DYNAMIC FEATURES OF STANDARD TANK CARS ON CURVED TRACK SECTIONS

Anisimov, Peter S., *Moscow State University of Railway Engineering (MIIT), Moscow, Russia.*

ABSTRACT

Nature of interaction of wheel sets of four-axle and eight-axle tank cars with rails when driving in curves of railway track is considered, including specifics of longitudinal and transverse elastic ties of wheel sets with bogie's side frames. The results of dynamic performance studies (coefficient of dynamic addition of vertical forces on sprung weight and transverse frame force) of tank cars in curves with radii of 350 and 650 m are provided. It follows that eight-axle tank car design under the terms of traffic safety from the point of view of dynamics is at the same level as a four-axle one, therefore, can be used without restriction on all railways under the same rules of maintenance for undercarriages of tank cars of both types.

Keywords: railway, theoretical mechanics, tank car, wheel sets, rails in curves, dynamic performance, elastic and frictional bonds, unextinguished transverse acceleration, coefficient of dynamic addition of vertical forces, frame force, comparative analysis.

Background. *As it is known, when cars are moving in curves of railway track, due to centrifugal shear force at speeds above the equilibrium, which depends on the radius and cant of the outer rail, and due to the action of the centripetal force at speeds below the equilibrium, there is a redistribution of vertical dynamic forces at the tank's sides, and transverse horizontal (frame) forces emerge caused by power curve negotiation and track irregularities in the transverse horizontal plane. Transverse horizontal forces are directed generally toward overloaded wheels, thereby increasing their vertical load on rails, which contributes to increase in resilience of loaded wheel against its rolling in the rail head under the influence of transverse horizontal (lateral) forces. However, less loaded second wheel will have such a resistance to a lesser extent.*

When wheel sets of tank cars are not on curve radii, wheel flanges climb on inner edge of rail head at a certain angle, and this leads to an increase in transverse horizontal forces and in wear of wheel flanges and of inner edge of rail head. The angle of attack of wheel flange on inner edge of the head rail depends on design features of the bogie, and of a total clearance in axle assemblies in a longitudinal direction, which can reach 13–15 mm in operation, as well as on resistance forces to curve negotiation: the larger are they, the larger is the angle at which wheel flange climbs on edge of rail head. Calculations and numerous dynamic tests of freight cars stated that this angle of attack reaches $1{,}5^o$, wherein contact area of wheel flange with inner edge of the rail head is significantly reduced and there is an increased tension and specific friction energy consumption per unit of contact area grows.

The contact area is dependent on the connection in the horizontal plane of bogie's side frames with each other through bogie bolster and spring group with wedge friction shock absorbers; on wheel tread profile, parameters of the shape of the rolling surface of the rail head, which during operation can vary greatly; on ratio of hardness and other mechanical characteristics of wheel flanges and wheel rims, rail heads, especially hardened; on non-radial installation of wheel sets in curves; on coefficient of friction between wheel flange and inner edge of the rail head, which in hardened wheel rims and rail heads is less than in non-hardened.

With increasing connection between bogie's side frames to each other their longitudinal relative displacement (overshooting) decreases, which reduces the angle of attack of wheel flange on the rail head, and consequently increases the contact area of wheel flange with the inner edge of the rail head. This area is able to grow when used in the construction of the bogie radially (following the radius of the curve) of installed wheel sets. Unfortunately, these bogies do not exist yet.

In two-axle bogies of four-axle tank and four-axle bogies of eight-axle tank forces of resistance to curve negotiation of bogies are mainly created by friction wedge shock absorbers of spring groups, axle and pivoted assemblies, side bearers of tank shell and bogie, which are closed on one side of the tank when driving in a curve. Calculation have shown that when the load of wheel sets on rails is of 240 kN torque resistance to rotation of the bogie in a curve generated by closed side bearers may reach 100–150 kNm, and the total guiding force on the wheel flange can reach 150–160 kN [1].

