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Temperature regime and energy capacity of bus friction clutches in urban operating conditions

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Abstract. Problem. *Friction clutches (FC) of motor vehicles operate under significant thermal and dynamic loads, especially in urban driving conditions characterized by frequent starts and stops. The reliability and service life of FC largely depend on the thermal state of friction pairs, which is influenced by the thermophysical properties of materials, clutch design, and heat transfer conditions. Existing methods for calculating and designing FC often do not sufficiently consider the interrelated dynamic, thermal, and frictional processes occurring during operation, which leads to a reduction in their service life and operational efficiency.* ***Goal.*** *The purpose of the work is to study the temperature regime and energy consumption of bus friction clutches in the urban operating cycle.* ***Methodology.*** *The temperature distribution in FC elements was described using the differential heat conduction equation in a cylindrical coordinate system with corresponding initial and boundary conditions. Due to the complexity of the clutch geometry and operating modes, a three-dimensional thermal model based on thermal resistance grids was developed. The slipping work of the friction clutch during operation of the Etalon bus under SAE-regulated driving modes was calculated to determine the boundary conditions of the second kind (heat generation). Boundary conditions of the third kind (heat transfer) were also substantiated. Numerical simulation was performed using the Fourier-2 x, y, z software package.* ***Results.*** *The study determined the temperature distribution in the friction pairs of the clutch under urban operating conditions and established the influence of heat transfer coefficients on the thermal state and energy consumption of the friction clutch. It was shown that changes in heat transfer conditions significantly affect the temperature indicators and operational characteristics of FC friction pairs.* ***Originality.*** *A three-dimensional thermal model of a bus friction clutch operating in an urban cycle was developed, taking into account the interrelated thermal and frictional processes as well as variable heat transfer conditions. The proposed approach allows a more accurate assessment of the thermal behavior and energy capacity of FC under real operating conditions.* ***Practical value.*** *The obtained results can be used in the design and optimization of bus friction clutches, improvement of their thermal reliability and durability, as well as in the selection of effective operating modes and materials for friction pairs.*

Keywords: *vehicle, friction clutch, reliability, thermal model, energy consumption, SAE standard.*

Introduction and analysis

For FC traction and transport vehicles, it is economically feasible for the durability of their main components and parts to be equal to the engine's service life before major repairs. Experience in operating MCV [1] indicates that the goal has not yet been achieved. That is why regulatory and technical documentation often provides for 1-2 replacements of driven discs during the specified service life, as well as additional repair actions (grinding of pressure and intermediate discs, etc.).

The basis of the reliability of the FC is determined by its main dimensions. Attempts to change the mass and dimensions should not be an end in themselves, since the FC operates in the "engine-transmission" system and its main specific energy indicators must be in a certain correspondence (for example, the ratio of the maximum engine torque to the area of the FC pads).

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The energy consumption of the FC is determined mainly by the heat resistance of the linings, which in modern conditions for various friction pairs of MCV are made of composite materials, which are characterized by high stability of the friction coefficient at high temperatures.[2] Therefore, the issue of studying the temperature regime of fuel injection systems under their load conditions during the movement of commercial vehicles in the urban cycle has become relevant.

The latest fundamental research on this issue includes the work [3], which considers the problems of increasing the efficiency of automatic transmission starting and improving the driver's working conditions and passenger comfort by improving the output indicators of automobile clutch control systems. In work [4], an inductive clutch control sensor is studied, which makes it possible to obtain a signal in digital form without converters, which ensures stability and compactness. Work [5] is devoted to the justification of the creation of a cyber-physical system that allows you to work out the algorithm for controlling transmission elements, including the clutch, under different operating conditions without the need to install it on a real automatic transmission. Work [6] analyzes the work processes that occur in an automated gear shifting mechanism. Work [7] is devoted to the selection of rational parameters of an automated clutch control system for a robotic transmission using simulation modeling, and works [8 – 10] consider the work processes in the automatic transmission of an automatic transmission and ways to improve them.

