o5}}\right)*\left(\frac{dP_5}{dt}\right)=\dot{m}_{35}+\dot{m}_{45}-\dot{m}_{56}-\dot{m}_{57} \quad \text{..........(4)}$$

$$\left(\frac{m_{o6}}{P_{o6}}\right)*\left(\frac{dP_6}{dt}\right)=\dot{m}_{56}-\dot{m}_{67} \quad \text{................(5)}$$

Own to the great flow cross section the pressures in volume 1, Figure (1) are practically identical to combustion camber pressure [12].

Mass flow rates of gases through cross section (a) can be described by the following equation [10]:

$$\dot{m}_a=\left(\frac{m_{o1}}{P_{o1}}\right)*\left(\frac{dP_1}{dt}\right)+\dot{m}_b \quad \text{....... (6)}$$

The change in mass filling of individual inter-ring volumes is dependent on time see Figure (1) and

$$\dot{m}_b=\dot{m}_{12}+\dot{m}_{13} \quad \text{..................... (7)}$$

$$\dot{m}_c=\dot{m}_{13}+\dot{m}_{23} \quad \text{............... (8)}$$

$$\dot{m}_d=\dot{m}_{34}+\dot{m}_{35} \quad \text{............... (9)}$$

$$\dot{m}_e=\dot{m}_{35}+\dot{m}_{45} \quad \text{............ . (10)}$$

$$\dot{m}_f=\dot{m}_{57}+\dot{m}_{67} \quad \text{............. (11)}$$

can be described by the following differential equations:

As has been mentioned above, it is assumed that the flow in region, behind and between rings is isothermal [9 ,10 ,and 15]. Due to isothermal assumption the crevice gas temperature is the same as the crevice wall temperature (piston, rings, and cylinder wall temperatures are assumed equal).

Mass flow rates through the ring-side clearance

A mass flow rates through the ring-side clearance is determined treating the flow as an isothermal compressible flow through a narrow channel of width (z) and (h). Mass flow rate through a narrow channel of length pressure upstream and downstream of P_i and P_j are described by the following equation [14]:

$$\dot{m} = A * \sqrt{\frac{P_i^2 - P_j^2}{R \cdot T \cdot \left(\frac{4 \cdot f \cdot z}{D_h} + 2\ ln\frac{P_i}{P_j}\right)}} \quad \text{.........}\ (12)$$

The flow through the ring side-clearance is laminar, and the friction coefficient can be expressed by the relation [8]

$$f = \frac{16}{R_e} \qquad R_e = \frac{\rho.V.D_h}{\mu_g} \quad \text{.......}\ (13)$$

If we substitute equation (Reynolds number) into equation (friction coefficient) we have:

$$f = \frac{16.\mu_g}{\rho.V.D_h} \quad \text{..............}\ (14)$$

Generally the expression $2\ ln\frac{P_i}{P_j} \ll \left(\frac{4 \cdot f \cdot z}{D_h}\right)$ thus equation (12) can be expressed as following equation:

$$\dot{m} = A * \sqrt{\frac{P_i^2 - P_j^2}{R \cdot T \cdot \left(\frac{4 \cdot f \cdot z}{D_h}\right)}} \quad \text{............}\ (15)$$

Now, if we substitute equation (13) in to (15) and rearrangement we have:

$$\dot{m} = \frac{(D_h)^2 \cdot A \cdot (P_i^2 - P_j^2)}{64 \cdot R \cdot T \cdot z \cdot \mu_g} \quad \text{............}\ (16)$$

The dynamic viscosity () of flowing gas can be calculated from the following equation [16]:

$$\mu_g = 3.3 * 10^{-7} * T^{0.7} \quad \text{.......}\ (17)$$

Mass flow rates through the ring gap

Mass flow rates through the ring gap are calculated by the orifice flow equation [7, 8]:

$$\dot{m} = C_d \cdot A_g \cdot P_i \cdot \sqrt{\frac{\gamma}{R \cdot T}} \cdot \eta_c \quad \text{..........}\ (18)$$

The gas leakage area consists of the ring gap clearance, the cylinder wall clearance and groove surface clearance. In Fig. (2), the gap area may be determined by ($C_1 . g$) where there is no ring chamfering, and where (g) represents the gap length.

From a number of calculations and experimental work [1] (chamfering case) we observed that the value equivalent flow area might be taken twice as large as ($C_1 . g$) in case of normal operation normally designed engine. Thus the end gap area takes the following empirical equation.

$$A_g = 2 \cdot c_1 \cdot g \quad \text{..........}\ (19)$$

Piston-cylinder crevice flow model

The engine Z-6901 4-cylinder type is used for experimental measurement. Fig. (1) show the passage area between the crevices regions of cylinder liner, piston and piston rings. The flow model is coupled with ring motion model. The solution of model is based on the knowledge of pressure cylinder in the working volume.

