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A Co-Simulation Approach for Hydraulic Percussion Units

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Cover:

Front: An overview of the co-simulation work flow.

Back: Simulation result showing the stress wave in the tool at impact.

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Preface

The work presented in this Licentiate of Engineering thesis has been carried out at the Division of Solid Mechanics, Linköping University and at Epiroc, Kalmar. The research project concerns the field of simulation methodology for hydraulic percussion units, and has been an industrial Ph.D. project in collaboration between Epiroc, Kalmar, the Division of Solid Mechanics, Linköping University and Dynamore Nordic, Linköping. The project has been fully funded by Epiroc, Kalmar.

I would like to thank my advisor Associate Prof. Daniel Leidermark¹ and my assistant advisor Prof. Kjell Simonsson¹ for all their help during this project. Furthermore, I would like to thank my assistant advisors at Dynamore Nordic; Dr. Daniel Hilding² and Dr. Mikael Schill² for their valuable knowledge they shared with me. I would also like to give a very special thanks to Lic. Eng. Peter Nordin³, co-author on my first paper, who introduced me into the world of TLM, L^AT_EX and scientific writing. Further, without Peter's skills in technical programming and knowledge in the Hopsan simulation tool the co-simulation interface had become much harder to realise. I am also most grateful for the contribution of Dr. Thomas Borrvall² who has made important contributions to this project. I would also like to thank my managers Erik Sigfridsson⁴ and Conny Sjöbäck⁴, who gave me the opportunity to Ph.D. studies and to carry out this research project.

Finally, I would like to thank my family, who with great patience have withstood the many hours of our free time I have spent on this work, and the time I have spent in Linköping.

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Abstract

This Licentiate of Engineering thesis concerns modelling and simulation of hydraulic percussion units. These units are often found in equipment for breaking or drilling in rock and concrete, and are also often driven by oil hydraulics, in which complex fluid-structure couplings are essential for their operation.

Current methodologies used today when developing hydraulic percussion units are based on decoupled analyses, which are not correctly capturing the important coupled mechanisms. Hence, an efficient method for coupled simulations is of high importance, since these mechanisms are critical for the function of these units. Therefore, a co-simulation approach between a 1D system simulation model representing the fluid system and a structural 3D FE-model is proposed.

This approach is presented in detail, implemented for two well-known simulation tools and evaluated for a simple but relevant model. The Hopsan simulation tool was used for the fluid system and the FE-simulation software LS-DYNA was used for the structural mechanics simulation. The co-simulation interface was implemented using the Functional Mock-up Interface-standard.

The approach was further developed to also incorporate multiple components for coupled simulations. This was considered necessary when models for the real application are to be developed. The use of two components for co-simulation was successfully evaluated for two models, one using the simple rigid body representation, and a second where linear elastic representations of the structural material were implemented.

An experimental validation of the co-simulation approach applied to an existing hydraulic hammer was performed. Experiments on the hydraulic hammer were performed using an in-house test rig, and responses were registered at four different running conditions. The co-simulation model was developed using the same approach as before. The corresponding running conditions were simulated and the responses were successfully validated against the experiments. A parameter study was also performed involving two design parameters with the objective to evaluate the effects of a parameter change.

This thesis consists of two parts, where Part I gives an introduction to the application, the simulation method and the implementation, while Part II consists of three papers from this project.

List of papers

In this thesis, the following papers have been included:

- I. H. Andersson, P. Nordin, T. Borrvall, K. Simonsson, D. Hilding, M. Schill, P. Krus, D. Leidermark (2017). A co-simulation method for system-level simulation of fluid–structure couplings in hydraulic percussion units, *Engineering with Computers*, Volume 33, Issue 2, Pages 317-333
- II. H. Andersson, K. Simonsson, D. Hilding, M. Schill, D. Leidermark (2017). System level co-simulation of a control valve and hydraulic cylinder circuit in a hydraulic percussion unit, *Proceedings of 15:th Scandinavian International Conference on Fluid Power, June 7-9, 2017, Linköping, Sweden*, Linköping University Electronic Press, Linköping University, Issue 144, Article Number 22
- III. H. Andersson, K. Simonsson, D. Hilding, M. Schill, E. Sigfridsson, D. Leidermark (2018). Validation of a co-simulation approach for hydraulic percussion units applied to a hydraulic hammer, *Submitted*.

Own contribution

I have had the main responsibility regarding the writing of the appended papers. Paper I is a collaboration with Lic. Eng. Peter Nordin, former Ph.D. student at the Division of Fluid and Mechatronic Systems, Linköping University, who was responsible for writing the section on the Implementation. Paper II and III were entirely made by me with support from my co-authors. The development of the code for the co-simulation interface, the communication library and the FMU-generator, have been done by Peter Nordin. The LS-DYNA implementation of the co-simulation interface was done by Dr. Thomas Borrvall at Dynamore Nordic AB, Linköping. The experiments in Paper III was performed in collaboration with Kenneth Johansson Birath, Epiroc, Kalmar, while the data analysis was made by me.