Besides, resistance forces to bogie's curve negotiation may increase when using constantly closed elastic-friction side bearers with partial transfer of the vertical load on them (for 20–25 kN) from the body. The theoretical and experimental studies have shown that in order to make resistance forces caused by such side bearers lower, it is necessary to create a moment of friction forces on them not above 10–12 kNm while transmitting from the car body of vertical load of 20–25 kN per one side bearer. In this case, lateral forces, when wheel flange climbs on inner edge of rail head, do not exceed 5–10 kN, which is permissible under the terms of a normal bogie's curve negotiation and force interaction between wheel sets and rails.

Objective. *The objective of the author is to study dynamic features of standard tank cars when they are moving on curved track sections.*

Methods. *The author uses general scientific and engineering methods, statistics, evaluation approach, graph construction.*

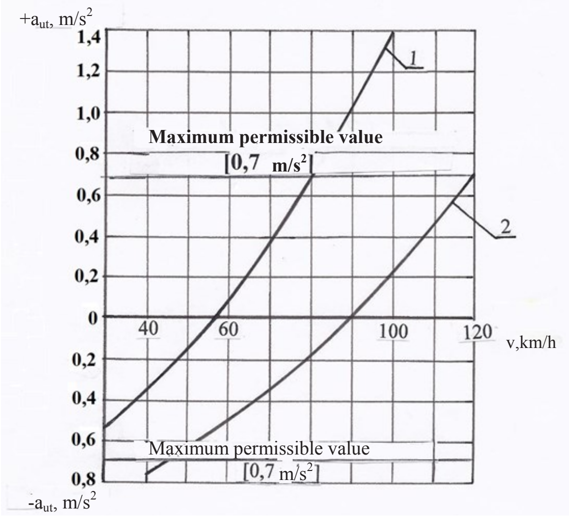
Results. *To reduce horizontal lateral guide forces on wheel sets of the car that is driving on curved sections of railway track it is necessary to:*

– Reduce the horizontal stiffness of rails with transverse elastic deformations of rail heads and exposed to guide efforts of the wheel flange;

– More accurately determine the rate of arrangement and maintenance of outer rail cant in curves (accounting for traction and braking forces, differentiated assessment of actual speeds of different types of cars, etc.);

– Minimize inertial forces and moments of inertia forces in the bogie by reducing its sprung weight;

– Optimize rigidity values of the horizontal transverse links of axle units with bogie's frame and of bogie with the car body;



Pic. 1. Dependence of unextinguished transverse acceleration on the speed of tank movement on curved sections of the track: 1 – curve with radius of 350 m; 2 – curve with radius of 650 m.

– Reduce the value of initial moments of the resistance forces to turn of the bogie with respect to the car body, which depend on rotation angles, wear of friction surfaces of center plate arrangement and side bearers at their closure;

– Weaken the influence of traction and braking forces on vertical loads of wheels on rails, and therefore change the nature of car's bogie curve negotiation.

The theoretical and experimental studies [3, 4] have shown that in order to facilitate curve negotiation of the bogie its frame should be as much as possible, «unfastened» in the horizontal plane. On straight sections on the contrary, it should be as much as possible «fastened» in the plan to reduce twisting movement of wheel sets and hence horizontal transverse forces, wear of wheel flanges and inner edge of the rail head.

This contradiction in the kinematic scheme of bogies when driving on curves and straight sections may be solved, for example, by the use of longitudinal and transverse elastic ties (with their initial tightening) of wheel sets with bogie's side frames. The angular stiffness of the frame should be about 2000–2200 tfm / rad.

Initial tightening takes into account the conditions of geometric unchangeability of position of wheels relative to bogie's side frames truck when driving on straight sections without climbing of wheel flanges on inner side edge of the rail head. When driving in a curve with climbing of wheel flanges with transverse attacks on rail head lateral displacement of wheel sets will arise when external forces exceed the amount of initial tightening of elastic tie of wheel set with bogie's side frames. In other words, wheel sets in this case will have a greater freedom of movement in the rail track and will fit easily into a curve with smaller transverse horizontal forces and wear of wheel flanges and inner edge of the rail head.