Piston in this engine has three compression rings and one oil ring. The flow through the oil ring can be neglected in this study. Table (1) illustrates the three cases wear of piston ring used in a solution of the model.

Experimental Work

Indication of pressure-curve (indicator diagram)

The indication of pressure curve as a function of crank angle are carried out on engine Zeetor Z-6901 for three piston group element for Case I, II and III at engine speed N=1200,1800 and 2200 rpm. in the laboratory test bench in University of Technology-Baghdad.

Layout of static measurement apparatus

For checking the validity of the results of the theoretical part for three cases of piston group, static measurement apparatus is designed and constructed in order to measure the pressure in the crevices region 1, 3 and 5 between piston rings.

Static apparatus

Fig. (6) Shows a schematic diagram of static measurement apparatus which is used to perform measurement, Fig. (4) and Fig. (5) show in details the connection of pressure gauges.

The main parts of static apparatus are

1- Cylinder of gas: that is filled by the Nitrogen gas up to pressure of 60 bar (6 MPa) and its connected hose (9) to the combustion chamber (7). The pressure gauge (1) indicates the pressure of the gas in the combustion chamber.

2- In order to maintain cylinder pressure uniform during measurements, pressure gauge (2) is connected to the cylinder head in order to measure the cylinder pressure as close as to the actual value during engine operation.

3- Pressure gauge (3) and (4) are connected to the cylinder at opposite side in order to measure the pressure inside the crevices region between the first, second and the third piston rings.

4- The upper plate (16) and lower plate (17) are sealed by rubber O-ring (15) in order to prevent leakage of gas. These plates are connected by four screws (13).

5- The static apparatus is supplied by safety valve (14): which is constructed at the lower plate and in order to allow the passage of gas from the combustion chamber through a piston rings gap and then to atmosphere.

Measurement of pressure

The static apparatus is used to measure the pressure inside the crevices region volume 3 and 5. The procedure of measurement is carried out as in the following steps:

1- Turn the control valve (6) in order to allow the passing of the gas from the cylinder gas to the combustion chamber until the pressure gauge reach 10 bar, the flow of gas continues to leaks through the ring gaps, after stabilization of reading of pressure gauge, the pressure in volume 3 and 5 were measured respectively. Then the increments of pressure were increased until 5 bar until attain a pressure of 35 bar.

2- In order to check the correctness or validity of measurement in point 1, the measurement is repeated from maximum valve to lower value with 5 bar.
3- The same procedure was repeated for piston group II and III.

Table (2) show the results of pressure measurements in three cases for three piston groups.

Construction of static measurements apparatus and assumptions

As mentioned above, the static apparatus is constructed and designed in order to perform the measurements of pressure inside the cylinder volume and through the region between the piston rings. The following steps are considered during construction of static apparatus:

1- The system is constructed as shown in Fig. (6)
2- The first circular hole is done by drilling holes through the cylinder liner with d = 2.5 mm, L= 6mm and the pressure gauge (3) is connected or tied to its region volume between 1^{st} and 2^{nd} piston rings.
3- The second circular hole is drilled at the opposite side of the cylinder at the region between the 2^{nd} and 3^{rd} piston rings in order to simplify the connection of pressure gauge (4) and also reading the pressure easily.
4- The distance between two center holes are approximately 7.5mm, which is equal to the actual distance between the two piston rings.
5- Hole in upper sealed plate is used to tied or connect the pressure gauge (1) to read the uniform pressure inside the combustion chamber.
6- In the apparatus four screws are used to tight the upper and lower plates of system that are sealed by a circular O-ring rubber type.
7- Safety valve (14) is designed and fabricated at the lower plate in order to allow the passage for gas leaking through the piston ring to the atmosphere.
8- Adjusting screw (13) will control the adjustment of the level of the piston movement inside cylinder.

During measurement process the following notes are considered:

1- The range of pressure during measurement is from 1 to 33.8 bars, which is approximately close to the range of pressure during compression in a diesel engine Z-6901 i.e. close to the point of ignition.
2- Neglecting the dynamic behavior of gases during static measurement as compared in the engine actually.
3- Assume that the instantaneous value of pressure measurement at any time during measurement process is close to dynamic value of pressure in actual engine during compression stroke.

Results and Discussion

The results obtained from theoretical and experimental work are discussed bellow. The computation results of experimental and theoretical part are carried out on a computer using MATLAB Programs.

1- Results of experimental work

1.1 Indication of pressure – curve

As mentioned in experimental work the results measurement i.e. indication of pressure curve as a function of crank angle in the working volume of the measured cylinder during engine operation under full load performance are shown in Figs. (7, 8, and 9) for the three cases of piston group i.e. I, II, and III at engine speeds of N =1200, N =1800, N =2200 rpm.