Purpose and Tasks

The purpose of the work is to study the temperature regime and energy consumption of bus friction clutches in the urban operating cycle.

To achieve this goal, it is necessary to solve the following tasks:

- to carry out a mathematical description of thermal processes occurring in FC pairs;
- create a grid thermal model of the FC;
- to investigate the change in the temperature regime of the FC during the MCV tests provided for by the SAE standard;
- to study the influence of heat transfer on the thermal state of the bus's fuel system during their operation;
- obtain and analyze research results.

Mathematical description of thermal processes occurring in clutch elements

The thermal process in the elements of the friction pairs of the clutch depends on the thermophysical properties of the materials and the design of the clutch itself, heat transfer from its surfaces and operating conditions [10,11]. To determine the influence of various factors on the thermal state of the clutch, it is advisable to focus on a mathematical model of the process.

The temperature distribution between the coupling elements (Fig. 1) is described by the equation [11]

$$d_i V\left[\lambda_i\left(t_i\right) grad t_i\right] = \\ = j_i \frac{\partial\left(C_i t_i\right)}{\partial\tau}, \rho \in D_i, \tau \triangleright 0, \left(i = 1,2,3,4\right), \quad (1)$$

where λ_i –coefficient thermal conductivity friction materials couple; C_i – specific heat capacity materials of the same elements; j_i – density of materials; D_i –areas, occupied by counter bodies; $\rho(x, y, z)$ –investigated point systems; t - temperature; τ – time, S_i -heat transfer surfaces, h_i -friction surfaces.

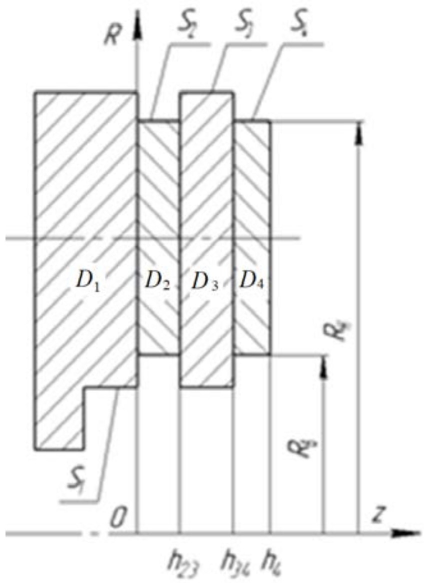

Fig. 1. Estimated thermal power scheme clutch

The above equation is supplemented with the following boundary conditions [14]:

$$\left[grad t_1\right]_{\rho\in s1} - \frac{\alpha_1}{\lambda_1}\left[t_1 - t_n(\tau)\right]_{\rho\in s1} = 0 \quad (2)$$

$$\left[grad t_2\right]_{\rho\in s_2} - \frac{\alpha_2}{\lambda_2}\left[t_2 - t_n(\tau)\right]_{\rho\in s2} = 0 \quad (3)$$

$$\left[gradt_3\right]_{\rho\in s_3} - \frac{\alpha_3}{\lambda_3}\left[t_3 - t_n(\tau)\right]_{\rho\in s_3} = 0 \qquad (4)$$

$$\left[gradt_4\right]_{\rho\in s_4} - \frac{\alpha_4}{\lambda_4}\left[t_4 - t_n(\tau)\right]_{\rho\in s_4} = 0 \qquad (5)$$

$$\left[\lambda_1 gradt_1 - \lambda_2 gradt_2\right]_{\rho\in s_{12}} = q_{12}(\tau) \qquad (6)$$

$$\left[\lambda_2 gradt_2 - \lambda_3 gradt_3\right]_{\rho\in s_{23}} = q_{23}(\tau) \qquad (7)$$

$$\left[\lambda_3 gradt_3 - \lambda_4 gradt_4\right]_{\rho\in s34} = q_{34}(\tau) \qquad (8)$$

The following notations are adopted in the above equations:

$\alpha_1, \alpha_2, \alpha_3, \alpha_4$ – coefficients heat transfer from surfaces; S_i; q_{12}, q_{23}, q_{34} – heat flux density on heat generating surfaces.

