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THEORETICAL AND EXPERIMENTAL STUDY OF PRESSURE PREDICTION BETWEEN PISTON RING FOR RECIPROCATING AIR COMPRESSOR

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ABSTRACT

An reciprocating air compressor type (V-AK 150) of three compression rings is employed in this work to accomplish the experimental work . The indicator diagram is recorded at different compressor speeds. The present work elucidates the measurement and indication of gas pressure within compression space as a function of crank angle. The study contains the prediction of pressure in inter-ring volume pressure behind piston ring during compressor 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 (1-50 bar).The comparison between mathematical and experimental results show good agreement.

Keywords: Reciprocating air compressor, Piston rings.

دراسة نظرية وعملية للتنبؤ بالضغط بين حلقات المكبس لضاغطة الهواء الترددية

الخلاصة :-

تم استخدا م ضاغطة هواء ترددية نوع (فى–اى كى 150) ذو ثلاث حلقات ضغط لا كمال متطلبات الجزء العملى من البحث و الحصول على مخططات الضغط كدالة لزوايا عمود المرفق بسرع مختلفة . يتضمن الجانب العملى قيلس منحني الضغط المبين فى حيز الانضغاط كدالة لزاوية عمود المرفق و كذ لك التنبؤ با لضغوط بين حلقات المكبس والضغوط خلف الحلقات اثناء اداء الضاغطة . تم بناء منظومة متكاملة لقياس الضغوط بين حلقات المكبس لعدة سرع من الاداء و لغرض التحقيق من صحة النتائج النظرية تم مقارنتها مع النتائج العملية .الضغوط المقاسة بين حلقات المكبس يتراوح مد اها 1− 50 بار . تم الحصو ل على نتائج جيدة تبين ان هنا ك توافقا جيداً بين النتائج العملية و النتائج التى تم الحصول عليها من النموذج الرياضى .

NOMENCLATURE

A: area (m^2)
D_h :Hydraulic diameter (m)
f :Friction coefficient
$m^{\cdot}$:Mass flow rate (kg/s)
P_i : Upstream pressure (bar)
P_j : Downstream pressure (bar)
R : Specific gas constant (kg .j /k)
R_e : Reynolds number
T : Temperature of ring (k)
Z : Length of channel (m)

Greeks

ρ **:** Density (kg/m^3)
μ : Dynamic viscosity of gases (kg/ m.s)

INTRODUCTION

Piston and Piston Rings

A piston is a cylindrical compressor component that slides back and forth in the cylinder bore by forces produced during the compression process . The piston acts a movable end of the compression space .The stationary end of the compression space is the cylinder head. Piston are commonly made of a cast aluminum alloy for excellent and lightweight thermal conductivity . Thermal conductivity is the ability of a material to conduct and transfer heat. Aluminum expands when heated ,and proper clearance must be provided to maintain free piston movement in the cylinder bore. Insufficient clearance can cause the piston to seize in the cylinder . Excessive can cause a loss of compression and an increase in piston noise[Akalin,2001].

Piston features include the piston head , piston pin bore ,piston pin , skirt, ring grooves, ring lands, and piston rings. The piston head is the top surface (closest to the cylinder head) of the piston which is subjected to tremendous forces and heat during normal air compressor operation. A piston pin bore is a through hole in the side of the piston perpendicular to piston travel that receives the piston pin. A piston pin is a hollow shaft that connects the small ends of the connecting rod to the piston. The skirt of a piston is the portion of the piston closest to the crankshaft that helps align the piston as it moves in the cylinder bore. Some skirts have profiles cut into them to reduce piston mass and to provide clearance for the rotating crankshaft counterweights.

A ring groove is a recessed area located around the perimeter of the piston that is used to retain a piston ring. Ring lands are the two parallel surfaces of the ring groove which function as the sealing surface for the piston ring .A piston ring is an expandable split ring used to provide a seal between the piston an the cylinder wall. Piston rings are commonly made from cast iron. Cast iron retains the integrity of its original shape under heat ,load ,and other dynamic forces. Piston rings seal the compression space ,conduct heat from the piston to the cylinder wall, and return oil to the crankcase .Piston ring size and configuration vary depending on compressor design and cylinder material.[H.Kind,2004]

Piston rings seal the compression space, transferring heat to the cylinder wall and controlling oil consumption .A piston ring seals the compression space through inherent and applied pressure. Inherent pressure is the internal spring force that expands a piston ring based on the design and properties of the material used.

Inherent pressure requires a significant force needed to compress a piston ring to a smaller diameter .Inherent pressure is determined by the uncompressed or free piston ring gap .Free piston ring gap is the distance between the two ends of a piston ring in an uncompressed state. Typically, the greater the free piston ring gap ,the piston ring applies when compressed in the cylinder bore[Akimoto,2000].

A piston ring must provide a predictable and positive radial fit between the cylinder wall and the running surface of the piston ring for an efficient seal . The radial fit is achieved by the inherent pressure of the piston ring. The piston ring must also maintain a seal on the piston ring lands. In addition to inherent pressure , a piston ring seals the compression space through applied pressure. Applied pressure is pressure applied from compression gases to the piston ring, causing it to expand . Some piston rings have a chamfered edge opposite the running surface.

