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VIBRATION ANALYSIS OF VERTICAL SHAFT MOTORS

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I. INTRODUCTION

Recent heavy electrical machinery for industry tends to employ vertical shafts. The main reasons for this are as follows:

(1) The installation space is only 60~70% of that required for horizontal shafts.
(2) Larger capacities and lighter weights are possible.
(3) Performance of thrust bearing is improved.

In the case of pumps, etc. the use of the vertical shaft is effective against the occurrance of cavitation. There are also many advantages in respect to performance and handling such as the fact that priming is not required when starting. The tendency recently has been toward more compact machinery because of higher speeds and there has been considerable demand for high speed vertical shaft electrical machinery. Because of the extremely wide use of speed control in order to make equipment more efficient, the problem of vibrations in vertical shaft machinery has come to the fore. Fuji Electric has long years of experience and technology concerning this type of vibration problem and has completely solved the vibration problem in vertical shaft machinery. This will therefore be outlined in this article.

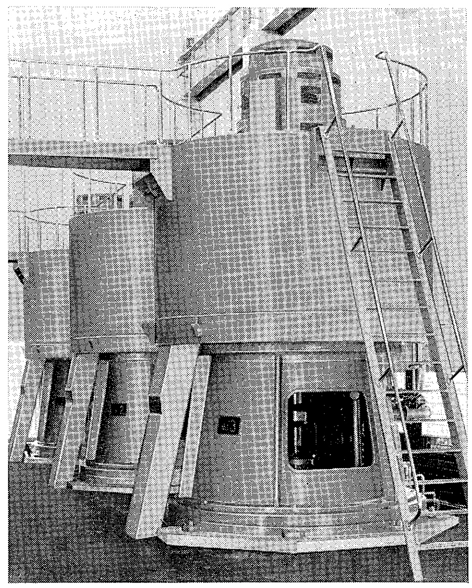

Fig. 1 Set of vertical shaft motor

II. CAUSES OF VIBRATIONS

There are many causes of vibration in vertical shaft machinery and it would be very difficult to describe each one in detail. Only the main causes will be considered here.

1. Exciting Force

Vibrations will occur due to the forced vibrations in vertical shaft machinery, especially when combined with drive motors.

1) Unbalance forced vibration

When vibrations of the same frequency as the rotational speed occur, these vibrations can be considered as due mainly to the unbalance forced vibration. The unbalance forced vibration is proportional to the product of the residual unbalance of the rotor and the square of the rotational speed. Therefore, during manufacture to the rotor, complete balancing must be performed (static or dynamic balancing) so that the amount of residual unbalance is within the permissible value (third class of JIS B0905 "Balance Quality of Rotating Machinery"). Balancing at the site (field balancing) must also be easy to perform since unbalancing will increase over the years.

2) Electromagnetic forced vibration

An electromagnetic pulling force can be caused by unbalances in the air gaps, windings, power source voltage, etc. and this force can cause vibrations. The frequency of these vibrations are generally twice that of the commercial power frequency or of the slip frequency. However, this can be checked at the test stage during manufacturing and eliminated. This type of vibration therefore presents on problem in actual use.

3) Bearing forced vibration

Vibrations caused by the bearing are oil whip and oil whirl in the case of journal bearings. In the case of ball and roller bearings, the vibrations are those passing through the drive unit and those due to the inclination of the inner and outer races. When single row angular contact ball bearings in thrust bearings or spherical roller thrust bearings are used as is generally the case in vertical shaft machinery, the vibrations are turning vibrations due to floating of the rotor. Such vibrations can be avoided if sufficient investigations are made at the design stage by such measures as maintaining the minium limit load of the bearing or using a guide bearing in conjunction with the other bearing.

4) Unbalance forced vibration due to direct coupling

This vibration force is caused by the force due to poor eccentricity or linearity of the shaft center when direct coupling the machinery with a motor, an unbalanced torque based on coupling machning error, or the unbalanced force of the other coupled machine. This problem is easy to solve at the time of setting or processing. It is also necessary to consider exciting force due to cavitation, water hammers, etc. in pumps.

5) Exciting force transferred from the foundation

There are cases of vibration forces being transferred from other machinery in the surrounding area. In such cases, it is necessary to decrease the vibration force of the machine causing the vibration or provide vibration isolation by means of some vibration-resistant support. Therefore, it is first necessary to investigate carefully when installing a machine which can cause large vibration forces.

2. Resonance of Vertical Shaft Sets

As was described above, there are many types of exciting force which can arise in vertical shaft machinery. However, the ordinary vibration problems in vertical shaft machinery are not limited only to such forced vibrations. It is rather a problem of the resonance between the natural frequency and the exciting force in vertical shaft sets. *Fig. 2* shows a typical example of the relation between the vibration characteristics and resonance frequency.

