An experimental study of the correlation between wear and entropy flow in machinery components

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Based on first principles, a hypothesis was made on the potential correlation between entropy and degradation of machinery components in an earlier investigation of stochastic characterization of degradation dynamics. This paper reports on an experimental study in which degradation in the form of wear of model machinery component pairs was made on an accelerated testing basis. Concomitant measurement of entropy flow was made by means of a simple calorimeter. Results show a strong correlation between the referenced wear and the production of entropy flow. © 2000 American Institute of Physics. [S0021-8979(00)09417-2]

I. INTRODUCTION

The booklet Competitive Edge drew a great deal of attention among researchers as well as sponsors who were interested in the relevance of research topics in manufacturing. One such topic can be deduced from Fig. 1, attributed to Intel, a which shows that increased investment at the concept and design stages can reduce manufacturing lifecycle cost; the reduction of a significant portion of operation and support cost can be realized through savings in maintenance. A subsequent National Science Foundation sponsored workshop, in which half of the participants were from industry, concluded that the most pressing need among the 19 topics was the development of better models of machinery component degradation. It was the science base of this topic that was of interest.

Based on first principles, it was hypothesized that the degradation of machinery components is a consequence of irreversible thermodynamic processes that “disorder” a component, and that degradation is the time-dependent phenomenon of increasing disorder. a This suggests that entropy, a fundamental thermodynamic characterization of disorder, offers a natural measure of component degradation, which should be a direct consequence of the second law of thermodynamics.

Thermodynamic entropy has been classically employed to measure the quality of a system. Entropy can predict the direction and reaction rate of chemical processes, heat and energy transfer, and the efficiency of engines, among other phenomena. Machine component degradation involves physical effects similar to the above-mentioned processes. That is, degradation is also a thermodynamically driven process. A natural question is—can entropy predict the direction and rate of degradation processes in machinery components?

This paper reports the findings of an experimental study to test the above-mentioned hypothesis. Section II describes the experiment, the model component pair, rationale for the design of the accelerated testing, and other factors. Section III describes the procedures and the results of the experiments. Section IV is a discussion of the results, and Sec. V provides the conclusions drawn.

II. THE EXPERIMENT

A. Apparatus

Figure 2 is a schematic diagram of the central portion of the apparatus. A photograph of the central portion of the apparatus is shown in Fig. 3. The apparatus contains a precision, high-speed rotating shaft mounted on ball bearings. The shaft is driven by a dc motor and is partially immersed in a fluid lubricant. The lubricant is drawn in between the slider and the rider component pair and is circulated by means of a peristaltic pump. In the diagram, (A) is a weight which provides a compressive load between the component pair; (B) is an arm, free to rotate in the horizontal plane, which allows friction force to be measured; (C) is a magnetic tachometer; (D) is the rider partner of the component pair that is mounted on the rotating shaft; (E) is the rider partner of the component pair; (F) is a simple calorimeter with the rider component as an integral part; (G) is a pipe fitting where lubricant, which wets the interface of the component pair, can be drawn for ferrographic analysis; (H) is the probe of a photonic sensor which provides an optical means of measuring the physical degradation of the rider partner of the component pair. Not shown is a built-in transducer, which restrains the arm (B) from moving and measures the friction force.

The rider component was a cylindrical rod, 0.3 cm in diameter and 2.5 cm in length, which widens into a rectangular plate of larger dimensions than the cylindrical diameter. The rectangular plate allows the component to be anchored to a plunger, which holds the component and allows it to move freely in the vertical plane. This vertical movement was measured by the photonic sensor (H). The cylindrical end of the rider component was encased in a cylindrical insulator. The insulating casing stops short of the contact end of the rider component to allow wear to occur without the interference of the insulating material. The exposed part of the component at the contacting portion was painted for insulation purposes.

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The cylindrical end of the rider and the insulator form the basic part of the simple calorimeter. To complete the simple calorimeter function, very fine iron-constantine thermocouples were inserted into holes drilled to the center of the components at 0.16, 1.27, and 2.38 cm from the beginning of the insulating casing nearest to the rider–slider interface. The thermocouples were anchored in place by epoxy.

