Effects of bearing clearance and atmospheric temperature on performance of pinion bearings of railway vehicles

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Abstract
In Japan, most of the gear units for transmitting driving force in railway vehicles employ one-stage speed reducers. The gear unit is composed of a pinion connected to a traction motor via a coupling mechanism, a gear installed on an axle, and a gear case that covers them. In their rotating parts, tapered roller bearings are mainly used, and are lubricated by gear oil splashed by the rotating gear. In order to prevent seizure of the bearings and to improve the reliability of the gear units, it is important to appropriately adjust the bearing clearance. The bearing clearance can change from its initial value during the travel of the vehicles due to an atmospheric temperature and its initial value when assembling the gear unit, and can affect the performance of bearings. In this research, an actual gear unit was subjected to bench rotation tests under various bearing clearances and various atmospheric temperatures. The bearing temperature and torque were measured, and changes in bearing clearance were estimated. As a result, it is found that the bearing clearance decreases immediately after the rotation starts, and this tendency becomes more remarkable as the initial bearing clearance is smaller and as the atmospheric temperature is lower.

Keywords: Railway vehicle, Gear unit, Pinion, Machine element, Tapered roller bearing, Bearing clearance

1. Introduction
In Japan, most of the gear units for transmitting driving force in railway vehicles (electric cars) employ one-stage speed reducers. The gear unit is composed of a pinion connected to a traction motor via a coupling mechanism, a gear installed on an axle, and a gear case that covers them (Kimura et al., 2013). In their rotating parts, tapered roller bearings are used in Japan (Ezaki et al., 2012). Two tapered roller bearings are installed to the pinion shaft, which are referred to as pinion bearings, to support the pinion shaft with respect to the gear case. Similarly, two tapered roller bearings are installed to the axle, which are referred to as gear bearings, to support the gear case. These bearings are lubricated by gear oil splashed by the gear. Among these bearings, the rotational speed of the pinion bearings is higher than that of the gear bearings, so the sliding speeds at the contact surface between the roller end and inner ring flange of the pinion bearings are higher than that of the gear bearings. In the pinion bearings, therefore, seizure may occur by metal-to-metal contact from inadequate lubrication between the roller end and the inner ring flange (SKF, 2017). In order to prevent seizure of the pinion bearings and to improve the reliability of the gear units, it is important to appropriately manage the clearance of the bearings in addition to the management of the lubricating oil. Although this bearing clearance is usually adjusted in a specified range when assembling the gear unit, it can change from its initial value due to temperature change of peripheral parts of the bearings during the travel of the vehicles (Johns, 1998) (Hayashi et al, 2001). Furthermore, the tendency and amount of this change of the bearing clearance can vary depending on the initial bearing clearance and the atmospheric temperature.

The authors have already reported the behavior of the bearing temperature and torque under various rotational conditions, lubricating conditions, and bearing clearances by performing basic rotational tests of the pinion bearings (Takahashi et al., 2019). That is, it was found that the torque of the bearings and the heating value from the bearings increased as the temperature of the gear oil decreased, the flow rate of the gear oil increased, and the bearing clearance

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decreased. On the other hand, the above research was conducted under the circulating lubrication with the gear oil, so there are many unverified items about the performance and the behavior of the pinion bearings mounted on the actual gear unit under the splash lubrication by the gear. Therefore, in this study, the effects of the initial bearing clearance and atmospheric temperature on the bearing temperature and torque were investigated by the bench rotation tests of the actual gear unit. In addition, the behavior of the bearing clearance during the operation of the gear unit was estimated from the temperature change of the peripheral parts of the bearings.

2. Test apparatus

2.1 Test rig of gear unit

Figure 1 shows the test rig of the gear unit used in this study. This rotation testing machine has the structure in which a pinion shaft is rotated by a motor through a flexible flanged shaft coupling, using an actual gear unit as a specimen. The gear unit is attached to an axle and supported by two bearings at the both ends of the axle and a link mechanism to suspend gear case. The rated output of the motor is 22 kW and the rated rotational speed is 6000 min\(^{-1}\). Blowers are installed in the vicinity of the gear unit and the support bearings in order to cool them which generate heat during rotation tests.

