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FUJI SPINNING POT MOTOR

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I. INTRODUCTION

The pot motor for rayon spinning has naturally been developing and improving together with the progress of rayon production. Domestic production of rayon was restored to the pre-war level in 1949. The production showed further growth in 1950 because of the sharp increase in facilities investment, continuing until 1957. In that year a recession occurred, resulting in a temporary sharp drop in rayon production. It is said, however, that production has since increased slowly but steadily, recently being restored to the condition formerly enjoyed. Large quantities of the Fuji Electric pot motor were manufactured for domestic use up until 1957 to meet the demands of the rayon industry. Since then, the production of the pot motor showed a slight drop, but production for export has increased since around 1959, a number of pot motors being delivered to overseas customers, such as Baroda Rayon, National Rayon, and Travancore Rayon of India.

Since the restoration of the technical tie-up of Fuji Electric with Siemens of West Germany after the war, Fuji Electric has manufactured a large quantity of pot motor Model RS 11–2, thus greatly contributing to the realization of higher speed rayon spinning and larger pots by superbly meeting the then increasing demands.

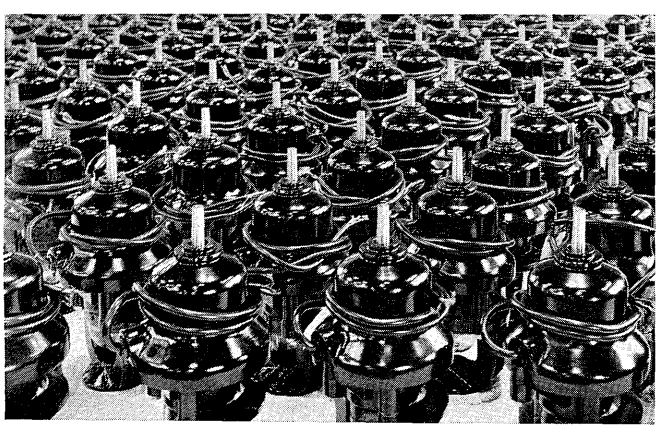

Fig. 1 Pot motors, RS 1112–2

Since then, further research has been performed based on the operation experience of users. Fuji Electric improved various points of the pot motor then in use, announcing a new type of pot motor in 1960. This new type of pot motor has been delivered to many companies in India, as previously mentioned, as well as to domestic users, showing unprecedented performance.

At present, pot motors are being manufactured for delivery to Dawood Rayon of Pakistan and to Viskoza Loznica of Yugoslavia.

Given in the following is a general description of the Fuji pot motor, with Model RS 1112–2 for Dawood Rayon taken as an example.

II. MODEL RS 1112-2 POT MOTOR

1. Specifications

Model:	RS 1112–2
Voltage:	133.3 v
Frequency:	133⅓ cps
Synch. speed:	8000 rpm
Phases and poles:	Three-phase, two-pole
Input:	230 w
Cake dimensions:	7″ diameter × 6″ height

An outer view of the pot motor model with the above specifications is shown in *Fig. 2*. *Fig. 3* shows the external dimensions.

The major basic requirements for the pot motor include the following: The motor is always placed under the most severe chemical conditions, exposed directly to the splash of Glauber's salt and sulfuric acid; with respect to the unbalance of the pot and cake formed in it, and also to the excessive rotational load produced by high speed rotation of approximately 7000 to 10,000 rpm, the vibration of the pot and the motor should be minimized, together with reduction of the bearing load; minimum profitable number of pot motors for the rayon industry is said to be c. 4000 units, most of which are operated 24 hours a day. Thus the motors should be of high

