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that $2kf_b=kf_r\{1-(D_a/d_m)^2\cos^2\alpha\}d_m/D_a$,

the rolling bearing being characterized in that vibrations at frequencies of mZf_i, $mZf_i \pm f_r$, nZf_c, $2kf_b$ and $2kf_b \pm f_c$ are generated in the rolling bearing due to the circumferential undulations of (mZ) waves and ($mZ \pm 1$) waves existing on the surface of the first raceway, to the circumferential undulations of (nZ) waves and ($nZ \pm 1$) waves existing on the surface of the second raceway and to the undulation of (2k) waves existing on the rolling surface of the respective rolling bodies, and

that the formulas of $(mZf_i) \neq jf_r$, $(mZf_i \pm f_r) \neq jf_r$, $(nZf_c) \neq jf_r$, $(2kf_b) \neq jf_r$ and $(2kfb \pm f_c) \neq jf_r$ are satisfied for all of n,m, k and j with respect to the frequencies.

~~According to a second aspect of the present invention there is provided a rolling bearing comprising a first race having a first raceway, a second race provided around the first race and having a second raceway, Z rolling bodies having a rolling surface and rollably disposed between the first raceway and the second raceway, and a retainer for holding the rolling bodies,~~

~~provided that under grease lubrication, the first race rotates at the frequency of f_r(Hz) while the rolling bodies held in the retainer rotate at the frequency of f_b(Hz) and~~

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~~revolve at the frequency of f_c(Hz), that n, m, k and j are~~ each a positive integer up to 100, respectively,

that D_a is the diameter of the rolling bodies,

that d_m is the pitch circle diameter of the rolling bodies,

that α is the contact angle between the rolling bodies and the first and second raceways,

that $f_i=f_r-f_c$(Hz),

that $nZf_c=(1/2)nf_r\{1-(D_a/d_m)\cos\alpha\}Z$,

that $mZf_i=(1/2)mf_r\{1+(D_a/d_m)\cos\alpha\}Z$, and

that $2kf_b=kf_r\{1-(D_a/d_m)^2\cos^2\alpha\}d_m/D_a$,

the rolling bearing being characterized in that vibrations at frequencies of mZf_i, $mZf_i\pm f_r$, nZf_c, $2kf_b$ and $2kf_b\pm f_c$ are generated in the rolling bearing due to the circumferential undulations of (mZ) waves and (mZ±1) waves existing on the surface of the first raceway, to the circumferential undulations of (nZ) waves and (nZ±1) waves existing on the surface of the second raceway and to the undulation of (2k) waves existing on the rolling surface of the respective rolling bodies, and

that the formulas of $(mZf_i)\neq(nZf_c)$, $(mZf_i\pm f_r)\neq(nZf_c)$, $(nZf_c)\neq(2kf_b)$, $(nZf_c)\neq(2kf_b\pm f_c)$, $(2kf_b)\neq(mZf_i)$ and $(2kf_b\pm f_c)\neq(mZf_i\pm f_r)$, are satisfied for all of n, m and k in ~~the natural frequency domain of a rotation system which is~~

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~~a rotation supporting portion having the rolling bearing incorporated therein.~~

According to a third aspect of the present invention there is provided a rolling bearing comprising a first race having a first raceway, a second race provided around the first race and having a second raceway, Z rolling bodies having a rolling surface and rollably disposed between the first raceway and the second raceway, and a retainer for holding the rolling bodies,

provided that under grease lubrication, the first race rotates at the frequency of f_r (Hz) while the rolling bodies held in the retainer rotate at the frequency of f_b(Hz) and revolve at the frequency of f_c(Hz), that n, m, k and j are each a positive integer up to 100, respectively,

that D_a is the diameter of the rolling bodies,

that d_m is the pitch circle diameter of the rolling bodies,

that α is the contact angle between the rolling bodies and the first and second raceways,

that $f_i=f_r-f_c$(Hz),

that $nZf_c=(1/2)nf_r\{1-(D_a/d_m)\cos\alpha\}Z$,

that $mZf_i=(1/2)mf_r\{1+(D_a/d_m)\cos\alpha\}Z$, and

that $2kf_b=kf_r\{1-(D_a/d_m)^2\cos^2\alpha\}d_m/D_a$,

the rolling bearing being characterized in that vibrations at frequencies of mZf_i, $mZf_i \pm f_r$, nZf_c, $2kf_b$ and $2kf_b \pm f_c$ are the rotational frequency.

