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SIMULATION OF A MICRO SHIFT VALVE WITH IMPACT ACTUATION

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Abstract. *A new concept of an impact actuated micro shift valve is presented. The impacts are transmitted to the interior of the valve through the casing. In order to predict the energy transmitted into the valve, the most important effects are discussed and two elastic multibody models using the Hertzian contact law are introduced and compared to a full finite element simulation. A simpler model with one degree of freedom for an elastic plate as transmission element proves to be too crude. But a more sophisticated model with axisymmetric finite elements for the plate shows good agreement. The simulations are compared to experiments performed with a scaled model for two different cases. The experiments show that so far neglected effects like plastic deformation occur and must be considered in the simulation if accurate predictions are required.*

1 INTRODUCTION

In many technical applications shift valves are used to control fluid flow. One important aspect for some uses of shift valves is media separation, meaning that there is no contact between the environment and the fluid within the valve. This is important, e.g. in cases where the purity demands for the media are extreme or the fluid is harmful. Usually this is achieved by using deformable membranes which either block or enable the flow. The disadvantage with membrane valves is that they keep the deformed state only if energy is supplied to the system which makes them less attractive if this position must be maintained for long times. A new concept is presented in the following based on impulse actuation including the simulation of impulse transmission into the valve. The simulations are performed for two different parameters and compared to experimental investigations.

2 THE CONCEPT OF AN IMPULSE ACTUATED SHIFT VALVE

The proposed concept of a shift valve with separated media is based on a body inside the valve which can be in two distinct positions. One allowing fluid flow and one blocking the fluid flow as depicted in Fig. 1. It is the basic idea to transmit energy by an impulse wave or deformation from actuators ("open" and "close") into the valve chamber to the sphere which will switch its position. The spring must be stiff enough to keep the sphere at its position but not too stiff such that shifting is not impaired.

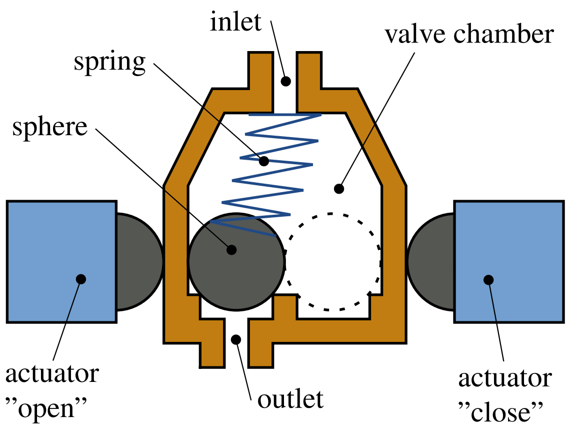

Figure 1: Concept of the shift valve

This concept is promising for applications where media separation is relevant and the periods in the same state are relatively long. It has already demonstrated as a prove of concept prototype that it can work nicely. It also allows the creation of very small valves with a radius for the inner sphere of approximately 1.5 mm. The following part will give a short description about the modelling of the impact.

3 MODELLING THE IMPACT IN AN ELASTIC MULTIBODY SYSTEM

The most important part in the simulation of this system is the impact and the impulse transmission through the casing of the valve. To achieve an understanding of the impact behaviour and the parameters influencing it, a very flexible and fast model which also allows optimisations and parameter studies must be created. Usually detailed analyses of impact are done with finite element simulations but here an elastic multibody system is used in order to reduce the model complexity and simulation time. A finite element simulation is only used to validate these simulations.

The model used for the simulations is shown in Fig. 2. The impact being the beginning of a shift process is modelled using two sphere-like bodies and a circular plate. The outer body has the initial velocity v_{o0}. The simplification of the actuator as a sphere with initial velocity is chosen in order to separate the impact process from effects that an actuator such as a piezo stack or a bending transducer has. For experimental verification a test-bed has been set up, see Fig. 3.