It should also be borne in mind that:

$$t_1 = t_2 \text{ at } \rho \in s_{12} \quad t_2 = t_3$$
$$\text{t } \rho s_{23}\,;\, t_3 = t_4 \text{ at } \rho s_{34} \qquad (9)$$

Initial conditions:

$$t_{i(\rho,0)} = t_0 = const, i = 1,2,3,4. \qquad (10)$$

Because thermophysical characteristics in this case do not depend on temperatures, i.e. $\lambda_i = const, (cj)_i = const$, then initial differential equation in a cylindrical coordinate system (z, y, φ) will take the following form:

$$\Delta^2 t_i = \frac{1}{\chi_i}\cdot\frac{\partial t_i}{\partial \tau}, \rho \in dD_i, \tau \triangleright 0, i = 1,2,3,4. \qquad (11)$$

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$$\text{at } \rho s_{23}\,;\, t_3 = t_4 \text{ at } \rho s_{34} \qquad (12)$$

Initial conditions:

$$t_{i(\rho,0)} = t_0 = const, i = 1,2,3,4. \qquad (13)$$

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$$\Delta^2 t_i = \frac{1}{\chi_i}\cdot\frac{\partial t_i}{\partial \tau}, \rho \in dD_i, \tau \triangleright 0, i = 1,2,3,4. \qquad (14)$$

where $\Delta^2 = \frac{\partial^2}{\partial r^2} + \frac{1}{z}\cdot\frac{\partial}{\partial r} + \frac{\partial^2}{\partial z^2}$ – LaPlace operator in cylindrical coordinates; $\chi = \frac{\lambda_i}{(cj)_i}, i = 1,2,3,4$ – coefficients thermal conductivity materials of clutch elements.

Therefore, the boundary conditions will take the form:

$$\left[gradt_1\right]_{\rho\in s_{1z\triangleleft 0}} - \frac{\alpha_1}{\lambda_1}\left[t_1 - t_n(\tau)\right] = 0 \qquad (15)$$

$$\left[\frac{\partial t_2}{\partial r}\right]_{r=R_{2B}} - \frac{\alpha_2}{\lambda_2}\left[t_2\left(R_{2B,z,\tau}\right) - t_n(\tau)\right] = 0 \qquad (16)$$

$$\left[\frac{\partial t_2}{\partial r}\right]_{r=R_{2H}} - \frac{\alpha_2}{\lambda_2}\left[t_2\left(R_{2H,z,\tau}\right) - t_n(\tau)\right] = 0 \qquad (17)$$

$$\left[\frac{\partial t_3}{\partial r}\right]_{r=R_{3B}} - \frac{\alpha_3}{\lambda_3}\left[t_3\left(R_{3B,z,\tau}\right) - t_n(\tau)\right] = 0 \qquad (18)$$

$$\left[\frac{\partial t_3}{\partial r}\right]_{r=R_{3H}} - \frac{\alpha_3}{\lambda_3}\left[t_3\left(R_{3H,z,\tau}\right) - t_n(\tau)\right] = 0 \qquad (19)$$

$$\left[\frac{\partial t_4}{\partial r}\right]_{r=R_{4B}} - \frac{\alpha_4}{\lambda_4}\left[t_3\left(R_{4B,z,\tau}\right) - t_n(\tau)\right] = 0 \qquad (20)$$

$$\left[\frac{\partial t_4}{\partial r}\right]_{r=R_{4H}} - \frac{\alpha_4}{\lambda_4}\left[t_3\left(R_{4H,z,\tau}\right) - t_n(\tau)\right] = 0 \qquad (21)$$

$$\left[\frac{\partial t_4}{\partial z}\right]_{z=h_4} - \frac{\alpha_4}{\lambda_4}\left[t_3(r, h_4, \tau) - t_n(\tau)\right] = 0 \qquad (22)$$

