Evaluation of Breakage of Keystone Rings with Two Types of Overall Side Angles

Kohei NAKASHIMA*[1], Yoshio MURAKAMI*[1] and Soichi ISHIHARA*[1]

*1 Department of Vehicle and Mechanical Engineering, Meijo University
1-501 Shiogamaguchi, Tempaku-ku, Nagoya 468-8502, JAPAN
nakasima@meijo-u.ac.jp

Abstract

Keystone rings with overall side angles of 6° and 15° are specified by the International Organization for Standardization (ISO), Japanese Industrial Standards (JIS) and SAE. This research investigated the effect of overall side angle on breakage of the keystone ring. We used simple models to calculate the side clearance between the ring and the ring groove, the mean colliding force between the ring and its groove, and the stress of ring edge under disc-spring deformation. Our results suggested that a keystone ring of a 6° angle (and thus a narrower side clearance) increased breakage resistance, as compared to a ring with a 15° angle (and a wider side clearance).

Keywords: piston ring, keystone ring, overall side angle, breakage, side clearance

1 Introduction

Heavy duty diesel engines use a keystone ring as the top ring. The keystone ring has a wedge-shaped cross section with tapered sides. When the keystone ring moves inside its piston ring groove in a radial direction, the clearance between the sides of the keystone ring and the ring groove decreases. The wedge shape of the moving ring pushes combustion residue (such as carbon) out of the ring groove while preventing the ring from sticking.

The essential dimensional features of the keystone rings are specified by the International Organization for Standardization (ISO), the Japanese Industrial Standards (JIS) and the SAE [1]-[4]. In all these standards, as well as existing heavy duty diesel engines, the keystone rings have an overall side angle (keystone angle) of either 6° or 15°. The clearance between the sides of the keystone ring and the ring groove depends on this keystone angle.

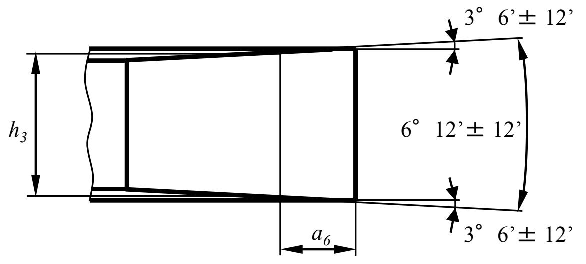
Future engine design might increase both engine output power and exhaust brake power, thus increasing the thermal load around the ring. This may increase the wear of the ring and the ring groove, and increase the force of the ring. Therefore, engine and piston ring designers should design the keystone ring taking into account not only wear resistance but also breakage resistance.

Several studies analyzed the behavior and lubricating oil consumption of keystone rings in diesel engines [5]-[9]. Another study evaluated ring breakage in a diesel engine [10]. However, we found no report examining ring breakage with different keystone angles.

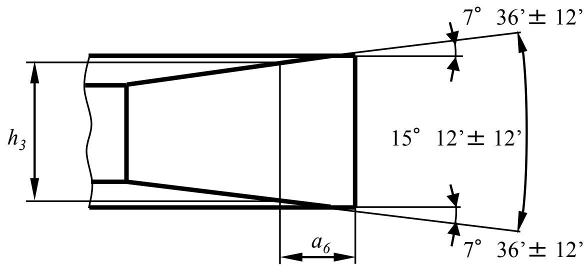
This study investigated the breakage resistance of the keystone ring with keystone angles of 6° and 15° using simple models: i.e., the effect of keystone angle on colliding force of the ring and the ring groove, and stress by ring deformation.

2 Keystone ring and its ring groove

Figure 1 shows the cross sections of keystone rings. The overall side angles (keystone angles) of the 6° and 15° rings are actually specified as 6°12’ ± 12’ and 15°12’ ± 12’, respectively. The width h_3 of the keystone ring is defined as the width dimension at the position of the reference distance a_6 inward from the ring periphery.



