Numerical Methods of Validation of Valve Systems of Railway Hydraulic Dampers

Piotr CZOP[1], Mateusz WOŹNIAK[2]

Summary
This work presents a method to accelerate the process of configuration and calibration of shim washer valve systems which are used in railway hydraulic dampers. The advantage of the method is possibility to reduce the shim fatigue failure risk during train operation. This failure causes a sudden drop of a damping force influencing negatively safety of passengers. Fatigue verification is an important phase of configuration and calibration process of a valve system. The verification is conducted based on the detailed simulation model of the valve system established (formulated) with a finite element method. The mesh density was determined using a sensitivity analysis regarding the number of finite elements. The force and flow balance were used to formulate the system model. The model was calibrated based on experimental measurements conducted on the servo-hydraulic tester. The verification process allows to determine the critical von Misses stress level in elastic components of a valve system. The work showed the feasibility of accelerated configuration and calibration process of shim washer valve systems modeling their mechanical and hydraulic properties.

Keywords: hydraulic damper, damping force, disc valve, numerical model, fatigue strength

1. Introduction

Model – based design (MBD) approach is considered to be productive in case of advanced engineering projects which require analytical insight and multi variant design analyses before a physical prototype is built in the workshop. MBD approach leads to shorten development cycle, lower prototype and validation costs. High speed bogies are one of those applications where MBD approach shows its advantages and effectiveness replacing as far as possible experimental tests with numerical modeling [13].

Suspension systems of high speed trains are strongly subjected to railway roughness which excites structural vibrations. The vibrations are passed from railways to the bogie and further to the train car body and their components. In turn, they negatively affect the train stability and passengers' comfort [6]. New methods, to reduce high-speed vibrations such as active suspension modules, are continuously developed; nevertheless passive systems are still commonly in use due to their standard design, better reliability and lower costs. The key suspension component is a hydraulic damper [9] which significantly influences the passenger comfort and train stability through chosen damping characteristic of a hydraulic damper.

Hydraulic damper design process considers a few criteria, e.g. specified damping forces at given velocities, weight, reliability, lifetime, and physical dimensions (diameter, fixation point distance). From the listed criteria, the most critical are lifetime and reliability ones. The reliability of a hydraulic damper is strictly related to the embedded valve system. Modern hydraulic dampers are usually equipped with shim washer valves [6]. The stable damping forces along damper lifetime and high fatigue reliability ensures the correct valve shim washers configuration.

The paper objective is to introduce a numerical method to select an optimal shim washer setting starting with a numerical computation instead a workshop activities and experimental testing. Diagram of the method shown in Fig. 1.

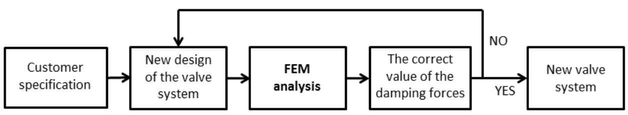


Fig. 1. Diagram for validation of valve systems using FEM analysis [Source: Self study]

The application of a finite element model (FEM) allows to validate shim washer settings without expensive experimental tests and on the other hand to predict the damper force. In turn, a new hydraulic damper project is launched into the market significantly faster and at lower costs. The method proposed

[1] Dr hab. inż.; Akademia Górniczo-Hutnicza, Kraków.
[2] Mgr inż.; Tenneco Automotive Eastern Europe Sp. z o.o., Gliwice; e-mail: wozniak753@gmail.com.

by authors is applicable to all hydraulic dampers equipped with shim washer valve systems. The method was demonstrated based on calculations performer for a double-tube and two-direction flow [6, 16], which is a typical design commonly use in passenger multi-section trains, coach cars or locomotives.

2. Working Principle of a Hydraulic Damper with a Shim Washer Based Valve System

The type of considered hydraulic damper, Fig. 2, is a double-tube type consisting of three chambers, two of variable volumes (rebound and compression chambers) and one of fixed volume (reserve chambers). The chambers are connected by flow restrictions (orifices and valves). The piston is kinematically forced to move inside the compression and rebound chambers, which are formed as a cylinder, a pressure differential is built across the piston and forces liquid to flow through restrictions located in the piston, cylinder-end assembly, and from the rebound chamber to the reserve chamber [2]. The action of the piston transfers the liquid surrounding the rod to the reserve chamber. The reserve chamber is partially filled with fluid (oil) and partially filled with gas (nitrogen). The combined volume of the compression and rebound chambers during piston movement changes by an amount equivalent to the inserted, or withdrawn, rod volume. The oil is transferred from the reserve chamber to the compression chamber through the cylinder – end assembly located at the bottom of the compression chamber. Two types of valves, intake valves and control valves, are used in a double-tube hydraulic damper to enable liquid flow from the compression to the rebound chamber and from the rebound to the compression chamber. The valve systems considered in this work are indicated in Fig. 2. with the labels 1 and 4.