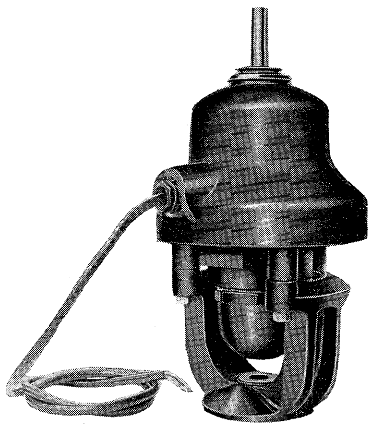

Fig. 2 Pot motor, RS 1112-2

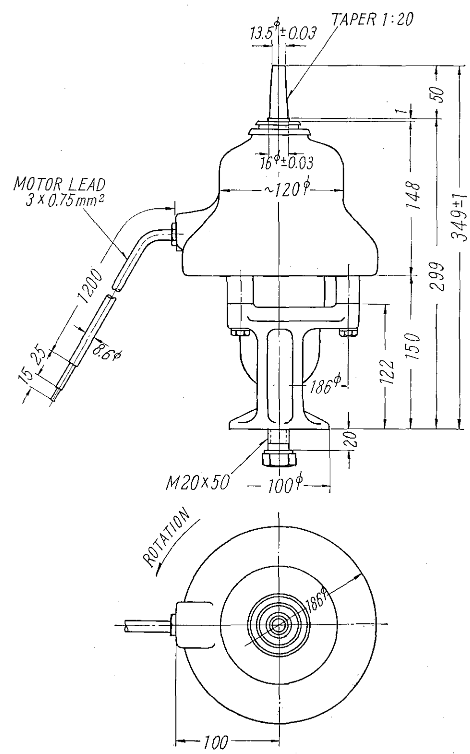

Fig. 3 Outline drawing of pot motor, RS 1112-2

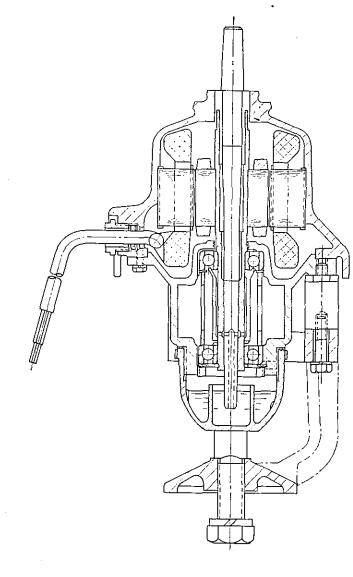

Fig. 4 Sectional drawing of pot motor, RS 1112-2

electrical efficiency; daily occurrence of trouble requiring maintenance calls should be a minimum. They should be light and small, with as few component parts as possible for ease in disassembling and reassembling.

2. Construction

The structure of the pot motor is shown in *Fig. 4*.

The pot motor can be roughly divided into four sections, that is 1) the frame, which is the electrical unit, 2) the spindle and vibration-proof support section which form a vibration-proof mechanism, 3) the bearing and lubrication mechanism, and 4) the mounting section, etc.

1) The frame accommodating the electrical unit is made of special cast iron, having an extremely smooth surface which is specially coated with utmost care for excellent resistivity to acid, alkali, shock and impact. This special coating is incorporated into the unique shape of the equipment to provide it with sufficient durability even in operation under the most severe chemical conditions.

   The rotor-equipped hollow shaft separated from the spindle extends upward, allowing a gap of only approximately 0.4 mm from the top section of the frame, so that the external atmospheric air cannot penetrate inside. The labyrinth formed at the top section of the frame and the pot holder improves the efficiency of such prevention. The thermal and oil resistive cab tyre cable is drawn out from the bottom section of the frame through the packing gland. A special silicon steel plate is used in order to reduce the input; for the insulation, thermal and acid-resistivity have been provided.

2) The vibration-proof mechanism is one of the most important factors. This pot motor features such a mechanism with simple construction and excellent vibration-proof characteristics.

   The rotor shaft is separated from the drive shaft and is called hollow shaft. This shaft is taper connected to the bottom of the spindle. Consequently a spindle which bends when absorbing the vibration of the pot does not affect the gap between the rotor and the stator. This permits, as described in the following section, a construction in which the overall height may be lowered for extreme stability from the point of view of vibration dynamics, thus increasing the strength of the spindle. For the vibration-proof mechanism, three synthetic rubber cushions are effectively used for simplicity.