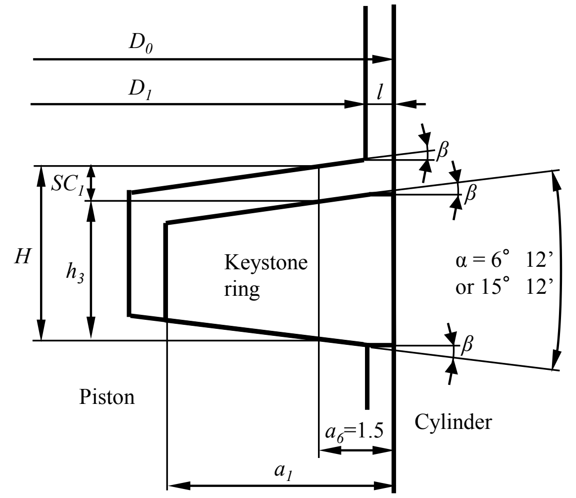
(i) Keystone ring 6°



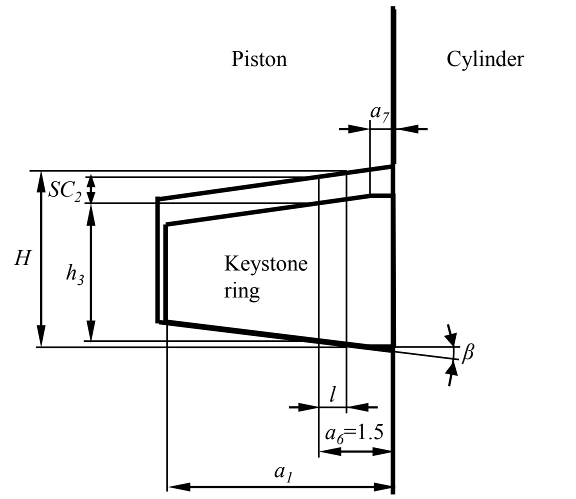
(ii) Keystone ring 15°

Fig. 1 Cross section of keystone rings

Figure 2 shows the positional relationship between the keystone ring and the piston ring groove, both when (a) the piston vertical axis is coincident with the cylinder axis, and when (b) the piston land contacts with the cylinder wall. **Figure 2** assumes that the keystone angle of the ring is equal to the overall side angle of the ring groove, and the piston top land diameter is equal to the piston second land. The radial tilt of keystone ring groove is zero; i.e., the ring groove axis is perpendicular to the piston vertical axis [11]. The reference distance a_6 inward from the ring periphery is 1.5mm. The width h_3 of the keystone ring is fixed, even when the keystone angle varies. In **Fig.2**, D_0 is the cylinder bore diameter, D_1 is the piston land diameter, a_1 is the ring thickness, and a_7 is the remaining flat thickness.



(a) Piston vertical axis is coincident with cylinder axis



(b) Piston land contacts with cylinder wall

Fig. 2 Positional relationship between keystone ring and piston ring groove

The side clearance between the keystone ring and the ring groove was calculated at the position of the reference distance a_6 inward from the ring periphery. Equations (1) and (2) indicate the side clearances, when the piston vertical axis is coincident with the cylinder axis, as shown in **Fig.2** (a), and when the piston land contacts with the cylinder wall, as shown in **Fig.2** (b), respectively.

$$SC_1 = H - h_3 \tag{1}$$

$$SC_2 = H - 2\ l\ \tan\beta - h_3 \geq 0 \tag{2}$$

where SC_1 is the side clearance [mm] when the piston vertical axis is coincident with the cylinder axis, SC_2 is the side clearance [mm] when the piston land contacts with the cylinder wall, H is the ring groove width [mm] at the position of the reference distance a_6 (1.5mm) inward from the ring periphery when the piston vertical axis is coincident with the cylinder axis, h_3 is the ring width [mm] at the position of the reference distance a_6 inward from the ring periphery, l is the radial clearance [mm] between the cylinder and the piston land, and β is one side angle [°] of the keystone ring and the ring groove.