Compression Ring

The compression ring is the top or closest ring to compression gases and is exposed to the greatest amount of chemical corrosion and highest operating temperature. The compression ring transfers 70% of the compression space heat from the piston to the cylinder wall. Most Briggs and Stratton engines use either taper-faced or barrel -faced compression rings. A taper faced compression ring is a piston ring that has approximately a one taper angle on the running surface .This taper provides amid wiping action to prevent any excess oil from reaching the compression space.[Kyung,2007]

Oil Ring

An oil ring includes two thin rails or running surfaces. Holes or slots cut into the radial center of the ring allow the flow of excess oil back to the oil reservoir . Oil rings are commonly one piece, incorporating all of these features. Some on-piece oil rings utilize aspiring expander to apply additional radial pressure to the piston ring. This increase the unit (measured amount of force and running surface size) pressure applied at the cylinder wall.

The function of a piston ring are to seal off the compression space, to distribute and control the oil, to transfer heat , and to stabilize the piston .The piston is designed for thermal expansion , with a desired gap between the surface and liner wall. The rings and the ring grooves from a labyrinth seal , which relatively well isolates the compression space from the crankcase .The position and design of the ring pack is shown in Fig-1 .The ring face conforms to the liner wall and moves in the groove , sealing off the route down to the crankcase. The sealing ability of the ring depends on a number of factors, like ring and liner conformability ,pre-tension of the ring , and gas force distribution on the ring faces. Some of the compression space heat energy is transferred through the piston to the piston boundaries ,i.e . the piston skirt and rings ,from which heat transfers to the liner wall . Furthermore , the piston rings prevent excess lubrication oil from moving into the compression space by scraping the oil from the liner wall during the down stroke. The piston rings support the piston and thus reduce the slapping motion of the piston .[Reinhard,2006&Shengyi,2010]

The following is summary of the most significant published work involved in the important function of piston rings in reciprocating air compressor (RAC).

Tian. [2000]. The ring twist affect the access of the gas pressure flow behind and between the piston rings , which causes non-uniform contact pressures and reduced conformity. In ring twisting , different sections of the face surface of the ring alternate to form the ring / liner contact, and this leads to a non-uniform contact pressure.

M. Voigt [2000]. Studied the dynamic behavior of piston ring and cylinder of large bore diesel engine.
Sutaria B.M. and Bhatt D.V. [2009] , studied a basic tribological parameters that influences performance of an internal combustion engine .
Richard Mittler and Albin Mierbach [2009] ,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 compressor operation of reciprocating air compressor 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.

THOREY

The engine compression space 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 compressor 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-2 .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 [sataria, 2009 ,Richard,2009].

Gas flow passages

Fig-3 . Show a schematic diagram of a piston with three compression rings, which represents the current of reciprocating air compressor . 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 prove and blocks that passage[Aungier,2000].

Gas flow equations

To model the flow, we follow the flow from the compression space 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 . 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 equation for the regions 2,3,4,5 and 6 can be written[G ,R. 1975] .

$$(m_{02} / dP_{02}) * (dP_2 / dt) = \dot{m}_{12} - \dot{m}_{23} \quad (1)$$

$$(m_{03} / dP_{03}) * (dP_3 / dt) = \dot{m}_{13} + \dot{m}_{23} - \dot{m}_{34} - \dot{m}_{35} \quad (2)$$

$$(m_{04} / dP_{04}) * (dP_4 / dt) = \dot{m}_{34} - \dot{m}_{45} \quad (3)$$

$$(m_{05} / dP_{05}) * (dP_5 / dt) = \dot{m}_{35} + \dot{m}_{45} - \dot{m}_{56} - \dot{m}_{57} \quad (4)$$

$$(m_{06} / dP_{06}) * (dP_6 / dt) = \dot{m}_{56} - \dot{m}_{67} \quad (5)$$

Mass flow rates of gases through cross section (a) can be described by the following equation.

$$\dot{m}_a = (m_{01} / P_{01}) * (dP_1 / dt) + \dot{m}_b \quad (6)$$

The change in mass filling of individual inter-ring volumes is dependent on time and can be described by the following differential equations:

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

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

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

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

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

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

Mass flow rate 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 [Dixon,2008].

$$\dot{m} = A * \{ P_i^2 - P_j^2 / R.T (4. f . z / D_h + 2 \ln P_i / P_j)\}^{0.5} \quad (12)$$

The flow through the ring side- clearance is laminar ,and the friction coefficient can be expressed by the relation[Hoefiner,1996] :

$$f = 16 / R_e \quad , \quad R_e = \rho .V . D_h / \mu_g \quad (13)$$

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

$$f = 16 \mu_g / \rho .V. D_h \quad (14)$$

Generally the expression $2 \ln P_i / P_j \ll 4. f\ z / D_h$) thus equation (12) can be expressed as following equation [Stefano,2007] :

$$\dot{m} = A * \{ (P_i^2 - P_j^2) / R. T . (4. f. z / D_h) \quad (15)$$

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

$$\dot{m} = D_h^2 .A . (P_i^2 - P_j^2) / 64 . R.T .Z . \mu_g \quad (16)$$

The dynamic viscosity of flowing gas can be calculated from the following equation[Namazian,1982].

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

Piston -cylinder crevice flow model

The air compressor (V-Ak 150) , two cylinder type is used for experimental measurement as shown in (Fig- 2). 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 compressor has three compression rings and one oil ring . The flow through the oil ring can be neglected in this work . 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 air compressor(V-AK150) For three piston group element for case A,B and C at compressor speed (N= 1000,1500 and 2000 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. Fig- 4 .Shows a schematic diagram of static measurement apparatus which is used to perform measurement.

The main parts of static apparatus are:

1. Cylinder of gas ; that is filled by the Nitrogen gas up to pressure of (70 bar) and its connected hose (9) to the compression space (7). The pressure gauge (1) indicates the pressure of the gas in the compression space.