~~Also, in the case of the rolling bearing according to Claim 2, a plurality of kinds of vibration frequencies generated due to the above described undulations do not coincide with each other, in the natural frequency domain of a rotation system which is a rotation supporting portion constituted by incorporating the rolling bearing.~~

In the case of the rolling bearing of the present invention constructed as described above, banks of grease are formed due to the self-excited vibration of the rolling bodies, but this vibration is not susceptible to growth even if vibration occurs due to the banks of grease. As a result, harmful abnormal vibration and offensive noise to the ear are unlikely to occur. The reason for this will be described below.

At first, a description is given of the reason why the vibration frequencies controlled by the relation with the frequencies in other parts, are limited to one due to the undulations of (nZ) waves and (nZ ± 1) waves existing on the raceway surface, and the undulation of (2n) waves existing on the rolling surface of the respective rolling bodies. Here, it is well known, as described for example in Japanese Unexamined Patent Publication No. Toku Kai Hei 8-247153 or the like, that the undulations existing on the raceway surface and the rolling surface exist in a bodies (Hz)}, $f_i = f_r - f_c$ (Hz), f_b : rotating frequency of the rolling bodies (Hz).

If the frequencies of the vibration due to undulations determined by the expression described in the above Table 1 are deviated from the frequency of $f\omega = n \cdot f_r$ described above, banks of grease formed corresponding to the frequency of $f\omega$ are crushed between each rolling surface and the surface of the first and second raceways and collapse, as the respective rolling bodies carry out the rotation movement and the revolution movement. That is to say, if the above described both frequencies coincide with each other, banks of grease once formed further grow due to the vibration based on the undulations, and the vibration itself also grows, thereby resulting in the above described abnormal vibration and noise. On the contrary, if the above described both frequencies do not coincide with each other, the track of the respective rolling bodies which generates vibration due to the undulations does not coincide with the shape of the banks of grease. Hence, the respective rolling bodies crush this bank, to thereby prevent the vibration generated due to the self-excited vibration of the rolling bodies from growing. Rather, the bank of grease collapses to thereby absorb the energy of vibration due to the undulations, and hence alleviate the vibration due to the undulations.

Moreover, ~~according to Claim 2,~~ even in the case where a plurality of kinds

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bearing in a first embodiment of the present invention is used.

FIG. 4 is a Campbell chart representing vibration generated when a rolling bearing in a second embodiment ~~of the present invention~~ is used.

Respective reference symbols denote the followings:

1: rotation supporting apparatus; 2: support bracket; 3: rolling bearing; 4: buffer material; 5: housing; 6a, 6b: elements; 7: outer ring; 8: inner ring raceway; 9: inner ring; 10: outer ring raceway; 11: rolling body (ball); 12: space; 13: seal ring; 14: propeller shaft; 15: acceleration sensor; 16: amplifier; 17: computer.

Best More for Carrying out the Invention

In a procedure for designing a rolling bearing satisfying the requirement of the present invention, a description will be made of a case where a radial ball bearing in which the rolling bodies are balls, and where in use the outer ring, being the first race, is kept stationary and the inner ring, being the second race, is rotated, is used for supporting the rotation of a propeller shaft 14 as shown in FIG. 1.

