THE ARCHIVE OF MECHANICAL ENGINEERING

VOL. LIV 2007 Number 2

DOI: 10.24425/ame.2007.131547

Key words: Adjustable; semi-active; damping; hydro-pneumatic cab suspension; proportional valve

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AN ADJUSTABLE DAMPER FOR CAB SUSPENSION

The aim of this study was to design and test an adjustable hydro-pneumatic damper for cab suspension. The goal was to make a simple and cheap solution for a damper, which is intended to be placed between the hydraulic cylinder and accumulator. Damping behaviour of different terrain types had to be taken into consideration. Terrain type varies from field to road driving and damping should react rapidly to varying conditions.

In this study, the semi-active damper has been built with a hydraulic direct acting cartridge type 2/2-way proportional flow control valve. Flow-pressure curves and dynamic tests were carried out in the laboratory. The dynamic test with forced vibration focused on stability in damping frequencies and step response between different states. Also, total damping force was measured in different damping states and the proportional valve's precise step responses and stability were investigated in a closed hydraulic system.

As a result, this research gave a lot of new information about the proportional valve's applicability to work as a semi-active damper and information about damping behaviour. Research showed that a proportional valve can work in a cab suspension damper as well as a multi-fixed orifice damper. Bi-directional flow in the proportional valve was found to remain stable in cab suspension working conditions. The proportional valve also has the ability to work as a continuous state damper, which could lead to better damping results with the appropriate control system.

Introduction

Vibration exists in many forms and one cannot avoid being exposed to it. One can estimate vehicles' driving conditions and make decisions on damping with that information. One always concludes in passive damping to compromise for different terrain types, and the result is an adequate damper for different situations. The problems increase if the vehicle's mass or terrain

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type varies considerably. Then, it is hard to get good damping for every situation with the traditional passive damper.

Semi-active dampers, where damping can be changed for each driving situation, have been produced to solve the problem. Then one can concentrate on design without compromising the optimization of damping for different terrain types. However, the main focus in semi-active damping is to get a flexible system, where damping control is performed in real time by sensors during driving. The simplest way to control damping is to change it with the driver's manual control, but a better way is to let sensor information to control damping, so that the driver can concentrate on the vehicle handling. Advanced control can also improve damping at resonance frequencies.

The goal of the work was to design and test a hydro-pneumatic damper solution for off-road vehicles' cab suspension. The damper should fulfil the demands of adjustability, simplicity and economy. The damper should have at least three damping states and shifting of damping states should be as fast as possible. The damper should be used as a multi-state or continuous-state damper.

1. Damper design

1.1. Hydro-pneumatic damper

The hydro-pneumatic suspension is composed of an accumulator and cylinder (Fig. 1.). The accumulator has a pre-charge pressure, and it works like a spring in a vibrating system. The cylinder is the actuator, which is connected between the cabin and the body. The cylinder is connected hydraulically to the accumulator, and damping is usually done by choking fluid flow to the accumulator. Another possibility is to choke the differential flow in a single rod cylinder. Differential flow is greater in standard hydraulic cylinders, and therefore disadvantageous for small damper design. In custom cylinders, differential flow can be made considerably smaller, and then it could be usable for damper design.

The starting point of this study was the need for a small, fast, adjustable, inexpensive and simple damper solution for hydro-pneumatic cab suspension. Some semi-active dampers for damping optimization in different driving conditions have been developed [3–5, 11]. Semi-active damper is needed when vehicle mass and terrain type varies [9]. The damping ratio has to be adjustable from stable to comfort driving to reduce the driver's whole-body vibration.