2. In order to maintain cylinder pressure uniform during measurement , pressure gauge (2) is connected to the cylinder head in order to measure the cylinder pressure as close as to actual value during compressor 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 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 plate 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 compression space 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 compression space until the pressure gauge reach (15 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 (50 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 B and C . Table -2. Show the results of pressure measurement in three cases for three piston groups.

RESULTS AND DISCUSSION

The result 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.[Won,2005

Results of experimental test

Induction of pressure-curve

As mentioned in experimental work the results measurement i.e . indication of curve as a function of crank angle in the working volume of the measured cylinder during compressor operation under full load performance are shown in Figs.(5,6 and 7) for the three cases of piston group i.e (A,B and C) at air compressor speeds of (N= 1000 , 1500 and 2000 rpm) .

It is clear that the pressure curve inside the cylinder for new air compressor (case –A) is higher than that in (case-B) and (case- C) due to different wear of piston group elements in each case. These figures also show the effect of compressor speed on the course of pressure in the compression space . 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 compression space .It is also clear that while the piston groups wear increases the pressure-curve level decrease, as in case (B and C) .

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_1) in the inter-ring volumes (1,3 and 5) are carried out as shown in (Fig- 17) to (Fig- 19) and Table- 2 , for the three cases of piston group within the range of pressure during compression stroke only. The results of experimental test rig as it is shown in Fig-17. To Fig- 19, indicates a good agreement with the predicated theoretical pressure. It is clear that at a constant compressor speed, the difference between the theoretical and experimental results are increased with increasing the wear of the pistoure uren group elements. This is due to higher leakage of gases through the piston ring end gap. These differences are decreased with increasing compressor speed due to less time for flow of gases or blow-by from compression space .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.

Results of computer simulation

The results of pressure changes in inter-ring volumes as a function of crank angle for three cases (A,B and C) at compressor speeds of (N=1000, 1500 and 2000)rpm are shown in Fig-8 to Fig- 16. From these results it is shown that the pressure in crevice region (1) is equal to pressure of compression space .They also show that any two adjacent regions connected by ring side clearance have the same pressure (i.e $P_1 = P_2$, $P_3 = P_4$, $P_5 = P_6$). These results indicate that the theoretical model is working reliably.

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 compression 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 compression space.

2. The maximum pressure value P_5 is attained later i.e at 180^0 after TDC due to continuous flow of gases from region 3 to region 5 for all cases.
3. The effect of compressor speed on the value of pressure in region 3 and 5 are shown in Fig-8 to Fig-16 for three cases, it is clear that when compressor speed decreases the value of pressure P_3 and P_5 are increased due to enough time of gases leakage from compression space to region 3,5 and 7 .
4. The effect of piston group wear 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 17.5 bar i.e at 25% of cylinder pressure for piston group (C).

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 air compressor . 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 evaluation 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 reciprocating air compressor. 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 compressor operating.
4. The computation results achieved with 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 compressor cylinder volume.

Table 1 . Summary chart of the three wear of piston rings

Case	A	B	C
1_{st} ring gap length (mm)	0.15	0.23	0.35
2_{nd} ring gap length (mm)	0.15	0.23	0.35
3_{rd} ring gap length (mm)	0.15	0.23	0.35

Case A : It represents a new piston ring (standard).
Case B : It represents halve wear which occurs in piston rings.
Case C : It represents full wear which occurs in piston rings.

Table 2.Show the results of pressure measurements in three cases for three piston groups.

Crank angle (deg)	Case- A			Case- B			Case- C		
	P_1 (bar)	P_3 (bar)	P_5 (bar)	P_1 (bar)	P_3 (bar)	P_5 (bar)	P_1 (bar)	P_3 (bar)	P_5 (bar)
70	1.7	1.4	1.1	1.7	1.5	1.3	1.7	1.6	1.5
80	2.9	2.2	1.3	2.9	2.5	1.5	2.9	2.7	1.7
100	5.7	2.7	1.4	5.7	2.9	1.6	5.7	3.1	1.8
120	9.8	3.1	1.6	9.8	3.5	1.9	9.8	3.7	2.1
140	20.2	3.8	1.7	20.2	4.2	2.1	20.2	4.8	2.4
160	39.7	5.1	1.8	39.7	5.8	2.3	39.7	6.3	2.6
170	50	6.2	1.9	50	6.4	2.5	50	7.6	2.9

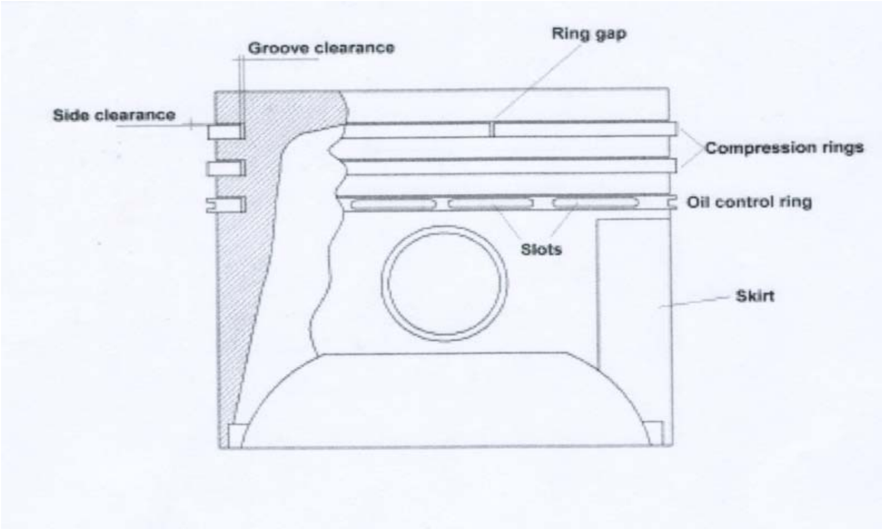

Fig-1. Piston and piston rings

Fig-2. Reciprocating air compressor (V-AK 150)