The axial vibration frequency nZf_c (Hz) of the outer ring, the axial vibration frequency mZf_i (Hz) of the inner ring, and the axial vibration frequency $2kf_b$ (Hz) of the balls, resulting from the undulations of the bearing parts, which are nZ with regard to the inner ring and the outer ring, and 2n with regard to the rolling bodies described in the above Table 1, are respectively represented by the following Expressions (1) to (3):

$$\underline{n}Zf_c = (1/2)\, nf_r \{1 - (D_a/d_m) \cos \alpha\} Z \qquad (1)$$

$$mZf_i = (1/2)\, mf_r (1 + (D_a/d_m) \cos \alpha\} Z \qquad (2)$$

$$2kf_b = kf_r \{1 - (D_a/d_m)^2 \cos^2 \alpha\}\, d_m/D_a \qquad (3)$$

Here, in these expressions (1) to (3), n, m and k are optional positive integers. In the above described Table 1, all are denoted by n. However, in order to distinguish the source of axial vibration generated in the radial ball bearing, consideration is given by dividing these into three kinds of positive integers. Moreover, d_m (mm) denotes a diameter of a pitch circle of a plurality of balls constituting the radial ball bearing, and α denotes a contact angle between these balls and the respective races. Other symbols have the same meaning as described in the Table 1.

In order to realize a radial ball bearing corresponding to Claim 1, this is constructed such that the axial vibration frequency generated in each constituent part of the radial ball bearing, as shown by Expressions (1) to (3), does not coincide with the frequency jf_r (j is an optional positive integer) proportional to the rotational other frequencies. Moreover, since the radial vibration frequency due to the undulations of the $nZ \pm 1$ components existing on the raceway surface can be obtained in a similar manner to for the axial vibration frequency due to the undulations of the nZ component existing on this raceway surface, the description thereof is omitted.

It is assumed that in order to satisfy the condition of $nZf_c \neq mZf_i$ so as to realize the radial ball bearing corresponding to Claim 2, these frequencies nZf_c and mZf_i need only be different by ± 2% or more. To realize this, it is necessary to satisfy the following Expressions (4) ~~and~~ or (5):

$$nf_c/mf_i \geq 1.02 \qquad (4)$$

$$nf_c/mf_i \leq 0.98 \qquad (5).$$

If the aforesaid expressions (1) and (2) are substituted in these expression (4) and (5) and rearranged, the following expression (6) can be obtained. Here, since a radial pre-load is applied on the radial ball bearing, it is assumed that $\alpha \fallingdotseq 0$.

$$\begin{aligned} nf_c/mf_i &= (d_m - D_a)\, n/(d_m + D_a)\, m \\ &= (D_i/D_e) \cdot (n/m) \qquad (6) \end{aligned}$$

In this expression (6), D_i denotes a groove diameter of the inner ring (diameter of the bottom portion of the inner ring raceway) and D_e denotes a groove diameter of the outer ring (diameter of the bottom portion of the outer ring raceway).

the above step (5) is No), the groove diameter D_i of the inner ring and the groove diameter D_e of the outer ring are changed as shown in the above step (2), and the operation up to the above step (5) is repeatedly performed until the respective values become at least 1 ~ 2 % (until the judgment result in the above step (5) become Yes). When these respective values become at least 1 ~ 2 %, the procedure proceeds to the next step (7).

(7) The calculation in the above described expression (6) is carried out by using the axial vibration frequencies nZf_c of the outer ring and mZf_i of the inner ring obtained by step (3). This calculation is performed with respect to all values of n and m that can be practically considered.

(8) Based on the calculation result in the above step (7), it is judged if the second design conditions described above, namely $(nZf_c/mZf_i) \geq 1.02$ ~~and~~ or $(nZf_c/mZf_i) \leq 0.98$, are satisfied. If both these conditions are not satisfied, the groove diameter D_i of the inner ring and the groove diameter D_e of the outer ring are changed as shown in the above step (2), and the operation up to the above step (7) is repeatedly performed until ~~both~~ one of the conditions ~~are~~ is satisfied.

