



# Analysis of the Blow-by in Piston Ring Pack of the Diesel Engine

Na Liu<sup>a</sup>, Zhongcai Zheng<sup>\*a</sup>, Guoxiang Li<sup>b</sup>

<sup>a</sup> School of Mechanical and Electrical Engineering, Shandong Jianzhu University, Jinan, 250101, China

<sup>b</sup> School of Power and Energy Engineering, Shandong University, Jinan, 250061, China  
zcc638@163.com

Favourable sealing performance of the piston ring pack is an important condition for the engine to maintain efficient, stable operation. Analysis on blow-by in piston ring pack for an inline six cylinder diesel engine was carried out, through the establishment of sealing system theory model. The Matlab program was compiled for numerical calculation. The results of the study provide a basis for improving the air tightness of the piston ring pack, reducing air leakage, thereby improving the performance of internal combustion engine.

## 1. Introduction

Piston rings are key components of engine, sealing, on the role of adjusting the oil, heat transfer, and guiding, which will directly affect the engine performance and fuel economy (Su (1998), Yu (2005)). Sealing is the main task of gas rings, namely sealing the gas, to prevent the gas of combustion chamber to flow into the crankcase, and to keep the amount of gas leakage as few as possible (Kong (1992), Lu (2014)). If bad sealing of piston rings occurs, the amount of leakage of gas will increase greatly, which leads to the reduction of engine power. At the same time, leakage increase will induce the modification of lubricating oil, affect forces acting on the piston ring, then indirectly induce severe wear of the system (Tian (2002)), eventually lead to the sharp decline in the service life of the engine. The study of Piston Ring Pack's gas pressure and leakage, not only can provide guidance for the design of piston ring, but also is the essential prerequisite to study the lubrication and friction performance of the cylinder piston ring system (Zavos (2015), Mohammed (2015), Senatore (2014), Poort (2014, 2015), Wolff (2014), Koszałka (2014), Krisada (2008)).

## 2. Leakage channel analysis

Many researchers have investigated and analyzed the gas flow in the system piston-rings-cylinder [7]. Leakage channel of piston ring pack was shown in figure 1. There were two ways for leakage passage. One way was to generate the relative motion of the piston ring and the ring groove, the gas flowed from the ring body and the ring groove clearance, such as shown in Figure 1 (a). The gap between the piston ring body and ring groove was very small, thus through this channel, the gas had small amount of leakage. It could be approximately considered as one-dimensional laminar flow, and one-dimensional laminar flow formula could be applied for study (Kong Long (2007)).

According to the analysis of piston ring pack movement, the piston ring body and the ring groove were attached together for most of the time, which formed a sealed space, so the form of leakage could be negligible. Another way of gas leakage was as shown as in Figure 1 (b), Gas leaked through the piston ring gap, which was the main form of piston rings blow-by.

## 3. Establishment of gas leakage calculation model

According to the analysis of leakage channel, leakage of piston ring pack is mainly produced by the opening gap of piston ring. Leakage channel can be formed in the adjacent piston rings, as shown in Figure 2.

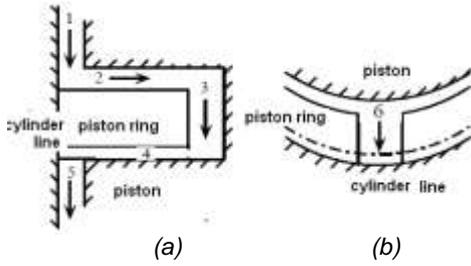


Figure 1: Leakage channel of the piston ring

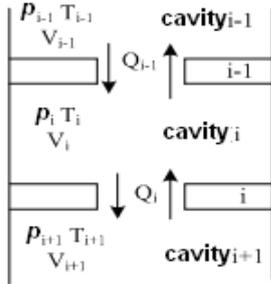


Figure 2: Leakage channel of the piston ring pack

Gas leakage on the opening gap of piston ring is analyzed based on the following assumptions,

- 1) Flowing mode for gas is reversible adiabatic flow;
- 2) Gas leakage caused by the crankcase and the combustion chamber pressure variation is ignored, the temperature in the combustion chamber and in the crankcase are considered are constants;
- 3) Leaked gas is assumed as ideal gas to maintain a constant chemical composition, gas constant and isentropic index remains unchanged in the process of leaking;
- 4) Deformation factors just as piston ring's bending and torsion in the ring groove are ignored, gas volumes in cavities are considered as constants.