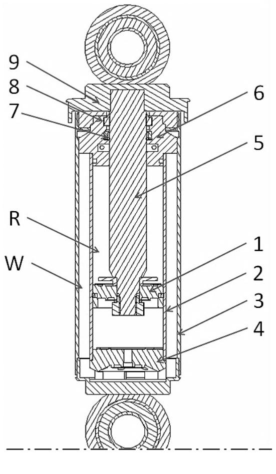


Fig. 2. Diagram of a double-tube hydraulic hydraulic damper;

Indicated: 1) piston and valves, 2) pressure tube, 3) reserve tube, 4) foot valves, 5) rod, 6) head, 7) rod guide, 8) oil and dirt seal, 9) cover and mounting;

W – reserve chamber, R – working chamber [Based on 6]

Two main operation phases are feasible in the hydraulic damper, namely rebound and compression phase, which activate different hydraulic flow paths in valve systems depending on the velocity of a piston-rod assembly. The piston-rod assembly moving in the base valve direction forces the flow through the piston valve as it is depicted in the schematic view in Fig. 3. While the piston moves slowly down, the pressure across the shim washer stack of the valve no. 2. (Fig. 3) increases slightly and in turn, the oil do not pass to the reserve chamber. The damping force is proportional to the square function of the piston-rod velocity according to the characteristic – M (Fig. 4).

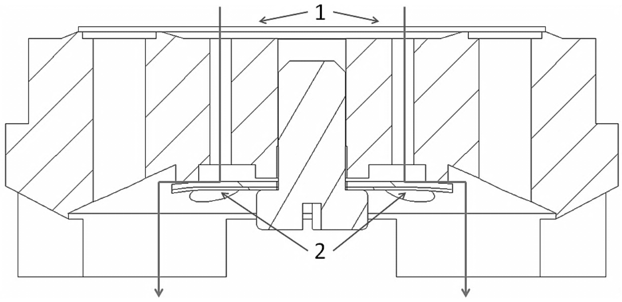


Fig. 3. Fluid flow path through the foot valve during the compression phase [Based on 6]

The working medium is simultaneously displaced through the calibrated bleeds to the chamber above the piston and causes the full opening of the piston valve indicated with the label no. 3 (Fig. 5). High speed train drive causes higher piston-rod velocities due to railway imperfections. The pressure below the piston increases and pushes to open the shim washer stack in the valve no. 2 (Fig. 3) till full opening mode. The velocity at which the shim washer stack opening is so-called critical velocity. The valve no. 2 allows to limit the increase of the damping force changing the curve into more flat section between points M – N as indicated in Fig. 4. The valve no. 1 (Fig. 3) is remained closed during the compression cycle due to back oil pressure.

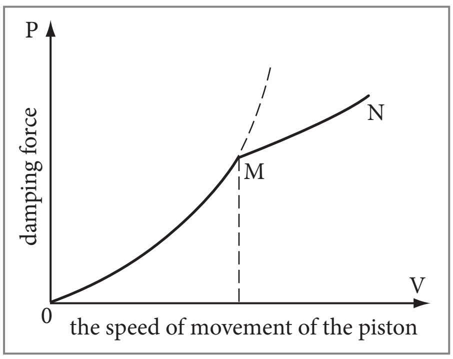


Fig. 4. Theoretical characteristics of the hydraulic damper during the compression movement [Based on 6]

During the phase of rebound piston moves upwards and at the same time it comes closer to the head of the damper. Working fluid will flow through the valve according to the diagram in Fig. 5. Slow movement of the piston makes that the oil pressure exerts on a stack of washers of the valve 4 (Fig. 5) and it is too small to cause its opening. The damping forces increase in accordance with the course of the curve O – M (Fig. 6). At the time when the vibration amplitude is being increased, piston begins to move with a higher speed. This results in opening a pressure valve 4 and oil displacement from upper space till lower space by calibrated openings in the piston valve. This phase is presented on a curve M – N (Fig. 6).

A small oil volume flows simultaneously through rod-guide clearance to ensure lubrication between the piston rod-assembly and the rod-guide. The oil passes through the deaeration holes in the rod-guide to the reserve chamber. In turn, the under pressure creates under the piston-rod volume and then the oil passes from the reserve chamber through the base valve. The valve no. 3 (Fig. 5) remains closed during this cycle.