A single fixed orifice works fine with a passive damper, but in a semi-active system the orifice has to be selectable or adjustable. One solution would be a multi-fixed orifice damper where a rotating element changes orifices or hydraulic solenoid valves change the flow channel. With a rotating fixed orifice element, it is hard to get a solution small enough for fast response. Also, the rotating actuator increases the size of a damper. Solenoid valves are very fast and one 3/3-way valve can produce three damping states. Increasing damping states needs more valves and therefore more space. The proportional valve can produce multiple states with a single valve and corresponds better to the requirements than other damping solutions, although proportional valves are not so commonly used, and therefore a suitable valve is hard to find. Another thing that complicated valve selection is pressure compensation, which is nowadays commonly used in proportional flow control valves, but it is a disadvantage in a damper. Pressure compensation limits the fluid flow to a certain level, so the damper operating range must be within these limits.

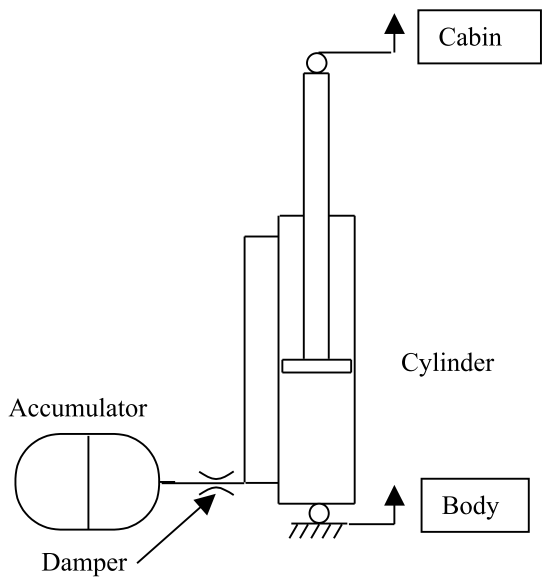

Fig. 1. Quarter model of hydro-pneumatic cab suspension

1.2. Solution for damper

It was decided to make the damper with a proportional hydraulic valve. The selected proportional cartridge valve was a little bit of a compromise. A proportional directional valve would have had a better flow-pressure curve and response time, but it would take a fairly large space. The cartridge valve body can be attached directly between the cylinder and accumulator.

Therefore, it is a very compact solution, and its performance is almost as good as in directional valves.

The Bosch Rexroth KKDSR1 normally open direct operated 2/2-way cartridge proportional valve was the best available compromise, which could fulfil all demands [1]. There is pressure compensation in another direction but it starts from a 70 bar pressure difference. It is high enough for the cab suspension system, where cabin mass is around 1,000 kg, and cylinder rod diameter is 18 mm. The accumulator has a 40-bar pre-charge pressure and nominal volume of 0.16 dm^3. With these specifications, the static pressure is 86 bar in the system, and thus pressure compensation is only a small problem when the pressure difference exceeds 70 bar in 1-2 directions. Then, damping increases rapidly to over critical even in low damping states. Valve speed is 50/40 ms (open/closed), so it is a bit slow when one compares it to the fastest solenoid valve response times. The valve seems to be somewhat asymmetric in flow directions, but the manufacturer's flow-pressure curves are so close to each other at low pressures that it is hard to find any precise asymmetries before measurements.

It is important to observe flow forces in damper design when the valve is used to control damping. There is both stable and unstable flow in a bi-directional system [6], [7]. Stable flow tends to close the orifice, while unstable flow tends to open it. There is the possibility that the valve spool can vibrate, because these forces influence in opposite directions in a bi-directional flow system. Pressure differences in cab suspension are relatively small compared to the valves' maximum operating pressure, and accordingly one can hypothesize that the valve should be stable in cab suspension.

