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RESEARCH

A Bearing Cartridge Assembly for Long-Term Performance of Momentum/Reaction Wheels used in Spacecraft

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A B S T R A C T

Momentum/reaction wheels are used in a spacecraft for the attitude control. They are used for the real time control of spacecraft attitude towards the desired direction as required by the mission. These are one of the few systems of a spacecraft which need to work continuously from the beginning to end of the mission. Thus the life of a space mission depends largely on the performance of the attitude control wheels. The common mode of failure in momentum/reaction wheel is mechanical and mostly related to the bearing and lubrication. This paper presents the design and development of a bearing cartridge assembly including the lubrication system for long-term performance of spacecraft attitude control wheels. The results of both theoretical and experimental analysis are presented. The experimental study of the assembly shows that it can perform continuously for a period of 30 years with reduced frictional losses.

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1. INTRODUCTION

Spacecraft, irrespective of the functional requirement, contain a number of electro-mechanical assemblies. There are oscillating, sliding and rotating systems. Also, there are high speed systems such as momentum/reaction wheels, gyroscopes etc. and low speed systems such as scanners, solar array drives etc. All these systems use ball bearings of different size. Most problems encountered with these moving systems are problems of tribology and, specifically, lubrication [1]. The friction and wear performance in space is very much different from the terrestrial systems, and therefore extensive on-orbit experiments are conducted using on-board tribometers to suggest new materials for future space tribosystems [2].

Momentum/reaction wheels are spacecraft actuators used for control and stabilization of spacecraft attitude to the required accuracy [3]. A momentum wheel (MW) has large angular momentum of 50 to 200 Nms and spins constantly in one direction with typical speeds between 3,000 and 12,000 rpm [3-5]. These are momentum exchange devices that work by the principle of conservation of angular momentum. The satellite and the momentum wheel system have an angular momentum equal to the sum of individual angular momentum and it is constant at all times provided there are no external disturbances on the satellite. The torque produced by changing the speed of rotation of the wheel is used to turn the satellite to the required direction. Since the inertia of the satellite is large compared to the inertia of the wheels, a very precise control of the satellite orientation is possible with momentum wheels. A reaction wheel (RW) is similar in construction as momentum wheels and they are particularly useful when the spacecraft must be rotated by very small amounts. Reaction wheels create torque on a spacecraft by applying an opposite torque on a flywheel, changing its spin rate. By accelerating or decelerating the speed of the wheels, the motors may apply a torque on the spacecraft in either direction about the axis of the wheel. Generally, reactions wheels are of low angular momentum capacity (typically 0.1 to 5 Nms) and operating speed of ±3000 rpm.

The internal detail of a typical momentum wheel is shown in Fig. 1 [6]. A momentum wheel essentially consists of a flywheel which is driven by an electric motor, generally, a brushless dc motor, its precise rotation about a fixed axis is ensured by mounting it over a bearing unit consisting of a pair of high precision angular contact ball bearings. The speed of the fly wheel is controlled through a drive electronics circuit. All these components are enclosed in a hermetically sealed metal casing purged with an inert gas. Usually the internal pressure is less than atmospheric, typically 15 torr [6].

Tribological failures of momentum/reaction wheels are related to lubricant breakdown, loss of lubricant due to evaporation and surface migration (insufficient lubricant) and retainer instability [3]. Lubricant breakdown failure occurs when the original liquid lubricant is chemically changed to solid friction polymer [7]. The working temperature, which is also a function of bearing friction torque, causes the lubricant to evaporate. The oil loss by migration is induced by temperature gradients and capillary forces. Retainer instability is the most dangerous mode of failure in momentum wheel bearings. It is characterized by large variation in bearing friction torque associated with severe audible noise. Uneven cage wear, lubricant degradation and insufficient lubrication are the prime causes for it. The retainer instability is related to number of factors like geometry and mass of the retainer, operating speed, lubricant quantity, etc. [8–13]. Therefore, with the selection of proper lubricant and proven retainer design, lubrication remains the principle life limiting problem on momentum wheels.

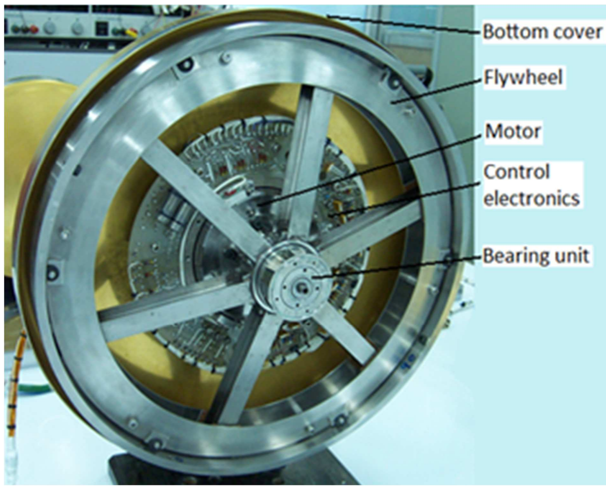

Fig. 1. Momentum wheel with top cover removed [6].