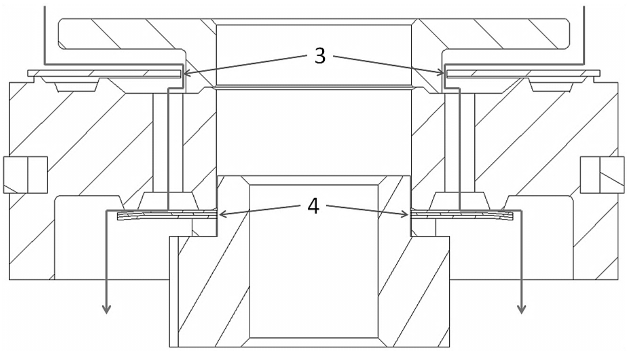


Fig. 5. Fluid flow path through the piston valve during the rebound phase [Based on 6]

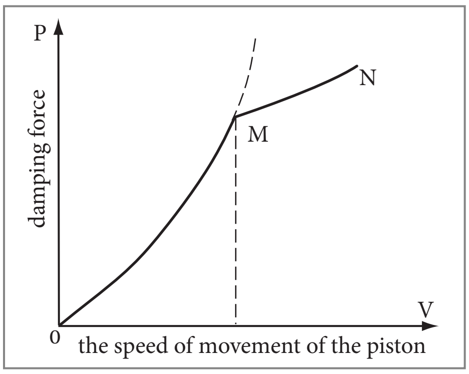


Fig. 6. Theoretical characteristics of the hydraulic damper during the rebound movement [Based on 6]

Valve system design and configuration determines the characteristic of damping force vs. rod velocity and damping force vs. rod displacement which are specified during the train bogie design process. The hydraulic damper characteristics are result of the train bogie design process. On the other hand, the major valve system durability contributor is the highest thickness of a valve shim washer. A valve system requires an adjustment process to achieve the damping forces at specified velocities within the given tolerance band (typically 15%). The adjustment process is mostly manually conducted by a trained operator in the prototype workshop using customized shim washers of different diameters and thicknesses. The objective is to meet the customer damping force while minimize the stress level trying to reduce the shim washer thickness manipulating the number of shim washers and their diameters. A typical damping force calibration process consists of the following steps:

- rebuilding the piston valve (change in number of shim washers, their diameter, or thickness);
- rebuilding the base valve (change in number of shim washers, their diameter, or thickness);
- changing the oil volume in the damper if an aeration effect occurred.

The aeration phenomenon results from releasing pressurizing gas dissolved in the working medium in the form of gas bubbles to form an emulsion. Such emulsion is not homogeneous and has a certain life – span during which, at least partial, re-adsorption of bubbles into the working liquid occurs. Bubble formation and bubble re-adsorption is caused by local changes of the pressure depending on the valve system characteristic. The damping force calibration is performed on the servo-hydraulic testers equipped with required force and displacement sensors. However the obtained valve shim washers' configurations require also to meet durability objectives, e.g. number of rebound-compression cycles until the shim washer failure. Hydraulic damper durability is validated servo-hydraulic tester using dedicated accelerated life-time programs with random sequences to reproduce the load to which the hydraulic damper is subjected [3].

Experimental validation requires long time testing procedures (days or weeks) which in turn are expensive, nevertheless the number of valve configuration is very limited. However the major disadvantage of such validation approach is in general lack of any constructive feedback from the tests which allows to improve the valve shim washers settings. Experimental validation does not provide the answer to the question, which shim washer failed first and in which valve (Fig. 7).

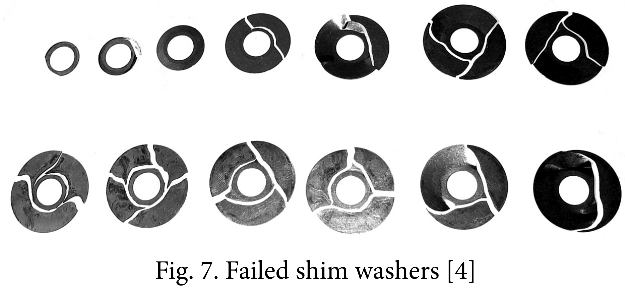


Fig. 7. Failed shim washers [4]

A significant improvement in hydraulic damper validation process involves a model-based approach which allows to obtain the most durable shim washer settings regarding the minimum stress optimum criterion. A model-based approach requires to formulate two models, namely the system model of entire hydraulic damper and detail numerical 2/3 D model of the shim washers stack including the boundary conditions. The work [15] presents a model – based approach to understand the hydraulic damper operation at the electrical locomotive. The system model provides damping forces using the following formula [16]:

$$F_d = p_{reb} \cdot A_{reb} + p_0 \cdot A_{rod} - p_{com} \cdot A_{com} \qquad (1)$$

where:

- F_d – damping force generated by the hydraulic damper,
- A_{rod}, A_{com}, A_{reb} – surfaces of the piston (rod, rebound, compression) [m^2],
- p_{com}, p_{reb} – the pressure in the compression and rebound chambers [Pa],
- p_0 – atmospheric pressure, p_0 = 1e5 [Pa].

