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SOME COMMENTS ON THE INTERPRETATION OF THE TRIBOLOGICAL WEAR COEFFICIENT

UWAGI O INTERPRETACJI WSPÓŁCZYNNIKA ZUŻYCIA TRIBOLOGICZNEGO

Key words: J.F. Archard equation, wear coefficient, real and nominal surfaces, energy dissipation zone, wear intensity, number of asperity contacts, unit pressures, alternative model.

Abstract: An original model of tribological wear is presented, an alternative to the commonly used J.F. Archard's model. The impossibility is established of a full conversion of mechanical work into the heat of dissipation and thereby of avoiding wear in the sliding friction of solids. The assumption is consequently questioned that only some contacts of surface asperities are subject to temporary wear. Material wastage is assumed to occur at each contact of asperities. The volume of worn material is dependent on the volumetric wear coefficient of the "energy dissipation zone" in friction. The dimensions of the zone are determined in both the elements in friction. Linear wear intensities and volumetric wear are described in analytical terms. The thermodynamic analysis of the tribological process indicates some limitations to these intensities. Energetic efficiencies of solid wear and heating as a result of friction are defined. Some new interpretations of the wear coefficient are proposed.

Słowa kluczowe: wzór J.F. Archarda, współczynnik zużycia, powierzchnie rzeczywista i nominalna, strefa dyssypacji energii, intensywność zużywania, liczba styków nierówności, naciski jednostkowe, model alternatywny.

Streszczenie: W pracy przedstawiono oryginalny model zużywania tribologicznego alternatywny do powszechnie stosowanego modelu J.F. Archarda. Stwierdzono niemożliwość zupełnej zamiany pracy mechanicznej na ciepło dyssypacji i tym samym uniknięcia zużycia przy tarciu ślizgowym ciał stałych. Na tej podstawie zakwestionowano słuszność założenia, że tylko część styków nierówności powierzchni podlega doraźnemu zużywaniu. Przyjęto, że ubytki materiału występują w każdym styku nierówności. Objętość startego materiału uzależniono od współczynnika zużycia i objętości tak zwanej strefy dyssypacji energii podczas tarcia. Ustalono wymiary tej strefy w obu trących się elementach. Opisano analitycznie liniowe intensywności zużywania i zużycie objętościowe. Przeprowadzona analiza termodynamiczna procesu tribologicznego wykazała istnienie ograniczeń tych intensywności. Określono sprawności energetyczne zużywania i nagrzewania ciał stałych wskutek tarcia. Przedstawiono nowe interpretacje współczynnika zużycia.

INTRODUCTION

In machine operation, friction is analysed at the macroscopic level of matter, that is, as a process among co-working machine parts. Their contact is limited to the geometrical (outline) or nominal surface. The design documentation of machinery shows unit pressures with reference to that surface. At the lower levels of the matter hierarchy, known as macroscopic layers and molecular aggregates **[L. 1]**, a contact of bodies in friction takes place on a very high number of small surfaces that carry the loading. Elementary contact surfaces are the simple connected areas of body contacts. The unit pressures in these areas are relatively large, comparable to the yield strength or material hardness. The real

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surface of body contact is defined as the sum total of elementary surfaces limited to the area of their nominal contact. Elementary friction and wear effects take place at contacts of solid surface asperities. They are manifested as the macroscopic processes of energy dissipation and wear of co-working solids.

In tribology, wear is a continuous process of destructive changes to an initial surface condition and to the mass, chemical composition, structure, and stresses of a body material top layer caused by co-working solids and the environment acting on its surface **[L. 2]**. These changes are reversible, disappearing on removal of the impacts that cause them, or permanent, remaining after these impacts are removed. In addition, temporary changes, occurring on a single contact of asperities, and accumulating changes that expand as identical external impacts recur, can be distinguished.

This paper addresses the interpretation of wear coefficient that conditions the generation of a wear particle in effect of accumulating and temporary changes. The major types of damage to a friction bond are discussed to begin with. Only some of them follow multiple loadings of an elementary contact of surface asperities.

J.F. Archard **[L. 3]** is greatly appreciated by specialist authors (more than 6650 citations according to Google Scholar). It presents an original way of defining volumetric wear based on a microscopic model of an elementary contact of solids. The addition of the volumes of small separated material fragments is the starting point for determining the wastage of a worn solid's volume. Wear coefficient, identified with the likelihood of detachment of a particle, plays an important role in the analytical description introduced in **[L. 3]**. J.F. Archard's model is described here, then an alternative is suggested where the coefficient is given a new interpretation. Relationships between volumetric wear and the area of energy dissipation in the particular friction couple elements are established. Some limitations to the transformation of mechanical energy into heat from the thermodynamic perspective are taken into account. The efficiency of solid material dispersion is described, and maximum and minimum values of its intensity are determined.

THE PULSE NATURE OF THE PROCESS OF TRIBOLOGICAL WEAR

Elementary, instantaneous contact forces on real solid contact surfaces manifest themselves on the nominal surface as phenomenologically averaged friction forces that execute some work along a certain path. In order to interpret the processes in the zone of energy dissipation, or the immediate environment of real solid contacts, their pulse nature must be taken into consideration. Material discontinuities in the contact area of solids in friction result in local wastage in the form of wear particles. This implies that energy dissipation and wear are of a pulse nature. Little material portions detached from the top layer cause a systematic change of the linear dimension or the volume (mass) of friction couple elements. Therefore, the interpretation of wear should also address the fact of material dispersion, conditioned by the generation of very small wear particles of characteristic dimensions to the order of 10^{-6} m.