This merit is one of the results obtained regarding the vibration-proof rubber which has been adopted on the basis of trial and error, and found to be most effective for vibration-proofing and support of vehicles and rolling stock, from the recent joint research of both high molecular chemistry and vibration dynamics.

3) For the bearing, two deep groove ball bearings 6204 are arranged with a sufficient separation between them for minimizing the bearing load. The lubricant is drawn in from the bottom of the spindle and allowed to run through the hollow shaft through a lateral hole from the taper bottom section. It then further runs along the internal wall of the oil pipe for lubricating the lower bearing, returning to the oil tank. A cylindrical separator is installed in the oil tank in order to separate fine sludge for safety purposes. The bearings used are limited to the ball bearing, with the lubricant, limited to spindle oil No. 1, having very low viscosity, so that the mechanical loss of the motor is minimized. Since electrically high grade material is used, in addition to this minimum mechanical loss, the load input of the lowest value may be obtained for a pot motor of this type.

   The bearing case and the oil tank are assembled by a long threaded portion and oil-resistive packing; since the portion where the hollow shaft runs through the bearing case is provided with a long labyrinth, there is no leakage of lubricant oil; in addition it prevents the penetration of acidic substances which may deteriorate the oil.

   It results in a remarkable long life span of the lubricant oil.

4) Installation of the pot motor on the spinning machine is rapidly and simply made by means of a single bolt. In all, the number of associated parts is slightly above 30. The pot motor is packaged with utmost care for overseas shipment, with its packing efficiency already proven. After the packaged equipment had been left in a location of high temperature and high humidity for a period of nearly one year, nothing occurred to impair the performance of its contents.

3. Features

The pot motor of this type has the following features:

1) Electrical characteristics: As is seen from the subsequent section, the no-load input is as small as approximately 23 w; the load input is also extremely low, but the motor provides excellent slip, power factor, and overload resistivity. This superiority is attributable to use of specially high quality electrical material, to the limited employment of ball bearings rather than plane bearings, to the fact that there is no portion, other than the rotor where a large air-resistance loss is caused by rotation, as well as to the use of a lubricant oil of low viscosity.

2) Vibration-proof characteristics: In order to drive the pot and pot case at the very high speed of 8000 rpm, which has a large mass and unbalance, the spindle makes full use of the gyro effect and is driven at a speed exceeding the primary critical speed. The relative positioning of the flexible unit and the vibration-proof supporting unit is one of the very important factors in the design.

   In this pot motor, a pure stacking method is not employed, in which the motor is mounted on the vibration-proof supporting unit and the flexible portion of the spindle brought onto the rotor, but another, hollow shaft method is adopted whereby the spindle is embedded under the rotor shaft, thus lowering the position of the pot. This method is highly stable from the point of view of vibration dynamics. This incorporates select vibration-proof rubber to provide unmatched vibration-proof characteristics when the speed exceeds the primary danger speed at the time of starting and braking, as well as at the running speed.

3) Spindle and hollow shaft: Since this method permits sufficient flexible portion and length to be obtained without the necessity for increasing the overall height, the diameter can be made larger under a constant flexibility. Also the method permits the lateral hole of the lubrication path to appear at the bottom of the tapered section, so that the effective cross-sectional area of the spindle is increased without causing a notch effect to the flexible section. This fact serves to enhance the fatigue resistance of the spindle. Replacing the spindle has no effect on the rotor, so that such replacement can be performed rapidly and economically. While the spindle bends dynamically, the hollow shaft is independent, so that even when the spindle has bent, the air gap between the rotor and the stator is not at all affected. Since the gap may be made smaller, the electrical characteristics can be improved.

