ISSN 0208-7774 T R I B O L O G I A 4/2023 p. 33–41
DOI: 10.5604/01.3001.0054.3941

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ENERGETIC CONDITIONS OF MAXIMISING TEMPERATURE AND ENERGY DENSITY IN THE ENVIRONMENT OF FRICTION CONTACT OF SOLIDS

ENERGETYCZNE UWARUNKOWANIA MAKSYMALIZACJI TEMPERATURY I GĘSTOŚCI ENERGII W OTOCZENIU STYKU TARCIOWEGO CIAŁ STAŁYCH

Key words: specific heat, friction, wear, energy balance, development of results.

Abstract: The paper is intended to determine maximum temperatures in the friction zone of solids. An original model of thermal processes in a miniature test object – environment of a selected contact of asperities – is proposed. Its volume is limited to an area where energy dissipation and wear take place. The discussion is based on an energy balance including temperature variations within the object. An original method of establishing the maximum value of the so-called flash temperature on the basis of an experiment addressing the thermodynamic nature of friction is proposed. A method of determining average density of energy dissipated in a tested volume is specified as well. An analytical description of elementary friction includes physical properties of a material: density, specific heat, hardness; parameters characterising friction and wear, such as coefficient of friction, coefficient of wear, nominal unit pressure, specific work of wear, unit work of mechanical dissipation, temperature of friction surface, temperature of the immediate environment of surface asperities contact, mass of energy dissipation area, mass wear, and structure of the energy balance. The proposed description of friction within a contact of surface asperities encompasses analytical dependences that relate all the physical quantities accepted as characteristics of an object and the process inside it to one another. A quantitative evaluation of the maximum temperature and density of dissipated energy is undertaken for a selected instance of tribological testing.

Słowa kluczowe: ciepło właściwe, tarcie, zużycie, bilans energetyczny, opracowanie wyników.

Streszczenie: Praca poświęcona jest ustaleniu maksymalnych temperatur w strefie tarcia ciał stałych. Zaproponowano oryginalny model procesów cieplnych zachodzących w miniaturowym obiekcie badań – otoczeniu wybranego styku nierówności. Jego objętość ograniczono do obszaru przestrzennego, w którym zachodzi dyssypacja energii oraz zużywanie. Podstawą rozważań był bilans energii, uwzględniający zmiany temperatury w obrębie tego obiektu. Zaproponowano oryginalny sposób wyznaczania maksymalnej wartości temperatury, tak zwanej temperatury błysku, w oparciu o eksperyment uwzględniający termodynamiczną naturę tarcia. Ponadto ustalono metodę wyznaczania średniej gęstości energii rozpraszanej w badanej objętości. W opisie analitycznym elementarnego zjawiska tarcia uwzględniono własności fizyczne materiału: gęstość, ciepło właściwe, twardość; parametry charakteryzujące tarcie i zużycie, m.in.: współczynnik tarcia, współczynnik zużycia, nacisk jednostkowy nominalny, pracę właściwą zużycia, pracę jednostkową dyssypacji mechanicznej, temperaturę powierzchni tarcia, temperaturę bezpośredniego otoczenia styku nierówności powierzchni, masę obszaru dyssypacji energii, zużycie masowe i strukturę bilansu energetycznego. Zaproponowany opis zjawiska tarcia w obrębie styku nierówności powierzchni stanowią zależności analityczne wiążące ze sobą wszystkie wielkości fizyczne przyjęte jako cechy obiektu i zachodzącego w nim procesu. Ocenę ilościową maksymalnej temperatury i gęstości rozpraszanej energii przeprowadzono dla wybranego przykładu badań tribologicznych.

