Air Spring and Magnetorheological Damper: an Integrated Solution for Vibration Control

F. Renno, S. Strano, M. Terzo

***Abstract*—A combined employment of an air spring and a magnetorheological damper is proposed for the vibration control. The proposed device is characterized by the possibility to change stiffness and damping, making it fully versatile and functional to overcame the several issues that can be found in vibration control. The integrated solution is based on the parallel of both an air spring and a magnetorheological damper. The solution allows to select the better compromise to enhance the opposite aspects of ride and handling.**

***Index Terms*—Air spring, magnetorheological fluid, damper, ride, handling**

I. Introduction

DIFFERENTLY from conventional coiling springs or leaf springs, air springs are characterized by many advantages, such as adjustable carrying capacity, reduced weight, variable stiffness with almost constant natural frequency, reduced structurally transmitted noise, variability of ride heights, and so on. As a result, air springs have been widely used in luxury passenger cars, sports utility vehicles and mini-vans in order to increase requirements for driving comfort and safety. Two are the variables to take under consideration for the design of a suspension system, the spring stiffness and the damping factor. There are different requirements for the spring stiffness as a car in different driving conditions. While accelerating, braking, cornering, or changes in load, the spring stiffness should be increased so as to reduce the dynamic suspension displacement or the change in ride height. However, while normal riding, the spring stiffness should be softer in order to improve ride smoothness. Variable stiffness, as one of the important properties of the air spring, can satisfy this requirement. At the same time, there are conditions that require also a variable damping factor. For example, a reduced damping factor is desired when the air springs are switched for a high ride height in order to improve comfort. Conversely, a low ride height is characterized by reduced stiffness that could cause high amplitude motions of the chassis and so, an augmented damping factor should be functional. This paper, taking into account the several conditions illustrated above, aims to present a device characterized by the controlled variability of both stiffness and damping. It is substantially based on the merging of an air spring with a magnetorheological (MR) damper. The MR dampers employ MR fluids that are suspensions that exhibit a rapid, reversible and tunable transition from a free-flowing state to a semi-solid state, after the application of an external magnetic field.

The paper is organized as follows: section II describes semi-active suspensions, section III illustrates the MR fluids, sections IV and V concern the air spring and MR damper respectively. The proposed idea is illustrated in Section VI.

F. Renno is with the Department of Industrial Engineering, University of Naples Federico II, Naples 80125, Italy (e-mail: fabrizio.renno@unina.it).
S. Strano is with the Department of Industrial Engineering, University of Naples Federico II, Naples 80125, Italy (e-mail: salvatore.strano@unina.it).
M. Terzo is with the Department of Industrial Engineering, University of Naples Federico II, Naples 80125, Italy (corresponding author phone: +39 081 7683277; fax: +39 081 2394165; e-mail: m.terzo@unina.it).

II. Semi-active Suspensions

Generally, a suspension, in its conventional configuration, is constituted by an elastic element, a damping element and a set of mechanical elements which links the suspended body (sprung) to the unsprung mass.

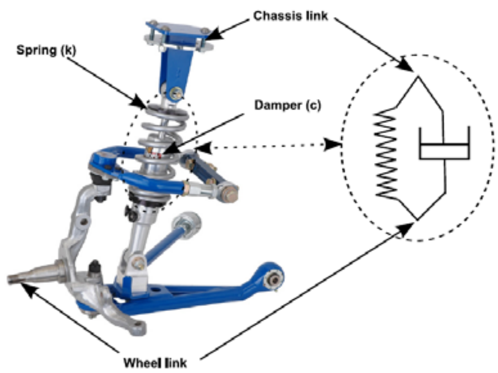

Fig. 1. Classical scheme of a wheel-to-chassis suspension in a car.

The mechanical links determine only the suspension kinematics; hence, from the dynamics point of view, the spring and the damper are the two key elements of a suspension.

As concerns an automotive suspension (Fig. 1), its design implies a tradeoff between ride comfort and vehicle stability. Ride quality is closely tied to the acceleration of the sprung mass and it can be quantified by the amount of energy transmitted through the suspension into the passenger compartment (sprung mass). Conversely, vehicle stability is strictly related to the road-holding ability.

This critical trade-off is worsened by the fact that a suspension has a limited travel: when the end-stop (bushing) of a suspension is reached, both the comfort and road-holding performances are dramatically deteriorated, and the occurrence of this situation must be carefully avoided.