The detail numerical valve model allows to obtain the hydraulic characteristic (differential pressure across the valve vs. volumetric oil flow) and durability characteristic (averaged / max stress level in a stack of shim washers vs. differential pressure across the valve). The hydraulic characteristic is used in a system model while the durability characteristic allows to verify the critical stress level for the given load of a hydraulic damper.

3. Development of a Valve and Damper Model

Within recent years the subject of durability of valve systems used in hydraulic dampers has increased in importance for unquestionable growth in quality demands in the automotive sector caused the warranty period requirements to be significantly prolonged with a clear tendency towards lowering the hydraulic dampers fatigue failures. The early durability prediction methods involved Roark's stress and strain formulas stated in the form of the pre-derived and parameterized equations (for historical details cf. [17]). Fundamentals of the disc stack prediction and interpretation of the fatigue wear process were deduced from theoretical studies [1, 2, 5, 10, 11, 12, 14] and adapted to prototypical and manufactured valve systems [7]. During the last two decades, existing approaches to valve system modeling and measurement data interpretation are being continually improved [11] and the finite element modeling approach is being implemented to support development of more advanced models.

A proposed method was demonstrated based on a typical mid-size railway hydraulic damper. The durability characteristics were obtained for the piston and base valves. This section describes however only the piston valve model formulation process (Fig. 8).

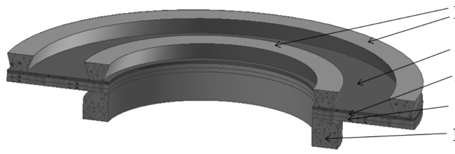


Fig. 8. CAD model of the piston valve shim washers stack and supporting components (piston base and back-up washer) [Source: Self study]

The shim washers stack configuration is presented in Table 1 and Table 2, respectively for piston and base vale. The shim washer mechanical properties are presented in Table 3.

Table 1

Piston valve shim washers stack configuration

Item	Component name	Shim washer dimensions (outer diameter x inner diameter x thickness)
1	Elastic Shim washer 0,2	Ø32 x Ø16 x 0,2
2	Elastic Shim washer 0,3	Ø32 x Ø16 x 0,3
3	Elastic Shim washer 0,3	Ø32 x Ø16 x 0,3
4	Disk washer	Ø20 x Ø16 x 2 | r = 0,7

Source: Self study.

Table 2

Base valve shim washers stack configuration

Item	Component name	Shim washer dimensions (outer diameter x inner diameter x thickness)
1	Elastic Shim washer 0,2	Ø 22 x Ø 6,2 x 0,2
2	Elastic Shim washer 0,2	Ø 22 x Ø 6,2 x 0,2
3	Disk washer	Ø 16 x Ø 6,2 x 2 | r = 0,7

Source: Self study.

Table 3

Mechanical properties of shim washers| used in valve systems

Parameter	Value	Unit
Young's modulus	210000	MPa
Poisson ratio	0,3	[-]
Yield strength (Remin.)	250	MPa
Tensile strength Rm	600–950	MPa
Hardness max.	215	HB

Source: Self study.

Simulation was conducted with the use of finite element methods in ANSYS Workbench 12.1. The piston component was modeled as ta non – deformable part, while the shim washers as elastic part with properties listed in Table 3. The large – displacement solver was involved to increase nonlinear effects occur at high pressure load. The contacts among particular components were defined.

The sensitivity analysis was performed to determine the best mesh density. The Quadratic Tetrahedron (Mechanical APDL Name: Mesh200) finite elements were used in simulation. The following boundary conditions were applied (Fig. 9):

- axisymmetrically fixation of the shim washers removing the rotational and vertical movement (Cylindrical Support);
- pre-load force of the threaded nut 120 N (on disk washer);
- equivalent oil pressure load of 5 MPa, increasing linearly with the span of 0,5 MPa.