When SC_2 was zero in eq. (2), we obtained eq. (3) for the minimum side clearance when the piston vertical axis is coincident with the cylinder axis.

$$SC_{1\ \min} = 2\ l\ \tan\beta \tag{3}$$

where $SC_{1\ min}$ is the minimum side clearance [mm] when the piston vertical axis is coincident with the cylinder axis.

In eq. (3), the minimum side clearance is proportional to the keystone angle. **Figure 3** shows the minimum side clearance with keystone angles of 6° (actually 6°12') and 15° (15°12'). **Figure 3** indicates that a keystone angle of 6° reduced the minimum side clearance by 59%, as compared to a keystone angle of 15°. We used these minimum side clearances in the following calculations.

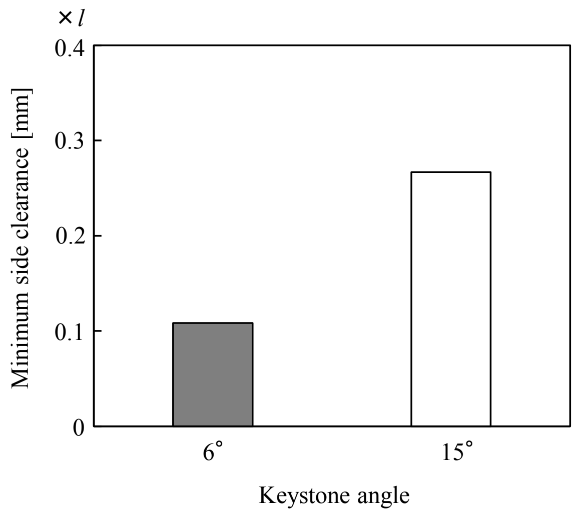


Fig. 3 Minimum side clearance between ring and ring groove

3 Colliding force of keystone ring to ring groove

According to Sasaki [10], during the exhaust brake operation, the top ring moves from the bottom side to the upper side in the top ring groove at the top dead center (TDC) of the exhaust stroke. Then, the mean colliding force F [N] of the top ring to the upper side of the ring groove is shown in eq. (4).

$$F = \frac{m(1+e)\sqrt{2\ a\ SC}}{\Delta t} \tag{4}$$

where m is the piston ring weight [kg], e is the coefficient of restitution, a is the acceleration of collision [m/s^2], SC is the side clearance between the ring and the ring groove [m], and Δt is the collision time [s].

First, we calculated ring weight utilizing the specifications for the keystone ring in **Table 1**. **Figure 4** shows the keystone ring weight with keystone angles of 6° and 15°. In the fixed ring width h_3 of 3mm, a keystone angle of 6° increased ring weight by 4%, as compared to a keystone angle of 15°.

Table 1 Specifications of keystone ring and engine

[mm]

Keystone ring	Keystone angle α (One side angle β)	6° 12' (3° 6') 15° 12' (7° 36')
	Width h_3	3.0
	Thickness a_1	4.5
	Remaining flat thickness a_7	0.5
	Material	Stainless steel
	Density ρ	7.7g/cm^3
Engine	Cylinder bore diameter D_0	114.0
	Stroke S	130.0
	Piston land diameter D_1	113.0
	Connecting rod length L	187.5