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INTRODUCTION

The discrete nature of friction contacts of solids restricts the area of mechanical energy dissipation to very low volumes of materials; in metals, friction zones reach depths of $10^{-4} - 10^{-6}$ m. Elementary instantaneous contacts of surface asperities are not immediately observable, which prevents examination of real thermal processes caused by friction that take place in close proximity to such contacts. Some information about the kinds of temperatures prevailing there can be gained indirectly from changes in the structure of the superficial layer material found at the end of tribological testing. This type of testing served to establish wear mechanisms **[L. 1**, **2]**. As part of experimentation, on the other hand, only maximum, approximate (because averaged) temperatures can be explored by means of natural thermocouples. This method of observation led Bowden and Ridler to discover and describe the so-called 'flash temperature' in 1936 **[L. 3]**. Bowden's continued testing with the aid of the photocell confirmed existence of such 'flashes' in the case of a friction couple where one element is transparent. Despite substantial progress of tribological research since the first observation of 'flash temperatures', the effects have not been examined in more depth. In the circumstances, an energetic analysis is reasonable of heating based on the balance of energy dissipated by a material directly involved in friction that would take into account energy spent on both wear and cooling. The application of a calorimeter to estimate maximum temperature increments in the process of dry friction of steel may be an example of this line of testing **[L. 4]**. Specialist literature also offers a number of attempts at analytical description of the temperature field characterising the environment of a friction contact, e.g. **[L. 5]**, **[L. 6]**. These analyses are commonly based on the equation of heat conduction. A boundary condition describing efficiency of a frictional source of heat as equal to power of friction is assumed when solving this equation, which means the energy spent on wear is ignored. Some initial attempts at including wear in energy balances are **[L. 7**, **8],** where discharge of friction heat is described with a third-type condition as heat transfer according to Newton's law. Flash temperature is central to this description. This approach suffers from a certain gap, namely, of ignoring mass heated up to the flash temperature.

This author has attempted to formulate a generalised balance of energy as an equation of the first law of thermodynamics for stationary open systems, where a mass of dissipation area reaching a maximum temperature is introduced. All components of the energy balance are described and given physical interpretations and tribological wear is taken into account as well **[L. 9**, **10]**. This paper is designed to develop a method of evaluating the maximum temperature of friction area of solids by expanding the model proposed in **[L. 11]**. The consideration of cooling and wear that accompany heating of friction microareas around momentary contacts of surface asperities is a novel element of this discussion, inspired by the results of theoretical and experimental analyses in **[L. 10]**. They demonstrate the need to clarify causes of flash temperatures lower than those derived from the model presented before **[L. 11]**. An attempt is also made to evaluate the average density of the energy dissipated in the environment of surface asperity contact with the highest temperature. The analysis of thermal processes is conducted for two cases. One addresses energy dissipation as part of the adiabatic process, the other – real friction and wear processes. A new way of establishing values of temperatures and energy densities in the immediate environment of contacts between surface asperities of solids in friction is developed in this paper. Quantitative aspects of the thermal process analysis are also presented on the basis of tribological testing, considering the thermodynamic nature of friction and its associated effects **[L. 12]**.

A MODIFIED MODEL OF A FRICTIONAL HEAT SOURCE

The modelling of the solids' friction process in this article begins with its underlying elementary effects. Contact of solids can be considered on two observation scales – as restricted to the environments of their surface asperity contacts or to the nominal friction surface, that is, at the macroscopic level of a material object. Energy dissipation processes and their associated effects are limited to very low volumes of material around real contacts of surface asperities. In order to answer the question concerning maximum densities of energy dissipated at the time of friction and maximum temperatures present in the circumstances, therefore, one must first of all consider momentary friction processes on the small observation scale.

Since duration of such a contact is very short, in the order of milliseconds, due to the low dimensions, of the order of micrometres, energy is dissipated in pulses and friction on this observation scale may be treated as an impact. Of course, momentary thermal processes and wear are non-stationary. It is only on averaging energetic effects of a great number of elementary effects of energy dissipation that stationary or non-stationary heating and wear of real objects in friction, e.g. machine parts, can be observed. Further discussion will be limited to stationary processes. It is assumed, therefore, that a friction couple has been ground in, external conditions and forces, that is, friction parameters and properties of the couple materials, do not change over time. This restrictive assumption is adopted to arrive at an unambiguous physical interpretation of the quantities describing the effects under analysis and to allow the use of experimental results needed to exemplify the model of microscopic phenomena. The employment of parameters characterising the friction and wear of a macroscopic object to establish the maximum temperature of a friction zone relies on two universal principles, namely, energy and mass conservation. Both laws enable derivation of analytical dependences between friction and wear processes at the two diverse levels of matter mentioned above.