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Part I

Application and Method

Introduction

The manufacturers of hydraulic percussion units need to find improved simulation methods due to continuously increasing requirements from legislation regarding, *e.g.* robustness, sustainability and environmental impact, and customer demands requiring that new such products are evaluated to a further extent during the development phase than today. The industry also strives for minimising the testing of physical prototypes, and must therefore introduce new simulation-based methods for the product development. The nature of these products holds a lot of complex mechanisms, where some of them are fluid-structure interactions that need to be handled by the simulation method. Other typical features are the short duration and high amplitude pressures and strains, occurring in both the fluid and the structure, which require that the wave propagations, and interactions, throughout the model are correctly reproduced by the simulation method.

Today Fluid-Structure Interaction (FSI) capabilities can be found in many software suits, *e.g.* LS-DYNA [1] and ANSYS [2], and are commonly used to study problems on the detailed level for a small part of the system. Transient FSI-simulations for complete systems are very rare and require extensive computational resources, not only for solving but also for the post-processing, and hence are not suitable for the general machine building industry where the computational resources are limited. Due to this, alternative methods must be utilised when performing coupled simulations for fluid power machinery developed by the general machine building industry. The fluid system of hydraulic percussion units can be simulated using 1D system simulation tools, where the structural parts are incorporated as 1D rigid bodies, to include the fluid-structure couplings [3, 4]. This method is based on sub-components that are described by different types of equations, *e.g.* differential and algebraic equations, for describing the behaviour of the component, which are connected in a network representing the full system. This method is computationally inexpensive and efficient for simulating the complex behaviour of a hydraulic percussion unit. However, if these products shall be evaluated to a further extent the elastic properties of the structural components must be incorporated to fully represent the important fluid-structure couplings, which effects not only can be observed at the component level but also on the overall system level. This can be done by using flexible bodies that are incorporated in the system model, and are defined by its modal properties, eigenfrequencies and eigenvectors, or simply by defining the wave equation for those bodies, which can be a good approach for conceptual studies. A more general method would be to incorporate a structural Finite Element (FE) model in the system simulation model, and by co-simulation transfer the fluid loads to the structural FE-model and send its response back to the fluid system, and thereby achieve the necessary fluid-structure couplings.

1.1 Aim of this work

The aim of the work presented in this Licentiate of Engineering thesis has been to develop a computationally inexpensive co-simulation approach for hydraulic percussion units, where the fluid-structure couplings affecting the overall behaviour are included. The approach shall facilitate evaluations not only on the overall system level, *e.g.* performance and efficiency, but also on the component level, *e.g.* evaluation of stresses and assessment of fatigue. Another feature important for an improved simulation method is the prediction of radiated noise.

1.2 Outline

Part I of the thesis gives an introduction to hydraulic percussion units in the application of hydraulic hammers and rock drills, and a presentation of essential and critical mechanisms in these units. A short introduction to co-simulation and to different approaches are given, followed by a detailed presentation of the implementation of the co-simulation interface developed in this project. Furthermore, an overview of the performed experimental validation is given. Part II contains the three papers produced in this project.

Hydraulic Percussion Units

Hydraulic percussion units are often found in mining and construction equipment for drilling or breaking of rock and concrete, *i.e.* rock drills and hydraulic hammers, see Fig. 1. By the percussive mechanism high impact forces are generated to break or crush the material during operation. Rock drills are mostly used for underground blast hole drilling in mine drifting or when driving tunnels. They are often mounted on special drill rigs with controlled installation, both mechanically and hydraulically, since high input of hydraulic power is needed during operation, which also requires high feeding forces in order to achieve an efficient drilling operation. The characteristic properties of the percussion unit in a rock drill are: high impact velocity, high impact frequency, moderate impact energy level, high working pressure and high oil flow. Hydraulic hammers are

Figure 1: Typical products using hydraulic percussion units, showing a) the Epiroc SB202 hydraulic hammer and b) the Epiroc COP MD20 hydraulic rock drill, courtesy of Epiroc.

often used in demolition and reconstruction works of concrete structures, but larger models can also be used in mining or in quarries for secondary rock breaking, and the largest models may even be used for primary rock excavation. They are often mounted on commercially available excavators and are connected to the present hydraulic system, which sometimes may limit the hydraulic input power that will end-up in a non-optimum operation. The characteristic properties of the percussion unit in a hydraulic hammer are: moderate impact velocity, low impact frequency, high impact energy level, moderate working pressure and moderate oil flow. A comparison of the most important parameters for these machines are presented in Table 1.