Momentum/reaction wheel bearings are generally lubricated with liquid lubricants. Since the bearings are required to operate with minimum frictional power loss, it is preferred to operate in the elastohydrodynamic (EHD) lubrication regime (see Section. 2.6). As mentioned above, due to the progressive loss of lubricant from the bearing contact surfaces, it become necessary to supplement the lubricant to ensure the long-term efficient functioning of the wheels. Also, the lubricant supply rate should be the least to avoid abnormal increase and variations in bearing friction. This paper presents the design of a bearing cartridge with lubricant supply system for long-term performance of momentum/reaction wheels. In this paper, the design details of the bearing cartridge are presented in Section. 2 and the design features of the lubrication system are presented in Section. 3. The experimental results are given in Section. 4.

2. DESIGN OF BEARING CARTRIDGE

The design features of a momentum wheel bearing cartridge typically used in communication satellites is presented here. The bearing cartridge is generally made of high quality steel (AISI 440C) to ensure high strength and dimensional stability. Usually, the bearing cartridge components are made of material same as the bearings to eliminate the effects of thermal stresses, because in service the wheels are subjected to wide ranges of temperatures. A typical momentum wheel used in communication satellite has angular momentum around 60 Nms at normal operating speed of 5,000 – 8,000 rpm [6,7,14,15]. A typical flywheel mass of 5 kg is assumed. In order to design the bearing cartridge and lubricator for long-term operation of a momentum wheel, it is essential to know the life of the bearing and the lubrication requirements. Also to ensure a long life to the bearings, it is essential that the fatigue life of the bearings should be higher than the mission life expected. The bearing contact stress is also important in determining the life of the bearings. Since the momentum wheel is required to operate with least possible bearing friction, it is necessary to know the friction torque of the bearing under consideration. Similarly, the film parameter is important to check whether the bearings are working in the elastohydrodynamic (EHD) region because the minimum frictional losses occur only when the bearings are operating in the EHD regime or below. The design features and important parameters of the bearing are presented in the following sub-sections.

2.1 Bearing Cartridge Features

The bearing cartridge developed in this project contains of a pair of high precision angular contact ball bearing. These bearings are assembled on a flanged hollow shaft. The flanged end of the shaft forms the interface with the base of the momentum wheel. To avoid radial play, transition fit (0-3 μm) is given between the bearing and the shaft. The bearings are mounted in back-to-back (DB mounting) arrangement to give good alignment and high moment rigidity. To increase the loading span, the bearings are separated by a set of spacers (outer and inner) of equal length. The length of the bearing spacer is dependent of the overall length of the bearing unit which in turn depends on the height of the stator and rotor of the drive motor. In this case, the length of the spacer considered is 50 mm. The outer spacer is in contact with the outer race of the bearing and the inner spacer is in contact with the inner race of the bearings as shown in Fig. 2. The bearings and the spacers are enclosed in housing and are held in position by means of a set of nuts – housing nuts and shaft nut. The bottom bearing is supported on a belleville spring, which is designed to have the resilience to protect the bearings from the launch vibrations. On the housing, a flange is provided which forms the interface with the flywheel and the rotor of the motor.

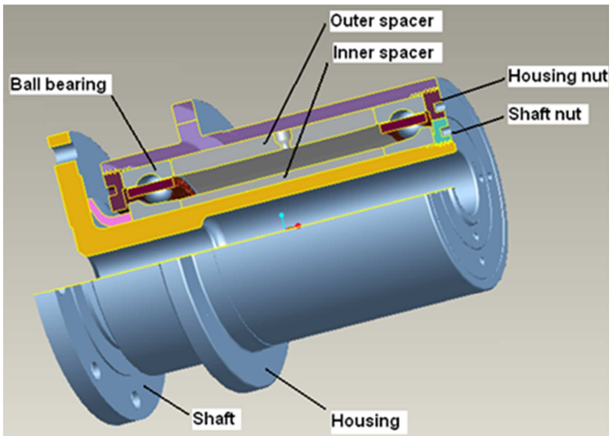

Fig. 2. Bearing cartridge assembly.

In this design, the shaft is stationary and the housing is rotating. Since angular contact ball bearings are used in the assembly, it needs to be preloaded to avoid the internal clearance [16-18]. Preloading will eliminate the axial and radial play, increase the system rigidity and prevent ball skidding during high accelerations of the system. In this case, a preload equal to the total radial load (50 N) is applied. The desired preload is obtained from the difference in length of the inner and outer spacers. The spacer length difference required to achieve the preload is obtained by loading each bearings separately by a known load (for example,50 N) and the stick out of the inner race with respect to the outer race is measured. The net amount of stick out of both bearings is the difference in spacer length required for the preload (50 N).

