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45 YEARS OF DELIBERATIONS ON THE THERMO-MECHANICAL INTERPRETATION OF FRICTION AND WEAR. PART I: THE MACROSCOPIC INTERPRETATION OF THE TRIBOLOGICAL PROCESS

45 LAT ROZWAŻAŃ O CIEPLNO-MECHANICZNEJ INTERPRETACJI TARCIA I ZUŻYWANIA. CZĘŚĆ I: INTERPRETACJA MAKROSKOPOWA PROCESU TRIBOLOGICZNEGO

Key words: thermodynamic system, friction couple, friction area, extensive parameters, friction couple, wear, dissipation heat, work of mechanical dissipation, enthalpy, internal energy, specific work of wear, the first law of thermodynamics.

Abstract: This first part will consider the friction couple or its part, identified with the open thermodynamic system. Dependences among extensive parameters and energetic interactions in the system and its boundaries with the environment are described in analytical terms. The dimensions of the energy dissipation zone, where the friction of solids takes place, are established. The thermo-mechanical nature of the tribological process is demonstrated. The wear conditions and distribution of energy generated by its particular elements are determined. A new quantity is introduced – the specific work of wear, which characterises the wear resistance of the tribological system and its parts. The effect of the reciprocal cover of solids' friction surfaces on the energy balance structure in calorimetric testing is analysed. The concept of generalised wear is introduced and standardised. The discussion is restricted to processes at the macroscopic level of matter organisation.

Słowa kluczowe: system termodynamiczny, para tarciowa, strefa tarcia, parametry ekstensywne, praca tarcia, zużywanie, ciepło dyssypacji, praca dyssypacji mechanicznej, entalpia, energia wewnętrzna, praca właściwa zużycia, pierwsza zasada termodynamiki.

Streszczenie: Przedmiotem rozważań w pierwszej części pracy jest para tarciowa lub jej fragment utożsamiona z systemem termodynamicznym otwartym. Opisano analitycznie zależności między parametrami ekstensywnymi i oddziaływaniami energetycznymi w systemie i na jego graniach z otoczeniem. Ustalono wymiary strefy dyssypacji energii, w której zachodzi tarcie ciał stałych. Wykazano cieplno-mechaniczną naturę procesu tribologicznego. Ustalono warunki zachodzenia zużywania i rozdziału energii rozproszonej przez poszczególne jego elementy. Wprowadzono nową wielkość – pracę właściwą zużycia, charakteryzującą odporność na zużywanie systemu tribologicznego i jego elementów. Zanalizowano wpływ wzajemnego przykrycia powierzchni tarcia ciał na strukturę bilansu energii w badaniach kalorymetrycznych. Wprowadzono pojęcie zużycia uogólnionego i jego standaryzację. Rozważania ograniczono do procesów zachodzących na makroskopowym poziomie hierarchicznym organizacji materii.

INTRODUCTION

The author proposed the hypothesis of the thermo-mechanical nature of friction resistance for the first time in his doctoral dissertation of 1977 **[L. 1]**. The thermal and mechanical parts of the friction coefficient were described based on the energy balance equation and an original model of the source of heat that relied on the third-type boundary condition of heat capture at the contacts

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of solids with rough contact surfaces. The thermo-mechanical interpretation of friction and its effects was developed in successive publications based on more general assumptions. Some significant progress was achieved when the friction couple was identified with the open thermodynamic system **[L. 2]**. Basic physical concepts and quantities were considered at that stage for the purpose of a systemic analysis of phenomena concomitant with the mechanical energy dissipation based on the equation for the first law of thermodynamics.

Furthermore, relying on the laws of mass and energy conservation and Newton's third law of dynamics on the interaction of bodies, a range of analytical dependencies were determined that characterise the effects between two bodies in friction. The shares of energy dissipated by the particular elements in friction were established. The wear resistance was characterised as a systemic quantity. The conditions of wear and its absence, frictional deposition, and the stages of grinding stabilised and accelerated wear were formulated. The necessary and sufficient condition of grinding as the process of interference with stabilised wear was provided. This condition implies the known practical conditions of grinding. The conclusion was drawn from the energy balance for stationary friction that there is a temperature characteristic of a maximum resistance to wear. Wear may result not only in a mass (volume) wastage but also in an increment of a system's internal energy, expressed as changes to the properties of its material. Thermodynamic conditions were determined for that case concerning the potential for and progress of wear.

J.F. Archard's classic formula for volumetric wear was another major assumption adopted for the purposes of the study. It helped define the energy dissipation zone out of the friction couple, treat it as a thermodynamic system, and ignore secondary phenomena in the analysis. An attempt at generalising and standardising the measures of wear caused by friction summarises discussions on the improvement of comparability of tribological test results. Both the friction couple and the zone of energy dissipation are macroscopic objects, and the author chose some regularities known from phenomenological tribology to describe the processes in these objects. In the second part of this study, the description is expanded to apply to a lower observation scale, where elementary contacts of solid asperities play significant roles. The energy dissipation and wear at this observation level are of a pulsed nature.

Some significant results arising from the microscopic analysis of friction include the interpretation of wear as a process of material disintegration, an effect of mechanical work of volumetric and superficial dissipation, new proposed measures of wear and resistance to wear, the discovery and interpretation of the cooling mechanism concomitant on wear, and a method of determining the chief components of the energy balance without a calorimeter. Moreover, the separation mechanism of wear product particles is modified compared to the initial interpretation suggested by J.F. Archard **[L. 3]**. Given the limited potential for converting mechanical work into friction heat and following an analysis of T.A. Afanasjeva-Ehrenfest's **[L. 4]** thermodynamic circuit, maximum and minimum linear intensities of wear are determined. The second law then restricts the condition of wear-free friction derived from the first law of thermodynamics.

