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TRIBOLOGY ANALYSIS IN ROLLING PISTON TYPE COMPRESSOR

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ABSTRACT

This paper refers to an improvement of the lubrication in a rolling piston type rotary compressor. The lubrication conditions of lubricated surfaces are examined in running compressor and some parameters of them are numerically analyzed. It is elucidated from the investigation that,when unfavorable condition occurs,the metallic contact between the journal and the bearing becomes severe, and sliding velocity at the vane-tip also gets so high,causing extremely high surface temperature of the vane-tip.

It is concluded that it is very important to select proper materials for the bearings ,vane and rolling-piston. Then the materials of the above-mentioned parts are evaluated with two types of friction and wear testers,and selected. The selected materials are applied to an actual compressor. As the result, the compressor is successfully improved in durability.

INTRODUCTION

In general,the rolling piston type rotary compressor are widely used in small air conditioning units because they satisfy the requirements, such as compact size, light weight,good performance and so on.

Recently this type of compressor is also coming into use for heat pump system, while requiring larger capacity and endurance against higher revolution in a variable frequency system. When heat pump is running, in addition to high temperature and discharge pressure, violent liquid return to the compressor causes lack of oil film. On the other hand,in case of larger compressor ,sliding velocity of each part in the compressor increases and also misalignment takes place. Furthermore,in a variable frequency system, lack of oil film occurs at low frequency,and the temperature at sliding surfaces increases rapidly at high frequency.

Thus, the discussions of the lubrication problem,which were reported previously(1,2) ,have become increasingly important in designing of the rotary compressor.

The objective of the present study is to investigate its lubrication condition and select proper materials in order to improve durability of the compressor.

In this paper,the lubrication was studied systematically based on the manner suggested by Czichos and Salomon (3). The lubrication system of the rotary compressor can be clasSfied as shown in Fig.1.

As indicated in Fig.1,we use the term "tribo-..." according to (3). The "tribo-element" is one of the basic elements in the lubrication system, which consists of three groups,-"input","structure" and "output". Load,velocity,temperature,etc. are presented as the "inputs". If one of the "inputs" reaches to a critical level, the failure would occur, causing sudden transitions in the "outputs" as friction, wear, surface temperature,metallic contact or noise. The "inputs" differ only with the operating parameters of the system(air conditioning unit),and the "outputs" present the characteristics of the "tribo-element". In order to clarify the mechanism of the "tribo-element" ,some parameters in Fig.1 were measured and others were numerically analyzed.

The measurement were carried out with some

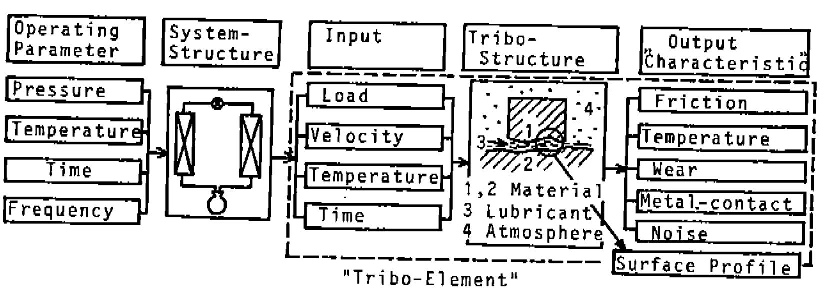


Fig.1 Lubrication system of rotary compressor

methods-visual observation of inner cylinder, electric resistance between lubricating surfaces.

On the other hand, the numerical analysis was carried out,taking boundary lubrication at an eccentric bearing into account, while the previous study (2,4) did not refer to such condition at the eccentric bearing.

Moreover,in selecting the proper materials,two types of friction and wear testers were constructed. The manner shown in Fig.1 was also applied to these test on the evaluation of the materials.

CLARIFICATION OF LUBRICATION CONDITIONS

The lubrication condition of journal bearings and vane-tip were investigated with two types of modified compressors in Fig. 2 (a),(b). 2(a) was used for the observation of metallic contact, bearing temperature , system behavior, vibration and their functional behavior. And the behavior of the metallic contact of each part was examined with electric resistance methods. 2(b) was provided for the visual observation of rolling-piston's motion and the behavior of gas or oil flow with high-speed-camera. Fig.3 shows a typical snapshot of inner cylinder.

