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Virtual Prototyping Methodology: Predicting Start Stop Movement of Pump Unit by Simulation

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ABSTRACT

In a typical, stable operation of a hermetic compressor, a pump unit mounted on a suspension system will constantly vibrate within its shell (housing). During starting and stopping of the compressor, the significantly increased forces in cylinder chamber and rotor-stator coupling will act on the pump unit which will have an increased dynamic response in form of large displacements in all directions. This paper investigates short, but unpredictable starting event and its effect on the pump unit movement. It proposes a fully virtual approach using numerical simulation tools in order to capture and define starting movement and use the data to benefit compressor development.

Combination of modern numerical tools was used to build a numerical model, define boundary conditions, calculate pump unit movement and lastly to map a gap profile based on the results. A part of results was also verified with experimental measurements. Such approach is valuable to the compressor development because it creates a virtual prototype which is cheaper, faster, more flexible and has a higher repeatability compared to alternative, hardware-oriented approaches. Methods and results from this work can be used to design a compressor shell more efficiently. Such development would otherwise require multiple physical shell prototypes, that are expensive and complex to produce, especially in the early stages of a development project.

1. INTRODUCTION

A typical hermetic compressor contains inside of its shell a pump unit, which sits on three or four helical cylindrical springs. Additionally, the pump unit is connected to the housing via a discharge tube, which also has spring-like behavior. In the compressor, the pump unit has 6 degrees of freedom (DoF), 3 translational and 3 rotational. Its movement is mostly determined by the suspension chosen by the designers. While the compressor is running in a stable operating condition, the pump unit vibrates but its amplitude of movement is low and predictable. Gap between the pump unit and shell is relatively constant. Figure 1 shows a cross section view of such a compressor.

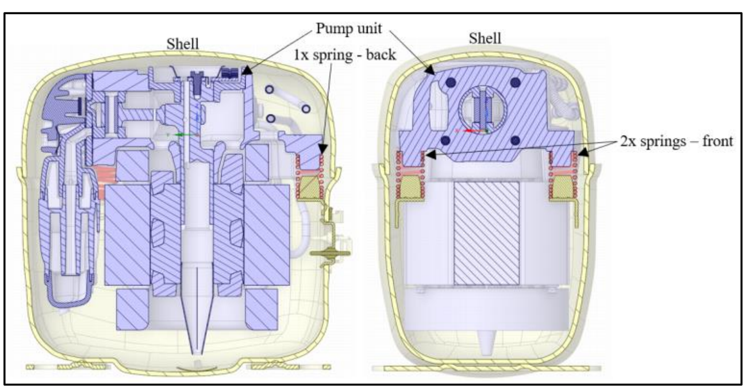

Figure 1: Schematic view: compressor with shell, pump unit and springs

When a compressor is started, the stator and rotor coupling convert electrical energy into mechanical rotational energy of the crankshaft. At the same time, the piston starts its reciprocating motion, moving along the cylinder walls from bottom dead center to top dead center, compressing gas inside the cylinder. As the cylinder pressure increases, forces transferred to the crank train components and crankcase are growing, leading to inertia and torque acting on the whole pump unit. Similar transfer of forces occurs when the compressor is stopped from operating condition. Here the remaining forces in the cylinder and the active inertia in the mechanism release their energy onto the pump unit, causing it to shake intensively before coming to a standstill. In both cases, significantly higher forces are imposed on the pump unit by the rapidly increasing motor torque and cylinder pressure than during continuous operation. The pump unit moves within the shell and at least partially consumes more of the gap between its stoppers and the shell´s inner contour.

The amplitude of this movement is determined by multiple factors, such as the motor torque characteristics, the gas conditions in the cylinder, the stiffness of suspension springs and discharge tube, and the inertia of the pump unit itself. The excitation and damping forces during start and stop are shown in Figure 2. In case of a correct suspension and shell design, the pump unit is not displaced enough to consume the gap between itself and the shell, in which case the stoppers would cause a mechanical impact with the inner shell wall.

This background of the pump unit movement was very well explained in a mathematical model prepared by Marriott (1998), where the author showed equations behind the slider-crank and motor models and combined them to calculate rotation and displacement of a compressor crankcase. Theoretical background of this is also described by research from Futakawa *et al.* (1980). Their conclusions based on theoretical and experimental analysis conclude that at starting, the difference between torque coming from motor and the torque coming from gas compression act on compressor body with transient vibrations, which are result of this varying and abrupt torque. Same occurs at stopping, but in this case only the left-over torque from gas compression will act on the compressor body displacement.