2.1 Main Mechanisms

Figure 2: The main components in the percussion unit of a hydraulic hammer.

Table 1: Comparison of the Rock drill and the Hydraulic hammer.

	Operating pressure	Oil flow	Impact Energy	Impact Frequency
Rock Drill	High	High	Medium	High
Hydraulic Hammer	Medium	Medium	High	Low

The main parts in the percussion unit of a hydraulic hammer can be seen in Fig. 2. The Housing is the outer shell of the unit, where all the internal parts are mounted. It also includes a number of channels for distributing oil to different cavities in the hammer. The Piston is the part that converts the hydraulic energy to mechanical energy. The Piston is guided by the Liner and due to special features of these parts a few different valve mechanisms are established. The Control valve controls the oil flow to the Piston. A typical piston displacement curve is shown in Fig. 3, where the position of impact is equal to zero and the upper turning point is equal to one.

Figure 3: A typical piston displacement of a hydraulic hammer. The working cycle consists of the return stroke, where the piston is lifted to the upper turning point, and the working stroke, where the piston is accelerated towards the tool and the point of impact.

At the start of the working cycle the Piston is resting on the top of the Tool, and at this point Cavity A is connected to the low pressure side of the hammer through the Control valve. Since Cavity B is always connected to the high pressure side, the Piston will start to move upwards, *i.e.* return stroke. At a certain distance, the signal groove, a feature on the Liner, will open up and thereby pressurise the signal line to the Control valve, which then shifts its position and connects Cavity A to the high pressure side. When Cavity A is pressurised the Piston starts to accelerate downwards towards the tool, *i.e.* working stroke. At another predefined distance the Piston will connect the signal- and the return groove, and the control pressure on the Control valve will be released, which causes the Control valve to shift position and connect Cavity A to the low pressure side. This means that directly after the impact the Piston will start to move upwards again, and the cycle is completed. During the working stroke the oil consumption is higher than can be delivered by the carrier, and therefore an intermediate energy storage must be used, *i.e.* the Accumulator. This is in short the working principle of the percussive mechanism of the hydraulic hammer used in this project. There are also other principles for hydraulic impact machines, see for instance the study by Gorodilov [5].

2.2 Critical Mechanisms

The hydraulic percussion unit is operating in a reciprocating manner, which is achieved by the opening and closing of valves. The mechanism of suddenly closed valves will force the flow of oil to change direction and by that generate pressure waves throughout the fluid system. The impact mechanism will generate stress waves in the structural components of the unit. Both the fluid pressure waves and the structural stress waves are critical for the operation and they will significantly influence not only the overall performance and durability but also the operation of each component. These mechanisms can be the source to several problems, *e.g.* wear, cavitation and contact surface related issues, and also, due to timing effects, cause low efficiency or even an incorrect function. Thus, it is of big importance to consider these mechanisms when designing a hydraulic percussion unit.

Simulation Methods

The need for a proper simulation tool when designing hydraulic percussion units is evident since several coupled mechanisms must be represented, which are also acting in different domains, *i.e.* fluid and structure, and hence, it will be very challenging to predict the total response. Several papers that concern the area of modelling and simulation of percussion units can be found in the literature. Gorodilov started his work in 1997 trying to establish mathematical models for hydraulic impact machines, and has published several papers on this subject until 2018, see for example [5, 6]. Giuffrida et al. [3] and Ficarella et al. [7–9] defined a 1D system simulation model of a hydraulic hammer using the simulation tool Amesim [10] for simulating the responses for the coupled fluid-structure system. In the work by Oh et al. [4] a hydraulic rock drill was simulated using the same methodology as Giuffrida et al. and Ficarella et al. Their approach to use 1D rigid bodies for the structural components and 1D fluid components, *e.g.* cavities, orifices etc., and the formulation of these components cannot handle internal wave propagation, since they only has one degree of freedom, and hence this mechanism is not represented in their simulation results.

The requirements on an improved simulation approach for hydraulic percussion units must be to include all critical mechanisms, facilitate the use of elastic 3D structural components, and also to require it to be computationally inexpensive. It has been found that system simulation models, which are described by *e.g.* 1D partial differential equations or ordinary differential equations, are often used to simulate the functional behaviour of a product and this approach will not only represent the critical fluid mechanisms but it will also be computational very efficient. 1D elastic bodies can also be incorporated in the system simulation model by using 1D partial differential equations to represent the axial stress waves that are critical for the operation of these units, and since it is in the same model the fluid-structure coupling are already established. However, the use of 3D structural components are necessary when detailed analysis and assessment of stress and fatigue or acoustical radiation must be performed. In order to incorporate 3D structural components in the simulation model co-simulation with an FE-model can be set-up, which is described in the coming sections.