It is clear that the pressure curve inside the cylinder for new engine case I is higher than that in case II and III due to different wear of piston group elements in each case.

These figures also show the effect of engine speed on the course of pressure in the combustion chamber. It is clear that the pressure value is higher in case of higher speed, and this is due to insufficient time of leakage of mass from combustion chamber. It is also clear that while the piston groups wear increases the pressure-curve level decrease, as in case II and III.

1.2 Results of pressure in the inter-ring volume by using static-test apparatus

In order to check the validity and accuracy of the results of mathematical model, the measured results of pressure P_i in the inter-ring volumes 1, 3 and 5 are carried out as shown in Fig. (19) to Fig. (21) and Table (2) for the three cases of piston group with in the range of pressure during compression stroke only.

The results of experimental test rig as it is shown in Fig. (19) to Fig. (21) indicates a good agreement with the predicted theoretical pressure. It is clear that at a constant engine speed the difference between the theoretical and experimental results are increased with increasing the wear of the piston group elements. This is due to higher leakage of gases through the piston ring end gap. These differences are decreased with increasing engine speed due to less time for flow of gases or blow-by from combustion chamber. It is clear also that the wear of piston group is important parameters which are effecting on sealing and performance of the rings inside its groove.

2- Results of computer simulation

The results of pressure changes in inter-ring volumes as a function of crank angle for three cases I, II, and III at engine speeds of N =1200, 1800 and 2200 rpm are shown in Fig. (10) to Fig. (18)

From these results it is shown that the pressure in crevice region 1 is equal to pressure of combustion chamber. They also show that any two adjacent regions connected by ring side clearance have the same pressures (i.e. P_1=P_2, P_3=P_4, P_5=P_6). These results indicate that the theoretical model is working reliably. The following conclusions which depend on the predicted pressure can be written:

1- In all cases P_3 in volume 3 reaches its maximum value after top dead center. In the course of further revolution of crankshaft, pressure P_3 decreases while the pressure in the combustion volume P_1 decreases in a high rate. At the instant when P_3 becomes larger than P_1, the reverse flow occurs from volume 3 into the combustion chamber.

2- The maximum pressure value P_5 is attained later i.e. at 180° after

TDC due to continuous flow of gases from region 3 to region 5 for all cases.

3- The effect of engine speed on the value of pressure in region 3 and 5 are shown in Fig. (10) to Fig. (18) for three cases , it is clear that when engine speed decreases the value of pressure P_3 and P_5 are increased due to enough time of gases leakage from combustion chamber to region 3, 5 and 7.

4- The effect of piston groups wear (i.e. sealing) are shown also at the same above mentioned figures, obviously when the wear of piston group increases the value of pressure in region 3 and 5 are increased as well, and the maximum value reaches about 1.75 MPa i.e. at 25% of cylinder pressure for piston group III.

Conclusions

In the present work an experimental static test rig for measuring pressure in the inter-ring volume of piston ring are designed and constructed.

The main conclusions which can be drawn from the results of the present work are:

1- An experimental static test rig are designed and constructed for measuring of pressure in crevices region between piston rings for diesel engine. It can be concluded that the results of measuring pressure from static test rig has a good agreement with that results obtained from the theoretical model within the range of a compression pressure.

2- In order to evaluate the effect of individual wear of piston ring and values of crevices volumes between the piston and cylinder wall, it has been a mathematical model for calculation of the pressure in the inter-ring pressure.

3- The applicability of the worked out model is verified on a Z-6901 engine. However, the model provides for the evaluation of piston group structures as early as in the design phase. From the viewpoint of wear and leakage increase it allows prognostic evaluation of these effects and in case the engine is operating.

4- The computation results achieved with the aid of the above model showed the importance of volume minimization above the first piston ring and between the compression rings. Another important role is played, the wear of the first piston ring, which due to deterioration or loss of its sealing capacity, contributes to a significant increase of the amount of gas leakage from the engine cylinder volume.

References

[1]. S. Furuhama and M. Hiruma, "***Piston Ring Motion and its Influence on Engine Tribology***", SAE paper 790860m 1979.

[2]. H. Lindgren, "***Approximate calculations of the pressure distribution in a scraper ring contact***", Wear, 1989.

[3]. Laurance, V. Wong and A. Brown, "***Effect of lubrication system parameters on diesel particulate emissions characteristics***", SAE paper 960318, 1996.

[4]. P. Reipert and M. Voigt, "***Simulation of Piston / Cylinder Behavior of Large Bore Engines***", www.Engine web.com/ media/

Fachvortrag/ 6/ MB68197C. Pdf, 2000.

[5]. Sutaria B. M., Bhatt D. V. and Mistry K. N. "***Simulation and modeling of friction force and oil film thickness in piston ring-cylinder liner assembly of an I. C. engine***" proceedings of the world congress on Engineering Vol II, London U. K. 2009.

