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Six Sigma methodology applied to minimizing damping lag in hydraulic shock absorbers

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Analysis and modelling

ABSTRACT

Purpose: The aim of this paper is to identify the root cause of the temporary decrease of the damping force during the early stage of the compression phase of the stroking cycle, the so called damping lag, to describe measures of the phenomenon and to present methods for optimizing the design towards minimizing this (negative) effect.

Design/methodology/approach: Theoretical background is presented in a constructive and computable manner with emphasis on measurement data analysis and MATLAB/Simulink modeling. Six Sigma tools were used to validate the model statistically and, more importantly, to propose a method of data-driven optimization of the design.

Findings: Root cause of the occurrence of the damping lag was confirmed during model validation to be caused by oil aeration. The dependence of the damping lag on parameters is nonlinear. Six Sigma methodology proved to be useful in achieving design optimality.

Research limitations/implications: Statistical model and conclusions drawn from it are only valid in the interior of the investigated region of the parameter space. Additionally, it might not be possible to find a local minimum of the aeration measure (damping lag) inside the selected region of the parameter space; global minimum located at the boundary might be the only possible solution.

Practical implications: Optimal value of parameters is not unique and thus additional sub-criteria (cost/durability) can be imposed. Conducting tests in an organized manner and according to the Six Sigma methodology allows for expediting the design optimization process and eliminating unnecessary costs.

Originality/value: : Improvements in understanding and measuring aeration effects constitute a clear foundation for further product optimization. Signal post-processing algorithms are essential for the statistical analysis and are the original contribution of this work.

Keywords: Statistic Methods; Six Sigma; Shock absorber; Aeration

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1. Background

Functionally, an ideal shock absorber (damper) should satisfy the following, sometimes viewed as mutually contradictory, criteria. In order of importance: (i) a car damper has to guarantee good road handling of the car, (ii) has to be designed for durability, (iii) radiated noise and emitted vibrations has to have as low power as possible, and (iv) it should ensure comfort of the passengers. In order to satisfy these design criteria to the largest possible extent, a number of formalized quality control methodologies has been proposed and developed. Among these, the most successful one is the Six Sigma approach. Six Sigma is a disciplined, data-driven approach and methodology to seek to eliminate defects and errors in processes and products. The fundamental objective of the Six Sigma methodology is the implementation of a measurement-based strategy that focuses on process improvement and variation reduction [1]. It uses a set of statistical methods, including DOE (design of experiments) techniques - a structurized, organized method for analyzing the influence of factors, and interactions between factors, on the output of a process [2]. Outcomes of the DOE analysis can be used for either identification or optimization purposes. This work presents application of DOE tools in design optimization.

1.1. Working principle of a shock absorber

This section presents fundamental working principles of the hydraulic shock absorber. A hydraulic double-tube damper presented in Figure 1 consists of a piston moving inside a liquid-filled cylinder. As the piston is forced to move inside the cylinder (pressure tube), a pressure differential is built across the piston and the base-valve assembly that the liquid is forced to flow through restrictions (orifices) and valves [3].

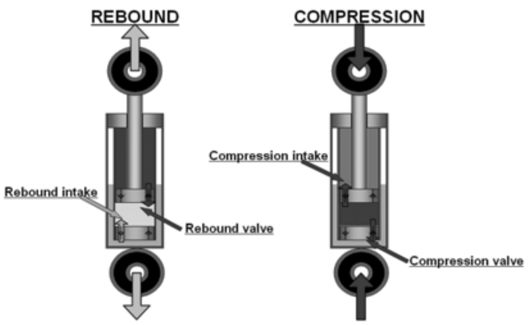

Fig. 1. Working principle of a double tube damper

The piston divides the cylinder space onto two chambers: (i) the rebound chamber, the portion of the cylinder above the piston and (ii) the chamber, the portion below the piston. Action of the piston transfers liquid to and from the reserve chamber, surrounding the cylinder, through the base-valve assembly located at the bottom of the compression chamber. Two types of valves, (i) intake valves and (ii) control valves, are used in a shock absorber. Intake valves are check valves providing small resistance to flow in one direction and preventing the flow in the opposite direction upon reversal of the differential pressure. A valve spring is used in order to preload control valves and prevent their opening until a specified pressure differential has built up across the valve. When the valve opens, stiffness of the valve spring controls the height of the opening. Depending on the required force-displacement range and tolerance, coil or disc springs are used. Orifices of various types are used in the shock absorber assembly to provide demanded flow restriction. Initial orifices, or notches in the valve seat, are used to provide a highly restricted flow path between two chambers when the connecting valves are closed. On the other hand, high speed valve restrictors placed in series with a valve provide a flow restriction when the valve is open.