3.1 Transmission Line Method

The transmission line element method (TLM) was proposed by Auslander [11], where it was referred to as bi-lateral delay line modelling, for modelling and simulation of distributed physical systems. The fundamental concept in this method is that a physical

motivated time delay T are introduced that effectively decouples the whole system, which facilitates parallel simulation of each sub-component. The physical time delay is here related to the wave propagation speed in each component, which ensures a weak coupling of the interface points, and thus, makes it possible to use explicit time integration methods for solving the system.

Figure 4: A TLM element illustrated as a pipe with fluid pressure p and input flow Q on each side as boundary values.

By solving the governing relations together with boundary conditions, the following two relations are found

$$p_2^{t+T} = c_1^t + Z_c Q_2^{t+T} \quad (1)$$

$$p_1^{t+T} = c_2^t + Z_c Q_1^{t+T}, \quad (2)$$

where the impedance Z_c and time delay T are given by the capacitance C and inductance L of the component according to

$$Z_c = \sqrt{\frac{L}{C}} \quad (3)$$

$$T = \sqrt{LC}, \quad (4)$$

and where the c entities can be recognised as the wave variables, which are calculated as follows

$$c_1^t = p_1^t + Z_c Q_1^t \quad (5)$$

$$c_2^t = p_2^t + Z_c Q_2^t \quad (6)$$

Eq. 1 and 2 show the relation between the pressure and flow on each side of the TLM element. They also show the pressure dependence on one side of the pressure at the other end from the previous time step, which represents the wave propagation through the element.

A comprehensive background and presentation of the TLM-method is for example given by Braun [12], who also has derived the Eq. 1–4 using a slightly different notation. Krus et al. presented a method of handling wave propagation in pipeline sub-models using the method of characteristics for the transmission line elements [13]. Further, this was applied to simulate the responses of a hydromechanical system using the TLM-based

simulation tool Hopsan, which is presented in more detail in Section 5. The TLM-technique used by Hopsan has during the years been found to be a very good tool for predicting the fluid responses of the hydraulic percussion units, which is mainly due to the handling of the wave propagation in the fluid system.

3.2 Explicit FE-simulation

When simulating stress wave propagation in solid structures or when analysing contact problems between structural parts the explicit FE-software LS-DYNA works very well, to which a brief introduction is made to point out the specific properties that are used in the simulation of hydraulic percussion units.

LS-DYNA is solving the general equation of motion stated in Eq. 7

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{C}\dot{\mathbf{u}} + \mathbf{f}_{Int} = \mathbf{f}_{Ext} \quad (7)$$

where $\mathbf{M}$ is the mass matrix, $\mathbf{C}$ is the damping matrix and $\mathbf{f}_{Int}$ is the internal force vector. The external force vector, $\mathbf{f}_{Ext}$, represents all external loads acting on the FE-model. The displacement $\mathbf{u}$ is solved explicitly by the central difference time integration scheme described by Eq. 8 and 9. For further description regarding the explicit integration scheme see for instance [14].

$$\dot{\mathbf{u}}^{n+\frac{1}{2}} = \dot{\mathbf{u}}^{n-\frac{1}{2}} + \ddot{\mathbf{u}}^n \Delta t^n \quad (8)$$

$$\mathbf{u}^{n+1} = \mathbf{u}^n + \dot{\mathbf{u}}^{n+\frac{1}{2}} \Delta t^{n+\frac{1}{2}} \quad (9)$$

3.3 Co-simulation

When complex mechanical systems, where the fluid-structure couplings are essential for the overall responses, shall be analysed the today well-known 3D FSI method has been used, in the context of co-simulation of 3D FE-models and 3D Computational Fluid Dynamic (CFD) models. These methods have been developed during the last decades and are today available in many software suits, *e.g.* LS-DYNA [1] and ANSYS [2]. A schematic figure for a general co-simulation procedure is shown in Fig.5.

Examples of industrial use of these types of simulations can be found in [15, 16], and further, these applications are often found in the specialised industry such as aerospace, nuclear or defense, where the massive computational resources required for large scale FSI-analysis can be found. The general machine building industry requires computationally more efficient methods for coupled simulations, especially when the complete system shall be analysed, since the computer resources are limited. One approach to achieve a computationally inexpensive method is to conduct a co-simulation between a 1D simulation method for the fluid system and a 3D FE-model for the simulation of the structural responses. Studies where a 1D system simulation model is co-simulated with a 3D model, structural or fluid, have been performed with the purpose to incorporate an

Figure 5: A schematic figure of a co-simulation procedure of two simulation models. The model parameters and functions f_n are sent to the simulator for each model. During the simulation the specified information, *i.e.* $\mathbf{x}$, $\mathbf{y}$, is exchanged between the simulators, thus affecting the response y_n in each model.

overall system behaviour in the detailed 3D-model [17–19]. The real advantage from this approach is that full 3D-results are available, which facilitates advanced detailed evaluations of stress, fatigue, acoustic responses, etc. The use of co-simulation between different simulation tools has been promoted by the Functional Mock-up Interface (FMI) standard [20], which today are implemented in many different simulation tools. This standard supports exchanging dynamic models using a standardised interface between different modelling and simulation tools, and a component that originates from this standard is called a Functional Mock-up Unit (FMU).

