Evaluation of the efficiency of the friction tread brake of the diesel motor car


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Abstract. The article considers and calculates the brake system of a two-axle service diesel motor car, consisting of two brake subsystems acting separately on two axles of the diesel motor car, provided that each axle is served by one brake cylinder 501Б. A calculation was made in order to evaluate the effectiveness of the friction tread brake of the service diesel motor car model AS-01uz.

1 The relevance of the task

In modern conditions, to ensure the safety of railway traffic, maintenance of the track, transportation of teams of specialists on lightly loaded non-electrified lines, it is advisable to use railcars. The advantage of this type of rolling stock is economy, versatility, reliability, ease of repair, and operation [1-5].

At present, JSC "Tashkent Plant for the Construction and Repair of Passenger Cars" has launched the production of railcars, namely the "Avtomotrisa" model AS-01uz (railcar). Railcars with diesel engines are equipped with a direct-acting automatic brake and a direct-acting non-automatic brake, and railcars with carburetor engines - only a direct-acting non-automatic brake.

The brake system in the presence of a carburetor engine is powered from a compressor installed on this engine; a compressor is mounted on railcars with a diesel engine.

In order to evaluate the effectiveness of the shoe friction brake of the AS-01uz service railcar, a calculation was performed in accordance with the procedure [1].

The brake is one of the most important elements in the organization of safety and speed control for rolling stock.

For wagons with braking (the impact of one or more brake cylinders on each bogie), the calculation is carried out for one or each bogie, taking into account the tare and load per bogie [5-7].

The braking system of a two-axle service railcar operates in such a way that there is one brake cylinder per two axles of the railcar. In this case, brake cylinder 501Б is used. The system is supplied with air from a compressor and a reserve tank with a capacity of 78 liters [1-8].

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The calculation of the brake is carried out in the following steps to confirm the provision:

- the required braking efficiency of the railcar as part of the train;
- skid less braking of the railcar (no skidding of the wheels during braking) [1-8];
- the design diagram of the braking equipment is shown in Figure 1;
- allowable power per block during braking;

Fig. 1. Calculation diagram of the braking system of the railcar AS-01uz:
1 - compressor; 2 - main part; 3 - reservoir; 4 - traverse; 5 - leverage; 6 - rake cylinder; 7 - cast iron block; 8 - brake shoe.

To carry out the calculation of braking equipment, the initial data given in Table 1 are required.

Table 1. Initial data for calculation.

<table>
<thead>
<tr>
<th>Value name</th>
<th>Designation</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum railcar weight, t</td>
<td>Q</td>
<td>15.7</td>
</tr>
<tr>
<td>Type of air distributor</td>
<td>292</td>
<td></td>
</tr>
<tr>
<td>Type of brake cylinder</td>
<td>510B</td>
<td></td>
</tr>
<tr>
<td>Brake cylinder diameter, m</td>
<td>d</td>
<td>0.356</td>
</tr>
<tr>
<td>Brake cylinder spring stiffness, tf/m</td>
<td>C</td>
<td>0.068</td>
</tr>
<tr>
<td>Force of preliminary compression of a spring of the brake cylinder, t</td>
<td>p</td>
<td>0.159</td>
</tr>
<tr>
<td>Estimated output of the brake cylinder rod, m</td>
<td>l</td>
<td>0.15</td>
</tr>
<tr>
<td>Number of brake pads for motorcars, pcs</td>
<td>m</td>
<td>8</td>
</tr>
<tr>
<td>Automatic Brake Linkage Ratio</td>
<td>n</td>
<td>2.955</td>
</tr>
<tr>
<td>Efficiency: – in the lever transmission of the automatic brake – in the brake cylinder</td>
<td>η</td>
<td>0.9 0.98</td>
</tr>
<tr>
<td>Design speed, km/h</td>
<td>V</td>
<td>80</td>
</tr>
</tbody>
</table>
The criterion for ensuring the required braking efficiency is the fulfillment of the condition for the calculated coefficient of the pressing force of the pads $\delta_p$:

$$\delta_p = \frac{K_p n}{T + Q} \geq \delta_p^*$$  \(1\)

where:

- $\delta_p$ - the calculated coefficient of the pressing force of the pads;
- $K_p$ - the calculated force of pressing the brake shoe, $t_f$;
- $n$ - the number of brake pads on the motorcar;
- $Q$ - the maximum gravity force of railcar, $t_f$;
- $\delta_p^*$ - the minimum allowable value of the calculated coefficient of pressing force of cast iron pads according to the availability of brake pressing $[1-5]$.