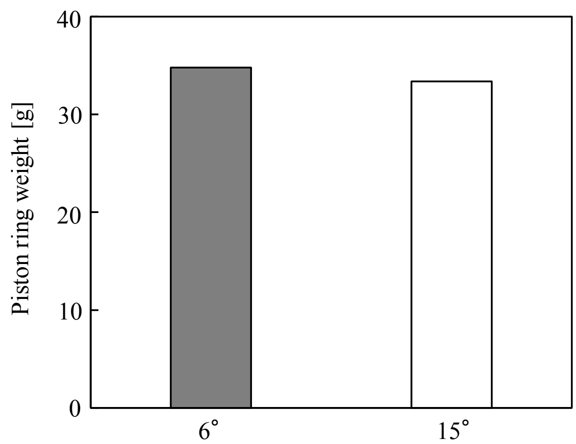


Fig. 4 Keystone ring weight with fixed ring width

Next, we calculated the mean colliding force of the top ring to the upper side of the ring groove, by substituting the side clearance from **Fig.3**, the ring weight from **Fig.4**, and the engine specifications in **Table 1** into eq. (4). **Figure 5** shows the mean colliding force with keystone angles of 6° and 15°, with a collision time of 53μs for 1° crank angle [10], a coefficient of restitution of 0.75 [12], at an engine speed of 2000rpm. It can be seen that a keystone angle of 6° reduced the mean colliding force by 34%, as compared to a keystone angle of 15°, because the mean colliding force is more affected by the side clearance than by the ring weight. Even a keystone angle of 6° with a wider side clearance of 0.100mm reduced the mean colliding force by 10%, as compared to a keystone angle of 15° with minimum side clearance, as shown in **Fig.6**. Again, the 6° keystone ring makes a ring groove width with narrower side clearance than a 15° ring. This narrower side clearance increased the breakage resistance of the keystone ring by the colliding force.

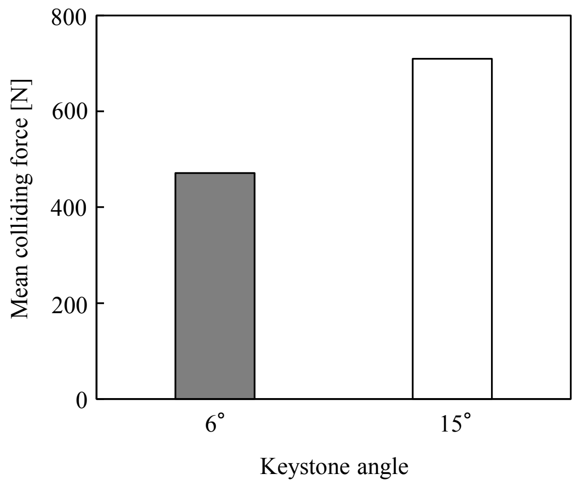


Fig. 5 Mean colliding force of ring to ring groove with minimum side clearance

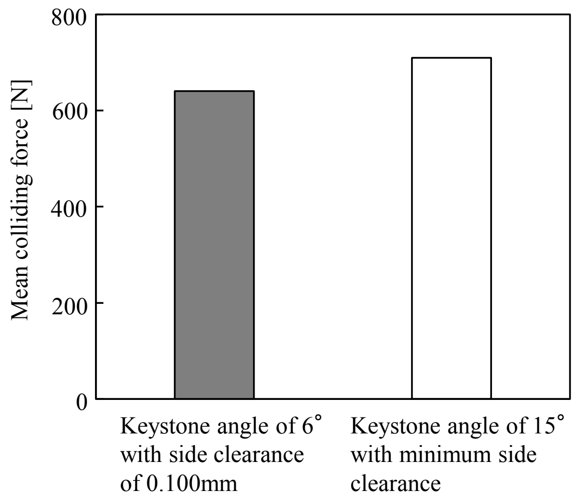


Fig. 6 Mean colliding force of ring to ring groove in keystone angle of 6° with side clearance of 0.100mm and 15° with minimum side clearance

4 Stress by deforming keystone ring

Sasaki's finite element calculation [10] shows that the ring deformation like disc spring dramatically increases stress, causing ring breakage. His tests of the combination of engine motoring and engine brake operations suggested the origin of the ring breakage is at the opposite side of the ring gap, at the edge of the periphery and the bottom side of the ring, which has the highest stress in the ring deformation like disc spring.

We investigated the stress in the tangential direction at the edge of the periphery and the bottom side of the ring using our calculated side clearance. We assumed that the force is applied to the rectangular ring without a gap, as shown in **Fig.7**. Using the formula of the stress of the disc spring as a reference [13], the stress σ_{t2} [Pa] in the tangential direction at the edge of the periphery and the bottom side of the ring can be indicated in eq. (5).