Based on the above assumptions, the flow equation of the reversible adiabatic process for ideal gas is used [8], mass flow rate through the gas piston ring gap can be obtained as,

$$Q_i = \begin{cases} \psi_i \times \left( \frac{P_{i+1}}{P_i} \right)^{\frac{1}{k}} \sqrt{1 - \left( \frac{P_{i+1}}{P_i} \right)^{\frac{k-1}{k}}} & \frac{P_{i+1}}{P_i} > 0.546 \\ \psi_i \times 0.546^{\frac{1}{k}} \sqrt{1 - 0.546^{\frac{k-1}{k}}} & \frac{P_{i+1}}{P_i} \leq 0.546 \end{cases} \quad (1)$$

In which,  $\psi_i = K_c A_i \sqrt{\frac{2k}{R_g (k-1) T_i}} P_i$ ,  $K_c$  meant the flow coefficient, which can be obtained from the formula as

$$K_c = 0.85 - 0.25 \left( \frac{P_{i+1}}{P_i} \right) \quad (\text{Fu Qinsheng (2007)}), \quad k \text{ meant the adiabatic index; when } i = 1, P_1 \text{ meant the}$$

combustion chamber pressure value in each crank angle which can be accepted by indicator diagram; when  $i = n$ ,  $P_n$  meant crankcase pressure, which can be considered as constants.

The cavities between the piston rings as shown in Figure 2 is solved by ideal gas equation of state [9], which can be obtained that,

$$P_i = \frac{m_i R_g T_i}{V_i} \quad (2)$$

In which,  $R_g$  means gas constant;  $T_i$  means thermodynamic temperature of the gas flowing through the ring cavity;  $V_i$  means ring cavity volume;  $m_i$  means gas quantity in the ring cavity.

According to the law of conservation of mass, the mass flowing through the ring cavity can be obtained by:

$$\dot{m}_i = Q_{i-1} - Q_i \quad (3)$$

Both sides of the equal sign for formula 3 are demanded on time derivatives, which can be obtained:

$$\frac{dp_i}{dt} = \frac{R_g T_i}{V_i} \dot{m}_i + \frac{R_g m_i}{V_i} \frac{dT_i}{dt} \quad (4)$$

That is,

$$\frac{dp_i}{dt} = \frac{R_g T_i}{V_i} (Q_{i-1} - Q_i) + \frac{p_i}{T_i} \frac{dT_i}{dt} \quad (5)$$

There existed relationship between the crankshaft rotation angular velocity and the crank angle,

$$\varphi = \frac{180\omega}{\pi} t \quad (6)$$

Formula (5) can be written as,

$$\frac{dp_i}{d\varphi} = \frac{\pi}{180\omega} \frac{R_g T_i}{V_i} (Q_{i-1} - Q_i) + \frac{p_i}{T_i} \frac{dT_i}{d\varphi} \quad (7)$$

Ignoring the influence of the ring gas temperature change rate for pressure change, formula (7) can be written as,

$$\frac{dp_i}{d\varphi} = \frac{\pi}{180\omega} \frac{R_g T_i}{V_i} (Q_{i-1} - Q_i) \quad (8)$$

Formula (8) is the calculation formula of leakage that finally established. The equation is the first-order differential equation, and can be solved by the Runge-Kutta method.

#### 4. Leakage calculation process

According to the leakage model, the MATLAB language programming is used for caculation. The piston ring pack with  $n$  rings had  $n-1$  closed cavities, and  $n-1$  first-order differential equations can be built and solved by the Runge-Kutta method.

