ISSN 1068-3666, Journal of Friction and Wear, 2016, Vol. 37, No. 6, pp. 523–528. © Allerton Press, Inc., 2016.
Original Russian Text © Yu.Yu. Osenin, I.I. Sosnov, O.V. Sergienko, A.V. Chesnokov, Yu.I. Osenin, Douma Mansur Al-Makhdi, 2016, published in Trenie i Iznos, 2016, Vol. 37, No. 6, pp. 677–684.

Increase in the Coefficient of Friction of the Rolling Stock Disc Brake via Fluid Cooling of its Friction Elements

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Received April 13, 2016

Abstract—The paper presents the results of theoretical and experimental investigations in the effect of a partial heat energy removal in the friction contact zone upon the disc brake coefficient of friction during braking. A fluid cooling system for the disc brake friction elements has been proposed based on a recuperative liquid-pneumatic heat-exchange apparatus.

Keywords: disc brake, coefficient of friction, fluid cooling, heat exchange, recuperative cooling system

DOI: 10.3103/S106836661606012X

INTRODUCTION

Growing velocities of the railway transport have complicated the problem of rolling stock braking due to the augmented generation of kinetic energy during deceleration, which raises the temperature of brake-system elements. This experience, along with investigations on the rolling stock, have shown that, all other conditions being equal, the temperature rise in friction elements is a reason for the reduced coefficient of friction of the brakes, which in turn lowers the braking efficiency of the rolling stock as a whole.

The problem of stabilizing the coefficient of friction in the high-temperature conditions is solved mainly by the elaboration of the novel materials, the friction properties of which are designed so as to level the temperature effect. Furthermore, it is natural to affect the coefficient of friction under a high temperature operation by cooling the working surfaces of the disc brake in order to stabilize the coefficient of friction.

The cooling of the friction elements aimed at raising their coefficient of friction is commonly used in modern disc brake designs. The designs with the cooled brake discs have special channels provided to force the atmospheric air to circulate during the motion of the rolling stock. This method helps to reject about 10% of the heat from the friction area that is generated when the train stops [1, 2].

This problem is solved most efficiently by the fluid cooling of the friction elements of disc brakes, which is restricted by rather high working temperatures on the elements of the disc brake, the complexity of removing heat from the body of the revolution (brake disc), and fire-hazardous heat carriers employed today.

The aim of the work was to increase the coefficient of friction of the disc brakes under high-temperature operating conditions using the fluid cooling of the disc-brake elements of the rolling stock.

INVESTIGATION METHODS

To achieve the goal we took the following steps:

—the mathematical simulation of the heat-exchange processes in the friction elements of the disc brakes in correspondence with the environment taking into account the heat conductance, as well as the convective and radiant heat exchange;

—experimental investigations of the processes of friction interaction between elements of the disc brakes with the goal of establishing the effect of the fluid cooling on the coefficient of friction and the mean integral temperature of the interacting working surfaces;

—the development of the fluid cooling systems for the disc brake friction elements.

Under these conditions, the friction processes in the contact region of the disc brake elements were considered to be the heat flow sources.

The main assumptions accepted in the work are as follows:

—the materials used for the elements of the cooling system are isotropic;

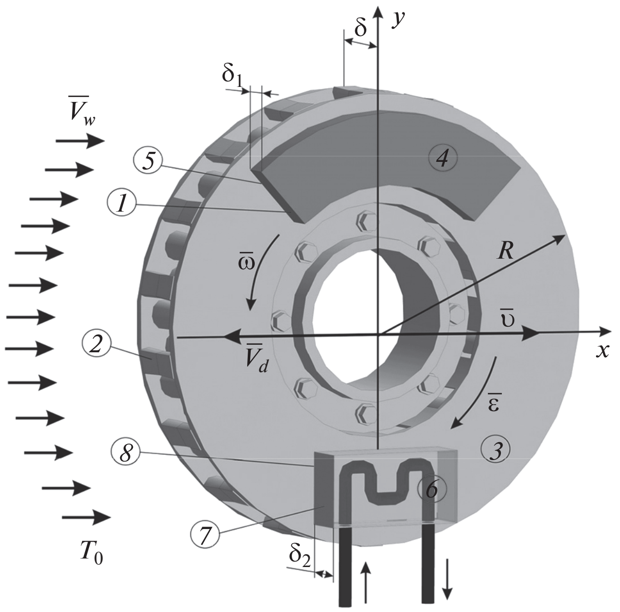

Fig. 1. Design diagram of the disc brake friction joint. (*1*) side surfaces of pads; (*2*) disc rim surface; (*3*) disc side surfaces; (*4*) external surfaces of pads; (*5*) inner surfaces of pads; (*6*) external surfaces of heat pickup elements; (*7*) side surfaces of heat pickup elements; (*8*) surfaces of heat pickup elements contacting brake discs.