4) Bearing: With two 6204 ball bearings used, the load capacity is greater and the mechanical loss smaller. Making full use of the hollow shaft, which permits minimum overall height, greater separation between bearings is possible and the relative positions of the vibration-proof supporting unit, bearing positions, and spindle supporting unit (lower tapered position) is properly selected, so that the load applied to the bearings is a minimum. This has also been confirmed by measurement of the bearing load, as mentioned in a later section. Because of the construction, replacement of the bearing is easy.

5) Lubricant oil: Defective lubrication causes serious harm such as thermal damage in the use of the pot motor. The lubricant oil does not run into the outside connecting space between the spindle and the hollow shaft, but is fully enclosed except at the labyrinth section of the bearing case run-through section of the hollow shaft. The joined portions of the container comprise only the bearing case and the oil tank, so that there will be no oil leakage at all, and deterioration due to penetration of acid is minimized.

The lubricant oil is deteriorated by heat. In this pot motor, the oil is exposed to the atmospheric air by means of the single wall of the oil tank of bearing case, so that the temperature rise of the oil is minimized and deterioration is not hastened. Only one type of oil is used. The oil tank can be removed simply by hand without employing any tools, so that handling of the lubricant oil is easy.

6) Number of parts, etc.: Thanks to the above-mentioned rational design, the number of parts is minimized, and consequently the weight is as light as approximately 12 kg, very effective in reducing the disassembling and assembling time, minimizing the cost of spare parts, transportation, and handling, as well as of general maintenance.

4. Characteristics

Fig. 5 shows the frequency variation characteristic curves for two pots, 7″ diameter × 6″ height (*A* of the figure) and what is generally called a small type pot (*B* of the figure). The Schneider formula is used for determination of the output of the pot motor. We have obtained 2.2 to 2.4 as an index of the speed. *Fig. 6* shows the dc braking characteristic curves for pot *A* of *Fig. 5*.