(1) For the journal bearings

As shown in Fig.2,there are three different bearings in rolling piston type rotary compressor, such as front bearing (or main bearing), rear bearing (or sub-bearing) and eccentric bearing. Vibration and temperature were also measured. The surface shape and roughness were measured with surface-profile-meter just before and after the run. The concentration of refrigerant 22 in oil was investigated in the running compressor and the oil viscosity was determined from it(See ASHRAE HANDBOOK,1976).

The left sise of Fig.4 shows the behavior of the lubrication condition when the heat pump started after being stopped through the night ("cold start"). It is found,in this case,that the quantity of refrigerant in the evaporater was more than that in the compressor before the "cold start". It is supposed because,without sunlight,the pressure difference between the condenser and the compressor was not so large enough to make the refrigerant flow through the small clearances in the compressor.

When the compressor started under such condition, the refrigerant in the condenser flowed back into the accumulater. Then,the refrigerant overflowed into the compressor and diluted the oil as shown in Fig.4. Accordingly the diluted oil made metallic contact of each bearing worse because of lowed viscosity.

The right side in Fig.4 shows the behavior during defrosting cycle. We had thought it impossible to avoid metallic contact during defrosting cycle as in the "cold start". However,the liquid refrigerant hardly flowed into the compressor in this cese, as found from the behavior of refrigerant 22 concentration in Fig.4. And the such behavior was also seen in the other air conditioning

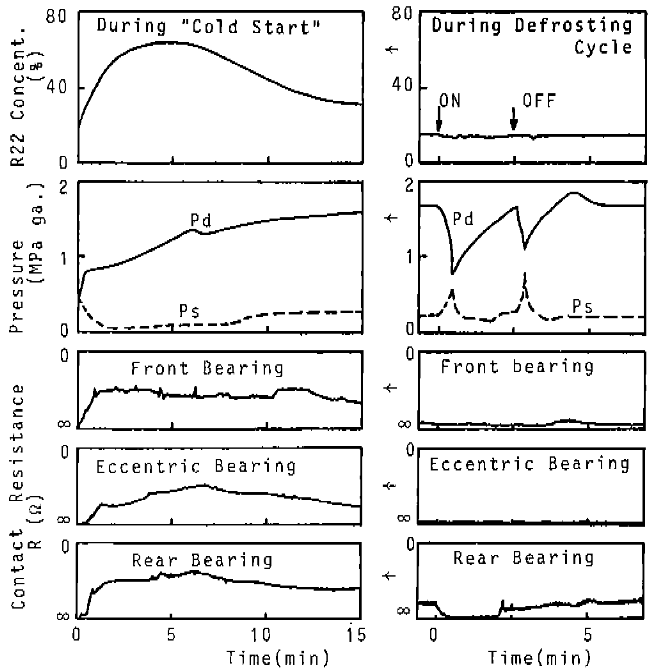
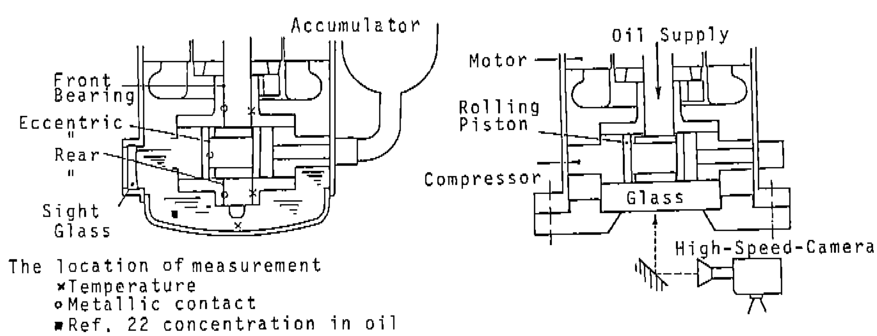


Fig.4 Lubrication condition under running conditions



(a) (b)

Fig.2 Modified compressors for measurement

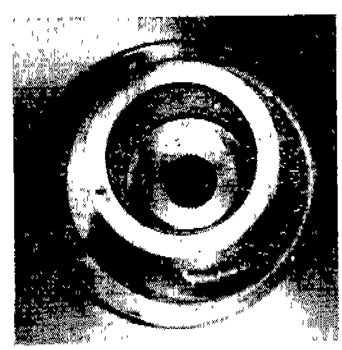


Fig.3 Inner cylinder

units. On the other hand, under running conditions of high oil temperature and high discharge pressure, the degree of the metallic contact were worse.