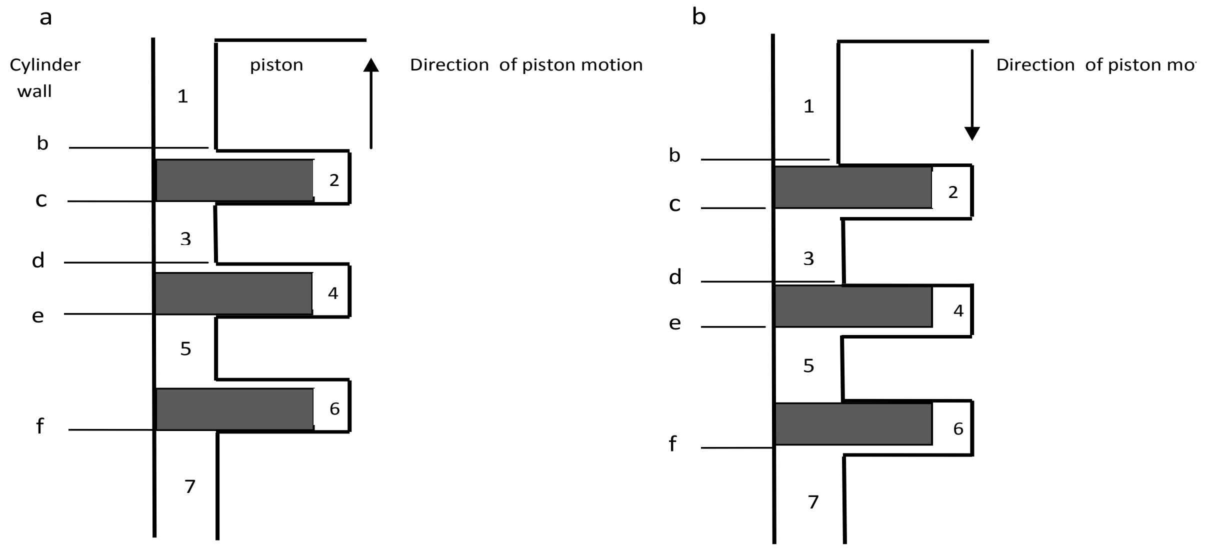

Fig-3. Schmatic diagram for the gas flow through piston rings.

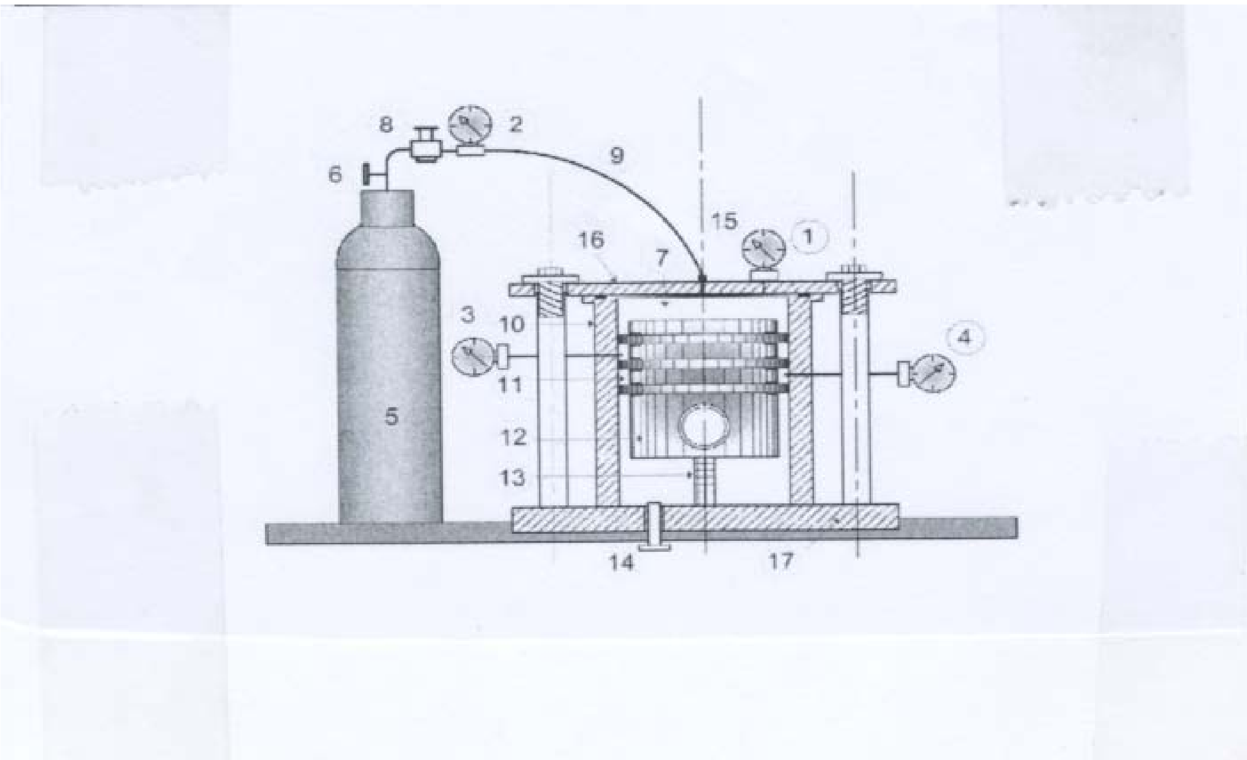

Fig-4.Schematic diagram of the static apparatus uses for experimental measurement .