The following assumptions are made to characterise the object of discussion and energy transformations inside it:

– Energy is dissipated in a volume V_{oi} around the area of a single contact, A_{ri}.
– Real surface of solid contact, A_r, is the sum total of individual areas, A_{ri}, within the zone of nominal contact A_n, that is: $A_r = \sum A_{ri}$.
– Momentary volume of the energy dissipation area at the time of real solids' friction is the sum total: $V_o = \sum V_{oi}$.
– Density of the energy dissipated in V_{oi} is e_t.
– Elementary work of friction A_{ti}, i.e. the energy dissipated within V_{oi}, is the product of its volume times average energy density, e_t: $A_{ti} = V_{oi}e_t$.
– Temperature of the material directly adjacent to the area considered as part of the proposed model equals Θ.
– Θ is also equal to the temperature of nominal surface of solids contact, A_n.
– Physical properties of the material of a friction couple element included in V_{oi}: density ρ, specific heat c_p and hardness H, are constant.
– Mass m_{oi} of V_{oi} is constant and expressed as $m_{oi} = \rho V_{oi}$.
– In the process of energy dissipation, the temperature of m_{oi} grows from Θ to Θ_o.
– After the contact between surface asperities is broken, the temperature of m_{oi} reverts to the original value of Θ as Q_i of the heat has been conducted.
– In the peculiar case of adiabatic process of frictional heating of the area under discussion, where $Q_i = 0$, its temperature reaches a theoretical maximum of $\boldsymbol{\Theta_o}$.
– If friction is associated with wear, mass $m_i = V_i\rho$ at Θ and with c_p is supplied to V_{oi} while mass $m_i = V_i\rho$ at Θ_o and with specific heat of c_p' leaves the area at the same time.

Two cases of temperature growth caused by frictional heating are then analysed. **Figure 1a** is a schematic illustration of heating in the adiabatic process, that is, where heat released into a friction couple element equals zero. The balance of energy expressing elementary heat of dissipation Q_{dyssoi} is simplified in this case to:

$$Q_{dyssoi} = A_{ti} = e_t V_{oi} = m_{oi} c_p(\boldsymbol{\Theta_o} - \Theta). \quad (1)$$

Maximum temperature $\boldsymbol{\Theta_o}$ results from the dependence:

$$\boldsymbol{\Theta_o} = \Theta + \frac{e_t}{\rho c_p} \quad (2)$$

[L. 11] proceeded along this path to assume energy density e_t is equal to maximum contact stress possible in a contact of bodies, that is, the product of friction coefficient μ times material hardness:

$$\Theta_{omax} = \Theta + \frac{\mu H}{\rho c_p} \quad (3)$$

Figure 1b shows the instance of frictional heating associated with wear and parallel release of heat Q_i from V_{oi} into a solid involved in friction. A cooled material inside this volume reaches Θ_o at the time of energy dissipation, a temperature lower than in the process of adiabatic heating. Thus, the following inequality applies: $\boldsymbol{\Theta_o} > \Theta_o$. Wear is designated as supply of a certain volume, V_i, at Θ, to the area analysed and release of the same quantity of material to the environment of a friction couple in parallel. A separated wear particle is assumed to exhibit Θ_o and c_p'. Therefore, the balance of energy expressing the elementary heat of dissipation in Q_{dyssi} conditions of wear for