(9) If ~~both~~ one of the conditions ~~are~~ is satisfied (the judgment result in the above step (8) becomes Yes), the groove diameter D_i of the inner ring and the groove diameter D_e

Table 2

	$D_i/D_e =$ $(d_m - D_a)/(d_m + D_a)$	n/m	(D_i/D_e) (n/m)
Comparative product	0.713	7/5 = 1.4	0.998 ≒ 0%
Present invention 1	0.73	7/5 = 1.4	1.022 ≒ 2.2%
~~Present invention~~Comparative product 2	0.722	7/5 = 1.4	1.010 ≒ 1.0%

Among the three calculation examples shown based on the result exemplified in this Table 2, the result of the conventional product is $nZf_c \fallingdotseq mZf_i$. That is to say, 1026 Hz of the frequency component of the vibration resulting from undulation on the outer ring raceway and 1027 Hz of the frequency component of the vibration resulting from undulation on the inner ring raceway, become almost the same.

On the contrary, the result in the present invention is such that these do not coincide with each other. For example, in the case of the present invention product 1, 1109 Hz with respect to 1135 Hz, and in the case of the ~~present invention~~comparative product 2, 1022 Hz with respect to 1033 Hz. This means that in the case of the conventional product, banks of grease are retained for a long time and lead to vibration and noise of the radial ball bearing, but in the case of the present invention products 1 and comparative product 2, banks of grease disappear, and are unlikely to cause the above described vibration and noise.

Moreover, with regard to resonance in the natural frequency domain of the rotation system, with the conventional product, this is conspicuous such as at 32 f_r and 64 f_r as shown in FIG. 2 described below, but on the contrary, with the present invention products 1 and comparative product 2, vibration becomes small.

In the above description of the calculation procedure for the designs for realizing the rolling bearing of the present invention, the description is concerned with the case where the initial values of n, m and j are designated as 1, and these values are incremented by +1 from the initial value. However, the vibration level (amplitude) of the frequency components nZf_c, mZf_i, and jf_r of the above described respective vibrations decreases as the order increases (as the value of n, m and j becomes large). Therefore, from the viewpoint of decreasing the vibration and noise, it is not necessary to limitlessly increase the value of each natural number n, m and j. For example, it is preferable to limit the upper limit of these natural numbers n, m and j to about 100, respectively, in view of reduction of the calculation time, while exerting a practically effective reduction effect on vibration and noise.

Examples

Claims 1 and 2, and a second embodiment not belonging to the present invention ~~and satisfying only the condition of Claim 2~~, as shown in the following Table 3, obtained by changing the diameter of the rolling bodies, the number of rolling bodies and the pitch circle diameter of the rolling bodies, without changing the inner and outer diameters and the width of the ball bearing, the relation between the axial vibration frequency with respect to the rotational speed and the level (size) of the generated vibration were obtained, while rotating the respective inner rings. The Table 3 shows a case where rotational frequency f_r of the inner ring is 32 Hz (=1920 min^{-1} in rotational speed).

close, these do not coincide with each other. Hence, as is obvious from the small circles present on the straight lines of $5Z \cdot f_i$ and $7Z \cdot f_c$, the generated vibration is small. Moreover, these straight lines of $5Z \cdot f_i$ and $7Z \cdot f_c$ exist between the straight lines of $32f_r$ and $36f_r$, but do not coincide with any of the straight lines of $(33 - 35) f_r$ existing between these two straight lines. As a result, the generated vibration can be made sufficiently small.

Moreover, in the case of the second embodiment described in the lower part of Table. 3, the frequency of the vibration based on the fifth order of the undulation component of the inner ring ($5Z \cdot f_i = 1022$ Hz) coincides with the 32nd order frequency ($32 f_r = 1024$ Hz) of the rotational speed. That is to say, this does not satisfy the conditions of Claim 1. ~~However, this still satisfies the conditions of Claim 2, such that the vibration frequency components due to undulations existing on the surfaces of the outer ring and inner ring raceways and on the rolling surfaces of the rolling bodies do not coincide with each other in the aforesaid natural frequency domain (600 – 1100 Hz).~~

In the case of the second embodiment, though the vibration becomes slightly larger than in the case of the first embodiment, the generated vibration is considerably smaller compared to the case of the conventional product described straight line representing the vibration frequency passes through the origin point on the Campbell chart. Hence, also in the case of the radial vibration, this is the same as in the case of the axial vibration in that if these do not coincide with each other at any rotational frequency, these do not coincide at other rotational frequencies.

