AFFECT OF PHENOMENON OF PISTON SEALING RINGS VIBRATIONS ON TIGHTNESS OF PRC UNIT

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Abstract

In this elaboration there is showed the affect of the phenomenon of piston rings vibrations on the tightness of the piston-piston rings-cylinder liner system (PRC). There has been executed an attempt to determine the mathematical model of the phenomenon of piston rings vibrations in the radial plane and a preliminary method for a model towards the cylinder axis that affect on the blow-by to the crankcase of the piston combustion engine.

Keywords: *blow-by, ring, piston ring oscillation, cylinder*

1. Introduction

Keeping the appropriate mating among the piston, piston rings and the cylinder liner bearing surface in a wide range of temperatures and the pressure alterations causes the load losses from the working space. The load losses in the form of the blow-by to the crankshaft of the piston combustion engine by leakages of the piston-piston rings-cylinder liner (PRC) cause a reduction of the engine output. Escalation of the cylinder leakage causes a considerable reduction of the engine working parameters. It is caused by the affect of the clearance among the PRC system elements and conditions of forming of the oil film among the piston rings and the cylinder liner bearing surface. As the research for providing the tightness of the entire PRC system showed, any kinds of the piston rings movements have the affect on the tightness of this system.

The phenomenon of the piston sealing rings vibrations is, in a certain extent of the engine rotational speed, the direct reason for the excessive blow-by. For example, the extreme of maximum of the blow-by takes place at the rotational speed about 1800rpm for the SW-680 engine. In the Fig. 1 there is shown the diagram of alterations of the blow-by for the 359 engine. There are two local extremes of the blow-by for this engine. The first takes place at the speed of 1500rpm approximately and the second at the speed of 2100rpm.

The piston ring makes various kinds of movements in the piston ring groove. Despite these movements, the piston ring has to adhere to the cylinder liner to provide the tightness. As the piston extorts the movement of piston rings towards the cylinder axis and radial movements of the piston rings, the piston ring has to have a possibility to move in various directions to provide the tightness.

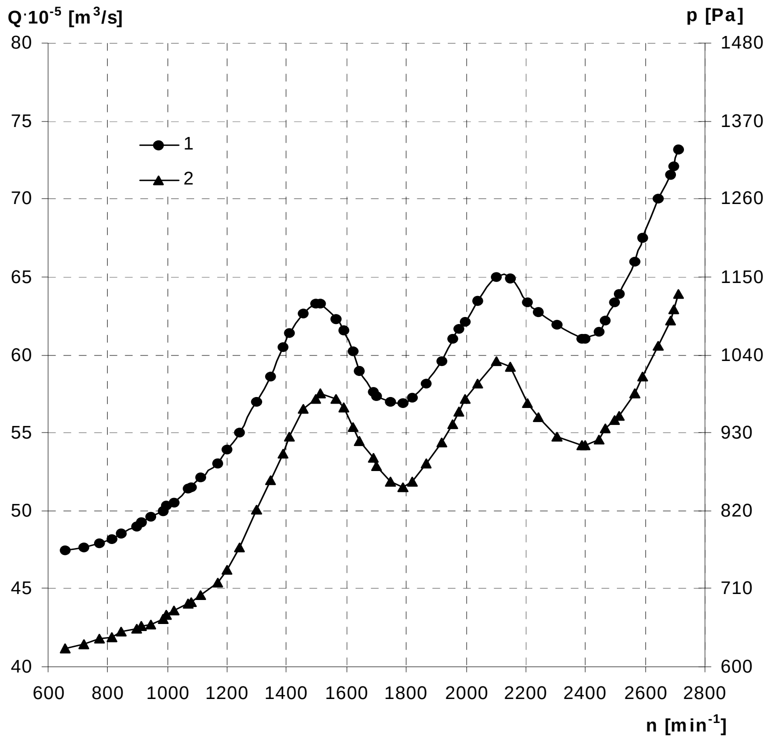

Fig 1. The characteristic of the blow-by in the function of the rotational speed of the 359 engine loaded to maximum, 1- the blow-by intensity, 2-stowing of gases in the crankcase [1]

2. Formulating a problem

According to A. Iskra, Professor, there are the most possible the vibrations with the amplitude towards the radial direction of the cylinder [4]. It can be assumed, that there are the small amplitudes of vibrations in the radial direction that result from the small harmonic amplitudes that extort vibrations, and a strong damping that causes moving of the piston ring in this direction.

However these vibrations do not cause such loss of tightness by the piston rings as the vibrations towards the cylinder axis. The piston sealing ring, detaching from the upper and lower edges of the piston ring groove, causes such a loss of the tightness as the PRC system would operate without the piston ring. This phenomenon leads to out-of-proportion escalation of the blow-by intensity to the engine crankcase. These vibrations are extorted by the reciprocating movement of the piston. The amplitude of vibrations can only be in the range of the piston groove size and is very strongly damped because of increased unit pressures in the areas of the piston-ring joint.