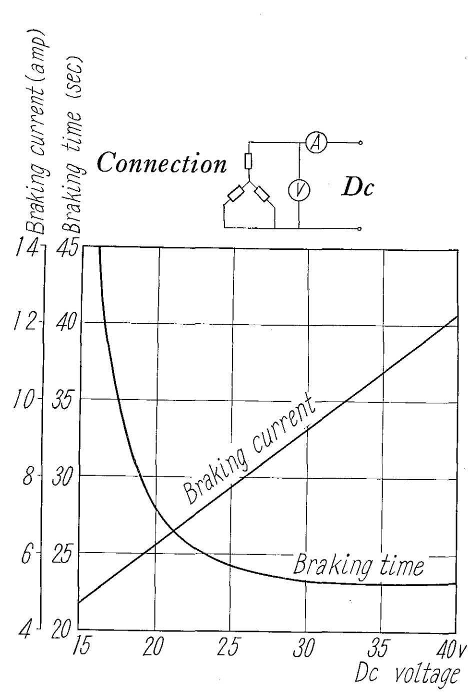

Fig. 6 Dc braking characteristic curves of pot motor, RS 1112-2

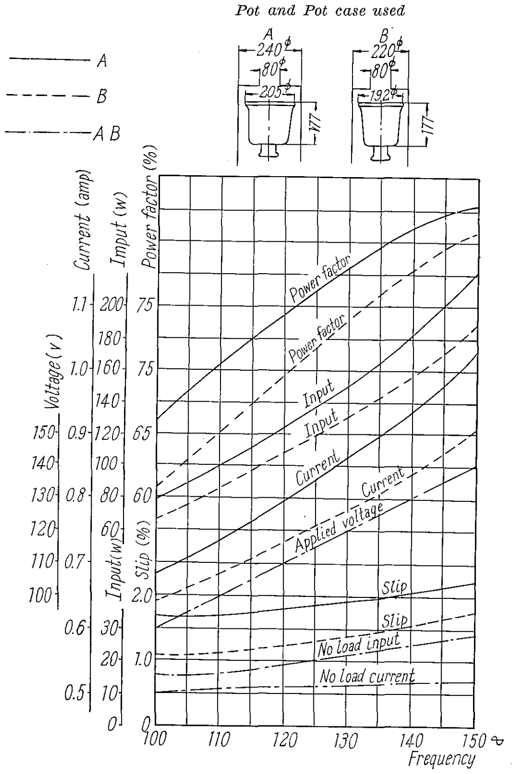

Fig. 5 Characteristic curves of pot motor, RS 1112-2

III. MEASUREMENT OF VIBRATION AND BEARING LOAD

1. Objective

Vibration of the pot motor is considered to be closely related to the bearing load. Consequently, when discussing the vibration of the pot motor, one of the important problems is the bearing load.

For measurement of the bearing load, there is a method available which uses a piezo-electric element and the resistance wire strain gauge. In this pot motor, the bearing-to-bearing separation is made large in order to reduce the bearing load, with the upper bearing inserted into the stator coil end, thus allowing no place to connect the measuring attachment. For this reason this method is not applicable. From the viewpoint of vibration dynamics, the elasticity and viscosity-supported motor (with the exception of rotating portions such as the rotor, etc.), receives the vibration load (equivalent to the bearing load) through the rubber cushions placed on the installation table, from the bearings; the motor is thus considered to cause vibration. Accordingly, by measuring the motor vibration, the bearing load may be obtained as given in detail in the following, with the results described.

2. Formula Relating Vibration Value and Bearing Load of Pot Motor

1) Relation of frame displacement to cushion displacement

Symbols (refer to *Fig. 7*)

a: Height from the cushioned surface of the vibration center

r: Installation radius of cushion

s: Cushion radius

θ: Angle of churning motion

ω: Angular speed of churning motion

$\delta, \delta_x, \delta_y, \delta_z, \delta_{\omega t}$: Cushion displacement vector and its components of x, y, z axis and the direction of inclination from the center axis.

Assuming that when the center axis VO is shifted by θ in the ωt direction, thus reaching point VO; A has come to point A', the following relation may be obtained, provided that θ is minute:

$$\delta_{\omega t}=AA'' \fallingdotseq \theta V'S=a\theta \quad \text{......(1)}$$

$$\delta_z=A'A'' \fallingdotseq \theta SA=r\theta \cos \omega t \quad \text{......(2)}$$

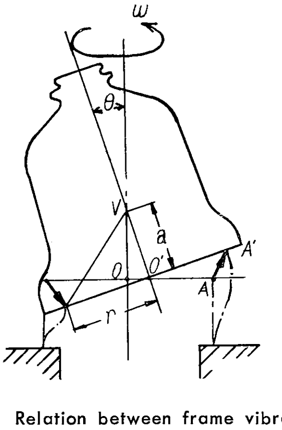

Fig. 7 Relation between frame vibration and cushion displacement

2) Relation of cushion displacement to reaction force

Of the A, B, and C cushions, cushion A's displacement rise is expressed by formulas (1) and (2), with the angle θ of the edge portion. After all, the reaction force due to the displacement of each of the cushions is obtained and the total value may be obtained as the total reaction force with respect to the frame. The relevant symbols are defined in the following:

$\phi=\theta$: Rotation distortion at the edge of cushions

$f=f_A=f_B=f_C$: Force in the xy plane, which is applied by $\delta_{\omega t}$ of A, B, and C cushions

f_{Az}, f_{Bz}, f_{Cz}: Force of direction applied by δ_z

$n=n_A=n_B=n_C$: Moment applied by θ

$F_{\omega t}$: Resultant force of cushions of A, B and C of reaction force by $\delta_{\omega t}$

