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RESEARCH

The Influence of Natural Frequency of the Experimental Set-up on the Friction Coefficient of Stainless Steel-304

The present paper investigates experimentally the effect of natural frequency of the experimental set-up on friction property of stainless steel-304. To do so, a pin-on-disc apparatus having facility of vibrating the test samples at different directions, amplitudes and frequencies was designed and fabricated. The natural frequency of the set-up was varied by adding dead loads of the setup from 0 kg to 50 kg. At each added load the friction coefficient has been measured. Results show that both the natural frequency and friction coefficient decrease with the increase of added loads. It has been also observed that the coefficient of friction increases with the increase of natural frequency of the experimental setup. The experimental results are also compared with those available in literature and simple physical explanations are provided.

Keywords: *Friction coefficient; Natural frequency; Dead load; Stainless steel-304.*

1. INTRODUCTION

The coefficient of friction of a material is dependent upon the interface or mating material, surface preparation and operating conditions [1-7]. It is also known that vibration and friction are dependent on each other. Friction generates vibration in various forms, while vibration affects friction in turns [8-18]. The vibration and the natural frequency of a machine have severe effect on its life time and performance. These two factors cause noise in industries. To control noise it is important to determine the natural frequency of the machine. The dynamic body, which is subjected to friction force inherently, changes its static friction property. The general conclusion drawn by W. W. Tworzydlo and E. Becker, 1990 [19] is that the ratio of reduction of friction depends very strongly on the characteristics of each experimental set-up, so that these experimental observations could not yield any general law which would estimate the reduction of friction under a wide variety of conditions. Asad and Helali [20-22] observed the reduction of the friction coefficient as a function of different amplitude and frequency of vibration. However, the frictional behavior of stainless steel-304 (SS-304) and their dimensional analysis under natural frequency of the experimental set-up is yet to be investigated. Considering the lack of correlation among friction coefficient, natural frequency of the experimental set-up and other operating parameters, the present research was started to find out suitable correlation and a way of reducing friction coefficient by applying known natural frequency of vibration at a particular direction. Therefore, in this study an attempt is made to investigate the frictional behavior of SS-304 under natural frequency of the experimental set-up. It is expected that the applications of these results will contribute to the improvement of different concerned mechanical systems.

In this study vibration is generated artificially in such a way that direction, amplitude and frequency of vibration can be controlled.

2. EXPERIMENTAL

Fig.1 shows a pin-on-disc machine which contains a pin that can slide on a rotating horizontal surface (disc).

A circular SS-304 test sample (disc) is to be fixed on a rotating plate (table) having a long vertical shaft welded from the bottom surface of the rotating plate.

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The shaft passes through three close-fit bush-bearings which are rigidly fixed with three-square plates such that the shaft can move only axially and any radial movement of the rotating shaft is restrained by the bush.

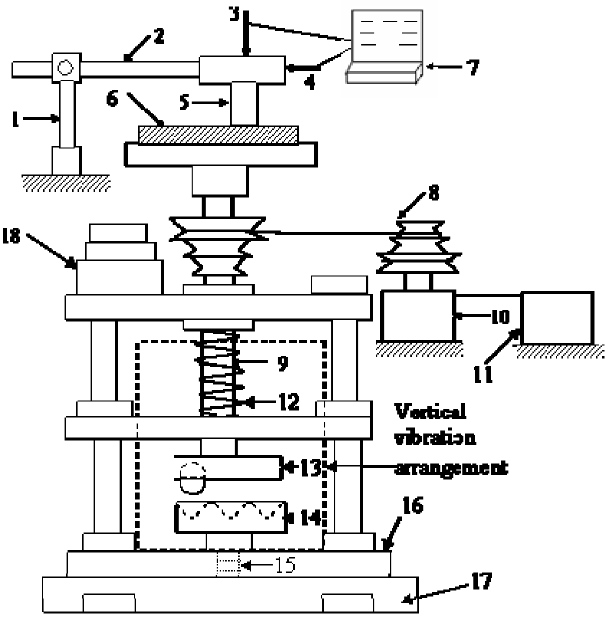

Figure 1. Schematic diagram of the experimental setup: 1. Load arm holder 2. Load arm 3. Normal load (dead weight) 4. Horizontal load 5. Pin sample 6. Test disc with rotating table 7. Computer 8. Belt and pulley 9. Main shaft 10. Motor 11. Speed control unit 12. Compression spring 13. Upper plate with ball 14. Lower plate with V-slots 15. Height adjustable screw 16. Base plate 17.Rubber block 18. Added dead load