1.3. Damping states

Damping state design is based on the flow-pressure curves of a proportional valve. There is static pressure in the hydro-pneumatic system and it affects seals, which cause friction between the seal and cylinder rod [10]. This friction affects low velocity damping passively, and with proportional flow control one can change damping with high velocities. The flow-pressure curve for turbulent flow thought a fixed orifice forms a linear damping ratio curve in the ideal case. Usually, the damping ratio is between 0.2-0.6, and it is a good basis for adjustable damper design [2]. There must be at least one state, which is over-critically damped, and therefore it is always stable. Racing cars use damping where the damping ratio is over 2, and it is a good estimate value for the stable state in cab suspension. The lowest reachable damping ratio depends on seal friction forces, when a 100 N friction force gives a damping ratio of 0.04 at a velocity of 0.7 m/s. This is the lowest possible damping ratio, which can be achieved at that speed, but in practice it is not reachable because of hydraulic losses in the valve. On the other hand, there is no significant difference in damping behaviour if the valve's damping force is smaller than the friction force. Therefore, the lowest proportional valve's damping state is designed using the lowest friction damping ratio.

Table 1.

Valve damping states

Valve control %	Damping ratio (0.35 m/s)	
	Rebound	Bound
58	0.15	0.09
66	0.3	0.15
72	0.5	0.3
78	1	0.63
82	3	2

Damping was limited to five different states. Damping states were determined by damping ratio in the rebound direction with the approximate accumulator spring constant 15 kN/m and average velocity 0.35 m/s (Table 1.).

2. Measurements

2.1. Flow-pressure

The valve was tested in an open hydraulic circuit to obtain accurate flow-pressure curves in low pressure difference (Fig. 2.). The test was performed in a hydraulic circuit where pressure gauges were in both valve body ports. A manually adjustable choke valve and flow meter was placed before the first pressure gauge. Measurement was carried out by measuring pressure difference and flow when the choke valve was screwed in a close-open-close sequence. Measurements were compared to the manufacturer's curves, which were obtained at 10x magnification in the manufacturer's specifications. It was noticeable that the difference in valve controls was over 70% to 200% compared to the manufacturer's curves at low pressures (Fig. 3.). The manufacturer's curves were quite accurate at high pressures. The result was expected and meant that the damper valve should be measured if accurate information is needed.

Valve compensation was also noticeable in measurements. The temperature of the valve body started to rise rapidly when the pressure difference was over 50 bar (Fig. 4.). Also, examination of the hysteresis loop in flow-pressure curves revealed the same result. The manufacturer's curves showed that the valve should not pressure compensate until 70 bar. Pressure compensating flow direction is in rebound and the accumulator static pressure limits the maximum rebound pressure difference to about 90 bar. Measurements showed that the valve starts to pressure compensate over 50 bar and the slope of the flow-pressure curve starts to rise. Pressure compensation should not be a problem, because the maximum pressure difference of the rebound direction is smaller than the pressure when flow starts to decrease.

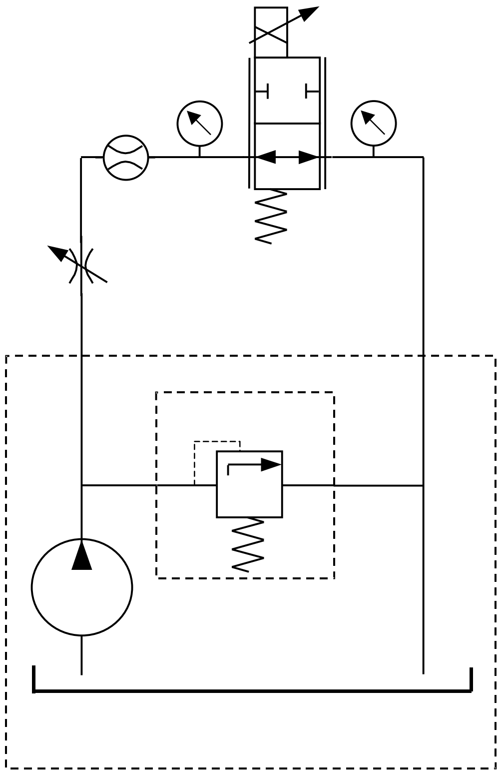

Fig. 2. Flow-pressure measurement hydraulic circuit

These flow-pressure curves give a roughly 2:1 damping ratio, and stiffer rebound damping can be obtained without any additional fluid choking method. With measured flow-pressure curves, damping states can be defined more accurately to the required damping states at desired velocities. Also then the accuracy of damping states can be obtained below the valve's positioning accuracy, which is 2%.