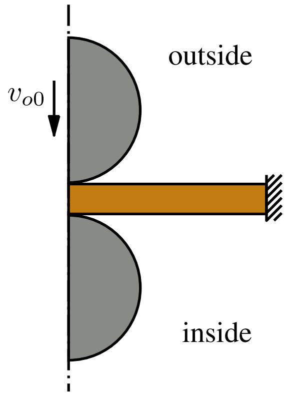

Figure 2: Model for impact simulations

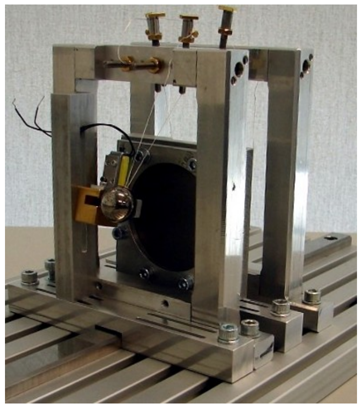

Figure 3: Test-bed for experimental verification

There are two contacts at the same time, one between the outer body and the plate and the other between the plate and the body inside the valve chamber. The most significant result of the simulation is the energy transmitted from the outer body to the inner body defined as the efficiency

$$\eta = \frac{T_{i1}}{T_{o0}} = \frac{m_i v_{i1}^2}{m_o v_{o0}^2} \, . \tag{1}$$

This efficiency is the ratio of kinetic energy T of the colliding bodies with mass m and velocity v. The subscript denotes the state just before impact (0) and after impact (1). Therefore, the contact must be modelled in a detailed manner. It will be shown that the full 3D finite element model is not necessary to capture all relevant effects and give high accuracy. An elastic multibody model will be refined in several steps until the model is accurate enough.

But first the effects are discussed which can be expected in impact processes which are highly dynamic and nonlinear. The most important effect is, of course, the elastic deformation of the impacting bodies. We separate them into two different parts, first the global deformations like vibrations or wave propagation within the bodies and second the local deformations in the contact region. The local deformations are influenced by all sorts of effects which are not captured in standard linear elasticity which is also used in the Hertzian contact law [1], [2].

Plastic deformations occur almost always and are specifically important if multiple plastic impacts occur at the same location. As a further complication due to the high speed and strain rate $\dot{\varepsilon}$, nonlinear material effects like visco-plasticity must be considered if plasticity is modelled, see [3] for a detailed analysis of these effects. Global effects like vibrations and wave transmission are very important in most cases except for bodies like spheres which can be modelled as rigid bodies. Elastic bodies can describe wave effects if the model is detailed enough to capture the waves.

The simplest approach for modelling impact in multibody systems is the use of the coefficient of restitution. This is not useful here because it is basically the result of the simulation and, therefore, shouldn't be used as input. This means that the impact must be simulated continuous and the simplest model that comes to mind is to put springs between the bodies and model the elasticity of the clamped plate with one degree of freedom. The following proposed models are already improved over simple linear springs using nonlinear springs according to the Hertzian impact law (A). Another model (B) is based on an improved description of the plate with axisymmetric plate elements. The two models (A) and (B) are simulated using the scientific libraries SciPy and SymPy. The reference solution for the simulations is a nonlinear explicit finite element simulation using Abaqus.

3.1 Model A: Elastic plate theory and Hertzian contact

The simplest investigated model is based on elastic plate theory modelling the global elasticity of the clamped plate with one degree of freedom and the local elasticity with nonlinear springs. This model is depicted in Fig. 4.