The reproducibility and comparability of results arrived at by different authors is another major issue in the field of tribological testing. These difficulties are conditioned not only by the complexity of friction and its consequences but also by the application of notions and quantities that do not always have unambiguous physical interpretation or by the omission of thermodynamic quantities characterising the nature of friction. For instance, an assessment of the impact of temperature on the process of friction depends on the location of a measurement point in the tribological system. This is due to the fact that the quantity is defined according to the zero law of thermodynamics and thus only applicable to this point; therefore, it cannot clearly characterise the thermal state of the system. The latter is better represented with the friction area's temperature gradient, related to the quantity of dissipation heat generated.

The testing of dissipation heat – the main component of the friction energy balance – is particularly complicated in technical terms and, therefore, rarely presented in the literature. The author's testing of steel dry friction corroborates a major share of dissipation heat in the energy balance and allows for an assessment of the maximum temperature of elementary contacts of surface asperities. It also discloses a pulsation of internal energy on the contact boundaries of the bodies tested, caused by a reciprocal covering of friction surfaces different from one. Unit pressures at the contacts of surface asperities of bodies in friction are an important parameter of friction, and the information on their values is obtained indirectly due to the discrete and pulse nature of body contacts at the small observation scale. To this end, the author uses a thermo-mechanical model of friction and wear while meeting the laws of mass and energy conservation.

Some major results arrived at during the last 45 years are introduced in the two parts of this paper. The first is devoted to interpreting the tribological process at the macroscopic level of matter organisation, including the friction couple or its selected fragment.

A LIST OF KEY TERMINOLOGY

A_n – nominal contact surface of solids 1 and 2 [m^2],
A_r – real contact surface of solids 1 and 2 [m^2],
A_z – wear surface [m^2],
A_{t1-2}– work of friction [J],
A_{dyss}– work of mechanical dissipation [J],
$\dot{A}$– friction power [W],
$\ddot{A}$– time derivative of friction power [W/s],
$\dot{A}_{dyss}$– power of mechanical dissipation [W],
a – dimension of A_n surface side measured in the direction of friction path l [m],
a_{dyss} – specific work of mechanical dissipation [J/kg],
b – width of friction path [m],
dF – elementary field of solid contact [m^2],
e_R^x – specific work of wear [J/kg],
e_R^z – generalized specific work of wear [J/kg], [J/m^3],
e_R^* – density of wear energy [J/m^3],
H – hardness [MPa],
h – linear wear along the friction path l [m],
h_o– linear wear along the friction path a [m],
i – mean specific enthalpy of wear products [J/kg],
ΔI – enthalpy change [J],
$\dot{I}$ – time derivative of enthalpy I [W],
J – specific intensity of wear [kg/m^{2}s],
k – coefficient of wear,
l – friction path [m],
Δm – change of the system mass [kg],
m – mass wear [kg],
$\dot{m}$ – flux of wear product mass [kg/s],
$\ddot{m}$ – time derivative of $\dot{m}$ [kg/s^2],
N – normal force affecting the surface A_n [N],
n_k– critical number of surface asperity contacts,
p – unit pressure [MPa],
q – unit friction power [W/m^2],
q_c – unit thermal flux [W/m^2],
q_m – unit power of mechanical dissipation [W/m^2],
Q_{1-2} – heat [J],
Q_{dyss} – heat of dissipation [J],
$\dot{Q}$ – heat flux [W],
$\ddot{Q}$ – time derivative of $\dot{Q}_{dyss}$ [W/s],
Q_{dyss} – flux of dissipation heat [W],
t – friction time [s],
T – friction force [N],
T_c – thermal component of T [N],
T_m– mechanical component of T (conditioning wear) [N],
ΔU – change of the system's internal energy [J],
$\dot{U}$ – momentary change of the system's internal energy [W],
$\ddot{U}$ – second time derivative of internal energy [W/s],
V – volumetric wear [m^3],
V_a^x – real (momentary) volume of friction area [m^3],
V_l^x– volume of friction area along the path l [m^3],
v – friction velocity [m/s],
Z – generalized wear [m^3], [kg],
δ – thickness of friction area [m],
λ – reverse coefficient of surface cover,
μ – coefficient of friction,
μ_c – thermal coefficient of friction,
μ_m – mechanical coefficient of friction,
Θ – temperature of friction contact [K],
Θ_o – 'flash' temperature [K],
Θ_x – characteristic temperature [K],
τ – time [s],
1, 2 – indices attributed to the elements 1 and 2 of the system, respectively;
1-2 – symbol of the system's state change between the states 1 and 2.

THE TRIBOLOGICAL SYSTEM AS AN OPEN THERMODYNAMIC SYSTEM [L. 5, 6]

The friction couple as a thermodynamic system

This is a list of basic dependences that characterise the tribological system – **Fig. 1 [L. 5]**, **[L. 6]**.

$$\Delta U = -\Delta I - Q_{1-2} + A_{t1-2}, \tag{1}$$

$$\Delta U_1 = -\Delta I_1 - Q_{1-21} + A_{t1-21}, \tag{2}$$

$$\Delta U_2 = -\Delta I_2 - Q_{1-22} + A_{t1-22}, \tag{3}$$

$$\Delta I = i\Delta m, \tag{4}$$

$$\Delta I_1 = i_1 \Delta m_1, \quad (5)$$

$$\Delta I_2 = i_2 \Delta m_2, \quad (6)$$

$$A_{t1\text{-}2} = \int_0^t T(\tau) \cdot v(\tau) \cdot d\tau, \quad (7)$$

$$A_{t1\text{-}21} = \int_0^t T_1(\tau) \cdot v_1(\tau) \cdot d\tau, \quad (8)$$

$$A_{t1\text{-}22} = \int_0^t T_2(\tau) \cdot v_2(\tau) \cdot d\tau, \quad (9)$$

where: $t \geq \tau \geq 0$.