[6]. Richard Mittler and Albin Mierbach "***Understanding the fundamentals of piston ring axial motion and twist and the effects on blow-by***" ASME 76080, 2009.

[7]. L. L. Ting, J.E. Mayer, "***Piston Ring lubrication and Cylinder Bore Wear Analysis, Part I, Theory***", Transactions of the ASME, July 1974.

[8]. L. L. Ting, J.E. Mayer, "***Piston Ring lubrication and Cylinder Bore Wear Analysis, Part II, Theory Verification***", Transactions of the ASME, April 1974.

[9]. G. R. Bishop and A. H. Leavitt, "***Performance Simulation of a Diesel Engine and Ring System***", SAE paper 750768, 1975.

[10]. M. Namazian and J. B. Heywood, "***Flow in the Piston-Cylinder-Ring Crevices of a Spark-Ignition Engine: Effect on Hydrocarbon Emissions, Efficiency and Power***", SAE paper 820088, 1982.

[11]. W. W. Pulkarbed, "***Engineering Fundamental of the Internal Combustion Engine***", New Jersey 1997.

[12]. J. B. Heywood, "***Internal Combustion Engine Fundamentals***", New York, 1988.

[13]. A. Kolchin and V.Demidov, "***Design of Automotive Engine***", MOSCK Ba 1984.

[14]. A. H. Shapiro,"***The Dynamics and Thermdynamics of Compressible Fluid Flow***", New York 1953.

[15]. M. Namazian, S. Hansen, "***Schieren Visualization of the Flow and Density Fields on the Cylinder of a Spark-ignition Engine***", SAE paper 800044, 1980.

[16]. Mansouri, S. H. And Heywood, "***Correlation's for the Viscosity and Prandtl Number of Hydrocarbon-air Combustion Products***", Combustion Science and Technology, vol. 23, pp. 251-256, 1980.

[17]. J. Ellermann, M. D. Rohrle and H. Schelling, "***Oil consumption and blow-by of truck diesel engines-test bench results***", SAE paper 810937, 1981.

[18]. V. N. Lukanin, "***Internal Combustion Engine***", Moscow, 1990.

[19]. Liang Liu "***Development and applications of an analytical tool for piston ring design***" SAE paper 01-3112, 2003.

[20]. Won Young Yang "***Applied Numerical method using MATLAB***" 2005.

[21]. Dan Richrdson "***Understanding the fundamentals of piston ring axial motion and twist and the effects on blow- by***" ASME, 2009.

Table (1) Summary chart of the three wear of piston rings

Case	I	II	III
1st ring gap length (mm)	0.2	0.28	0.36
2nd ring gap length (mm)	0.2	0.28	0.36
3rd ring gap length (mm)	0.2	0.28	0.36

Case I: It represents a new piston ring (standard).
Case II: It represents a halve wear which occurs in piston rings.
Case III: It represents a full wear case occurs in piston rings.

Table (2) show the results of pressure measurements in three cases for three piston groups.

Crank angle α (deg.)	Case I			Case II			Case III `		
	P_1 (MPa)	P_3 (MPa)	P_5 (MPa)	P_1 (MPa)	P_3 (MPa)	P_5 (MPa)	P_1 (MPa)	P_3 (MPa)	P_5 (MPa)
70	0.138	0.116	0.113	0.138	0.136	0.133	0.138	0.146	0.153
80	0.17	0.118	0.115	0.17	0.138	0.135	0.17	0.148	0.155
100	0.212	0.138	0.119	0.212	0.158	0.139	0.212	0.168	0.159
120	0.38	0.157	0.124	0.38	0.177	0.144	0.38	0.187	0.164
140	0.88	0.182	0.138	0.88	0.202	0.158	0.88	0.28	0.19
160	2.25	0.3	0.147	2.25	0.32	0.167	2.25	0.515	0.235
170	3.38	0.43	0.16	3.38	0.45	0.18	3.38	0.76	0.288

