Abstract: The paper presents a structural and topological analysis of the braking mechanisms in the railway vehicles. The braking force is transmitted from the piston of a pneumatic cylinder to the axle of the railway vehicle through a linkage. In the case of a single cylinder the braking mechanism has a complex topological structure consisting of three planar kinematic chains: a central horizontal chain and two vertical chains. By highlighting the components of these kinematic chains the paper describes the sequential operation of a multi-mobile mechanism with a single input element (actuator). Most rail cars have four axles meaning two two-axle bogies. The modern trend is to use disk brakes with one or two disks on each axle. In this way the braking mechanism becomes less complicated than a classic linkage, but it requires more brake cylinders at least one per axle.

Key-Words: braking mechanisms, railway vehicles, kinematic chains, disk brakes, structural synthesis, mobilities

1. General considerations

The braking mechanisms of the railway vehicles (rail cars and locomotives) are called braking linkages by the railway operators [1]. The classic braking mechanisms consist of planar articulated bars by which the motion and force of the brake piston are transmitted to the brake shoes.

In the case of two-axle rail cars the braking mechanism consists of a central braking mechanism which operates horizontally and two axle shoe-brake mechanisms that operate vertically. If the shoes exert pressure on the wheel from two opposite sides the mechanism operates symmetrically, and if they exert pressure from only one side the mechanism operates asymmetrically [1, 2].

In the case of a symmetric shoe-brake mechanism the pressure on the axle is exerted from two opposite sides, thus holding the axle in a steady position – perpendicular on the direction of the pressure force. By using two symmetric shoes on each wheel of the axle the pressure exerted on one of the shoes is reduced, thus reducing the wear of the shoes and increasing the friction coefficient. As a result the effect of the braking action is highly efficient.

In the case of an asymmetric shoe-brake mechanism the structure is less complicated and exerts less pressure, but the shoes have to be more robust to withstand a greater pressure and thus on a higher wear. Most shoe-brake mechanisms are symmetrical and they are used on the four-axle freight and passenger rail cars (two-axle bogies).

The asymmetric shoe-brake mechanisms are used on certain types of the freight rail car bogies, whose specific frame design allows the shoes to be installed only on the inside of the bogie wheels.

In the case of certain railway vehicles such as locomotives, motorized and special rail cars, due to the lack of space, the braking mechanism has to be divided in several disk-brake mechanisms that operate individually using a brake cylinder.

In the case of the diesel-hydraulic, diesel-electric and electric locomotives, specific braking mechanisms that have brake cylinders on each wheel are used.

The paper presents the analysis of the symmetrical and asymmetrical shoe-brake mechanisms as well as the disk-brake mechanisms used on the freight and passenger rail cars.

2. The topological structure of the braking mechanism

The automated braking mechanism (fig. 1) consists of two distinct mechanisms that operate as mechanical subsystems [3]:
- the actuation mechanism called the central braking mechanism (fig. 1) which is a planar linkage and it operates horizontally (xy) underneath the rail car frame;
- the end-effector mechanism called the axle shoe-brake mechanism (two of them in fig. 1) which is also a planar linkage and it operates vertically (xz) perpendicular on the axis of the axle.

The central braking mechanism and the two axle shoe-brake mechanisms (in the case of two-axle rail cars) are connected through two rods 1(0_s) and 1(0_d) that have a relatively high length (2–3 m).

By means of these rods the motion is transmitted from the horizontal plane xy (where the central braking mechanism operates) to the vertical plane xz (where the axle shoe-brake mechanisms operate).
The two articulations of both longitudinal rods 1 (fig. 1) have perpendicular axes, one of them is vertical ($D_s$, respectively $D_d$, on exiting the horizontal kinematic chain) and the other axis is horizontal ($A$ on entering the vertical kinematic chain).

2.1. The central braking mechanism

The central braking mechanism (fig. 2) consists of a fixed cylinder 0 with a piston 1 which slides in it, a rod 2 (with three mobile joints A, B and $D_s$) connected via joint A to the piston 1, a rod 3 (with two mobile joints B and C) and a rocker 4 which is geometrically identical to the rod 2 and it has a fixed joint $C_0$ and two mobile joints C and $D_d$.

The rod 2 and the rocker 4 (fig. 2a) are also connected through a stressed helicoidally spring ($a_{24}$) which serves to bring the piston 1 back in its initial position. The kinematic scheme (fig. 2b) does not display the helicoidally spring because it is not considered a kinematic element. By means of the rods 5 and 6 the sliding movement of the piston 1 is transmitted to the end-effector mechanisms on each axle (fig. 1). The mobility of the central mechanism (fig. 2b), which consists of a closed kinematic chain (0, 1, 2, 3 and 4), can be calculated using the following formula [4]:

$$M = 3n - 2C_5 - C_4$$  

(2.1)

where the parameters are: $n = 4$, $C_5 = 5$, $C_4 = 0$.