F_z: Resultant force of reaction force by δ_z

M_z: Resultant moment of reaction moment by δ_z

M_θ: Resultant moment of reaction moment by θ

F: Vector total of $F_{\omega t}$, F_z, M_z, and M_θ, total cushion reaction force

b: Length from O of point on the VO axis applied to F

k_1 c_1: Shearing-direction spring coefficient and viscosity coefficient of cushion

k_2, c_2: Tensile direction

k_3, c_3: Rotation direction of edge

$\varepsilon=\varepsilon_1=\varepsilon_2=\varepsilon_3$: Mechanical loss factor of cushion as rubber substance

u: Ratio of rubber shearing elasticity factor G to nominal tensile elasticity factor E_{ap}

v: Ratio of G to nominal shearing elasticity factor G_{ap}

The following relations axists between the constants of the cushion: (Because of limited space the explanation cannot be given in detail). Consideration of the type and dimensions of the rubber with regard to the theory of the rubber mechanical motion will help to obtain the required relations.

$$k_2=\frac{u}{v}k_1 \quad \text{......(3)}$$

$$k_3=\frac{s^2u}{4v}k_1 \quad \text{......(4)}$$

$$\varepsilon=\varepsilon_1=\varepsilon_2=\varepsilon_3=\frac{c_1\omega}{k_1}=\frac{c_2\omega}{k_2}=\frac{c_3\omega}{k_3} \quad \text{......(5)}$$

(1) Reaction force $F_{\omega t}$ by δ_{wt}

$$f_x=k_1\delta_x+c_1\dot{\delta}_x$$
$$=k_1a\theta \cos \omega t-c_1a\theta\omega \sin \omega t$$
$$=a\theta \sqrt{k_1^2+c_1^2\omega^2} \cos(\omega t+\beta)$$

where,

$$\beta=\tan^{-1}\frac{c_1\omega}{k_1}=\tan^{-1}\varepsilon \quad \text{......(6)}$$

Likewise

$$f_y=a\theta \sqrt{k_1^2+c_1^2\omega^2} \sin(\omega t+\beta)$$

$$\therefore f=\sqrt{f_x^2+f_y^2}=a\theta \sqrt{k_1^2+c_1^2\omega^2}=a\theta k_1\sqrt{1+\varepsilon^2} \quad \text{...(7)}$$

Since the phase angle and $\alpha_{\omega t}$ of f_A, f_B, and f_C are the same, f may be used to explain them.

$$\begin{cases} F_{\omega t}=3a\theta k_1\sqrt{1+\varepsilon^2} & \text{......(8)} \\ \alpha_{\omega t}=\omega t+\beta+180° & \text{......(9)} \end{cases}$$

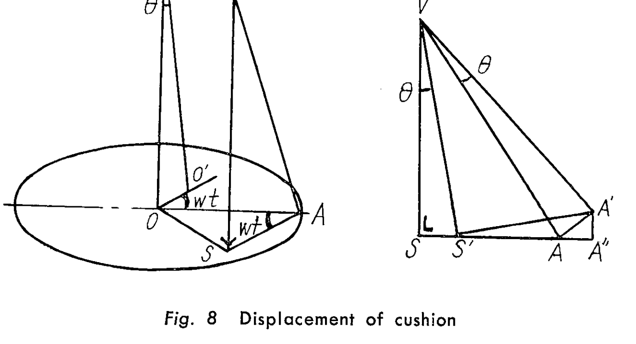

Fig. 8 Displacement of cushion

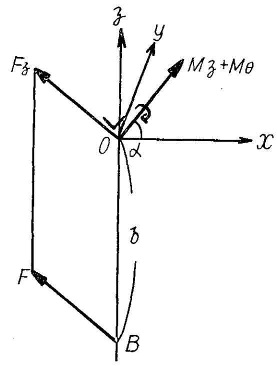

Fig. 9 Composition of reactive forces of cushions

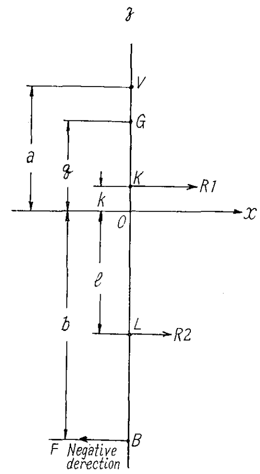

Fig. 10 Forces and their points acting on frame

(2) Reaction force F_z and M_z by δ_z

With remarks given to $f_{Az} \neq f_{Bz} \neq f_{Cz}$, calculation should be performed in a manner identical to that used for the formula (8).