1) At first, assume the initial value  $p_{i0}$  ( $i = 2, 3, \dots, n-1$ ) for  $p_i$  ( $i = 2, 3, \dots, n-1$ ) under the crank angle of  $0^\circ\text{CA}$ ; the mass flow of the ring opening gap through the ring opening gap then can be obtained by solving the 2-4 equation;

2) Set the increment as  $1^\circ\text{CA}$  crank angle, obtain  $p_{ij}$  ( $i = 2, 3, \dots, n-1$   $j = 1, 2, \dots, 720$ ) for each  $1^\circ\text{CA}$  by solving the 2-11 equation using the Runge-Kutta method;

3) Calculate the corresponding cavity pressure  $p_{i720}$  ( $i = 2, 3, \dots, n-1$ ) under the crank angles of  $720^\circ\text{CAs}$ ,

if  $\sum_{i=2}^{n-1} (p_{i0} - p_{i720})^2 \geq 1e-5$ , let  $p_{i0} = p_{i720}$ , go to step (2),

until  $\sum_{i=2}^{n-1} (p_{i0} - p_{i720})^2 < 1e-5$ .

4) Complete the calculation, output the results of the cavity pressures.

The calculation process is shown in figure 3.

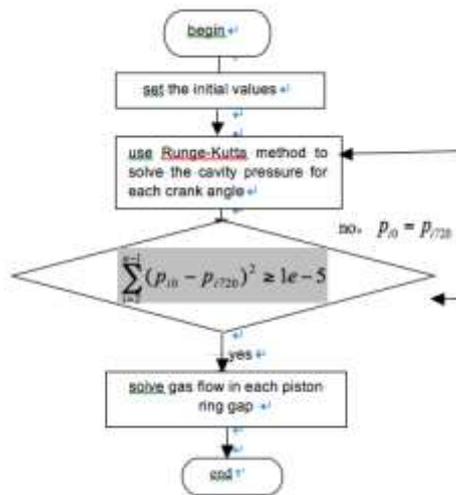


Figure 3: Leakage analysis flow chart

## 5. The result analysis of leakage

Through the calculation model, calculation and analysis of leakage for an inline six cylinder diesel engine were carried out. The piston ring pack consisted of two air rings and an oil ring, gas leaked through the ring gaps. The pressures in the combustion chamber were obtained by the indicator diagram. The isentropic exponent was taken as 1.3; the gas constant was taken as 288.4J / (kg.k); the combustion chamber temperature was taken as 650 °C; The crankcase pressure was taken as 0.1MPa ; the crankcase temperature was taken as 80 °C.

Figure 4 and figure 5 were shown as the calculated gas pressures and gas flow distribution between the piston ring pack.

0°CA in figure 4 and figure 5 was selected as the crank angle of the intake stroke TDC. As can be seen from figure 4, the combustion chamber gas pressures varied strongly with the crank angle. The diesel engine was a turbocharged diesel engine, so in the whole working cycle, the combustion chamber gas pressures were higher than the crankcase gas pressures. In the earlier power stroke, after about TDC 10°CA, the combustion chamber pressure reached the maximum value, 17.0MPa. Flowing through the piston ring, the gas pressure dropped rapidly with the throttling effect of the piston ring gap, the maximum value of the gas pressures between the top ring and the second air ring had been reduced to 3.4MPa, far below the combustion chamber gas pressure, only about 1/5 of the maximum combustion chamber pressure, which showed that the sealing effect of the top ring was significant. Because of the throttling effect of the piston ring gap, flowing through the ring gap, not only the maximum pressure value of the gas decreased rapidly, but also the position of the crank angle of which maximum value occurred was delayed. Flowing through the first piston ring, the maximum gas pressure value occurred at 400°CA, 30°CA later than the maximum combustion chamber pressure. Flowing through the second piston ring, the gas pressure value continued to decline, the maximum value was reduced to 2.7MPa, about 3/4 of the gas pressure between the top ring and the second air ring, the pressure decreased significantly than the top ring, the position that the maximum pressure value occurred also continued to delay, appeared at about 420° CA. After flowing through the oil ring, the gas pressure would reduce to the crankcase pressure at a constant value 0.1MPa, which was not shown in the figure. It also can be seen from the figure, because of the maximum values of the pressure hysteresis flowing through the piston ring, and the pressure in the combustion chamber decreased significantly in the power stroke, the pressure value between the top and the second air ring was higher than the pressure in the combustion chamber after the 420° CA, which provided conditions for back flowing.