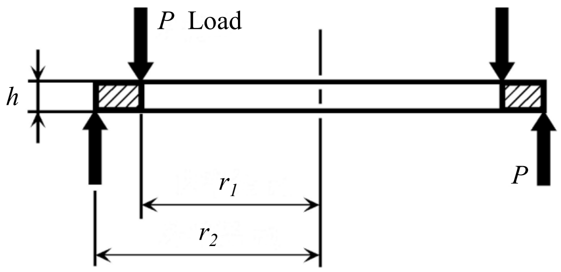


Fig. 7 Ring deformation like disc spring

$$\sigma_{t2} = K_{t2} \frac{Eh^2}{r_2^2} \tag{5}$$

where

$$K_{t2} = \frac{C\delta}{\{(1-\nu^2)h\}} \left(\frac{\delta}{2h} C_4 + C_5 \right)$$

$$C = \left(\frac{\lambda+1}{\lambda-1} - \frac{2}{\log_e \lambda} \right) \pi \left(\frac{\lambda}{\lambda-1} \right)^2$$

$$C_4 = \left(\frac{\lambda-1}{\log_e \lambda} - \lambda \right) \frac{6}{\lambda\pi \log_e \lambda}$$

$$C_5 = \frac{3(\lambda-1)}{\lambda\pi \log_e \lambda}$$

$$\lambda = \frac{r_2}{r_1}$$

where r_1 is the internal radius of the ring [mm], r_2 is the outer radius of the ring [mm], h is the ring width [mm], E is the modulus of elasticity [Pa], ν is the Poisson ratio, and δ is deflection [mm].

Finally we examined the stress in the tangential direction at the edge of the periphery and the bottom side of the ring in a rectangular ring made of stainless steel, with a ring width of 3mm and a ring thickness of 4.5mm (the same as the keystone ring in **Table 1**), a modulus of elasticity of 203GPa, and a Poisson ratio of 0.3. We calculated the stress in the tangential direction at the ring edge when the same ring displaced the distance of the calculated side clearance with keystone angles of 6° and 15° in **Fig.3**. We verified that the sum of the calculation result of the stress at the ring edge in eq. (5) and the closure stress was almost the same as the measured result of the maximum stress on the bottom side of the keystone ring at the periphery side and the opposite side of the ring gap. **Figure 8** shows the stress in the tangential direction at the ring edge without closure stress. We see that the side clearance calculated with a keystone angle of 6° reduced the stress in the tangential direction at the ring edge by 59%, as compared to that with a keystone angle of 15°. Not shown in a figure, even a side clearance of 0.100mm reduced the stress of the ring edge by 25%, as compared to the minimum side clearance calculated with a keystone angle of 15°. Again, the ring groove width with narrower side clearance can be designed with a keystone angle of 6° than with 15°, and this narrower side clearance increased the breakage resistance of the keystone ring by the deformation like disc spring.

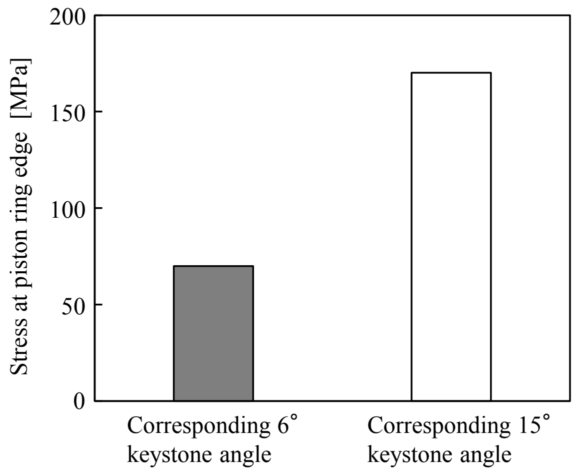


Fig. 8 Stress in tangential direction at edge of periphery and bottom side of ring by deforming like disc spring, without closure stress