![Fig. 1 Test rig of the gear unit](image)

The specifications of the used gear unit are shown in Table 1, and a schematic view of the gear unit is shown in fig. 2. The reduction gear mechanism is composed of helical gears with a left-hand pinion and a right-hand gear. The pinion shaft is installed with pinion bearings on both sides of the pinion, which supports the pinion shaft with respect to the gear case. The axle is installed with gear bearings on both sides of the gear, which supports the gear case with respect to the axle. The inner rings of the pinion bearings are fitted to the pinion shaft, and the inner rings of the gear bearings are fitted to the axle. On the other hand, the outer rings of these bearings are fitted to the housings which are fastened to the gear case. The gear case is made of aluminum alloy, and the gear oil is stored inside.

![Fig. 2 Schematic view of testing gear unit](image)

| Table 1 Specification of testing gear unit |
|------------------------------------------|
| **Gear ratio**                          | 2.79 |
| **Material of gear case**               | Aluminum alloy AC4CH (JIS H 5202) |
| **Material of housings**               | Steel S25C (JIS G 4051) |
| **Material of pinion and pinion shaft** | Steel SNCM420 (JIS G 4053) |
| **Material of gear**                   | Steel S40C (JIS G 4051) |
| **Dimensions of pinion bearings**       | Outside : ø150 mm, Bore : ø70 mm, Width : 38 mm |
| **Dimensions of gear bearings**         | Outside : ø280 mm, Bore : ø195 mm, Width : 58 mm |
2.2 Measuring items

While the gear unit is operating, the temperature and the vibration acceleration of the gear unit, and the rotational speed and the torque of the pinion shaft can be measured (Fig. 3). The temperatures were measured at 7 locations in total, including the outer diameter surface of the pinion bearings and the gear bearings, the surface of the gear case, the gear oil, and the atmosphere. These were all measured by K-type thermocouples. The vibration accelerations were measured at the pinion bearings. These were measured by piezoelectric accelerometers. The rotational speed of the pinion shaft was measured by a photoelectric rotation detector. The torque of the pinion shaft was obtained by a calculation from the current value of the motor. Furthermore, the temperature at the end of the pinion shaft was measured with a small temperature data logger (Fig. 4).

![Fig. 3 Measuring items. The temperature of each part of the gear unit indicated by the red arrow was measured.](image)

2.3 Test bearing

In this study, we focused on the pinion bearing (hereinafter referred to as “bearing”). As shown in Table 1, the bearing is a tapered roller bearing with an outside diameter of 150 mm, a bore diameter of 70 mm, and a width of 38 mm. The outer ring, inner ring, and rollers are made of high-carbon chromium bearing steel (JIS SUJ2). The cage is made of low-carbon steel. The bearing clearance (combination clearance in the axial direction of the two bearings) can be changed by inserting shims of various thicknesses between the gear case and the pinion housing (Fig. 5).

![Fig. 5 Method of changing the bearing clearance. The bearing clearance (combination clearance in the axial direction of the two bearings) can be changed by inserting shims of various thicknesses between the gear case and the pinion housing.](image)
3. Test method

The rotation tests were performed under the constant conditions shown in Table 2. Figure 6 shows the rotation pattern of the pinion shaft. The rotational direction of the pinion shaft was set to a clockwise direction in a view from the motor side (the direction indicated by the arrow in Fig. 2). After the start of rotation, the rotational speed of the pinion shaft reaches to 6000 min\(^{-1}\) in 315 s. This rotational speed corresponds to a vehicle speed of approximately 320 km/h. The amount of gear oil is 2.95 L. During the test, the gear case and the support bearings were air-cooled when the rotational speed of the pinion shaft was 200 min\(^{-1}\) or higher.