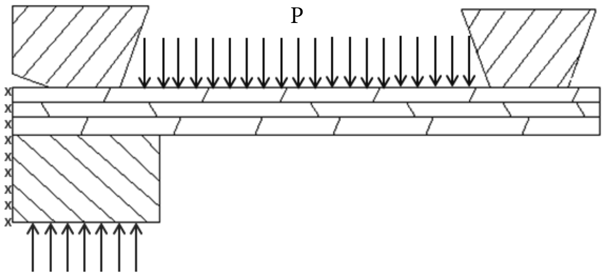


Fig. 9. The boundary conditions assumed in the FE model [Source: Self study]

There are two steps essential to loading and unloading a shim washer stack in the model:

- applying preload;
- applying the loading pressure.

During the preload step, the rod nut (rigid part) is moved down, while the piston hub (rigid) is held fixed. The nut moves until the clamping force is equal to the preload force resulted from the thread reaction (120 N). In the second step, the oil pressure equivalent load is applied to the shim washers stack during the rebound cycle.

4. Sensitivity Analysis of Mesh Density

The sensitivity analyses were conducted for shim washers stacks configuration presented in Table 1 and 2. The results are presented in Table 4 regarding the number of used finite elements.

Table 4

Mesh size of the analyzed shim washer stacks

FEM element size [mm]	The number of elements FEM	Number of nodes
The piston valve		
1,2	14885	29872
1,1	17224	35208
1	21177	42637
The foot valve		
0,5	12339	24029
0,4	18537	36581
0,3	28120	55980

Source: Self study.

The results are presented in Fig. 10–12 in a form of stress or displacement (clearance) vs. the applied equivalent oil pressure load. The displacement is obtained in the cross section of a shim washer above the supporting piston edge.

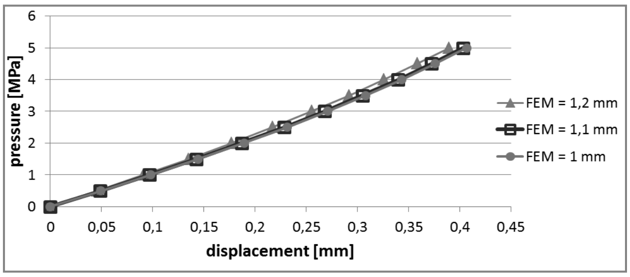


Fig. 10. The displacement characteristic of shim washers at the support location under equivalent oil pressure load regarding the mesh density [Source: Self study]

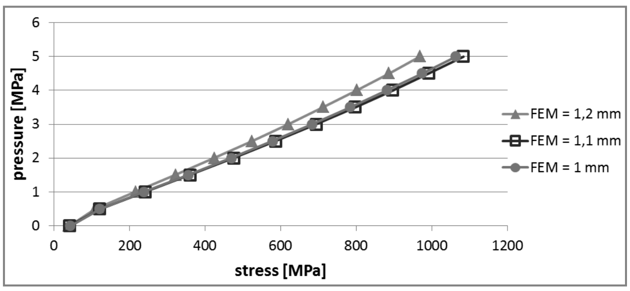


Fig. 11. The stress characteristic of shim washers at the support location under equivalent oil pressure load regarding the mesh density [Source: Self study]

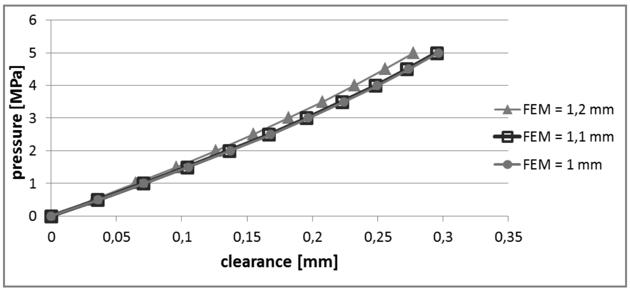


Fig.12. The clearance characteristic of shim washers at the support location under equivalent oil pressure load regarding the mesh density [Source: Self study]

The same analysis was conducted for base valve system and the results were summarized in Table 5.

Table 5

The critical values of selected parameters of the analyzed valve model

FEM element size [mm]	Displecement [mm]	Stress [MPa]	Clearance [mm]
The piston valve			
1,2	0,38939	968,42	0,27773
1,1	0,40321	1083,4	0,29524
1	0,40684	1063,7	0,29667
The foot valve			
0,5	0,12841	758,09	0,083815
0,4	0,13819	905,58	0,092086
0,3	0,14368	1092,2	0,09621

Source: Self study.

Maximal deflection as expected occurred at the edge of shim washers (Fig. 13). The deflection value was significantly differed model by model.