As can be seen from Figure 5, gas leakage flow of the top was more than the second air ring and the oil ring, The gas flow rate of the top ring reached the maximum value near the compression TDC, approximately at the same time that the combustion chamber pressure occurred the maximum value, about 10° CA after the compression TDC. With the downward movement of the piston, the combustion chamber pressure reduced sharply, the gas flow also decreased, until 420° CA, the combustion chamber pressure value dropped below the pressure value between the top ring and the second ring, then the gas flowed back into the combustion chamber, and the gas flow rate became negative. After 420° CA crank angle, the combustion chamber pressure continued to decline, while the gas pressure between the top ring and the second gas ring declined slightly due to throttling effect of the piston ring gap. So in a wide range of crank angle after 420° CA, the

combustion chamber pressure was less than the pressure between the top ring and the second ring when the gas flowed from the gas ring to the combustion chamber and the gas flow rate was negative. Until the end of the exhaust stroke, the combustion chamber pressure was almost equal to the pressure between the top ring and the second ring, and the gas flow rate is almost zero. The gas flow rate through the oil ring was the gas leakage across the piston ring pack, As can be seen from the figure, the gas pressure before the oil ring was higher than the gas pressure in the crankcase, so the gas flow rate were positive, the gas flowed from the oil ring to the crank chamber.

## 6. Conclusion

This paper analyzed the blow-by in piston ring pack for an inline six cylinder diesel engine, through the establishment of sealing system theory model and used Matlab program for numerical calculation. Calculation results accord with the actual situation and provide a basis for improving the air tightness of the piston ring pack, reducing air leakage, thereby improving the performance of internal combustion engine.