To provide the rigidity to the main structure of this set-up all these three supporting square plates along with a base plate are rigidly fixed with four vertical square bars. The base plate was bolted with the foundation. A 50mm thick neoprene rubber bearing pad was used between the base plate and the foundation. Foundation bolts were passed through this bearing pad to fix base plate with this concrete foundation. Between the top two supporting square plates a compound V-pulley was fixed with the shaft to transmit rotation to the shaft from a motor. A compression spring between the second and the third supporting plates is fitted with the shaft such that any vertical movement of the shaft can be controlled. There are two circular plates near the bottom end of the shaft, one is fixed with the shaft end and another is fixed with the base plate through a height adjusting screw. The circular plate fitted with the shaft has a spherical ball on its lower surface. There are a number of slots on the top surface of the other circular plate. At the time of the rotation of the shaft, the ball of the upper circular plate will slide on the slotted surface of the lower circular plate and the shaft along with the rotating plate will vibrate due to the spring action. The direction of vibration is vertical i.e perpendicular to the sliding direction of the pin. Due to the cantilever action of the holding arm the pin with the holder has rocking mode of vibration. Considering the small area of contact of the pin and diameter of the rotating disc the sliding velocity can be taken as linear though the sliding surface. Frequency of vibration can be varied (i) by changing the shaft rotation and (ii) the number of slots of the lower circular vibration generating plate the. Sliding velocity can be varied by two ways (i) by changing the rotation of the shaft when frequency also changes and (ii) by changing the radius of the point of contact of the sliding pin when frequency does not change. But it can be noted that the change of curvature may affect resisting force measurement. The amplitude of the vibration can be varied by adjusting the height of the slotted plate.

A half-horsepower motor is mounted vertically to rotate the shaft with the table on a separate base having rubber damper. This separate base was used to reduce the effect of vibration of the motor, which may transmit to the main structure. The speed of the motor is varied as required by using an electronic speed control unit.

Contacting foot of a 6mm diameter cylindrical pin is flat made of SS-304, fitted on a holder is subsequently fitted with an arm. The arm is pivoted with a separate base in such a way that the arm with the pin holder can rotate vertically and horizontally about the pivot point with very low friction. Pin holder is designed including the facility of putting dead weight on it so that required normal force will act on the test sample through the pin. To avoid the loss of surface material of the pin the contacting surface will remain almost constant and for this the shapes of pin were maintained cylindrical. The natural frequencies of the setup are varied by adding the dead loads of 0 to 50 kg at the upper base plate of the setup. A load cell (TML, Tokyo Sokki Kenkyujo Co. Ltd, CLS-100NA, Serial no. MR2947) was used to measure the vertical force acting on the pin. A data acquisition system was used to measure the force continuously when the system is on and these data are sent directly to the computer. Vibration was measured by using a digital vibration meter (METRIX Instrument Co., Miniature Vibration Meter, Model no. 5500B.). The load cell along with its digital indicator (TML, Tokyo Sokki Kenkyujo Co. Ltd, Model no. TD-93A), calibrated against a standard proving ring was used for measuring loads. Losses of frictional forces at pivot points of the pin holder were determined and incorporated in the results. The total set-up was placed inside a chamber whose relative humidity can be adjusted by supplying requisite amount of moisture. A hygrometer (Wet and Dry Bulb Hygrometer, ZEAL, England) was used to measure the relative humidity of the chamber. A tachometer was used to measure the rpm of the rotating shaft. The surface roughnesses of the test sample were also measured by surface roughness tester (Taylor Hobson Precision Roughness Checker). The average roughnesses of the SS-304 before test were found to be 0.20 μm (RMS).All experiments were conducted at about 70% relative humidity. During tests each experiment was repeated several times with new sample of pin and disc

3. RESULTS AND DISCUSSIONS

3.1 Experimental Results

Fig. 2 and 3 show the variation of amplitude of vibration with the variation of frequency of vibration. From these figures, it is shown that the natural frequencies of the experimental setup are 60 and 32 Hz for 0 (without added load) and 50 kg dead loads respectively.