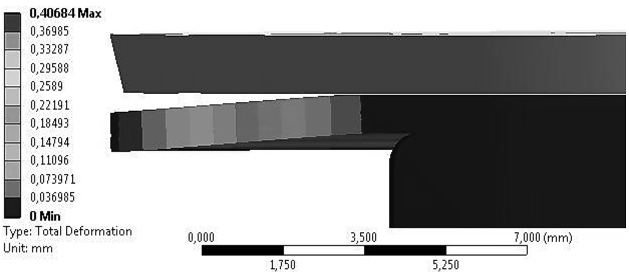


Fig. 13. Total deformation map obtained for shim washers in the piston valve. The size of a finite element FEM=1 mm [Source: Self study]

The maximal stress level occurred in the proximity of the contact location between a shim washer and back-up washer (Fig. 14). The stress values are moderately high compared to hydraulic damper analyzed in a similar work [8], where stress levels from 1600 MPa up to 1800 MPa were reported.

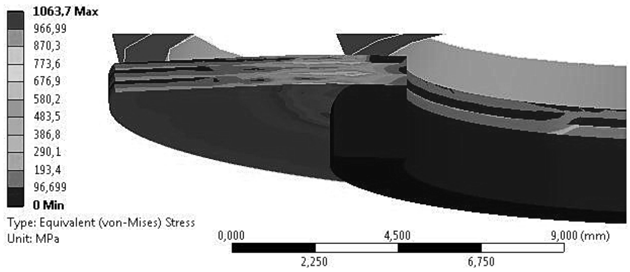


Fig. 14. Equivalent stress map obtained for shim washers in the piston valve. The size of a finite element FEM=1 mm [Source: Self study]

5. Validation of Valve and Damper Models

The numerical analysis was validated with the use of the system model and experimental tests. The valve opening vs. equivalent oil pressure load characteristics were obtained using approximation formulas as presented in Fig. 15.

A

B

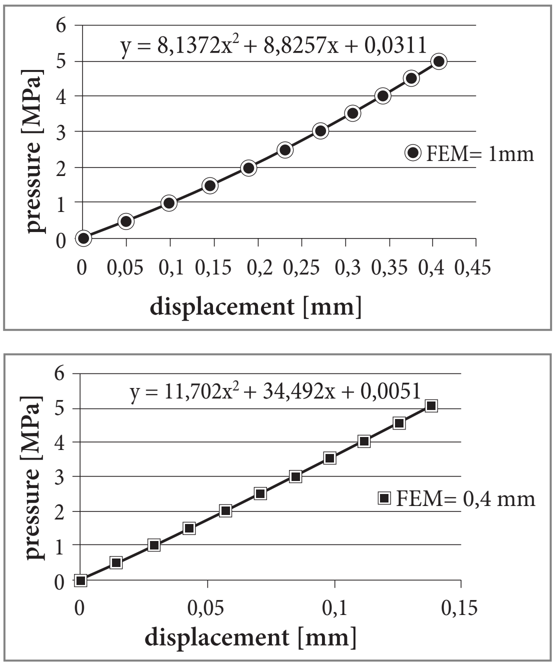


Fig. 15. Approximating curves for analysis: A – piston valve, B – foot valve [Source: Self study]

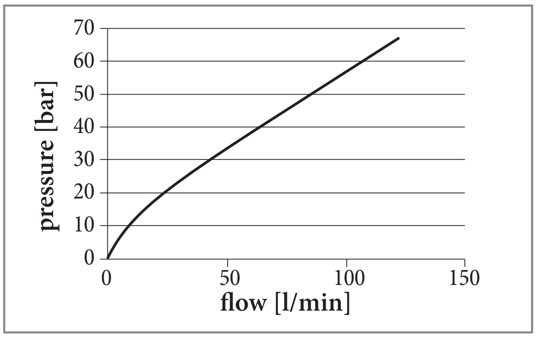
The valve flow vs. equivalent oil pressure load characteristics were obtained in the second step as presented in Fig. 16. The volumetric flow rate was determined using the formula [16]:

$$q = C_d \cdot (\pi \cdot d \cdot x) \cdot \left(\frac{2p}{\rho} \right)^{\frac{1}{2}} \quad (2)$$

Where:

- q – flow rate through valve [m^3/s],
- p – pressure drop across valve assembly [Pa],
- C_d – flow (discharge) coefficient for valve = 0,35,
- ρ – fluid density = 850 [kg/m^3],
- x – valve disk lift [m],
- d – the outflow valve diameter = 0,029 [m] (piston valve) and 0,020 [m] (foot valve).