The problem of resonance in a vertical shaft set has been handled in the past as a problem of the individual motor in which the greatest amount of vibration occurred because the electric motors used as drive motors were always installed at the top part. However, it is essential that the problem be considered as arising due to resonance between the previously described exciting force and the natural vibration in the case of vertical shaft sets consisting of a combination of motor and motor bed as described later. In other words, the problem becomes very easy and can be completely solved if the natural frequency of the set is calculated at the design stage and measures are taken so that is does not match the frequency of the exciting force measured beforehand.

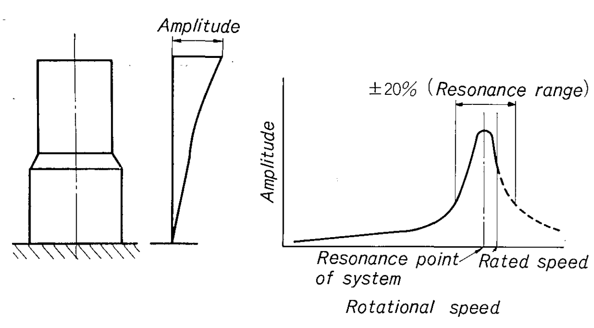

Fig. 2 An example of resonance of 160 kW, 8 pole pump motor

III. NATURAL FREQUENCY OF VERTICAL SHAFT SETS

1. Preliminary Conditions

When calculating the natural frequency of the system, the following conditions have been established from measured data in vertical shaft sets.

(1) The directional characteristics of the natural frequency due to the influence of the coupling holes of the motor bed and the discharge nozzle in the case of pumps are almost constant.
(2) Coupling directly with another machine has almost no influence on the natural frequency or the vibration changes (amplitude, frequency etc).
(3) In the case of pump sets, the natural frequency is almost not altered by the shape of the part below the ground level or the water level.
(4) The mode of vibration is generally like that shown in *Fig. 2*.

Therefore, for motors and motor beds above the ground level, the natural frequency of vertical shaft sets is calculated as the curved natural frequency considering it as the rigidity of the motor and motor bed and a cantilever which has mass. An accurate

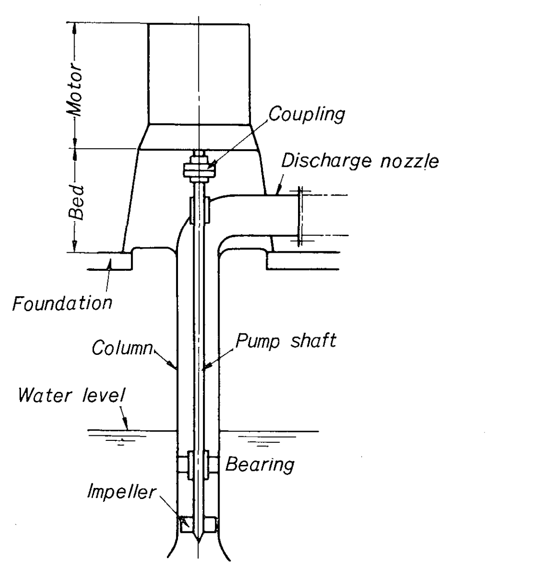

Fig. 3 Construction of vertical shaft pump set

calculation can be made under such conditions using a computer but in this article a simpler method with good accuracy is described.

2. Calculation of Natural Frequency

1) Calculation assuming the motor bed to be a portal type frame

For the simplified calculation, the construction of the vertical shaft pump set is to be as shown in *Fig. 3* and the model is to be as shown in *Fig. 4.* Under the above preliminary conditions, the natural frequency can be expressed as follows:

$$N_C = 5\sqrt{\frac{1}{\frac{Wl_1^3}{3E_1I_1} + \frac{Wl_2^3}{3E_2I_2}\left(1+3\frac{l_1}{l_2}+3\frac{l_1^2}{l_2^2}\right)}} \quad \cdots\cdots (1)$$

From various experimental results, the accuracy is very good as shown in *Fig. 6* when the geometrical moment of inertia of the motor bed I_2 is the equivalent value obtained by assuming the portal type frame construction as shown in *Fig. 5.*

However, the equivalent I_2 of the motor bed with the portal type frame construction can be expressed as follows:

$$I_2 = \frac{2I_2b}{1-\frac{3/4}{1+\frac{1}{6}\times\frac{I_2b}{I_2f}\times\frac{b}{l_2}}} \quad \cdots\cdots (2)$$

where I_2b : geometrical moment of inertia of motor bed with portal type frame construction

I_2f : geometrical moment of inertia of motor bed in thickness direction of motor side flange

b : distance between neutral axes of motor assuming portal type frame construction

l_2 : height of motor bed

2) Simplified calculation method

As will be described later in 4., when the natural frequency of the motor is above 60~80 Hz, the motor I_1 has almost no influence on the natural frequency of the system when considered as a rigid unit and therefore, equation (1) can be simplified as follows:

$$N_C' = 5\sqrt{\frac{1}{\frac{Wl_2^3}{3E_2I_2}\left(1+3\frac{l_1}{l_2}+3\frac{l_1^2}{l_2^2}\right)}} \quad \cdots\cdots (3)$$

Therefore, it can be said that when designing the motor bed floor, the only information needed in respect to the motor is (1) the motor weight W_1 and (2) the center of gravity l_1.