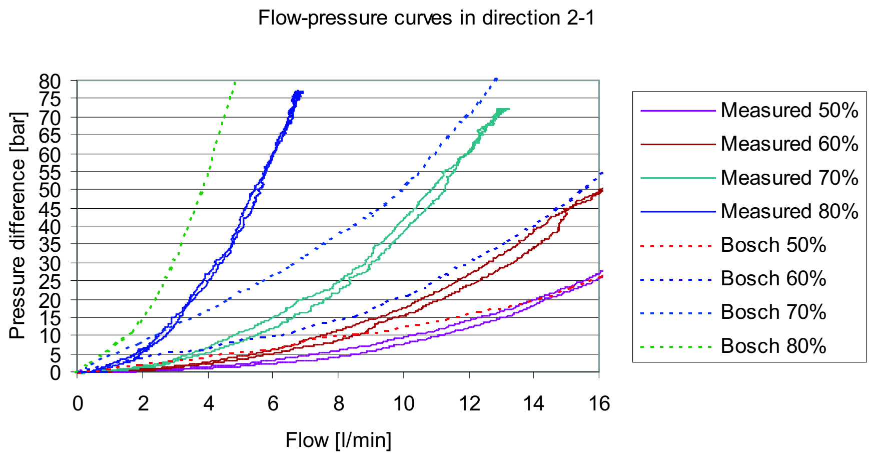

Fig. 3. Measured flow-pressure curve compared to the manufacturer's curves

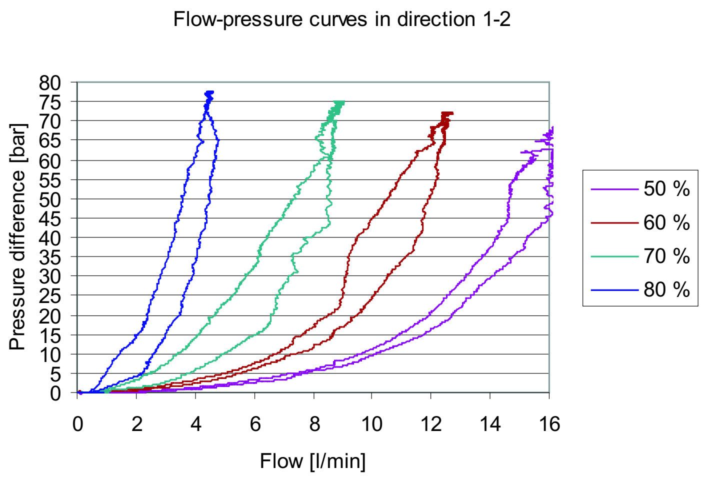

Fig. 4. Measured flow-pressure curve in the pressure compensation direction

2.2. Response time

Valve response time was tested in a closed and open hydraulic circuit with a 1 Hz square-wave signal in the valve control. Closed circuit tests were used to verify the open circuit measurements' validity to meter response time. The open circuit was used because the valve had to be measured over 30 l/min flow. It was impossible to achieve the high flow in the closed circuit where control hydraulic cylinder dynamics limited damper velocity and therefore maximum flow. The valve was connected as in flow-pressure measurements and flow was adjusted by manual choke. Measurements showed that response time was proportional to the pressure differential in the valve and depended on flow direction (Fig. 5., Fig. 6.). A decrease in closing time along flow, which was a couple of seconds at very small pressure difference, was noticeable. The valve needs at least a 10-bar pressure difference to achieve the manufacturer's 40 ms response time, which is measured at a constant 10-bar pressure difference. The valve must be selected in a different way for better response time. The lowest damping state should be as near as possible to the valve's maximum flow to achieve the best results in response time. In this case, over 3 l/min flow is needed to have noticeable differences in viscous damping. At lower flows, damping is mainly friction. With this information, one can define the highest valve operating frequency as 2 Hz. Then, the damping state shift occurs always when damper velocity is in the viscous damping range and it affects damping instead of seal friction.