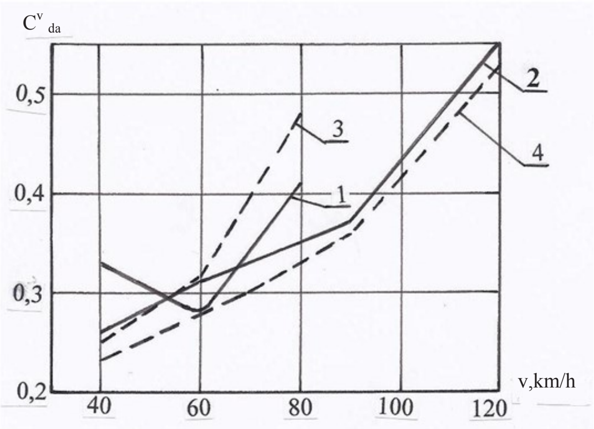
To reduce transverse horizontal forces and wear of wheel flanges and inner edge of the rail head elastic-frictional connection of wheel sets with bogie's side frames having rational parameters can be also used. Structurally, the connection must be one in which friction part will «fasten» bogie's frame in the plan when driving on straight sections and «unfasten» when driving along curves. At the same time, to avoid the shock climbing of wheel flanges on the inner edge of the rail head, in the latter case only elastic part of the connection of wheel sets with bogie's side frames should work. When the car is leaving the curve, this part «turns off» and it is replaced with the friction part.

In constructions of typical bogies there are no such connections, therefore, in curves force interaction of wheel sets with rails increases, especially at low speeds (50–60 km / h), increasing horizontal transverse forces and wear of wheel flanges and the lateral edge of the rail head. This pattern is confirmed by dynamic (running) tests of four-axle and eight-axle tank cars [5].

The basic dynamic performance in the tests was shown by coefficient of dynamic addition of vertical forces for sprung weight of tanks (bolster of bogies) and transverse horizontal (frame) force, transmitted from the wheel set on the frame of bogies. The estimation was made, in particular, for curves with radius of 350 m (S-shaped curve) and 650 m. In the first case, cant of the outer rail was 115 mm, the length of the ease curve was 115 m, retraction of cant of the outer rail was 1 mm per meter length of the ease curve, equilibrium speed – 58 km / h, the maximum speed – 80 km / h, based on the normalized unextinguished transverse acceleration of 0,7 m / s^2. In the curve with radius of 650 m cant of the outer rail was of 150 mm, length of ease curve was 150 m, retraction of cant of the outer rail was 1 mm per meter length of the ease curve, the equilibrium speed – 90 km / h, the maximum – 120 km / h, based on the same acceleration of 0,7 m / s^2.

Pic. 1 shows that under the equilibrium speed unextinguished transverse accelerations are negative, and at speeds higher than the equilibrium those acceleration rates are positive. When the equilibrium speed acceleration is zero, and the nature of the movement of tank cars in circular curves is almost the same as on straight sections of the track.

The graphs in Pic. 2 show that the value of the coefficient of dynamic addition of vertical forces in four-axle and eight-axle tanks differ slightly. Therefore, we can assume that they are less than the maximum permissible 0,7 m / s^2 under the terms of traffic safety. Therefore, this dynamic indicator points out that the amplitude and the character of oscillations in a vertical plane of four-axle and eight-axle tanks are practically



Pic. 2. The maximum value of the coefficient of dynamic addition of vertical forces on bolster of the bogie's while laden tanks driving along curved track sections: 1 and 2 – four-axle tank in curves with a radius of 350 and 650 m, 3 and 4 – eight-axle tank in the curves of the same radius.

identical. However, theoretical calculations indicate that for the same values of relative friction force of friction dampers and static deflection of spring suspension of four-axle and eight-axle tank cars coefficient of dynamic addition of vertical forces in the latter is 20–25% less than that of four-axle one due to more (twice more) wheel sets in the bogie of the eight-axle tank car.

Earlier theoretical studies and calculations have shown that the highest values of the frame force arise from the first (climbing along the car) wheel set of the first bogie. Moreover, a characteristic feature of values measured in curves using wire strain gauge sensor is the existence of periodically changing dynamic and constant components of the frame force. Consequently, the frame force arising in the curve is the sum of its dynamic and constant components.