Cyclic operation of a damper can be split into four transient stages *A, B, C, D* as follows

$$
\begin{aligned}
A:\ & (p_{com}-p_{reb}\le 0),\quad (p_{com}-p_{res}\ge 0),\\
& q_{intake,reb}=0,\quad q_{intake,com}=0,\\
B:\ & (p_{com}-p_{reb}\le 0),\quad (p_{com}-p_{res}\le 0),\\
& q_{intake,reb}=0,\quad q_{control,com}=0,\\
C:\ & (p_{com}-p_{reb}\ge 0),\quad (p_{com}-p_{res}\le 0),\\
& q_{control,reb}=0,\quad q_{control,com}=0,\\
D:\ & (p_{com}-p_{reb}\ge 0),\quad (p_{com}-p_{res}\ge 0),\\
& q_{control,reb}=0,\quad q_{intake,com}=0,
\end{aligned}
\tag{1}
$$

where $q_{intake,reb}$ is the flow through the rebound intake valve; $q_{control,reb}$ is the flow through the rebound control valve; $q_{intake,com}$ is the flow through the compression intake valve; $q_{control,com}$ is the flow through the compression control valve; p_{com} is the pressure in the compression chamber; p_{reb} is the pressure in the rebound chamber; p_{res} is the pressure in the reserve chamber.

Stage *A* represents conditions at the beginning of a rebound stroke phase, where initially all valves are closed, and liquid from the rebound chamber is forced to flow through “rebound orifices” in the piston into the compression chamber. The rebound control valve opens, allowing the liquid to flow through the rebound control valve and through rebound notches; opening occurs when the pressure differential across the piston becomes sufficiently large. Continuation of the rod movement further into the rebound stroke, stage *B*, causes a volume of the liquid, equal to the displacement of the piston rod being withdrawn, to be pumped out from the reserve chamber, via the compression intake valve, to the compression chamber. The rebound intake valve and the compression control valve remain closed during this stage. In order for the liquid to flow from the reserve chamber to the compression chamber the pressure p_{com} therein must be lower than the pressure p_{res}. Pressure drop in compression chamber is accompanied by an expansion of fluid which may cause the formation of entrapped gas bubbles, or if the pressure level is sufficiently low, vaporization of the liquid. At the beginning of the compression stroke (stage *C*), the presence of entrapped gas or liquid vapor results in a large piston displacement before a significant pressure differential across the piston is built. This lag in the pressure build-up at the beginning of the compression stroke is referred to as the “compression lag”. During stage *C*, the across the piston exceeds the pressure differential across the base-valve assembly, and bubbles of entrapped gas or liquid vapor form in valve restrictions. Presence of gas/vapor bubbles entrapped in the liquid is highly undesirable since, at the beginning of the compression phase, bubbles have to collapse causing performance deterioration. Stage *C* and its effects on shock absorber performance have been minimized by shock absorber designs utilizing certain ratios of rebound intake valve to compression control valve restrictions. As discussed earlier, formation of entrapped bubbles and the accompanying symptoms of the compression lag have been minimized with the use of a gas volume which is pressurized well above the 4 atmospheric pressures. Well into the compression stroke (stage *D*), after collapse of entrapped bubbles, the pressure differential across the base valve is reversed and become sufficiently large, for the compression control valve to open and allow the flow of the liquid into the reserve chamber through both, the valve and the compression orifices. During this stage, when liquid is flowing from the compression chamber into the other two chambers, the highest pressure exists in the compression chamber.

1.2. Aeration terminology

Aeration is the process by which air/gas is circulated through, mixed with or dissolved in a liquid. Gas (nitrogen) is included in dampers under certain pressure, separately from the oil, to provide compressibility to allow for the rod displacement volume compensation. A liquid that was exposed to a soluble gas (i.e. the liquid had a contact surface with the atmosphere of a gas that can dissolve in it) can be in one of three forms - liquid-gas solution, liquid-gas bubbles emulsion or foam. The liquid-gas solution is prone to bubble formation, when the pressure of the liquid-gas solution decreases below the so called saturation pressure. In this state the liquid is no longer capable of retaining all the gas in dissolved form and so bubbles occur. This paper describes properties of the liquid-gas bubbles emulsion and identifies those of its parameters that influence amount of the force lost due to aeration. An attempt to modeling the influence of increased oil compressibility is presented along with the methodology, derived from the Six Sigma approach, by which these negative effects can be minimized.

