Optimizing the braking system for handling equipment

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Abstract. The article is focused on optimizing the friction disc plates of the braking system integrated into the drive wheel of an automated guided vehicle. The introduction is focused on the application of these types of braking systems. In the next step, the individual mathematical parameters and the acting forces in the braking system are expressed. Using the equation of motion that is written in Matlab, the braking distance and the stop time are graphically expressed. During the Matlab simulation, different diameters of the braking plates are monitored to determine the braking distance. The permitted pressure in the braking plates during the test must not exceed the specified pressure of 2 MPa. The functionality of the braking plates is ensured by hydraulic power. Part of the wheel is a sensor that monitors the speed during braking. Ultimately, a continuous reduction in kinetic energy is achieved without vigorous blocking of the slats during braking. The acquisition of the paper is the output of numerical calculations, which are necessary for the correct design of the wheel with the integrated braking system.

1. Introduction

The new technologies are coming to our market with industrial development. This is also the case in the sphere of logistics. Automated handling transport equipment is implemented in manufacturing processes to increase the productivity of work. These equipment are also referred to as AGVs (automated guided vehicles). The market offers transport equipment (see figure 1) that is braked by the engine brake (electric engine) in the present time [6]. This type of braking is only suitable in certain situations. This way of braking is successful at small speeds. Once the vehicle is driven at a speed of more than 2 m.s⁻¹, it should be equipped with a friction brake system [7].

Figure 1. Automated guided vehicle in present [6].
The reason why the braking system should be implemented is clear. To increase the protection of property and life. This fact is very important today. There is a danger for a persons or property damage in certain situations [12]. This isn’t acceptable. This project will design a wheel with integrated braking system which maximum speed is 5 m.s\(^{-1}\) (see figure 2). The handling devices have great applications in the automotive industry. These devices are found in three car factories in Slovakia. They allow transporting hundreds of kilograms. In this case, the device is designed to carry a load of 1100 kg. The weight of the reserve is set on 100 kg, which is thinking on the calculations. The reason is a safety at overload [11]. The handling equipment uses the energy of the battery, which powers an electric engine. Electric vehicles and automated guided vehicles have a significantly shorter range compared to conventional vehicles with an internal combustion engine [8]. These devices need to charge the battery more often.

The transport handling equipment is oriented in the space using sensors. These sensors monitor the space (warning zones) in front of the vehicle (see figure 3). The software evaluates the warning zones according to the distance to the vehicle. If there is an obstacle in the yellow or orange zone, the vehicle will slow down. If the obstacle is in the red zone, the vehicle stops. In crisis situations, the vehicle must be stopped immediately. For this purpose, the friction disc brake will be used as engine braking is ineffective in these cases [7].

![Figure 2. Positioning the brake system in the wheel. Source: [3].](image)

![Figure 3. The safety zones for vehicle. Source: [7].](image)

### 2 Technical knowledge of the topic

The design of the wheel will be created to base on the boundary conditions listed in table 1. It is intended to equip the wheel with a friction disc plate in the original wheel space. Therefore, it is necessary to optimize the size of the friction disc plate. The space in the wheel is between 80 and 220 mm. It should be noted that the entire brake assembly must be located inside the space of the wheel. It mustn’t exceed pressure 2 MPa between the friction disc plates during braking. If the braking pressure between the friction disc plates exceeds 2 MPa during braking, the brake system will be damaged [10]. These boundary conditions must be borne in mind.

| Table 1. Boundary conditions. |
|--------------------------------|
| Maximum speed of the handling equipment | 5 m.s\(^{-1}\) |
| Internal space of the drive wheel | 80 – 220 mm |
| Maximum allowed pressure at braking (between the friction disc plate) | 2 MPa |
Figure 4 represents the drive wheel in the graphic view with multiple dimensions of friction disc plates. As it can be seen from the figure, the wheel space inside (80-220 mm) is very limited.

![Legend of the Figure 4.](image)

1. Original drive wheel
2. FDP-D
3. FDP-C
4. FDP-B
5. FDP-A

Figure 4. Original wheel with considered friction disc plates [9].

3. The theory on the theme of the braking system

Tangential and normal force act in the brake system with friction disc plates (see figure 5). The normal force imagines the pressure force in the friction disc plate. The tangential force characterizes the braking force, the size of which depends on the friction coefficient of the friction disc plate.