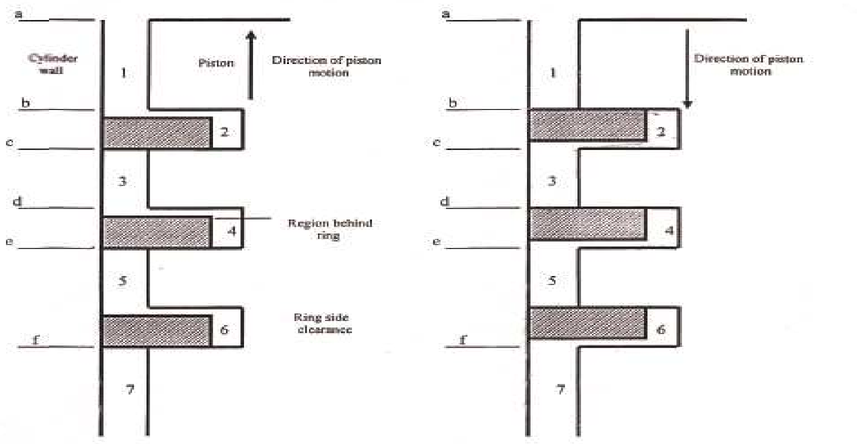
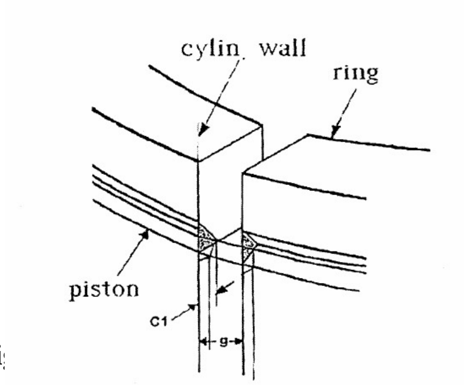


Fig. (3.1): Schematic Diagram for the Gas Flow Through Piston Rings [9].



Fi ... ring gap

Figure (3) Diesel engine Z-6901

Figure (4) Front view of static test rig

Figure (5) Top view of static test rig

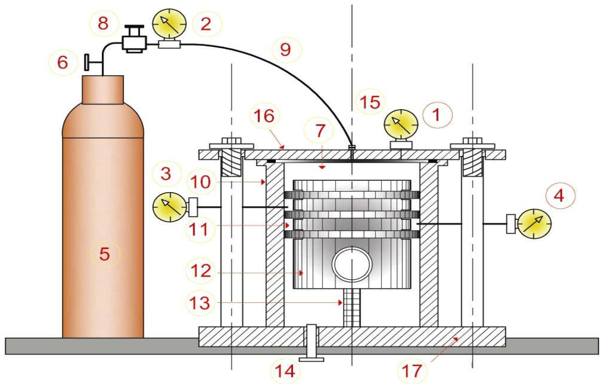
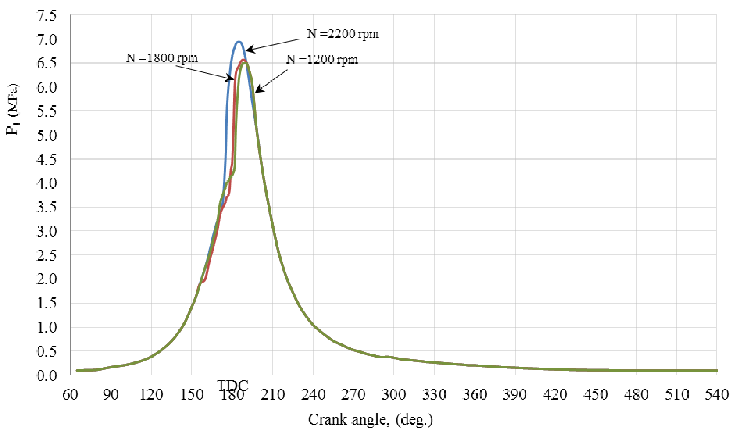


Figure (6) Schematic diagram of the static apparatus uses for measurement 1, 2, 3, 4.Pressure gauge 5. Compressed gas bottle 6. Control valve 7. Combustion chamber 8. Regulator valve 9. Connected hose 10. Cylinder liner 11. Piston ring 12. Piston 13. Moving supported screw 14. Safety valve 15. Rubber O-ring 16. Upper plate 17. Lower plate