Figure 1 illustrates the basic types of damage to a friction bond after I.V. Kragielski **[L. 4]**. Except

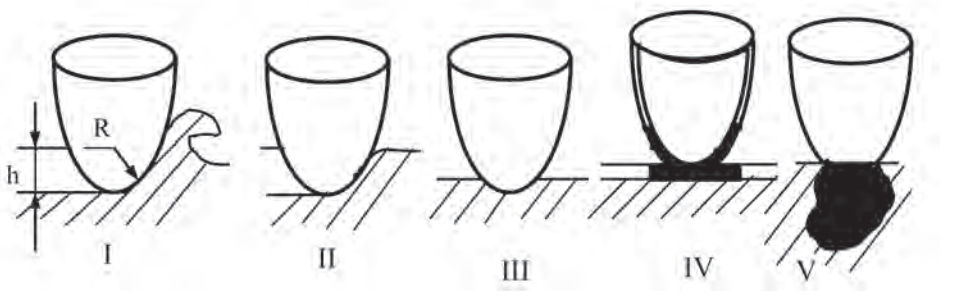

Fig. 1. A schematic representation of the basic types of damage to the friction bond: I – shear, II – plastic deformation, III – elastic deformation, IV – adhesive material bond and separation of the bonds, V – bonding of both the surfaces and material pull-out [L. 4]

Rys. 1. Schematyczne przedstawienie podstawowych rodzajów uszkodzenia więzi tarciowej: I – ścinanie, II – odkształcenie plastyczne, III – odkształcenie sprężyste, IV – sczepianie adhezyjne materiału i rozdzielanie powstałych sczepień, V – sczepianie obu powierzchni i wyrywanie materiału **[L. 4]**

III and IV, these are temporary destructive changes, whereas the plastic deformation (chasing) does not cause a wear particle to separate. Accumulating changes comprise elastic deformation and adhesive bond that result, after a sufficient number of repetitions, in the detachment of a material fragment I.V. Kragielski assigned to each a number of cycles n_k conditioning damage to a worn material. The number may be related to the following:

- The ratio of depth to which asperities penetrate into a worn material to the radius of its apex rounding R;
- The ratio of the material shear stress τ to its yield strength σ_T;
- The gradient of shear stresses on the surface of a worn material τ/dh.

The key characteristics of the friction bonds shown in **Fig. 1** are listed below **[L. 4]**.

Micromachining (shearing) I:

$$n_k \rightarrow 1,\ \frac{h}{R} \geq 0.5\left(1-\frac{2\tau}{\sigma_T}\right).$$

Plastic deformation II:

$$1 < n_k < \infty,\ \frac{h}{R} > 0.01 \text{ – for ferrous metals;}$$

$$\frac{h}{R} > 0.001 \text{ – for non-ferrous metals.}$$

Elastic deformation III:

$$n_k \rightarrow \infty,\ \frac{h}{R} < 0.01 \text{ – for ferrous metals;}$$

$$\frac{h}{R} < 0.001 \text{ – for non-ferrous metals.}$$

Adhesive bond IV:

$$n_k \rightarrow \infty,\ \frac{d\tau}{dh} > 0.$$

Material bonding and pull-out V:

$$n_k \rightarrow 1,\ \frac{d\tau}{dh} < 0.$$

The methods of breaking a friction bond as illustrated in **Fig. 1** may serve to construct some models of the wear process. In the case of accumulating destructive changes, the number of recurrences (energy pulses) n_k prior to the detachment of a wear particle is important. A first attempt at establishing a dependence between tribological wear and the likelihood of its occurrence was undertaken by R. Holm **[L. 5]** as early as 1946. He treated particular atoms as potential wear particles, assuming plastic contacts between bodies in friction. In fact, these are not individual atoms but their aggregates that are distributed between friction elements, as demonstrated by J.T. Burwell and C.D. Strang in **[L. 6]**. The likelihood of wear particles detaching from friction surfaces is defined assuming the particular real contact surfaces are constant in time and their number rises as the external loading increases. This means the dimensions of particles are independent from that loading. J.F. Archard **[L. 3]**, of fundamental significance to tribology even today, was published in 1953. The author assumed tribological wear is an effect of adhesion. The model of surface asperities contact is presented in **Fig. 2**.

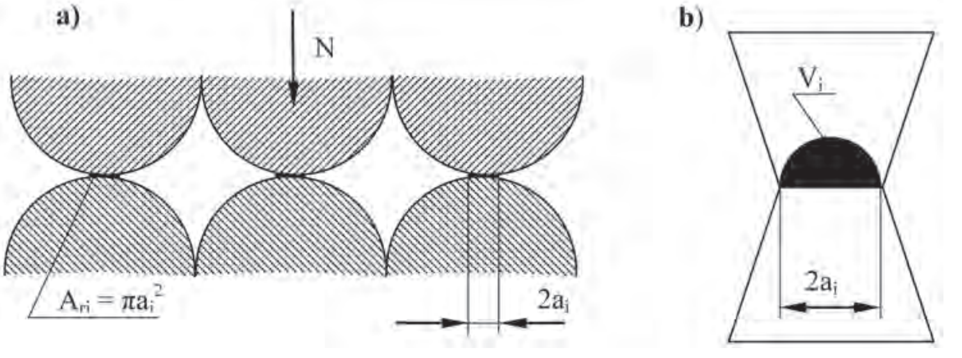

Fig. 2. The model of surface asperities contact after J.F. Archard: a) contacts between semi-spherical surface asperities, b) the generation of wear particles with a volume V_i [L. 3]

Rys. 2. Model styków nierówności powierzchni według J.F. Archarda: a) kontakty półkulistych nierówności powierzchni, b) powstawanie cząstki zużycia o objętości V_i **[L. 3]**

The sum total of the circular elementary contact surfaces $A_{ri} = \pi a_i^2$ forms a real contact surface A_r, where a_i – the radius of elementary circle. If elementary micro-contacts are of the same size and their number is n_o, the real contact surface is described with the following **[L. 3]**:

$$A_r = n_o \pi a_i^2 = \frac{N}{H}, \tag{1}$$

where: N – normal force, H – the hardness of the softer friction couple material.

The wear particles are additionally assumed to be semi-circular and have an elementary volume V_i:

$$V_i = \frac{2}{3} \pi a_i^3. \tag{2}$$

Assuming each particle arises on an elementary displacement $2a_i$ with a likelihood k', the volumetric wear of an entire friction couple is described as follows:

$$V = k' n_o V_i \frac{1}{2a_i}, \tag{3}$$

where: l – friction path and the ratio V_V/l, on considering (1) and (2), is formulated as **[L. 3]**:

$$\frac{V}{l} = k' \frac{N}{3H} \tag{4}$$

or the current formula

$$\frac{V}{l} = k \frac{N}{H}, \tag{5}$$

where: k = k'/3 – non-dimensional coefficient of wear.