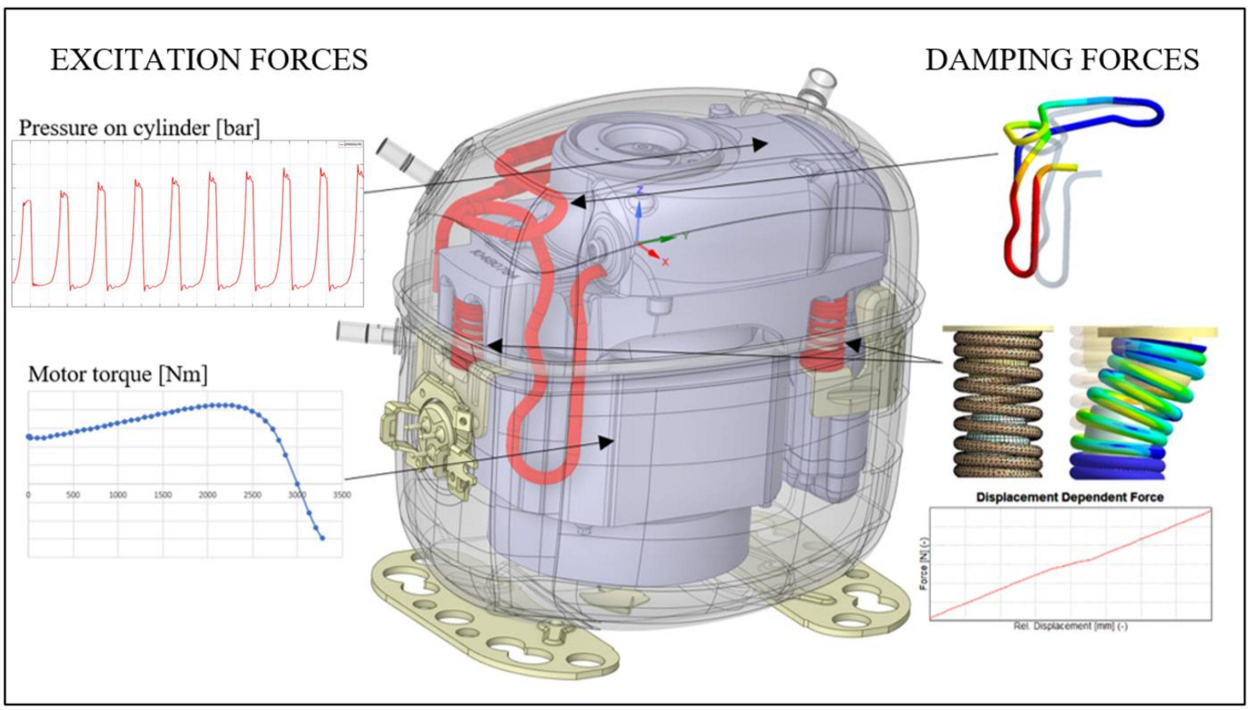

Figure 2: Excitation and damping forces affecting pump unit movement

Existing literature from Dropa de Bortoli *et al.* (2010), Elson (1978) and McWilson (1972), already showed in spring fatigue analysis that the stress on springs and the resulting displacements are the highest at start and stop events, with Elson (1978) specifically suggesting that during start stop the vibration amplitude of movement is three to four times larger compared to steady state operation. Study from Zaccone (2001) adds similar conclusions to this topic with a root cause investigation of failed compressor springs. Another analysis of transient motion produced in the compressor body during starting and stopping from Futakawa *et al.* (1980), contributes to the theoretical background behind the presenting paper. Authors compared theoretical and experimental results of shear stresses in springs and connected them to acting fluctuating torque during start and stop operations. In fact, the oldest similar investigations found during the literature review for this paper were done already by Gatecliff (1974). He combined analytical and experimental results to propose a model combining slider-crank dynamics, forces from gas pressure and motor torque. Objective of his research was to describe a six-degree-of-freedom motion of a compressor crankcase. Lastly, Marriott (1998) built upon this mathematical model and has proposed another calculation of compressor crankcase movement at the startup phase.

The objective of this paper is to use state of the art numerical tools to develop a virtual prototype that supports the design decisions in compressor development, providing boundary conditions of maximum pump unit movement at starting (e.g., benefit to shell development). The paper proposes a method to predict pump unit movement completely numerically and investigates at least partial calibration and validation by experimental results. Commercial Multi-body dynamics (MBD) software and Multi-Physics Platform are used to perform numerical simulations, calculate the pump unit movement and to determine the necessary boundary conditions, like gas pressure inside the cylinder. Commercial FE (Finite Element) tools are also employed to prepare the compressor parts for the MBD software, post-process results and determine gap to shell considering pump unit start motion. This result can be used to design multiple compressor components and parameters related to displacement and vibration before the hardware prototypes are even manufactured.

2. THEORETICAL BACKGROUND

2.1 Effects of starting of the pump unit

It is important to remember that the pump unit movement is limited by a suspension, which is rigidly connected to the compressor shell. This suspension absorbs the energy and reduces the movement to acceptable levels. In a typical compressor, springs and discharge tube act as a suspension together and are a source of damping forces and elastic potential energy, as shown in Figure 2. Similar principles apply in the technology of internal combustion engines. Study by Adhau and Kumar (2013) shows how an engine mount is designed to act as an absorber of vibration and isolator of unbalanced excitation forces (torque from reciprocating crank train parts and pulsation coming from fuel combustion in the cylinder).