The Co-Simulation Approach

The used co-simulation approach is presented in Paper I, which was further developed in Paper II. This approach is based on a 1D system simulation model, representing the fluid in the hydraulic circuit, that is co-simulated with a 3D FE-model for the structural mechanics simulation. The co-simulation interface is based on the FMI-standard, and the FMU is incorporated in the system simulation model for handling the communication to the structural simulation. The method is illustrated in Fig. 6, where the top row represents the system simulation, the middle row is the co-simulation interface and the bottom row is the FE-simulation.

Figure 6: Overall simulation sequence

During the simulation the fluid pressure p , simulated in the fluid system simulation, is communicated over the co-simulation interface to the structural mechanics simulation, where displacements u , velocities $\dot{u}$ and forces f are calculated and sent back to the fluid system simulation. The pressure and the flow Q in the fluid system is then updated due to the structural responses, and the coupled simulation is herewith completed. When the simulation is completed the full response in the fluid is available for component analysis of pressure and flow, but overall analysis of the system performance and efficiency can also be conducted. A full set of 3D FE-results are also available which facilitates stress analysis, fatigue assessment or acoustic analysis for the structure. This method was implemented and verified for simple but relevant test models, including main mechanisms in hydraulic percussion units, for co-simulation of only one component in Paper I. In Paper II multiple components for co-simulation were successfully evaluated for a simple model of verification.

Implementation

In Section 3 the simulation methods were presented, and it was proposed to make use of the TLM-technique for the 1D system simulation and explicit FEM for the 3D FE-simulation. Further, a co-simulation interface should be based on the FMI standard to facilitate a more general use of this tool. In this section, the implementation of the proposed co-simulation approach used in this project is presented.

5.1 The Hopsan Simulation Tool

The first version of Hopsan was launched in 1977 and it was written in Fortran using a graphical interface written in Visual Basic. It was developed by the Division of Fluid and Mechatronic Systems at Linköping University [21]. The first version of Hopsan has historically been used as the main system simulation tool when developing hydraulic percussion units at Epiroc. The fast simulations and the handling of the wave propagation throughout the fluid system, which were facilitated by the use of the TLM-technique, have made Hopsan a very appreciated simulation tool. However, as this version became out-of-date, a research project was initiated to develop a new generation of Hopsan. This version was based on the object oriented programming language C++ and was launched 2010 [22]. A more detailed description of the Hopsan simulation tool can, for instance, be found in [23].

An important property implemented in Hopsan is the concept of power ports, which are used as the interface points between sub-components. For each domain, *e.g.* fluid, mechanical and electric, there are different variables for calculating the power transfer in the system. In hydraulic systems, the power is calculated by the pressure and the flow, while the velocity and the force can be used for a mechanical system. These variables are defined by the power port and are describing the power transfer in that domain. The concept of power ports also facilitates the use of physically motivated connection points, similar to what is used in the real world, *e.g.* the hydraulic hose or the electric cable.

Functionalities for co-simulation between Hopsan and other software packages have already been implemented in the early versions, see [24]. In the new generation of Hopsan the FMI standard has been implemented, which supports both model exchange and co-simulation interoperability [20].

5.2 LS-DYNA FE-software

LS-DYNA is a general-purpose finite element program capable of simulating highly non-linear and transient dynamic problems [1]. The program includes both implicit and explicit solvers, where the latter one has been found to be very suitable when analysing hydraulic percussion units, since the important mechanisms are highly non-linear and of short duration. Further, the stress wave propagation in the solid structures and the contact mechanisms are very important when simulating the responses for these units, and these are areas where the explicit solver have been found to work very well. LS-DYNA also supports customised external loads through the user-defined function (UDF) code, which was found to be a suitable interface point for communication at co-simulation.

5.3 Co-simulation Interface

While the Hopsan simulation tool already are supporting the FMI standard, these functions are not implemented in LS-DYNA. For the realisation of the co-simulation interface an FMU that communicates with the LS-DYNA UDF code was developed, and this function for applying custom external loads was used as the entry point into the software. A TCP/IP socket communication library was developed using the C programming language, since C-functions can be called directly from the Fortran language code, which is used in LS-DYNA. This library is used by both the FMU and the LS-DYNA UDF, and the TCP/IP communication also facilitates the use of dedicated computer clusters for the FE-simulation, while another computer on the network runs the system simulation.