The estimated load is $Q_{est}$ accepted in accordance with the terms of reference. Quantity values $\delta_p$ with composite and cast iron brake pads are shown in table 2.

Table 2. Minimum, allowable values of the design coefficients of the pressing force of cast iron and composite brake pads.

<table>
<thead>
<tr>
<th>Brake type</th>
<th>Value $[\delta_p]$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Cast iron standard pads</td>
</tr>
<tr>
<td>Cast iron standard pads</td>
<td></td>
</tr>
<tr>
<td>Composite pads</td>
<td></td>
</tr>
</tbody>
</table>

$$K_p = 1.714 \cdot \frac{Q}{n}$$

$$K = \frac{d_k p_{\mu} \eta_{\mu} F_1 \eta_1}{m}$$  \(3\)

where:

- $m$ - the number of brake pads of the railcar, which are subjected to the force from one brake cylinder;
- $d_k$ - brake cylinder diameter, $m$;
- $p_{\mu}$ - design air pressure in the brake cylinder, generally accepted as $3.8 \, kgf/cm^2$ when determining the effectiveness of the brake and $4.2 \, kgf/cm^2$ when checking for the absence of skidding of wheel sets.
For other values of limiting pressures in the brake cylinders, the minimum and maximum pressures in them are respectively taken $11 - 17$;

The obtained value $K = 1.306$ $t_f$ meets the requirements for brake linkages.

The force of the inner spring of the brake cylinder is determined by the formula:

$$F = P_{	ext{pre}} + C_l l_{\text{est}}$$

where $P_{	ext{pre}}$ is the pre-compression force of the inner spring of the brake cylinder $12 - 20$; $C_l$ is the stiffness of the corresponding springs, $t_f/m$; $l_{\text{est}}$ is the estimated output of the brake cylinder rod; $F$ is the force of the inner spring of the brake cylinder determined by the formula $F_1 = 0.167 t_f$, thus the condition is met under which the output of the brake cylinder rod is calculated.

Calculations for the absence of skidding of wheel sets of railcars during braking are performed:

- for the minimum load on the axle of the railcar from the mass;
- at the maximum design pressure in the brake cylinders for railcar speeds of 20, 40, 60, and 80 km/h.

Estimated friction coefficient of brake pads $\phi_{\text{kr}}$, is determined by the formula:

$$\phi_{\text{kr}} = 0.27 + V$$

where $V$ is the speed of movement, km/h.

The braking distance of the railcar during emergency braking on the site $[1 - 4]$ was determined by the formula:

$$S = S_\text{pre} + S_\text{d}$$

where $S_\text{pre}$ is the preparatory (before braking) path, m; $S_\text{d}$ is the actual stopping distance, m.

The preparatory stopping distance was determined by the formula:

$$S_\text{pre} = 3.6 V_t$$

where $V_t$ is the speed of the railcar before braking, m/s.
The resulting value $S_t = 88.89$ m meets the requirements.

The actual stopping distance was determined by the formula:

$$S_A = \hat{a} \frac{\sum B_i \hat{V} - V_i}{b \omega}$$

where $\omega$ is the main specific resistance to movement at an average speed $(7-9)$;

$$b_t = \sum \frac{\hat{B}_i \hat{V} - V_i}{\omega b}$$

The value of the specific resistance to movement was determined by the formula

$$\omega = 0.0023 \cdot 0.7 + q$$

where $V$ is the average speed in the interval, km/h.