2. Theory underlying the damper model

This section presents the review of the literature and equations of the damper model, and briefly describes the theory underlying the two-phase flow model. To the best knowledge of the authors, principles of the 2-phase flow theory were not used in modeling the aeration effects in the car shock absorbers In that respect, the model described in this paper is a novelty.

2.1. Literature reviews of a shock absorber model

Shock absorber models are widely presented and discussed in the literature. A first-principle dynamic model of a hydraulic shock absorber was discussed by Lang in his early monograph [7]. This model was further simplified assuming massless valve systems [8]. A similar model has been created for a monotube shock absorber [9]. Another monotube model is focused on the understanding of hydraulic behavior of a piston valve. The model of the piston valve was developed using advanced combined simulation/measurement approach [10]. A measurement setup was equipped with a laser sensor and pressure sensors to evaluate the pressure drop across the valve assembly. The model developed in this paper is based on the new approach, namely te mass-flow rate. This provides more transparent model structure considering an aeration effect.

2.2. Equations of a shock absorber model

Damper models presented in the literature [7-8] have been extended to enable modeling of aeration effect, i.e. the ”compression lag”. Models introduced herein, involves the following assumptions: (i) independent value of a bulk module for each chamber can be applied, (ii) valve opening and closing is abrupt in a completely symmetric manner, (iii) pressure in each chamber of the shock absorber is uniformly distributed, (iv) fluid velocities in the chambers are sufficiently small so that minor dissipative losses can be neglected, (v) temperature and density are constant in all the chambers.

Force balance equation for the damper piston is

$$F = p_{com} \cdot A_{com} - p_{reb} \cdot A_{reb} + F_{friction} \tag{2}$$

where F is the force; A_{com} is the area of compression side of piston; A_{reb} is the area of rebound side of piston.

Dry friction force between the piston and the wall of the tube is modeled by

$$F_{fricition} = F_{friction\ max} \cdot tanh\left(\frac{\dot{x}}{\dot{x}_{ref}}\right) \tag{3}$$

where, $F_{friction\ max}$ is the maximal dry friction force; $\dot{x}$ is the piston/rod velocity; $\dot{x}_{ref}$ is the reference velocity, on which the speed of the fast dry friction force increases depends.

Flow restriction allows the required force characteristics of a shock absorber to be formed. Restrictions are adjusted to reach a desired relationship, i.e. force vs. piston displacement and force vs. piston velocity. Based on Figure 1 the following dependencies are formulated

$$\begin{aligned} q_{intake,com} &= f_{intake,com}(p_{com} - p_{reb}) \\ q_{intake,reb} &= f_{intake,reb}(p_{res} - p_{com}) \\ q_{control,com} &= f_{control,com}(p_{com} - p_{res}) \\ q_{control,reb} &= f_{control,reb}(p_{reb} - p_{com}) \end{aligned} \tag{4}$$

Mass flux in the compression and the rebound chambers are given by the following equations

$$\begin{aligned}\dot{m}_{com} &= \left(q_{intake,reb} - q_{control,com}\right)\cdot \rho_{com,res} \\ &\quad -\left(q_{intake,com} - q_{control,reb}\right)\cdot \rho_{reb,com} \\ \dot{m}_{reb} &= \left(q_{intake,com} - q_{control,reb}\right)\cdot \rho_{reb,com}\end{aligned} \tag{5}$$

The average density of the oil passing trough the valves is calculated as follows

$$\begin{aligned}\rho_{reb,com} &= \frac{\rho_{reb} + \rho_{com}}{2} \\ \rho_{reb,res} &= \frac{\rho_{reb} + \rho_{oli\_res}}{2} \\ \rho_{com,res} &= \frac{\rho_{com} + \rho_{oil\_res}}{2}\end{aligned} \tag{6}$$

Volumes in the compression and the rebound chambers are obtained from the following formula

$$\begin{aligned}V_{reb} &= V_{reb\_ini} - A_{reb}\cdot x \\ V_{com} &= V_{com\_ini} + A_{com}\cdot x \\ V_{res} &= const\end{aligned} \tag{7}$$

where V_{reb_ini} is the initial volume of the rebound chamber; V_{com_ini} is the initial volume of the compression chamber; x is the piston/rod displacement.