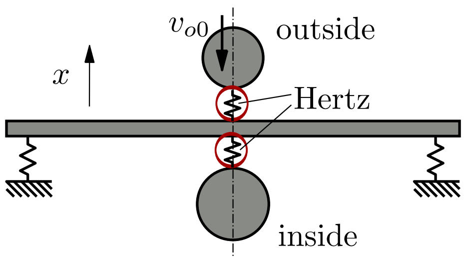

Figure 4: Simplified model using elastic plate theory

The nonlinear springs representing the local elasticity are modelled with the Hertzian contact law, see e.g. [1] or [2]. The clamped elastic plate is modelled as a point mass with one degree of freedom connected by a spring to the environment. The mass and stiffness are determined using elastic plate theory. The stiffness c_P is calculated according to the static force displacement relation from [4]. The point mass

$$m_P = \frac{c_P}{\omega_1^2} = \frac{16\pi D}{r^2\omega_1^2} \quad \text{with} \quad D = \frac{Eh^3}{12(1-\nu^2)} \tag{2}$$

can be found by using the first eigenfrequency ω_1 of an axisymmetric plate, see [2] and the already determined stiffness c_P. The term named D is the flexural rigidity of the plate, E is the Young's modulus, ν the Poisson ratio and h the thickness. This model can be seen as an approach to separate local elasticity effects in the contact region from global elasticity effects of the whole body which is the plate here. The spherical bodies on either side of the plate are modelled as point masses. It will be shown later that the approximation of the plate with one degree of freedom is not satisfying. Therefore, the plate must be modelled more detailed by using more degrees of freedom in model (B) which proves to be very close to the full FE model.

3.2 Model B: Elastic plate using finite elements and Hertzian contact

In order to improve the accuracy of the dynamic behaviour of the plate, additional eigenmodes or more general degrees of freedom must be added. This is done by using finite elements as depicted in Fig. 5.

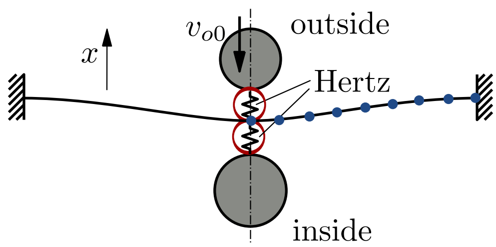

Figure 5: Improved model for the plate using finite elements

This is realised in two steps. First the plate is modelled with linear axisymmetric finite elements similar to those proposed in [5]. The mass matrix which is defined in [5] is derived according to [6]. The resulting linear system for the plate is transformed to modal coordinates. This gives the advantage of much faster time integration and the possibility to perform a modal reduction by transforming the model to a subspace of the modal coordinates. Basically high frequency eigenmodes are assumed to be negligible. By doing that, the time integration is even faster and the influence of the number of eigenmodes can be studied. This approach of using a modal model for the global behaviour and a local contact force law such as the Hertzian law for the local behaviour is described in [7] and [8]. However, the transformation to modal coordinates is not the only possibility, in [9] more powerful transformations are presented. By using such methods the accuracy can be increased or the number of ansatz functions decreased.

3.3 Finite Element Model

The reference solution is computed with the commercial finite element program Abaqus. The simulation is an explicit dynamic simulation with nonlinear deformations but using a linear elastic material model. The model is set up with axisymmetric elements which simplifies meshing with quadrilaterals. The elements are linear and a kinematic master-slave contact algorithm is used. The mesh is refined in the contact region so that approximately 10 to 15 elements are across the contact radius according to [7] and [8]. The largest elements are small enough to capture wave effects which might occur. A typical mesh with the stress distribution is shown in Fig. 6.