—the temperature of the brake disc with the pads at the initial moment of braking is similar and equals that of the ambient air;

—the braking process is running at constant deceleration without slippage of the wheel pairs against the railheads;

—the brake power is uniformly distributed between all pads participating in braking;

—the external surface roughness does not affect heat emission (but effects the contact heat exchange). The mean equivalent asperity height does not exceed the dynamic boundary layer thickness formed longitudinally by the air flows over the surface;

—all other conditions being equal, the heat emission values of the brake disc, pads, and the heat pickup elements are fully determined by the velocity of the incoming air flow (depending on the current speed and the cross wind velocity), as well as rotation frequency for the brake disc;

—the processes of the convective and radiant heat exchange are not affected by the bodies located round the brake disc, pads, and heat pickup elements;

—the actual contact area of the conjugated surfaces equals an insignificant portion of the nominal one and is formed of contact spots that are the roughly similar in size and evenly scattered over the contact surface;

—the heat models of the rough surfaces used to obtain the integral dependencies look like cylinders with spherical contacting surfaces (each surface has a single contact spot).

Mathematical Simulation

The base of the mathematical model that describes the heat exchange constitutes the differential Fourier–Kirchhoff heat conductivity equation in Cartesian coordinates without internal heat sources that contain a convection-induced summand (a substantive derivative that accounts for variations in the convective temperature) as follows [3, 4]:

$$\rho c_{pm}\frac{\partial T}{\partial t}=\left[\frac{\partial}{\partial x}\left(\lambda\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\lambda\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(\lambda\frac{\partial T}{\partial z}\right)\right] - \rho c_{pm}\left(v_x\frac{\partial T}{\partial x}+v_y\frac{\partial T}{\partial y}+v_z\frac{\partial T}{\partial z}\right). \quad (1)$$

The design scheme is shown in Fig. 1. The orthogonal system of coordinates x, y, z is movable and attached to the brake disc whose axis is the applicate one. The dimensions and form of the surfaces in question can be in general different and vary depending on the structure.

The brake disc with the pads and heat pickup elements are washed during motion by the atmospheric air (its temperature being much lower than that of the disc and pads excluding the initial moment preceding braking when they are equal according to our assumptions). The brake disc moves forward at a speed V_d (for the disc it is a current velocity of the rolling stock) and rotates at an angular velocity ω.

For surfaces *1*–*4*, *6*, *7* (see Fig. 1) we have used the third-order boundary conditions (a combination of convective and radiant heat exchange without internal heat sources) of type (2). For surfaces *5* and *8* (sliding contact regions of the brake disc with a pad and a heat pickup element, correspondingly) the fourth-order boundary conditions are used with a surface heat source (3) as follows:

$$\lambda_m\left(\frac{\partial T}{\partial n}\right)_s=\alpha\left(T_s-T_0\right)+\varepsilon\sigma\left(T_s^4-T_0^4\right), \quad (2)$$

$$\lambda_{m1}\left(\frac{\partial T_1}{\partial n}\right)_s \pm q=\lambda_{m2}\left(\frac{\partial T_2}{\partial n}\right)_s. \quad (3)$$