2.2 Bearing Selection

Bearings for the cartridge assembly are selected considering the expected maximum loads acting on it. The major load on the bearings is due to the vibration of the flywheel and motor rotor assembled on it during launching of the spacecraft. A higher design safety margin is considered and is usually greater than 5 [6]. Therefore, a 20 mm bore (104 size) angular contact ball bearing is selected, which is the commonly used size in medium capacity (65 Nms) momentum wheels [3,19]. The tolerance class is ABEC 9P, which is the highest precision class commonly used for aerospace application. The load applied on each bearing are: 50 N axial load applied as the preload and 25 N radial load due to the weight of the flywheel mounted on the bearing unit (the flywheel weight (50 N) is shared equally by two bearings). The load ratings of the bearing are: static radial 4,806 N, static axial 12,262 N and the dynamic capacity is 7,337 N [6].

2.3 Bearing contact stress

In a ball bearing, the shape of the contact surface where the ball touches the outer/inner race is elliptical. The maximum bearing contact stress (Hertzian contact stress) occurs at the geometrical center of the contact ellipse. This is the localized stresses that occur as the two curved surfaces of the bearing contact and deform slightly. The magnitude of maximum contact stress (σ_{max}) can be determined by Eq. (1) [16,17];

$$\sigma_{\max} = \frac{3Q}{2\pi a b} \tag{1}$$

where, a and b are semi major and semi minor axis of the contact ellipse and Q is the maximum load per ball. Using Eq. (1), the Hertz contact stress for the bearings are calculated based on 50 N axial load is 848.2 N/mm^2 for the inner race and 636 N/mm^2 for the outer race contacts (since the gravity is negligible in space, the effect of radial load is also negligible). The results of the calculation show that the maximum contact stress is 848.2 N/mm^2 which occurs at the inner race-ball contact point. It is clear that the obtained stress value is much lower than the yield strength of the material of the bearing (AISI 440C steel), which is about 1,800 N/mm^2 [18].

2.4 Bearing Fatigue Life (L_1 life)

Fatigue in a bearing is caused by the dynamic stressing of the bearing material during operation. The time at which fatigue occurs depends on the stress level. In a well-lubricated ball bearing, failure is mainly due to fatigue. Therefore the fatigue life is considered as the life of a bearing [17]. The fatigue life of the bearing (L_1 and L_{10} life) is calculated using the modified Lundberg and Palmgren [20,21] equation;

$$L_n = a_1\, a_2\, a_3 \left(\frac{C}{Q}\right)^3 \times 10^6 \;\; revolutions \tag{2}$$

where a_1 is the life adjustment factor for failure probability, a_2 is the life adjustment factor for material, a_3 is the life adjustment factor for operating conditions, C is the dynamic load capacity (7,400 N) and Q is the equivalent dynamic load, $Q = (\, X\; j\; Q_R + Y\, Q_A\,)\, s$; here X and Y are the radial and axial factors (in this case, $X = 0.44$ and $Y = 1.54$), Q_R and Q_A are the radial and axial loads on the bearings, s is the impact factor (1.0 for constant or steady load) and j is the race rotation factor (1.2 for outer ring rotation). Using Eq. (2), the fatigue life of the bearings with 99 % (L_1 life) and 90 % (L_{10} life) reliabilities are obtained as 41 and 194 years. This means that under normal operating conditions, the failure of bearing unit will not occur due to fatigue and it may occur due to other factors like lubricant starvation, contamination, excessive vibration caused by imposed load etc.

2.5 Bearing Friction Torque

A reasonable estimate of the friction torque of a ball bearing under moderate load and speed conditions is the sum of the load friction torque, viscous friction torque and friction due to spinning motion of the balls about an axis normal to the race way contact area;

$$M = M_L + M_V + M_S \tag{3}$$

where M_L is the torque due to applied load, M_V is the torque due to lubricant viscosity and M_S is the torque due to spinning. The load-induced friction can be calculated from the Palmgren equation [20];

$$M_L = f_1\, F_\beta\, d_e \tag{4}$$

where f_1 is a factor depending upon the bearing design and relative bearing load, F_β is a factor depending on the magnitude and direction of the applied load, and d_e is the pitch diameter of the bearing.

The viscous friction can be calculated from the equation [20]:

$$M_V = 10^{-7} f_o\, (v_o n)^{2/3} d_e^3 \qquad v_o n \geq 2000 \tag{5}$$

$$M_V = 160 \times 10^{-7} f_o\, d_e^3 \qquad v_o n \leq 2000 \tag{6}$$

where f_0 is a factor depending upon the type of bearing and method of lubrication, ν_0 is the lubricant viscosity in centistokes (cSt) and n is the rotational speed.