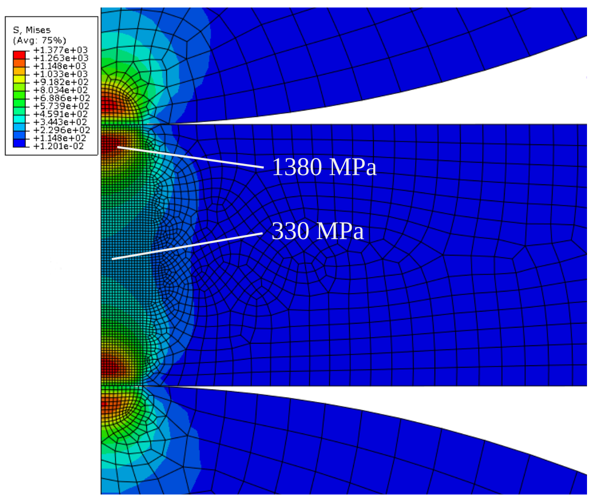

Figure 6: Example of a mesh and the resulting stress distribution

4 SIMULATIONS

The two simplified models are compared to the full transient finite element (FE) simulation. This allows first to determine the accuracy of the simplified models compared to the full FE simulation which has fewer simplifications. Second one can show whether the introduced simplifications in the models (A) and (B) are valid. This allows to determine the required model complexity and investigate the influence of parameters such as material and geometry of the bodies.

4.1 Comparison of the different simulation models

For comparison a scaled model is used which is also used for the experiments. The scaled model is chosen in order to have more accurate experiments. The geometry is consistently scaled such that the large model behaves similar to the small valve. If the methodology for simulating such models is validated, smaller models can be simulated without further experiments. The material in this model is steel for all bodies with density 7780 kg/m^3, Young's modulus 210 GPa and Poisson ratio 0.3. The material and geometrical data of the chosen model is summarized in Table 1 for two different sphere radii and their respective initial velocity.

Table 1: Geometrical data and initial condition of the scaled model

	outer sphere	plate	inner sphere
radius (mm)	6.35 / 10	33	6.35 / 10
thickness (mm)	-	2	-
initial velocity (mm/s)	-437 / -550	0	0

This scaled model is simulated as transient fully nonlinear linear-elastic FE model and also with models A and B. The simulation results are shown in Fig. 7 for two different sphere radii and different initial velocities of the outer sphere. The three figures on the left are for the larger spheres with higher initial velocity, the figures on the right are for the smaller spheres with the lower initial velocity. Each case has three plots with the same time scale where all three bodies simulated with the three simulation models are shown. The plots of the displacement and contact force are annotated as it is difficult to see which line belongs to which body. On the first glance it is apparent that model B is in good agreement with the FEM and model A is obviously too simple. The resulting efficiencies as defined in Eq. (1) are displayed in the grey boxes. Again it is clear that model A and the FEM are in good agreement but model B is not good enough.

A more detailed investigation shows that model B has a slightly longer contact time and lower contact forces. This is of course related, the main influence here is the stiffness of the plate in longitudinal direction which is neglected. This influences the stress distribution of the Hertzian contact law which assumes an infinite half space. In Fig. 6 the problem can be seen clearly. The stress between the two contact points is not zero, therefore the stress distributions influence each other. The assumption in the Hertzian law would require zero stress between the contact points.

The stresses occurring in the simulation models are well above yield, so that plastic deformations occur which are neglected. Elastic, ideal-plastic behaviour is not precise enough due to the extremely high strain rate $\dot{\varepsilon}$. Therefore, the material must be modelled visco-plastic. Further, multiple contacts occur which lead to an effect called shakedown which basically deforms the colliding bodies plastically during each impact until the geometry has changed enough so that