Figure(7) Pressure indicator diagram of as a function of crank angle , case I

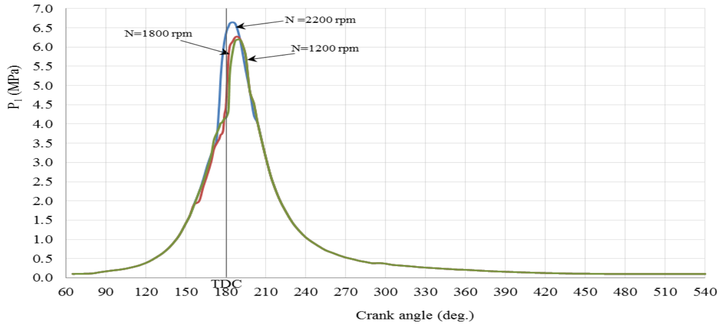


Figure (8) Pressure indicator diagram of as a function of crank angle , case II

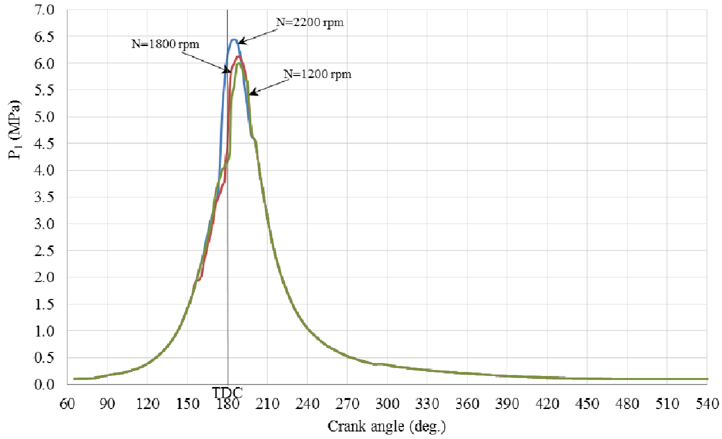


Figure (9) Pressure indicator diagram of as a function

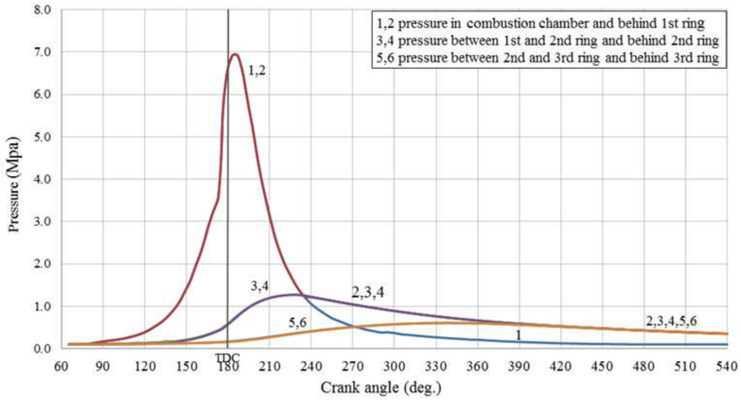


Figure (10) Pressure distributions between and behind piston rings as functions of crank angle case (I) ¸ N =1200 rpm

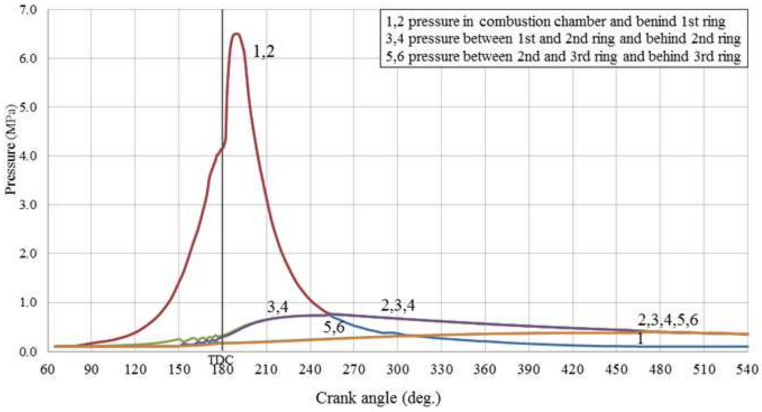


Figure (11) Pressure distributions between and behind piston rings as functions of crank angle case (I) , N =1800 rpm

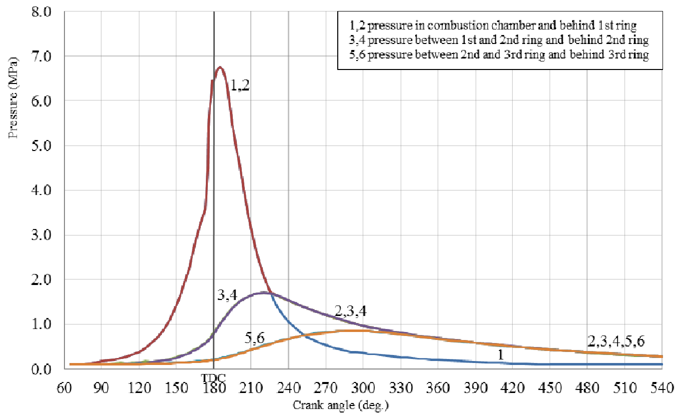


Figure (12) Pressure distributions between and behind piston rings as functions of crank angle case (I) , N =2200 rpm