5 Conclusion

We investigated the breakage resistance of keystone rings with keystone angles of 6° and 15°, as specified in the ISO, JIS and SAE Standards, using simple models to calculate the mean colliding force of the ring with the ring groove, and the stress of the ring edge from the deformation like disc spring. The results indicated that the side clearance between the ring and the ring groove affects the colliding force and the deformation stress in the ring. Compared with a keystone angle of 15°, a ring with an angle 6° enjoys a narrower side clearance and thus an advantage in resisting breakage.

In an actual engine, with increased operation period, the wear on the sides of the ring and the ring groove would tend to accumulate, increasing side clearance, and making the colliding force and the deformation stress in the ring even higher than our calculated results above.

There is a trade-off between keystone angles of 6° and 15°. Compared to a keystone angle of 15°, a keystone angle of 6° increases both ring breakage resistance and ring sticking. Therefore, engine and ring designers should take into account not only ring breakage resistance bur also ring sticking resistance.

Simple models evaluating ring breakage can help design the clearance between the sides of the keystone ring and the ring groove at a fixed keystone angle. Such evaluation reduces the time needed to model and analyze the ring in FEM.

References

[1] ISO Handbook Piston Rings and Pins, ISO Standard 6624-1, Internal Combustion Engines –Piston Rings- Part 1: Keystone Rings Made of Cast Iron, ISO, (2006), pp.1-25.

[2] ISO Handbook Piston Rings and Pins, ISO Standard 6624-3, Internal Combustion Engines –Piston Rings- Part 3: Keystone Rings Made of Steel, ISO, (2006), pp.1-23.

[3] JIS Handbook Automobiles II, JIS Standard B8302-9, Internal Combustion Engines –Small Diameter Piston Rings- Part 9: Keystone Rings, JIS, (2011), pp.107-117.

[4] 2005 SAE Handbook, SAE Standard J2000, Internal Combustion Engines –Piston Rings-Keystone Rings-, SAE, (2005), pp.26.111-26.120.

[5] Hayes, B., "Piston Rings for Diesel Engines –Effect of Keystone Angle on Oil Consumption", SAE Paper 710815, (1971).

[6] Bishop, G. R. and Leavitt, A. H., "Performance Simulation of a Diesel Piston and Ring System", SAE Paper 750768, SAE Transactions, Vol. 84, Section 3, (1975), pp.1834-1842.

[7] Furuhama, S., Hiruma, M. and Tsuzita M., "Piston Ring Motion and Its Influence on Engine Tribology, SAE Paper 790860, SAE Transactions, Vol. 88, Section 4, (1979), pp.2929-2941.

[8] Mihara, K. and Inoue H., "Effect of Piston Top Ring Design on Oil Consumption", SAE Paper 950937, SAE Transactions, Journal of Engines, Vol. 104, Section 3, (1995), pp.1560-1567.

[9] Herbst, H. M. and Priebsch, H. H., "Simulation of Piston Ring Dynamics and Their Effect on Oil Consumption", SAE Paper 2000-01-0919, SAE Transactions, Journal of Engines, Vol. 109, Section 3, (2000), pp.862-873.

[10] Sasaki, I., "A Study on Mechanism and Technical Measures of Piston Ring Breakage", Isuzu Technical Journal, No. 95, (1996), pp.61-66.

[11] 2005 SAE Handbook, SAE Standard J2275, Internal Combustion Engines –Piston Ring-Grooves-, SAE, (2005), pp.26.195-26.203.

[12] Goldsmith, W., Impact The Theory and Physical Behaviour of Colliding Solids, Dover Publication Inc., (2001), pp.258.

[13] Japan Society for Spring Research, Spring, Maruzen Publishing Co., Ltd. (1982), pp.283-287.

Received on December 12, 2013
Accepted on January 22, 2014