The need to optimize the suspension compromise has

ISBN: 978-988-14047-0-1
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

given rise to several new advancements in automotive suspensions.

If the damper is replaced with a force actuator, the suspension becomes a fully active suspension. The idea behind fully active suspensions is that the force actuator is able to apply a force to the suspension in either jounce or rebound. This force is actively governed by the control scheme employed in the suspension.

When the suspension is modified without energy insertion, the suspension is called semi-active. A semi-active suspension is typically composed by a spring type element and a damper that is continuously adjustable [1].

The most interesting features of semi-active suspensions are [2]:

• negligible power-demand: they are based on the regulation of the damping-ratio and, consequently, the power-absorption is limited;

• safety: the stability is always guaranteed in a semi-active suspension since the whole system remains dissipative;

• low cost, low weight: the main damping-modulation technologies (Electro-hydraulic, Magneto-Rheological, Electro-Rheological, Air-damping) can be produced (for large volumes) at low cost and with compact packaging;

• significant impact on the vehicle performance: by changing the damping ratio of a suspension, the overall comfort and road-holding performance can be significantly modified.

A. Applications and technologies of semi-active suspensions

Semi-active suspensions are widely used in a vast domain of applications. In particular, for vehicle applications, semi-active suspensions can be used at wheel-to-chassis layer, at the chassis-to-cabin layer and at the cabin-to seat layer.
Semi-active suspensions for chassis-to-cabin layer are used for large vehicles where the driver cabin is separated from the main chassis (e.g. earth-moving machines, large agricultural tractors, trucks, etc.).
Fig. 2 shows an example of a semi-active suspension system for a cabin-to seat layer. This particular suspension design is useful in large off-road vehicles in order to reduce the vibration transmitted to the driver during the long working-time spent on the vehicle.

Semi-active dampers are available in three main technologies based on the system adopted to regulate the damping (see Fig. 3).

Electro-Hydraulic technology, based on solenoid valves located inside or outside the main body of the damper; they can change the damping ratio by modifying the size of orifices. Magneto-Rheological technology is based on fluids which can change their viscosity when exposed to magnetic field. Electro-Rheological technology is based on fluids which can change their viscosity when exposed to electric fields.

Such technologies are in competition on the basis of many features and parameters, such as: response-time, controllability range, cost, weight and packaging, maintenance requirements, power-electronics requirements, etc.

Fig. 2. Example of cabin-to-seat (by SEARS) semi-active suspension systems.

a) b) c)

Fig. 3. Examples of semi-active dampers, using three different technologies, a) solenoid-valve Electro-Hydraulic damper, b) Magneto-Rheological damper, and c) Electro-Rheological damper.

B. Mathematical model of a semi-active suspension

Fig. 4 shows a scheme of a conventional semi-active suspension, where k is the stiffness of the spring element and β is the adjustable damping.

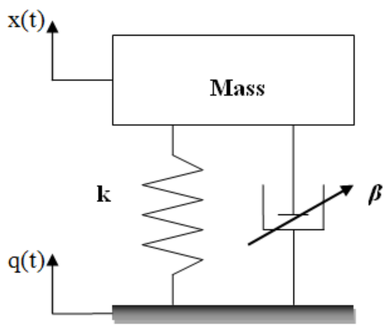

Fig 4. Scheme of a semi-active suspension.

The damper characteristic β is continuously variable and is controlled by a computer algorithm.

The linear idealized model for this system is:

$$m\ddot{x} + \beta(t)\dot{x} + kx = \beta(t)\dot{q} + kq \qquad (1)$$

where β varies over time as defined by a control law. The linear equation as written assumes that the damping force is a linear function of the relative velocity with slope equal to

ISBN: 978-988-14047-0-1
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

the damping coefficient. This is easier to see if (1) is rewritten in terms of a summation of forces as follows:

$$m\ddot{x} + f_d + f_s = 0 \qquad (2)$$

where f_d is the damping force defined as

$$f_d = \beta(t)(\dot{x} - \dot{q}) \qquad (3)$$

and f_s is the spring force defined as

$$f_s = k(x - q) \qquad (4)$$

where $(x-q)$ and $(\dot{x}-\dot{q})$ are the relative displacement and relative velocity, respectively.