Thus the mobility results: $M = 3 \cdot 4 - 2 \cdot 5 = 2$

The two mobilities show that the central braking mechanism (fig. 3) that have a single driving element operate differentially, putting in motion the first bar $1(O_s)$, in stage I, a situation in which the rocker 4 remains fixed (point C, the centre of the joint, acts as a fixed joint).

After putting in motion the brake shoe ($S_1$ and $S_2$) and braking the left axle wheels $O_s$ (fig. 3a), the movement of the piston 1 is transmitted through the rocker 4 to the bar $1(O_d)$ (fig. 1) and from here to the braking end-effector mechanism of the right axle $O_d$. So, the braking process is not simultaneous in the very beginning to both axles of the rail car.

2.2. The symmetrical shoe-brake mechanism for the axle braking

Let’s now consider a first type of an axle shoe-brake mechanism as a constructive scheme (fig. 3a) and as a kinematic scheme (fig. 3b) where the fixed joints are located on frame 0 of the two-axle rail car (fig. 1).
On the schemes of the symmetrical shoe-brake mechanism (fig. 3) it can be identified the following kinematic elements: 1, 2, 3, 4, 5, 6, 7, 8, 9, 10 and 11. As well as the kinematic joints with fixed axis: \( B_0, D_0, E_0, F_0 \) and \( H_0 \), and the kinematic joints with mobile axis: \( A, B, C, D, E, F, G \) and \( H \).

![Fig. 3. Constructive (a) and kinematic scheme (b) of the symmetric shoe-brake mechanism (type 1)](image)

The modeling of the mechanism can be obtained using an input element \( m \) (fig. 4a) by which a kinematic element and two kinematic joints are introduced \( (O_m, A_m) \).

![Fig. 4. The modeling of the symmetric shoe-brake mechanism (type 1): the initial position (a), the phase I as the rocker 4 is driven (b) and the phase II as the rocker 11 is driven (c)](image)

The geometrical mobility of the planar mechanism (fig. 4a) consisting of several closed kinematic chains [3, 4] can be calculated using the following formula (2.1):

\[
M = 3n - 2C_5 - C_4
\]  

(2.1)

where the numerical values of the structural parameters are: \( n = 12, C_5 = 17, C_4 = 0 \).

Thus the mechanism mobility results:

\[
M = 3 \times 12 - 2 \times 17 - 0 = 2
\]

Considering the braking mechanism of an axle (fig. 3a) the two independent motions correspond to the sequential motion through which the two brake shoes \( (S_1, S_2) \) are successively pushed towards the rim of the wheel.

This operation requires two phases.

In the phase I when joint B is immobile (fig. 4b) the rocker 4 rotates clockwise and the brake shoe \( S_1 \) is pushed towards the rim of the wheel through the dyadic chain \( LD(3,4) \), after which the joint D is immobilized (fig. 4c).

The structural and topological formula of the MM (“Motorized” Mechanism) [4] in the phase I, using the element \( m \) as actuating component (fig. 4b), is:

\[
MM = MF(0,m) + LD(1,2) + LD(3,4)
\]  

(2.2)

In the phase II (fig. 3c) the structure of the mechanism is modified and the structural and topological formula is as follows [4]:

\[
MM = MF(0,m) + LTr(1,2,3,5) + LD(6,7) + LD(8,9) + LD(10,11)
\]  

(2.3)

The formula (4.3) highlights a triadic chain \( LTr(1,2,3,5) \) and three dyadic chains \( LD(6,7), LD(8,9) \) and \( LD(10,11) \). By continuing the trigonometrically rotation of the input element \( m \)
(fig. 4c), joint D is blocked and the motion is transmitted to the rocker 11 which rotates counterclockwise and pushing joint H and brake S₂ towards the rim of the wheel.

The second type of the axle shoe-brake mechanism with symmetrical brake shoes is shown in fig. 5. In this case there are fewer kinematic elements than in the type 1 (fig. 3).

![Fig. 5. Constructive (a) and kinematic scheme (b) of the symmetric shoe-brake mechanism (type 2)](image)

Considering the modeling of the planar mechanism (fig. 6a) by adding an input element \( m \) and two joints \( (O_m, A_m) \) within another closed kinematic chain, the mobility can be calculated using the following formula [4]:

\[
M = 3n - 2C_3 - C_4
\]

(2.1)

where the numerical values of the structural parameters are: \( n = 10, C_3 = 14, C_4 = 0 \).