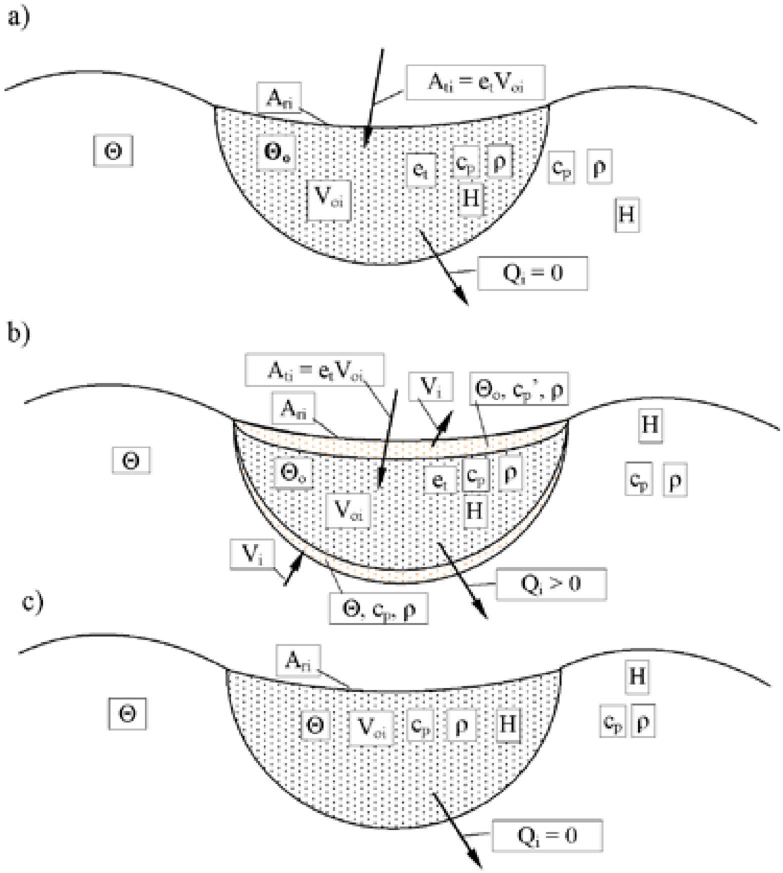

Fig. 1. Elementary volume of energy dissipation V_{oi} in the environment of a real area of surface asperity contact A_{ri} as a frictional source of heat. a) –adiabatic heating of an elementary volume, b) – heating of this volume combined with parallel wear and heat release into the material of a solid in friction. c) – energetic condition of V_{oi} at the end of heat exchange

Rys. 1. Elementarna objętość dyssypacji energii V_{oi} w otoczeniu rzeczywistej powierzchni styku nierówności powierzchni A_{ri} jako tarciowe źródło ciepła. a) – przypadek nagrzewania adiabatycznego objętości elementarnej, b) – przypadek nagrzewania tej objętości z jednoczesnym zużywaniem i oddawaniem ciepła w głąb materiału ciała trącego się. c) – stan energetyczny objętości V_{oi} po zakończeniu wymiany ciepła

the case of elementary friction under discussion will build on the dependence (1):

$$Q_{dyssi} = (1-\eta)A_{ti} = (m_{oi}c_p + m_i c_p')(\Theta_o - \Theta), \quad (4)$$

where η is the portion of dissipated energy not converted into frictional heat yet causing material dispersion.

(4) details two components of dissipation heat (1-η) A_{ti}. One causes heating of mass m_{oi}

$$Q_{oi} = m_{oi}c_p(\Theta_o - \Theta), \quad (5)$$

while the other causes heating of wear products' mass m_i

$$Q_i' = m_i c_p'(\Theta_o - \Theta). \quad (6)$$

In the absence of wear, $m_i = 0$, $Q_i' = 0$ and

$$A_{ti} = Q_{oi} = m_{oi}c_p(\Theta_o - \Theta_x), \quad (7)$$

and Θ_o and Θ_x are reached where heat is discharged from the area of energy dissipation. Θ_x fulfils the condition:

$$\Theta_x = \Theta_o - \frac{A_{ti}}{m_{oi}c_p} \geq 0, \quad (8)$$

Hence, the inequality:

$$m_{oi}c_p\Theta_o \geq A_t \quad (9)$$

and the zero value of Θ_x:

$$\Theta_x = 0 \quad (10)$$

as the quantity of the dissipation heat generated cannot be greater than the mechanical energy dissipated in the elementary area of friction **[L. 9]**, **[L. 10]**. Its mass m_{oi} can, thus, be determined considering the analytical conditions (9) and (10):

$$m_{oi} = \frac{A_{ti}}{c_p\Theta_o} = \frac{e_t V_{oi}}{c_p\Theta_o}. \quad (11)$$

Therefore, Q_{oi} can be expressed in a new form on the basis of (5) and (11):

$$Q_{oi} = A_{ti}\left(1 - \frac{\Theta}{\Theta_o}\right) \quad (12)$$

Upon introducing (12), the formula of thermal balance (4) can be represented as:

$$(1\text{-}\eta)A_{ti} = A_t\left(1-\frac{\Theta}{\Theta_o}\right) + m_i c_p'(\Theta_o - \Theta), \quad (13)$$

(4) also implies an interesting regularity, namely, its conversion into

$$A_{ti}(1\text{-}\eta) - m_{oi}c_p(\Theta_o - \Theta) = m_i c_p'(\Theta_o - \Theta) \quad (14)$$

the heat released by conduction from V_{oi} into a solid involved in friction can be said – according to the left side of (14) – to equal enthalpy, that is, energy associated with mass m_i of the wear products.