$$\lambda_1\left[\frac{\partial t_1}{\partial z}\right]_{z=0} - \lambda_2\left[\frac{\partial t_2}{\partial z}\right]_{z=0} = q_{12}(\tau)$$
$$t_1(r, o.\tau) = t_2(r, o, \tau) \qquad (23)$$

$$\lambda_2\left[\frac{\partial t_2}{\partial_z}\right]_{z=h_{23}}-\lambda_3\left[\frac{\partial t_3}{\partial z}\right]_{z=h_{23}}=q_{23}(\tau) \quad (24)$$
$$t_1(r,h_{23},\tau)=t_2(r,h_{23,}\tau)=q_{23}(\tau)$$

$$\lambda_3\left[\frac{\partial t_3}{\partial_z}\right]_{z=h_{34}}-\lambda_4\left[\frac{\partial t_4}{\partial z}\right]_{z=h_{34}}=q_{34}(\tau) \quad (25)$$
$$t_1(r,h_{34},\tau)=t_2(r,h_{_{34},}\tau)=q_{34}(\tau)$$

Initial conditions:

$$t_i(r,z,o)=t_o=const, i=1,2,3,4. \quad (26)$$

The above equations for such systems cannot be solved analytically at present. Certain difficulties also arise with incorrect initial and boundary conditions caused by the cyclic operation of the FC, which necessitates the creation of a grid thermal model using the Fourier-2 (x, y, z) software package [12] with a given reproduction of repeated short-term cycles [13] that occur during the operation of the MCV. Therefore, it is necessary to dwell on the methodology creation of a three-dimensional mesh thermal model of adhesion

The "Fourier – 2 x, y, z" complex is mostly adapted to work with a two-dimensional grid of thermal resistances. This complex performs automatic calculation of grid resistances with given constant steps in X, Y coordinates and makes it possible to determine the Z dimension only over the entire grid.

To construct a three-dimensional thermal model of the coupling, we will use a grid of thermal resistances [2]. In this case, all equations obtained by the method of transition to algebraic equations, the coefficients of which are thermal resistances, are identical with respect to the grid of thermal resistances.

Considering the above regarding automobile clutch, three-dimensional heat conduction processes can be reduced to a two-dimensional system of differential equations with correction of thermophysical coefficients and boundary conditions from the Z coordinate.

Fig. 2 shows a diagram of the mesh model of the clutch sector, where the sector of the flywheel, driven and pressure discs will be modeled according to the Z coordinate.

It should be noted that the change in the average sector size along the Y coordinate with a step Δ Y correlated by the change in thermophysical coefficients for each horizontal row of the grid.

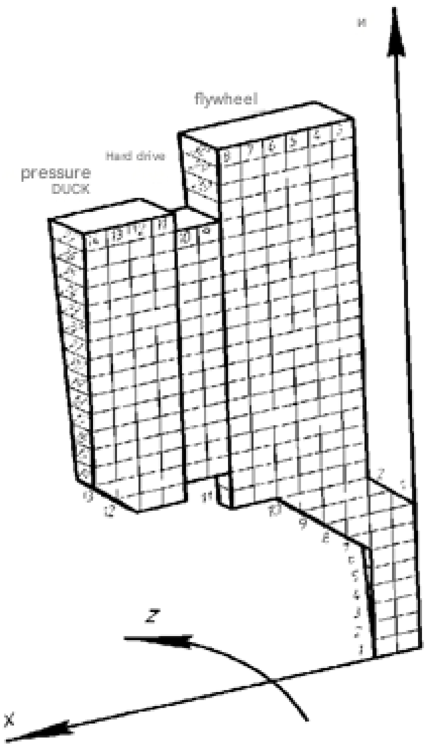


Fig. 2. Scheme of the mesh model of the clutch sector (breakdown 14x30 nodes, node size – 5x5 mm)

The first upper horizontal row of the coupling model corresponds to the set steps in space Δ X, Δ Y, Δ Z, and it specifies the actual values of the thermophysical coefficients of the materials FC. In the following rows, their values change in accordance with the change in size along the Z coordinate depending on the spatial step along the Y coordinate. The change is carried out by introducing the coefficients kz. Similarly, taking into account this coefficient, the heat flux density in the friction zone and the heat transfer coefficient will be correlated q from the clutch discs.