A



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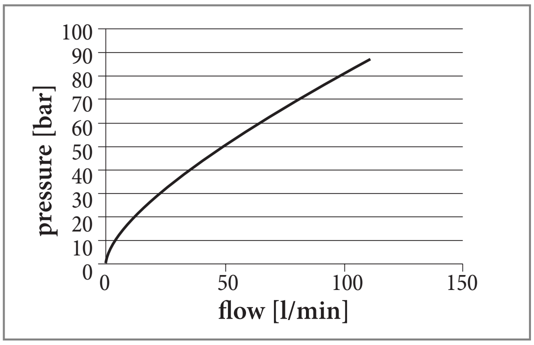

Fig. 16. Flow graph: A – piston valve, B – foot valve [Source: Self study]

The obtained valve characteristics allowed to compute damping forces based on the formula (1) representing the damper system model, respectively for rebound and compression stroke. Model parameters were listed in Table 6.

Table 6

The parameters used to determine the damping forces of the hydraulic damper

Parameter	Value	Unit
Rebound area	0,001433	m2
Compression area	0,001963	m2
Rod area	0,000531	m2
Stroke	25,0	mm
Velocity	0,20	m/s

Source: Self study.

The damping force is presented for the selected velocity v = 0.2 m/s as a diagram force vs. Piston rod displacement (Fig. 17).

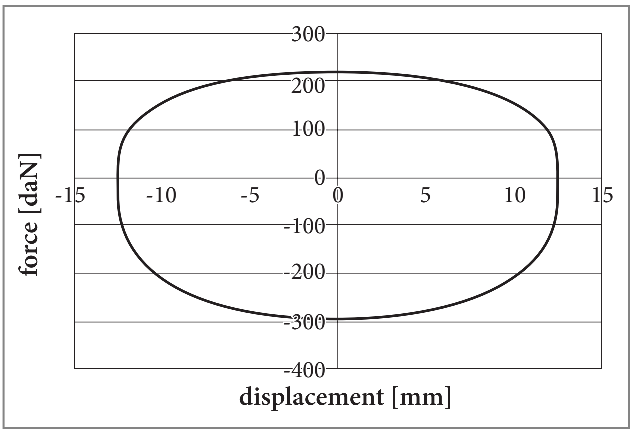

Fig. 17. Damping forces graph for the entire system [Source: Self study]

The results obtained with the numerical model were compared with the experimental results obtained with the use of servo-hydraulic MSP25 IST tester (Table 7). The relative error between simulated and measured force – displacement curve was calculated as [16]:

$$E_r = \frac{\sum F_d - \sum F_c}{\sum F_d} \cdot 100\% \tag{3}$$

where:

- F_d – the expected value of the force (with experimental measurements) [daN],
- F_c – the calculated value of force (measured number) [daN],
- E_r – relative error [%].The flow coefficient in formula (2) was additionally adjusted to minimize the relative error.

Table 7

Comparison of numerical analysis with the experimental analysis

Damping force	Numerical Analysis [daN]	Experience Analysis [daN]	Error [%]
Rebound	228	300	24
Compression	298	320	6,8

Source: Self study.

The modes shows high accuracy regarding the compression stroke and acceptable accuracy regarding the rebound stroke.

6. Discussion

The paper aimed at demonstration of model-based approach towards shorten the work in a prototype shop during new hydraulic damper adjustment and calibration to meet the customer damping force and durability requirements. The paper is focused on railway hydraulic damper equipped with shim washer based valve systems.

Fundamental outcome of this work is the detail finite element model of a valve system with the use of ANSYS workbench software. The model accuracy is determined by the number and type of finite elements. ANSYS allowed to control the fit of finite elements thanks to a split of finite elements into mesh deformation groups in relation to the ideal finite element shape and obtained „Element Quality" parameter for each group. The analyses indicate the optimal ones, i.e. Quadratic Tetrahedron.

The sensitivity analyses were conducted to show how the valve model accuracy depending on the number of finite elements. While the finite elements number increases, then the model accuracy increases accordingly. Nevertheless, simulation time significantly increases too. It was shown that the optimal size of a finite element is 1 mm and 0.4 mm, respectively for piston and base valve.

The simulation results were obtained in a form of durability valve characteristics where the stress level was similar and do not exceeded 1000 MPa for maximal specified working velocity $v = 0.2$ m/s. These results have enough safety life-time margin. It is noticeable that the highest stress value occurs at contract between the diameter of supporting shim washer and elastic shim washers. It confirms that the diameter of a shim washer has significant impact on potential shim washer failures (e.g. crack). The maximal shim washer deflection was equal to 0.4 mm for the piston valve and 0.14 for the base valve. The investigated valve systems differed by the number of shim washers, their thickness and diameter.