Figure 2b shows a semi-spherical contact of surface asperities and the generation of wear particles in J.F. Archard's model. Volumetric wear is a function of only two variables – friction path and real friction surface. Two more quantities, the normal loading N and hardness H of the softer friction couple material, are required to describe the real friction surface. They are of huge practical significance. The coefficient of proportionality k, or wear coefficient, means the likelihood of generating wear particles, expressed as the reverse of the number of recurrences n_k of elementary asperities contacts necessary to produce a wear particle:

$$k = \frac{1}{n_k}. \tag{6}$$

J.F. Archard's model in **Fig. 2** illustrates the generation of a wear particle with a volume V_i out of the less hard material of the friction couple element. The wear of the other element is ignored. In fact, both the elements are worn; therefore, the model needs to be generalised. This paper will present a more general wear mechanism that addresses the material hardness of each element and the presence of very hard wear particles in the friction contact. Equation (5) describes wear as a superficial mass flow across elementary contact asperities. This is a narrow approach, as it ignores the process of energy dissipation in friction in a volume far larger than that of a worn material. The following section will characterise the geometry of the "friction zone," situated immediately under the nominal surface of bodies in friction. The energy dispersed therein produces thermal effects (heating and heat release to the environment) and material dispersion. In the model proposed then, the wear coefficient will be a function of the volume of worn material and the volume of the friction zone.

FRICTION AND WEAR IN THE AREA OF ENERGY DISSIPATION

The top layer plays a major part in tribological research. Its testing provides grounds for classifications of wear mechanisms. The particular mechanisms correspond to certain changes of material properties near a friction surface. The changes of material hardness and structure and of the depth at which wear products are detached can be observed most commonly. The information gained in this way helps to determine the approximate depth down to which external effects penetrate into the material of a body involved in the process of friction. The concept of friction volume, described as a space in which energy is dissipated, was only introduced by G. Fleischer **[L. 7]** as a quantity characterising friction in 1980. The depth of friction is a derivative notion, associated with the former. G. Fleischer related the volume of separated material and the depth of friction, the depth of wear (thickness of a removed layer) and the depth of friction by means of a non-dimensional wear number **ν**. In 2007, I proposed an

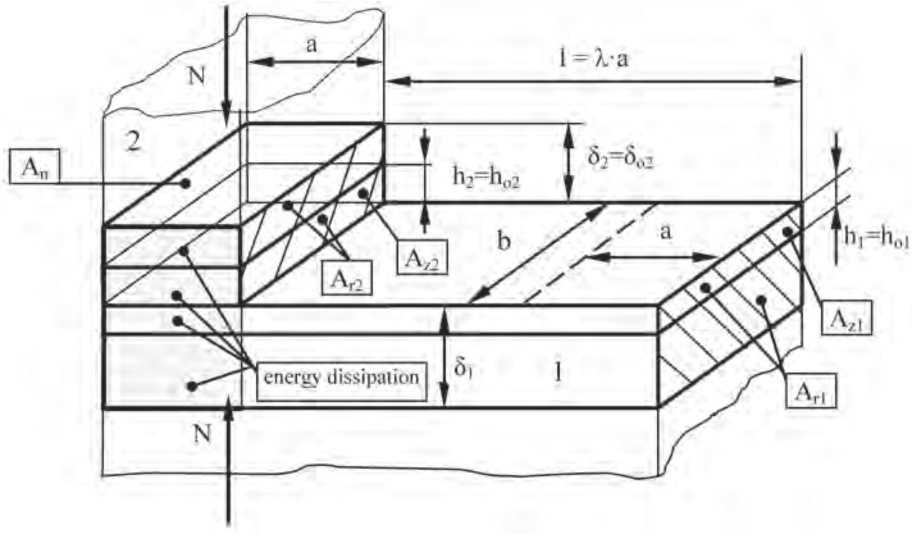

Fig. 3. A diagram of a system of solids 1 and 2 in friction, highlighting the area of energy dissipation [8]
Rys. 3. Schemat układu trących się ciał 1 i 2 z zaznaczonym obszarem dyssypacji energii **[L. 8]**

original method of determining the area of energy dissipation in both solids in friction **[L. 8]**. The argument intended to characterise the geometry of the energy dissipation by friction is summarised below.

In the general case, machine elements in friction differ in their dimensions. That is why **Fig. 3** shows a system of two solids – larger, 1, and smaller, 2. The dimensions of their contact, or nominal surface A_n, restrict the dimensions of the smaller element, a and b. Element b is the width of the friction path as well. The friction path l is a multiple of the length a, i.e., $l = \lambda \cdot a$, where λ – a coefficient whose reverse is referred to as the coefficient of mutual surface cover. A pressure force N acts on the nominal surface A_n, causing a unit pressure p. The physical properties of these solids are characterised by hardness and density, H_1, ρ_1 and H_2, ρ_2, respectively. Both the elements are worn by friction. The linear wear is expressed as h_1 and h_2, respectively, where $h_2 = \lambda \cdot h_{o2}$, where h_{o2} – linear wastage along the friction path. Meanwhile, $h_1 = h_{o1}$ after a single displacement of 2 in relation to 1. During friction, mechanical energy is dissipated in the spatial areas V_1^x and V_2^x, delimited with the dimensions δ_1 and δ_2 in the particular elements: $V_1^x = \delta_1 \cdot b \cdot l$; $V_2^x = \delta_2 \cdot A_n$. The real contact surfaces of 1 and 2, A_{r1} and A_{r2}, respectively, and the surfaces of worn layer cross-sections, A_{z1} and A_{z2}, respectively, are marked on the diagram based on **Fig. 3**. Friction force T acts on the contact surface A_n and performs work A_t along the displacement 1. It results in a volumetric wastage of V, the total volume wastage of both the solids, $V_1 = l \cdot h_1 \cdot b = k_1 \cdot A_{r1} \cdot l = k_1 \cdot V_1^x$ and $V_2 = h_2 \cdot A_n = \lambda \cdot h_{o2} \cdot A_n = k_2 \cdot A_{r2} \cdot l = k_2 \cdot A_{r2} \cdot \lambda \cdot a = k_2 \cdot \delta_2 \cdot A_n = k_2 \cdot V_2^x$. The mass wear of the particular elements is expressed as the products of material densities and the volumes of their wastage caused by friction: $m_1 = \rho_1 \cdot V_1$ and $m_2 = \rho_2 \cdot V_2$. The mass wear of the system totals $m = m_1 + m_2$.