Here and below, index "s" indicates external surface, index "1" indicates the brake disc, and index "2" indicates the pad or the heat pickup element. The use of the incoming velocity on the brake-disc air flow (taking into account the counter wind velocity and disc rotation) as a of characteristic heat transfer during braking makes possible to consider it to supersede a given stage of convective heat exchange, namely, forced convection, the combined action of natural and forced convections, and natural convection. As a result, we have the following criterion equations [2, 5]:

$$\left.\begin{array}{l} V_d = 0\text{–}15 \text{ km/h}: \\ \mathrm{Nu} = 0.18[(0.5\mathrm{Re}_\omega^2 + \mathrm{Gr})\mathrm{Pr}]^{0.315}, \\ V_d = 10\text{–}35 \text{ km/h}: \\ \mathrm{Nu} = 0.4(\mathrm{Re}_\omega^2 + \mathrm{Gr})^{0.25}, \\ V_d = 30\text{–}57 \text{ km/h}: \\ \mathrm{Nu} = 0.135[(0.5\mathrm{Re}_\omega^2 + \mathrm{Re}_a^2 + \mathrm{Gr})\mathrm{Pr}]^{0.33}, \\ V_d = 42\text{–}125 \text{ km/h}: \\ \mathrm{Nu} = 0.037(\mathrm{Re}_a^{0.8} + \mathrm{Re}_\omega^{0.4})\mathrm{Pr}^{0.33}. \end{array}\right\} \quad (4)$$

The specific heat flow generated by a unit surface S on the contact region in time t incorporated in boundary condition (3) is

$$q(r,t) = \frac{mR_\kappa^2\varepsilon}{nr_0S} r(\omega_0 - |\varepsilon|t). \quad (5)$$

To simulate the operation of the recuperative heat-exchange apparatus that has a cross scheme of the motion of heat carriers, the following relations can be used [6]:

$$\left.\begin{array}{c} Q = \int_0^F k_i \Delta t_i dF_i; \quad Q = Q_1 = Q_2 + \Delta Q; \\ Q_1 = W_1(t_1^{'} - t_1^{''}); \quad Q_2 = W_2(t_2^{''} - t_2^{'}); \\ \delta t_1 = t_1^{'} - t_1^{''} = (t_1^{'} - t_2^{'})Z; \\ \delta t_2 = t_2^{''} - t_2^{'} = (t_1^{'} - t_2^{'})\dfrac{W_1}{W_2}Z; \\ Z = \dfrac{1 - e^{-(1-W_1/W_2)(kF/W_1)}}{1 - (w_1/W_2)e^{-(1-W_1/W_2)(kF/W_1)}}; \\ \Delta t = \dfrac{(t_1^{'} - t_2^{''}) - (t_1^{''} - t_2^{'})}{\ln\dfrac{t_1^{'} - t_2^{''}}{t_1^{''} - t_2^{'}}}; \end{array}\right\} \quad (6)$$

The results of a mathematical simulation of the above-considered models (1)–(6) and a comparison to the experimental data are presented below.

RESULTS AND DISCUSSION

The experimental investigations have been carried out to determine the effect of the forced partial removal of heat generated during braking upon the mean integral temperature of the interacting working surfaces, as well as the coefficient of friction of the disc-brake elements. The experiments were conducted on a natural-size test bench capable of reproducing the real operating conditions of friction elements of the brake [6]. Figure 2 shows a functional scheme of the bench.

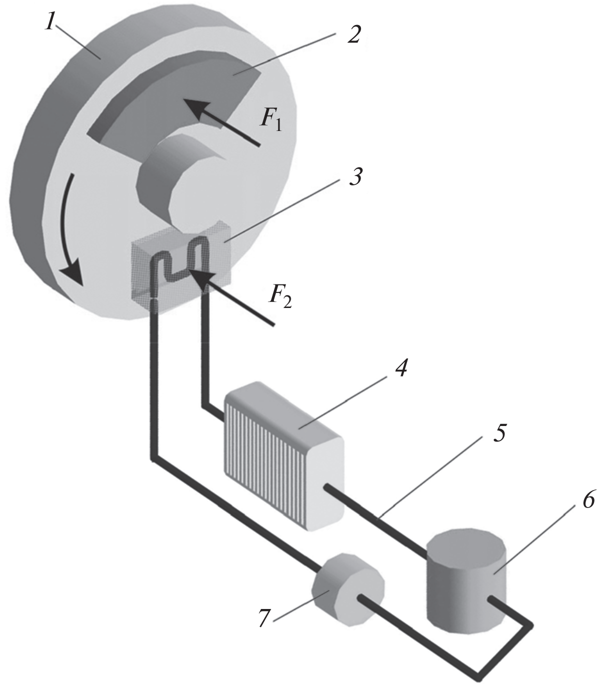

Fig. 2. Functional diagram of the bench: (*1*) brake disc; (*2*) brake pad; (*3*) heat pickup element; (*4*) recuperative liquid-air heat-exchanging apparatus; (*5*) tubing; (*6*) expanding tank; (*7*) pump.