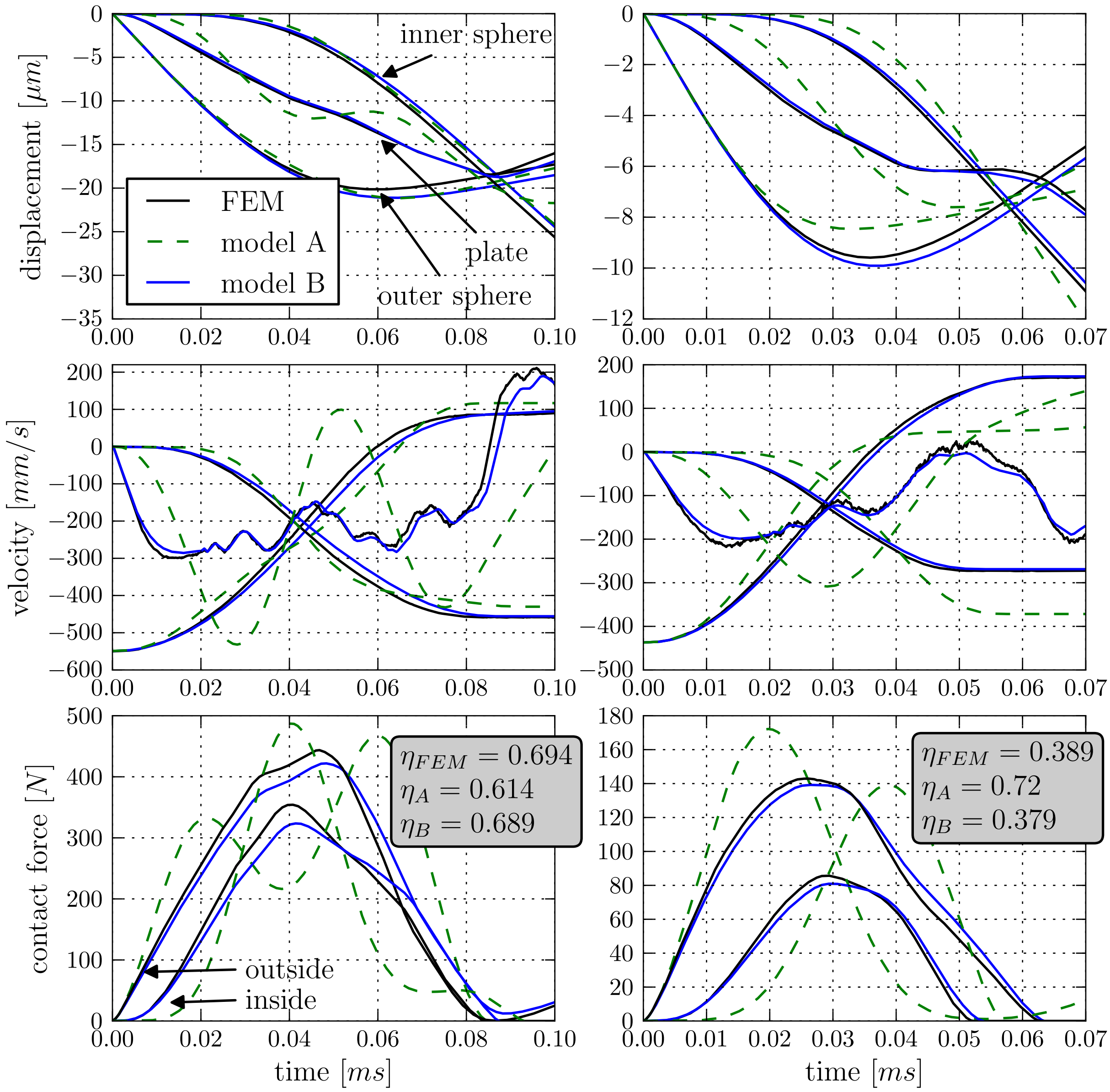

Figure 7: Simulation results for $r = 10mm$ spheres (left) and $r = 6.35mm$ spheres (right).

the stress is below yield and the impact is elastic again. Visco-plastic material modelling and multiple plastic impacts are investigated in detail in [3].

5 EXPERIMENTAL VERIFICATION

In computational models simplifications are necessary and useful to reduce the model complexity. However, the assumptions must be verified. So it is good practice to compare simulations to experiments wherever possible. An experimental set-up is used similar to the set-up used by [7] which is based on Laser-Doppler-Vibrometers (LDVs) to measure displacements and velocities at high frequencies ($f < 100kHz$).