The magnitude of the spinning torque can be determined from the equation given in [17]:

$$M_S = \frac{3\, f_c\, Q\, a\, \varepsilon}{8} \qquad (7)$$

where ε is the complete elliptical integral of the second kind with modulus $\sqrt{1-(b/a)^2}$, f_c is the friction coefficient, q is the load on the bearing, in this case the axial load 50 N which is the maximum load acting on the bearing, and a and b are the semi major and semi minor axis of the contact ellipse.

The frictional torque of the momentum wheel bearing unit operating at 40 °C and 5,400 rpm is calculated by using Eq. (3) to (7). The components of the friction torques obtained are: torque due to applied load 1.864 mN-m, viscous torque 8.638 mN-m and torque due to spinning of balls 0.204 mN-m. The lubricant selected for the calculation of viscous friction is Kluber Isoflex-PDP 65, synthetic diester oil whose properties are given in Table 1.

Table. 1. Properties of kluber PDP 65 oil [22].

Base stock	Synthetic diester oil
Viscosity at 40 °C at 100 °C	73 cSt 16 cSt
Viscosity index	235
Pour point	-60 °C
Flash point	230 °C
Density	0.915 gm/cc
Surface tension	30 Dyne
Vapor pressure at 20 °C at 100 °C	7.8×10^{-8} Torr 7.6×10^{-4} Torr
Minimum shelf life	20 years

The estimated total friction torque of the bearing unit is 10.71 mN-m. However, the actual friction torque in a momentum wheel bearing unit will be less than this and is usually about 5 mN-m [19]. It is observed that the component the friction due to the viscosity of lubricant forms the major part. This part of the friction in a momentum wheel bearing is reduced to a minimum by operating the bearings in the preferred EHD or boundary lubrication regimes.

2.6 Bearing Lubrication Parameters

The purpose of lubrication is to separate two surfaces sliding past each other with a film of some material which can be sheared without causing any damage to the surfaces [23]. A lubricant is any substances that are used to reduce friction and wear and to provide smooth running and satisfactory life for machine components [24]. The introduction of a film of lubricant between the components with relative motion forms the solution of a vast number of tribological problems in engineering. In many cases, the viscosity of the lubricant and the geometry and relative motion of the surfaces may be used to generate sufficient pressure to prevent solid contact without any external pumping agency [25]. Depending on the thickness of the lubricant present between the interacting surfaces, several distinct lubrication regimes are identified by a number of researchers. One way of depicting these regimes is by the use of Stribeck curve shown in Fig. 3 [24,26]. Depending on the characteristics, they are divided in to four distinct regimes and are named as: hydrodynamic or fluid-film lubrication, elastohydrodynamic (EHD) lubrication, boundary lubrication and mixed lubrication.

Stribeck originally devised a convenient way of relating roughness of interacting surfaces and lubricant film thickness by a parameter $\boldsymbol{\lambda} = \frac{\boldsymbol{film\ thickness}}{\boldsymbol{roughness\ height}}$, the roughness height being some representative value of the undistorted roughness features, and the film thickness is being measured from it. He devised the Stribeck curve where the coefficient of friction ($\boldsymbol{\mu} = \frac{\boldsymbol{friction\ force}}{\boldsymbol{load}}$) is plotted against λ, which explains various lubrication regimes as shown in Fig. 3.

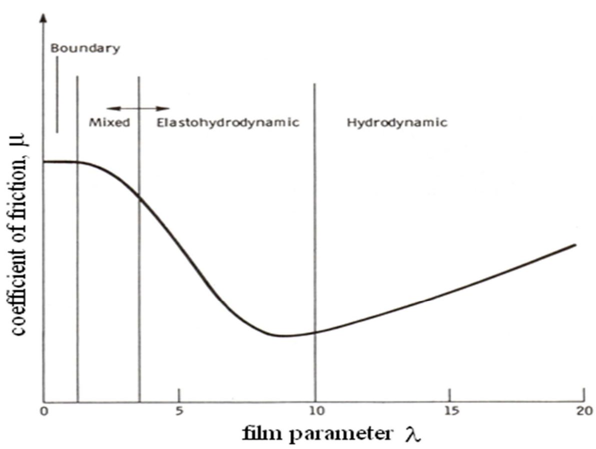

Fig. 3. Stribeck curve showing lubrication regimes.