Since there is little possibility of a situation where only the conditions of Claim 1 are satisfied and the conditions of Claim 2 are not satisfied, any experiment for such a case was not performed. However, as is obvious from the above description, it is believed that even a structure satisfying only the conditions of Claim 1 can practically reduce vibration sufficiently. Needless to say, the most preferable structure is the one ~~described in Claim 3~~, wherein both conditions of Claims 1 and 2 are satisfied as with the first embodiment. Moreover, the above description has been made for the case of a deep groove type ball bearing, being a radial ball bearing, and for the case of axial vibration. However, the present invention can be similarly applied to radial vibration, by using the vibration frequency generated in the radial direction, instead of the vibration frequency generated in the axial direction. Also, the same idea can be applied not only to a radial ball bearing but also to a thrust ball bearing or a radial or thrust roller bearing.

circumferential undulations of (mZ) waves and $(mZ\pm1)$ waves existing on the surface of the first raceway, to the circumferential undulations of (nZ) waves and $(nZ\pm1)$ waves existing on the surface of the second raceway and to the undulation of $(2k)$ waves existing on the rolling surface of the respective rolling bodies, and

that the formulas of $(mZf_i)\neq jf_r$, $(mZf_i\pm f_r)\neq jf_r$, $(nZf_c)\neq jf_r$, $(2kf_b)\neq jf_r$ and $(2kfb\pm f_c)\neq jf_r$ are satisfied for all of n,m, k and j with respect to the frequencies.

~~2. A rolling bearing comprising a first race having a first raceway, a second race provided around the first race and having a second raceway, Z rolling bodies having a rolling surface and rollably disposed between the first raceway and the second raceway, and a retainer for holding the rolling bodies,~~

~~provided that under grease lubrication, the first race rotates at the frequency of f_r(Hz) while the rolling bodies held in the retainer rotate at the frequency of f_b(Hz) and revolve at the frequency of f_c(Hz), that n, m, k and j are each a positive integer up to 100, respectively,~~

~~that D_a is the diameter of the rolling bodies,~~

~~that d_m is the pitch circle diameter of the rolling bodies,~~

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~~that α is the contact angle between the rolling bodies~~ and the first and second raceways,

that $f_i=f_r-f_c(\mathrm{Hz})$,

that $nZf_c=(1/2)nf_r\{1-(D_a/d_m)\cos\alpha\}Z$,

that $mZf_i=(1/2)mf_r\{1+(D_a/d_m)\cos\alpha\}Z$, and

that $2kf_b=kf_r\{1-(D_a/d_m)^2\cos^2\alpha\}d_m/D_a$,

the rolling bearing being characterized in that vibrations at frequencies of mZf_i, $mZf_i\underline{+}f_r$, nZf_c, $2kf_b$ and $2kf_b\underline{+}f_c$ are generated in the rolling bearing due to the circumferential undulations of (mZ) waves and $(mZ\underline{+}1)$ waves existing on the surface of the first raceway, to the circumferential undulations of (nZ) waves and $(nZ\underline{+}1)$ waves existing on the surface of the second raceway and to the undulation of $(2k)$ waves existing on the rolling surface of the respective rolling bodies, and

that the formulas of $(mZf_i)\neq(nZf_c)$, $(mZf_i\underline{+}f_r)\neq(nZf_c)$, $(nZf_c)\neq(2kf_b)$, $(nZf_c)\neq(2kf_b\underline{+}f_c)$, $(2kf_b)\neq(mZf_i)$ and $(2kf_b\underline{+}f_c)\neq(mZf_i\underline{+}f_r)$, are satisfied for all of n, m and k in the natural frequency domain of a rotation system which is a rotation supporting portion having the rolling ~~bearing incorporated therein.~~