2.2 Pump unit dynamics

In a multi-body system, such as pump unit within a compressor, translational and rotational motion can be distinguished. Furthermore, one must consider small motions (vibrations) and global motion with high amplitudes. The mathematical model can be deduced using Newton's equation of momentum and Euler's equation of angular momentum. The bodies can be handled as rigid mass points or further discretized with the help of an FE tool into multiple regions (elements) to get more accurate results. To consider the elastic deformation of the bodies is also possible.

The motion of the compressor parts is described by the vibration equation (1), where M, K and D are the mass, stiffness and damping matrices and the variables q, $\dot{q}$ and $\ddot{q}$ denote the position, velocity and acceleration vectors, respectively. The matrices are built up from the mass stiffness and damping properties of the individual mass points. The term f in the right-hand side consists of the connecting joint forces, moments and the external loads, meanwhile p^* contains the nonlinear inertia terms, see Offner (2011).

$$M \cdot \ddot{q} + D \cdot \dot{q} + K \cdot q = f + p^* \tag{1}$$

In the model of this paper the bodies are connected by joints that are mathematically represented by spring-damper elements. Employing elasto-hydrodynamic connection elements for main bearings and piston-cylinder contact would be possible but is not necessary for given analysis, and it would increase computational resources (e.g., time).

The joint force in equation (2) is a combination of stiffness and damping forces where Δx and $\Delta\dot{x}$ are the change in distance and relative speed of the connected nodes and k_{joint} and d_{joint} are the stiffness and damping coefficients, respectively. The nonlinear spring-damper characteristics, k_{joint} and d_{joint}, are determined by equations (3) and (4), where the parameters k_0, k_B and d_0, d_B define the nonlinear behavior.

$$f = k_{joint} \cdot \Delta x + d_{joint} \cdot \Delta\dot{x} \tag{2}$$

$$k_{joint} = k_0 \cdot \left(\frac{k_B}{k_0}\right)^{\left|\frac{\Delta x}{x_B}\right|} \tag{3}$$

$$d_{joint} = d_0 \cdot \left(\frac{d_B}{d_0}\right)^{\left|\frac{\Delta x}{x_B}\right|} \tag{4}$$

For further details on the theoretical background please refer to AVL (2021).
The MBD model used as numerical method in this paper consists of the main flexible parts below:

- Piston assembled with piston pin,
- Connecting rod,
- Block assembled with valve plate, cylinder head and stator,
- Crankshaft assembled with oil pump, counterweights and rotor and
- Suction muffler.

These parts perform:

- global motion (with the pump unit),
- rotation and translation as part of their functionality (piston, connecting rod, crank shaft) and
- elastic deformation.

Spring holders on the block are connected to the rigid spring holders at the bottom of shell by three springs. In the MBD model this system is modeled by spring-damper elements. Similarly, the discharge tube, modelled again by spring-damper element, connects the discharge chamber of the block to the rigid shell. While investigating the numerical methodology for this paper, both elastic and rigid bodies were used, and it was concluded that there is no significant difference in the two cases regarding global motion of the pump unit. Only benefit in consideration of bodies elastic properties is that some high frequency torsional vibrations of the crank shaft can be observed.

2.3 Boundary conditions

The developed virtual prototype consists of the MBD model which is built from the compressor parts with their corresponding material properties and the joints connecting the bodies. However, in order to be able to run a simulation and predict the pump unit movement, certain other physical properties must be provided as well. These include the suspension spring's force versus displacement characteristics and damping, stiffness and damping of the discharge tube, torque characteristics of the electric motor over crank shaft speed and the gas pressure inside the cylinder. These are the most important boundary conditions of the virtual prototype. The sources of these boundary conditions can be different, like numerical simulation, analytical calculation, or experimental data.
For this paper, different approaches are used to define boundary conditions. The cylinder pressure is simulated by a Multi-Physics Platform that considers refrigerant gas properties, condensing and evaporating temperatures of the given operating condition, motor torque characteristics and the compressor's geometrical properties like stroke, bore and eccentricity. For the purpose of experimental validation actual measured cylinder pressure is used additionally as boundary condition to achieve best possible comparison between numerical and experimental methods. In all cases, initial position of the piston is at the bottom dead center (BDC), marked by -180° crank angle. The motor torque curve is obtained from experimental torque measurement at constant voltage. It is defined as torque over relative velocity of stator and rotor. Following figure shows two main inputs for pump unit excitation: motor torque and cylinder load (both normalized).