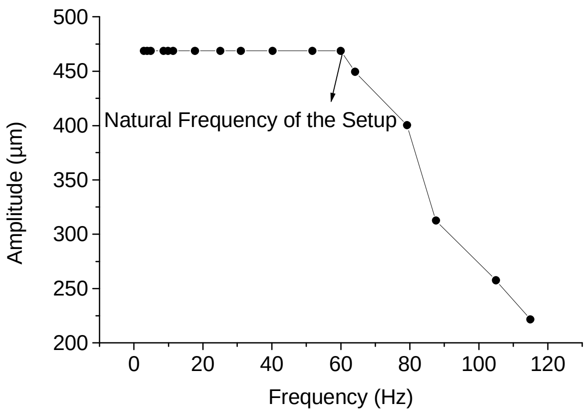

Figure 2. Amplitude vs. frequency curve of the test setup for without added dead load

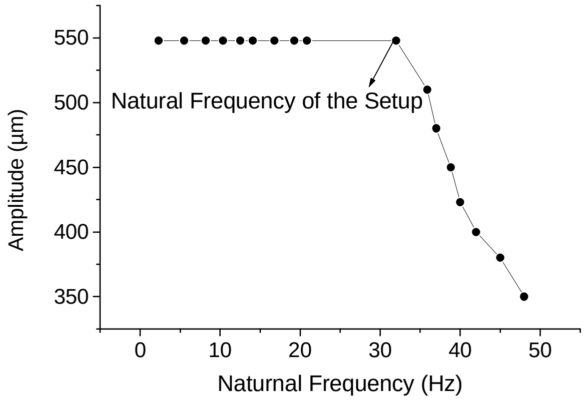

Figure 3. Amplitude vs. frequency curve of the setup for 50 kg added dead load

The method of measurement of the natural frequency of the setup is taken from the book of “Mechanical Measurements” [23]. Variation of natural frequencies with added dead loads of 0, 13.6, 20, 36.1 and 50 kg are presented in Fig. 4.

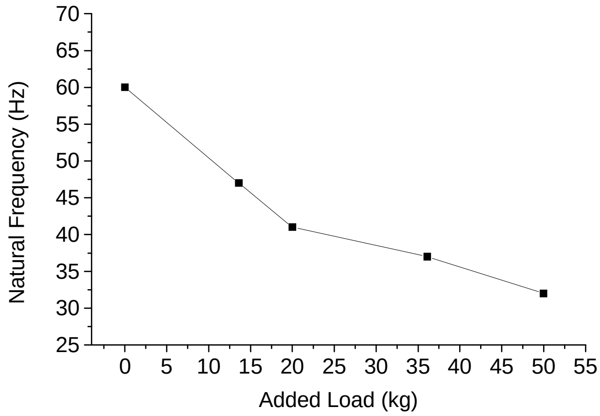

Figure 4. Variation of natural frequency with the variation of dead added loads

The curve shown in Fig.4 shows that the natural frequency of vibration decreases with the increase of added load. The decrease of natural frequency of vibration with the increase of added load can be explained by the equation $\omega^2 = \frac{k}{m}$ (where ω= natural frequency, k= stiffness of the system and m= mass of the system). From this equation, it is clear that the natural frequency only depends on the stiffness and the mass of the system. Therefore by keeping the stiffness constant if the mass of the system is increased the natural frequency of the system will be decreased.

Fig. 5 shows the variation of friction coefficient with the duration of rubbing at different natural frequency of vibration for copper.

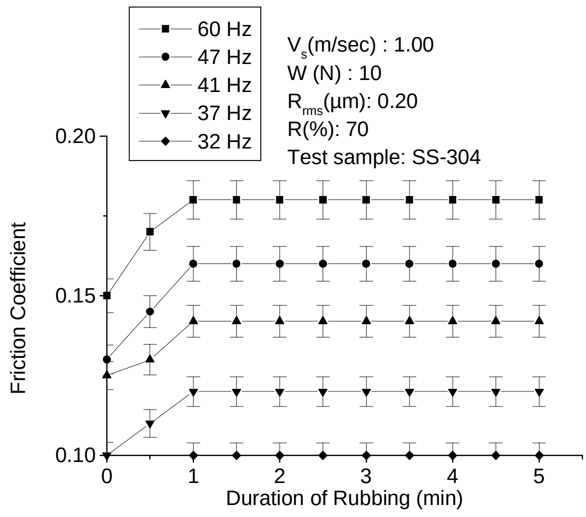

Figure 5. Variation of friction coefficient with the variation of natural frequency of the experimental setup