This configuration makes it necessary to synchronise the simulation models and to define which variables to be used in the communication between these models. Hence, a custom-made configuration file based on the native *Keyword* format in LS-DYNA was developed. An automatic FMU-generator was also developed, which together with the configuration file compiles the FMU to be used in the system simulation model. The configuration file is also used by LS-DYNA during the simulation, which effectively establish the connection points between the simulation models and defines which variables to be used for each port. Due to this set-up all low-level work of defining the FMU and the LS-DYNA UDF are taken care of for the user, and it is only necessary to define the interface points in the models. The connectivity and the work flow that is used by the co-simulation interface are shown in Fig. 7.

This implementation requires that the same fixed time steps are used both by Hopsan and LS-DYNA, which is most commonly determined by the smallest element in the FE-model in LS-DYNA and the Courant condition [14], and this makes it also possible that a shorter time step than necessary is used in Hopsan. However, this was considered appropriate since the time for simulating the Hopsan model can be considered negligible compared to the corresponding time for LS-DYNA.

Another important issue for a co-simulation is synchronisation of the communication, and when to transfer data between the simulation tools. In this implementation, Hopsan was set for administering the simulation and trigger each time step in LS-DYNA. The synchronisation throughout the simulation is illustrated in Fig. 8. The simulation starts

Figure 7: The work flow and the connectivity of the implemented co-simulation interface.

when the programs have connected and the communication protocol is verified. The simulation in Hopsan or in LS-DYNA can be started first, and will be waiting for the other to connect. Then the Hopsan simulation will be initialised and started by first simulating the Signal models, then the C-type models and finally the Q-type models. After this, the Send point is reached and data is sent to the LS-DYNA simulation. In parallel the LS-DYNA simulation has established the external forces through the UDF module and has advanced to the Receive point, where it waits for data from the Hopsan simulation. When data is received from Hopsan the simulation step in LS-DYNA begins, and if the simulation is not finished, the FE-model responses are sent back using the UDF module to the Hopsan simulation at the Send point on the LS-DYNA side. The Hopsan simulation is updated with the responses from LS-DYNA, and after the finishing Check, the simulation loop will be repeated for the next time step. The simulation steps for the programs are synchronised by using blocking socket receive calls, which means that each simulation is waiting until data is correctly received, and if not a failure time-out is set that terminates the simulation.

5.3.1 Functional Mockup Unit

A component, where the model content is wrapped in an archive with an extension of ".fmu", and with an interface according to the FMI standard, is called Functional Mockup Unit (FMU) [20]. The FMU can be used in any tool that is supporting the FMI standard. Normally the FMU is exported from one tool and then imported into the other, but since this functionality is not implemented in LS-DYNA a standalone FMU-generator was needed. The FMU for co-simulation consists of two mandatory parts; first an XML-file and secondly, a pre-compiled C-code interface. The XML-file

Figure 8: The co-simulation sequence and simulation step synchronisation between Hopsan and LS-DYNA. The receive points block execution until data have been received.

contains all model information and definitions of all variables used by the interface. The pre-compiled C-functions are in this implementation used for the TCP/IP communication with LS-DYNA.

The FMU-generator simply consists of a Python program that compiles the communication library, builds the XML-file describing the FMU and its variables, and finally wrap these files in an archive with the file extension ".fmu".

5.3.2 Configuration File

In the configuration file all ports and variables to be used for co-simulation are defined. First, a list of the port IDs is specified, and then the corresponding port and variables in Hopsan and in LS-DYNA are defined. Further, a scaling procedure was implemented in the communication library to facilitate the use of symmetry FE-models, and this procedure automatically scale the values communicated over the co-simulation interface. The values that are to be used by the scaling procedure must be specified in the configuration file.

5.3.3 LS-DYNA User Load Interface

In LS-DYNA the *Keyword* USER LOADING will activate the subroutine LOADUD in the custom load file from LS-DYNA, *i.e.* the "dyn21.f"-file, where the code for communication to the FMU and the routines developed for this application were implemented. This file is compiled together with the main files to build the user specific version of the executable file for LS-DYNA, which supports co-simulation with Hopsan. The USER

LOADING *Keyword* was used for the definition of which segment set IDs that belong to each port, and by the use of the USER LOADING SET *Keyword* the linking between the segment set ID and the segment set was specified.

One of the routines that was developed and implemented in this project was the control of the pressurisation of each segment depending on its position. This is an important feature for hydraulic percussion units where pressurised components are moving in and out from hydraulic cavities and the pressurised surface changes with its position.