3. The model description

In general, the piston rings vibrations were considered in two causes: the radial vibrations and the axial vibrations (towards the cylinder axis). The radial movements of the piston ring are caused by change of the cylinder liner shape, on which the piston ring moves, with change of the oil film thickness and variable pressure force resulting from the blow-by and the pressure alterations in the cylinder working space. According to A. Iskra, Professor, in the case of the piston ring radial vibrations, the potential energy is bound almost entirely in the piston ring deflections. The radial deflections cause the longitudinal stresses, the value of which alters in a linear way.

The equation of the piston ring vibrations can be received from the condition of balance of the forces, as the non-trivial solution of the homogenous equation [4, 5]:

$$\frac{\partial^2}{\partial x^2}\left[E(x)J(x)\frac{\partial^2 y(x,t)}{\partial x^2}\right]+\rho(x)A(x)\frac{\partial^2 y(x,t)}{\partial t^2}=p(x,t) \qquad (1)$$

where:

x	-	the coordinate of situating of the intersection considered,
y(x,t)	-	the deflection of the intersection form the neutral state at the t moment,
E(x)	-	the Young's modulus,
J(x)	-	the axial moment of inertia of the intersection,
ρ(x)	-	the mass density of the piston ring material,
A(x)	-	the piston ring cross-sectional area,
P(x,t)	-	the external force that extorts vibrations.

Finally, after conversions in accordance with the literature [4, 5], the piston ring movement is described with the dependency:

$$y_n(x)=F_n\left[U\left(z_n\frac{x}{l}\right)-\frac{U(z_n)}{V(z_n)}V\left(z_n\frac{x}{l}\right)\right] \qquad (2)$$

where:

U, V - Krylov functions

Concerning the quoted theoretical basis of the vibration that cause deflections in the radial direction, they can be adapted for the real piston ring, what was confirmed in the literature [4].

The vibrations in the axial direction shall be considered separately for each working cycle of the engine, with the forces acting on the piston rings taken into consideration. However the biggest losses of the load are present during the working cycle of the engine. Therefore the initial stage of the research will refer to determination of possibilities of the piston rings vibrations in this cycle.

The author proposes to use the Rayleigh method of reduction of the vibrating system with parameters split to the energy-equipollent equivalent system with one degree of freedom. A relatively high accuracy can be achieved at calculations of the first natural frequency. Vibrations of the continuous system approximated with one degree of freedom is described by the function with variables separated:

$$u(P,t) = T(t)U(P) \qquad (3)$$

where u(P,t) is the function that describes displacements of the point P at the moment t, $P\in\Omega$, and Ω is the area of the technical system considered. A point of reduction that moves identically as the equipollent system, it means T(t), shall be chosen in this method. The point of reduction shall be chosen there, where the vibrations of the system are the most intensive, or there, where the concentrated mass is situated. This is the best to assume, for the considered case of the ring, the point situated 180^o from the piston-ring joint. The substitute mass m_z is determined from the formula:

$$m_z = \int_\Omega \rho(P)U^2(P)dv_\Omega + \sum_{k=1}^{m} m_k U^2(P_k) \qquad (4)$$

where ρ(P) is the mass density and m_k is the concentrated mass connected with the system in the P_k point.

Because the piston rings have a tendency to rotate in the piston ring grooves, in the expression defining the kinetic energy there must be the appropriate term that corresponds to the energy of rotation of the masses connected with the system. Comparing the potential energy of the continuous system, which energy is the form of squares of displacements and deformations for the linear system, with the potential energy of the substitute system, we can receive the formula that defines the substitute stiffness. From the equality of the work of external forces in the continual system on the virtual displacements:

$$\delta L = \int_{\Omega} f(P,t)\delta u(P,t)dv_{\Omega} + \sum_{k=1}^{m} F_k(t)\delta u(P_k,t) \tag{5}$$

and the work of the substitute force $P_z(t)$ on the corresponding displacement $\delta T(t)$:

$$(\delta L)_z = P_z(t)\delta T(t) \tag{6}$$

that takes the following equality into consideration:

$$\delta u(P,t) = U(P)\delta T(t) \tag{7}$$

the formula that defines the external force results:

$$P_z(t) = \int_{\Omega} f(P,t)U(P)dv_p + \sum_{k=1}^{m} F_k(t)U(P_k) \tag{8}$$

where: the f(P, t) is the density of the split force, the $F_k(t)$ is the concentrated force that acts on the system in the P_k point.

In Figure 2 is presented the equivalent system of vibrations.