As noted above, the problem was solved using the software package "Fourier - 2 x, y, z"[2,12,13]. The methods described in the aforementioned works can be adapted to the study of thermal processes in clutches.

In the general case, the determination of the temperature fields of a solid consists of integrating the differential equations of non-stationary thermal conductivity [11] under appropriate boundary conditions, since there are no external sources for coupling, it is possible to reduce the determination of temperature fields in its elements to the solution of the equations written above in the cylindrical coordinate system under known boundary conditions.

As studies have shown [2] during computer modeling temperature fields disks in transverse intersections enough take into account the change in thermal resistance along the radius and axis, which will reduce the solution to axisymmetric tasks. For such situations the model is simplified because there is no need to simulate over flow heat along the coordinate φ.

According to the requirements for the solution axisymmetric tasks for location are as modeling on a grid and for step selection, the modeling object depicted in certain scale, are divided into rectangular nodes, with sides, parallel to the coordinate axes R and Z (Fig. 3).

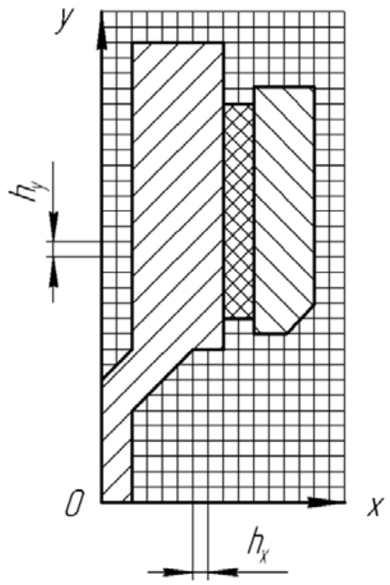


Fig. 3. Layout of the modeling area on the grid

Modeling allows get curves of temperature changes of the clutch elements both during one specific cycle (Fig. 4) and during 10 cycles MCV tests (Fig. 5), defined by the SAE standard [14].

Design parameters for the bus “STANDARD” A08128 made up: $L_б$ =16.47 kJ; slip time τ = 3.2 s, and the heat flux density Q = 149.7 kJ/(m^2 s) according to the method [15].

Computer modeling allows replace periodic, equivalent for work load test cycles, averaged aperiodic mode tests (due to a drop in the friction coefficient). The heat transfer coefficient from the outer surfaces of the pressure plate and flywheel is taken by analogy with disc brakes [15] in between 34 and 36W/(m^2·degree), and from the inner surfaces of the friction pairs at heirs breeding– 25–26W/(m^2·degree).

Computer modeling was used to study the impact of heat transfer on the thermal state of the clutch, which took place at given method tests. As can be seen from Fig. 6, a 4-fold increase in the coefficient heat transfer relative to the nominal value ($\alpha_{ном.зовн.}$ = 36W/(m^2·degree)) only from the outer surface of the pressure disc leads to a decrease in temperature at the end of the tenth cycle tests by approximately 14%. In the case of simultaneous provision of heat drains from the internal surfaces of the friction pairs during their dilution within $\alpha = 4 \cdot \left(\alpha_{ном.вн.}\right)$, where $\alpha_{ном.вн.}$ =25W/(m^2·degree), then the temperature value will decrease by 22%.