Under the constant conditions shown in Table 2, the axial clearance of the pinion bearings and atmospheric temperature were variously changed. They are shown in Table 3 and Fig. 7. Since the gear case and the pinion shaft are made of different materials shown in Table 1, the bearing clearance varies depending on the temperature of the gear unit. In this paper, the bearing clearances are shown as values obtained by converting the measured values at assembling to the values at 20 °C.

| Test number | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 |
|-------------|---|---|---|---|---|---|---|---|---|----|----|----|----|----|----|
| Axial clearance of pinion bearings [mm] | 0.06 | 0.06 | 0.06 | 0.07 | 0.09 | 0.11 | 0.11 | 0.12 | 0.12 | 0.14 | 0.14 | 0.24 | 0.26 | 0.26 | 0.31 |
| Atmospheric temperature [°C] | 22.6 | 25.1 | 27.4 | 8.7 | 28.5 | 19.3 | 24.8 | 9.1 | 13.7 | 7.5 | 16.4 | 6.6 | 24.3 | 29.1 | 20.0 |

4. Temperature change of each part of the gear unit and torque of pinion shaft

4.1 Influence of the bearing clearance

Figure 8 shows the temperature change of each part of the gear unit and torque of the pinion shaft from the start of rotation to 3600 s for test numbers 1, 6, and 15 as combinations with different bearing clearances under conditions where the atmospheric temperature is close to 20 °C. Regardless of the test conditions, the temperature measured at the end of the pinion shaft was the highest among the temperatures of each part, and increased rapidly from the start of rotation to about 600 s as the bearing clearance was smaller. In particular, when the bearing clearance is 0.06 mm, it
reached about 80 °C in 600 s after the start of rotation, and after the rising stopped a while, it rose slowly again. This is thought to be due to an increase in frictional resistance and rolling viscous resistance inside the bearing. The increase in these resistance is resulted from wide range of the load zone and large number of the rollers which loads were applied by relatively small clearance. Although it is not as significant as the trend of temperature at the end of the pinion shaft described above, the temperature of the outer ring of the bearing increases rapidly after the start of rotation, as the bearing clearance is smaller for both the motor side and counter motor side bearings. The tendency described above is further weakened at the temperature of the gear case surface and gear oil. The strength of these trends may be attributed to the main heat source in the tests being the pinion bearings, and the temperature rise of the inner rings being the greatest (Otaki and Haruyama, 2018) (Xianwen et al, 2019). The torque of the pinion shaft reaches its maximum value immediately after the start of rotation, then decreases, and becomes almost constant after about 1000 s. The maximum value of the torque of the pinion shaft increases as the bearing clearance decreases. Although the torque of the pinion shaft includes torques other than the torque of the pinion bearings (for example, the torque of the gear bearings and the stirring resistance of gear oil), it has been confirmed that the torque of the shaft increases as the bearing clearance decreases in the rotation test performed by the authors with only the pinion bearings (Takahashi et al, 2019). For this reason, it is considered that the bearing clearance has the most influence on the above torque difference.

### 4.2 Influence of the atmospheric temperature

Figure 9 shows the temperature change of each part of the gear unit and torque of pinion shaft from the start of rotation to 3600 s for test numbers 6, 7, 8, and 9 as combinations with different atmospheric temperatures under conditions where the bearing clearance is close to 0.11 mm. Regardless of the conditions, the temperature measured at the end of the pinion shaft was the highest among the temperatures of each part, and increased rapidly from the start of rotation to about 600 s as the bearing clearance was smaller. The maximum value of the torque of the pinion shaft increases as the bearing clearance decreases.
bearing, resulted from relatively low atmospheric temperature. In the rotation test performed by the authors with only the pinion bearings, it has been confirmed that the heating value from the bearings increases with a decrease in the temperature of the gear oil and an increase in the viscosity of the gear oil, which can explain the trend of the temperature rise above (Takahashi et al, 2019). The trends of temperature changes of the outer rings of bearings, gear case, and gear oil are the same as those described in Section 4.1. The torque of the pinion shaft reaches its maximum value immediately after the start of rotation, then decreases, and becomes almost constant after about 1000 s. The maximum value of the torque of the pinion shaft increases as the atmospheric temperature decreases. This is because the viscosity of gear oil increases as the atmospheric temperature decreases, and the rolling viscous resistance inside the bearing and the stirring resistance of gear oil increase.