$$F_z = f_{Az} + f_{Bz} + f_{Cz} = 0 \quad \text{.................(10)}$$

The following may be obtained with components separated into M_{zx} and M_{zy}

$$M_z = \frac{3}{2} r^2 \theta \sqrt{k_2^2 + c_2^2 \omega^2} = \frac{3r^2 \theta u k_1}{2v} \sqrt{1+\varepsilon^2} \quad \text{......(11)}$$

$$\alpha_Z = \omega t + \beta + 90° \quad \text{.....................................(12)}$$

(3) Reaction force M_θ by θ

By the former method of calculation, the following is obtained:

$$M_\theta = \frac{3s^2 \theta u k_1}{4v} \sqrt{1+\varepsilon^2} \quad \text{...........(13)}$$

$$\alpha_\theta = \alpha_z = \omega t + \beta + 90° \quad \text{.................(14)}$$

(4) All the cushion reaction forces are the same, but lag 90° behind $F_{\omega t}$, so that $F_{\omega t}$, F_z, M_z, and M_θ may be replaced by a separate single force F which is applied to the portion, lower by b than the origin O of the same direction and length of F_1 (refer to *Fig. 9*).

Consequently, $F = F_1$

$$F \times b = M_Z + M_\theta$$

$$\therefore \quad b = \frac{M_z + M}{F} = \frac{(2r^2 + s^2)u}{4av}$$

After all, the reaction force which the frame receives from the cushion may be expressed by a single force F applied at a phase angle α in a plane lower by b than the center O of the cushion effect.

$$\begin{cases} F = 3a\theta k_1 \sqrt{1+\varepsilon^2} & \text{....................(15)} \\ b = \dfrac{(2r^2+s^2)u}{4av} & \text{..........................(16)} \\ \alpha = \omega t + \beta + 180° & \text{.......................(17)} \\ \beta = \tan^{-1}\varepsilon & \text{...............................(6)} \end{cases}$$

3) Relations between frame displacement and bearing load.

Symbols

R_1, R_2: Radial load of upper and lower bearing K and L

κ, λ: Phase angle of R_1 and R_2

I_v: Moment of inertia of horizontal axis passing through V

I: Moment of inertia of horizontal axis passing through G

m: Mass of vibrating object (except for rotating parts such as the rotor)

X, Y: x and y components of G displacement (Refer to *Fig. 10*)

Considering that the vibrating object receives forces of R_1, R_2, and F and causes vibration a, θ, and ω, the following equation of motion is obtained and solved:

$$\begin{cases} R_1 = -\dfrac{3a\theta\varepsilon k_1(b+k)}{(k+l)\sin\kappa} & \text{.............................(18)} \\ R_2 = \dfrac{3a\theta\varepsilon k_1(b-l)}{(k+l)\sin\lambda} & \text{................................(19)} \\ \cot\kappa = \dfrac{1}{\varepsilon} + \dfrac{\omega^2\{m(a-q)(q+l)-I\}}{3a(b-l)\varepsilon k_1} & \text{...........(20)} \\ \cot\lambda = \dfrac{1}{\varepsilon} + \dfrac{\omega^2\{m(a-q)(q-k)-I\}}{3a(b+k)\varepsilon k_1} & \text{...........(21)} \end{cases}$$

In formulas (18) through (21) and in (16), above, the values of k, l, r, and s are obtained from the drawing; m, I, and q are actually measured; θ, ω, and a are measured for vibration; ε, k, u, and v are constants of the cushion.

This enables the theoretical values of R_1, and R_2 to be obtained.