It was observed that it is hardly possible to avoid metallic contact completely when unfavorable running conditions occur, such as violent liquid return, high temperature and discharge pressure. Therefore, it is important to select the proper materials which can endure these bad condition. This point will be discussed later on.

(2) For the vane-tip

It is supposed that the vane-tip is under boundary lubrication condition as understood from the observation of metallic contact(1) and that of the coefficient of friction(4), since the film thickness calculated by Dowson-Higginson's equation at the part is no more than 0.1 micron.

The surface temperature may be most significant factor in the characteristics of "tribo-element" in present case, because the effect of lubricant depends on the temperature under such boundary lubrication condition. The coefficient of friction was measured to be within the range 0.12-0.17 in previous study(4).

As for the "inputs" of Fig.1, the load was calculated, and the bulk temperature was measured. Moreover, the sliding velocity was evaluated with the equation of rotational speed of the rolling-piston, which can be written as

$$I_p\dot{\omega} = \mu_e W_e - \mu_v r W_v - M_t \quad ---(1)$$

Where I_p, ω, r are the inertia, angular speed and outer radius of the piston, μ_e and μ_v are the coefficient of friction at eccentric bearing and vane-tip, W_e and W_v are the load on eccentric bearing and vane-tip, M_t is the friction moment of thrust bearing (rolling-piston face). In eq.(1), M_t is negligible, W_e and W_v are determined from the pressure condition and μ_v is slightly changeable as stated above.

On the other hand, in evaluating the μ_e, the lubrication condition of the eccentric bearing were observed in connection with piston's motion. Fig.5 shows the behavior of metallic contact at the eccentric bearing, the motion of rolling-piston and the trajectory of the piston. Under unfavorable running condition, the metallic contact is found to be severe at the region around shaft angle 360 deg. Where the partial breakdown of oil film must occur and the friction becomes large, so that the revolution of the piston is accelerated. The trajectory of rolling-piston in Fig.5 indicates the result of the accelerated revolution of rolling-piston.

With increasing in sliding velocity and load under boundary lubrication conditions, the suface temperature at rubbing parts increases. Fig.6 present the relation among velocity, load and temperature at vane-tip during one revolution. These temperatures were calculated by Archard's theory(5), where the coefficient of friction at vane-tip was assumed to be as 0.15-0.17. (Archard's theory will be discussed in detail later on.) These temperatures within the range 200-300 °C seem to be the dangerous level, since it is said that the lubricant failures occur around the temperature 200-300°C and the film wear rate increases rapidly due to the absence of lubricant effects.

The improvement of such lubrication, therefore, would be achieved either reducing this temperature or using proper meterials to endure this high temperature. In the present paper, the latter case will be discussed later on.