The volumes V_1^x and V_2^x may be termed apparent volumes or geometric locations where real friction volumes $V_{a1}^x = (1/\lambda) \cdot V_1^x$ and $V_{a2}^x = (1/\lambda) \cdot V_2^x$, respectively, moved – **Fig. 3**. The linear dimension can be treated as a unit of friction path following the end of which another cycle of energy dissipation and wear recurs.

The wear number v refers to the particular solids. In the case of the larger element, it is defined with the following relations:

$$\mathbf{v}_1 = \frac{V_1}{V_1^x} = \frac{h_1}{\delta_1} = k_1, \tag{7}$$

while in the case of the larger element

$$\mathbf{v}_2 = \frac{V_2}{V_2^x} = \frac{h_2}{\delta_2} = \frac{h_{o2}}{\delta_{o2}} = k_2 . \tag{8}$$

The wear coefficients are equal to wear numbers. If a material wastage caused by friction is known, knowing **ν,** one can establish the friction volume. From the physical point of view, friction is not sufficiently explored if the location of the process is unknown.

This discussion allows for the determination of depths δ_1 and δ_2. On considering (7) and (8), the following dependences result:

$$\delta_1 = h_1 \frac{Nl}{H_1 V_1} = \frac{N}{bH_1} = \frac{ap}{H_1}, \quad (9)$$

$$\delta_2 = \delta_{o2} = h_{o2} \frac{Na}{H_2 V_{a2}} = \frac{N}{bH_2} = \frac{ap}{H_2}. \quad (10)$$

The volumes of real dissipation zones in the particular elements 1 and 2 result from (9) and (10) once the surface of nominal body contact A_n is introduced, since the load N and the friction force T act only on that surface:

$$V_{a1}^{x} = \frac{aN}{H_1}, \quad (11)$$

$$V_{a2}^{x} = \frac{aN}{H_2}. \quad (12)$$

The dimensions of the wear particles will be expressed as functions of the dimensions of energy dissipation zones in the particular solids in friction and of wear coefficient.

By introducing the linear intensity of wear I_{h1}, a major characteristic of the wear process, another interpretation of wear coefficient k_1 can be developed. With regard to the larger element, it is the relation of the linear wastage h_{o1} caused by friction to the friction path while element 2 is displaced relative to element 1. Therefore,

$$I_{h1} = \frac{h_1}{a} = \frac{h_{o1}}{a}. \quad (13)$$

The above quantity characterises the macroscopic body contact and the nominal contact surface A_n. The literature also posits the linear intensity of wear i_{h1} related to an elementary real contact surface A_{ri}, referred to as the specific linear intensity of wear. A known relation occurs between them, namely:

$$i_{h1} = I_{h1} \frac{H_1}{p}. \quad (14)$$

The dependence (14) between the wear intensities and i_{h1}, I_{h1} and the material hardness H divided by the nominal unit pressure p can be shown to result directly from (5). When the volume of worn material as λ of the nominal contact surface A_n along Section l of the friction path divided by the linear wastage h_1 is expressed with the dependences: $V_1 = \lambda \cdot A_n \cdot h_1 = k_1 \cdot l \cdot N/H_1$, k_1 can be defined as follows:

$$k_1 = \frac{A_n \lambda h_1 H_1}{lN} = \frac{\lambda h_1 H_1}{\lambda ap} = I_{h1} \frac{H_1}{p} = i_{h1}. \quad (15)$$

The tribological wear of the second element in **Fig. 3** is defined as follows: $V_2 = k_2 \cdot A_{r2} \cdot l = h_2 \cdot A_n = \lambda \cdot h_{o2} \cdot A_n = k_2 \cdot A_{r2} \cdot \lambda \cdot a$; hence the wear coefficient k_2 is expressed as follows:

$$k_2 = \frac{A_n h_{o2}}{A_{r2} a} = I_{h2} \frac{H_2}{p} = i_{h2}, \quad (16)$$

where the linear intensity of wear $I_{h2} = h_2/l = h_{o2}/a$, i_{h2} – specific linear intensity of wear, and h_{o2} – linear wear of the solid 2 along the friction path. A very important result is arrived at, namely, that the likelihood of generating a tribological wear particle k_1 out of an element of a greater friction surface is the specific linear intensity of wear i_{h1}. In parallel, the likelihood of generating a tribological wear particle k_2 out of an element of a smaller friction surface is the specific linear intensity of wear i_{h2}.

The geometrical quantities in **Fig. 3** and their analytical description, the equations (7 – 16), make up the macroscopic description of a friction couple and the tribological process. This discussion will continue to present a microscopic interpretation of objects and phenomena to supplement the macroscopic interpretation.

Figure 4 depicts the nominal surface of a friction contact A_n, a rectangle with the length a and width b. The direction of a is assumed to coincide with the vector of friction v and friction path l. The nominal surface is a geometric location of all elementary real contact surfaces. In turn, the real surface A_r of the contact of 1 and 2 is equal to the sum total of elementary real surfaces A_{ri} and constitutes a 10^{-5}–10^{-2} part of the area A_n. This can be schematically represented as a rectangle with the sides a_o and b_o. Tribological literature most commonly assumes the nominal surface divided by real surface n_o is equal to the relation of material hardness H and nominal unit pressure p. This condition obtains if plastic material deformation takes place, which it, in fact, does in the process of metal and plastic friction. Where elementary real surfaces A_{ri} are evenly distributed across the surface A_n, n_o corresponds to the number of these elementary surfaces. For the figure (rectangle) describing the real contact surface $A_r = a_o \cdot b_o$ to be similar to the figure expressing the nominal surface $A_n = a \cdot b$, two conditions must be fulfilled: $a_o = a/\sqrt{n_o}$ and $b_o = b/\sqrt{n_o}$. A_r can be therefore represented as a sum total n_o of identical elements A_{ri} evenly distributed across A_n. In the event, the sides of elementary fields are described as follows: $a_x = a_o/\sqrt{n_o} = a/n_o$; $b_x = b_o/\sqrt{n_o} = b/n_o$. A shear stress $\mu H = \mu \cdot n_o \cdot p$ is present in each surface element $A_{ri} = a_x \cdot b_x = A_n/n_o^2$.