In reality, the damping and spring forces are nonlinear; a more realistic modelling approach for both the forces is provided in Sections IV and V, focusing the study on two particular devices: an air spring and a magnetorheological damper.

III. Magnetorheological Fluids

These materials demonstrate changes in their rheological behaviour in presence of an applied magnetic field [3]. In the recent years MR fluids have attracted considerable interest because they can provide a simple and rapid response interface between electronic controls and mechanical systems [4 – 7]. The initial discovery and development of MR fluids and devices can be credited to Rabinow at the US National Bureau of Standards in the late 1940s [8 – 10]. These fluids are suspensions of micron-sized, magnetizable particles in an appropriate carrier liquid. Normally, MR fluids are free-flowing liquids having a consistency similar to that of motor oil. However, in the presence of an applied magnetic field, the particles acquire a dipole moment aligned with the external field that causes particles to form linear chains parallel to the field. This phenomenon can solidify the suspension and restrict the fluid movement. Consequently, yield stress is developed. The degree of change is related to the magnitude of the applied magnetic field and can occur in only a few milliseconds. The primary advantage of MR fluids stems from their large and controllable dynamic yield stress due to the high magnetic energy density that can be established in the fluids. The behaviour of the MR fluid is divided into pre-yield and post-yield regions, depending on whether the fluid is stressed below or above a critical yield stress value. The post-yield behaviour is non-Newtonian and the apparent viscosity of the fluid increases due to the increase in the yield stress with an external magnetic field.

The MR fluids can operate in three different modes (Fig. 5): the flow mode, direct shear mode and squeeze film mode. The flow mode, direct shear mode and their combination are used to develop all kinds of linear and rotary MR dampers while the squeeze film mode is only used to develop linear MR dampers with limited amplitudes.

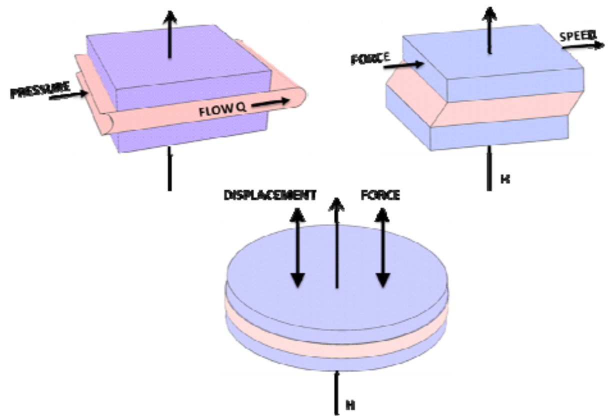

Fig 5. Operational modes of MR fluids.

IV. Air Spring

The air spring is a suspension element that consists of two chambers (primary and additional volume) filled with air at a desired pressure and connected to each other by means of a pipeline system. The stiffness of the air spring depends on the total volume and an electromagnetic valve is adopted to link the additional volume and, so, to change the stiffness. The change of the spring stiffness is controlled by an electromagnetic valve, while damping ratio is defined by dimensions and construction characteristics of the interconnection pipeline.

Considering the dimensions and the construction of the pipeline, there is a phase difference between the pressures in the two volumes which result in dynamic stiffness characteristics. The air spring systems require air supply equipment and pneumatic control valves, so they need more available space compared to the conventional suspension systems.

Air springs can also provide load leveling functions by adding or removing air. From the standpoint of vehicle suspensions, air springs can be found in many types of vehicles, such as automobiles, large trucks, buses, railway vehicles, construction and agricultural vehicles. In large truck applications, air springs can be used for cab and seat suspensions in addition to truck and trailer chassis suspensions. In addition, air springs are also used in factory operations.

Air springs are designed and manufactured in various shapes and sizes to meet a wide variety of applications as shown in Fig. 6. Two common types are the reversible sleeve or rolling lobe and the convoluted or bellows as shown in Fig. 7. A reversible sleeve type air spring contains a piston over which the elastomeric material moves as the height of the air spring changes in relation to load variations. Thus the piston plunges in and out of the air cavity. The convoluted air spring contains one or more lobes. There is no piston associated with this type of air spring. In general, the reversible sleeve air spring has an advantage over the convoluted by the fact that the piston can be shaped to fine tune the spring rate. In addition, it is possible with the reversible sleeve design to maintain a constant load for a given internal pressure over a range of heights [1].

ISBN: 978-988-14047-0-1
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

Fig 6. Air springs.