According to the scheme, the heat generated at brake-disc contact *1* with pads *2* is removed from the external side surfaces of the brake disc. The heat is removed from the surface of the brake disc by a special heat pickup element *3* that contains a system of internal channels to enable the circulation of the cooling fluid. The dimensions of the pickup element depend on the designed heat amount subject to rejection from the working zone. A cast-iron block has been used in experiments with special channels made for circulating cooling fluid as a heat pickup element.

Pickup element *3* was found in a constant mechanical contact with the brake disc surfaces. A pump *7* and ducting *5* were made for the circulation of the fluid. The diffusion of the heat in the environment was provided by a recuperative liquid-pneumatic heat exchanger *4.* The brake block *2,* together with the heat pickup element *3,* were pressed against the brake disc *1* under the forces F_1 and F_2, correspondingly.

The temperature of the friction surface was measured by seven thermocouples of type TR-01 mounted in the body of the brake block along its vertical axis. A diesel locomotive section 2TE16 (effective heat-exchange area equals to 52 m^2) was used as a heat exchanger cooled by a VOK-4,0 axial ventilator (max capacity 4500 m^3/h). The results derived in the experiments were recorded and processed by a PC/AT with a SDI-ADC14-32F board meeting certification standard ISO 2000. Furthermore, a module has been used

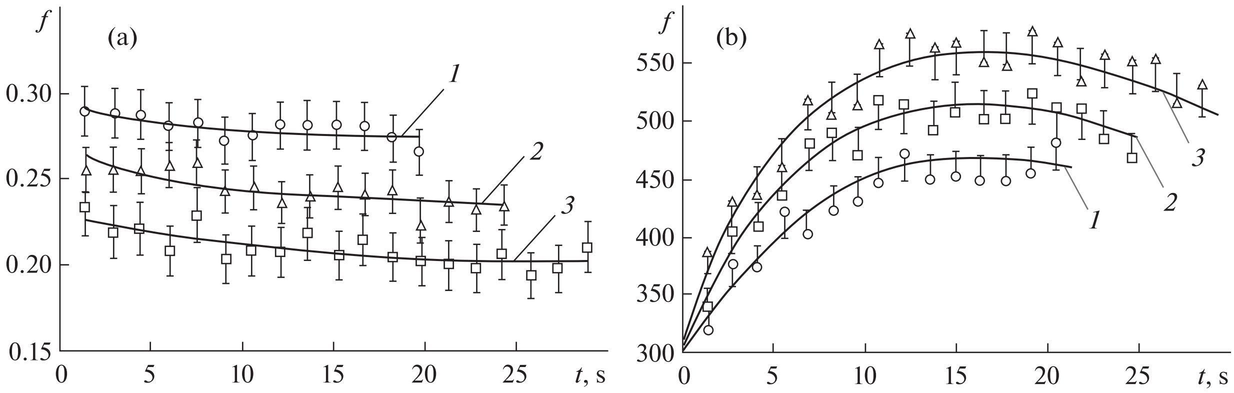

Fig. 3. (a) Experimental results on surfaces the effect of the fluid cooling of the mechanical brake on the coefficient of friction of its working friction and (b) results of experimental and theoretical investigations of the effect of fluid cooling of a mechanical brake on average temperature of working friction surfaces: (*1*) ~50% of the generated heat energy is rejected; (*2*) ~25% of the generated heat energy; (*3*) cooling system is not functioning.

for measurements able to record the temperature, as well as the static and dynamic loads. Corresponding software has been used for the ADC [7, 8].

During the tests, the following external factors were varied within the preset limits: the initial angular velocity value of the brake dis rotation was 1595 rpm (that corresponds to the linear velocity 60 km/h). The linear deceleration value during braking was about 1 m/s^2, the pressing force of the heat pickup element against the brake disc was varied within 105–180 N that, according to the theoretical estimate, corresponds to the case when the cooling system rejects 25–50% of the friction-induced heat energy.