The set-up is shown in Fig. 3, however, the lasers are not visible on the picture. The plate which is ideally clamped in the simulation is mounted on an aluminium frame in the centre. The two spheres are suspended by thin nylon wires which are connected to the spheres by small copper plates which are squeezed on them. The sphere which represents the actuation of the valve is released by a flap which is released by an electromagnet. For lower velocities a different mechanism is used. A Venturi tube is used to create low pressure which draws the sphere to a seal on a thin pipe. It is released by stopping the air flow through the Venturi tube.

The LDVs measuring the displacements and velocities are made by Polytec GmbH. One LDV is placed on the outside, measuring the sphere at the impact. This means that this sphere moves into the laser beam just before impact. The second LDV is placed on the opposite side which represents the interior of the valve, also measuring the centre of the sphere. Using both LDVs gives a displacement signal and a velocity signal for each sphere.

5.1 Analysis of the measurements

A typical measured signal is plotted in Fig. 8. The displacement is only a relative measurement, but its slope can be compared to the velocities before and after impact.

The velocity signals clearly show the behaviour one expects from the system. The outer sphere has an initial velocity v_{o0} which is constant within the short time-frame displayed in Fig. 8 until the impact begins. After the impact the outer sphere has a rebound velocity v_{i1} which is in general not zero but should be as small as possible for minimal losses. The displacement signals also shows the expected characteristics - a negative slope where the velocity is negative and vice versa. The smoothed jump representing the impact in the velocity signal is a turning point in the displacement signal. In order to calculate the ratio of kinetic energy as defined in Eq. (1) the two velocities v_{o0} and v_{i1} must be evaluated as accurate as possible.

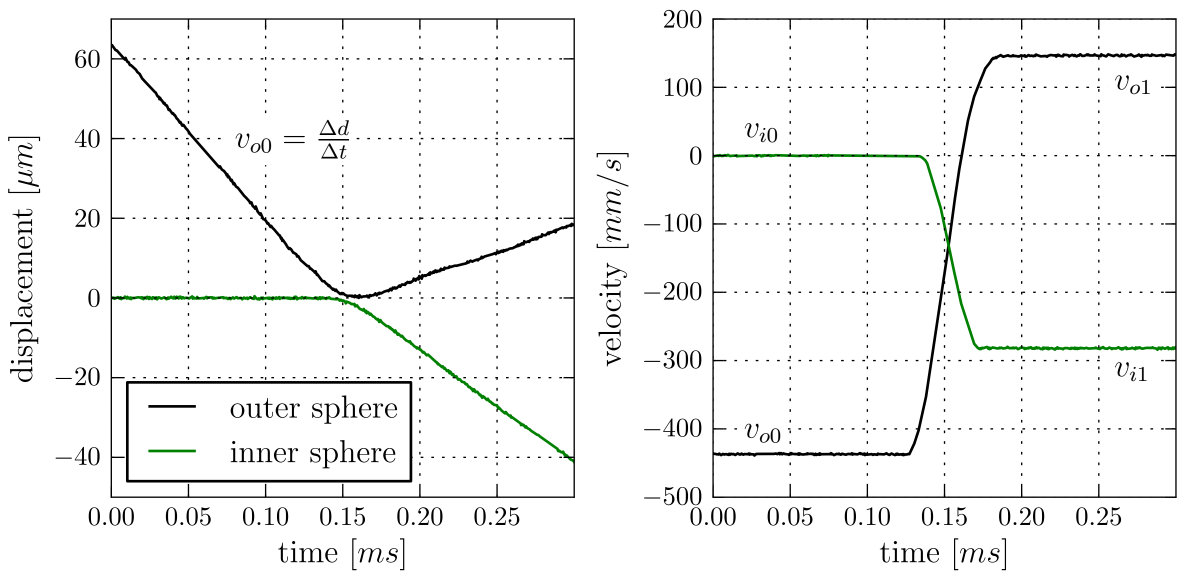

Figure 8: Displacement and velocity signals measured by LDVs.