Figure 7. Reversible sleeve and convoluted designs

Depending on the application, advantages of the air spring

over a steel spring are often cited in the literature. Some of the advantages enumerated in [3] are as follows:

1) an air spring has variable load-carrying capability. If a higher load needs to be accommodated, air can be added to the spring (automatically or manually) to increase the pressure and, at the same time, maintain the required height of the suspension;

2) the spring rate of an air spring can be adjusted. This happens, for example, when an additional load is applied and air is added to the spring to maintain a specified height. The internal pressure increases to accommodate the load, but without a significant shift in the suspension natural frequency.

3) the height of the load can be adjusted when necessary by increasing or decreasing the amount of air in the air spring. This allows for load leveling operations, or for the "squatting" capabilities such as are found in some transit bus applications.

4) an air spring has low friction dynamics. Since there is a flexible elastomeric member separating the rigid attachment points of the suspension, the air spring can move in six degrees of freedom without the resistance and squeaks associated with conventional steel spring suspensions.

Since air springs and steel springs contain relatively small amounts of damping, they are typically used in conjunction with separate dampers or shock absorbers. Although, integrated air spring and damper units are available for some applications as shown in Fig. 8, the spring and damper functions are typically separate. In many cases, air or some other gas is the fluid medium for the air spring, and a hydraulic fluid is the medium for the damper.

Fig 8. Integrated air springs and damper.

A. Air Spring Modelling

For a classical modelling of an air spring, the stiffness characteristic is presented by [11]:

$$k = \frac{PnA_{ef}^{\ 2}}{V} \tag{5}$$

where P is the absolute pressure in the air spring, A_{ef} is the effective area, V is the volume and n is the polytrophic coefficient. The stiffness characteristic (5) is useful to determine the static reaction of an air spring. An air spring model has been proposed in [12] in order to take into account dynamic conditions. The air spring is considered as the parallel connection of a pneumatic cylinder, capable of heat exchange, and an equivalent viscous damper (Fig. 9).

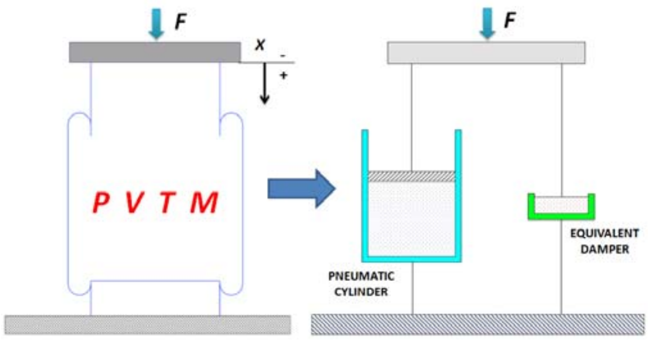

Fig 9. Air spring and its equivalent model.

The dynamic model is derived in the light of energy conservation and gas state equation, giving the following expression for the reaction force of the air spring:

$$F = A_{ef}(P - P_a) + c_{eq}\dot{x} \tag{6}$$

where P_a is the atmospheric pressure, c_{eq} the equivalent damping coefficient and x the displacement.

V. MR Damper

MR dampers are controllable damper based on the controllable property, defined MR effect, of the MR fluid to change its rheology when exposed to a magnetic field. MR dampers, which utilize the advantages of MR fluids, are semi-active control devices that can generate a magnitude of force sufficient for large-scale applications, while requiring only a battery for power. These devices offer highly reliable

ISBN: 978-988-14047-0-1
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

operation and their performance is relatively insensitive to temperature fluctuations and/or impurities in the fluids. The applications of MR dampers can be found over the range from civil structures such as buildings and bridges to automobiles and railway vehicles.

A. MR Damper Modelling

The nonlinear Bingham plastic model can be used to model the MR damper force [13]. It is based on the scheme of Fig. 10 that is based on a viscous damper combined in parallel with a Coulomb friction element.