Validation

The main objective for this project was to develop an improved simulation approach for hydraulic percussion units, which also has been presented in Paper I & II, cf. [19, 25]. In these studies the approach was verified for virtual and simple models representing main mechanisms from percussion units. In order to conduct a more thorough validation of the approach a number of experiments were executed on a real hydraulic hammer, where the results were compared to the responses from the corresponding simulation model. The complete study of the validation is presented in Paper III, and only a brief overview is given in this section.

6.1 Experiments

A series of experiments were performed on the hydraulic hammer using an in-house test rig, see Fig. 9.

The hydraulic hammer was an Epiroc SB202, see Fig. 1a, which is a 200 kg hammer developed mainly for reconstruction work in concrete structures. This hammer suits excavators, or other carriers, in the weight class of 2.5–6 tonnes. The nominal hydraulic input is 150 Bar at a flow rate of 35–65 l/min, and the typical impact frequency is in the range of 15–30 Hz. The parts of the hammer that are referred to in this section are shown in Fig. 2 and described in Section 2. The hammer was set-up in the test rig in order to operate under stable and controlled conditions. The working material was represented by the anvil, a steel block, which properties in this application can approximately be compared to high strength concrete. The function of the feed-force is to ensure contact between the hammer and the working material. A number of sensors were used to register the characteristic behaviour of the different parts of the hammer, *i.e.* component position, acceleration, strain and oil pressure, see Fig. 10. Experiments were conducted using four different running conditions, see Table 2, where the operating pressure and the restrictor, limiting the oil flow, were varied.

6.2 Simulation Model

The simulation model representing the hydraulic hammer was developed using the previously proposed co-simulation approach, and also using the same implementation as in [19, 25]. The 1D system simulation model was developed in Hopsan representing the fluid system, see Fig. 11, and the 3D structural component assembly was developed

Figure 9: Schematics of the measurement set-up.

using the FE-method for simulation in LS-DYNA, see Fig.12. The system model was a complicated network consisting of different types of sub-components, such as *e.g.* cavities, valves and restrictors, each representing a feature in the fluid system. The major part of the sub-components were collected from the standard component library in Hopsan, but the functionality had been modified for some to achieve a more relevant behaviour for the hydraulic percussion unit.

The FMU was configured for co-simulation of the piston and the control valve, for which both hydraulic and mechanical ports were defined. Mechanical ports were also defined for the rig tool, the housing and the valve cover, in order to monitor the position of these components during the simulation. The positions of the piston, the control valve and the housing were used to control the opening and the closing of the valve components

Table 2: Experimental running conditions of the four cases. D_R is the diameter of the Restrictor.

	Operating pressure (Bar)	Oil flow (l/min)	$\varnothing D_R$ (mm)
Case 11	150	80	6.0
Case 12	150	66	5.4
Case 21	100	59	6.0
Case 22	100	50	5.4

Figure 10: Showing a) the sensors on the upper side and b) the sensors on the left side of the hammer.

in the simulation model, which in turn controls the operation of the percussion unit. The running conditions from Table 2 were realised by changing the inlet pressure and the restrictor diameter in the system model.

6.2.1 Finite Element Model

The FE-model represents the main structural parts of the hydraulic hammer, the rig tool and the anvil, see Fig. 12, while the test rig was represented by discrete mass, spring and damper elements at the upper and lower boundary. The spring stiffness and the viscous damping values were tuned to fit the measured response of the housing movement as close as possible. The mass loading from the test rig was represented by discrete mass elements that was evenly distributed on the adapter plane of the housing. Due to symmetry a half model was used in order to reduce the time for simulation. All needed contacts were defined for each part to represent the correct behaviour, and for those where oil and grease could be found the damping value of the contact was increased, from 10% to 50% of the critical viscous damping value. The control valve is totally surrounded by oil and at the end points an oil film is created at the contact surfaces, which will reduce the contact forces. This effect was represented by implementing a routine in LS-DYNA that calculates a damping force, which will reduce the velocity of the control valve just before it hits the endpoint and the contact forces will by this be reduced, see Paper III for a detailed description of this routine. The fluid loads in the hammer were simulated in Hopsan and communicated over the co-simulation interface to the FE-model for each

Figure 11: The Hopsan simulation model. The components belonging to each of the main functions are encircled: pressure inlet, pressure outlet, impact piston and control valve. The restrictor was used to control the oil flow through the percussion unit. The friction force sub-model calculates the force from the piston sealing with respect to hydraulic pressure and the low pass filtered signal of the piston velocity. The gauges are sensors for different quantities; x =Displacement u , v =Velocity $\dot{u}$ and p =Pressure p .

time step. The pressure value was applied to each segment that belongs to each respective cavity for the piston and the control valve. A control routine in LS-DYNA keeps track of which segment is inside or outside the cavity, if a segment moves outside the cavity the pressure is removed, and when the segment moves inside again the pressure is restored. The friction force from the hydraulic sealings on the piston was simulated by Hopsan and transferred to LS-DYNA over the mechanical port for the Piston. The force is acting in the opposite direction to the piston movement and is thereby reducing its velocity. The feed-force on the hammer was simulated by a prescribed displacement of the spring elements attached to the adapter plane, and the compressive force was generated when the spring elements were compressed a certain distance. The pre-loading of the FE-model was conducted as an initialisation from a prescribed geometry using the explicit dynamic relaxation routine in LS-DYNA.