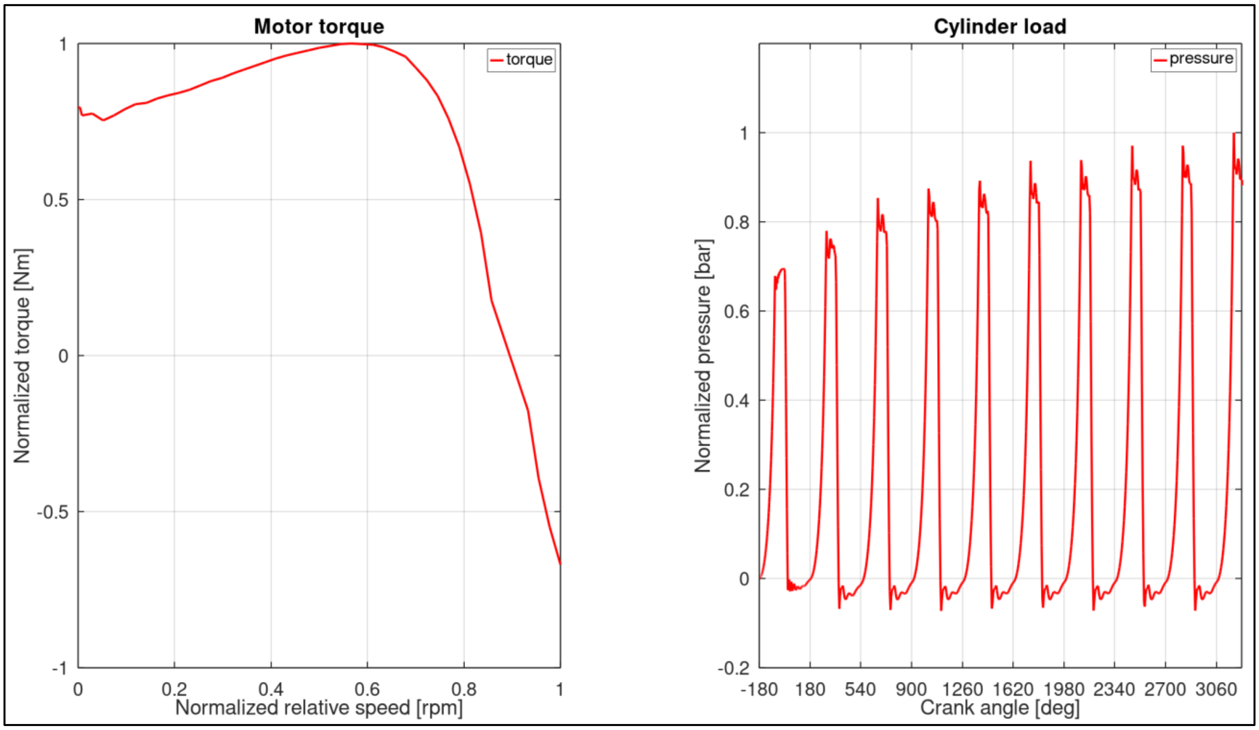

Figure 3: Motor torque and cylinder load used for numerical analysis

On the other side, the pump unit is sitting on three springs which, together with discharge tube, act as suspension that limits the excitation of pump unit movement. Figure 4 shows the FE model of spring and an arbitrary and normalized force versus displacement graph.

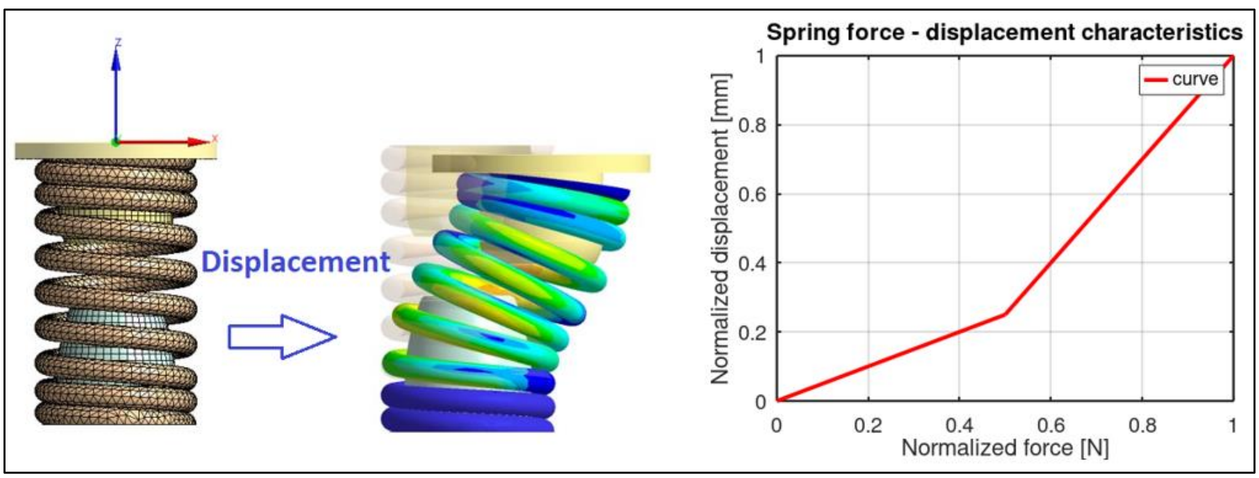

Figure 4: Force versus displacement function in a spring

Expected results of the pump unit motion are lateral and torsional motion with an oscillating function. This expectation corresponds to the investigation of spring fatigue from Dropa de Bortoli, *et al* (2010). Authors of this investigation mentioned limitations and proposed improvements, both of which the current paper is taking into consideration. Their analytical approach was limited by no interaction between springs and the stoppers, universal spring motion for all springs, only two directional motion and linear analysis. The presented paper uses Finite Element Method (FEM) along with multi-body dynamics simulation, which is a significant step forward in precision of the numerical method.