The dynamic component of the frame force is conditioned by unevenness of rails in the horizontal plane of the railway track, appearing due to the fact that in curves rails are actually located not on the design circular curve, but on the curved polygon. Because of this change in curvature causes the emergence of dynamic component of the frame force.

The constant component of the frame force is conditioned by the value of unextinguished transverse acceleration at the lack of cant of the outer rail of the curve, design of bogies and resistance forces to the installation of the wheel set on the radius of the circular curve. Consequently, according to the value of constant component it is possible to indirectly judge on the impact of design features of bogies of four-axle and eight-axle tank cars on an angle of attack of a wheel flange on the inner edge of the rail head and their wear.

Four-axle and eight-axle tanks have bogies of basically the same design, the only difference is the presence of connecting beams in four-axle bogie, in relation to which two-axle bogies in case of curve negotiation are rotated in the horizontal plane. Span bolster is involved only in the transmission of vertical forces on two-axle bogies, in the perception of longitudinal horizontal forces and provides insubstantial resistance to rotation of four-axle bogie at curve negotiation. In this connection, we can assume that the value of the frame force in eight-axle and four-axle tank should be substantially the same.

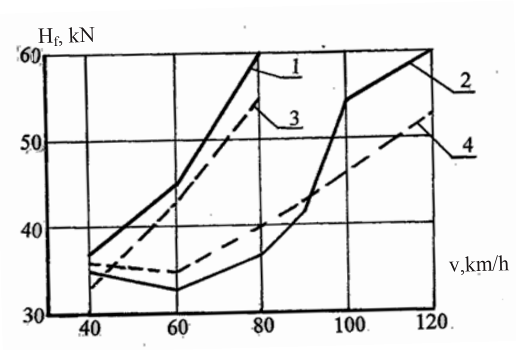
To prove this the values of total frame force and its constant component have been identified separately. Pic. 3 shows that for a four-axle and eight-axle tank car maximum total frame force does not exceed 60 kN, which is less than the maximum permissible value of 84 kN in conformity with the condition of wheel stability against rolling into by a wheel flange on the rail head and with the stability of rail track with rails type R50 and R60 on the transverse shift ballast stone under the influence of frame force under static load from wheel set on rails of 210 kN. At a load from 225,6 kN to 245,25 kN the maximum permissible value of frame force is equal to, respectively, 94 and 100 kN. When driving in a curve with a radius of 350 meters at speed 40–80 km / h, value of the total frame force in eight-axle tank is smaller than that in a four-axle one, by 2,0–5,0 kN. When driving on a curve with a radius of 650 m at speeds of 40–95 km / h values of the total frame force in both tank cars are practically identical, and at speeds of 96–120 km / h total frame force in the four-axle tank is more than in the eight-axle one by 5,0–8,0 kN.

Thus, in this case, the total frame force of eight-axle tank is almost the same as that of a four-axle one.

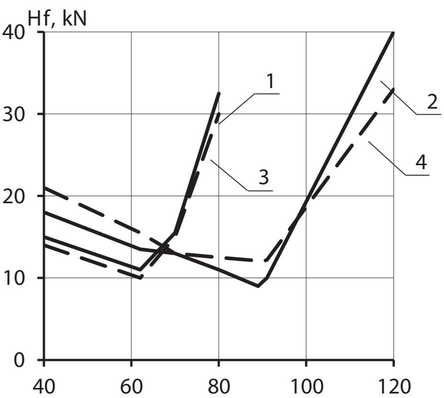
The graphs in Pic. 4 show that the minimum values of the constant component of the frame force on climbing wheel set are observed at equilibrium speed of tank car movement. If the constant component of the frame forces depends only on the unextinguished transverse acceleration, then at the equilibrium speed it will be equal to zero. But there is also a dependence on resistance forces to bogie's curve negotiation. Therefore, the values of the constant component of the frame force at the equvilbrium speed are conditioned only by resistance forces to bogie's curve negotiation. The picture shows that in two tank cars constant components are almost identical. According to their value it is possible to indirectly judge on the degree of wear of wheel flanges and the inner edge of rail heads.