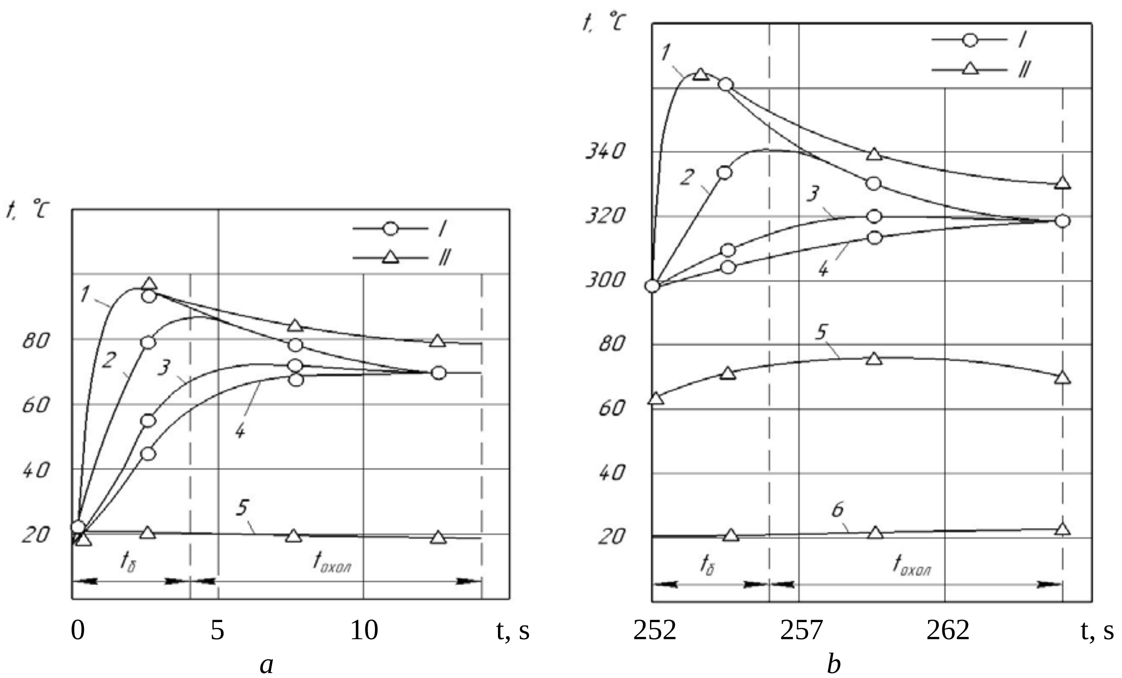


Fig. 4. Change clutch element temperatures bus "ETALON" A08128 by sometimes during tests (modeling results): a – first cycle; b – tenth cycle; I – pressure disc; II – pad; 1 – on the friction surface; 2, 3 and 4 – respectively at a distance of 6, 12 and 18 mm from the friction surface; 5 and 6 – at a distance of 1 and 2 mm from the friction surface

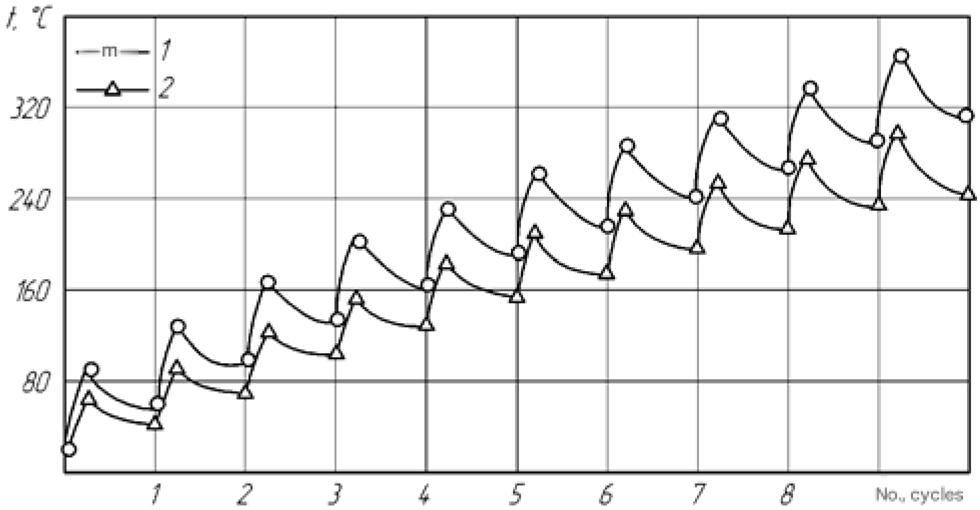


Fig. 5. Change temperatures of friction surfaces of clutch elements bus "ETALON" A08128by cycles (simulation results):1 — pressure plate; 2 — flywheel