1,2,3,4 ----- Pressure gauge 5. Compressed gas bottle 6. Control valve 7. Compression space
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

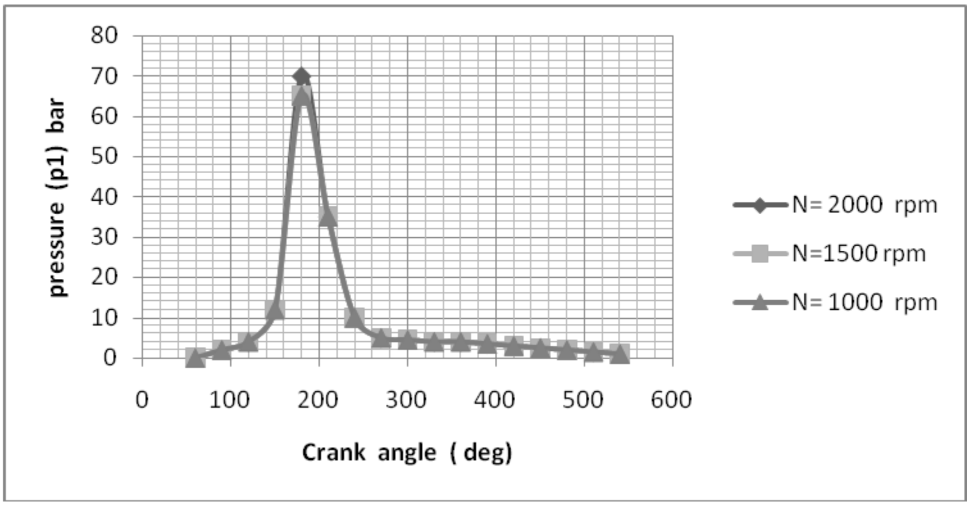

Fig-5. Pressure indicator diagram of as a function of crank angle , case A.

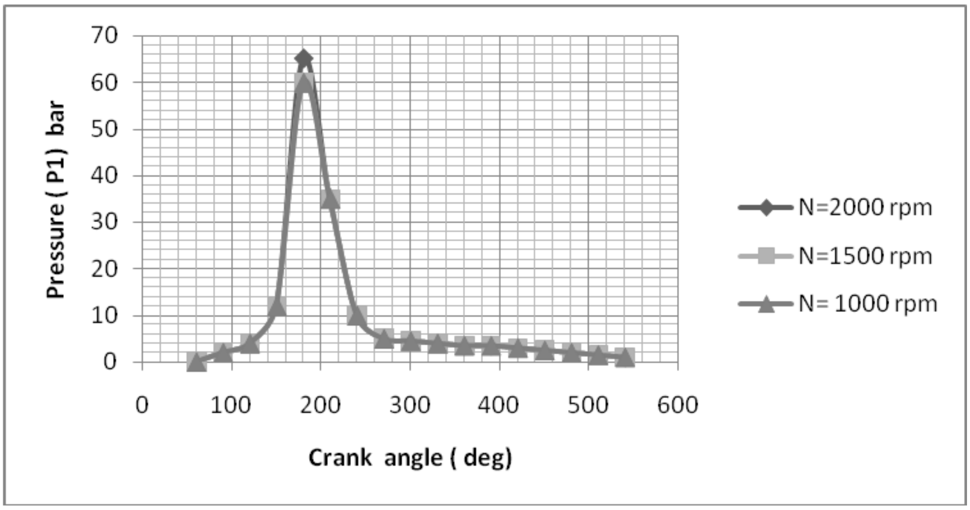

Fig -6 . Pressure indicator diagram of as a function of crank angle, case B.

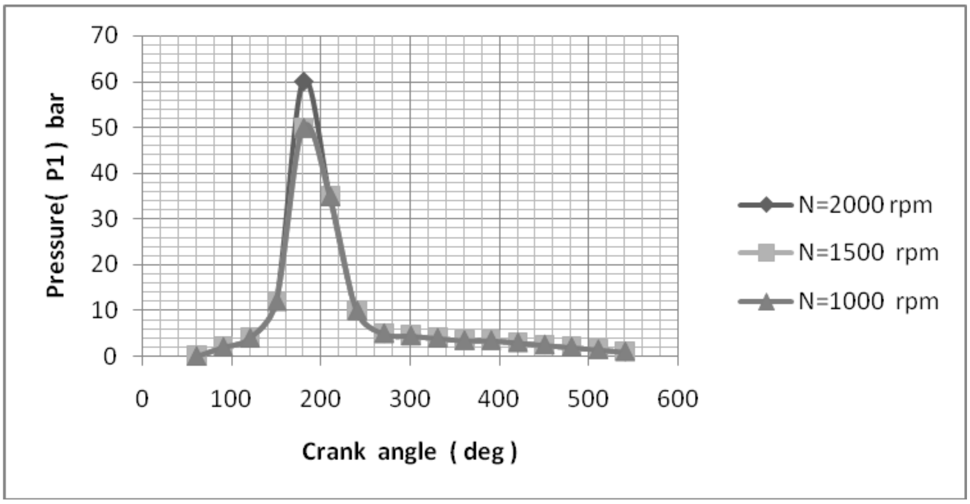

Fig-7. Pressure indicator diagram of as a function of crank angle ,case C.