The nominal surface in **Fig. 4** with some elementary contact surfaces of asperities distributed across it can refer to either friction couple elements made of a material with a specific hardness. Materials in friction are normally of different hardnesses; moreover, a quantity of wear products 3 is present in a friction contact as schematically shown in **Fig. 5**. The hardness of wear products material is assumed to be greater than that of the particular elements' material. In the case of metal friction, hard oxides are the particle 3, which substantiates the assumption. The solids 1 and 2 are pressed together with a normal force N, which causes a nominal unit pressure p against the nominal surface, designated with a horizontal broken line. The solids' material is physically characterised by density and hardness, respectively, ρ_1, ρ_2 and H_1, H_2. Wear particles between the friction surfaces form elementary real contacts with the particular elements of surfaces A_{ri1} and A_{ri2}, respectively; they make up the real contact surfaces A_{r1} and A_{r2}, respectively. The discussion below will employ the general symbols without indexing a friction couple element.

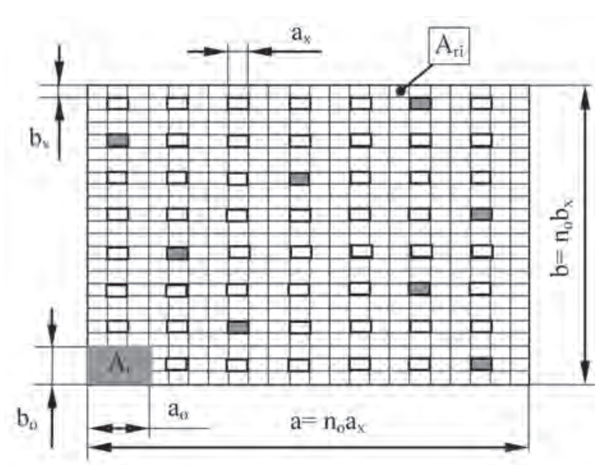

Fig. 4. Nominal surface A_n divided into n_o^2 identical parts A_{ri}; there are n_o parts marked with bold lines, $n_v = k \cdot n_o$ parts marked with bold lines and shaded; the real surface A_r is represented with a rectangle of the sides a_o and b_o [9]

Rys. 4. Powierzchnia nominalna A_n podzielona na n_o^2 jednakowych części A_{ri}; liczba części oznaczona grubymi liniami wynosi n_o; liczba elementów oznaczonych grubymi liniami i zaciemnionych wynosi $n_v = k \cdot n_o$; powierzchnię rzeczywistą A_r przedstawia prostokąt o bokach a_o i b_o **[L. 9]**

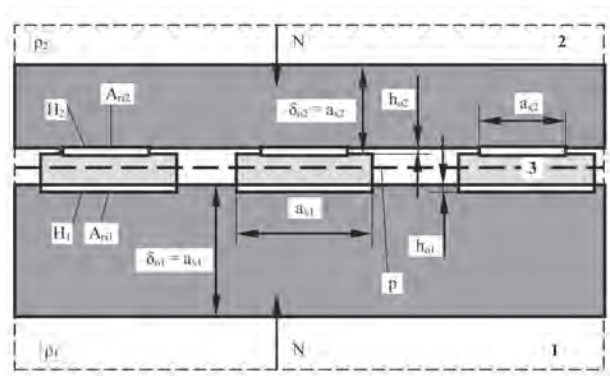

Fig. 5. A generalised diagram of the friction area [10]

Rys. 5. Uogólniony schemat obszaru tarcia **[L. 10]**

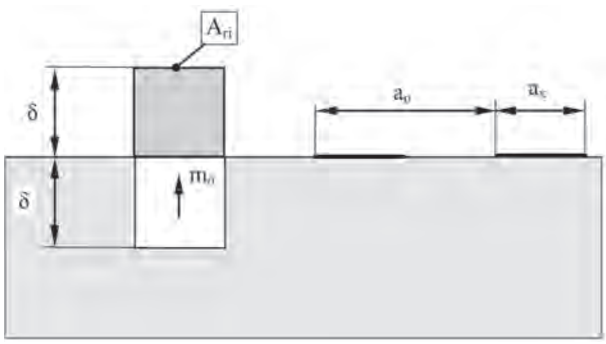

Fig. 6. A scheme of the detachment of a wear particle of the volume $A_{ri} \cdot \delta$ and mass m_o on the absorption of n_k energy pulses; the mutual situation of the contacts with sides a_x on the nominal surface is designated with bold lines

Rys. 6. Schemat oddzielania cząstki zużycia o objętości $A_{ri} \cdot \delta$ i masie m_o po wchłonięciu n_k impulsów energii; wzajemne usytuowanie styków o boku a_x na powierzchni nominalnej oznaczono grubymi liniami

A wear particle is generated in n_v shaded locations delineated with bold lines in **Fig. 4** following n_k energy pulses. Accepting J.F. Archard's postulate, the energy dissipation in the contacts marked only with bold lines does not result in temporary wear. This can be illustrated schematically in **Fig. 6**. A wear particle with a mass

m_o is a cuboid with a base $A_{ri} = a_x \cdot b_x$ and height δ, since an n_k – time energetic load applies to this material volume. After a friction path equal to the dimension a is covered, $n_v \cdot n_o$ of these particles are produced. Thus, the volumetric wear along this path is $V_a = a_x \cdot b_x \cdot \delta \cdot n_v n_o = (A_n/n_o^2)\cdot(a\cdot p/H)\cdot k\,(H/p)^2 = k\cdot a\cdot A_n \cdot p/H = k\cdot a\cdot N/H$. The result conforms to J.F. Archard's formula (5), where the friction path is equal to length a.