The mass of the oil in each chamber is obtained based on the mass flux

$$\begin{aligned}m_{reb} &= \frac{d\dot{m}_{reb}}{dt} + m_{reb\_ini} \\ m_{com} &= \frac{d\dot{m}_{com}}{dt} + m_{com\_ini}\end{aligned} \tag{8}$$

where m_{reb_ini} is the initial mass of oil in the rebound chamber; m_{com_ini} is the initial mass of oil in the compression chamber.

The oil mass equilibrium equation

$$m_{oil\_res} + m_{reb} + m_{com} = m_{oil} \tag{9}$$

where m_{oil_res} is the mass of oil in the reserve chamber, has to be satisfied.

Oil density in each chamber is obtained as follows

$$\begin{aligned}\rho_{reb} &= \frac{m_{reb}}{V_{reb}} \\ \rho_{com} &= \frac{m_{com}}{V_{com}}\end{aligned} \tag{10}$$

Pressure in each chamber is obtained as follows

$$\begin{aligned}p_{reb} &= p_{ini} + ln\left(\frac{\rho_{reb}}{\rho_{ini}}\right)\frac{1}{\beta_{reb}} \\ p_{com} &= p_{ini} + ln\left(\frac{\rho_{com}}{\rho_{ini}}\right)\frac{1}{\beta_{com}}\end{aligned} \tag{11}$$

where p_{ini} is the initial pressure (all chambers); ρ_{ini} is the initial oil density; β_{reb} is the compressibility of the oil in the rebound chamber; β_{com} is the compressibility of the oil in the compression chamber.

Compressibility in all chambers is assumed to be the same. Mathematical equation for an ideal fluid undergoing an adiabatic process is

$$p\cdot V^{\gamma} = const \tag{12}$$

where p is the pressure and V is the volume.

The ratio of specific heats is calculated as follows

$$\gamma = \frac{c_p}{c_v} = \frac{c_p}{c_p - R} \tag{13}$$

where c_p is the molar specific heat for constant pressure; c_v is the molar specific heat for constant volume; R is the gas constant $R = 8.314$ $[J/(mol\cdot K)]$. For a diatomic gas (such as nitrogen) $\gamma = 7/5$.

The pressure in the reserve chamber is derived from the adiabatic process equation, as follows

$$\begin{aligned}p_{res} &= p_{ini}\cdot\left(\frac{V_{gas\_res\_ini}}{V_{gas\_res}}\right)^{\gamma} \\ &= p_{ini}\cdot\left(\frac{V_{gas\_res\_ini}}{V_{res} - \dfrac{m_{oil} - m_{reb} - m_{com}}{\rho_{oil\_res}}}\right)^{\gamma}\end{aligned} \tag{14}$$

where ρ_{oil_res} is the oil density in the reserve chamber; m_{oil} is the mass of the oil; $V_{gas\_res\_ini}$ is the initial volume of the gas in the reserve chamber.

The volume of the oil in the reserve chamber V_{oil_res} and the volume of the gas in the reserve chamber V_{gas_res} are calculated as follows

$$\begin{aligned}V_{oil\_res} &= \frac{m_{oil\_res}}{\rho_{oil\_res}} = \frac{m_{oil} - m_{reb} - m_{com}}{\rho_{oil\_res}} \\ V_{gas\_res} &= V_{res} - V_{oil\_res}\end{aligned} \tag{15}$$

where m_{oil_res} is the mass of oil in reserve chamber; V_{oil_res} is the volume of oil in reserve chamber.

Oil density in the reserve chamber is obtained from the equation

$$\rho_{oil\_res} = \rho_{ini}\cdot e^{\beta\cdot\left(p_{res} - p_{ini}\right)} \tag{11}$$

where ρ_{ini} is the initial oil density; β_{res} is the compressibility of the oil in the reserve chamber; e is the Euler's Number.

2.3. Two-phase flow model theory

This section briefly describes the theory underlying the two-phase flow model. To the best knowledge of the authors, principles of the 2-phase flow theory were not used in modeling the aeration effects in the car shock absorbers. In that respect, the model developed in Tenneco is a novelty. In a typical double-tube car shock absorber the oil and gas are not separated and so gas dissolution occurs. Presence of dissolved gas has negligible effect on the viscosity of the oil. However, in dynamic conditions, when the oil-gas mixture flows through valve restrictions, local pressure in the flow drops below the saturation pressure and a portion of the gas forms bubbles entrained in the bulk of the oil. The delay in the build-up of pressure in the chambers and the hysteresis loop in the force-velocity response is attributable to (a) fluid compressibility, (b) the existence of either a gas or liquid vapor phase at certain stages of stroking cycle. Solubility of gas in a liquid is directly proportional to the absolute pressure above the liquid surface (Henry's law), and normally decreases with rising temperature [4]. The homogenous flow model assumes that the gas and the liquid have the same velocity and are in thermal equilibrium. The χ value is a ratio of the mass of gas in bubbles to the total mass of oil and gas.