3. RESULTS AND DISCUSSIONS

3.1 Experiment

The proposed virtual prototype is built up by using various numerical methods to simulate pump unit movement but also to provide the necessary boundary conditions, like suspension characteristics and cylinder pressure. Due to the complexity of the model and the assumptions behind every individual numerical simulation, these results are prone to errors if not monitored thoroughly. That is why an experimental validation and if necessary, calibration of the virtual prototype is inevitably required to increase confidence in the numerical methodology. For this reason, a test bench was set up with several sensors to measure movement of the pump unit, speed and position of the crankshaft and pressure within the cylinder.

There is a scarcity of similar experiments in the literature, those found provide only limited amount of details. One article by Dropa de Bortoli, *et al* (2010) shows a function of spring motion during start/stop phases with regards to spring fatigue. Authors conclude about the importance of experimental data when evaluating the motion of the pump unit. Marriot (1998) is another author to suggest that experimental validation of his analytical model for predicting the motion of a compressor crankcase during starting plays a significant role.

The following parameters are measured during starting of the compressor:

- Angular velocity and position of the crankshaft,
- Pressure within the cylinder and
- Acceleration of a chosen point on the crankcase in every three translational directions.

All signals require post-processing in order to obtain a usable result. Acceleration data from the accelerometer sensors is integrated two times over function of time to calculate displacement.

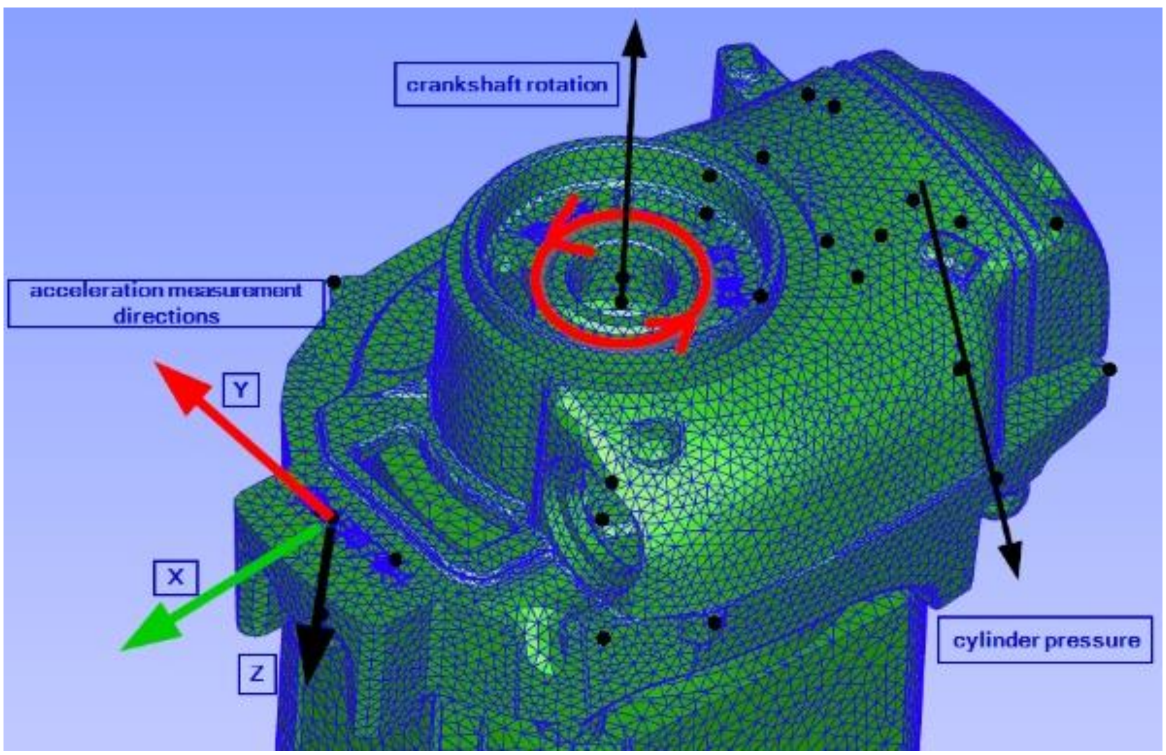

Figure 5: Experimental setup - measured characteristics

3.2 Pump unit movement calibration

For a complete simulation validation, all the boundary conditions and the final result – the pump unit movement – should correlate well with the measurement data. However, this is a very ambitious target to set and achieve due to the highly complex virtual prototype model and uncertainties of the experiment. The most important aspects for pump unit movement are following boundary conditions:

- Stiffness and damping of the suspension springs and discharge tube,
- cylinder pressure and

- motor torque characteristics.

The pump unit movement is calibrated to the experimental displacement measurement by using the measured crankshaft angular velocity and cylinder pressure as input in the MBD model. The data can be seen in Figure 6, blue color stands for the measurement curves and red for the implemented simulation input, in time domain.