METHOD FOR DETERMINING MAXIMUM TEMPERATURE AND ENERGY DENSITY E_T IN THE DISSIPATION AREA

The description of thermal processes characterising dissipation of mechanical energy in the environment of a momentary contact of surface asperities introduced in the preceding section is the starting point for further discussion intended to develop an experimental method for establishing the maximum temperature in the dissipation area.
Simplification of (13) to

$$A_{ti}\frac{\Theta}{\Theta_o} = A_{ti}\eta + m_i c_p'(\Theta_o - \Theta) \quad (15)$$

and introducing a definition of the specific work of wear, e_R^x, as the quotient of work of friction divided by the mass of the worn material:

$$e_R^x = \frac{\eta A_t}{m_i} \quad (16)$$

produces the following relation:

$$e_R^x\frac{\Theta}{\Theta_o} = a_{dyss} + c_p'(\Theta_o - \Theta) \quad (17)$$

where a_{dyss} is the specific work of mechanical dissipation:

$$a_{dyss} = \frac{\eta A_{ti}}{m_i} = \eta\, e_R^x \quad (18)$$

Another modification to (17) consists in the introduction of a system quantity, D, defined as the temperature above flash temperature Θ_o arising from dissipation of a_{dyss} in volume V_{oi} **[L. 9, 10]**:

$$D = \frac{a_{dyss}}{c_p'} + \Theta_o \quad (19)$$

Therefore,

$$e_R^x\frac{\Theta}{\Theta_o} = c_p'\left(\frac{a_{dyss}}{c_p'} + \Theta_0 - \Theta\right) = c_p'(D - \Theta) \quad (20)$$

and, considering m_{oi} as described by (11), and the definition of e_R^x according to (18), the following analytical relations can be formulated:

$$D - \Theta = \frac{A_{ti}\Theta}{m_i c_p'\Theta_o} = \frac{m_{oi}c_p\Theta}{m_i c_p'} \quad (21)$$

$$m_{oi}c_p = m_i c_p'(D - \Theta)\frac{1}{\Theta} \quad (22)$$

which can then serve to arrive at new formulations of the thermal balance:

$$A_{ti}(1-\eta) = m_i c_p'(D-\Theta)\frac{1}{\Theta}(\Theta_o - \Theta) + m_i c_p'(\Theta_o - \Theta) \quad (23)$$

$$A_{ti}(1\text{-}\eta) = m_i c_p'(\Theta_o - \Theta)\frac{D}{\Theta} \quad (24)$$

$$e_R^x(1\text{-}\eta)\frac{\Theta}{D} = c_p'(\Theta_o - \Theta) \quad (25)$$

The above dependence produces temperature D

$$D = \frac{e_R^x(1-\eta)\Theta}{c_p'(\Theta_o - \Theta)} \quad (26)$$

which can be reformulated as follows upon the introduction of (20):

$$D = \frac{c_p'(D-\Theta)(1-\eta)\Theta_o\Theta}{c_p'(\Theta_o - \Theta)\Theta} = \frac{(D-\Theta)(1-\eta)\Theta_o}{\Theta_o - \Theta} =$$

$$= \frac{(1-\eta)\Theta\Theta_o}{(1-\eta)\Theta_o - \Theta_o + \Theta} \quad (27)$$

Parameter η as a function of Θ, Θ_o and D

$$\eta = \frac{(D-\Theta_o)\Theta}{\Theta_o(D-\Theta)} \quad (28)$$

fulfils the initial assumptions adopted as part of the thermal process model (4), which means that for: $\eta = 0$; $\Theta = 0$ and for $\eta = 1$; $\Theta = \Theta_o$. Expressing the D/Θ_o relation, assumed to be constant in the range of temperatures $\Theta_1 \div \Theta_2$, with a system of dependences:

$$\frac{D}{\Theta_o} = \text{const} = \frac{(1-\eta)\Theta}{\Theta - \Theta_o \eta} = \frac{(1-\eta_1)\Theta_1}{\Theta_1 - \Theta_o \eta_1} = \frac{(1-\eta_2)\Theta_2}{\Theta_2 - \Theta_o \eta_2} \qquad (29)$$

provides the foundation for an experimental method of determining Θ_o. The dependence describing this temperature is the following solution to (29):

$$\Theta_o = \frac{(\eta_1 - \eta_2)\Theta_1\Theta_2}{\eta_1\Theta_2(1-\eta_2) - \eta_2\Theta_1(1-\eta_1)} \qquad (30)$$

Θ_1 and Θ_2 can be measured in a stationary friction process, e.g. by means of a thermocouple at a minimum distance from a frictional contact. On the other hand, η_1 and η_2 can be established on the basis of **[L. 13]**:

$$\eta_1 = k_1 \frac{H}{p} \qquad (31)$$

$$\eta_2 = k_2 \frac{H}{p} \qquad (32)$$

where: k_1, k_2 – coefficients of wear after J.F. Archard, corresponding to Θ_1 and Θ_2, whereas H – material hardness, p – unit pressure against nominal surface of friction.