4.3 Temperature at the end of the pinion shaft and torque of the pinion shaft

As described in Section 4.1 and Section 4.2, the difference in the test conditions most clearly affects the temperature at the end of the pinion shaft. In particular, it is found that the temperature rise from the start of rotation to about 600 s varies greatly depending on the test conditions. Therefore, from the results of each test shown in Table 3, the maximum rate of increase in the temperature at the end of the pinion shaft (temperature rise per 10 s) is summarized in a contour diagram with respect to the initial bearing clearance $EP_{20}$ (Axial clearance of pinion bearings shown in Table 3) and atmospheric temperature, and shown in Fig. 10. The white area in the diagram is the area for which data has not been obtained because the rotation test is not performed. The maximum rate of increase in the temperature at the end of the pinion shaft tends to increase as $EP_{20}$ decreases and the atmospheric temperature decreases. In particular, when $EP_{20}$ is smaller than 0.14 mm, the maximum rate of increase in the temperature at the end of the pinion shaft increases rapidly as $EP_{20}$ decreases.

It is found that the difference in the test conditions also affects the torque of the pinion shaft. Therefore, from the results of each test shown in Table 3, the maximum value of the torque of the pinion shaft is summarized in a contour diagram with respect to the initial bearing clearance $EP_{20}$ and atmospheric temperature, and shown in Fig. 11. The
Fig. 10  Maximum rate of increase in the temperature at the end of the pinion shaft. The maximum rate of increase in the temperature at the end of the pinion shaft tends to increase as $EP_{20}$ decreases and the atmospheric temperature decreases. In particular, when $EP_{20}$ is smaller than 0.14 mm, the maximum rate of increase in the temperature at the end of the pinion shaft increases rapidly as $EP_{20}$ decreases.

Fig. 11  Maximum value of the torque of the pinion shaft. The maximum value of the torque of the pinion shaft tends to increase as $EP_{20}$ decreases and the atmospheric temperature decreases.

5. Study on change of bearing clearance due to temperature change

As shown in Chapter 4, the temperature of each part of the gear unit greatly changes immediately after the start of rotation. For this reason, the bearing clearance changes due to the thermal expansion of each part of the gear unit. There is a concern that a seizure of the bearing may occur when the bearing clearance decreases to 0 mm. Therefore, the change in the bearing clearance is estimated from the temperature change in each part of the gear unit.

5.1 Method of calculation of the bearing clearance

Simplifying the structure around the pinion bearings of the gear unit shown in Fig. 2 (Fig. 12), we calculate the bearing clearance $EP$ during the operation of the gear unit, using Eq. (1) that takes into account only the axial thermal expansion of each component.

$$EP = EP_{20} + (t_c - 20) \cdot \alpha_c \cdot L_c - (t_h - 20) \cdot \alpha_h \cdot (L_c - L_h) - (t_s - 20) \cdot \alpha_s \cdot L_h$$  \hspace{1cm} (1)$$

where $EP_{20}$ is the initial bearing clearance shown in Table 3, $t_c$, $t_h$, and $t_s$ are the temperatures of the gear case surface, the pinion bearing housing, and the pinion shaft. $\alpha_c$, $\alpha_h$, and $\alpha_s$ are the coefficients of thermal expansion of the gear.
case, the pinion bearing housing, and the pinion shaft. \( L_c \) is the distance between the housing fastening surfaces of the gear case, and \( L_b \) is the distance between the bearing centers. These values are shown in Table 4. Since the temperatures of the pinion bearing housings are not directly measured, the temperature of the outer ring of the bearing near the housing (average value of the temperatures of the two bearings) is used, and the temperature of the pinion shaft is used for the temperature of the shaft end.

Table 4 Parameters used to calculate the bearing clearance

| Parameter | Value |
|-----------|-------|
| \( \alpha_c \) | \( 2.2 \times 10^{-5} \) K\(^{-1} \) |
| \( \alpha_h \) | \( 1.2 \times 10^{-5} \) K\(^{-1} \) |
| \( \alpha_s \) | \( 1.2 \times 10^{-5} \) K\(^{-1} \) |
| \( L_c \) | 178 mm |
| \( L_b \) | 128 mm |

5.2 Examination of the deformation of the gear case

Equation (1) assumes that the gear case, housings, and pinion shaft are thermally expanded in all directions uniformly. Of these, the pinion shaft and the inner rings of the bearings are both iron based materials, and it is considered that they do not restrain each other’s deformation, so the above assumption is considered to hold for the pinion shaft. On the other hand, as shown in Fig. 5, since the gear case made of aluminum alloy is fastened with the housing made of carbon steel by bolts, it is possible that the above assumption may not hold because the mutual deformation is restrained. Therefore, in order to verify whether the axial thermal expansion of the gear case conforms to Eq. (1), the dimensional change of the gear case was measured by a rotation test of the gear unit, and the deformation was analyzed by the finite element method.