B. Model component pair

Referring to the discussion in Sec. II A, rider (E) and slider (D) form the core of the model machinery component pair. The components can assume many shapes and functions. Our model configuration was chosen for the present study for the following reasons: (a) the component pair allows relative motion between the components, and is not a single component where motion may only be occasional; (b) the component pair operates in boundary lubrication regime, a state of lubrication in machinery most prone to wear, wherein intermittent contact between sliding pairs occurs; (c) while still simple for laboratory study, the configuration is sufficiently realistic to represent a real-life component pair; (d) the design is amenable to accelerated testing.

C. Materials and accelerated testing

For the component-pair configuration chosen, the flat surface of the rider was brought into contact with the slider’s curved surface, see Sec. II A. Under ideal conditions, when the rider touches the slider under load, the contact is a line contact. With sufficient load and wearing, in time, the contact surface of the rider becomes concave and conforms to the slider’s curved surface.

To satisfy an important condition of machine component design, the load is smaller than the product of the geometric area of contact and the hardness pressure or yield strength of the metal. The geometric area of the rider in contact is called the apparent area of contact to differentiate it from the real area of contact. The real area of contact is smaller than the apparent area of contact and is given by the ratio of the normal load and the hardness pressure of the metal.

The real area of contact is composed of a number of constantly shifting contact spots. The manner of constant shifting of these contact spots where “flash temperatures” occur has been studied. It should be noted that, henceforth in this paper, temperature on the surface will always mean macroscopic temperature, i.e., the large number of transient temperature spikes are ignored.

In view of the above discussion and the discussion in Sec. II B, AISI 1020 steel and ASTM B187 certified C110 copper were chosen for the slider and rider material, respectively. Copper is softer and wears faster than steel and thus satisfies one of the demands for accelerated testing. Copper is also very conductive. Inasmuch as it is the temperature gradient in a direction normal to the sliding surface that is
important in the calculation of thermodynamic entropy, the uniformity of temperature over the cross section of the copper rider component made it easier to measure temperatures along the insulated portion of the component.

As part of the plan to accelerate testing, a 9.1 kg dead weight, which provided the compressive load between the rider–slider interface, was selected along with the slider component’s rotational speed of 1000 rpm, which corresponds to a tangential surface speed of \(3.3 \text{ ms}^{-1}\). It was shown that for the above-described set of conditions, the macroscopic surface temperature, calculated from measurements, had reached steady state long before 1 h elapsed, see Fig. 4. As such, a standard testing period of 1 h was adopted for this study.

III. PROCEDURES AND TEST RESULTS

A. Procedures

The slider component ring was mounted on the shaft. The pool of lubricant from a previous test was drained, and while the peristaltic pump was running, a measured pool of fresh lubricant was used to flush the system. The lubricant level was set to provide boundary lubrication. The rider component was then installed in the plunger. The rider component and insulating casing were prepared the day before a given test; the thermocouples were mounted according to the description in Sec. II A, and the paint insulation around the tip of the component was allowed to dry. Altogether, there were five temperature probes—three for the simple calorimeter and one each for the room and oil temperatures. Thermocouple wires were connected to a set of recorders, as was the output of the linear transducer that was used to measure the frictional resistance.

The 9.1 kg deadweight, described earlier, was put in place after the rider component was installed in the plunger. The photonic sensor was then calibrated. At this point, a preliminary test was run at a shaft speed of 1000 rpm and a normal load of 9.1 kg. The purpose of this preliminary test was to wear in the tip of the rider component so it became curved and conformed to the curvature of the slider ring. This took about 5 min. Following this test, the motor was shut down and the system was allowed to cool to the temperature of the circulating oil.

Data collection consisted of 13 readings at 5 min intervals. The initial 5 min run smoothed out the sliding and allowed the specimen to heat up. More important, this process gave a baseline datum for the photonic sensor reading from which the remaining 12 readings were made. The rider’s macroscopic surface temperature reached steady state within the first 15 min. This was a requirement for the planned accelerated testing. At the end of the 1 h duration, an oil sample was taken for ferrographic analysis.

B. Test results

The surface temperature of the rider component as a function of time for a series of six tests is shown in Fig. 4. Degradation by wear of the rider component as a function of time for the series is shown in Fig. 5. Entropy flow as a function of time for the series is shown in Fig. 6. We use the term entropy flow to emphasize the fact that the entropy measured is that of a very thin layer of material at the rider’s surface, which is constantly wearing away or changing. The

FIG. 4. Surface temperature of the rider component as a function of time. Data represent a series of six tests. Rider component is ASTM B187 certified C110 copper and slider component is AISI 1020 steel. Normal load is 9.7 kg, rotational speed of the slider component is 1000 rpm. Tests were conducted under boundary lubrication regime.