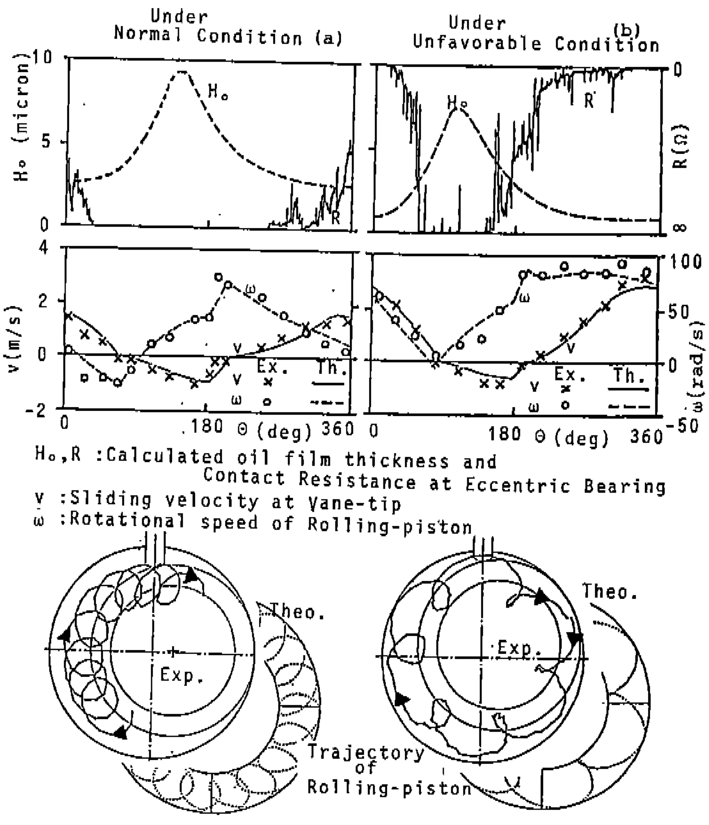


Fig.5 Behavior of rolling-piston and eccentric bearing under running conditions

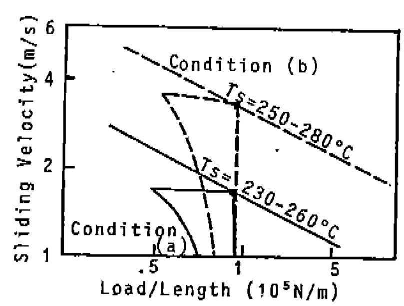


Fig.6 Orbits of load, velocity and temperature at vane-tip

EVALUATION OF MATERIALS
WITH FRICTION AND WEAR TESTERS

In evaluating the proper materials, much attention was paid to the evaluating test with friction and wear testers. The previous manner indicated in Fig.1 was applied to this test in order to understand the correlation between the results for the evaluating test and that for an actual compressor.

(1) For the bearing materials

The evaluation of characteristics,such as compatibility,conformability, embeddability and so on,was carried out using the tester of the stepwise loading types. Fig.6 present the relation between the scuffing load and refrigerant concentration. The results show that the refrigerant atmosphere is most sensitive to scuffing behavior. In the refrigerant atmosphere the scuffing load for nonferrous metal bearings decreases with the increase of refrigerant concentration. On the contrary,the scuffing load for ferrous metal bearings have a tendency to increase. It may be the reason why the chloride film formed by chemical reaction has considerable effect on the bearing ability in the case of ferrous metal bearings as reported by Murray *et al.* (6).

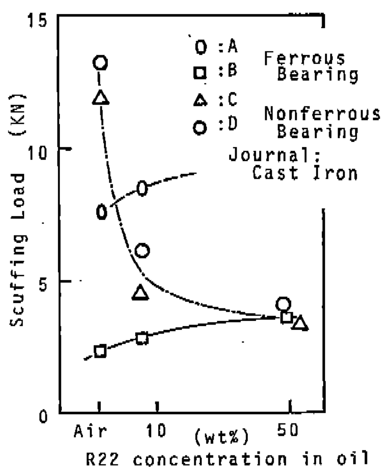


Fig.7 Scuffing load in refrigerant atmosphere

During the run ,"outputs"(friction torque ,temperature beneath a bearing, metallic contact) were also measured. Fig.8 shows the typical results for the "outputs",where the reasonable transition can be seen in the metallic contact and the friction torque at the load W=2940 N ,since the film thickness H_{min} calculated is nearly equal to the surface roughness R_m (r.m.s of interacting surfaces) The friction torque and the contact resistance R for specimen A were recovered quickly from such transition state although those for specimen D remained high until the scuffing occured at the load,2940

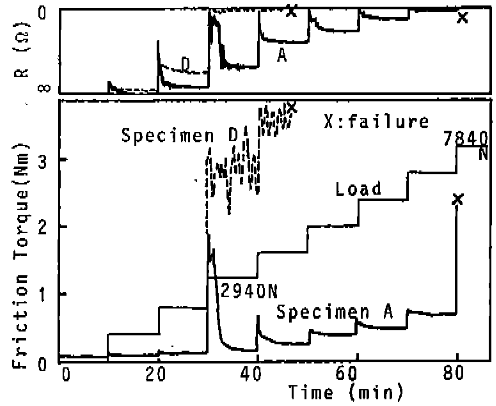


Fig.8 Lubrication condition during friction and wear test

-3920 N. Specimen A failed at 6860-7840 N. Thus,based on the present data,it is confirmed that in refrigerant atmosphere specimen A are surpassed both in compatibility and conformability in the bearing materials under our consideration.