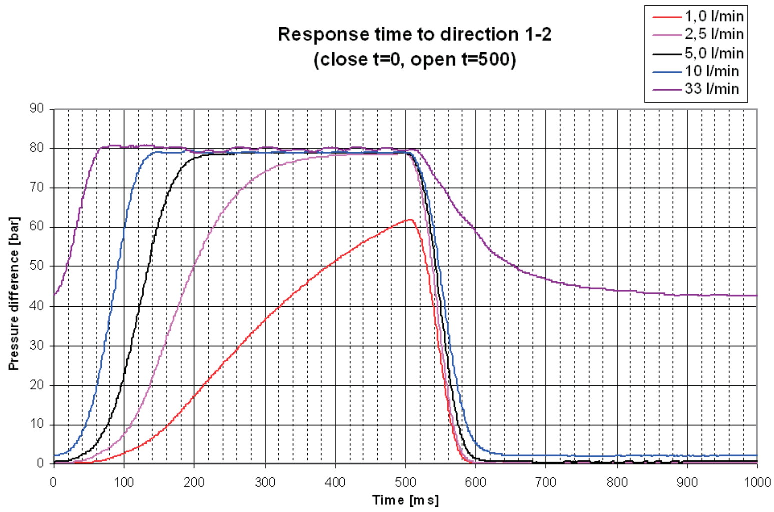

Fig. 5. Valve response time in flow direction from port 1 to 2 (rebound)

In this measurement, the flow determines pressure difference as in a real hydro-pneumatic damper, and then the flow also has its own influence on response time. Flow force tends to close the orifice in the 1-2 flow direction and open in the 2-1 direction. Comparing opening response in different directions, one can see that response time is greater in the 1-2 flow direction than when the flow force is in the opposite direction.

Closing time variation is much greater than that of opening time, and it also creates a greater problem in damping control. Stability is more important than comfort, which means that closing response should be as fast as possible.

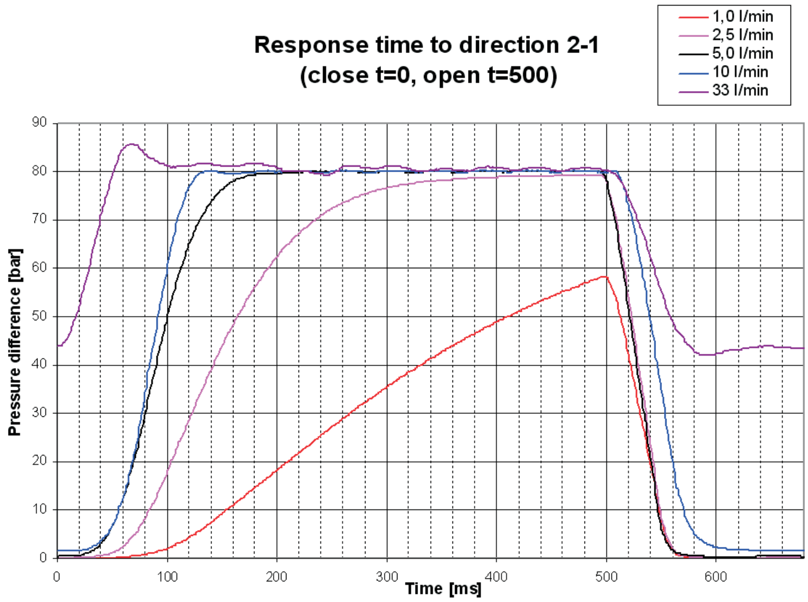

Fig. 6. Valve response time in flow direction from port 2 to 1 (bound)

Closing time was also measured with another proportional valve, which was a normally closed poppet valve. The same effect was also found in that valve, which indicates that it did not depend on the valve type.