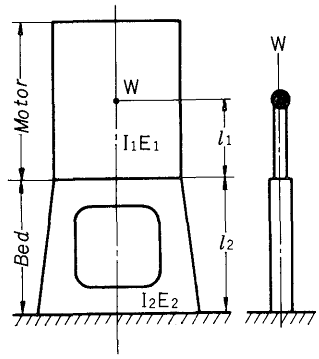

Fig. 4 **Relation between stiffness of motor, motor bed and their natural frequency**

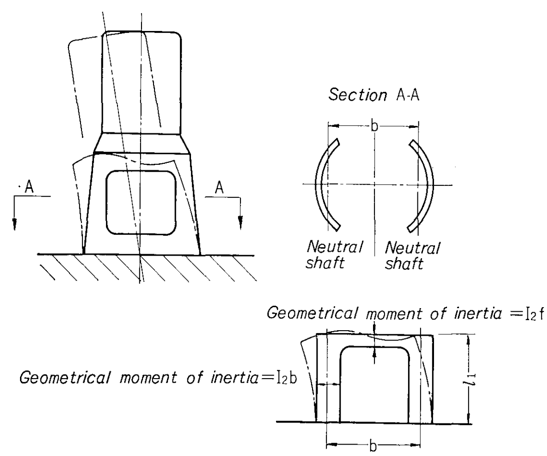

Fig. 5 **Distortion mode of motor bed**

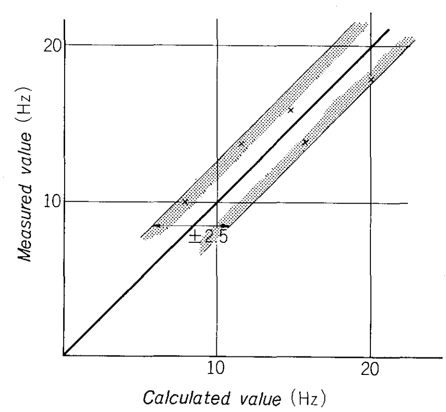

Fig. 6 **Comparison with calculated and measured values of natural frequency**

3. Setting the Natural Frequency of the System

As can be seen from *Fig. 2*, is generally recommended that the natural requency of vertical shaft sets be about 20% above the frequency of the exciting force. The problem is whethei the support should be rigid or elastic, but for ordinary use, it is best that the naturrl frequency be higher, i. e. the rigid support be used. However, as the speeds increase, both economic and technical limits arise.

Therefore, elastic supports become unavoidable and the rigidity has to be decreased.

This problem will now be considered from the standpoint of the limiting value of the vibration amplitude due to the forced vibrations.

The main forced vibration forces in the vertical shaft sets as described above can be considered as the rotational unbalanced force which has a frequency corresponding to that of the rotational speed. In this case, the natural frequency must be separated from the rotation cycle and set so that the vibration amplitudes due to the unbalanced vibration force are within the permissible values.

The balance of ordinary motors is standardized at the third class given in JIS B 0905 (1967) "Balance Quality of Rotating Machinery" and therefore it is necessary to consider the unbalance force and the third class maximum value (balancing quality=2.5). The eccentricity ε in this case can be expressed as follows:

$$\varepsilon=\frac{23900}{n}(\mu) \qquad (4)$$

where n=rotational speed (rpm)

The vibration amplitude δ caused by the unbalanced force due to eccentricity ε becomes as follows in the system shown in *Fig. 4*.

$$\delta=\frac{W_R}{g}\times\omega_2\times\varepsilon\left\{\frac{l_1^3}{3E_1I_1}+\frac{l_2^3}{3E_2I_2}\left(1+3\frac{l_1}{l_2}+3\frac{l_1^2}{l_2^2}\right)\right\} \qquad (5)$$

where:
δ : amplitude (μ)
W_R : rotor weight
ω : angular velocity
g : acceleration due to gravity

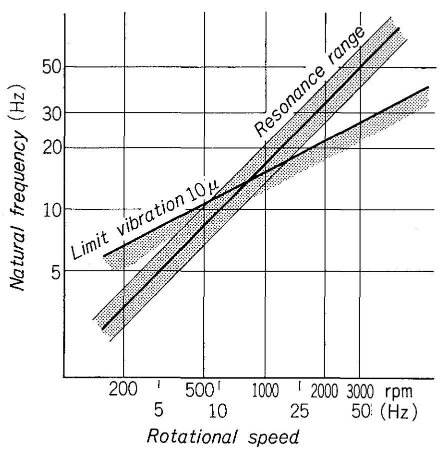

Fig. 7 Selecting range of motor bed natural frequency

Therefore, for normal motors, the vertical shaft machine's vibration can be expressed as follows equations (1), (4) and (5) and when (total motor weight W) ≑ (motor rotor weight W_R)×3:

$$\delta=2.23\times\frac{n}{N_C} \qquad (6)$$

Fig. 7 was obtained from this equation.