$$\Delta U = \Delta U_1 + \Delta U_2, \quad (10)$$

$$Q_{1\text{-}2} = Q_{1\text{-}21} + Q_{1\text{-}22}, \quad (11)$$

$$A_{t1\text{-}2} = A_{t1\text{-}21} + A_{t1\text{-}22}, \quad (12)$$

$$\Delta I = \Delta I_1 + \Delta I_2, \quad (13)$$

$$\Delta m = \Delta m_1 + \Delta m_2, \quad (14)$$

$$T = T_1 = \left| -T_2 \right|, \quad (15)$$

$$v = v_1 + v_2, \quad (16)$$

$$l = l_1 + l_2, \quad (17)$$

$$A_{t1\text{-}2} = Q_{dyss} + A_{dyss}, \quad (18)$$

$$A_{dyss} = A_{dyss1} + A_{dyss2}, \quad (19)$$

$$Q_{dyss} = Q_{dyss1} + Q_{dyss2}, \quad (20)$$

$$Tl = T_1 l_1 + T_2 l_2, \quad (21)$$

$$\eta = A_{dyss}/A_{t1\text{-}2}, \quad (22)$$

$$e_R^x = A_{t1\text{-}2}/\Delta m, \quad (23)$$

$$e_R^* = A_{t1\text{-}2}/V. \quad (24)$$

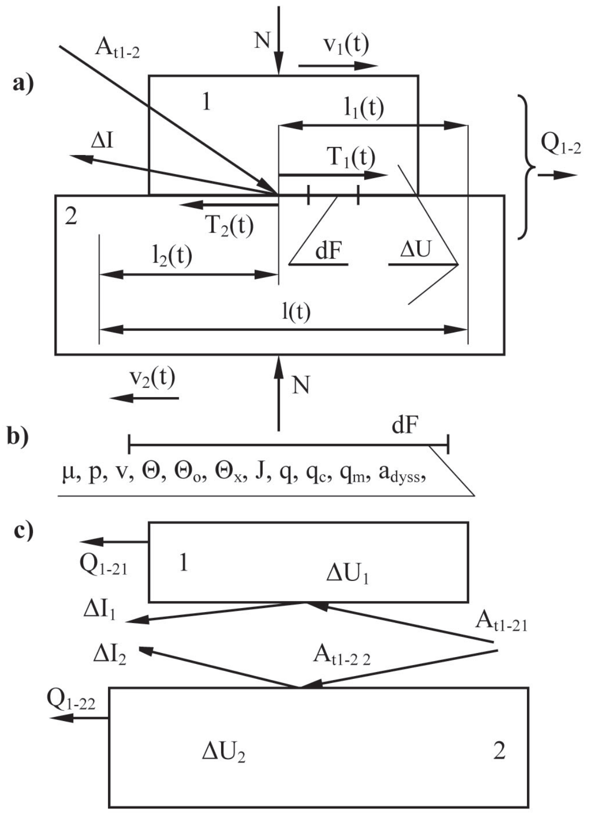

Fig. 1. The tribological system as an open thermodynamic system: a) a system of solids 1 and 2 in friction as a thermodynamic system, b) elementary area of friction contact dF and its assigned physical parameters, c) each of the solids in friction as a thermodynamic system [L. 5]

Rys. 1. System tribologiczny jako system termodynamiczny otwarty: a) układ trących się ciał 1 i 2 jako system termodynamiczny, b) elementarne pole styku tarciowego dF i parametry fizyczne przypisane do niego, c) każde z trących się ciał jako systemy termodynamiczne **[L. 5]**

The thermo-mechanical structure of kinetic friction resistances

Thermo-mechanical tribological hypothesis

The friction force T is a sum total of the following components: thermal T_c, responsible for heating (the generation of dissipation heat Q_{dyss}), and mechanical, T_m, responsible for tribological wear (in effect of dissipation work A_{dyss}):

$$A_{t1\text{-}2} = \int_0^t T(\tau) \cdot v(\tau) \cdot d\tau = \int_0^t T_c(\tau) \cdot v(\tau) \cdot d\tau + \int_0^t T_m(\tau) \cdot v(\tau) \cdot d\tau \quad (25)$$

$$Q_{dyss} = \int_0^t [1 - \eta(\tau)] \cdot T(\tau) \cdot v(\tau) \cdot d\tau \qquad (26)$$

$$A_{dyss} = \int_0^t \eta(\tau) \cdot T(\tau) \cdot v(\tau) \cdot d\tau \qquad (27)$$

where: $0 \leq \eta \leq 1$.

(25) and (28) illustrate the relationships among A_{t1-2}, its components Q_{dyss}, A_{dyss}, heat Q_{1-2}, enthalpy change ΔI, and internal energy change ΔU of the thermodynamic system.

$$\Delta U = -\Delta I - Q_{1-2} + \overbrace{\overbrace{\int_0^t T_c(\tau)\cdot v(\tau)\cdot d\tau}^{Q_{dyss}} + \underbrace{\int_0^t T_m(\tau)\cdot v(\tau)\cdot d\tau}_{A_{dyss}}}^{A_{t1-2}} \qquad (28)$$

Based on the theorem of integral derivative as a function of its upper boundary for any time, (25) implies the following relationships:

$$T(t) \cdot v(t) = \frac{[1 - \eta(t)] \cdot T(t) \cdot v(t)}{T_c(t)} + \frac{\eta \cdot T(t) \cdot v(t)}{T_m(t)} \qquad (29)$$

or

$$T(t) = T_c(t) + T_m(t). \qquad (30)$$

The force of kinetic friction T(t) is a sum total of the thermal $T_c(t)$ and mechanical $T_m(t)$ components. The coefficient of kinetic friction μ is a sum total of the thermal μ_c and mechanical μ_m coefficients of friction:

$$\mu = \frac{T}{N} = \frac{T_c}{N} + \frac{T_m}{N} = \mu_c + \mu_m \qquad (31)$$