The dimensions $a_x = a/n_o$, $\delta = a\cdot p/H$; therefore, the linear specific intensity of wear is described with the following dependence:

$$i_h = \frac{\delta}{a_x} = 1, \qquad (17)$$

and the linear intensity I_h with:

$$I_h = i_h \frac{p}{H} = \frac{p}{H}. \qquad (18)$$

These are very high values.

A NEW INTERPRETATION OF VOLUMETRIC WEAR

A particle of tribological wear is here assumed to detach from each elementary contact of surface asperities. This is illustrated in **Fig. 7**. The geometric properties of the elements shown are the same as in **Fig. 4**, but the number of real contacts n_o marked in bold outlines is equal to the number of contacts subject to wear, n_v. Therefore, the likelihood of wear on an elementary real surface A_{ri} is 1 and $n_k = 1$. In keeping with J.F. Archard's formula (5), the wear coefficient $k \neq 0$ should be included in the proposed scheme (**Fig. 7**). This can be illustrated on a projection perpendicular to the nominal surface A_n, shown in **Fig. 8**.

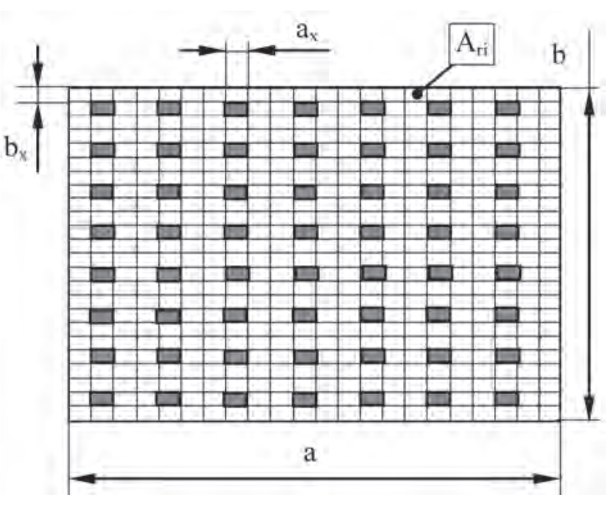

Fig. 7. Nominal surface $A_n = a\cdot b$ divided into n_o^2 identical elements, with a surface $A_{ri} = a_x \cdot b_x$; the number of elements marked with bold lines and shaded is $n_v = n_o$, these are the real surface asperities contacts subject to wear

Rys. 7. Powierzchnia nominalna $A_n = a\cdot b$ podzielona na n_o^2 jednakowych elementów każdy o powierzchni $A_{ri} = a_x \cdot b_x$; liczba elementów oznaczonych grubymi ramkami i zaciemnionych wynosi $n_v = n_o$, są to rzeczywiste styki nierówności powierzchni podlegające zużywaniu

The principle of volumetric wear after J. F. Archard addresses the likelihood of a wear particle detaching itself from an elementary contact of solid surface asperities. According to the principle, the number of contacts n_v distributed over a nominal surface A_n and subject to wear is the product of the number of real contacts n_o times the wear coefficient k: $n_v = k\cdot n_o$. This fails to address the role of the volume of energy dissipation zone situated immediately under the nominal surface. Therefore, the detachment of a wear particle can be mutually conditioned on the volume of the dissipation zone and wear coefficient k.

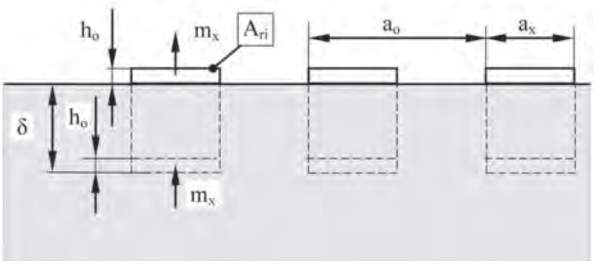

Fig. 8. A scheme of the detachment of a wear particle with the volume $A_{ri}\cdot h_o$ and mass m_x on the absorption of a single energy pulse

Rys. 8. Schemat oddzielania cząstki zużycia o objętości $A_{ri}\cdot h_o$ i masie m_x po wchłonięciu jednego impulsu energii

According to the new interpretation of temporary tribological wear (**Fig. 8**), a particle of mass m_x and volume $V_x = A_{ri}\cdot h_o$ is generated. The elementary linear wear h_o is the product of the dissipation zone depth δ times the wear coefficient k:

$$h_o = k\delta. \qquad (19)$$

The elementary volumetric wear V_x is in direct proportion to the elementary volume of the energy dissipation zone $V_o^x = A_{ri}\cdot\delta$:

$$V_x = kV_o^x. \qquad (20)$$

The volumetric element wear along the friction path equals $V_x \cdot n_o = k \cdot a_x \cdot b_x \cdot \delta \cdot n_o^2 = k \cdot (A_n/n_o^2) \cdot (a \cdot p/H) \cdot (H/p)^2 = k \cdot a \cdot A_n \cdot p/H$, that is:

$$V_a = ka\frac{N}{H}. \tag{21}$$

This result is produced assuming $(1/n_k)$ – the part of the volume of energy dissipation zone detached every time in effect of friction – is identical with J.F. Archard's dependence, based on the initial assumption of the involvement in the wear of $(1/n_k)$ – the number of surface asperities contacts n_o distributed over the nominal surface A_n.

The following can be described on the basis of the model of volumetric wear presented in this section:

The specific linear intensity of wear $i_h = h_o/a_x = (k \cdot \delta)/(a/n_o) = k \cdot (a \cdot p/H)/(a\ p/H)$, i.e.:

$$i_h = k, \tag{22}$$

and the linear intensity I_h:

$$I_h = k\frac{p}{H}. \tag{23}$$

(22) is identical with (15) and (16), which is proof the wear coefficient k and the specific linear intensity of wear i_h arrived at in the proposed model of wear are equal.