The hydraulic damper system model was calibrated based on the experimental measurements. Hydraulic damping forces of 228 daN and 298 daN were predicted by the system model at rod velocity $v = 0.2$ m/s, respectively for the rebound and compression stroke. The model prediction differs from the measurements in term of relative error of 24% and 7%, respectively for the rebound and compression stroke. The prediction results inconsistence for rebound and compression stroke can be caused by difficulties in reproduction of boundary conditions for each stroke.

The average simulation time for the high density mesh (FEM element size = 0,3 mm) was not higher than 60 min. The valve model provides also a stress distribution map which allows to identify the critical stress at early design stage. The model-based approach allows to analyze many shim washer settings in a short time without necessity to conduct experimental works at prototype workshop. In turn, the simulation process improves understanding of fatigue root causes and is less expensive and faster compare to conventional workshop-based approach for new hydraulic damper designs or configurations.

The work requires further improvement towards a rapid prototyping tool which becomes an engineering tool.

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Numeryczne metody walidacji systemów zaworowych amortyzatorów kolejowych

Streszczenie
W artykule zaprezentowano metodę umożliwiającą skrócenie czasu wymaganego w procesie konfiguracji oraz kalibracji podkładowych (dyskowych) systemów zaworowych stosowanych w tłumikach hydraulicznych przeznaczonych do pojazdów szynowych. Zasadniczą zaletą metody jest możliwość zmniejszenia ryzyka uszkodzenia zmęczeniowego zaworu w trakcie eksploatacji, wiążącego się z nagłym spadkiem siły tłumiącej w układzie zawieszenia wózka pojazdu szynowego, co ma bezpośredni wpływ na bezpieczeństwo pasażerów. Zasadniczą operacją w procesie konfiguracji oraz kalibracji jest weryfikacja wytrzymałościowa systemów zaworowych. Weryfikacja jest prowadzona na podstawie szczegółowego modelu symulacyjnego systemu zaworowego wykonanego metodą elementów skończonych. Dobór siatki modelu przeprowadzono na podstawie analizy wrażliwości wyników modelu na liczbę użytych elementów dyskretnych. W celu symulacji działania kompletnego tłumika hydraulicznego, sformułowano model systemowy na podstawie bilansu sił oraz przepływów wewnętrznych tłumika. Model systemowy skalibrowano na podstawie pomiarów eksperymentalnych przeprowadzonych na stanowisku serwo-hydraulicznym. Weryfikacja wytrzymałościowa umożliwiła wyznaczenie granicznych dopuszczalnych naprężeń złożonych w podatnych elementach systemu zaworowego. Artykuł wykazał możliwość wykonania przyśpieszonej konfiguracji oraz kalibracji podkładowych (dyskowych) systemów zaworowych uwzględniając ich weryfikację niezawodnościową przez modelowanie ich właściwości mechanicznych i hydraulicznych.

Słowa kluczowe: tłumik hydrauliczny, siły tłumienia, zawór dyskowy, model numeryczny, wytrzymałość zmęczeniowa

Цифровые методы проверки (контроля) систем клапанов железнодорожных амортизаторов

Резюме
В работе представлен метод который дает возможность сокращения необходимого времени в процессе конфигурации а также калибровки подкладных систем (дисковых) клапанов применяемых в гидравлических глушителях (демпферах) предназначенных для железнодорожных машин. Главным преимуществом метода является возможность уменьшени риска усталостного повреждения клапана во время эксплуатации в связи с неожиданным уменьшением силы глушения в рамной системе тележки железнодорожной машины, что имеет непосредственное влияние на безопасность пассажиров. Принципиальной операцией в процессе конфигурации и калибровки является проверка сопротивления клапанных систем. Проверка осуществляется на основании точной копии симуляционной клапанной системы изготовленной в соответствии с методом законченных элементов. Выбор сетки копии был осуществлен на основании анализа чувствительности результатов копии по отношении к количеству используемых мягких (деликатных) элементов. В целях осуществления полного цикла работы гидравлического глушителя была сформулирована системная копия на основании итога сил а также внутренних потоков глушителя. Системная копия была скалибрирована на основании экспериментальных измерений произведенных на серво-гидравлическом стенде. Проверка прочности (сопротивления) дает возможность определить гранично допустимые составные напряжения в поддатливых элементах клапанной системы. Работа показала возможности ускоренной конфигурации и калибровки дисковых клапанных систем с проверкой на надежность благодаря моделированию механических и гидравлических свойств.

Ключевые слова: гидравлический демпфер (глушитель), сила глушения (демпфирования), дисковой клапан, цифровая копия (модель), усталостная прочность