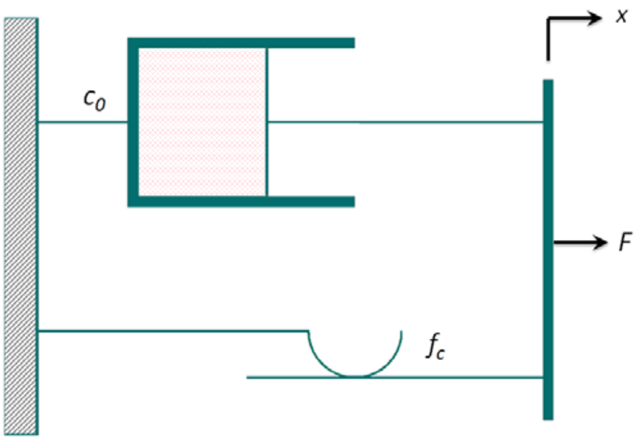

Fig. 10. The Bingham plastic model for MR dampers: a Coulomb friction element in parallel with a viscous damper.

The MR damper force is given by:

$$F_D = c_0\dot{x} + f_c\,\mathrm{sgn}(\dot{x}) + f_0 \qquad (7)$$

where c_0 is the damping coefficient, f_c is the frictional force related to the field-dependent yield stress and f_0 is the offset in the force.

VI. MR Damper Integrated with Air Spring

The idea proposed in this paper is characterized by the integration of an air spring with an MR damper. The idea takes place starting from the passive device illustrated in Fig. 11 and based on the combination of an air spring with a common viscous damper. The architecture of the device is based on the parallel connection between the spring and the damper.

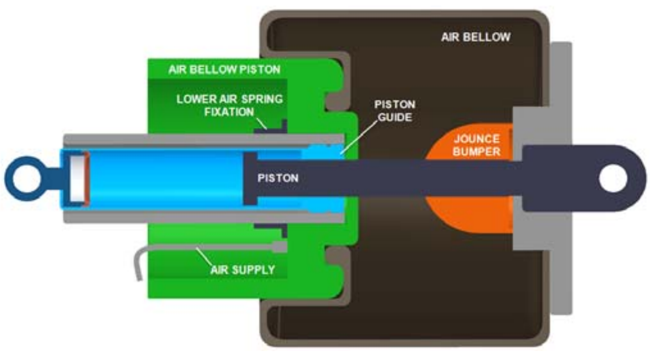

Fig. 11. Air spring integrated with a viscous damper.

The proposed device employs an air spring integrated with an MR damper. A first scheme (solution *a*) is illustrated in Fig. 12: the MR fluid flows through the annular gap between the piston and the cylinder. A magnetic circuit, supplied by the excitation coil located in the piston, is used to generate controllable magnetic field by varying the coil current. The resultant damping force is due to both shear damping and valve damping forces. It changes dynamically with the magnetic field generated by the input current.

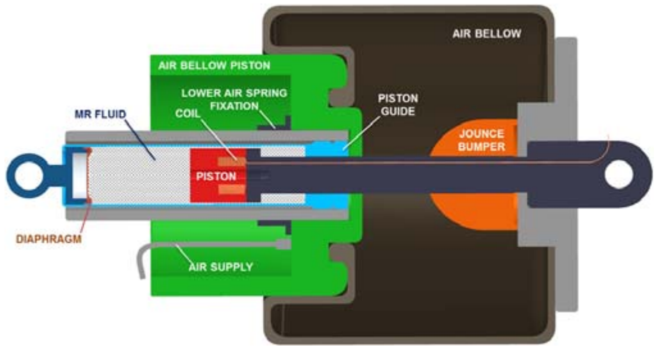

Fig. 12. Air spring integrated with an MR damper (solution *a*).

Another scheme is proposed in Fig. 13 (solution *b*). It is based on the employment of the valve mode: indeed, suitable orifices have been realized in the piston and seals have been adopted between the piston and the cylinder.

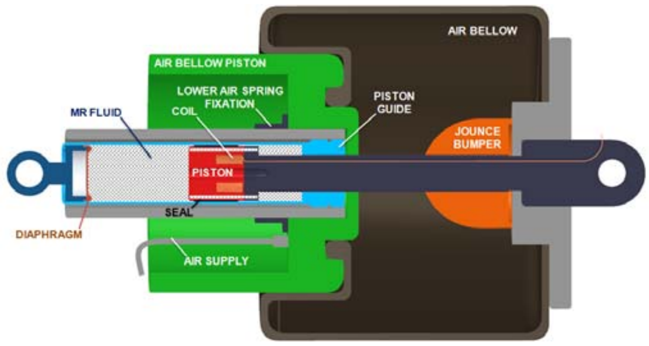

Fig. 13. Air spring integrated with an MR damper (solution *b*).