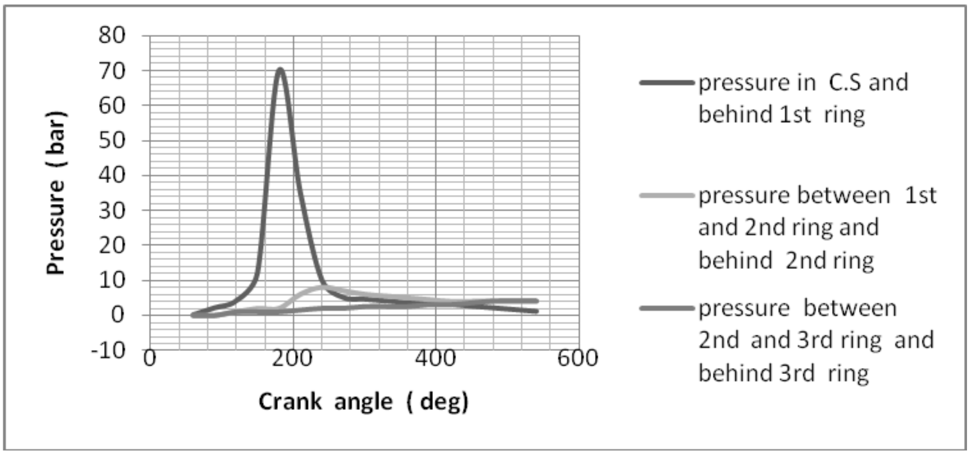

Fig-8. Pressure distribution between and behind piston ring,case A,N=1000

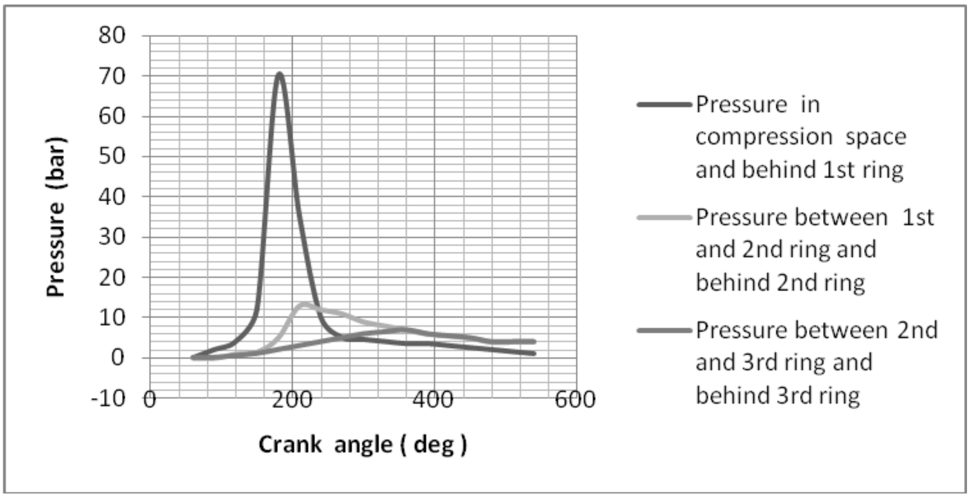

Fig-9. Pressure distribution between and behind piston ring,case A,N=1500 rpm.