5.2.1 Measurement of the dimensional change of the gear case

The axial dimensional change (expansion) of the gear case was measured with two laser displacement meters. As shown in Fig. 13 the object to be measured was the dimension between two housings of the pinion bearings. The test conditions are the same as in Chapter 3 except for the items shown in Table 5. In addition, the theoretical value of the axial expansion of the gear case \( \Delta L \) is calculated by Eq. (2), and is compared with the above measured value.

\[
\Delta L = (t_c - t_{c0}) \cdot \alpha_c \cdot L_c + (t_h - t_{h0}) \cdot \alpha_h \cdot (L_b - L_h) \quad (2)
\]
where $t_{c0}$ and $t_{h0}$ are the temperature of the gear case surface and the housing at the start of rotation, and $L_h$ is the distance between the housing surfaces (196 mm) shown in Fig. 12.

Figure 14 shows the measured and theoretical values of the axial expansion of the gear case from the start of rotation to 2400s, along with the change in the temperature of the gear case. The measured value (gray line) varies greatly due to the influence of vibration caused by the operation of the gear unit, so the values obtained by averaging adjacent data for 60 s (blue line) are also shown. The measured values for the axial expansion of the gear case almost agreed with the theoretical values obtained by Eq. (2).

![Fig. 14 Dimensional change of the gear case. The measured values for the axial expansion of the gear case almost agreed with the theoretical values.](image1)

5.2.2 Finite element analysis of the deformation of the gear case

Figure 15 shows a cross section of the peripheral portion of the pinion bearings to be analyzed, where some parts are simplified. The analysis model is shown in Fig. 16, and the analysis conditions are shown in Table 6. The analysis model is created based on the cross sectional shape (red line flame) in Fig. 15. The outer ring of the bearing is excluded from the model because it is smaller than the other parts. Although the gear case is not strictly cylindrical, it modeled as a cylindrical shape to simplify the calculation. All parts are assumed to be elastic, and Young’s modulus, Poisson’s ratio, and the coefficient of thermal expansion are set for the housing and the gear case according to those materials. The temperature of each part is assumed to change uniformly without any distribution, and the analysis is performed for a case where the temperature changes from the initial temperature by +10 ~ +60 K.

Figure 17 shows the deformation analysis results when the temperature changes +50 K from the initial temperature. The contour diagram shows the amount of axial displacement compared to the initial position with different colors, and the deformation is displayed at a higher magnification. The gear case and the housing expand from the center to the axial direction due to thermal expansion, but the housing with a relatively low coefficient of thermal expansion constrains the deformation of the gear case. As a result, the side surface of the gear case is pulled in the radial direction.
Table 6 Analysis conditions

| Analysis code | Analysis code | NX Nastran V8.5 |
|---------------|---------------|----------------|
| Element breakdown | Hexahedral elements | Some with pentahedral elements |
| Young’s modulus | 70 GPa |
| Poisson’s ratio | 0.3 |
| Coefficient of thermal expansion | $2.2 \times 10^{-5} \text{K}^{-1}$ |
| Young’s modulus | 210 GPa |
| Poisson’s ratio | 0.3 |
| Coefficient of thermal expansion | $1.2 \times 10^{-5} \text{K}^{-1}$ |
| Distribution | Uniform distribution |
| Amount of change | +10, +20, +30, +40, +50, +60 K |

Fig. 17 Deformation of the gear case and the housing. The gear case and the housing expand from the center to the axial direction due to thermal expansion, but the housing with a relatively low coefficient of thermal expansion constrains the deformation of the gear case.

and bent in the gear case inner direction, but these have little effect on the axial deformation of the gear case. Figure 18 shows the dimensions between two housings of the pinion bearings obtained by the finite element analysis. The measured and the theoretical values in Fig. 18 are the same as those shown in Fig. 14. The values obtained by the finite element analysis roughly agree with the measured and theoretical values.

Fig. 18 Dimensional change of the gear case obtained by finite element analysis. The values of the axial expansion of the gear case obtained by the finite element analysis roughly agree with the measured and theoretical values.