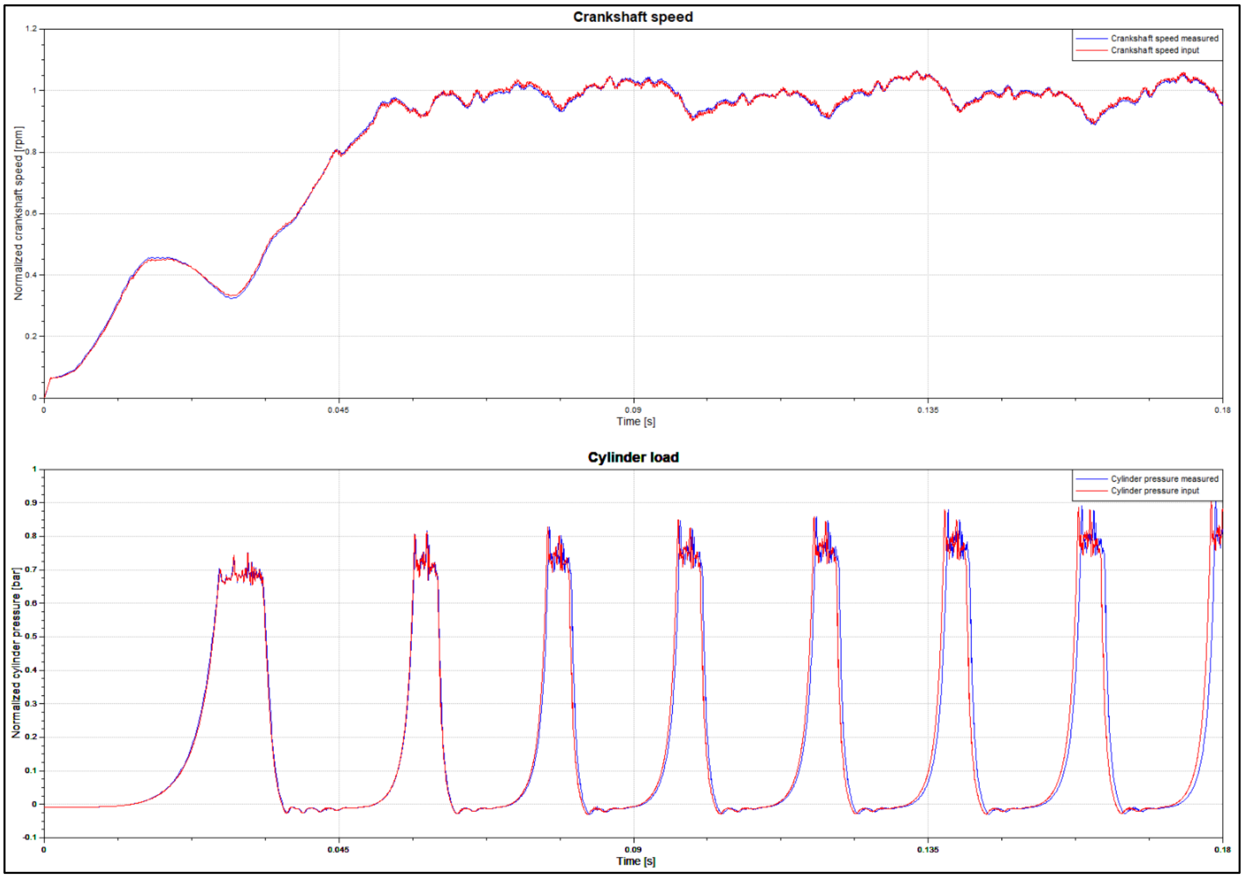

Figure 6: Crankshaft speed and cylinder load used for numerical analysis

This approach eliminates some of the possible differences in the input data, leaving motor torque characteristics and suspension stiffness as unknowns. The motor characteristics can be assumed to be accurate since it is also a measurement data, although isn't directly measured within this work. The suspension characteristics are estimated by a complex Finite Element Analysis, using multiple contact models. This approach might be prone to differences compared to the assembled suspension; therefore, adjustment of the stiffness might be necessary. Taking the measured displacement as standard for the calibration, the simulation results can be seen after adjustment of the numerical model in Figure 7. Blue color denotes the measurement while red color shows the simulation result. Direction X and Y are shown in the same coordinate system as depicted in Figure 5, direction Z is the axial one, mostly dominated by gravity and can be neglected for current analysis. Since the crankshaft speed in the simulation and experiment is identical, the timing of the main influencing effects like rising cylinder pressure and increasing motor torque are matching very closely.

Although the results show good correlation, there are points identified where improvement is needed. First, for a fully functioning virtual prototype that can replace expensive and time-consuming measurements, a complete validation is inevitable, meaning that all its input parameters must be provided without the need for complex measurements. Currently, the calculated and the measured crankshaft angular velocity aren't matching closely enough, therefore the frequency of pump unit movement also differs. Second, the standard for a very important parameter, the maximum amplitude of the pump unit movement is currently the displacement calculated from the accelerometer measurement which has its challenges and needs validation.