Considering that effective means temperature reduction there are heat removal from internal friction surfaces during clutch slippage, provided that heat transfer is ensured within $\alpha_{в.н.}$ =100–110 W/(m2·degree), the thermal regime will decrease by 35%. This indicates that for MCV operating in severe conditions with cyclic shutdowns clutches (e.g. city buses, forklifts, tractors), it is relevant to use wet clutches or dry clutches with air cooling [4, 9, 12].

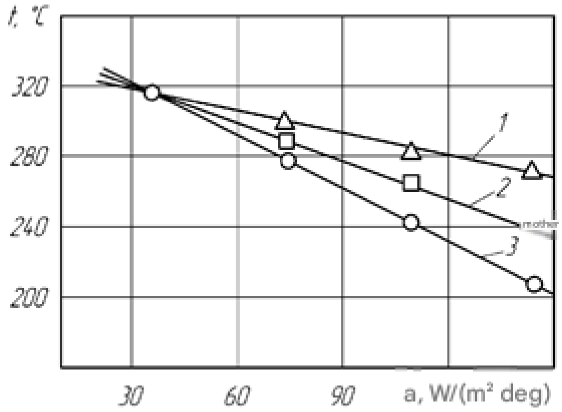
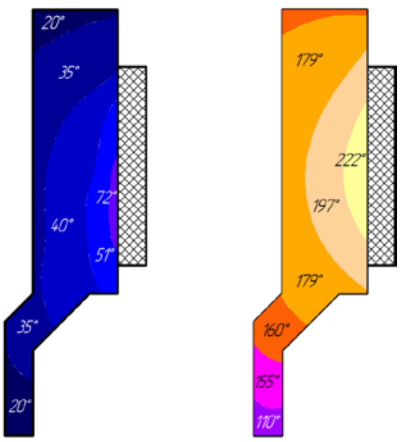


Fig. 6. Clutch pressure plate temperature drop bus "ETALON" A08128 at the end of the tenth cycle tests by heat transfer: 1– from the outer surface of the disc; 2 – from the outer surface of the disc and the inner surfaces of the friction pair during their separation; 3 – from the outer surface of the disc and the inner surfaces of the friction pair during slipping and their separation

Results modeling at tested [6] that the temperature regimes of the flywheel and the pressure plate are different due to their different masses. This negatively effects on the overall durability of the clutch, as the lining on the pressure plate side wears out faster due to the different temperatures. In view of this, during designing clutches must strive for leveling fly wheel and pressure plate masses or use materials with different thermophysical properties to compensate for uneven wear of the linings.

So, computer modeling allows you to quickly and efficiently already at the design stage determine not only the temperature values of the friction surfaces of the clutch, but also to evaluate their temperature fields (Fig. 7) in every preset moment time.

In addition, it allows you to compare different friction lining materials, in terms of the time it takes to reach specified critical temperatures. To deepen the study of thermal processes. What are happening in the lining at small distances from the friction surface and for a more accurate determination of the temperature gradient, mesh models with a finer step should be used.



a *b*

Fig. 7. Engine flywheel temperature fieldbus "ETALON" A08128at the end heating: a – first cycle tests; b - tenth cycle tests

Studies have shown that friction materials used in bus clutches "ETALON" A08128, ensure its sufficient energy intensity during operation in the urban work cycle, since they do not reach critical temperature values in the friction zone.

Conclusion

A mathematical description of the course of thermal processes in friction clutch pairs has been carried out.