Since the value of the film parameter λ determines the working regime of lubrication, it is essential to estimate this for the momentum/reaction wheel bearings. The lubricant film thickness required for calculating the film parameter can be estimated by using the equation derived by Hamrock and Dowson [24]:

$$\tilde{H}_{min} = 3.63 U_s^{0.68} G^{0.49} W^{-0.073} \left(1 - e^{-.68k}\right) \quad (8)$$

where $\tilde{H}_{min}$ is the dimensionless minimum EHD film thickness, U_s is the dimensionless speed parameter, G is the dimensionless material parameter, W is the dimensionless load parameter and k is the ellipticity parameter
The non-dimensional film parameter λ can be found from the equation [24]:

$$\lambda = \frac{h_{min}}{\left(s_r^2 + s_b^2\right)^{1/2}} \quad (9)$$

where $h_{min} = \tilde{H}_{min} \times R_x$ is the minimum film thickness, R_x is the effective radius in x-direction, s_r is the r.m.s surface finish of the bearing race and s_b is the r.m.s surface finish of the bearing balls. The value of λ indicates exactly the lubrication regime of a bearing under the prevailing operating conditions. Also, it provides the thickness of the lubricant film at the bearing contact zone. In momentum/reaction wheels, as mentioned above, the bearings are preferred to operate in the elastohydrodynamic (EHD) regime. Elastohydrodynamic lubrication is a form of fluid film lubrication where the elastic deformation of the bearing surfaces becomes significant. EHD lubrication is characterized by very low friction torque as seen from Fig. 3.

The calculated value of lubricant film thickness h_{min} using Eq. (8), for momentum wheel bearings operating under 50 N axial load at 40 °C is 0.628 µm at the inner race contact and 0.76 µm at the bearing outer race contact For this calculations, the viscosity at 40 °C (0.06225 N-s/m^2) is used. The film parameter λ for the above condition is calculated by using Eq. (9). The calculation shows that the film parameters are 11 and 13 for the inner and outer race contacts, respectively. This clearly shows that the momentum wheel bearings are working near the upper boundary of EHD lubrication regime as can be observed in Fig. 3. This together with the results given in Sec. 2.5 shows that friction torque of momentum wheels during the initial stage of running will be the maximum (10 mN-m) since it will be in the upper boundary of EHD regime. Once the lubricant starts depleting from the adjacent region of contact, the film thickness decreases and as a result, the friction torque also decreases. If no makeup oil is supplied, the decrease in lubricant film thickness continues and direct metal to metal contact occurs gradually. During this transition, the bearing pass through all the four lubrication regimes namely, EHD, parched EHD, mixed and boundary lubrication. This shows the necessity for a supplementary lubrication system for long-term performance.

3. THE LUBRICATOR

This section presents the design of a supplementary lubrication system called *centrifugal lubricator* developed in this work to provide long-term uninterrupted lubrication to the bearings. It is understood that to operate the bearing at a lower frictional power loss, the bearings are to be continuously supplied with least amount of lubricant. Literatures reveal that only 0.2 µg/h flow rate is needed to maintain a continuous film of lubricant at the working surfaces of a bearing of similar size selected in this work [27]. This is a too low value and difficult to achieve practically. Therefore the centrifugal lubricator developed for the bearing cartridge is designed to supply lubricant at the lowest possible rate, preferably less than 10 µg/h, for a period of 30 years. The design features and its working are explained in the following sub-sections.

3.1 Design Features

In the centrifugal lubricator, the mechanism to store the supplementary lubricant is a crucial part of the lubricator developed for the bearing cartridge. The centrifugal lubricator consists of an outer cup and an inner sleeve made of aluminum. The space formed between the outer cup and inner sleeves serves as the reservoir of lubricant. The capacity of the reservoir is about 5 cc. To make the reservoir leak proof, both ends of the assembly are sealed using a space proven adhesive or by electron beam welding. On one of the faces of the outer cup, a small hole is provided to fill the lubricating oil in to the reservoir. Mounting holes are provided on the periphery of the outer cup to attach the lubricator to the rotating outer spacer. A small hole of diameter 150 μm is drilled on the periphery of the outer cup for the lubricant to flow out as shown in Fig. 4. As the lubricant goes out of the reservoir space, it may create a vacuum inside the reservoir. This negative pressure tries to reduce the flow rate by reducing the net pressure head acting at the orifice. This is avoided by drilling a small hole on the wall of the inner sleeve which balances the internal pressure.

The lubricator assembly consists of two lubricators attached to the outer spacer of the bearing unit. The outer spacer of the bearing unit is designed in such a way that it can accommodate two lubricators and can direct the lubricant coming out of the lubricator into the bearings. The bearing unit with the lubricator assembly is shown in Fig. 5. Stress analysis of the lubricator shows a maximum circumferential stress of 2.034 N/mm² for a pressure of 60 kPa which corresponds to 10,000 rpm. The strength of the material used for the lubricator (AA 2024) is 296 N/mm². Hence the design satisfies the safety requirement.