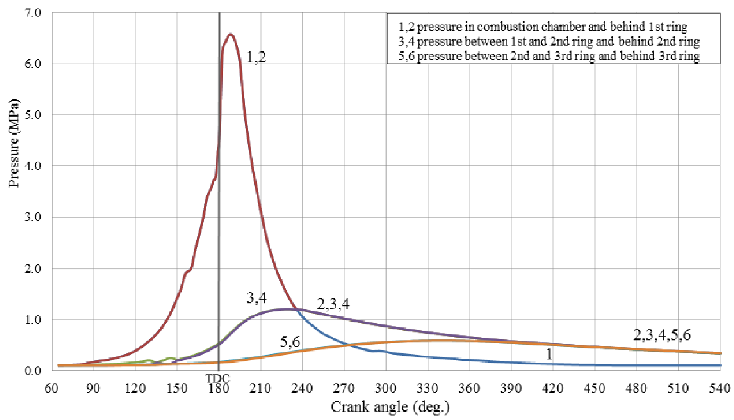
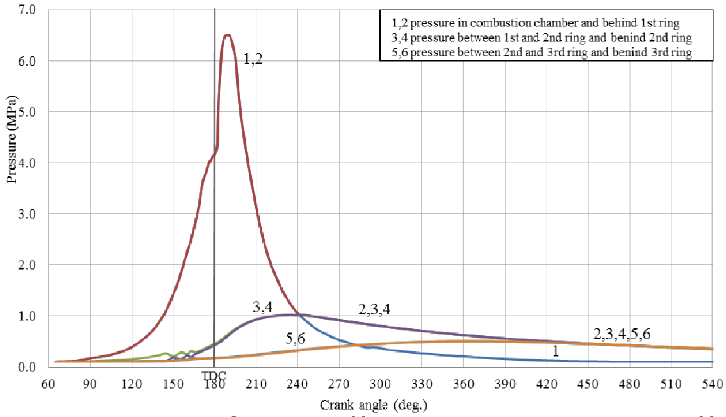
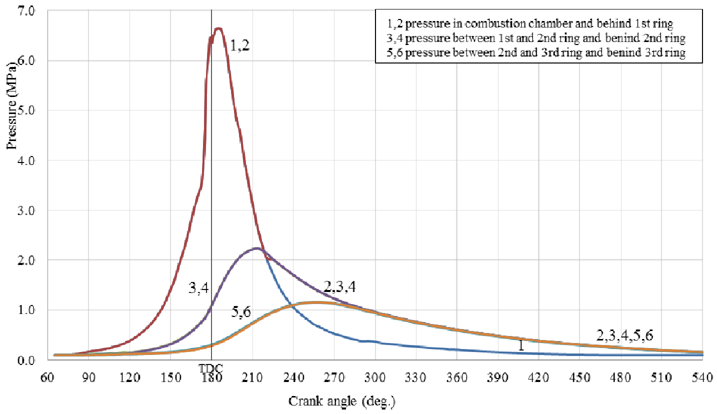


Figure (13) Pressure distributions between and behind piston rings as functions of crank angle case (II) , N =1200 rpm



case (II)¸ N =1800 rpm



Figure(15) Pressure distributions between and behind piston rings as functions of crank angle case (II)¸ N =2200 rpm

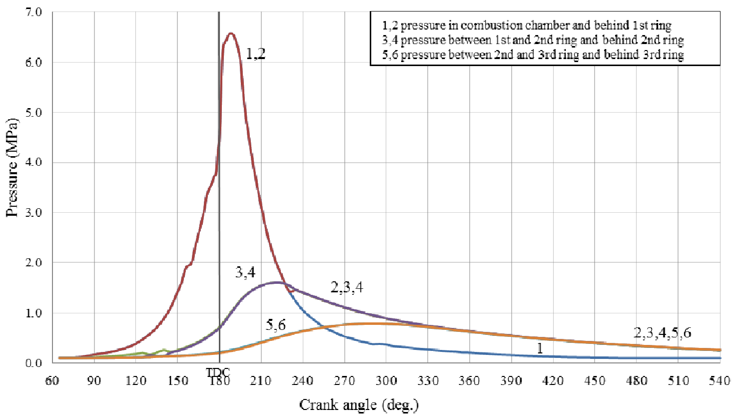


Figure (16) Pressure distributions between and behind piston rings as functions of crank angle case (III) ¸ N =1200 rpm

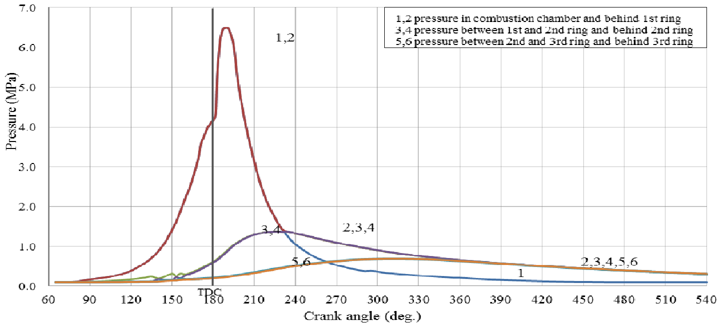
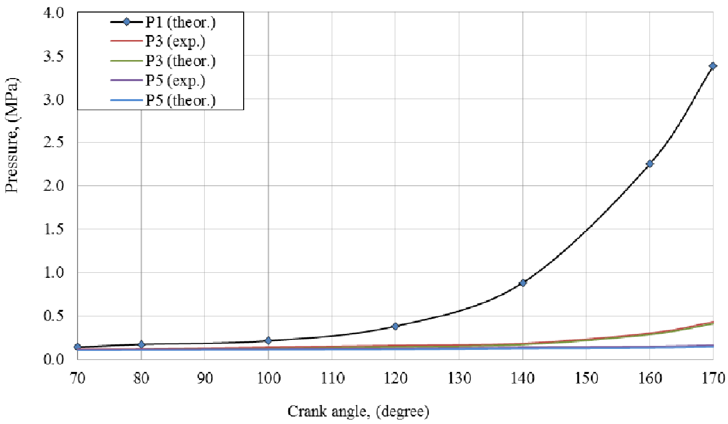