The method of computer modeling of the process of heating and cooling of the MCV clutch on a three-dimensional mesh model is described and the study on a two-dimensional one is justified by the solution of the axisymmetric problem. The patterns of changes in the temperature state of the bus clutch during tests according to the SAE standard, which is equivalent to the operation of the MCV in the urban cycle, are obtained.

Effective means reducing clutch temperature there are ensuring heat removal from friction surfaces within $\alpha_{в.н.}$ =100 – 110 W/(m^2·degree), which will reduce its thermal state by 35%. Therefore, for MCVs operated under heavy loads, it is recommended to use wet clutches or dry clutches with forced air cooling.

Computer simulations showed that Due to the different masses of the flywheel and the pressure plate, their temperature states are different. This is negative affects on the service life of the clutch, due to the fact that the driven disc lining on the pressure side wears out more intensively than on the flywheel side.

The simulation results showed that the energy consumption of the bus friction clutch "ETALON" A08128 sufficient, since during operation its friction pairs do not heat up to critical temperatures, due to a drop in the friction coefficient.

It is advisable to take the results of the research into account at the design stage of MCV friction clutches for a preliminary assessment of their thermal state and energy consumption.

Conflict of interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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Температурний режим та енергетична міцність фрикційних зчеплень автобусів в міських умовах експлуатації

Анотація. Проблема. *Фрикційні зчеплення (ФЗ) автотранспортних засобів працюють під значними тепловими та динамічними навантаженнями, особливо в міських умовах руху, що характеризуються частими стартами та зупинками. Надійність та термін служби ФЗ значною мірою залежать від теплового стану пар тертя, на який впливають теплофізичні властивості матеріалів, конструкція зчеплення та умови теплопередачі. Існуючі методи розрахунку та проектування ФЗ часто недостатньо враховують взаємопов'язані динамічні, теплові та фрикційні процеси, що відбуваються під час експлуатації, що призводить до зниження їх терміну служби та експлуатаційної ефективності.* ***Мета.*** *Метою дослідження є визначення температурного режиму та енергоємності фрикційних зчеплень автобусів в міських умовах експлуатації та оцінка впливу умов теплопередачі на тепловий стан пар тертя.* ***Методика.*** *Розподіл температури в елементах ФЗ було описано за допомогою диференціального рівняння теплопровідності в циліндричній системі координат з відповідними початковими та граничними умовами. Через складність геометрії зчеплення та режимів роботи було розроблено тривимірну теплову модель на основі сіток теплового опору. Для визначення граничних умов другого роду (тепловиділення) було розраховано роботу ковзання фрикційного зчеплення під час роботи автобуса Etalon в режимах руху, регульованих SAE. Також було обґрунтовано граничні умови третього роду (теплопередача). Чисельне моделювання було проведено за допомогою програмного пакету Fourier-2 x, y, z.* ***Результати.*** *У дослідженні визначено розподіл температури в парах тертя зчеплення в міських умовах експлуатації та встановлено вплив коефіцієнтів теплопередачі на тепловий стан та енергоспоживання фрикційного зчеплення. Показано, що зміни умов теплопередачі суттєво впливають на температурні показники та експлуатаційні характеристики пар тертя фрикційного зчеплення.* ***Оригінальність.*** *Розроблено тривимірну теплову модель фрикційного зчеплення автобуса, що працює в міському циклі, з урахуванням взаємопов'язаних теплових та фрикційних процесів, а також змінних умов теплопередачі. Запропонований підхід дозволяє точніше оцінити теплову поведінку та енергоємність фрикційного зчеплення в реальних умовах експлуатації.* ***Практична цінність.*** *Отримані результати можуть бути використані при проектуванні та оптимізації фрикційних зчеплень автобусів, підвищенні їхньої термічної надійності та довговічності, а також при виборі ефективних режимів роботи та матеріалів для пар тертя.*

Ключові слова: *транспортний засіб, фрикційне зчеплення, надійність, теплова модель, енергоспоживання, стандарт SAE.*

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