THE THERMODYNAMIC ANALYSIS OF THE MODELS OF TRIBOLOGICAL WEAR DISCUSSED

The pulse nature of energy dissipation in friction is a result of the matter's discontinuity across a nominal contact of elements in friction. In a model attributing wear at a given time only to n_v contacts out of their real number n_o, two types of energetic interactions can be distinguished. The generation of n_v particles is a result of mechanical work necessary for some defects of the material structure, especially the new surface of these particles, to develop. At the time, there is no wear in $(n_o - n_v)$ contacts and the work of friction is transformed into dissipation heat. An energy balance reflecting this situation can address the work of shear stress across elementary contact, equal to the friction coefficient times material hardness H. The elementary shear force $\mu \cdot H \cdot A_{ri}$ along the displacement a_x carries out elementary friction work $A_{tx} = \mu \cdot H \cdot A_{ri} \cdot a_x = \mu \cdot n_o \cdot p \cdot (A_n/n_o^2) \cdot a/n_o = \mu \cdot N \cdot a/n_o^2$. All n_o^2 elementary contacts will of course execute work $A_{ta} = \mu \cdot N \cdot a$ along a relative displacement of a elements. n_v of the contacts are subject to elementary mechanical work that rubs the material $A_{dyssx} = \mu \cdot H \cdot A_{ri} \cdot a_x\ n_v = \mu \cdot n_o \cdot p \cdot (A_n/n_o^2) \cdot (a/n_o) \cdot k \cdot n_o = \mu \cdot N \cdot (a/n_o) \cdot k = A_{tx} \cdot \eta = \mu \cdot N \cdot a/n_o^2 \cdot \eta$, since it constitutes a fraction η of the entire work of friction A_{tx}, hence:

$$\eta = kn_o, \tag{24}$$

which means the surface number n_v situated in the field of the nominal surface A_n and subject to wear along the displacement a_x is the efficiency of the wear process: $\eta = n_v$. η is below 1; therefore, fewer than one field of contact subject to wear is situated on the nominal surface at a given moment on average. The number of wear particles on covering a path of friction equal to the dimension a is $k \cdot n_o^2$. (24) was arrived at in **[L. 11]** by assuming the following sequences of transformations $(\mu H = \mu \cdot n_o \cdot p = \mu \cdot n_k \cdot \eta \cdot p)$ then simplifying and producing: $\eta = (n_o/n_k) = k \cdot n_o$.

Given that the wear coefficient k is equal to the specific linear intensity of wear i_h, its upper limit can be determined on the basis of (24), namely:

$$i_{hmax} = \frac{1}{n_o} = \frac{p}{H}, \tag{25}$$

whereas

$$I_{hmax} = \left(\frac{p}{H}\right)^2. \tag{26}$$

If only one wear particle is generated on the surface A_n following a friction path equal to the dimension a, that is, $k \cdot n_o^2 = 1$, the minimum specific linear intensity of wear i_{hmin} meets the following dependence:

$$i_{hmin} = \frac{1}{n_o^2} = \left(\frac{p}{H}\right)^2, \tag{27}$$

while

$$I_{hmin} = \left(\frac{p}{H}\right)^3. \tag{28}$$

The heating efficiency in the process of tribological wear η_c can be expressed with the following difference:

$$\eta_c = 1 - \eta. \qquad (29)$$

It reaches maximum at η_{min}, i.e., where $k_{min} = i_{hmin} = \left(\frac{p}{H}\right)^2$, hence the following results on the basis of (24):

$$\eta_{cmax} = 1 - \frac{p}{H} = \eta_{A\text{-}E}. \qquad (30)$$

This dependence conforms to the limited conversion of mechanical work into heat in T.A. Afanasjeva-Erhenfest thermodynamic circuit **[L. 12]**. Since mechanical work cannot be fully transformed into dissipation heat, the absence of wear in any friction contact cannot be expected from a thermodynamic point of view. Wear is an energy-consuming process, after all. Wear must occur with the minimum intensities described with (27) and (28) in sliding friction. Thus, the principal types of damage to a friction bond shown in **Fig. 1** (II, III, IV) should be treated as ideal, simplified models, given the thermodynamic limitations. They become more realistic for $n_k \to 1$. The model of volumetric wear in accordance with J.F. Archard's theory (**Figs. 2, 4, 6**) gives rise to certain doubts from the thermodynamic perspective as well. A lack of wear in most real contacts of surface asperities within a nominal surface cannot be assumed. The generation of few relatively large wear particles is a consequence of this assumption. In this way, the volume of worn material is correctly computed on the basis of a mechanism that is unacceptable from the physical point of view. The intensity of wear (17) and (18) is far above the values acceptable as per the equations (25) and (26).

The same argument can be replicated for the model of wear according to Figures 6 and 7 and the same conclusions can be produced. Namely, efficiency is derived from: $\delta = n_k \cdot h_o$ and $n_v \cdot n_k = n_o$:

$$\eta = \frac{h_o}{\delta} n_o, \qquad (31)$$

thereby including the geometric parameters of the friction zone δ and $\cdot h_o$ and eliminating n_v.

The assumption the removal of a material portion from the volume of energy dissipation is conditional on the likelihood the process provides for a new model of wear that is free from the above deficiencies. In particular, the volumes of worn material calculated on its basis correspond to the results of J.F. Archard's well-known formula.