The thermodynamic conditions of tribological wear

Based on the first law of thermodynamics for open systems (1), the mass of wear products is defined by:

$$\Delta m = \frac{-\Delta U - Q_{1-2} + A_{t1-2}}{i}. \qquad (32)$$

The wear takes place where:

$$A_{t1-2} > \Delta U + Q_{1-2}, \qquad (33)$$

The wear doesn't take place where:

$$A_{t1-2} = \Delta U + Q_{1-2}, \qquad (34)$$

The condition of friction deposition:

$$A_{t1-2} < \Delta U + Q_{1-2}, \qquad (35)$$

The condition of accelerated wear:

$$\ddot{m} > 0;\ \ddot{A} > \ddot{U} + \ddot{Q} \qquad (36)$$

The condition of stabilized wear:

$$\ddot{m} = 0;\ \ddot{A} = \ddot{U} + \ddot{Q}, \qquad (37)$$

The condition of grinding:

$$\ddot{m} < 0;\ \ddot{A} < \ddot{U} + \ddot{Q}. \qquad (38)$$

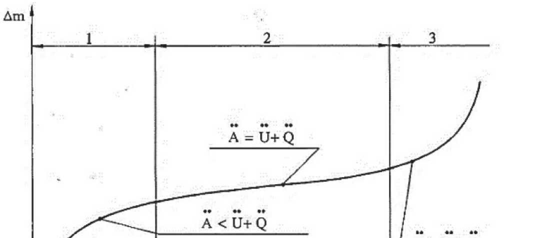

Fig. 2. The course of wear Δm over time t – Lorenz curve and its thermodynamic waveform: 1 – the period of grinding, 2 – the period of stabilised wear, 3 – the period of accelerated wear [L. 5]

Rys. 2. Przebieg zużycia Δm w czasie t – krzywa Lorenca i jej termodynamiczna charakterystyka: 1 – okres docierania, 2 – okres zużywania ustabilizowanego, 3 – okres zużywania przyspieszonego **[L. 5]**

Wear may also be measured with the internal energy increment ΔU. In the event:

The wear takes place where:

$$A_{t1-2} > \Delta I + Q_{1-2}, \qquad (39)$$

The wear doesn't take place where:

$$A_{t1\text{-}2} = \Delta I + Q_{1\text{-}2}, \tag{40}$$

Grinding takes place where:

$$\dot{A} < \dot{I} + \dot{Q}, \tag{41}$$

Accelerated wear takes place where:

$$\dot{A} > \dot{I} + \dot{Q}, \tag{42}$$

Stabilized wear takes place where:

$$\dot{A} = \dot{I} + \dot{Q}. \tag{43}$$

The thermodynamic conditions of resistance to wear [L. 6]

These are the energetic measures of resistance to wear:

specific work of wear:

$$e_R^x = \frac{A_{t1\text{-}2}}{\Delta m} = \frac{i}{1 - \frac{\Delta U}{A_{t1\text{-}2}} - \frac{Q_{1\text{-}2}}{A_{t1\text{-}2}}}, \tag{44}$$

and mean enthalpy of wear products:

$$i = \frac{\Delta I}{\Delta m}. \tag{45}$$

If the wear resistance of elements 1 and 2 of a system is defined as:

$$e_{R1}^x = \frac{A_{t1\text{-}21}}{\Delta m_1}, \tag{46}$$

$$e_{R_2}^x = \frac{A_{t1\text{-}22}}{\Delta m_2}, \tag{47}$$

and

$$i_1 = \frac{\Delta I_1}{\Delta m_1}, \tag{48}$$

$$i_2 = \frac{\Delta I_2}{\Delta m_2}, \tag{49}$$

then, relying on (12), (13), and (46) – (49) leads to the equality of the specific works of wear:

$$e_R^x = e_{R1}^x = e_{R2}^x, \tag{50}$$

and of the specific enthalpies of wear products:

$$i = i_1 = i_2. \tag{51}$$

Thus, resistance to wear is a characteristic of a tribological system that assumes identical values for the system and its particular elements. There is no resistance to material wear. (44) implies that the resistance to wear is the greater, the greater the specific enthalpy of wear products that characterises the mechanism of tribological wear in quantitative terms and the greater the increment of the system's internal energy ΔU and the energy released to the environment as heat $Q_{1\text{-}2}$ after the work of friction $A_{t1\text{-}2}$ is executed.

A contribution to the nature of the work of friction forces [L. 6]

The share of rubbing solids in the process of mechanical energy dissipation

The equality for the specific works of wear (50) implies the friction work components assigned to the tribological system's particular elements.

$$A_{t1\text{-}21} = A_{t1\text{-}2}\frac{\Delta m_1}{\Delta m} = \Delta m_1 e_R^x, \tag{52}$$

$$A_{t1\text{-}22} = A_{t1\text{-}2}\frac{\Delta m_2}{\Delta m} = \Delta m_2 e_R^x. \tag{53}$$

In addition, the system of (1) – (3) and (50) implies more relationships:

$$\Delta U_1 + Q_{1\text{-}21} = \frac{\Delta m_1}{\Delta m}(\Delta U + Q_{1\text{-}2}), \tag{54}$$

$$\Delta U_2 + Q_{1\text{-}22} = \frac{\Delta m_2}{\Delta m}(\Delta U + Q_{1\text{-}2}). \tag{55}$$

Conditions (12), (17), (21) as well as (50), (52) and (53) express the displacements l_1 and l_2 (**Fig. 1**) of friction forces T_1 and T_2 as follows:

$$l_1 = \frac{\Delta m_1}{\Delta m} l, \tag{56}$$

$$l_2 = \frac{\Delta m_2}{\Delta m} l. \tag{57}$$

The velocities of T_1 and T_2 displacements for the time t:

$$v_1 = \frac{\Delta\, m_1}{\Delta\, m} v, \tag{58}$$

$$v_2 = \frac{\Delta\, m_2}{\Delta\, m} v, \tag{59}$$

and for any time τ ($0 \leq \tau \leq t$) are described similarly:

$$v_1(\tau) = \frac{\dot{m}_1(\tau)}{\dot{m}(\tau)} v(\tau), \tag{60}$$

$$v_2(\tau) = \frac{\dot{m}_2(\tau)}{\dot{m}(\tau)} v(t_a). \tag{61}$$

The friction zone as a thermodynamic system[L. 7]

The dissipation of kinetic energy and the wear of solids are restricted to the friction zone, an area in space situated immediately beneath the surface of these solids' contact. The macroscopic interpretation of the zone is illustrated in **Fig. 3**. Its characteristic quantities are specified below:

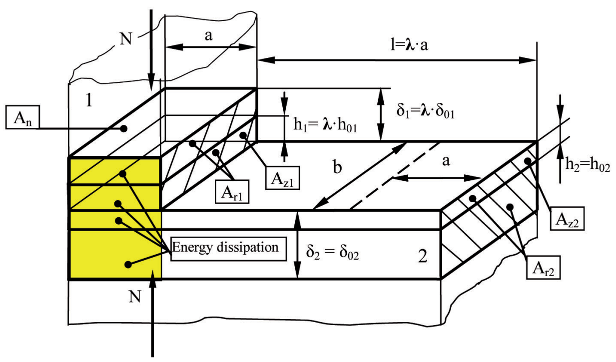

Fig. 3. A diagram of solids 1 and 2 in friction highlighting the energy dissipation zone [L. 7]

Rys. 3. Schemat układu trących się ciał 1 i 2 z zaznaczoną strefą dyssypacji energii **[L. 7]**

Analytical dependences among the quantities in the above Figure are determined on the basis of volumetric wear V, described by I.F. Archard's classic formula **[L. 3]**:

$$V = klN/H. \tag{62}$$

The real surface of solid contact:

$$A_r = \frac{N}{H}. \tag{63}$$

Wear surface:

$$A_z = kA_r. \tag{64}$$

Volumetric wear:

$$V = lA_z. \tag{65}$$

The linear wear of the particular elements:

$$h_1 = \frac{V_1}{A_n}, \tag{66}$$

$$h_2 = \frac{V_2}{\lambda \cdot A_n}. \tag{67}$$

The volumes of friction zones on the path l:

$$(V_l^x)_1 = \delta_{01}\lambda A_n = \delta_1 A_n, \tag{68}$$

$$(V_l^x)_2 = lA_{r2} = \delta_2 \lambda A_n. \tag{69}$$

In fact, energy is dissipated within the volume $A_n\delta$ because the force N only acts on the nominal surface A_n or along the path of friction a. Thus, the volume of a real (momentary) friction is described with the formulas for elements 1 and 2, respectively:

$$(V_a^x)_1 = aA_{r1} = a\frac{N}{H_1} = \delta_{01}A_n, \tag{70}$$

$$(V_a^x)_2 = aA_{r2} = a\frac{N}{H_2} = \delta_{02}A_n, \tag{71}$$

while the thickness of the friction zone, with:

$$\delta_1 = \frac{ap}{H_1}, \tag{72}$$

$$\delta_2 = \frac{ap}{H_2}. \tag{73}$$

The analysis of tribological processes in a thermodynamic system limited to a space which is part of the friction couple (**Fig. 4b**) enables us to ignore secondary processes concomitant on friction. The dimensions of the space, particularly the zone thicknesses δ_{01} and δ_{02}, are important for the production technologies of wear-resistant coatings.

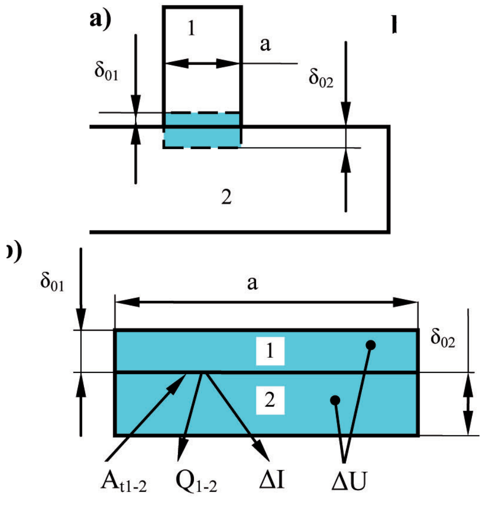

Fig. 4. The boundary of the tribological system: a) – where it is constituted by two elements 1 and 2 in friction and by a friction volume V_a^x, b) – where a friction volume of the dimensions a·(δ_{01} + δ_{02}) constitutes a tribological system; the width of friction path b is perpendicular to the plane of the Figure [L. 8]

Rys. 4. Granica systemu tribologicznego: a) – w przypadku, gdy system tworzą dwa elementy trące 1 i 2 wraz z objętością tarcia V_a^x, b) – w przypadku, gdy objętość tarcia o wymiarach a·(δ_{01} + δ_{02}) stanowi system tribologiczny; szerokość ścieżki tarcia b jest prostopadła do płaszczyzny rysunku **[L. 8]**

The impact of reciprocal covering of friction surfaces on the structure of energy balance [L. 9]

Three components are distinguished in calorimetric testing of the energy balance characterising stationary tribological processes, namely: friction power $\dot{A}$, flux of dissipation heat $\dot{Q}_{dyss}$, and power of mechanical dissipation $\dot{A}_{dyss}$ **[L. 9, 10]**:

$$\dot{A} = \dot{Q}_{dyss} + \dot{A}_{dyss}. \tag{74}$$

$\dot{A}$ is determined when A and t are known. When $\dot{Q}_{dyss}$ is calculated, the equality of the fluxes of dissipation heat $\dot{Q}_{dyss}$ and the heat discharged to the environment $\dot{Q}$ is assumed, which allows for an assessment of the power of mechanical dissipation work $\dot{A}_{dyss}$ based on the relation $\dot{A}_{dyss} = \dot{A} - \dot{Q}$. It is correct to assume that only stationary processes take place in a tested object. If the degree of reciprocal covering of the friction specimen and counter specimen is below 1, stationary thermal processes occur in the smaller element and quasi-stationary processes in the larger element. This denotes some momentary internal energy variations in the friction couple's larger element. The following dependence applies to this case of friction, **Figs. 5** and **6**:

$$\dot{Q}_{dyss} = \dot{Q} + \dot{U}. \tag{75}$$

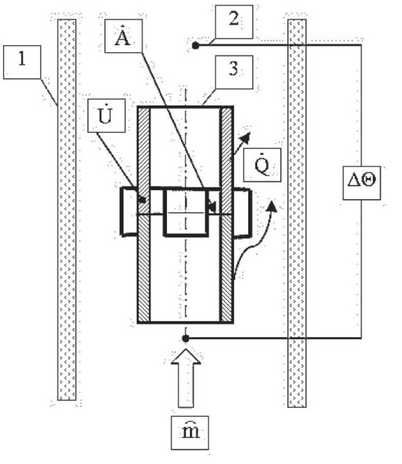

Fig. 5. A schematic representation of friction couple 3 in a flow calorimeter and the major components of the energy balance: 1 – the insulation wall of the calorimeter, 2 – differential thermocouple, $\hat{m}$ – the flux of calorimetric fluid, ΔΘ temperature increment caused by the heat flux $\dot{Q}$ [L. 9, 10]