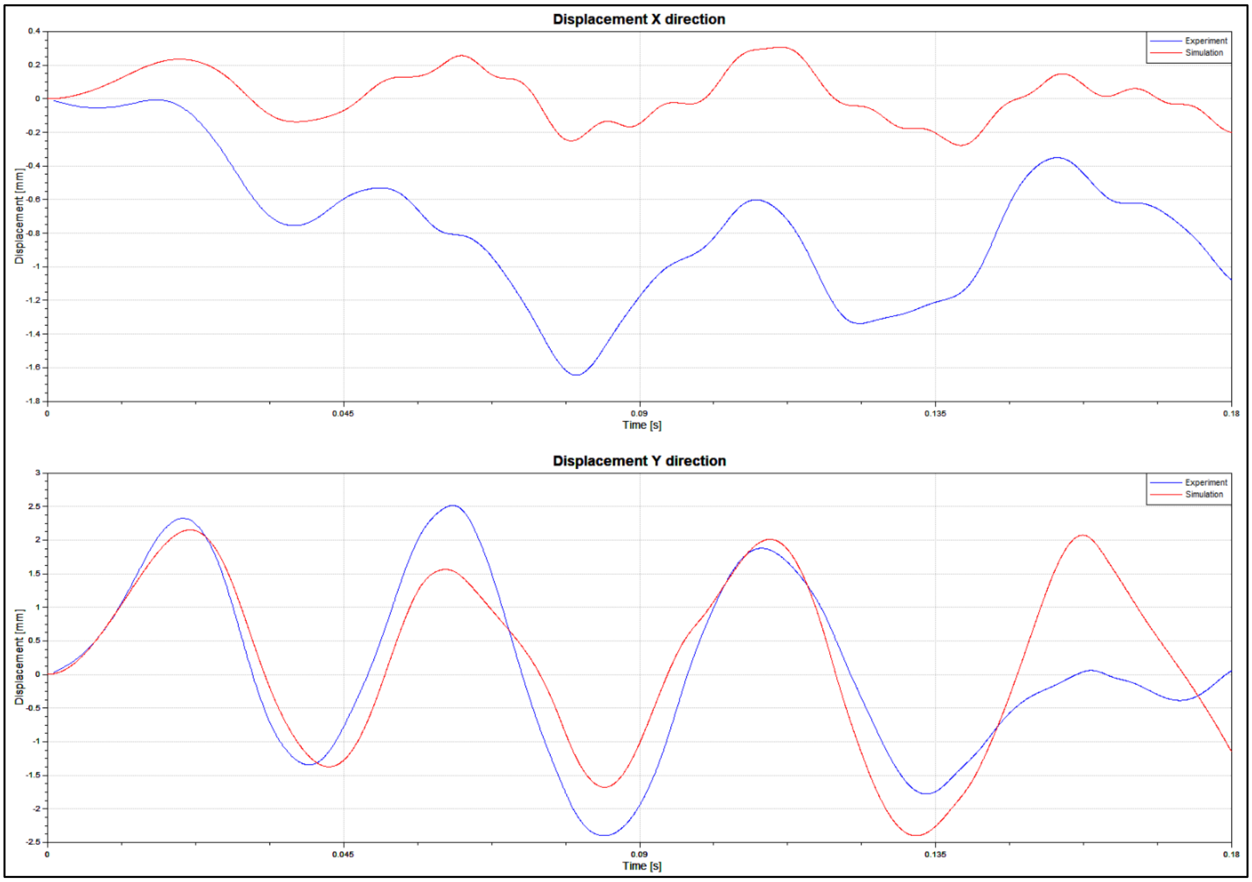

Figure 7: Displacement result comparison for simulation and experiment

3.3 Minimum gap identification

The outcome of the pump unit movement simulation can be seen basically as a 3D movement envelope in space which at some point in time is occupied by the pump unit during its operation. Designing a gap between pump unit and shell is crucial for the prevention of mechanical contact, which can have several negative effects, including a mechanical impact noise. To oversize the gap on the other hand is also disadvantageous as this increases the weight and size of the compressor. The 3D movement envelope can directly support the design of the shell as a boundary condition, showing where the optimum position is for the boundaries of the inner shell walls around the pump unit – considering a certain safety margin.

The main practical benefit for compressor development is that such virtual prototype can already be prepared at a very early stage as soon as all the necessary geometries, materials and boundary conditions are available. It can be ready long before the first hardware prototypes are manufactured. A virtual prototype offers also higher degree of flexibility, more options for quick adjustments and the possibility to do full scale DoEs (design of experiments). All of which would require extensive financial and time resources to replicate on physical prototypes. Prerequisite to gather these benefits from such a fully numerical approach is a high degree of trust in simulation results which can be achieved only by thorough and rigorous experimental validation.

Figure 8 shows final results of minimum gap to shell identification. A Finite Element tool is used to map the gap to shell profile, based on the 3D movement envelope. The pump unit is basically moved along all of the points on this envelope and gap to shell is calculated in each of the steps. The critical area for potential impact with shell is defined as the region where pump unit and shell have the smallest gap and highest movement amplitude, which is in this case the region on the top of cylinder head group. Second most critical area is on the block stopper, both seen in Figure 8. In practical application of shell design using this method also correct safety margin must be considered, along with any accuracy errors coming from the virtual prototype itself.