FIG. 5. Wear of the rider component as a function of time. Data represent the same series of six tests.

FIG. 6. Entropy flow as a function of time. Data represent the same series of six tests.
ordinate of Fig. 7 shows the wear for each test, normalized by the maximum value of the set in each test, and the abscissa shows the entropy flow for each test, normalized by the maximum value of the set in each test. Entropy flow was calculated by the discretized formula,

\[ S_n = \sum \frac{\Delta Q^{(n)}}{T^{(n)}} \]

where \( \Delta Q^{(n)} \) is the increment of heat input to the rider during the \( n \)th time interval, and \( T^{(n)} \) is the average absolute surface temperature of the rider during the \( n \)th time interval. Figure 8 is a photograph of a typical ferrogram showing the magnetically precipitated copper wear debris at the end of the test. The overall margin of error of measurements and method of calculations leading to Fig. 7 is 2.5%.

**IV. DISCUSSION**

The prior sections described the measurement of rider wear, and the temperatures that enabled the calculation of entropy flow. Since the heat associated with this entropy was generated by friction, this entropy is irreversible. The correlation found between wear and entropy flow links the irreversible degradation by wear to the production of irreversible entropy flow. We believe this to be a general principle of machinery component wear: irreversible degradation is always accompanied by concomitant production of irreversible entropy. By studying this entropy and its dependence on the energetics of the irreversible process that created it, the direction and rate of machinery component degradation processes can be determined. Thus, physical mechanisms of degradation can be compactly described by equations such as Eq. (1).

Even though the normal load and the rotational speed of the slider component are the same for the series of six tests, Fig. 4 shows a spread of surface temperatures of approximately 15 °C. There are many possible reasons for the temperature variations. Two factors are predominant; they are the variations in the hardness of the machined rider, and the partition of frictional heat between the rider component and the moving slider component, see Ref. 3. It should be noted that each test employed a freshly machined rider and an unused portion of the slider ring.

Figure 5 shows a wide range of wear. Here hardness plays an important role. For example, two sets of data shown as inverted triangles and open squares, have the least hardness, i.e., Rockwell B of approximately 50.

Figure 6 shows a wide spread of entropy flow. We note here only one observation. A set of four data sets, represented by open circles, open diamonds, open squares, and closed circles, are growing with time in a fairly linear fashion; the other two, represented by open triangles and inverted triangles are growing as the previous four, but their rise is tapering off with time.

Inasmuch as the spreads described in Figs. 4–6 represent the complex nature of tribology, the science of surfaces under normal load and relative motion, we believe the data set carries very important information relative to the primary goal of this study. When normalized as described under Sec. III B, Fig. 7 shows two salient features. (1) All data in the sets represented by open circles, inverted triangles, closed circles, and open squares show a nearly direct correlation of normalized wear and normalized entropy flow. (2) The data sets represented by open triangles and open diamonds, while also showing a strong correlation between the two normalized variables, show collectively the effect of hardness. As the relative difference of hardness between the copper rider and the steel slider is small, the abrasive mechanism for degradation commands not only smaller degradation, but also a decreasing rate of degradation with time. Thus the relative higher hardness of RB 60 causes the rate of degradation of the rider component to taper off with time.

Returning to the main feature of Fig. 7, let

\[ w = F_w(S), \]

where \( w \) denotes wear, and \( F_w \) denotes a function of entropy \( S \). Differentiating,

\[ \frac{dw}{dt} = \frac{dF_w}{dS} \frac{dS}{dt}. \]
Equation (2) depends on the change of wear with entropy $dF_w/ds$, and the rate of entropy production $dS/dt$. We will show later how $dF_w/ds$ can be extracted from Fig. 7.

Note that we use wear as a general term and it is measured by the volume of material removed. In the experiment, we use the term degradation, $d$, as a one-dimensional measure of wear; in this case $w$ is proportional to $d$ and the constant of proportionality is the cross-sectional area of the cylindrical rider.