Used empirically calculated (based on Henry's law) or obtained by experiment value of χ the density of homogenous gas-oil mixture illustrated in Figure 2 can be calculated with use of the following formula

$$\rho_{hom} = \left(\frac{\chi}{\rho_{gas}} + \frac{1-\chi}{\rho_{oil}} \right)^{-1}, \quad \chi = \frac{m_{gas}}{m_{hom}} \tag{12}$$

where ρ_{oil} is the density of oil in considered chamber; ρ_{gas} is the density of gas in considered chamber; m_{hom} is the mass of oil-gas mixture in considered chamber; m_{gas} is the total mass of gas (resolved and non-resolved in oil).

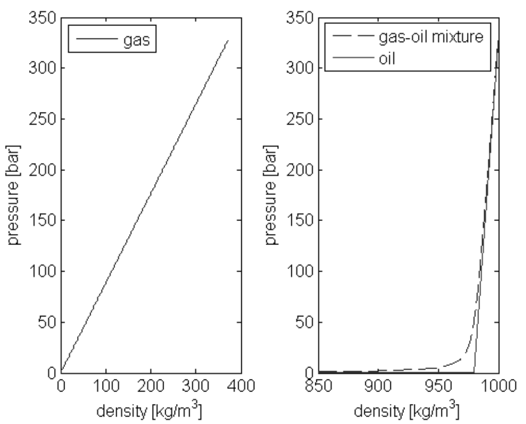

Fig. 2. Density-absolute pressure relation for gas and oil, and gas-oil mixture at constant temperature (ideal gas equation, isothermal fluid compressibility)

3. Damping lag minimization

3.1. Overview of the transparent damper

For the purpose of determining the value of χ, an experimental double tube shock absorber equipped with a transparent reserve tube (Figure 3) has been developed.

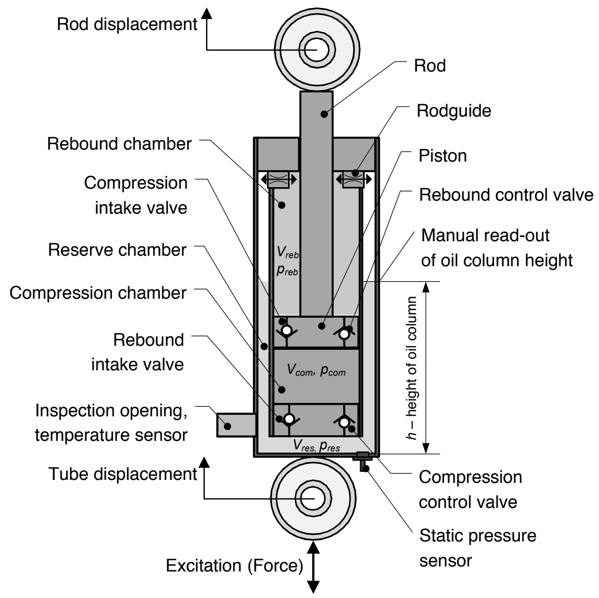

Fig. 3. Diagrammatic representation of transparent shock absorber

The transparent reserve tube makes it possible to view the effects the aeration process has on the oil and to measure the height of the oil in the reserve tube during dynamic tests thus investigating time dependence of the parameter χ. The height of the oil and the temperature of the unit and the internal pressure are measured periodically (usually every 100 cycles) after stopping the movement of the rod and placing the piston in the mid-stroke position.