The coefficients of wear can be described with the following dependences **[L. 13]**:

$$k_1 = \frac{\mu V_1 H \rho}{\mu N l \rho} = \frac{\mu m_1 H}{A_{t1-2}\rho} = \frac{\mu H}{e^x_{R1}\rho} \qquad (33)$$

$$k_2 = \frac{\mu V_2 H \rho}{\mu N l \rho} = \frac{\mu m_2 H}{A_{t1-2}\rho} = \frac{\mu H}{e^x_{R2}\rho} \qquad (34)$$

that include magnitudes measurable in experimental conditions. These comprise: μ – coefficient of friction, V – volumetric wear, m – mass wear, N – normal pressure against nominal surface, ρ – material density, A_{t1-2} – work of friction, and l – path of friction.

Knowledge of Θ_o allows for determining the density of energy e_t dissipated in an elementary volume V_{oi}. It is described in accordance with (11)

$$e_t = \rho c_p \Theta_o \qquad (35)$$

and, on the basis of (2):

$$\boldsymbol{\Theta}_o = \Theta + \Theta_o \qquad (36)$$

The above relation helps to represent the thermal balance (4) in yet another form:

$$(m_{oi}c_p + m_i c_p')(\boldsymbol{\Theta}_o - \Theta_o) = (m_{oi}c_p + m_i c_p')(\Theta - 0). \qquad (37)$$

This means the difference between the energy accumulated in mass $m_{oi} + m_i$ heated in the adiabatic process and by real friction and wear is equal to the absolute energy accumulated in the same mass on breach of the asperity contact and reaching temperature Θ.

QUANTITATIVE EVALUATION OF MAXIMUM TEMPERATURE AND ENERGY DENSITY IN THE ENVIRONMENT OF FRICTIONAL CONTACTS OF SELECTED METALS

Maximum temperature and energy density in the environment of a frictional contact can be evaluated through tests whose results carry some information on magnitudes in the model of process of energy dissipation by friction as developed above. Temperature Θ of nominal surface A_n is an important independent thermodynamic parameter. Unfortunately, it commonly results from testing and is not always explored properly, making such research of no use to the present discussion. Θ can be determined as part of tribological testing by means of a specially designed heat exchanger and measured with a thermocouple at the same time. This procedure was adopted by **[L. 12]**, therefore, temperature was treated as an independent parameter of the friction process. The further analysis relies on the test results in **[L. 12]** to supplement the description of the friction process with the information about the maximum temperature and energy density that characterise spaces around momentary contacts of surface asperities.

A pin-on-disc friction couple is tested. A 1.5 mm thick 145Cr6 ring with an external diameter of 121 mm and an internal diameter of 104 mm was the larger element. The hardness of 145Cr6 steel was 6970 MPa. A smaller element – a 5 × 5 × 0.5 mm metal pin (nominal friction surface:

5 × 5 mm²) mounted in a dedicated copper hold – worked together with the ring. Eight elements of this type were employed, made of Armco iron (ferrite), C45 steel (ferrite + pearlite), C80U steel (pearlite), copper, aluminium, zinc, lead, and LC60 (Sn+Pb) alloy. The test stand enabled setting of desired Θ with a heat exchanger positioned 0.4–0.5 mm away from the friction surface, on the pin side. An iron – constantan thermocouple was used to test the temperature. The relatively small nominal contact surface of the solids in friction, A_n, allows for a practical evaluation of the average Θ of the surface. The mass wear of the pin on grinding-in of the frictional couple was measured with analytical scales. **Table 1** provides values of the wear coefficient, characterises some physical properties of the test specimen pin materials: density ρ, unit pressure p against the nominal surface A_n, hardness H and specific heat c_p of the specimen material, two temperatures, Θ_1 Θ_2 and their corresponding specific work of wear, e^x_{R1} and e^x_{R2}. The hardness of the test materials, H, was far lower than of the ring material, therefore, the latter's wear was ignored in this analysis. **Table 2** contains coefficients of wear k_1, k_2 (33, 34), parameters η_1, η_2 (31, 32), maximum temperatures Θ_o(30), $\mathbf{\Theta_{o1}}$ $\mathbf{\Theta_{o2}}$(36), the system quantity – temperature D (27), density of energy dissipated within the volume e_t (35), and unit work of mechanical dissipation a_{dyss} (18) established on the basis of figures from **Table 1**.