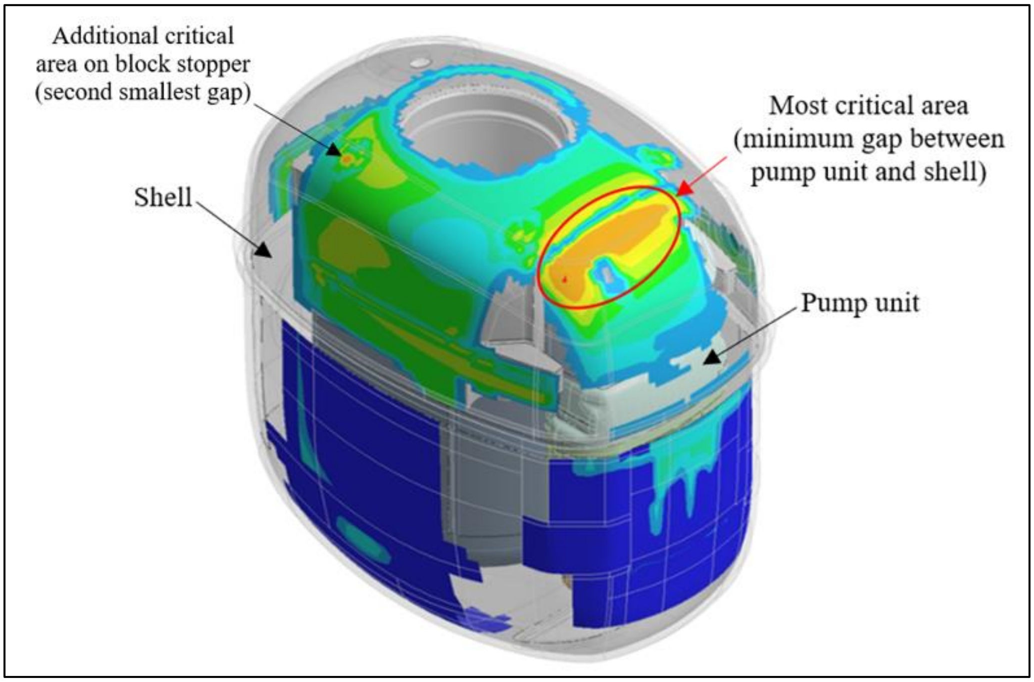

Figure 8: Minimum gap identification visualized

4. CONCLUSIONS

This paper proposes a fully numerical approach to predict pump unit movement by creating a virtual prototype. Boundary conditions, such as cylinder pressure, motor torque and suspension properties are defined and modern simulation tools, such as MBD, are used to project excitation forces onto pump unit and calculate its lateral and rotational displacements. Following conclusions are made:

- Crankshaft speed at compressor start is increasing relatively linearly, with one very short period of slow down and ramp up again. This happens due to piston reaching top of cylinder where a sudden increase of pressure slows down the movement. After this, on the piston´s way back, crankshaft speed ramps up again. These two effects can be seen oscillating also at steady operation (after start).
- Cylinder pressure ramps up and reaches constant behavior after approximately 5-6 rotations. First rotation is, as expected, the one with lowest values and slowest pressure development.
- Forces on the pump unit during starting lead to an increased dynamic effect and cause significantly larger pump unit movement within the shell compared to those during normal operation.
- Pump unit movement in the experiment and simulation is oscillating when looking at the test node, with highest amplitude in cross direction (Y).
- Proposed numerical approach is partially validated with experiment. After suspension stiffness adjustment on the virtual prototype, frequencies between simulation and experiment match very well. Amplitudes still show slight differences and could not be perfectly matched.
- Proposed virtual prototype is in the end able to predict pump unit movement with partial experimental validation. It is also shown that it can be used to define a 3D movement envelope and detect minimum gaps to compressor shell. Outcome which is very beneficial to shell development in terms of costs, time and equipment resources.

Limitations and difficulties of this work are most noticeable in the experimental validation of the numerical calculations. On the numerical side, simulations have a relatively easy access to parametrization once they are fully modeled. Many compressor displacements, conditions and even different motor torques can be simulated, while on the other hand experimental validation requires high amount of financial, time and equipment resources for any modification. This paper presents experimental validation on only one compressor model, with one motor and one condition. Future work could build upon this by performing experiments of pump unit movement on multiple compressors, multiple compressor models or under different conditions. Another limitation that could be improved

upon in future work is the number and locations of accelerometers during experiment. In this paper, only one accelerometer on one position is validated with its corresponding node in the MBD model. Increasing the number of accelerometers to fully capture pump unit global motion and validating the MBD model would provide more confidence in the numerical model. Additionally, the experimental process itself could be validated with a high-speed camera that captures the pump unit movement and an algorithm that calculates displacement of multiple pre-defined points. Furthermore, the second increased excitation event during compressor operation, the stopping, could be simulated and experimentally validated. Similar increased amplitude and unpredictability of pump unit movement is expected.
To exhaust the full potential of a virtual prototype, its boundary conditions need to come also from numerical simulations, therefore motor torque, spring stiffness calculation and cylinder pressure calculation must be calibrated to experimental results.

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