Curve 1 of Fig. 5 shows the variation of friction coefficient of SS-304 with the variation of rubbing duration at 60 Hz natural frequency of vibration. During the starting, value of friction coefficient is 0.15 which remains constant for few seconds then increases almost linearly up to 0.18 over a duration of 60 seconds of rubbing and after that it remains constant for the rest of the experimental time. Other curves of this figure show the values of friction coefficient at 47, 41, 37 and 32 Hz natural frequency of vibration. All these curves show similar trend as that of curve 1. Other parameters such as sliding velocity (1 m/sec), normal load (10 N), surface roughness (0.20 μm) and relative humidity (70%) are identical for these five curves. These findings are in agreement with the findings of Chowdhury for different amplitude of vibration [22]. The friction at the time of starting is low and remains at its initial value for some time and the factors responsible for this low friction are due to the presence of a layer of foreign material. This surface in general comprises of (i) moisture, (ii) oxide of metals, (iii) deposited lubricating material, etc.. SS-304 readily oxidizes in air, so that, at initial duration of rubbing, the oxide film easily separates the two material surfaces and there is little or no true metallic contact and also the oxide film has a low shear strength. During initial rubbing, the film (deposited layer) breaks up and clean surfaces come in contact which increase the bonding force between the contacting surfaces. At the same time due to the inclusion of trapped wear particles and roughening the substrate, the friction force increases due to the increase of ploughing effect. Increase of surface temperature, viscous damping of the friction surface, increased adhesion due to microwelding or deformation or hardening of the material might have some role on this increment of friction coefficient as well. After a certain duration of rubbing, the increase of roughness and other parameters may reach to a certain steady state value and hence the values of friction co-efficient remain constant for the rest of the time.

In the curves of Fig. 5, it is also seen that the values of friction co-efficient increases with the increase of natural frequency of vibration. These results are presented in Fig. 6. If a body (either static or dynamic) is in contact with another moving (either rotation or translation) body, where the second body is vibrating, the contact of those two bodies takes place at some particular points of the second body instead of continuous contact. When the natural frequency of vibration of second body is more, for a constant length of contact, the contact points as well as the area of contact between two bodies will be more (Fig. 7(b)) compared to the situation when the natural frequency of second body is less (Fig. 7(a)). As the area of contact or the points of contact between two bodies are more, they experience more frictional resistance for a constant length of contact. Hence, the friction factor between the two bodies will increase with increased natural frequency.

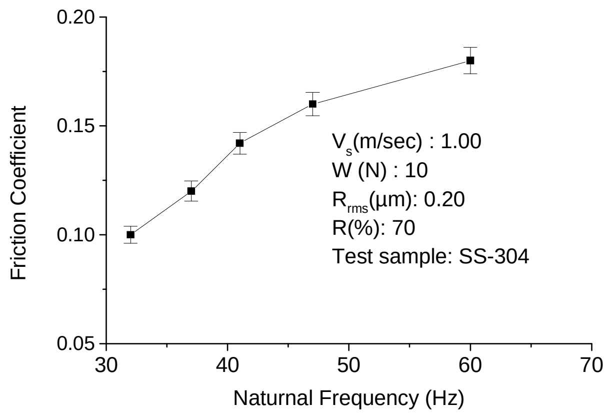

Figure 6. Variation of friction coefficient with the variation of natural frequency of the experimental setup for SS-304

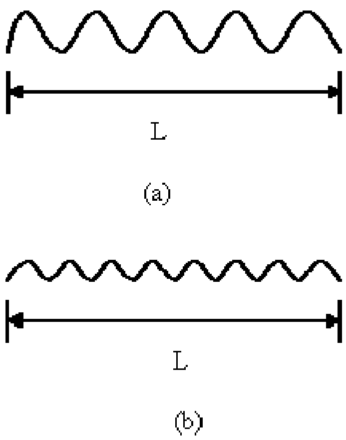

Figure 7. (a) The points of contact of a body with low natural frequency, (b) The points of contact of a body with high natural frequency for constant length of contact (L)