The cooling system operated at a given output (max heat carrier rate was 0.9 and 4500 m^3/h for the water and air, respectively). The nominal contact area of the brake disc with the heat pickup element equaled to 0.0055 m^2. The pressing force of a brake pad to the brake disc was 1500 N.

The external conditions of the experiments were kept as follows: the environment temperature 21–23°C, atmospheric pressure 55 mm Hg, air humidity 65%. Figure 3 illustrates theoretical and experimental results concerning the determination of the coefficient of friction and temperature of the working surfaces of the brakes under different operating modes of the fluid cooling of the disc brake system.

It follows from the data obtained that the coefficient of friction of the friction elements of the disc brake under the conditions of fluid cooling is 30% higher, while the mean integral surface temperature of the friction elements found in interactions is on average 15% lower compared to the case when the fluid cooling is absent (the theoretical value of the heat rejected during braking is about 50%).

According to theoretical estimates, about 50 kWt rejection (about 25% of the total heat amount generated by the brake disc during braking with an initial speed of 120 km/h up to a full stop) can be achieved by using the liquid-pneumatic heat exchanger having the efficient surface area at least 82 m^2 for the average heat-transfer coefficient 38 Wt/(m^2 K), using a fan of 4750 m^3/h capacity, and 1.02 m^3/h. hot heat-carrier consumption (water).

Thus derived theoretical and experimental results prove the efficiency of the fluid cooling of friction elements concerning an increase in the coefficient of friction on the disc brake and the reduction of the working temperature on its friction elements, as well as its use in the rolling stock.

Practical Realization of the Obtained Results

The application of the results in practice related to the fluid cooling of friction elements of the disc brakes requires the creation of a corresponding system based on a recuperative liquid-pneumatic heat exchanging device. A diagram of this system is shown in Fig. 4 [9].

It is proposed to use liquids based on polypropylene glycol $C_3H_6(OH)_2$ as a heat carrier for the fluid cooling of disc brakes in a working temperature range of ambient air of –2 to –20°C, as well as acetate heat carriers (CH_3COOK) for temperatures below –20°C. For climate zones with a minimum temperature of 0°C, it is recommended to use process water.

The calculations have shown that, in the case when process water is used in a fluid cooling system or means close to it with regard to the heat and technical properties with a maximum operation temperature 90°C that is cooled down by atmospheric air and has an initial temperature of 25°C, in order to diffuse 100 kWt of the heat energy (50% of the heat energy generated by a brake disc of a train disc brake at an initial speed of 120 km/h down to a full stop), the minimum surface area of the heat exchanger should be 190 m^2 (once-through system). In this case, the heat-

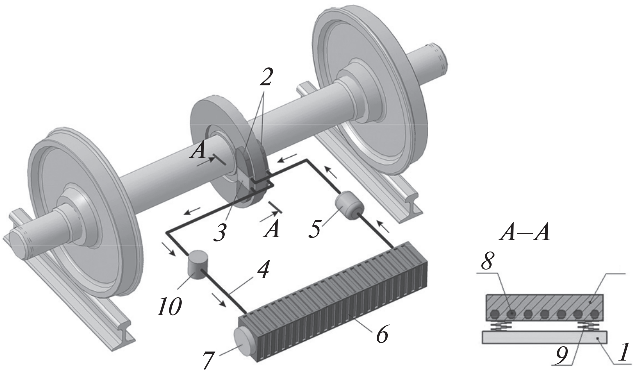

Fig. 4. Fluid cooling system of a locomotive disc brake based on a recuperative liquid-pneumatic heat-exchanging apparatus: (*1*) block; (*2*) friction pads; (*3*) heat pickup elements; (*4*) tubing; (*5*) pump for liquid pumping; (*6*) heat exchanger; (*7*) air fan; (*8*) ducts for cooling liquid; (*9*) elastic element; (*10*) expansion tank.

transfer coefficient is 40Wt/(m^2 K), liquid consumption is 2×10^{-5} m^3/s, air rate 6.0 m^3/s. For the case of the counterflow crossing scheme of the heat carrier motion, the area is 100 m^2 under the same conditions.