Table 1. Specific work of wear, e^x_{R1} and e^x_{R2}, determined at Θ_1 and Θ_2, unit pressure p, coefficient of wear μ, and velocity v = 1m/s for dry friction of eight different metal specimens with H and specific heat c_p in accordance with the testing discussed in [L. 12], as well as temperatures Θ_{omax1} and Θ_{omax2}, corresponding to Θ_1 and Θ_2

Tabela 1. Praca właściwa zużycia e^x_{R1} i e^x_{R2} wyznaczona przy temperaturach Θ_1 i Θ_2, naciskach jednostkowych p, współczynniku tarcia μ i prędkości v = 1m/s dla przypadków tarcia suchego ośmiu różnych próbek metali o twardości H i cieple właściwym c_p według badań opisanych w publikacji [L. 12] oraz temperatury $\mathbf{\Theta_{omax1}}$ **i** $\mathbf{\Theta_{omax2}}$ odpowiadające temperaturom Θ_1 i Θ_2

Material	μ	ρ kg/m³	H MPa	p MPa	c_p kJ/kgK	Θ_1 K	e^x_{R1} MJ/g	Θ_2 K	e^x_{R2} MJ/g	Θ_{omax1} K	Θ_{omax2} K
Fe	0.6	7,860	1,746.18	0.785	0.452	298	26.76	333	3.38	592.90	627.90
C45	0.6	7,860	2,158.2	1.177	0.452	298	62.41	333	9.08	662.49	697.49
C80U	0.6	7,860	2,687.94	1.177	0.452	298	6.74	333	1.24	751.95	786.95
Cu	0.51	9,830	1,236.06	0.392	0.383	293	14.51	333	6.61	460.44	500.44
Al.	0.43	2,700	794.61	0.392	0.896	293	19.26	333	8.44	434.24	474.24
Zn	0.5	7,130	431.64	0.392	0.385	293	12.19	333	4.03	371.62	411.62
Pb	0.8	11,340	58.86	0.020	0.129	293	0.79	313	0.43	325.19	345.19
LC60	0.5	8,500	78.48	0.078	0.129	293	2.39	313	0.94	328.79	348.79

Table 2. Coefficients of wear k_1 and k_2, parameters η_1 and η_2, temperatures: Θ_o, Θ_{o1}, Θ_{o2} and D, energy density e_t and unit work of mechanical dissipation a_{dyss}

Tabela 2. Wartości współczynników zużywania k_1 i k_2, parametrów η_1 i η_2, temperatur: Θ_o, $\mathbf{\Theta_{o1}}$, $\mathbf{\Theta_{o2}}$ i D, gęstości energii e_t i pracy jednostkowej dyssypacji mechanicznej a_{dyss}

Material	k_1	η_1	k_2	η_2	Θ_o K	Θ_{o1} K	Θ_{o2} K	D K	e_t MJ/m³, MPa	a_{dyss} MJ/g
Fe	4.98E-06	0.011	3.94E-05	0.088	338.24	636.24	671.24	338.75	1,201.67	0.297
C45	2.64E-06	0.005	1.81E-05	0.033	339.56	637.56	672.56	339.79	1,206.38	0.302
N8E	3.04E-05	0.070	1.65E-04	0.378	338.58	636.58	671.58	342.06	1,202.87	0.469
Cu	4.42E-06	0.014	9.70E-06	0.031	374.47	667.47	707.47	375.94	1,409.82	0.202
Al	6.57E-06	0.013	1.50E-05	0.030	371.34	664.34	704.34	372.69	898.35	0.257
Zn	2.48E-06	0.003	7.51E-06	0.008	356.86	649.86	689.86	357.08	979.60	0.033
Pb	5.26E-06	0.015	9.66E-06	0.028	339.93	632.93	652.93	340.78	497.27	0.012
LC60	1.93E-06	0.002	4.91E-06	0.005	327.42	620.42	640.42	327.49	359.01	0.005