3.2 Dimensional Analysis

Let,

$$F = f(f_n, V, N, R) \tag{1}$$

Where,

F = Frictional force = MLt $^{-2}$

f_n = Natural frequency = t $^{-}$

V = Sliding velocity = Lt $^{-1}$

N = Normal load = MLt $^{-2}$

R= Root mean square roughness of the tested surface = L

Let "k" be a dimensionless constant, then (1) can be written as,

$$F = k\,[f_n^{a}.\ V^{b}.\ N^{c}.\ R^{d}] \quad (2)$$

Substituting the dimensions of each physical quantity, (2) reduces to

$$[MLt^{-2}] = k\,[(t^{-1})^{a}.\ (Lt^{-1})^{b}.\ (MLt^{-2})^{c}.\ (L)^{d}]$$

or, $[MLt^{-2}] = k\,[L^{b+c+d}.\ t^{-a-b-2c}.\ M^{c}]$ (3)

Since (3) must be dimensionally homogeneous, equating the powers of M, L and t and obtain,

$$c = 1 \quad (4)$$

$$b + c + d = 1 \quad (5)$$

$$-a-b-2c = -2 \quad (6)$$

From (4) and (5),

$$d = -b \quad (9)$$

From (4) and (6),

$$a = -b \quad (10)$$

Therefore, $F = k\,[f_n^{-b}.\ V^{b}.\ N.\ R^{-b}]$

or, $$F = kN\left[\frac{V}{f_n R}\right]^{b} \quad (11)$$

or, $$\frac{F}{N} = k\left[\frac{V}{f_n R}\right]^{b} \quad (12)$$

or, $$\mu = k\left[\frac{V}{f_n R}\right]^{b} \quad (13)$$

Where, 'b' and 'k' are arbitrary constants

The dimensional wear parameter $\frac{V}{f_n R}$ is hereby called 'Chowdhury Number' and abbreviated as Cd No.

Fig. 8 shows the plot of Friction coefficient (μ) versus Cd No.

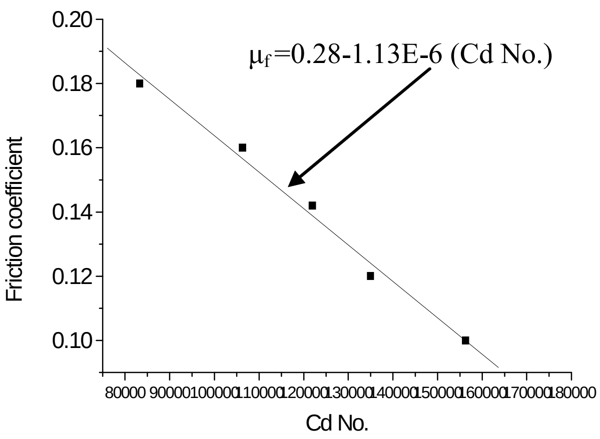

Figure 8. Friction coefficient as a function of chowdhury no.

The curve shows that μ decreases linearly with the increase of Cd No. for SS-304 and is represented by the equation:

$$\mu = 0.28 + (-1.13E-6)\left(\frac{V}{f_n R}\right)$$

In Fig. 8 square scatter points shows the experimental results of friction coefficient with Cd No. Using these experimental values, linear regression and correlation is done. Continuous line shown in this figure is the regression line and corresponding equation of this line is $\mu = 0.28 + (-1.13E-6)\left(\frac{V}{f_n R}\right)$.

This figure indicates strong negative relationship (coefficient of correlation=-0.99) between wear rate and Cd. No.

The coefficient of determination for the relationship between wear rate and Cd No. drawn in Fig. 8 is almost 99%. That is, Trend line or Cd. No. can explain 99% of the variation in friction coefficient. Therefore it may be concluded that experimental results are in good agreement with the theoretical calculations.

3. CONCLUSIONS

The presence of natural frequency of vibration indeed affects the friction force considerably. The natural frequency of vibration decreases with the increase of added dead loads to the experimental setup. The values of friction coefficient increase with the increase of natural frequency of vibration of the experimental setup. As the friction coefficient increases with increasing natural frequency of vibration, therefore maintaining appropriate level of natural frequency vibration friction may be kept to some lower value to improve mechanical processes.

The empirical formula of wear rate is derived from the dimensionless analysis. The friction coefficient obtained from the correlation shows better relationship with experimental results.

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