The cooling system operates as follows. The heat generated in the brake disc contact with the pads *2* is removed from the external surfaces of the brake disc by the heat pickups *3,* which contain a system of inner channels through which the heat carrier flows. The contact between the pickup elements and the side surfaces of the brake disc is exercised with the help of elastic elements *9.* The heat pickup elements *3* are made of material with a high coefficient of heat conduction. These elements do not participate in the braking process but are only pressed against the braking disc slightly. Above-described solution makes conditions for good heat contact with the brake disc and minimizes wear during sliding. The carrier is transported and cooled by a pump *5* with a pipeline system *4* and a recuperative liquid-air heat exchanger *6.* To compensate temperature changes in the heat carrier volume the system is fit with an expansion tank *10.*

CONCLUSIONS

(1) Both theoretical and experimental results have proved the efficiency of the liquid cooling of the friction elements of disc brakes, providing conditions for raising the coefficient of friction and lowering the operating temperature on the working surface of the friction elements. As a result of the liquid cooling of the friction elements the mean integral temperature on the interacting surfaces of the disc brake drops by about 1.5% and the coefficient of friction increases by roughly 30%.

(2) When the technical water is used in the liquid cooling system or the liquid close to it with regard to its thermophysical properties, in order to dissipate 100 kWt of the heat energy, the minimum efficient surface area of the heat exchanger should be no less than 190 m^2 (for the once-through scheme). In this case, the heat-transfer coefficient should be above 40Wt/(m^2 K), liquid consumption should be 2×10^{-5} m^3/s, and air rate should be 6.0 m^3/s. For the case of a counterflow crossing scheme of heat-carrier motion, the surface area of the heat exchanger should be no less than 100 m^2 under similar conditions.

(3) The design and dimensions of the liquid-pneumatic cooling system of the disc brake make it possible to mount it and prepare it for operation under the conditions of an existing vehicle of rolling stock.

NOTATION

ρ_m	density
t	time
c_{pm}	specific isobar heat capacity
λ_m	coefficient of heat conductivity
T	absolute temperature
$T = f(x, y, z, t)$, x, y, z	orthogonal coordinates
v_x, v_y, v_z	vector projections $\bar{V}$ of the linear velocity of a point on the external surface of the brake disc with coordinates x, y, z, on the corresponding coordinate axes
R, δ	correspondingly, radius and thickness of the brake disc
R_{w}	wheel radius of the locomotive
r	mass center radius of the pad (heat pickup element) of the disc brake (mean friction radius)
$\delta_{1,2}$	pad and heat pickup element thicknesses, correspondingly
V_d, ω	linear and angular velocities of the brake disc, correspondingly
α	heat-transfer coefficient between corresponding surface and ambient air
ε	degree of surface blackness (emissivity coefficient)
σ	Stefan-Boltzmann constant
n	unit vector (normal to the studied region boundary)
λ_m	heat conductivity coefficient of corresponding material of the brake disc, pads or heat pickup elements
q	specific heat flow generated by the sliding contact of the brake disc and pad; ("+" sign) or removed by the heat pickup ("–" sign)
Re_ω, Re_a	Reynolds numbers conditioned by disc rotation and blowing by air counterflow, correspondingly

m vehicle mass

n number of brake pads participating in braking

r_0 mass center radius of the brake pad

ω_0 angular velocity of the disc in the moment preceding braking

$Q_{1,2}$ quantity of heat transferred in hot state (cooling liquid) and accepted cold (atmospheric air) by the heat carrier, correspondingly

ΔQ heat loss into the environment

Δt mean-integral temperature difference of heat carriers lengthwise the heat exchanger

$t_1^{'}$, $t_1^{''}$ temperatures of hot heat carrier at inlet and outlet of heat exchanger, correspondingly

$t_2^{'}$, $t_2^{''}$ temperatures of cold heat carriers at inlet and outlet of heat exchanger, correspondingly

k gross heat-transfer coefficient of the heat exchanger

F efficient surface area of the heat exchanger

$W_{1,2}$ water equivalent for hot and cold heat carrier, correspondingly (in the general case, $W = Gc_p$)

G mass heat-carrier discharge

c_p specific isobar heat capacity of the heat carrier

f coefficient of friction between the brake disc and pad during braking

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Translated by L. Pushkina

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