To apply the same evaluation in an actual compressor,the testing compressors with the bearing of specimen A or D were constructed and the lubrication conditions were compared. The data in Fig.9 show that the lubrication condition for specimen A is fairly good compared with that for specimen D even in the actual compressor.

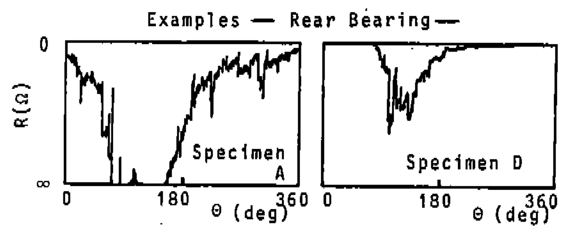


Fig.9 Difference between specimen A and D under running condition

(2) For the vane and rolling-piston materials

In the present evaluation,the tester of the circumscribed circle type was used,and the bulk temperature and the friction force were continuously recorded with stepwise loading at constant velocity.

The surface temperatures were estimated by modified equation deduced from Archard's theory(5). In case of point contact,many reports(7) suggested that the critical surface temperatures calculated by Archard's equation showed a good agreement within the range,500-650°C. To apply Archard's theory to the present problem, the determination of $a_\circ$ ($a_\circ$ is a contact distance in sliding direction) in Archard's equation was modified according to Tsukada and Yanagi(8). In their study, the band width of the contact area is weakly dependent on the hardness, since elastic deformation becomes larger as the hardness of material is high, while plastic deformation becomes larger as the hardness is low. And the $a_\circ$ can be written as

$$a_\circ = \alpha(R_m, W_1) a_h \qquad \text{---(2)}$$

Where R_m is surface roughness, $a_\circ$ is Hertzian deformation and W_1 is the load per length.

Using the $a_\circ$ and assuming $\alpha(R_m, W_1) \cong 1.0$, the surface temperature rise at the rolling-piston ΔT_p gives

$$\Delta T_p = 0.435\ N\ L^{\frac{1}{2}} \qquad \text{---(3)}$$

Where $N = \pi \mu_v P_y g/(J\rho c)$, $L = a_\circ V/2k$, P_y is the yield pressure, J is the mechanical equivalent of heat, ρ is the density, c is the specific heat, V is the sliding velocity , k is the thermal diffusivity, g is the acceleration due to gravity and μ_v is the coefficient of friction.

The surface temperature rise of vane-tip ΔT_v is estimated from the solution for 2-d thermal conduction heat problem. Finally the surface temperature T_s of the lubricated surfaces at this part is estimated as

$$T_s = T_b + \Delta T \qquad \text{---(4)}$$

according to (5).
Where $1/\Delta T = 1/\Delta T_p + 1/\Delta T_v$ and T_b is the bulk temperature. The scuffing points for the present tests are plotted in fig.10 , in connection with the calculated surface temperature.

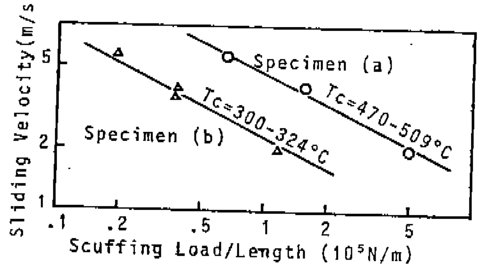


Fig. 10 Scuffing load,velocity and surface temperature

The data show that the scuffing would occur as T_s attains to T_c (T_c is the scuffing temperature) , and also T_c depends on the characteristic of the material including lubricants.

Therefore it could be said that the estimation of the scuffing temperature T_c is a useful manner on the evaluation of materials used under boundary lubrication condition.

As an example for the evaluation in an actual compressor, the surface profiles of the vane-tip and the rolling-piston after the durability test are shown in Fig. 11. Specimen (a) conformed even under extremely sever conditions, but specimen (b) suffered the violent scuffing.