We can extract the velocity of the bodies either from the displacement signal by differentiation or directly from the velocity signal. However, the velocity signal always has an offset and can not be taken directly to determine the velocity. The velocity signal in general has a higher resolution and less noise, so the velocity is preferred over the displacement signal. The reason for the offset is a high pass filter in the Controller of the LDVs which results in a drifting velocity signal while the velocity should be constant. In the short time of the measurement the offset is mostly constant. The signal of the inner sphere can be corrected easily because it is known that the velocity must be zero at the beginning. The signal of the outer sphere is more difficult to correct. We determine the numerical differentiation of the displacement signal and use the average of the difference of the differentiated displacement and the velocity. This average is then the offset for the velocity signal.

5.2 Comparison to the simulations

The impact obtained from simulation and the measured signal are shown in Fig. 9. For comparison with the FE simulation the time axis is aligned to match the signals. The displacement signals which also have an arbitrary offset are also aligned in vertical direction to match the simulation results. We can see that it shows similar behaviour but the contact duration and the velocities after contact differ slightly.

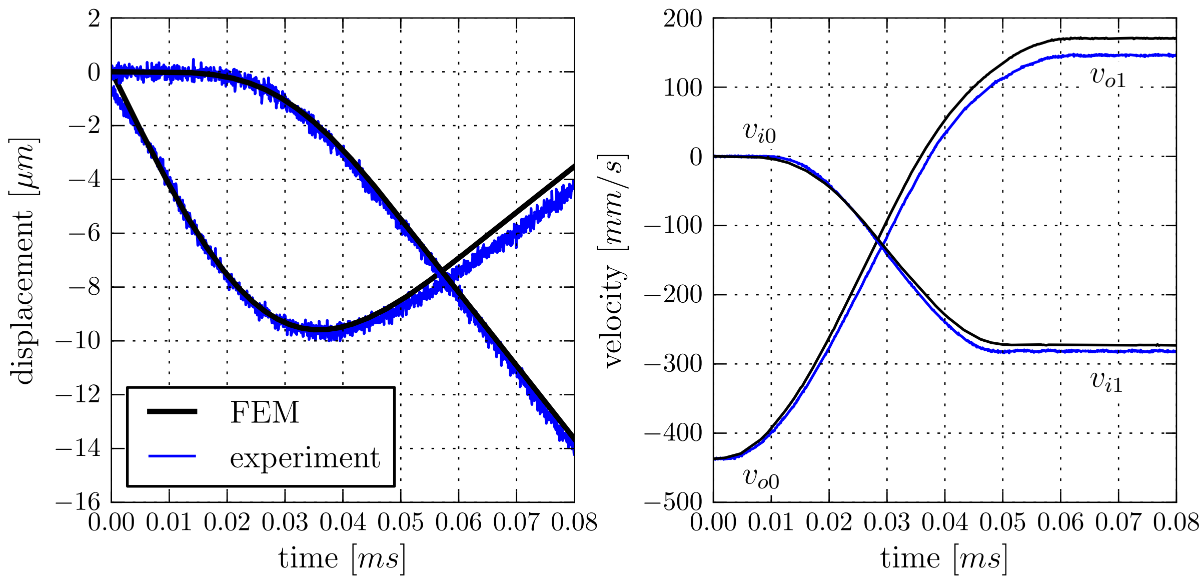

Figure 9: Comparison of the FEM simulation and the experiment.

The efficiency as defined in Eq. (1) for repeated impacts on the new and unused plates is not constant, see Fig. 10. The efficiency for sphere radii $r = 10mm$ is much lower at the first impact and increases with each additional impact until the steady state efficiency η_{EXP} is reached. The reason for that effect are permanent changes, which are most likely plastic deformations. This effect is studied in detail in [3] where the same behaviour occurs. Normally impacts with plastic deformation have a lower coefficient of restitution and, therefore, also a lower efficiency in our case. So it is correct that the predicted efficiency η_{FEM} is higher than the first data point. The steady state efficiency η_{EXP} should be even higher than the predicted efficiency η_{FEM} since the deformation is elastic again and the new deformed geometry has a higher contact radius which typically leads to higher efficiencies. However, the predicted efficiency is not below the steady state efficiency which must be related to another effect or simplification in the model.