Conclusion. *Under the same operating conditions of cars, state of curves of railway track, the same parameters of bogies and characteristics of elastic-friction relations of wheel sets with side frames of bogies and friction wedge dampers in both tanks the similar result can be expected referring to wear of wheel flanges and the inner edge of the rail head when driving on curves of the same degree.*

The constant component of the frame force depends mainly on the unextinguished transverse acceleration, the values of which are conditioned by the curve radius, cant of the outer rail in a curve and the speed of movement of tank cars. As for the construction of bogies, it does not affect the value of the constant component. The share of the constant component of the frame force caused only by unextinguished transverse acceleration, is approximately 70–75% of its total value.



Pic. 3. The maximum values of the total frame force on the climbing wheel set at movement in curves of the rail track: 1 and 2 – four-axle tank in curves with a radius of 350 and 650 m, 3 and 4 – eight-axle tank in the curve of the same radius.



Pic. 4. Dependence of the constant component of frame force on climbing wheel set on the speed of movement of tanks on the railway track curves: 1 and 2 – four-axle tank in curves with a radius of 350 and 650 m, 3 and 4 eight-axle tank in curves of the same radii.

At speeds above or below the equilibrium the major share of the total frame force is also owned by its constant component, which when tank cars are driving, for example, at a speed of 120 km / h in a curve of 650 m radius is greater than the dynamic component, by 1,7 times in eight-axle tank and by 2 times in four-axle tank. When driving four-axle and eight-axle models in curve with a radius of 350 m, for example, at a speed of 89 km / h the constant component of the frame force is greater than its dynamic component by 1,2 times in both tanks.

These data suggest the importance during designing bogies to pay attention to the elastic-friction relations of wheel sets with bogie's side frames and to reduce resistance forces to bogie's curve negotiation, especially of small radius.

When driving in curves, as well as on straight sections with irregularities in the horizontal transverse plane, at the equilibrium speed, the most important part of the total frame force is due to its dynamic component.

For example, when driving in a curve with a radius of 350 m and at equilibrium speed of 58 km / h the dynamic component of the frame force is more than its constant component by 3,9 times for the four-axle tank and by 3 times for eight-axle tank. When driving in a curve with 650 m radius and at equilibrium speed of 90 km / h dynamic component is more than the constant by 3,6 times in four-axle tank and by 2,9 times in eight-axle tank.

From the analysis of the values of the frame force it is clear that under the same conditions of operation of four-axle and eight-axle tank cars, the same rules of maintenance of their chassis and railway track in curves, design dynamic performance, as well as the degree of deterioration of wheel flanges and the inner edges of the rail heads should be almost identical.

REFERENCES

1. Verigo, M. F. Interaction of track and rolling stock in curves of small radius and the fight against side wear of rails and wheel flanges [*Vzaimodejstvie puti i podvizhnogo sostava v krivyh malogo radiusa i bor'ba s bokovym iznosom rel'sov i grebnej koljos*]. Moscow, PTKB CP MPS publ., 1997, 207 p.

2. Anisimov, P. S. Methods of Evaluation of the Impact of Freight cars on the Track. *World of Transport and Transportation*, Vol. 12, 2014, Iss. 1, pp. 24–32.

3. Vershinskiy, S. V., Danilov, V. N., Khusidov, V. D. The dynamics of the car [*Dinamika vagona*]. Moscow, Transport publ., 1991, 360 p.

4. Anisimov, P. S., Zhelnin, G. G., Kulikovskiy, B. S. Permissible speed of eight-axle tanks dimensions TC [*Dopuskaemye skorosti dvizhenija vos'miosnyh cistern gabarita Tc*]. *Trudy MIIT*, 1986, Iss. 780, pp. 10–24.

5. Anisimov, P. S., Grachiova, L. O. Tests of eight-axle tanks for BAM [*Ispytanija vos'miosnyh cistern dlja BAMa*]. *Zheleznodorozhnyj transport*, 1986, Iss. 5, pp. 35–36.

Information about the author:
Anisimov, Peter S. – D.Sc. (Eng.), professor of Moscow State University of Railway Engineering (MIIT), Moscow, Russia, +7 (495) 684–22–10.

Article received 29.06.2015, accepted 30.10.2015.