3.2. Analysis of experimental data

This section describes how to process measured data (temperature, pressure and oil height) in order to obtain value of χ at certain moment during the test. From damper geometry, the actual volume of the emulsion is equal to $V_E = V_{PT} + h \cdot \left(d_{RT}^2 - d_{PT}^2 \right)$, where d_{PT} is the outer diameter of the pressure tube and d_{RT} is the inner diameter of the reserve tube; h is the height of oil column; V_{PT} is the volume of the pressure tube. The volume of gas in bubbles is the difference between the total volume of the emulsion and the theoretical volume of oil at a given temperature $V_{G\_bubbles} = V_E - V_{oil}(T)$, where the theoretical volume of pure oil in the function of the temperature is

$$V_{oil}(T) = \rho_0 V_0 / \rho(T) \quad (13)$$

It is assumed that volume V_0 at given temperature has density ρ_0. Typically, the volume V_0 is the volume computed at the beginning of the first test and density ρ_0 is computed for the temperature measured at the beginning of the first test. Mass of gas entrapped in bubbles is therefore readily computed using the formula

$$m_{G\_bubbles} = \frac{P}{R \cdot T} \cdot V_{G\_bubbles} = \rho_{gas}(P,T) \cdot V_{G\_bubbles} \quad (14)$$

where it is assumed the gas bubbles are small so that the compression is isothermal.

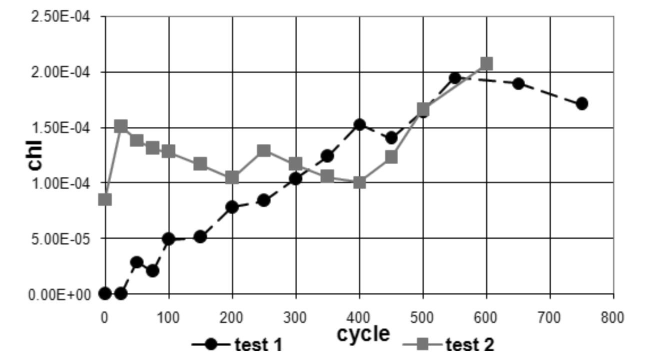

Fig. 4. Evolution of the average value of the parameter χ as a function of the damper cycle number

The ratio of the mass of gas in bubbles to the mass of the emulsion, at given pressure and temperature, is therefore equal to

$$\chi = \frac{m_{G\_bubbles}}{m_E} = \frac{P \cdot V_{G\_bubbles}}{R \cdot T \cdot m_{oil} + P \cdot V_{G\_bubbles}} \quad (20)$$

where m_E is the mass of the emulsion; T [K] is the temperature; P [Pa] is the absolute pressure; R [J/(kg K)] is the specific gas constant, which for nitrogen is R_N=296.80 [J/kg K].

Figure 4 shows an evolution of the parameter χ averaged over sequence of two tests performed on the transparent damper. The number of damper cycles in this figure represents single upward and downward movement of the damper piston. Black dashed-line represents the first part of the test, i.e. the test with the fresh oil; while the gray line represents the second part of the test, i.e. the continuation after approximately 2 hours of cooling the unit down and without changing the oil. The parameter χ was taken in the MATLAB model to simulate damper operation with fully developed oil-gas bubbles emulsion. The results (force-displacement characteristic) obtained from the Matlab model simulation (Figure 5) and physical tests (Figure 6) indicate that the force loss during the initial phase of the compression motion caused by aeration (presence of gas bubbles in the oil) can be successfully modeled i.e. the parameter χ in Figure 5 can be referred to the cycle number in Figure 6. A sine wave excitation with 1 [m/s] velocity and 50 [mm] amplitude was applied to these models.

3.3. Aeration measurement and controllable parameters

Analyses of the results were performed using Six Sigma tools (DOE analysis). The team decided to start experiment by varying 2 most controllable and measurable parameters, i.e. the initial oil volume (V_{oil}) and the initial (before saturation) gas pressure (p_{fill}). The reason supporting such a choice of parameters comes directly from the Henry's law of solvability which relates the mass of dissolved gas to the volume of the liquid and the pressure of the gas. There are several different ways (the so called aeration measures) to quantitatively describe degree of aeration and the effects it has on the performance (the so called damping lag or free stroke). For the purpose of this presentation the free stroke (μ), which is the length of the displacement from the maximal displacement to the point of change of the sign of the second derivative of the force signal (the inflection point), was used.

The major difficulty with this approach is the influence of the measurement noise on the inflection point selection method. Iterative use of the Savitzky-Golay filter permits for the selection of only the most relevant inflection point. Filter parameters (degree k of the polynomial and the size n of the frame, for details see [11]) were adjusted iteratively to obtain a single inflection point from which the free stroke was calculated, relative to the length of the test stroke.

The main task and focal point of this paper, namely design optimization by minimizing (negative) aeration effects is achieved by selecting such a combination of parameters, i.e. the gas pressure and the oil volume, for which a chosen aeration measure yields minimal value [3].