3. Vibration Measurement of Pot Motors and Bearing Load

In the manner described in the preceding section, measurement has been conducted, with the bearing load of the pot motor RS 1112–2 compared with that of the old model RS 11–2. Three fixed points of the motor main unit were measured with respect to the vibration. The values of these three points were used to obtain a and θ values with their geometric relations noted. Upon measurement of I, the value was obtained with the motor suspended horizontally by means of two thin steel wires, and the following results obtained:

1) Results

Figs. 11, 12, and *13* illustrate the important results relevant to this paper.

2) Conclusion

(1) The bearing load of the pot motor RS 1112–2 is obtained within approx. 11 kg to 64 kg for both the R_1 and R_2 from experiment. With respect to this range, the maximum value of R_1 is 268 kg and R_2 27 kg, demonstrating that the absolute value of RS 1112–2 is small and the balance between the upper and lower bearing is extremely satisfactory for $R_1 \approx R_2$.

However in the case of RS 11–2, the absolute value is great and the upper and lower balance of $R_1 \gg R_2$ is poor.

(2) The bearing loads of RS 1112–2 and RS 11–1 are as shown in *Fig. 11*, showing some improvement for RS 1112–2. This fact is also applicable to vibration.

(3) There exists a similar relation between the vibration, which varies according to the frequency, unbalance, and size of the pot load and bearing load, so that in subsequent experiments the bearing load may be estimated with ease by simply measuring the vibration (Refer to *Figs. 12* and *13*).

(4) The fact that $a<0$ in the case of RS 1112–2 and $a>0$ in the case of RS 11–2 is related to the previous Item (1) when considering the effect of points R_1 and R_2.

IV. SUMMARY

Taking advantage of this opportunity, when pot motors are being manufactured for delivery to the Dawood Ryon and Viskoza Loznica Company, explanation is given above concerning the construction, features, and performance of pot motor RS 1112–2, as well as calculation formulas and their results for the bearing load.

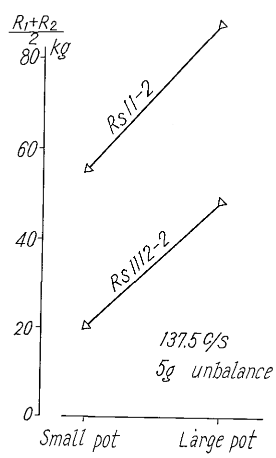

Fig. 11 Comparison of bearing loads

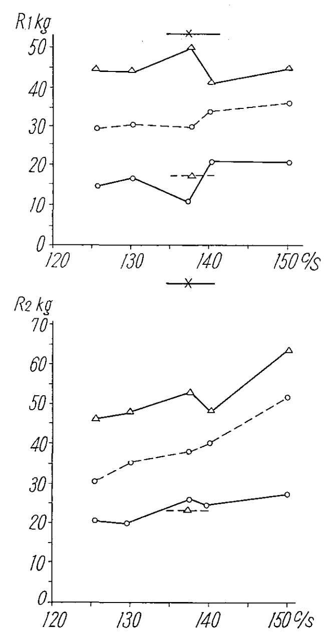

Fig. 12 Bearing loads of RS 1112–2

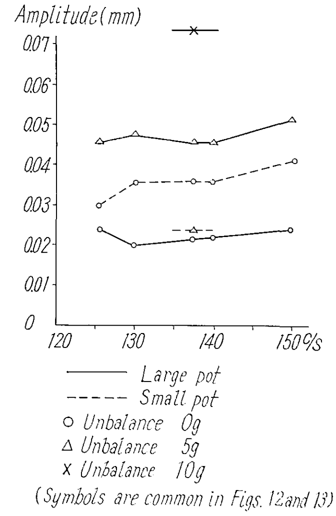

Fig. 13 Vibration of RS 1112–2

A great quantity of these pot motors have already been delivered both domestically and abroad, displaying most satisfactory performance results. It is hoped that the pot motor may be further improved with respect to construction, theory, and new concepts, thus rationally meeting the demands of operation of the chemical, electrical and mechanical fields.

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