Thus the mechanism mobility results:

\[
M = 3 \times 10 - 2 \times 14 - 0 = 2
\]

The two independent motions correspond to the sequential motion of the two brake shoes \( (S_1, S_2) \) which implies the operation of the axle shoe-brake mechanism (fig. 5a) in two phases.

In the phase I when joint B is immobile (fig. 6b) the rocker 4 rotates clockwise and the brake shoe \( S_1 \) is pushed towards the bandage of the wheel through a dyadic chain \( LD (3,4) \), after which the joint D is immobilized (fig. 6c).

![Fig. 6. The modeling of the symmetric shoe-brake mechanism (type 2): the initial position (a), the phase I as the rocker 4 is driven (b) and the phase II as the rocker 9 is driven (c)](image)

The structural and topological formula for phase I using an input element \( m \) (fig. 6b) is:

\[
MM = MF(0,m) + LD(1,2) + LD(3,4)
\]

(2.4)

In the phase II (fig. 6c) the structure of the mechanism is modified and the formula becomes:

\[
MM = MF(0,m) + LTr(1,2,3,5) + LD(6,7) + LD(8,9)
\]

(2.5)

In the formula (2.5) it can be identified one triadic chain \( LTr(1,2,3,5) \) and two dyadic chains \( LD (6,7) \) and \( LD (8,9) \). By continuing the trigonometrically
3. The asymmetrical shoe-brake mechanism for the axle braking

The asymmetrical shoe-brake mechanism (fig. 7) is used for the bogies of some of the freight cars that can be found in CFR’s vehicle exploitation fleet (Romanian railways).

The modeling of the mechanism (fig. 7a) allows to the rod 1 to be put in motion by the input element m which has two joints: one fixed Oₘ and one mobile Aₘ.

The mobility of the mechanism (fig. 7a) can be calculated using the following formula [4]:

\[ M = 3n - 2C_3 - C_4 \]  \hspace{1cm} (2.1)

where the numerical values of the structural parameters are: \( n = 10, C_3 = 14, C_4 = 0 \).

Thus the mechanism mobility results:

\[ M = 3 \times 10 - 2 \times 14 - 0 = 2 \]

The two independent motions correspond to the sequential motion of the two brake shoes \( (S_1, S_2) \) that implies the operation of the braking mechanism for the bogie (fig. 7) in two phases.
In phase I the joint E is fixed and the structural topological formula is:

\[ MM = MF(0,m) + LTr(1,2,3,4) + LD(5,6) \]  

(3.1)

This structure allows the counterclockwise rotation of the rocker 6 (fig. 7b) which implies the motion of joint D (brake shoe S1) from the left to the right.

Fig. 7b. The kinematic scheme of the asymmetric shoe-brake mechanism in phase I of the shoe driving S1

In the phase II the joints D and B are fixed (fig. 7c) and the structural topological formula is:

\[ MM = MF(0,m) + LD(1,2) + LD(4,7) + LD(8,9) \]  

(3.2)

which implies the clockwise rotation of the rocker 9 and the joint G (brake shoe S2) will move towards the left.

Fig. 7c. The kinematic scheme of the asymmetric shoe-brake mechanism in phase II of the shoe driving S2

4. The simple disk-brake mechanism

The topological structure of the disk-brake mechanisms (fig. 8) is similar to the structure of the central braking mechanism (fig. 2) used on two and four-axle rail cars. At present the disk-brake mechanisms are the most used.

The solutions are with one (fig. 8a) or two disks (fig. 8b) that are installed in a rigid position on each axle of the rail car. In this way the braking mechanism is less complicated but it requires a higher number of brake cylinders, at least one per axle.
Based on the constructive scheme of the disk-brake mechanism (fig. 8a) the kinematic scheme of the planar mechanism has been created (fig. 9a).

The mobility of the mechanism can be calculated using the following formula:

\[ M = 3n - 2C_3 - C_4 \]  
(2.1)

where the numerical values of the structural parameters are: \( n = 6, \ C_3 = 7, \ C_4 = 0 \)

Thus when the mechanism is in the neutral position (fig. 9a) the mobility results: \( M = 3 \times 6 - 2 \times 7 = 4 \).