Figure (17) Pressure distributions between and behind piston rings as functions of crank angle case (III) ¸ N =1800 rpm



Figure(18) Pressure distributions between and behind piston rings as functions of crank angle case (III)¸ N =2200 rpm

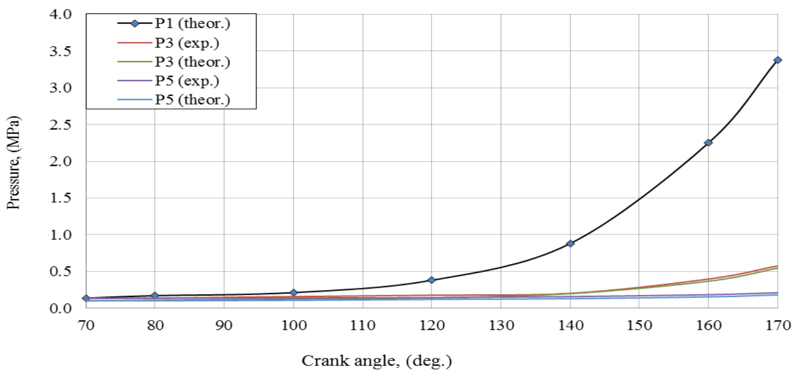
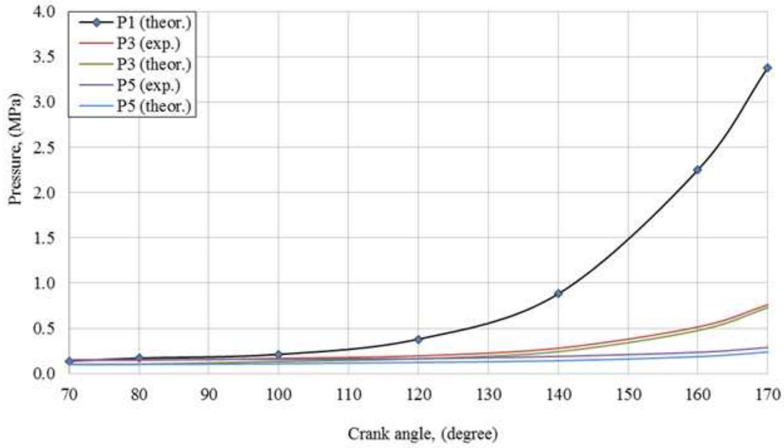


Figure (19) Theoretical and experimental results of pressures inside the inter-ring crevices region P_1, P_3&P_5 as functions of crank angle case (I)



Figure(20) Theoretical and experimental results of pressures inside the inter-ring crevices region P_1, P_3&P_5 as functions of crank angle case (II)

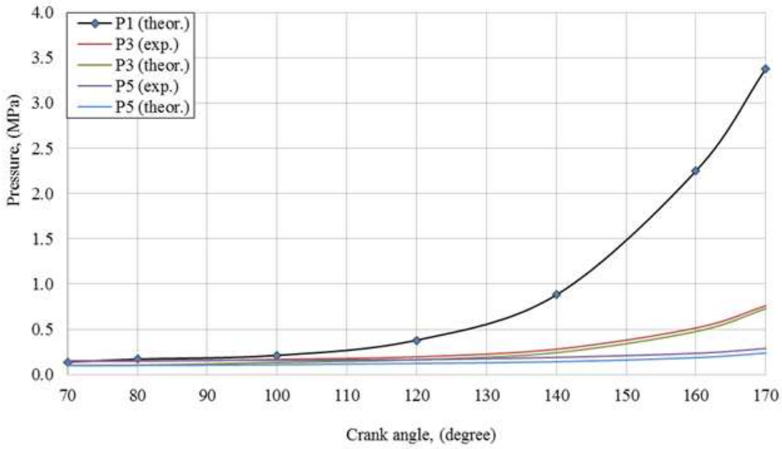


Figure (21) Theoretical and experimental results of pressures inside the inter-ring crevices region P_1, P_3&P_5 as functions of crank angle case (III)