2 ~~3~~. A rolling bearing comprising a first race having a first raceway, a second race provided around the first race and having a second raceway, Z rolling bodies having

to the undulation of (2k) waves existing on the rolling surface of the respective rolling bodies,

that the formulas of $(mZf_i) \neq jf_r$, $(mZf_i \pm f_r) \neq jf_r$, $(nZf_c) \neq jf_r$, $(2kf_b) \neq jf_r$, and $(2kf_b \pm f_c) \neq jf_r$ are satisfied for all of n, m, k and j with respect to the frequencies, and

that the formulas of $(mZf_i) \neq (nZf_c)$, $(mZf_i \pm f_r) \neq (nZf_c)$, $(nZf_c) \neq (2kf_b)$, $(nZf_c) \neq (2kf_b \pm f_c)$, $(2kf_b) \neq (mZf_i)$ and $(2kf_b \pm f_c) \neq (mZf_i \pm f_r)$, are satisfied for all of n, m and k in the natural frequency domain of a rotation system which is a rotation supporting portion having the rolling bearing incorporated therein.

3~~4~~. The rolling bearing of ~~one of claims 2 and 3~~claim 2, wherein the formulas of $nf_c/mf_i \geq 1.02$, ~~and~~ or $nf_c/mf_i \leq 0.98$ (5) are satisfied, wherein $nf_c/mf_i = (d_m - D_a)n/(d_m + D_a)m$.

~~5~~4. The rolling bearing of ~~one of claims 2 and 3~~claim 2, wherein the natural frequency domain of the rotation system is in the range of Fn±250 (Hz) where Fn is the natural frequency which is determined by detecting an acceleration generated by impulse excitation of the rotation system by way of a hammer and processing the acceleration with FFT.

5~~6~~. A method of establishing the conditions of a rolling bearing as set forth in claim 3~~4~~, the method comprising the steps of :

(a) in the formulas $nZf_c=(1/2)nf_r\{1-(D_a/d_m)\cos\alpha\}Z$ and $mZf_i=(1/2)mf_r\{1+(D_a/d_m)\cos\alpha\}Z$,

increasing the values n and m from the initial value of 1 to 100 so as to obtain the axial vibration frequency nZf_c(Hz) of the outer race and the axial vibration frequency mZf_i(Hz) of the inner race due to the number of waves of undulation on the outer and inner raceways,

(b) increasing the value of j from the initial value of 1 to 100 to obtain the frequency jfr proportional to the speed fr of the inner race,

(c) obtaining $|nZf_c-jf_r|/nZf_c$ and $|mZf_i-jf_r|/mZf_i$,

(d) if the value obtained at the step (c) is at least 0.02, going to the step (f),

(e) changing the groove diameters of the inner and outer races to obtain the pitch circle diameter of the rolling bodies, and going back to the step (a) and repeating the steps (a) to (d) until the condition of the step (d) is satisfied,

(f) obtaining nZf_c/mZf_i for all of nZf_c and mZf_i which are obtained at the step (a) wherein n, m=1 to 100,

(g) examining all the values obtained at the step (f) on the conditions of $(nZf_c/mZf_i) \geq 1.02$ ~~and~~ or $(nZf_c/mZf_i) \leq 0.98$, and if the conditions are satisfied, going to the

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step (h) and if the conditions are not satisfied, going back to the step (e), and

(h) deeming the values which satisfy the conditions of the step (g) proper, and finishing the calculation.

6 ~~7~~. The method of claim 5 ~~6~~, wherein in the step (e), the diameter of the rolling bodies, the number of the rolling bodies and the pitch circle diameter of the rolling bodies are changed without changing the inner and outer diameters and the width of the ball bearing.