In the phase I joint C is fixed (fig. 9b) and the “Motorized” Mechanism (MM) with the fundamental mechanism MF (0,1) operates within a single closed chain, based on the following structural-topological formula:

\[ MM = MF(0.1) + LD(2,3) \]  
(4.1)

Two of the four mobilities correspond to the free rotation movements of the brake friction pads 5 and 6 around their joints D and E. The other two geometrical mobilities represent the controlled movement of the brake piston 1 and the other is either the rod 2 or the rocker 4 connected to the elements 5 and 6 until they reach the brake disk Df.

In the phase II joint C becomes mobile and the pad 5 translates radial on the surface of the brake disk creating a translation joint (fig. 9c). The structural-topological formula of the mechanism consisting of two closed chains becomes:

\[ MM = MF(0.1) + LD(2,5) + LD(3,4) \]  
(4.2)
5. The complex disk-brake mechanism

The disk brake is tested by CFR on the passenger rail cars since 70’s. In 1973 they had reached the top speed of 200 km/h on these tests obtaining high techno-economical performances with respect to the high torque shoe brake. These good results can be translated on the material economy (4-5 times decreasing on shoe material usage), doubling the wheel rim life and an increase of the braking capacity, in other words a reduction of the braking distance.

The kinematic scheme of the complex disk-brake mechanism (fig. 10) is used by CFR on the passenger rail cars, type 74.000.

![Fig. 10. The kinematic scheme of the complex disk-brake mechanism](image)

The disk-brake mechanism is individual on each axle being actuated by one pneumatic brake cylinder of 8 inch used as actuator.

The geometrical mobility of the disk-brake mechanism (fig. 10) is calculated by the formula

$$M = \sum_{m=1}^{6} mC_m - \sum_{r=2}^{6} rN_r$$  \hspace{1cm} (5.1)

where the values of the parameters are:

- \(m = 1\), the number of the mobilities of each kinematic joint between the mechanism elements;
- \(C_1 = 15\), the number of the identified kinematic joints;
- \(r = 3\), the rank of the adjacent space of the closed kinematic contours (in parallel planes);
- \(N_3 = 3\), the number of the closed kinematic contours of the rank 3.

Thus the mechanism mobility results:

$$M = 1 \times 15 - 3 \times 3 = 6$$

The six mobilities allow the displacement of the four brake friction pads, two on each brake disk.
Due to the fact that there is only one driving element (actuator) represented by the brake piston 1 of the brake cylinder (fig. 10), the four brake friction pads (B1, B2, B3, B4) are not simultaneous driven.

The stepping operation of the multi-mobile mechanism implies the approaching of the brake friction pads in the order of their kinematic chain length with respect to the actuator 1.

The maximum value of the friction force $F_f = \mu_s F_s$ (between the pad and the disk) is deduced from the condition that the wheel not to be locked during the braking process.

This condition is written by means of the static equilibrium equation:

$$2\mu_s F_s \cdot r_m = \frac{1}{2} \mu_s Q \cdot R$$  \hspace{1cm} (5.2)

where:

$r_m$ - average radius of the brake disk;

$R$ - wheel radius at the contact point with the rail;

$Q$ - load force on the axle;

$\mu_s = 0.3-0.4$ - friction coefficient between the pad and disk;

$\mu_a = 0.15$ - adherence coefficient.

From equation (5.2) the pressure force $F_s$ of the pad on disk, depending on the load force on the axle $Q$, results:

$$F_s = \frac{\mu_s R}{4\mu_s r_m} Q$$  \hspace{1cm} (5.3)

Knowing $F_s$ there have been calculated the driving force $F_m$ and thus the necessary pressure depending on the brake cylinder (piston) diameter.

With all techno-economical advantages of the disk brake the braking force of it remains yet limited by the adherence between the rail and wheel.

For this reason the disk brake can be considered as main brake for speeds under 140 km/h with the corresponding extension of the braking distance (fig. 11).

![Fig. 11. The diagrams of the braking distance variation depending on the speed](image-url)

### 6. Conclusions

As it can be observed in figure 11 the disk brake is more efficient than the shoe brake in terms of shortening the braking distance of the railway vehicle.

In addition, at higher speeds the difference between the two types of braking systems is more obvious, the advantages of the disk brake being more evident.

Also the disk-brake mechanisms being simpler than the shoe-brake linkages in terms of the number of the kinematic elements it is clear that the manufacturing cost is lower. In addition, fewer components mean higher reliability, so cheaper maintenance.

Moreover, the usage of the wheel rims is lower too due to the fact that friction has moved on the brake disks.

Thus the maintenance of the rail car is simplified, not requiring the disassembling of the bogies for the changing of the wheel rims.

The disk brake is much more used than the shoe brake not only on passenger rail cars but also on freight cars.
References