![Figure 5. The acting forces in the friction disc plate](image)

The braking torque is generated during braking, the size is calculated according to the following equation (see equation 1). The value braking torque is very important in designing the brake body (static part of the brake).

\[ M_b = \frac{F_{t1} + F_{t2}}{2} \cdot \tau_s \rightarrow M_b = F_t \cdot \tau_s \, \text{[Nm]} \]  

(1)

where: \( M_b \) is braking torque [Nm]; \( F_{t1} \) a \( F_{t2} \) is tangential force [N]; \( \tau_s \) is the median radius of the friction disc plate [mm].

The construction of brake is designed with push springs, which return the brake to its original condition after releasing the brake. To calculate the real normal force, it is necessary to subtract the force from the compression springs (see equation 3). The normal force acting on both sides of the friction disc plates is the same (see equation 2) [10].

\[ F_{n1} = F_{n2} \, \text{[N]} \]  

(2)

where: \( F_{n1}, F_{n2} \) is normal force [N].

\[ F_{zb} = \frac{G \cdot d^4 \cdot s}{8 \cdot D^3 \cdot n} \, \text{[N]} \]  

(3)
where: $F_{sp}$ is the force exerted by one spring [N]; $d$ is the diameter of the spring [mm]; $D$ is the outside diameter of the spring [mm]; $s$ is length after push [mm]; $n$ is the number of spring thread active [-]; $G$ is Modulus of Rigidity [GPa].

The sum of the two tangential forces acting on both sides’ friction disc plate represents the result of the total tangential force. And also the multiplication of the normal force by the coefficient of friction for the brake disc plate (see equation 4) leads to tangential force.

$$\frac{F_{t1}}{2} + \frac{F_{t2}}{2} = F_t \rightarrow F_t = F_n \cdot f \ ; \ [N] \quad (4)$$

where: $f$ is the coefficient of friction [-]

Figure 6 shown the drive axle the handling equipment [14]. Individual arrows show the acting forces in the braking system and driving direction of the vehicle. The right part of the figure shows detail of the wheel with braking system [13].

3.1. Mathematical analysis of forces during braking

Calculations for the friction disc plates will be listed below. They are needed to express the individual forces and parameters acting (occurring it) during braking [14]. The first step is calculating the middle radius of the friction disc plate (see equation 5), then next to calculate the effective area of the hydraulic piston (see equation 6) [10].

$$r_5 = \frac{2}{3} \left( R_2^3 - R_1^3 \right) \ ; \ [mm] \quad (5)$$

where: $R_1$ is the inside radius of the friction disc plate [mm]; $R_2$ is outside radius of the friction disc plate [mm]; $r_5$ is middle radius of the friction disc plate [mm].

$$S_p = 3.14 \left( R_2^2 - R_1^2 \right) \ ; \ [mm^2] \quad (6)$$

where: $S_p$ is effective area of the hydraulic piston [mm$^2$].

As said, the design of brake is created with push springs. The reason is clear. The brake system must be released after braking. The acting force of springs (see equation 7) must be subtracted from the result of normal force [10]. In short, it is the force that pushes the direction out piston the friction disc plates, when the brake is released.
For proper operation, the brake design must be equipped with three or more springs. This always depends on the design of the brake. In this case, three springs are used (see equation 8) [10].

\[ F_{pr} = F_{z b} \times P_p ; [N] \]  
(8)

where: \( F_{pr} \) is a force at compressed brake [N]; \( P_p \) is the number of push springs [-].

In the hydraulic brake system, the pressure is 1 Mpa, which acts on the friction disc plates [10]. Using the equation (see equation 9), the compressive force on friction disc plates (normal force) is expressed [10].

\[ F_N = (S_p \times p_{pr}) - F_{pr} ; [N] \]  
(9)

where: \( p_{pr} \) is the pressure of the hydraulic oil [MPa].

In the next step, the effective area of the friction disc plate is expressed (see equation 10).

\[ S_L = \pi \times (R_2^2 - R_1^2) ; [\text{mm}^2] \]  
(10)

The value of the tangential force is expressed as a multiple of the normal force (see equation 7) with a coefficient of friction (see equation 11) [10].

\[ F_T = F_N \times \mu ; [N] \]  
(11)

Braking torque is calculated with safety in mind at this point. This step is very important for the next calculation (see equation 12) [10].

\[ M_B = F_T \times r_s \times k; [\text{Nm}] \]  
(12)

where: \( k \) is coefficient of safety [-], \( M_B \) is the braking torque including the safety factor [Nm].