RESULTS AND DISCUSSION

The example of experimental testing **[L. 12]** applied to relatively mild conditions of friction between eight metal arrangements. The range of Θ_o found: 327.37–374.50K, given the set range of Θ: 293–333K, means the flash temperature is but several to several dozen degrees greater than Θ_2 in each case of the materials studied. The temperatures established for processes of adiabatic heating, $\mathbf{\Theta_{o1}}$, $\mathbf{\Theta_{o2}}$, are relatively high, qualitatively comparable with $\mathbf{\Theta}_{omax1}$ and $\mathbf{\Theta}_{omax2}$, derived from (3) on the assumption the density of energy $e_t = \mu H$. Temperature D, on the other hand, is very close to Θ_o. Energy densities e_t established for real conditions of friction and wear become comparable to densities expressed by μH in the case of iron and steel and considerably greater than the densities expressed quantitatively by the contact stress μH in the remaining cases. Since unit pressures in the event of friction between relatively soft metals were assumed to be far lower than for steel and iron for the purposes of experimentation, V_o, was lower and, as a consequence, higher energy densities e_t were possible.

CONCLUSIONS

The issue of temperature growth at contact of solids in friction, particularly metals, is important for both cognitive and utilitarian reasons. Temperature substantially conditions physical properties of machine part materials, particularly in areas of direct dissipation of mechanical energy caused by friction. Although the role of temperature is noted and appreciated, especially since Bowden and Ridler discovered the so-called ‘flash temperatures,’ maximum instantaneous temperatures of contacts of solids’ surface asperities have not been conclusively determined. A range of experiments and calculation models following on Bowden’s and Ridler’s experiment are but attempts at approximate solutions to the problem.

This paper refers to frictional heating of momentary contacts of surface asperities. An original model of thermal process in a miniature test object – environment of a selected contact between asperities – is proposed. Its volume is limited to an area where energy dissipation and wear take place. The discussion is based on a balance of energy considering temperature variations within the object. Its increments are maximum in the presence of an adiabatic partition on the boundary of the energy dissipation zone. Under actual conditions established by tribological testing, lower flash temperatures are found. Impact of wear on thermal processes, represented as parameter η characterising the structure of the energy balance, is another major part of this discussion.

The balance of energy equation (4) and its modification (13) define a dependence between a quantity of energy dissipated, mass of the dissipation area, mass of worn material, and temperatures of nominal friction surface and surface of a momentary contact between surface asperities of solids in friction. These balances, appropriately developed and with the addition of the system quantities (a_{dyss}, e_R^x, D), provided a starting point for an analytical description of temperature Θ_o (30), treated as another system quantity, constant within the range of experimentally established temperatures, $\Theta_1 \div \Theta_2$, as part of the model submitted here. The discussion also gave rise to a description of energy density e_t dissipated in the environment of a frictional contact and conditioning the contact’s maximum temperature. A known Θ_o enables determination of temperatures Θ_o which characterise the process of adiabatic heating – equation (36).

NOMENCLATURE

a_{dyss}	– specific work of mechanical dissipation (J/g)
A_n	– nominal friction surface (m^2)
A_r	– real surface of solids’ contact (m^2)
A_{ri}	– elementary contact surface of bodies in friction (m^2)
A_{ti}	– elementary work of friction (J)
e_t	– density of energy in dissipation zone (J/m^3)
c_p	– specific heat of specimen material (J/g K)
c_p'	– specific heat of wear product material (J/g K)
D	– tribological system constant (K)
e_R^x	– specific work of wear (J/g)
H	– material hardness (Pa)
k	– wear coefficient according to J. F. Archard
l	– riction distance (m)
m_{oi}	– mass of elementary friction area (g)
m_i	– elementary mass wear (g)
N	– normal force against the nominal surface (N)
p	– unit pressure (Pa)
V_{oi}	– elementary volume of dissipation zone (m^3)
V_o	– volume of dissipation zone (m^3)
V	– volumetric wear (m^3)

Q_i – heat released from volume V_{oi} (J)
Q_{oi} – heat heating mass m_{oi} (J)
Q_i' – heat heating mass m_i (J)
Q_{dyssi} – elementary heat of dissipation discharged in conditions of wear (J)
Q_{dysso} – elementary heat of dissipation discharged when wear is absent (J)
η – quotient of mechanical dissipation work and friction work
μ – friction coefficient
ρ – material density (kg/m^3)
Θ – material temperature out of dissipation zone (K)
Θ_o – material temperature in dissipation zone (K)
$\boldsymbol{\Theta}_{\mathbf{o}}$ – material temperature in dissipation zone generated by adiabatic process (K)
$\boldsymbol{\Theta}_{\mathbf{oma}}$ – maximum material temperature in dissipation zone generated by adiabatic process (K)

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