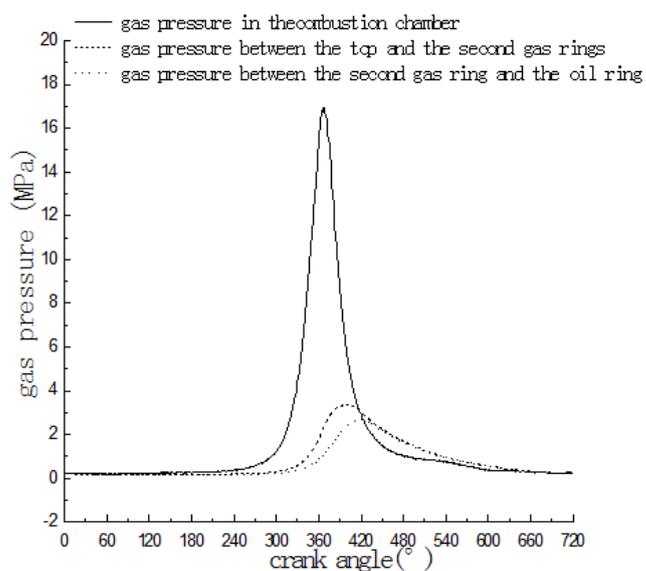


Figure 4: The pressure distribution between the piston ring pack

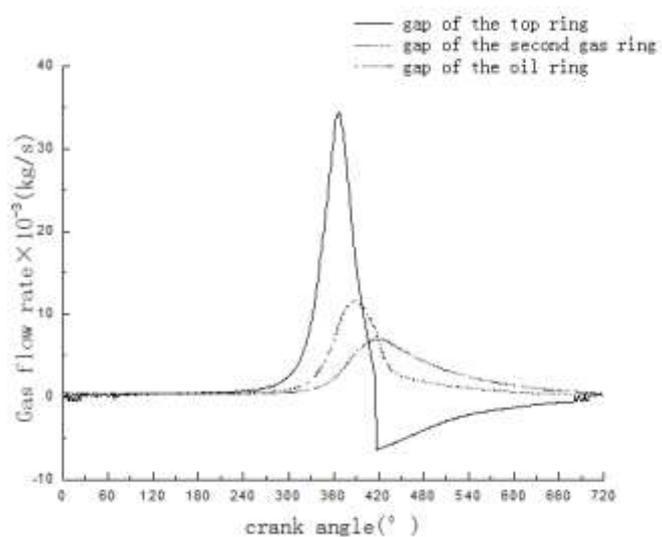


Figure 5: The gas flow distribution between the piston ring pack

## References

- Fu Q.S., 2007, *Thermodynamic fundamentals and Applications (Second Edition)*, Mechanical Industry Press, Beijing
- Kong L.J., Xie Y.B., 1992, Calculation of blow-by gas, lubrication, friction and wear in cylinder liner-piston ring tribo-systems [J]. *Transactions of CSICE*, 10(3), 267-274
- Kong L., 2007, *Engineering fluid mechanics [M]*, China electric power press, Beijing
- Koszałka, G., 2010, Application of the Piston–Rings–Cylinder Kit Model in the Evaluation of Operational Changes in Blowby Flow Rate, *Eksplotacja i Niezawodność [Maintenance and Reliability]*, 48 (4), 72-81
- Krisada W., Somchai C., 2008, Surachai S, Simulation algorithm for piston ring dynamics, *Simulation Modelling Practice and Theory*, 16(1), 127-146, DOI: 10.1016/j.simpat.2007
- Lu P., Zhang Y., Zhang M.H., Liu X.R., 2014, Simulation and analysis of the effect of piston ring group parameters on blow-by, *Journal of Chinese Agricultural Mechanization*, 35(2), 193-197, DOI: 10.13733/j.jcam.issn.2095-5553.2014.02.047
- Mohammed, U., Gokhale N., Pardeshi S., Gokhale U., Kumar M.N., 2015, Analysis of parameters affecting liner bore distortion in diesel engines, *SAE Technical Papers*, DOI: 10.4271/2015-26-0178
- Poort M., Cheng C., Richardson Dan., Schock H., 2014, Piston ring and groove side wear analysis for diesel engines, *ASME 2014 Internal Combustion Engine Division Fall Technical Conference*, DOI: 10.1115/ICEF2014-5670
- Poort M., Cheng C., Richardson D., Schock H., 2015, Piston ring and groove side wear analysis for diesel engines, *Journal of Engineering for Gas Turbines and Power*, 137(11) DOI: 10.1115/1.4030290
- Senatore A., Aleksendric D., 2014, Engine piston rings improvement through effective materials, advanced manufacturing methods and novel design shape, *Industrial Lubrication and Tribology*, 66(2), 298-305, DOI: 10.1108/ILT-01-2012-0010
- Su Y.Y., Wang B.J., 1998, The blow-by analysis of engine piston ring, *Journal of University of Shanghai For Science and Technology*, 20(4), 330-334.
- Tian T., 2002, Dynamic Behaviors of Piston Rings and Their Practical Impact—Part II: Oil Transport, Friction, and Wear of Ring/Liner Interface and the Effects of Piston and Ring Dynamics, *Proceedings of the Institution of Mechanical Engineers - Part J: Journal of Engineering Tribology*, 216(4), 229–247, DOI: 10.1243/135065002760199970
- Wolff A., 2014, Simulation Based Study of the System Piston-Ring-Cylinder of a Marine Two-Stroke Engine, *Tribology Transactions*, 57(4), 653-667, DOI: 10.1080/10402004.2014.895886
- Yu Z.Z., Dong G.N., Xie Y.B., 2005, A New Method to Reduce Frictional Power Consumption of Piston Ring in Internal Combustion Engine, *Lubrication Engineering* (6), 18-20,
- Zavos A.B., Nikolakopoulos P.G., 2015, Simulation of piston ring tribology with surface texturing for internal combustion engines, *Lubrication Science*, 27(3), 151-176, DOI: 10.1002/lis.1261