The value of the specific braking force was determined by the formula

$$b_T = 1000 \varphi_{tr} \cdot \delta_r$$

Comparison of the dependence of braking equipment and path (cast iron pads, pneumatic brakes) in Table 3.

<table>
<thead>
<tr>
<th>Average speed in the interval, km/h</th>
<th>Friction coefficient $\varphi$ in the interval</th>
<th>Specific braking force $b_t$, kgf/t</th>
<th>Basic resistivity $\omega_0$, kgf/t</th>
<th>Actual stopping distance, $S_D$, m</th>
</tr>
</thead>
<tbody>
<tr>
<td>75</td>
<td>0.099</td>
<td>86.875</td>
<td>4.896</td>
<td>68.159</td>
</tr>
<tr>
<td>65</td>
<td>0.105</td>
<td>91.548</td>
<td>4.282</td>
<td>56.569</td>
</tr>
<tr>
<td>55</td>
<td>0.112</td>
<td>97.466</td>
<td>3.726</td>
<td>45.329</td>
</tr>
<tr>
<td>45</td>
<td>0.120</td>
<td>105.205</td>
<td>3.230</td>
<td>34.611</td>
</tr>
<tr>
<td>35</td>
<td>0.133</td>
<td>115.759</td>
<td>2.791</td>
<td>24.622</td>
</tr>
<tr>
<td>25</td>
<td>0.150</td>
<td>131.003</td>
<td>2.412</td>
<td>15.628</td>
</tr>
<tr>
<td>15</td>
<td>0.177</td>
<td>154.957</td>
<td>2.091</td>
<td>7.966</td>
</tr>
<tr>
<td>5</td>
<td>0.227</td>
<td>198.076</td>
<td>1.828</td>
<td>2.420</td>
</tr>
</tbody>
</table>

Based on the diagram, we can determine the length of the braking distance depending on the friction coefficient of the railcar.
2 Power per block during braking

\[ N = \frac{q V}{S m} \]

- \( q \) is the maximum design axial load of the railcar, tf;
- \( V \) is the maximum allowed speed, km/h;
- \( S \) is the minimum estimated stopping distance of a railcar with a full load, m;
- \( m \) is the number of pads acting on the wheel pair;
- \( N \) is the maximum allowable average power per block during emergency braking [21].

The value \( N \) is assumed to be 70 kW for composite pads and 35 kW for cast iron standard brake pads [19-21].

3 Conclusions

1. Obtained as a result of calculating the braking efficiency, the calculated coefficients of the force of pressing the brake pads \( \delta_p = \) 0.873 are greater than the minimum allowable \( \delta_p = 0.6 \); hence the condition is met. The calculated pressing force on the brake shoe is 1.714 tf with the actual pressing force of 1.306 tf [1-5].

2. Realized wheel-rail adhesion coefficients at 20 km/h - 0.150, which is less than the calculated coefficients. Therefore, there is no user interface. Realized wheel-rail adhesion coefficients at 40 km/h are 0.139, which is less than the calculated coefficients. Therefore, there is no user interface.

3. Implemented wheel-rail adhesion coefficients at 60 km/h - 0.129, which is less than the calculated coefficients. Therefore, there is no user interface. Realized wheel-rail adhesion coefficients at 80 km/h are 0.121, which is less than the calculated coefficients. Therefore, there is no user interface.

4. Checking the skid at speeds of 20 km/h, 40 km/h, 60 km/h and 80 km/h, the developed brake system showed that there is no slip to skid.

5. Checking the maximum value of the operating power attributable to the brake cast-iron shoe during braking (at the maximum allowable speed for a railcar of 80 km/h), showed that the calculated power \( N = 21,535 \text{ kW} \) does not exceed the maximum permissible standard value \( N = 70 \text{ kW} \).

References


14. Development of a preliminary design and calculation methods for a body with a removable load-bearing roof adapted for large-block assembly. Participation in the development of the technical design of the body and blocks of internal equipment: Final stage / KF VNIIV; Ruk. topics Meister V.M. – Subject code 30/2–73; No. GR 7307294. Kalinin, 1974. 86 p.


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