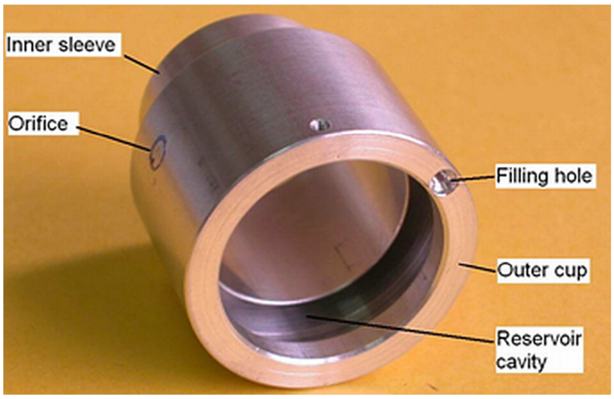

Fig. 4. Centrifugal lubricator.

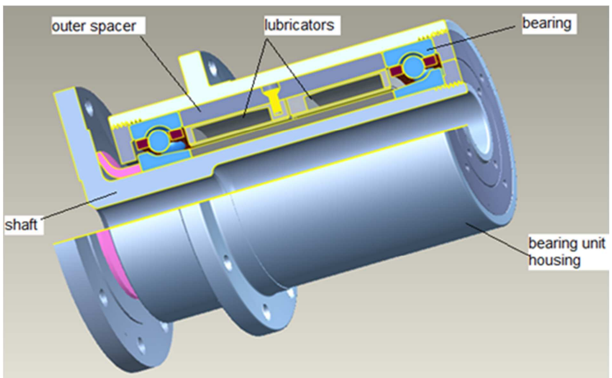

Fig. 5. The bearing cartridge assembly with the lubricators [28].

3.2 Working Principle

In centrifugal lubricators, as the name indicates, the centrifugal force due to the rotation of the bearing unit is utilized for supplying lubricant to the bearings. When the bearing unit rotates, the lubricator attached to the rotating outer spacer of the bearing unit also rotates. This causes an increase in the pressure head of oil filled in the lubricator. The pressure varies along the radial direction and is the maximum at the outer surface where the orifice is located. The estimated maximum pressure at the orifice is about 13 kPa when the reservoir is full and rotating at 5,000 rpm. The pressure causes the oil to flow through the orifice. The oil particles coming out of the orifice are thrown in to the outer spacer by centrifugal force. In the outer spacer, a small taper is provided which causes axial movement of oil film towards the bearings. From the outer spacer, the oil flows into the bearings as shown in Fig. 6.

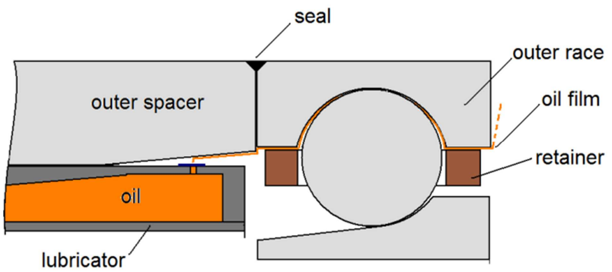

Fig. 6. Schematic view of oil flow from the lubricator.

The interface between the bearing outer race and the outer spacer is sealed to prevent the leakage of oil.

3.3 Flow Rate Control

The oil pressure developed due to rotation of the lubricator at the normal operating speed (5,000 rpm) is estimated to 13 kPa. Under this pressure, it is only a matter of hours to empty the oil filled in the reservoir of the lubricator through the 150 μm orifice. To avoid this and to control the flow rate to the required level, a flow restrictor is incorporated at the orifice. Here, the flow rate is controlled to the targeted value of 10 μg/h using micro/nano orifice created on a metal foil and rigidly mounted over the 150 μm orifice as shown in Fig. 7. The flow control orifice is made using a pulsed laser system.

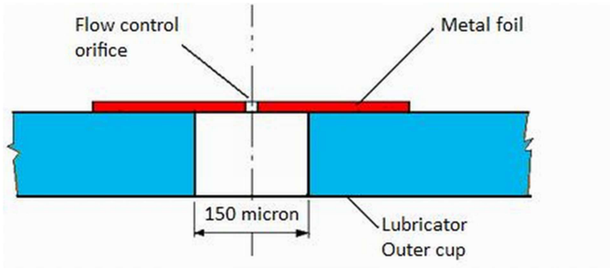

Fig. 7. Arrangement of flow control orifice.

A mathematical model of the lubricator is developed to determine the diameter of the flow control orifice required to obtain the targeted flow rate of 10 µg/h. This model is also useful to predict the life and performance of the lubricator at different operating conditions of the lubricator. The equation for flow rate from the lubricator is [29]:

$$q = K\,\frac{\pi\,\rho^2\,\omega^2\,r^4}{8\eta}\left[\frac{R_3^2 - R_1^2}{R_3 - R_2}\right] \qquad (10)$$

where K is the flow coefficient (0.326), ρ is the density of the lubricant (kg/m^3), η is the dynamic viscosity of the lubricant (kg/m-s), ω is the angular speed (rad/s), r is the radius of the orifice (m), R_1 is the instantaneous radius of oil inner layer in the reservoir (m), R_2 is the radius at which oil enters the orifice (m) and R_3 is the radius at which oil leaves the orifice (m). In this case, R_2 and R_3 are constants and the difference between the two gives thickness of the orifice plate. q is the mass flow rate (kg/s).