The actual selection process is performed visually based on the contour plot of the best-fit surface, the so called response surface, created separately for each set of DOE measurements. Example of response surface contour plots for different aeration measures are shown in Figure 7 (the response surface of the length of the free stroke) and Figure 8 (the response surface of the parameter χ). Such an approach to optimization is feasible thanks to the small number of parameters. In case of higher-dimensional parameters spaces other visualization techniques or more automated optimization algorithms may be used.

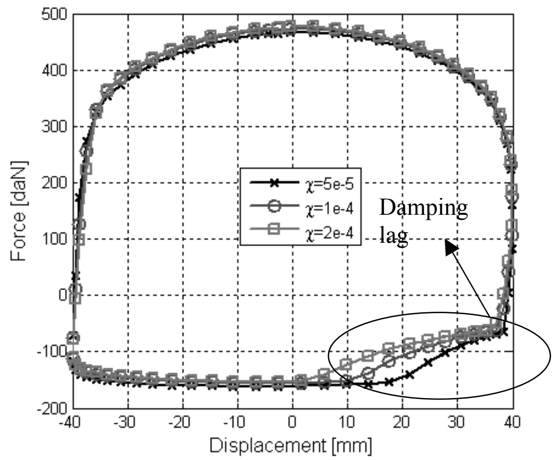

Fig. 5. The force-displacement characteristic of the transparent damper simulations

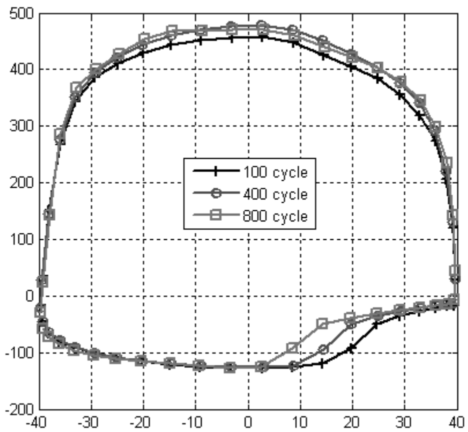

Fig. 6. The force-displacement characteristic of the transparent damper physical first test

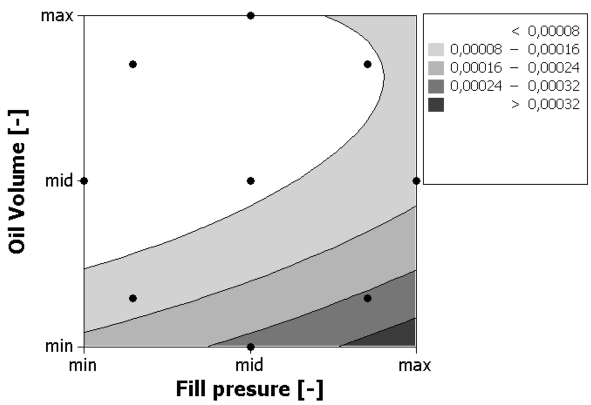

Fig. 7. Contour plot of the response surface of the parameter χ vs. the oil volume V_{oil} and the fill pressure p_{fill} obtained after 500 cycles

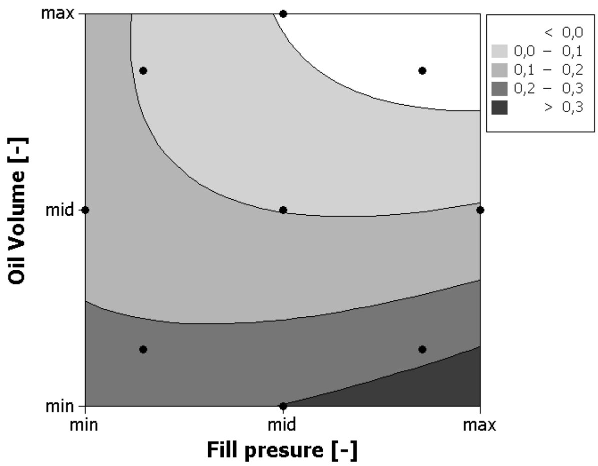

Fig. 8. Contour plot of the response surface of the free stroke length μ vs. oil volume V_{oil} and the fill pressure p_{fill} obtained after 500 cycles

In terms of accuracy, one should note that this optimization approach yields a region of the parameter space and permits other criteria (like cost or durability requirements) to be easily, that is without additional measurements and analyses, imposed.

In order to validate the numerical model for which the parameter χ is an input, the correlation between the value of the χ and the free stroke length measure corresponding to this value of χ, for both the experiment and the simulation, was investigated. Figure 9 and Figure 10 show the correlation plots of the free stroke in the function of the parameter $\chi(V_{oil}, p_{fill})$ indicating a good agreement between the model and the experiment.