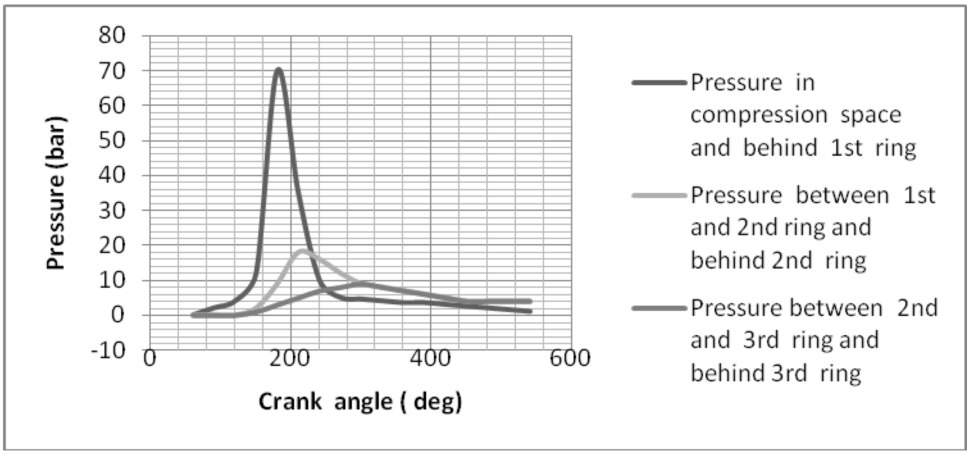

Fig-10.Pressure distribution between and behind piston ring,case-A, N=2000rpm

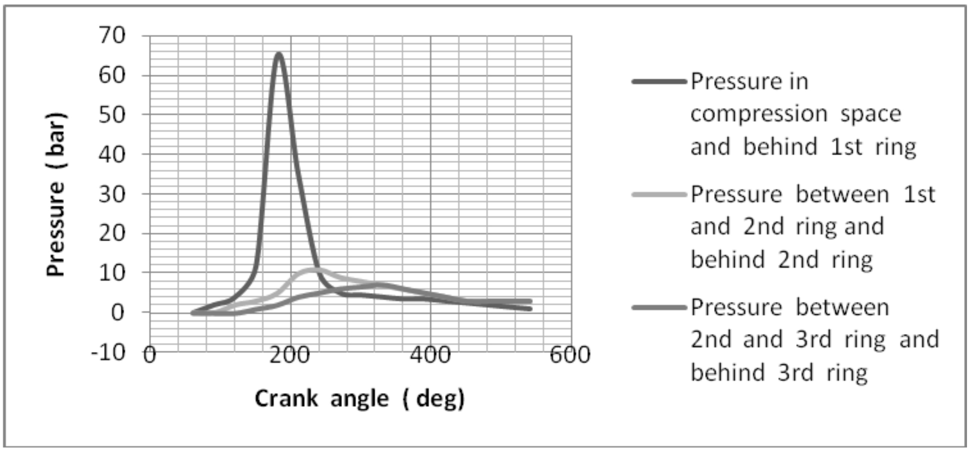

Fig-11.Pressure distribution ,case- B , N= 1000 rpm.

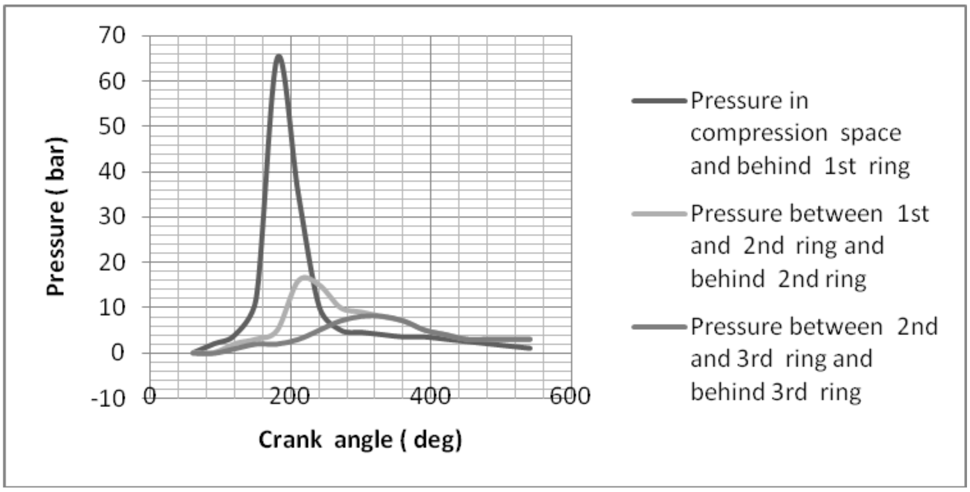

Fig-12. Pressure distribution , case B ,N= 1500 rpm.

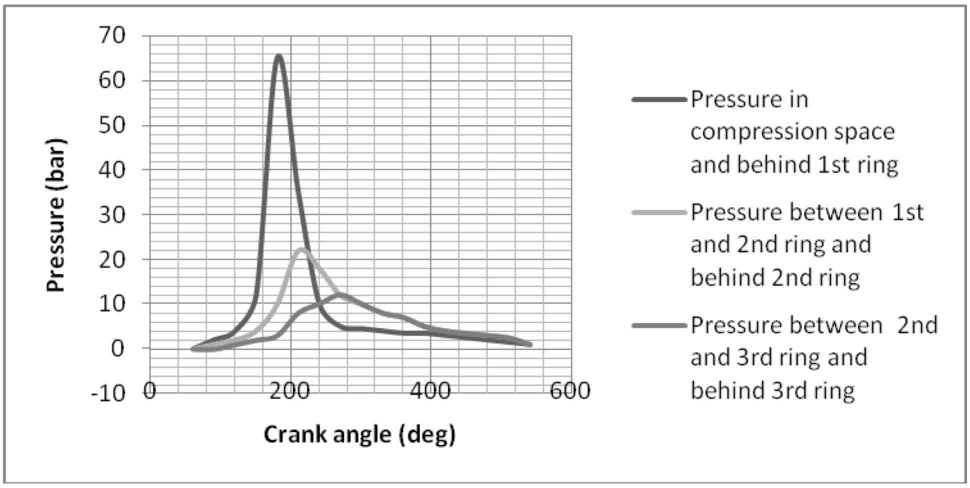

Fig-13. Pressure distribution , case –B ,N =2000 rpm.

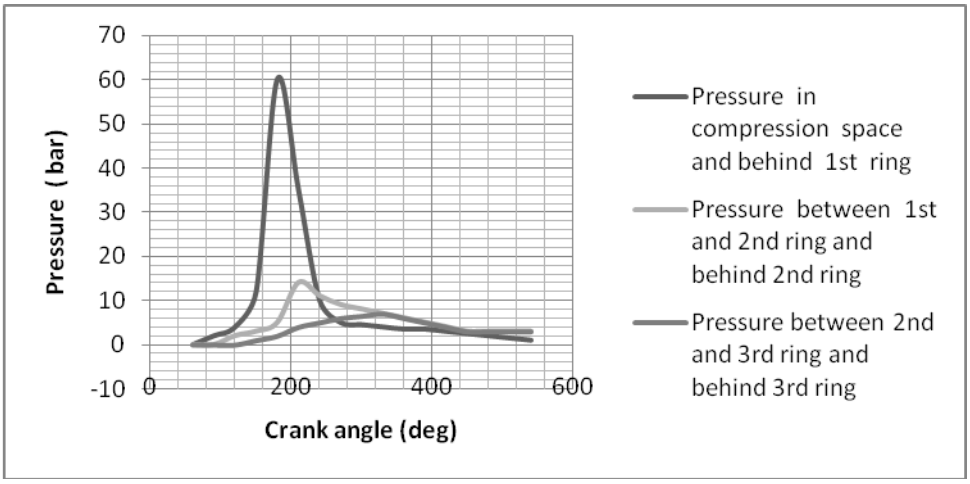

Fig -14. Pressure distribution, case- C, N=1000 rpm .