5.3 Calculation result of bearing clearance

Figure 19 and 20 show the bearing clearances calculated by Eq. (1) for the test results shown in Fig. 8 and 9. After about 600 s from the start of rotation, the bearing clearance increases due to the temperature rise in each part of the gear case, mainly due to the difference in the coefficient of thermal expansion between the gear case and the pinion shaft. However, for about 600 s from the start of rotation, the bearing clearance decreases with rotation, and this tendency is more pronounced as the initial bearing clearance is smaller (Fig. 8) and the atmospheric temperature is lower (Fig. 9). This is because, in the initial stage after the start of rotation, the temperature of the pinion shaft $t_s$ accompanying the rotation of the bearing rises faster than the temperature of the gear case $t_c$, and the thermal expansion of the pinion shaft becomes larger than the thermal expansion of the gear case. The trend that the bearing clearance once decreases after the start of the rotation and then increases agrees with the trend confirmed by the rotation test performed by the authors with only the pinion bearings (Takahashi et al, 2019). As described in Chapter 4, when the initial bearing clearance was
the smallest and the atmospheric temperature was the lowest, the temperature at the end of the pinion shaft increased rapidly in the initial stage after the start of rotation, and after the rising stopped a while, it rose slowly again. It is thought that this is because the bearing clearance (EP value) started to increase and the heat generation was suppressed after the heat generation of the bearing increased due to the decrease in the bearing clearance.

As shown in Fig. 19 and 20, the bearing clearance varies during the rotation of the pinion shaft. There is a concern that the temperature of the bearing can rise rapidly when the bearing clearance decreases the most, resulting in seizure of the bearing. Therefore, from the results of each test shown in Table 3, the minimum values of the actual bearing clearances $EP_{\text{min}}$ obtained by Eq. (1) is summarized in a contour diagram with respect to the initial bearing clearance $EP_{20}$ and atmospheric temperature, and shown in Fig. 21. Figure 22 shows the value obtained by subtracting $EP_{\text{min}}$ from the initial bearing clearance $EP_{20}$, that is, the maximum amount of decrease in bearing clearance during the rotation.

As shown in Table 3, the maximum amount of decrease in bearing clearance tends to increase as $EP_{20}$ is smaller and the atmospheric temperature is lower.
The white area in the diagram is the area for which data has not been obtained because the rotation test is not performed.

When compared at the same atmospheric temperature, the smaller $EP_{20}$ becomes, the smaller $EP_{min}$ is. Further, when compared with the same $EP_{20}$, the lower the atmospheric temperature is, the smaller the $EP_{min}$ is. Along with that, the maximum amount of decrease in bearing clearance tends to increase as $EP_{20}$ is smaller and the atmospheric temperature is lower. In the range in which the test was performed, the maximum amount of decrease in bearing clearance was relatively large value of 0.05 mm or more under the conditions that $EP_{20}$ was 0.10 mm or less and the atmospheric temperature was 10 °C or less.

6. Conclusions

The findings obtained through this study are summarized below.

(1) The temperature measured at the end of the pinion shaft is the highest among the temperatures of each part, and increased rapidly from the start of rotation to about 600 s as the initial bearing clearance is smaller and the atmospheric temperature more decreases. Therefore, the maximum rate of increase in the temperature at the end of the pinion shaft (temperature rise per 10 s) also increases as the initial bearing clearance is smaller and the atmospheric temperature more decreases.

(2) The maximum value of the torque of the pinion shaft tends to increase as the initial bearing clearance decreases and the atmospheric temperature more decreases.

(3) The influence of restraint on the axial thermal expansion of the gear case by the housing is extremely small, and the actual deformation in the axial direction of the gear case is almost the same as that obtained from the calculation of thermal expansion. Therefore, the actual bearing clearance during operation of the gear unit can be easily calculated using an equation that considers only the thermal expansion in the axial direction of each part.

(4) The bearing clearance during the operation of the gear unit is calculated from the temperature change of each part. As a result, it is found that the bearing clearance decreases with rotation for about 600 s from the start of rotation, and this tendency is more pronounced as the initial bearing clearance is smaller and the atmospheric temperature is lower.

(5) The minimum value of the bearing clearance during the operation of the gear unit becomes smaller as the initial bearing clearance decreases and the atmospheric temperature is lower.

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