In the penultimate step, it is necessary to determine the number of friction areas needed (see equation 13) [10].

\[ i \geq \frac{M_B}{F_N \times F_s \times r_s} ; [-] \]  
(13)

where: \( i \) is count of the friction disc plate [-].

The result from equation 13 is very important for the next calculation of a number of friction disc plates (see equation 14). The one plate is added because of design reasons.

\[ n = i + 1 ; [-] \]  
(14)

The calculation for the braking torque has already been shown above. In this calculation, the coefficient of friction and the total number of friction disc plates are taken to account (see equation 15). This is a real braking torque for the existing situation at the braking. [10]

\[ M_B = f \times i \times F_N \times r_s ; [\text{Nm}] \]  
(15)

The above calculations must be applied to every friction disc plate. The aim will be to compare the individual calculated values (see table 2). The suitability of using the friction disc plate is determined by its size and maximum permitted braking pressure between the friction disc plates. These requirements were identified at the beginning [10]. The size of the friction plate must be between 80 and 220 mm considering the complete braking system.
Table 2. Calculated values of the friction disc plate.

| Friction disc plate (next only FDP) | FDP-A | FDP-B | FDP-C | FDP-D |
|------------------------------------|-------|-------|-------|-------|
| The median radius of the FDP \( (r_S) \) [mm] | 72.6  | 77.9  | 83.4  | 89.0  |
| Effective area of the hydraulic piston \( (S_P) \) [mm²] | 10 597.5 | 13 188.0 | 18 840.0 | 21 901.5 |
| Normal force \( (F_N) \) [N] | 9855.9 | 12 446.4 | 18 098.4 | 21 159.9 |
| Effective area of friction disc plate \( (S_L) \) [mm²] | 4553 | 9734 | 15 543 | 21 980 |
| Tangential force \( (F_T) \) [N] | 1576.9 | 1991.4 | 2895.7 | 3385.6 |
| Braking torque \( (M_B) \) [N] | 114.5 | 155.2 | 241.5 | 301.4 |
| Required number of friction areas \( (i) \) [-] | 2 | 2 | 2 | 2 |
| Selected number of friction disc plates \( (n) \) [-] | 3 | 3 | 3 | 3 |

The calculation below expresses maximum pressure (see equation 16) between the friction disc plates. Between the friction disc plates mustn’t exceed pressure 2 MPa. The results are in table 3.

\[
P_{\text{max}} = \frac{F_N}{S_L} \quad \rightarrow \quad P_{\text{max}} < P_{\text{Allow}} , \text{[MPa]} \tag{16}
\]

Table 3. Pressure between the friction disc plates.

| The friction disc plate | FDP-A | FDP-B | FDP-C | FDP-D |
|-------------------------|-------|-------|-------|-------|
| Pressure between friction disc plates [MPa] | 2.165 | 1.279 | 1.164 | 0.963 |

As mentioned above, it is important that pressure greater than 2 MPa is no generated in the friction disc plates. For this fact, a graph is created that will present the braking pressure. It can be seen in figure 7, that the value of pressure for FDP-A isn’t satisfactory [10].

![Specific pressure in the friction disc plates](image)

Figure 7. Specific pressure in the friction disc plates.

The size of the friction disc plate FDP-D cannot be applied to the intended inside size of the drive wheel. This is not possible due to size. This dimension only suits the permissible pressure. The most suitable friction disc plate is selected from FDP-C and FDP-B types.
3.2. Simulation the braking system with the friction disc plates
In the previous subsection 3.1 the two most suitable friction disc plates were selected. Subsequently, a braking simulation will be created in Matlab for both friction disc plates.

\[ F_N - F_B - F_G = F_Z \]  

Figure 8. The forces to act on the handling equipment [1-2].

where: \( F_Z \) is a inertia force [N]; \( F_N \) is a normal force [N]; \( F_B \) is a braking force [N]; \( F_G \) is a gravitational force [N]; \( v \) is speed [m.s\(^{-2}\)].

The values of braking forces are calculated and shown in table 2. Since the braking system is always designed in pairs, simulation in the Matlab is set for two wheels. Figure 8 shows the forces acting on the vehicle when braking [1]. The equation of motion (see equation 17) in the vector shape is created according to figure 8 [4]. The forces \( F_G \) and \( F_N \) act in the y-axis direction. However, it is not important for the equation of motion.

\[ x: -F_B = m \cdot \ddot{a} \]  
\[ y: F_N - F_G = 0 \]  

where: \( x, y \) is coordinate system [-]; \( \ddot{a} \) is vector of acceleration [m.s\(^{-2}\)].