Rys. 5. Schematyczne przedstawienie pary ciernej 3 w kalorymetrze przepływowym oraz główne składowe bilansu energetycznego: 1 – ścianka izolująca kalorymetru, 2 – termopara różnicowa, $\hat{m}$ – strumień cieczy kalorymetrycznej, ΔΘ przyrost temperatury wywołany strumieniem ciepła $\dot{Q}$ **[L. 9, 10]**

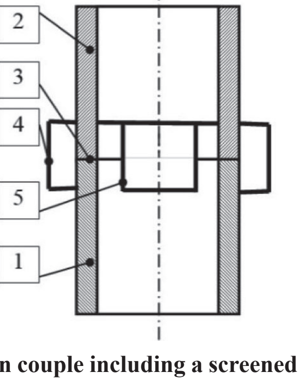

Fig. 6. A friction couple including a screened calorimetric fluid screen: 1, 2 elements in friction, 3 – friction surface, 4 – outside guard, 5 – inside guard

Rys. 6. Para cierna z ekranowanym stykiem od cieczy kalorymetrycznej: 1, 2 elementy trące, 3 – powierzchnia cierna, 4 – osłona zewnętrzna, 5 – osłona wewnętrzna

Two pipes with their fronts in contact are the rubbing elements in the calorimeter; their contact is inaccessible to the calorimetric fluid owing to dedicated guards. The top, revolving specimen has a ring contact surface, and the bottom specimen, the form of three protrusions – **Fig. 7**.

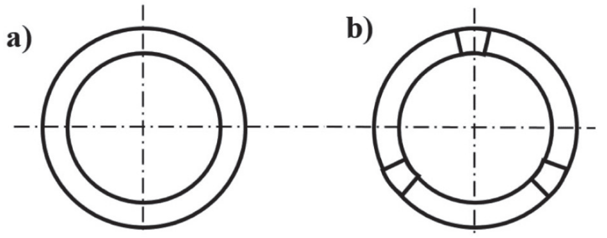

Fig. 7. The friction surfaces of specimens: a) the ring friction surfaces of the upper, revolving specimen 2, b) Three segments of the friction surface of the bottom, non-revolving specimen 1 [L. 9, 10]

Rys. 7. Powierzchnie tarcia próbek: a) pierścieniowa powierzchnia tarcia górnej próbki 2 obracającej się, b) Trzy segmenty powierzchni tarcia dolnej próbki 1 – nie obrotowej **[L. 9**, **10]**

The reciprocal covering of the friction surfaces – the relationship of nominal surface areas – corresponds to the relationship of friction times t_o/t. t_o can be defined as the time of the heat pulse acting on the ring surface, t – the time of its relaxation. t_o/t can be established experimentally as a function of powers: $\dot{A}$ and $\dot{Q}$. The flux of dissipation heat $\dot{Q}_{dyss2}$ from element 2 can be described as:

$$\dot{Q}_{dyss2} \frac{\Delta m_2}{\Delta m} = \dot{Q}_2 + \dot{U}_2, \tag{76}$$

where: $\dot{U}_2$ – the variations of internal energy.

The variations of internal energy $\dot{U}_2$ as a component of $\dot{Q}_{dyss}$ – (76) means $\dot{Q}$ received by the calorimetric fluid from the friction couple dependent on t_o/t is lower than $\dot{Q}_{dyss}$. The temporal variation of internal energy can be expressed, therefore, as a function of heat flow, namely:

$$\dot{U}_2 = \dot{Q}_2 \left(1 - \frac{t_o}{t}\right). \tag{77}$$

(76) becomes:

$$\dot{Q}_{dyss2} = \dot{Q}_2 \left(2 - \frac{t_o}{t}\right). \tag{78}$$

If $t_o = t$, then obviously $\dot{Q}_{dyss2} = \dot{Q}_2$ and $\dot{U}_2 = 0$.

Since the internal energy does not change in element 1, i.e., $\dot{U}_1 = 0$ and $\dot{U}_2 = \dot{U}$, the flux of dissipation heat $\dot{Q}_{dyss1}$ can be described as follows:

$$\dot{Q}_{dyss1} = \dot{Q}_{dyss} \frac{\Delta m_1}{\Delta m} = \dot{Q}_1. \tag{79}$$

Considering the dependence

$$\dot{Q}_2 = \dot{Q} - \dot{Q}_{dyss1}, \tag{80}$$

(76) can be converted into:

$$\dot{Q}_{dyss2} = (\dot{Q} - \dot{Q}_{dyss1})\left(2 - \frac{t_o}{t}\right). \tag{81}$$

Since $\dot{Q}_{dyss} = \dot{Q}_{dyss1} + \dot{Q}_{dyss2} = \frac{\Delta m_1}{\Delta m} Q_{dyss} + \frac{\Delta m_2}{\Delta m} Q_{dyss}$, the following dependence can be formulated:

$$\dot{Q}_{dyss}\left[\frac{\Delta m_2}{\Delta m} + \frac{\Delta m_1}{\Delta m}\left(2 - \frac{t_o}{t}\right)\right] = \dot{Q}\left(2 - \frac{t_o}{t}\right), \tag{82}$$

$$\dot{Q}_{dyss} = \frac{\dot{Q}\left(2 - \frac{t_o}{t}\right)}{1 + \frac{\Delta m_1}{\Delta m}\left(1 - \frac{t_o}{t}\right)}. \tag{83}$$

If there is no wear, then $\dot{A} = \dot{Q}_{dyss} = \dot{Q}\,(2 - t_o/t)$, that is:

$$\frac{t_o}{t} = 2 - \frac{\dot{A}}{\dot{Q}}, \tag{84}$$

where $\dot{A}$ and $\dot{Q}$ are measured with a calorimeter.