If the limit value of the vibration amplitude is less than 10μ, the following conclusion can be obtained from *Fig. 7*.

(1) In low speed machines of under 1,000 rpm, the rigid support should be used, i. e. natural frequency should be higher. In high speed machines of over 1,000 rpm, the natural frequency should be lower.

(2) In the case of high speed machines over 1,000 rpm, it is necessarry to investigate carefully and choose an appropriate natural frequency at the time of design since the safety range is small.

4. Setting the Natural Frequency of Motors

As can be seen from *Fig. 7*, it is necessary to set the natural frequency in the 10~30 Hz range. In this case, the setting of the natural frequency of the individual motor will be considered.

In the natural frequency formula shown in equation (1), the natual frequency becomes as follows if the geometric moment of inertia of the motor I_1 is a maximum.

$$N_C'=5\sqrt{\frac{1}{\frac{Wl_2^3}{3E_2I_2}\left(1+3\frac{l_1}{l_2}+3\frac{l_1^2}{l_2^2}\right)}} \qquad (7)$$

and from equations (5) and (1):

$$\left(\frac{N_C'}{N_C}\right)^2=\frac{1}{1-\left(\frac{N_C}{N_{CM}}\right)^2} \qquad (8)$$

Therefore, the ratio of N_C'/N_C in repsect to the natural frequency of the individual motor

$$N_{CM}=5\sqrt{\frac{1}{\frac{Wl_1^2}{3E_1I_1}}}$$

becomes as shown in *Fig. 8* with N_C as the system parameter.

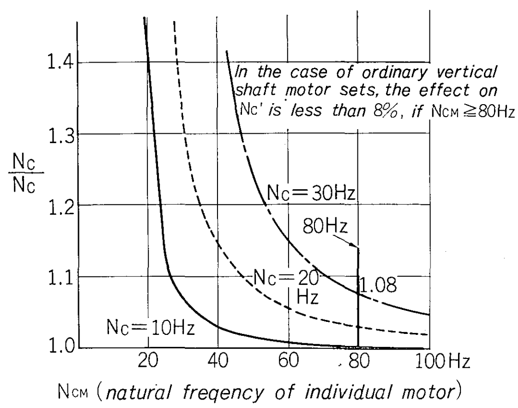

Fig. 8 Effect of natural frequency of individual motor to total pump set

In other words, if the natural frequency of the individual motor is set above 70~80 Hz, the influence of the geometrical moment of inertia of the motor I_1 on the natural frequency can be neglected.

IV. METHODS OF AVOIDING RESONANCE

The measures for avoiding resonance will be discuseed in reference to *Fig. 9*.

This figure shows, in the natural frequency of equation (1), the natural frequency when $l_1=l_2$, $E_1=E_2$ and $I_1=I_2$ versus the variable ratio of the natural frequency when any one of I_1, I_2, W and l_1+l_2 are changed. From this figure, the following points are evident:

(1) Even when the geometrical moment of inertia of the motor I_1 is altered somewhat, the natural frequency does not change and this is therefore not an effective method.

(2) However, when the geometrical moment of inertia of the motor bed is changed by 50%, a change of about 20% in the natural frequency can be expected.

In other words, this latter change is very effective. Changes in l_1+l_2 at the design stage are also very effective.

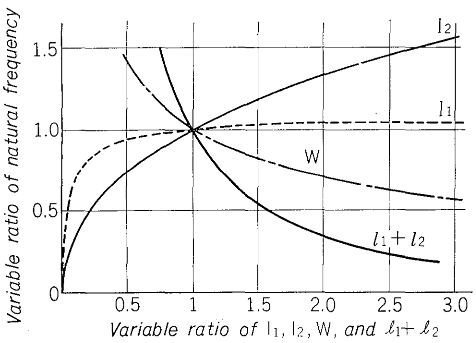

Fig. 9 Variable ratio of I_1, I_2, W, l_1+l_2 and natural frequency

V. CONCLUSION

This article has dealt with the problems of vibration in vertical shaft sets and especially of resonance. Practical methods of solving these problems were introduced and the authors wish to thank all those persons who aided in the establishment of these methods.