From the equation in the x-axis direction (see equation 18) is expressed relationship for acceleration (see equation 20) [1]

\[ a = -\frac{F_B}{m} ; [m.s^{-2}] \]  

where: \( m \) is load [kg].

The acceleration (deceleration) of the handling equipment is constant. In this case, the device slows down using the braking force \( F_B \), so the acceleration is negative. By integrating the acceleration of acceleration (see equation 20), the dependence of velocity in time is expressed (see equation 21). [1]

\[ v(t) = \int a \ dt = \int -\frac{F_B}{m} \ dt = -\frac{F_B}{m} \cdot t + c_1 ; [m.s^{-1}] \]  

where: \( c_1 \) is constant that is determined from the initial conditions; \( a \) is acceleration [m.s\(^{-2}\)] ; \( t \) is time [s].

In the next step, the equation above (see equation 21) is rewritten to the following shape (see equation 22).

\[ v(t) = -\frac{F_B}{m} \cdot t + v_0 ; [m.s^{-1}] \]  

where: \( v_0 \) is initial speed [m.s\(^{-1}\)].
By integrating the velocity from the relationship above (see equation 22), the dependence of the path from time is expressed (see equation 23).

\[ s(t) = \int v(t) \, dt = \int \left( -\frac{F_B}{m} \cdot t + v_0 \right) dt = -\frac{F_B}{2m} \cdot t^2 + v_0 \cdot t + c_2; \quad [\text{m}] \]  

(23)

where: \( c_2 \) is constant that is determined from initial conditions; \( s \) is route [m].

Braking distance (see equation 25) and stopping time (see equation 24) are check calculation and listed in table 4 below.

\[ t_b = \frac{m \cdot v}{F_B}; \quad [\text{s}] \]  

(24)

\[ s_b = \frac{m \cdot v^2}{2 \cdot F_B}; \quad [\text{m}] \]  

(25)

Table 4. Parameters of the braking distance, and the braking time.

| Type of the friction disc plate | FDP-B | FDP-C |
|--------------------------------|-------|-------|
| Time braking [s]               | 3     | 2.1   |
| Braking distance [m]           | 7.53  | 5.18  |

On the graph (see figure 9), it can be seen that from speed 5 m.s\(^{-1}\) the vehicle is stopped in 2.1 seconds (see figure 9). The device passes the distance of 5.18 meters (see figure 9). This graph belongs to friction disc plate FDP-C. The red point in the graph shows the moment when the device stops.

Similarly to the previous example, a simulation for FDP-B is created. The braking time is 3.01 seconds (see figure 10). In this case, the device will pass 7.53 meters (see figure 10). The red point in the graph shows the moment when the device stops. The results in Matlab were verified by calculations (see equation 24 – 26) and are listed in table 4.
The braking distance values have been satisfactory for the type of friction disc plate C (FDP-C). However, additional calculations are required for the final design. The brake design is a complex problem that involves many technical and operating factors. Such as heat dissipation during braking, etc. It is also necessary that the wheel is equipped with a sensor to monitor the speed during braking. The braking principle is similar to a car with an automatic braking system (ABS).

4. Discussion
By changing the mean radius of the friction plate surface, the size of the normal force changes. When the normal force changes, the tangential force and moment of the brake also change. As a result, the change in the size of the friction disc plate affects the distance of the braking distance of the AGV device [13]. Figure 4 above, shows the drive wheel and more types of friction discs plates (figure 11 is illustrative only).
5. Conclusion
The aim of the work was to optimize the size of the friction disc plate for the drive wheel of the handling equipment. While the size of the brake system had to insert to the inside space of the original wheel of existing handling equipment. Of the four friction disc plates considered, two sizes (FDP-B and FDP-C) were recommended. The other two (FDP-A and FDP-D) did not fit the size and allowable pressure during braking. The simulation in the software Matlab is setting for two friction disc plates. The results were compared, and the most suitable friction disc plates were chosen. In this case, the friction disc plate FDP-C is accepted. At this setting, the equipment handling stops from the maximum speed of 5 m.s\(^{-1}\) to zero in 2.1 seconds. The distance is 5.18 meters. The weight considered was 1200 kg. It must be said, that more effort is needed for the final design (heat dissipation during braking etc.) The braking system is a complex problem that involves many technical and operational factors.

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