When the power of mechanical dissipation is derived from the balance: $\dot{A}_{dyss} = \dot{A} - \dot{Q}_{dyss}$ should be assumed to be a function of $\dot{Q}$ measured with a calorimeter.

Generalised and standardised wear

The problem of comparability and reproducibility of the assessments of wear and its intensity arises from several causes. The complexity of friction is the key, and it predetermines the great diversity of the test methods and measures of wear used. Own analysis has shown that only some major measures have unambiguous physical interpretations **[L. 8]**. The quantities of high practical importance are most commonly applied. Therefore, a systemic approach is recommended to address the tribological process's thermodynamic aspect. The laws of mass and energy conservation should be adopted as the foundations for correct wear assessments. This means both friction couple elements must be considered in parallel in experimentation. The specific work of wear (44), the energetic measure of resistance to wear that is identical to the tribological system and its elements (50) **[L. 8]**, is taken to have an unambiguous physical interpretation and be easy to determine. The fluxes or densities of mass or volume of a worn material are useful to the analytical descriptions of tribological wear.

If wear is compared based on the testing of mass and volume wastage in materials of different densities, different conclusions are arrived at. A lighter material will display greater volumetric wear than a heavier material given the same worn volume. This means such an assessment depends on a measure which is adopted. The generalised measure of wear introduced by this author allows for clearly finding which solid exhibits more wear if the results of mass and volume wastage testing do not provide such a finding **[L. 11]**.

The mass wear m, the volumetric wear V, and the work of friction $A_{t1\text{-}2}$, represented on appropriate scales in **Fig. 8**, make up a rectangular reference system. The relationship of mass to volume is the density of material ρ equal to the tangent of the angle γ:

$$\rho = \frac{m}{V} = \mathrm{tg}\,\gamma. \tag{85}$$

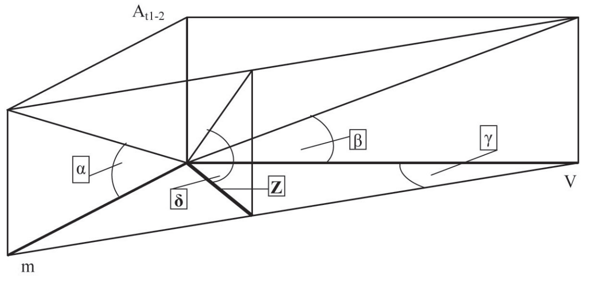

Fig. 8. The characteristic of tribological wear and resistance to wear in a rectangular system of coordinates $A_{t1\text{-}2}$ – V – m [L. 12–14]

Rys. 8. Charakterystyka zużycia tribologicznego i odporności na zużywanie w prostokątnym układzie współrzędnych $A_{t1\text{-}2}$ – V – m **[L. 12–14]**

α is shown in the plane $A_{t1\text{-}2}$ – m. Its tangent is the relationship of friction work to mass m or the specific work of wear e_R^x :

$$e_R^x = \frac{A_{t1\text{-}2}}{m} = \mathrm{tg}\alpha. \tag{86}$$

The angle β is present in the plane $A_{t1\text{-}2}$ – V. Its tangent is the relationship of friction work to the volume of worn material or the density of friction energy e_R^* :

$$e_R^* = \frac{A_{t1\text{-}2}}{V} = \mathrm{tg}\beta. \tag{87}$$

The proposed system of coordinates illustrates key relationships among friction ($A_{t1\text{-}2}$), wear (m, V), material density (ρ), and resistance to wear (e_R^x and e_R^*) synthetically and clearly. The height **Z** of a right triangle built on sides m and V is a generalised measure of wear between mass and volume. The relationship of friction work to **Z**, in the graphic scale adopted equal to the tangent of **δ**, is a new generalised energetic measure of resistance to wear e_R^z .

$$e_R^z = \frac{A_{t1\text{-}2}}{\mathbf{Z}} = \mathrm{tg}\boldsymbol{\delta}. \tag{88}$$

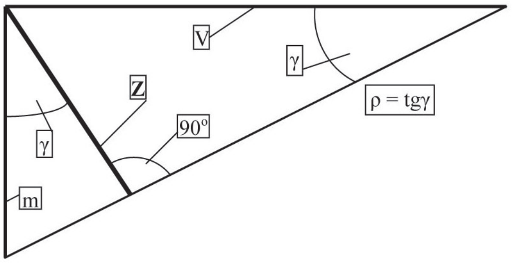

Fig. 9. Interdependences among mass m, volumetric V, and generalised wear Z [L. 12–14]

Rys. 9. Współzależności między zużyciem masowym m, objętościowym V i uogólnionym **Z [L. 12–14]**

Fig. 9 contains a geometrical interpretation of the interdependences among the mass, volumetric, and generalised wear. The generalised wear **Z**, expressed in terms of mass, is a direct result of the triangle resting on sides m and **Z**:

$$\mathbf{Z}_m = m\cdot\cos\gamma. \tag{89}$$

Z in terms of volume is derived from the triangle based on sides V and **Z**:

$$\mathbf{Z}_V = V\cdot\sin\gamma. \tag{90}$$

Since sin γ = ρ·cosγ, an equality for the value of wear is produced regardless of its initial assessment in terms of weight or volume.

$$\mathbf{Z}_m = \mathbf{Z}_V = \mathbf{Z}. \tag{91}$$

Thus, the introduction of **Z,** as shown in **Fig. 9**, allows for a detailed assessment of tribological wear. As a result, the measures of wear intensity, defined as the flux density **Z** and resistance to wear, defined as the reverse of **Z,** become unambiguous as well. On the other hand, resistance to wear expressed in energetic terms is the same in quantitative terms if the work of friction is referred both to the mass and the volume of the used material:

$$e_R^z = \frac{A_{t1\text{-}2}}{\mathbf{Z}} = \frac{A_{t1-2}}{m \cdot \cos\gamma} = \frac{A_{t1-2}}{V \cdot \sin\gamma}. \tag{92}$$

In order to standardise the wear measures, generalised unit wear **Z** = 1 will be applied, and the remaining unit measures will be derived from there.