The second case with $r = 6.35mm$ has less accurate measurements due to the smaller sphere and the different release mechanism which was used to get a lower velocity. Plastic deformations also occur here but slightly less due to the lower impact velocity. The first data points also show an increase in efficiency as expected. Here the simulation is below the steady state results of the experiments which was expected. The higher variation of the points might be that the impact point is not always exactly the same so that some of them are again influenced from plastic deformations.

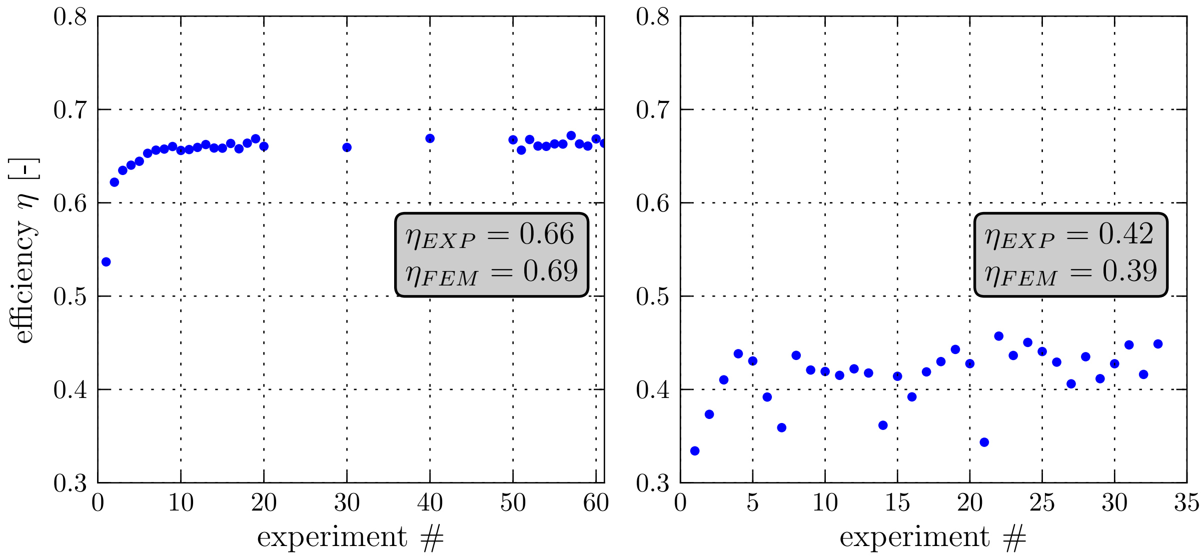

Figure 10: Efficiency of multiple impacts for the two cases with $r = 10mm$ spheres (left) and $r = 6.35mm$ spheres (right).

There are several reasons why the experiments differ slightly from the simulation.

- The plate can not be clamped ideally in the experiments so that there is a vibration of the whole aluminum frame.
- The plate is not hit exactly in the center.
- The alignment of the spheres is not exactly on one line, so the impact is not exactly central.
- The suspension of the spheres certainly has an influence on the movement of the spheres.

6 CONCLUSIONS

It is shown that the elastic multibody model captures the most prevalent effects by giving results similar to the FE simulation. One result of this modelling approach is that wave effects in impact direction within the plate can be neglected because the simplified models do not contain this effect but give accurate results. The simulations generally agree with the experiments. In order to increase the accuracy, the material must be modelled visco-plastic and several successive simulations must be performed to get the steady-state efficiency. Further development includes the possibility to describe a bending transducer with a simple beam model which can be used instead of the outer sphere. Additional to the dry contact, fluid flow between the casing of the valve and the inner sphere can be simulated and experimentally investigated.

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