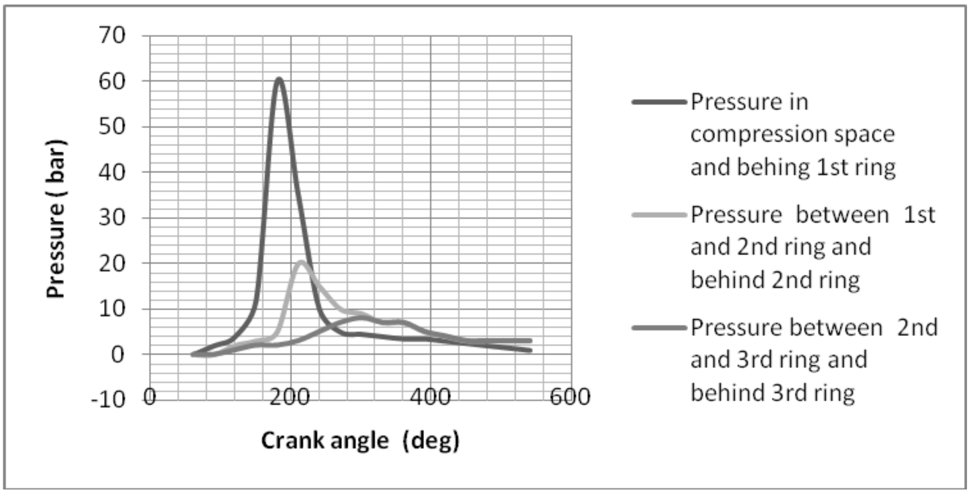

Fig-15. Pressure distribution case –C ,N=1500 rpm

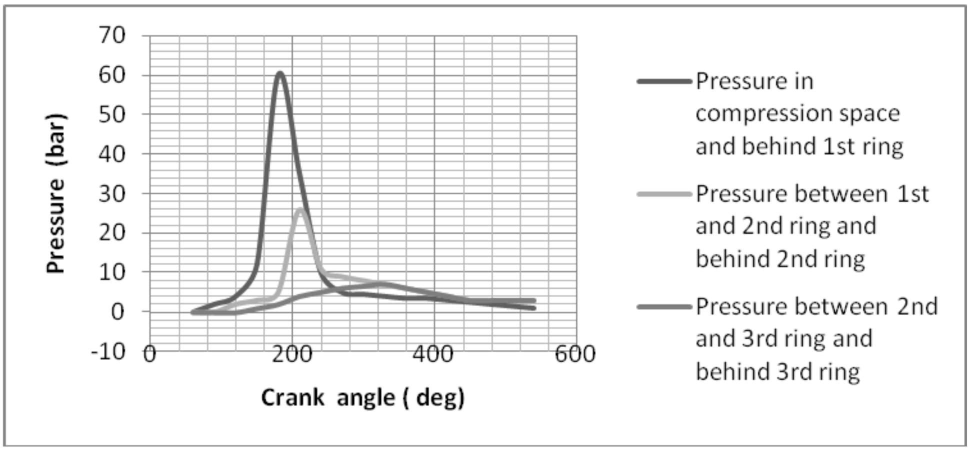

Fig-16 .Pressure distribution case-C ,N= 2000 rpm

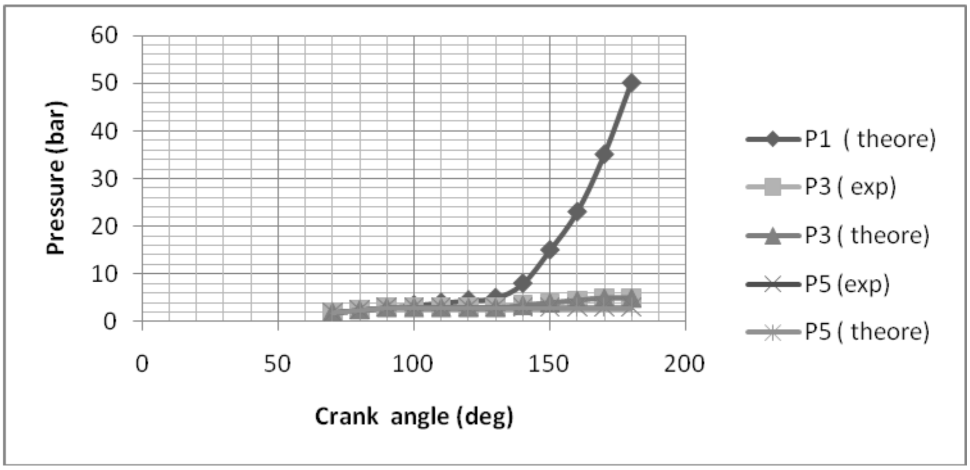

Fig-17. Theoretical and experimental results of pressures inside the inter-ring crevices region P1,P2,P3 as function of crank angle case-C.

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