Fig. 10 depicts a rectangular system of coordinates $A_{t1\text{-}2} - V - m$, applied when determining the standardised wear measures. Defining unit wear and its derivatives requires assuming that **Z** = 1 and the friction work $A_{t1\text{-}2} = 1$ [J]. **Table 1** collects the unit measures of wear and resistance to wear that are not derived from **Figs. 8** and **11**.

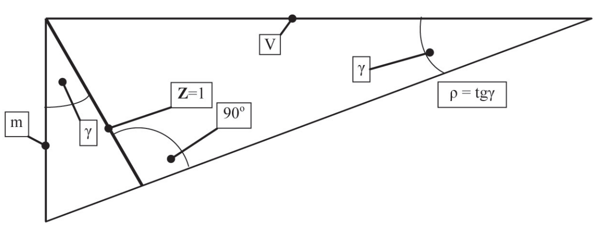

Fig. 10. Interdependences among the mass m, volumetric V and unit generalised wear Z = 1; material density ρ = tgγ [L. 13, 14]

Rys. 10. Współzależności między zużyciem masowym m, objętościowym V i jednostkowym uogólnionym **Z** = 1; gęstość materiału ρ = tgγ **[L. 13, 14]**

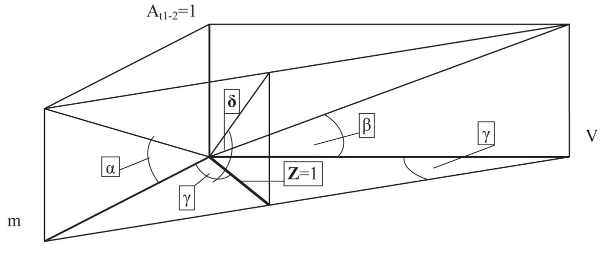

Fig. 11. The characteristic of tribological wear and resistance to wear in a rectangular system of coordinates $A_{t1\text{-}2}$ – V – m for Z = 1 and $A_{t1\text{-}2}$ = 1 [L. 13, 14]

Rys. 11. Charakterystyka zużycia tribologicznego i odporności na zużywanie w prostokątnym układzie współrzędnych $A_{t1\text{-}2} - V - m$ dla przypadku **Z** = 1 i $A_{t1\text{-}2} = 1$ **[L. 13, 14]**

Table 1. Unit generalised measures of wear and resistance to wear [L. 13, 14]

Tabela 1. Jednostkowe uogólnione miary zużycia i odporności na zużywanie **[L. 13, 14]**

Measure	Formula	Unit	Comments
Generalized wear	**Z** = 1	g, m^3	mcosγ = V·sinγ
Mass wear	m = 1/cosγ	g	m/V = tgγ
Volumetric wear	V = 1/sinγ	m^3	
Specific work of wear	e_R^x = cosγ	J/g	e_R^x = 1/m = tgα
Resistance to mass wear	1/m = cosγ	1/g	
Density of friction energy	e_R^* = sinγ	J/m^3	e_R^* = 1/V = tgβ; tgβ = tgγtgα
Resistance to volumetric wear	1/V = sinγ	1/m^3	

Table 2. Measures of wear based on Table 2; the unit measures are given in brackets [L. 13, 14]

Tabela 2. Miary zużycia ustalone według danych podanych w tabeli 2; miary jednostkowe podano w nawiasach **[L. 13, 14]**

Measure	Formula	Unit	Comments
Generalised wear	**Z**	g, m^3	m·cosγ = V·sinγ
Mass wear	m = [cosγ]·**Z**	g	m/V = tgγ
Volumetric wear	V = [1/sinγ]·**Z**	m^3	
Specific work of wear	$e_R^x = (A_{t1\text{-}2}/m)$ = [cosγ]·($A_{t1\text{-}2}$/**Z**)	J/g	e_R^x = (1/m)·A
Resistance to mass wear	1/m = [cosγ]·(1/**Z**)	1/g	
Density of friction energy	$e_R^* = (A_{t1\text{-}2}/V)$ = [sinγ]·($A_{t1\text{-}2}$/**Z**)	J/m^3	e_R^* = (1/V)·A
Resistance to volumetric wear	1/V = [sinγ]·(1/**Z**)	1/m^3	
Intensity of mass wear	m/t; m/l	g/s; g/m	m/V = tgγ
Intensity of volumetric wear	V/t; V/l	m^3/s; m^3/m	
Generalized intensity of wear	**Z**/t; **Z**/l	g/s, g/m, m^3/s, m^3/m	-

Since real wear is not equal to unit wear, the assumptions **Z** = 1 and $A_{t1\text{-}2}$ = 1 need to be abandoned, considering the real values of **Z** and $A_{t1\text{-}2}$. The method of calculating standardised measures is presented in **Table 2**.

CONCLUSIONS

The first part of this paper introduces a set of analytical dependences among the key components of the energy balance that characterise friction and its effects in the open thermodynamic system and its boundaries with the environment. The thermo-mechanical nature of friction is demonstrated. The heat of friction is a cause of thermal processes in the system, while dissipation of a mechanical nature, of disintegration of solids, or wear. The conditions of wear emergence and progress are established. The discussion is based on the first law of thermodynamics. The importance of a proper selection of the measures of tribological wear is emphasised. Interpreting the tribological process on the macroscopic hierarchical level of matter organisation is narrow and will be supplemented, in the second part of this study, with a description of impulse phenomena that prevail in the area of contacts of solid surface asperities.

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