CONCLUSIONS

The discussion in the present paper leads to the following conclusions:

1. A friction couple forms a system of bodies in pressure disequilibrium as it includes two surfaces, nominal and real, with different unit pressures. The work of friction in such a system cannot be fully converted into dissipation heat.
2. Energy dissipation in friction takes place in the real dissipation zone of a volume V_a^x whose dimensions are described with (9 –12).
3. Once a friction path l is covered, an apparent dissipation zone of a volume $V^x = \lambda \cdot V_a^x$ is produced, a trace of the mobile real zone where energy is not dissipated directly in effect of friction.
4. The linear dimension a of the smaller friction couple element has the property of a friction path unit following which another cycle of energy dissipation and wear is replicated in both the friction couple elements and the apparent volume of friction is multiplied.
5. Part of the friction work, dissipation heat, is equal to heat in T.A. Afanasjeva-Erhenfest cycle, and its remaining portion is mechanical and causes the material to disperse.
6. Material is always separated in the effect of friction at each elementary contact of asperities.
7. The model proposed in this paper addresses the occurrence of wear at each elementary surface contact and enables a correct evaluation of volumetric wear as per (5).
8. Wear coefficient k can be interpreted as the following: the wear number **ν**, specific linear intensity of wear i_h, the ratio of linear wear to the depth of friction zone h_o/δ, the ratio of wear surface to real surface A_z/A_r, and the ratio of the wear volume to the friction volume V/V^x.
9. The boundary values of wear coefficient k and of wear intensity i_h, I_h are described with (25–28) as functions of unit pressures p and H or the number of real asperities contacts distributed over the nominal surface A_n.
10. The efficiency of tribological wear process η is the product of wear coefficient k times the

number of real contacts n_o – (24) – or the ratio h_o/δ times n_o.

11. The detachment of n_v tribological wear particles, after n_k replications of the energetic loading of surface asperities contact and the absence of wear across (n_o – n_v) contacts, is reasonable in mechanistic terms, not in the light of the thermodynamic analysis of the friction process of solids.
12. Wear coefficient k should not be interpreted as a share of the number of elementary contacts n_v subject to wear in the number n_o of real contacts on the nominal surface A_n, but as the share of the volume of a generated particle in the volume of energy dissipation zone at the time of friction.
13. The discussion based on the model as postulated by J.F. Archard, **Figs. 2, 4,** and **6**, allows for a correct evaluation of volumetric wear as per (5); however, a wear particle generated on n_k replications and its corresponding specific linear intensity of wear i_h reach very high values that have no grounding in thermodynamics.

REFERENCES

1. Leszek W.: Metodologiczne podstawy badań tribologicznych, PWN. Warszawa–Poznań 1981.
2. Solski P., Ziemba S.: Zagadnienia zużycia elementów maszyn spowodowanego tarciem, PWN, Warszawa 1969.
3. Archard J.F.: Contact and rubbing of flat surfaces, "J. Appl. Phys." 1953, 24, No. 8, pp. 981–988.
4. Kragielski I.V., Dobyčin M.N., Kombalov V.S.: Osnovy rasčetov na trenie i iznos, "Mašinostroenie", Moskva 1977.
5. Holm R.: Electrical Contacts, H. Gerbers, Stockholm 1946, p. 398.
6. Burwell J.T., Strang C.D.: On the Empirical Law of Adhesive Wear, "J. Appl. Phys." 1952, 23, 1, pp. 18–28.
7. Fleischer G., Gröger H., Thum H.: Verschleiß und Zuverlässigkeit, VEB Verlag Technik, Berlin 1980.
8. Sadowski J.: Die Verschleißzahl, "Tribologie und Schmierungstechnik" 2007, No. 4, pp. 52–57.
9. Sadowski J.: Über die Wahrscheinlichkeit der Lostrennung der Verschleißpartikel aus den Reibkörpern, "Tribologie und Schmierungstechnik" 2013, No. 4, pp. 41–51.
10. Sadowski J.: Reibungsvolumen als tribologisches System, "Tribologie und Schmierungstechnik" 2008, No. 5, pp. 43–48.
11. Sadowski J.: Zür Energiebilanzierung bei der Festkörperreibung, "Tribologie und Schmierungstechnik" 2009, No. 6, pp. 27–31.
12. Sadowski J.: Irreversibility of friction and wear processes, "Tribologia" 2020, No 5, pp. 51–60.

NOMENCLATURE

a – length of the smaller friction element, measured in the direction of friction velocity [m],
a_i – radius of the circle of elementary asperities contact in J.F. Archard's model[m],
a_o – linear dimension of area l contact surface parallel to [m],
a_x – linear dimension of an elementary asperities contact surface parallel to [m],
A_{dyssx} – elementary mechanical work of dissipation [J],
A_n – nominal contact surface of solids in friction [m^2],
A_r – real contact surface of solids in friction [m^2],
A_{ri} – elementary surface of asperities contact [m^2],
A_t – work of friction [J],
A_{ta} – work of friction performed along the displacement a [J],
A_{tx} – elementary work of friction performed along the displacement a_x [J],
b – width of the friction path [m],
b_o – linear dimension of area l contact surface perpendicular to a [m],

b_x – linear dimension of an elementary asperities contact surface perpendicular to a [m],
h – linear wear [m],
h_o – elementary linear wear[m],
H – hardness [MPa],
i_h – specific linear intensity of wear,
I_h – linear intensity of wear,
k – wear coefficient,
l – friction path[m],
m – mass wear [kg],
m_o – mass of a wear particle as per J.F. Archard's model [kg],
m_x – mass of a wear particle as per the proposed model [kg],
n_o – number of real asperity contacts on the nominal contact surface,
n_k – critical number of contacts conditioning the generation of a wear particle,
n_v – number of friction particles generated on the nominal contact surface,
N – normal force [N],
p – nominal unit pressure [MPa],
R – radius of an asperity apex rounding [m],
T – friction force [N],
v – friction velocity [$m \cdot s^{-1}$],
V – volume, volumetric wear [m^3],
V_i – volume of a semi-spherical wear particle [m^3],
V_a – volumetric wear along the friction path [m^3],
V_a^x – real volume of the friction zone [m^3],
V_o^x – elementary volume of energy dissipation zone [m^3],
V_x – volume of wear particle [m^3],
V^x – apparent volume of the friction zone [m^3],
δ – depth of the friction zone [m],
η – efficiency of the wear process,
η_c – thermal efficiency of frictional heating,
η_{A-E} – efficiency of T.A. Afanasjeva-Erhenfest cycle,
λ – reverse of the coefficient of mutual surface cover,
μ – friction coefficient,
ν – wear number,
ρ – density [$kg \cdot m^{-3}$],
σ_T – limit of material plasticity [MPa],
τ – shear stress [MPa],

$\frac{d\tau}{dh}$ – gradient of shear stress [MPa/m],

1, 2 – an index of a friction pair element.