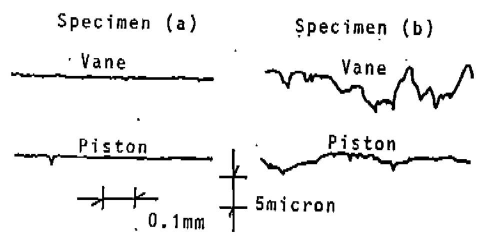


Fig. 11 Surface profile after durability test

Incidentally thrust bearings may be also evaluated in the same manner. In this case, however, the estimation of contact distance is more difficult to treat, so that a reasonable theory cannot be suggested at present. According to our experiment, however, it is confirmed that the relation between the scuffing load W_c and the scuffing temperature T_c gives equation (5) as shown in Fig.12.

$$T_c \cong W_c{}^{\alpha} V^{\beta} \qquad \text{---(5)}$$

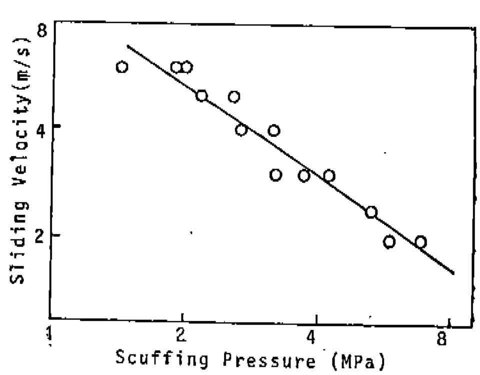


Fig.12 Scuffing pressure and velocity

CONCLUSION

1. The lubrication conditions in running compressor were investigated:
   1-1 It is impossible to avoid metallic contact of journal bearings when unfavorable running conditions,such as violent liquid return,high temperature and high discharge pressure,occur.
   1-2 Under the unfavorable lubrication condition for eccentric bearing,the revolution of rolling-piston is accelerated. And it results in higher sliding velocity and higher temperature of vane-tip, which causes the failure of lubricant and high wear rate.

2. Friction and wear testers were effectively used on the selection of materials. The measurement of the parameters in the characteristics of "tribo-element" during such test is very important to understand the correlation between the result with the friction-wear-tester and that with an actual compressor.

3. The estimation of the surface temperature is a useful manner to evaluate materials used under boundary lubrication condition ,such as vane-tip.

REFERENCES

1. Ozu,M. and Itami,T. , Proc.Purdue Comp.Tech.Conf.,1980,pp.105-111

2. Okada,K. and Kuyama,K. , Proc.Purdue Comp.Tech.Conf.,1982,pp.178-184

3. Czichos,H. and Salomon,G. , JSLE-ASLE Int.Lubri.Conf.,1975,pp.428-441

4. Yanagisawa,T. *et al.* , Proc.Purdue Comp.Tech.Conf.,1982,pp.185-192

5. Archard,J.F. , Wear 2,1959,pp.438-455

6. Murray,S.F. *et al.* , Mechanical Engineerings,1956,pp.233-236

7. Sethuramiah,A. *et al.* , Wear 26,1973,pp.187-206
   or
   Begelinger,A. and de Gee,A.W.J. , Jour.Lubri.Tech.,1976,pp.575-579

8. Tsukada,T. and Yanagi,K. , Jour.JSLE 26,1981,pp.34-40

NOMENCLATURE

- a_o contact distance at vane-tip in sliding direction
- a_h contact distance for Hertzian deformation
- g acceleration due to gravity
- H_o thickness of oil film
- H_{min} minimum thickness of oil film
- I_p inertia of rolling-piston
- J mechanical equivalent of heat
- M_t friction moment of thrust bearing
- P_d discharge pressure
- P_s suction pressure
- r outer radius of rolling-piston
- R contact resistance at lubricating surface
- R_m surface roughness (r.m.s of interacting surfaces)
- T_b bulk temperature
- T_c scuffing (or critical) temperature
- T_s surface temperature
- ΔT temperature rise of surface
- ΔT_p temperature rise of rolling-piston's surface
- ΔT_v temperature rise of vane
- v sliding velocity
- W_c scuffing (or critical) load
- W_e load on eccentric bearing
- W_l load per lengh on vane
- W_v load on vane
- c specific heat
- k thermal diffusivity
- Θ angular position of shaft
- μ_e coefficient of friction at eccentric bearing
- μ_v coefficient of friction at vane-tip
- ρ density of material
- ω rotational speed of rolling-piston