The diameter of the flow control orifice to achieve the targeted 10 µg/h flow rate is calculated using Eq. (10). The calculated diameter of the orifice at the normal working environment of 23°C, running at 5,000 rpm, using a lubricant of kinematic viscosity 99 cSt (Kluber ISOFLEX PDP-65 oil) is 0.28 µm. A set of lubricators are made with orifices of different diameters such as 2.2 µm, 2.4 µm and 2.7 µm. These lubricators are then filled with clean and filtered lubricant by vacuum filling process [30]. A series of experiments are conducted to observe the flow rate at different operating conditions of the momentum wheel. The experimental setup is designed to test more lubricators simultaneously as shown in Fig. 8. Experiments are done under vacuum using a thermo-vacuum chamber and the internal pressure is maintained at 375 torr (internal pressure of a typical momentum wheel).

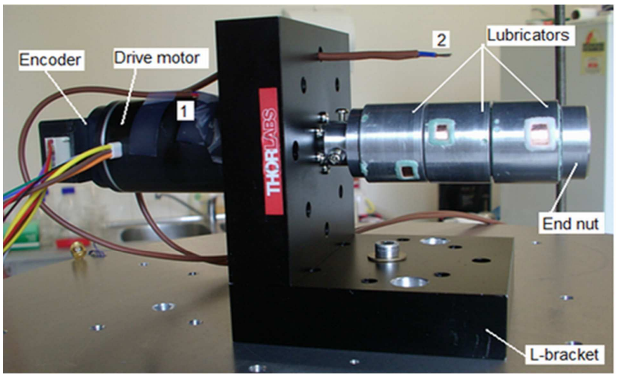

Fig. 8. Experimental setup [30].

4. RESULTS AND DISCUSSIONS

At first, lubricators are subjected to flow rate measurements, during which the lubricators are run at constant a speed of 5,000 rpm at 23 °C continuously for 15 – 20 days. The weight of the lubricator is recorded before and after each run and weight loss (ΔQ) is calculated by subtracting the current measurement from the initial measurement. From the weight loss, the flow rate is calculated by dividing it with the duration (ΔT) of running;

$$Flow\ rate = \ \Delta Q/\Delta T \ \ mg/h \qquad (11)$$

This process is repeated for a number of times to verify the consistency of flow rate. This process is called as calibration. In order to obtain a reliable calibration chart, the calibration is done for a period of 325days. In addition to the calibration test, the following tests are also done to prove the functionality of lubricator at various operating conditions of the momentum wheel.

1) Measurement of flow rate at different operating speeds from 3,000 to 10,000 rpm
2) Flow rate by varying the quantity of oil in the lubricator.

Figure 9 shows the results of the test conducted to verify the flow rate of the lubricator. Here, the theoretical results obtained using Eq. (10) are compared with the experimental data for a lubricator having 2.7 µm orifice. It is seen that the experimental results are very close to the theoretical results. Also seen that the flow rate is almost consistent and it is about 8 µg/h. It is to be noted that the total oil expelled in 7798 h is only 60 mg, whereas the quantity of oil filled in the lubricator is about 5,000 mg. At this initial flow rate, simple calculation shows a life expectancy of more than 50 years.

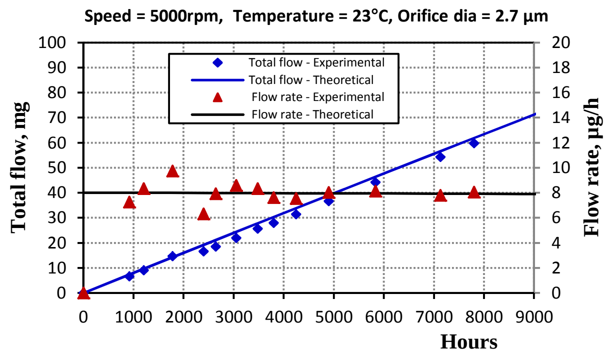

Fig. 9. Theoretical and experimental results of a lubricator with 2.7 µm diameter orifice.

Figure 10 shows the variation of flow rate at different rotating speeds of the lubricator. It shows that when the speed is doubled from 5,000 rpm to 10,000 rpm, the flow rate increased by a factor of 4 (i.e. 8.1 µg/h at 5,000 rpm and 35.84 µg/h at 10000 rpm). This increase is because of the increased oil pressure as a result of higher centrifugal force. The figure also shows that the experiment results are very close to the theoretical predictions. The few wild points observed in the experimental results were due to the combined effect of variations in temperature, chamber vacuum, and error in weight measurement. The results of the speed test confirm the suitability of the lubricator for higher running speeds.