The last scheme (solution *c*) is based on the presence of orifices and annular gap (Fig. 14).

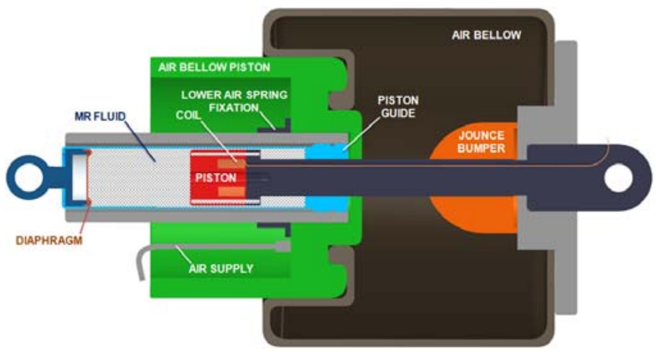

Fig. 14. Air spring integrated with an MR damper (solution *c*).

The three solutions differ for the MR fluid operational modes. The solution *a* is based on the combination of both the shear and the valve mode. The magnetic circuit involves inevitably the piston and the cylinder. The solution *b* is characterized by the only valve mode and the magnetic circuit is located only in the piston. The solution *c* presents the operational modes of the solutions *a* and *b*, with a magnetic flux that crosses the orifices and the annular gap. The increasing of the fluid volume to be controlled is the principal feature of this last solution. At the same time, an augmented magnetic reluctance is due to the presence of both the orifices and the annular gaps.

VII. Conclusion

An innovative idea concerning the combination of an air spring with an MR damper has been presented in the paper. The possibility of changing both the stiffness and the damping represents a functional solution for several issues concerning the vibration control. The proposed integrated system has been illustrated in three different configurations that allow to control independently the stiffness and the damping.

References

[1] R. W. Daniel, “A pneumatic semi-active control methodology for vibration control of air spring based suspension systems,” Graduate Thesis and Dissertations, Paper 12555, 2012.

[2] S. Savaresi, C.P. Vassal, C. Spelta, O. Sename, L. Dugard, “Semi-Active Suspension Control Design for Vehicles,” Butterworth-Heinemann, 2010.

[3] J.D. Carlson, K.D. Weiss, “A growing attraction to magnetic fluids,” *Mach. Des.* vol. 8, pp. 61 – 66, 1994.

[4] W. Kordonsky, “Magnetorheological effect as a base of new devices and technologies,” *J. Magn. Magn. Mater.,* vol. 122, pp. 395 – 398, 1993.

[5] W. Kordonsky, “Elements and devices based on magnetorheological effect,” *J. Intell. Mater. Syst. Struct.,* vol. 4, pp. 65 – 69, 1993.

[6] A. Lanzotti, M. Russo, R. Russo, F. Renno, M. Terzo, “A physical prototype of an automotive magnetorheological differential,” *WCE 2013 - Lecture Notes in Engineering and Computer Science*, vol. 3, pp. 2131 – 2135, 2013.

[7] A. Lanzotti, F. Renno, M. Russo, R. Russo, M. Terzo, “Design and development of an automotive magnetorheological semi-active differential,” *Mechatronics*, vol. 24, no. 5, pp. 426 – 435, 2014.

[8] J. Rabinow, “The magnetic fluid clutch,” *AIEE Trans.* vol. 67, pp. 1308–1315, 1948.

[9] J. Rabinow, “Magnetic fluid clutch,” *Natl Bur. Stand. Tech. News Bull.* vol. 4, pp. 54–60, 1948.

[10] J. Rabinow, “Magnetic fluid torque and force transmitting device,” *US Patent Specification,* 2, 575, 360, 1951.

[11] V. Gavriloski, J. Jovanova, G. Tasevski, M. Djidrov, “Development of a New Air Spring Dynamic Model,” *FME Transactions,* vol. 42, pp. 305-310, 2014.

[12] H. Liu, J. Lee, “Model development of automotive air spring based on experimental research,” *Proceedings - 3rd International Conference on Measuring Technology and Mechatronics Automation*, ICMTMA 2011.

[13] D.H. Wang, W.H. Liao, “Magnetorheological fluid dampers: A review of parametric modelling,” *Smart Materials and Structures*, vol. 20, no. 2, 023001, 2011.