A fast response time (20 ms) was sought because then one can use valve control to prevent damping cylinder end-stop impacts. Proportional valves do not give the required performance in small direct-operated valves, and thus an additional end-stop impact system has to be present for safe operation. Also, a passive end-stop impact system is safer than an active one, but it increases the cost of the whole hydro-pneumatic suspension system. Then, the cylinders must be custom-made, or must be expensive end-stop hydraulic cylinders, while the active system can operate with cheap, standard cylinders.

2.3. Dynamics and stability

The proportional valve is a second order vibrating system in hydro-pneumatic suspension. Therefore, it is important to determine the valve's stability in cab suspension. The valve was tested with damping frequencies 1.5-10 Hz and velocities 0-0.35 m/s with forced vibration. The test signal was a triangle wave whose amplitude was increased linearly from zero to maximum (Fig. 7.). Also, the state shift at a velocity of 0.35 m/s was investigated. The vibration effect was explored by pressure difference in the valve.

A triangle wave causes high acceleration at turning points, which simulates strong impulses in the cab suspension.

Measurements showed that valve is stable in cab suspension. The valve started to vibrate slightly in the lowest damping state (control 58%) and vibration was unnoticeable in the hardest damping state (control 82%). This also proves that a normally open valve is better for stable behaviour than a normally closed valve. The proportional magnets generating force against spool spring are at their maximum in the closed position and thus the behaviour is understandable. Vibration in low damping states was noticeable but only slightly greater than the pressure gauge's signal noise. Also, the measurements' test (velocity is changed rapidly to the opposite) simulates a very hard impact, which is rare in normal free vibration, and thus the comfort state should also remain stable.

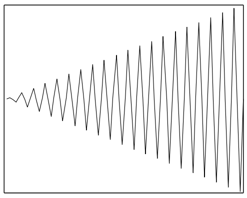

Fig. 7. Increasing amplitude triangle wave

Total damping force was measured using an increasing amplitude triangle wave (Fig. 7.). There velocity was constant between turning points, and the force-velocity curve could be drawn with velocity fractions.

Total damping force measurement reveals the role of static friction in hydro-pneumatic suspension (Fig. 8.). The suspension works as a friction damper at low velocities and state shifting does not have any influence on the damping force. Vibration at low amplitudes and high frequencies is poorly damped because of friction. Static friction should be minimized for a continuous variable system where friction also limits the usable control area [10]. Static friction is a relatively difficult problem to solve, because there is always static pressure in a hydro-pneumatic suspension, and it has to be sealed somehow. A rubber spring in series is an easy solution to the problem, but it gives only an fairly good performance. Another solution is to keep the cylinder rod in rotating movement in which case the influence of the Stribeck region of friction decreases.