Experiments are also conducted to measure the flow rate with different amount of oil in the lubricator. It is important to know the flow rate at various stages of the life of lubricators to ensure continuous supply of sufficient amount of lubricant to the bearings. It is understood that as the oil flows out of the lubricator, the quantity of lubricant in the lubricator decrease with respect to time. Therefore, at various stages of the life, the quantity of oil left in the lubricator will be different from the initial quantity. The lubricator was first filled with 1 g oil and the flow rate was measured by running at 5,000 rpm. After a few readings were obtained, the quantity was increased by adding oil in steps of 1 g each. The experimental results are compared with the theoretical results as shown in Fig. 11. During the experiment, the operating speed and temperature are maintained at 5,000 rpm and 23 °C. The results indicate that the flow rate is 2.8 µg/h when the amount of oil in the lubricator is just 1gm. It proves that the lubricator still can supply lubricant even when a very small amount of oil remained in the reservoir. This is a promising result and it proves the longevity of the lubricator.

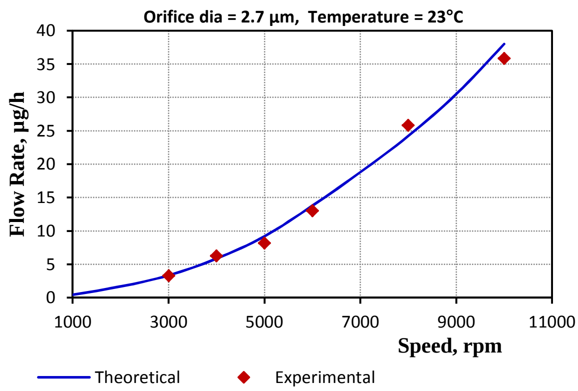

Fig.10. Variation of theoretical and experimental flow rate with operating speed.

The results of the stress analysis of the orifice plate (50 µm thick copper foil) show a maximum tensile stress of 45.1 N/mm^2 at the normal operating speed of 5,000 rpm [20]. At this speed, the maximum oil pressure acting on the plate is about 0.013 N/mm^2. This shows a higher safety factor of 6. Also, at 10,000 rpm, which is double the operating speed, the maximum stress value obtained is 181.72 N/mm^2 and safety margin is 2. This shows that to operate the lubricator at speeds higher than 10,000 rpm, the thickness of the orifice plate has to be increased. However, the results prove that the selected thickness of 50 µm of the plate is sufficient for operation up to 10,000 rpm.

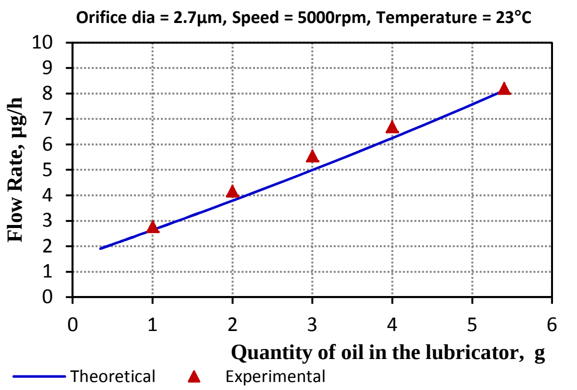

Fig. 11. Variation of theoretical and experimental flow rate with quantity of oil in the reservoir.

In summary, the theoretical analysis done on the bearing cartridge and the theoretical and experimental study on the centrifugal lubricator proves that the assembly will be promising and worthy development in the spacecraft arena. This assembly would be able to solve the bottlenecks in the development of reliable attitude control systems required for the future long-term spacecraft missions.

5. CONCLUSIONS

In this paper, the design of a typical rotating housing type bearing unit and a lubrication system for its long-term operation are presented. The analysis shows that the selected bearing has a fatigue life (L_1 life) of 41 years. This is much higher than the life expectancy of 30 years for the future spacecrafts. Under the normal loading conditions, the calculated contact stress shows a very good safety margin. The calculated film thickness and film parameter shows that the bearings are operating near the upper boundary of EHD regime. Also, the estimated bearing frictional torque proves that the power consumption of the bearing unit will be much lower.

The analysis carried out on the lubricator shows that the design has a very high safety margin under the intended rage of working. The results show that the thickness of the metal foil on which the flow control orifice is created has a higher safety margin under pressure corresponds to a speed of 10,000 rpm. The experimental results prove the suitability of the lubricator for long-term application of longer than 30 years. It also proves its suitability for variable speed operation. The experiment conducted by varying the quantity of lubricant in the reservoir shows that the lubricator bleeds out oil even when the leftover oil in the reservoir is just 1 g. This is a very promising result, which confirms its suitability for future spacecrafts requiring longer mission life.

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