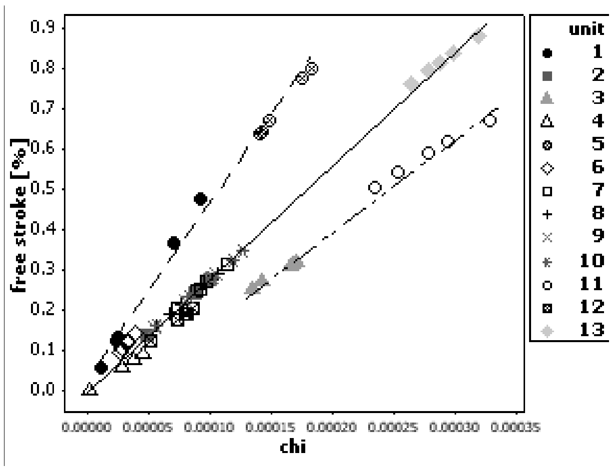

Fig. 9. Correlation plots relating the length of the free stroke to the value of χ in simulations of shock absorber units

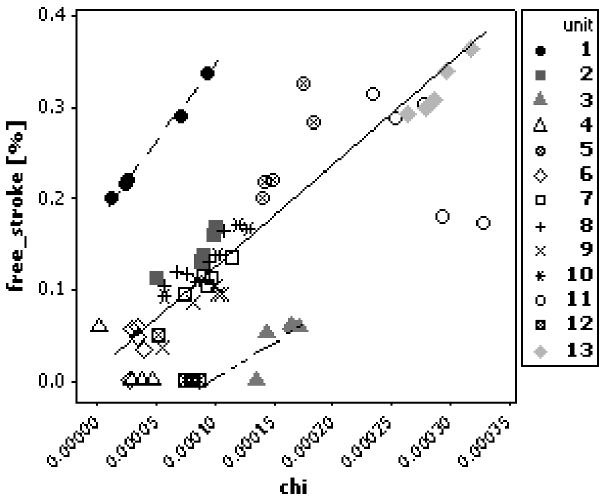

Fig. 10. Correlation plots relating the length of the free stroke to the value of χ in experiments on shock absorber units

One should realize that had the length of the free stroke depended solely on value of the parameter χ, all the data points would form a single, straight line. One notices that the regression coefficient depends on the particular combination on the parameters (marked as the unit index - this notation should not be misleading since each test with a new combination of the oil volume and the gas pressure required changing the oil and pressurizing the transparent damper).

Differences between experiment and simulation are attributable to errors in the oil height measurements as well as unstable behavior of the unit for some combination of parameters (correlation between the parameter χ and the free stroke in the experiment is acceptable, at the level of 0.8 while the liner model explains approximately 64% of the variation). Moreover, the model does not take into account the loss of viscosity caused by the increase of temperature and the emulsification - in reality such a loss of viscosity changes the P-Q curves, while in the model P-Q curves are held constant.

4. Conclusions

Modeling the dynamics of bubble formation and transport is a task very difficult for several reasons. Most important ones are difference between time scales in which aeration processes occur (order of minutes) and the time scales of oil flow through a damper (order of seconds), existence of uncontrollable parameters on which bubble size depends and the bubble size itself (e.g. oil impurities and sharp edges), re-absorption of gas from bubbles surface, etc. At present, the most efficient approach is to use experimental, average characteristics of the oil/gas emulsion (ratio of the mass of gas in bubbles to the mass of the oil-gas bubbles emulsion) at model initialization and predict the force response of the modeled shock absorber. During the work on this paper authors realized the level of complication of the aeration phenomenon and decided that more reliable data and data processing algorithms are necessary before any attempt to create a prediction tool can be made. It was decided that by using Six Sigma methodology and carefully organizing data gathering and analysis process two goals, namely the continuous improvement of the product and the aeration model identification, can be achieved.

Six Sigma tools (DOE analysis) of the measurement and simulation results indicate that there might be an optimal choice of values of the controllable parameters (there exists a local minimum of the aeration measure in function of the gas pressure and oil volume), and the exact values of the optimal combination of the parameters depend on the aeration measure. One should however notice that in some cases the minimum in only global, i.e. the minimum of the aeration measure is located on the boundary of the region of the parameter space for which the statistical model was fitted. In general, conclusions drawn from the response surface are not valid beyond these boundaries.

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