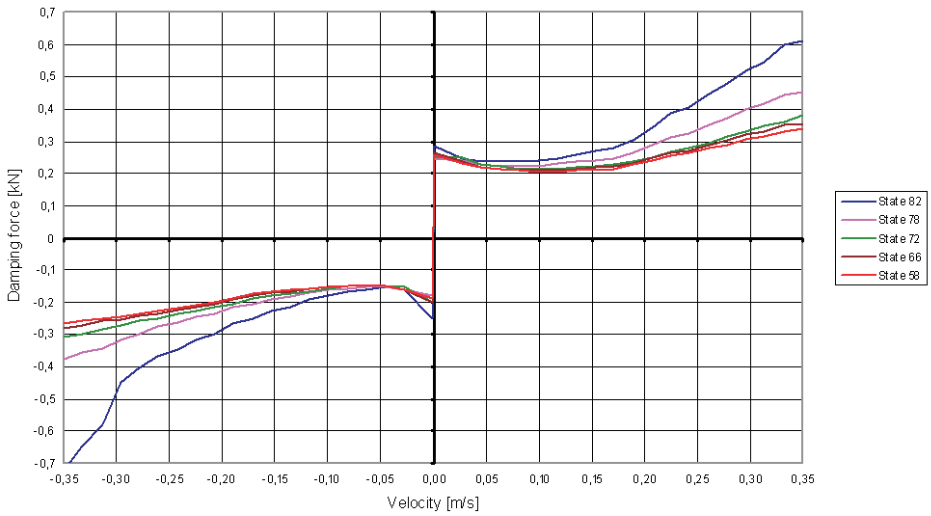

Fig. 8. Total damping forces in different damping states (rebound in negative direction)

3. Conclusions

The proportional valve was found to be stable in bi-directional flow at cab suspension frequencies. Measurements showed that valve spool vibration does not have any significant effect on damping in any damping state. Damper response time demand (20 ms) was not achieved and it would be quite difficult to achieve with any direct-operated valve. Valve response time was found to be dependent on fluid flow through the valve, and thus the maximum 20 Hz frequency is not reachable in this system. In this case, the proportional valve is suitable for damping control when control frequency is below 2 Hz.

Total damping force was also measured. Seal friction created a "knee" in the total damping force, but it is a disadvantage at zero velocity. Friction should be decreased, or another damping system for low amplitudes should be designed. Static friction is a major problem in particular for continuous damping control.

The proportional valve is suitable for hydro-pneumatic cab suspension and it can produce multi- or continuous-state damping. Damper construction is simplified to one controllable commercially available valve and it has a competitive price comparing to other semi-active dampers.

Manuscript received by Editorial Board, March 20, 2007

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Regulowany amortyzator zawieszenia kabiny

Streszczenie

Przedmiotem pracy było projektowanie i badanie regulowanego, hydro-pneumatycznego amortyzatora przeznaczonego do zawieszenia kabiny. Za cel przyjęto znalezienie prostego i taniego rozwiązania, w którym tłumik jest umieszczony między cylindrem hydraulicznym i zasobnikiem. Wzięto pod uwagę wpływ rodzaju nawierzchni na działanie amortyzatora. Typy nawierzchni zmieniały się od jazdy po drogach do jazdy terenowej, a tłumik miał natychmiastowo reagować na zmieniające się warunki. Półaktywny amortyzator był w tej pracy zbudowany przy użyciu dwukierunkowego zaworu hydraulicznego typu kasetowego 2/2 o działaniu bezpośrednim z proporcjonalnym sterowaniem przepływem. Pomiary charakterystyk ciśnienia przepływu i testy dynamiczne były wykonane w laboratorium. W testach dynamicznych badano stabilność tłumienia dla różnych częstotliwości drgań wymuszonych i dla wymuszenia skokowego między różnymi stanami. Mierzono także całkowitą siłę tłumienia w różnych stanach, a dokładna proporcjonalna odpowiedź zaworu na skok jednostkowy i jego stabilność były badane w zamkniętym systemie hydraulicznym.

W rezultacie badań uzyskano wiele nowych informacji o przydatności zaworu proporcjonalnego do zastosowania w półaktywnym amortyzatorze, oraz o właściwościach tłumienia. Badania wykazały, że zawór proporcjonalny może pracować w amortyzatorze zawieszenia kabiny, a także jako tłumik kryzowy. Stwierdzono, że dwukierunkowy przepływ płynu w zaworze proporcjonalnym był stabilny w roboczych warunkach pracy zawieszenia kabiny. Zawór proporcjonalny może także pracować jako amortyzator dla stanów zmieniających się w sposób ciągły, co przy odpowiednim sterowaniu może prowadzić do lepszych właściwości tłumiących.