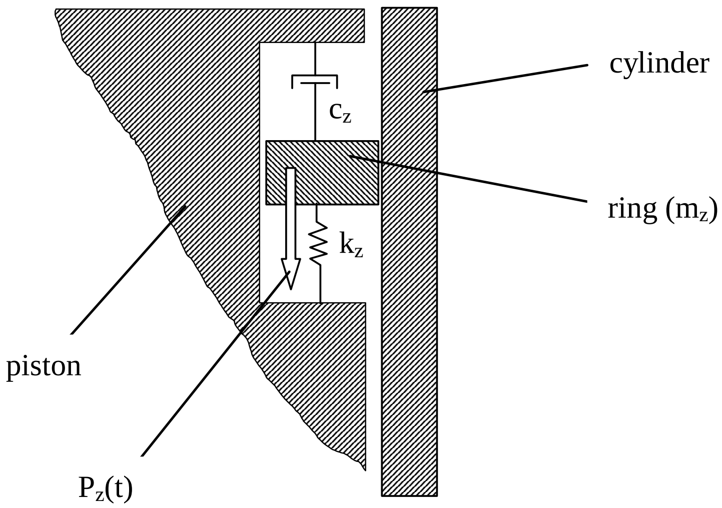

Fig. 2. Diagram of the equivalent system of vibrations (c_z – damping constant, k_z – substitute rigidity)

Comparing the energy lost by the continual system and by the substitute system we can calculate the damping constant c_z caused by the friction and presence of the oil and carbon deposits and lakes on the cylinder liner surface. The author realises that it is necessary to determine the concrete form of the formula that depends on the accepted rheological model. The curve that is the eigenfunction of the similar system can be used as the example. The assumption of the form of vibrations is equivalent with imposing of constraints on it.

4. Conclusions

Because, in accordance with the author's assumptions, the vibrations shall be considered separately for each working cycle of the engine, therefore it is necessary to collect, as much as possible, the data concerning changes of the pressure course in the cylinder, and thanks to it, alterations of the piston ring pressure to the lower edge of the piston ring groove and alterations in the thickness of the oil film, alterations of the speeds and accelerations and thanks to it, the piston ring inertia. Determination of the resultant of the force variable in time that acts on the ring, can lead to determine of the concrete equation of vibrations that describes the displacement of the piston ring towards the cylinder liner axis.

When determining the model of packing ring vibrations towards the cylinder axis according to Rayleigh's method, the external force extorting vibrations $P_z(t)$ should be set. This force is a resultant force from: in-time non-uniform piston ring inertial force resulting from piston to-and-fro motion with non-uniform acceleration, non-uniform gaseous force (non-uniform pressure of gases exerting an effect on piston ring), and non-uniform friction force, depending on oil film thickness. This method does not take into account changes in the angular position of piston rings, and consequently their effect on variation in the exciting force. The vibration damping is examined as a constant of silencer c_z, although as a matter of fact it is not constant, depending on the thickness of oil film, pressure pattern of piston ring, and changes in its position (in particular in relation to the cylinder liner). Nevertheless, for simplicity of calculations, it was assumed that vibration damping is constant and invariable in time.

References

[1] Abramek, K. F., *Charakterystyka przedmuchów w funkcji prędkości obrotowej silników SB-3.1, SW-680, 359. Zeszyt naukowy nr 4: Eksploatacja Silników Spalinowych,* Wydawnictwo Katedry Eksploatacji Pojazdów Samochodowych Politechniki Szczecińskiej, Szczecin 2001.

[2] Abramek, K. F., *Wpływ warunków pracy pierścieni tłokowych na ilość gazów przedostających się do skrzyni korbowej. Zeszyt naukowy nr 1: Eksploatacja Silników Spalinowych,* Wydawnictwo Katedry Eksploatacji Pojazdów Samochodowych Politechniki Szczecińskiej, Szczecin 2000.

[3] Abramek, K. F., *Wpływ zastosowania pierścieni wg Patentu PL 170524 na wielkość przedmuchów. Zeszyty Naukowe Wyższej Szkoły Morskiej nr 66,* Dział Wydawnictw Wyższej Szkoły Morskiej, Szczecin 2002.

[4] Iskra, A., *Wpływ drgań własnych pierścienia uszczelniającego na warunki pracy zespołu tłokowo-cylindrowego,* Journal of KONES Vol. 1, No 1, Warszawa-Lublin 1994.

[5] Kaliski, S., *Mechanika techniczna drgania i fale. Tom III,* Państwowe Wydawnictwo Naukowe, Warszawa 1986

[6] Smoczyński, M., Sygniewicz, J., *Analiza wpływu obciążeń mechanicznych na kątowe położenie tłoka względem tulei cylindrowej,* Interkonmot 1998.

[7] Smoczyński, M., Sygniewicz, J., *Przemieszczenia uszczelniającego pierścienia tłokowego w rowku pierścieniowym tłoka,* Journal of KONES Vol. 2, No 1, Warszawa-Poznań 1995.