We recall Archard’s wear law$^5$ or Holm’s wear law$^6$, which states

$$w = \frac{kLx}{H}. \quad (3)$$

Here $k$ is a dimensionless wear coefficient, $H$ is the hardness pressure, $L$ is the normal load, and $x$ is the distance slid, e.g., between a rider on a slider. Upon differentiating $\rho$, and introducing the coefficient of friction $\mu = f/L$, where $f$ is the frictional resistance,

$$\frac{dw}{dt} = \frac{kP_\mu}{\mu H}, \quad (4)$$

and the power dissipated by friction force, $P_\mu = \mu L(dx/dt)$. Noting that $P_\mu/T = dS/dt$ is the rate of irreversible entropy produced by friction we have

$$\frac{dw}{dt} = \frac{kT}{\mu H} dS, \quad (4a)$$

whence

$$w = \frac{kT}{\mu H} S. \quad (4b)$$

We note that Archard’s wear law or Holm’s wear law, Eq. (3), is subsumed in Eq. (1). That is to say, while it was stipulated that there is a constant wear coefficient, Eq. (1) allows, generally, the wear coefficient to be a function of entropy, among other factors such as the coefficient of friction.

Subject to the earlier discussion of the six sets of data, Fig. 7 shows that the group of four sets of data in the first approximation is represented by

normalized wear = normalized entropy flow, \quad (5)

where normalized wear $= (d_n - d_0)/(d_{12} - d_0)$ in which wear $d_0$ is the datum, and $d_n$ $(n = 1, ..., 12)$ is the wear at the end of the $n$th interval of data taking. Of course $(d_{12} - d_0)$ is the maximum wear in each set. Normalized entropy flow $= S_n/S_{12}$ where $S_n = \Sigma^n (\Delta Q^{(n)}/T^{(n)})$, $\Delta Q^{(n)}$ is the increment of heat input to the rider at the rider–slider interface during the $n$th time interval of data taking, and $T^{(n)}$ the average absolute temperature of the interface of the rider and slider in the same time interval.

As noted before, $w$ in Eq. (4b) can be easily expressed in terms of $(d_n - d_0)$. Applying Eq. (5) to Eq. (4) with suitable substitution,

$$k = \frac{dF_w}{S} \mu H T. \quad (6)$$

Using Eq. (6) with $dF_w/\rho$ from the slope of Fig. 7, one can calculate the value of $k$ for each set of data shown in Fig. 7: inverted triangles, open circles, closed circles, and open squares. We have $k = 1.2 \times 10^{-4}$, $k = 8.6 \times 10^5$, $k = 10^{-4}$, $k = 10^{-4}$, respectively. The average value of the four is $k = 1.01 \times 10^{-4}$. We note that this value coincides with that given by Rabinowicz$^7$ for compatible metals under poor lubrication, i.e., $10^{-4}$.

Finally, we point out that two different measuring schemes, measuring physical states with very different physical dimensions, produced very similar values of wear coefficient $k/H$. Rabinowicz’s value for $k/H$, based on Eq. (3), required measurements of normal force $L$, wear volume $w$, and distance slid $x$. Our value, based on Eqs. (4b) and (6), required measurements of wear volume $w$, coefficient of friction $\mu$, and temperatures for $T$ and $S$. This agreement between coefficients implies that the entropy/degradation by wear hypothesis is correct.

V. CONCLUSION

This study started with an experimental design which satisfied the following requirements: (1) a model component pair which allowed relative motion between the pair; (2) the component pair operated in boundary lubrication regime, a state of lubrication most prone to wear; (3) though simple, the configuration was sufficiently realistic to represent a real-life component pair; (4) the design allowed entropy flow to be measured, though of necessity indirectly; (5) the design was amenable to accelerated testing. The primary result was a strong correlation between component wear and entropy flow. It was shown that normalized wear is a function of normalized entropy flow. In particular, within first approximations, normalized degradation is equal to normalized entropy flow. Moreover, it was shown that the classical Archard’s wear law or Holm’s wear law, which allows only a constant of proportionality between wear and quantity of normalized load $L$, the distance slid $x$, and divided by the hardness pressure, is a thermodynamic consequence and thus is subsumed in our generalized functional relationship between wear and entropy flow.

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$^6$R. Holm, Electric Contacts (Gerbers, Stockholm, 1946).