6.2.2 Time Step and Mass Scaling

A fixed time step of 2.7×10^{-7} s was used in both Hopsan and LS-DYNA, which was determined from the smallest element in the Piston FE-model and the relevant Courant condition [14]. However, this choice of time step resulted in a minor mass scaling in

Figure 12: The a) showing the complete FE-model and b) details of the hammer.

some of the parts in the FE-model, which is also discussed in Paper III. Components from fluid power machinery often tend to have narrow sections that generates small elements when modelled by finite elements, and thus generating small time steps. If these small time steps are used without mass scaling, the clock time to finish the simulation will be increased. It is however important to monitor the effects of the mass scaling since it can significantly affect the dynamic properties of the body, and for components with large movement the mass scaling should be reduced to a minimum. In this study the maximum mass scaling was 1.8% for the control valve, which has a large movement, and 5.8% for the valve cover, which has a small movement, which was considered acceptable for this application.

6.2.3 Restrictor Tuning

From in-house experience at Epiroc it is known that the simulated flow through the percussion unit is underestimated, which depends, among other things, on the modelling of the restrictor. The running condition Case 11 was used when estimating the diameter of the restrictor, which was tuned to meet the impact frequency from the experiment. An equally large restrictor was also used for the simulation of Case 21, but for Case 12 and 22 the diameter was derived using the same area relation as for the restrictors used in the experiments.

Figure 13: Simulation and experimental results from Case 11. Marked regions or events of the signals, *e.g.* region N in Fig. 13a, where deviations or typical behaviour have been noticed.

6.2.4 Execution

The simulation was initialised by the pre-loaded stress state for the FE-model and by start values of positions, velocities and pressures for the fluid model. For all mechanical components the starting positions were related to the corresponding positions in the FE-model, and all initial velocities were set to zero. The initial pressures were set to 15 MPa on the high pressure side, and to 100 kPa on the low pressure side. When the simulation started the flow source was set to deliver a constant oil flow to the percussion unit. The pressure and flow are building-up through the model as the simulation advances, and will eventually reach the Piston and the Control valve, and will due to this start to move. A pressure relief valve on the inlet side was set to adjust the operating pressure in the simulation. After a few working cycles the piston reaches a steady state behaviour, where transients from the start-up phase have faded out. The corresponding running conditions from the experiments were simulated by changing the operating pressure and the restrictor diameter. A total time of 0.3–0.4 s was simulated that resulted in five to six complete working strokes.

6.3 Outcome

Since this study was a validation of the co-simulation approach rather than an investigation of the absolute behaviour of the hydraulic hammer, all simulated responses were normalised against the corresponding results from the experiments. All signals, both from the measurements and the simulations, except the stress in the rig tool, were low pass filtered at 500 Hz to make their basic characteristic appear. Time domain signals from the simulation together with the corresponding signal from the measurement are presented in Fig. 13 for the running condition Case 11. These signals are normalised against the maximum value from the experimental signal.

Normalised values from all running conditions are presented in Table 3, and the calculation of these values are presented in Paper III.

From the time domain signals certain important operating parameters for the hydraulic hammer were calculated, *e.g.* from the piston displacement signal the stroke length and the impact frequency for the piston were estimated. Another important property of a percussion unit is the impact energy, and this was estimated from the primary stress wave in the tool stress signal, see for instance [26]. The impact energy W was calculated using the following equation

$$W = \frac{A c_0}{E_{Steel}} \int_{t_1}^{t_2} \sigma_{Tool}^2 dt, \quad (10)$$

where A is the cross section area of the rig tool and c_0 is the speed of sound in this

Table 3: Normalised percentage values from the simulations with respect to the experiments.

	Case 11	Case 12	Case 21	Case 22
W_{Norm}	97.3	95.3	95.3	96.6
f_{Norm}	100.1	98.4	101.9	101.3
$\Delta u_{P_{Norm}}$	99.2	99.3	99.6	99.8
$Q_{In_{Norm}}$	105.3	108.8	114.9	117.3
$p_{In_{Norm}}$	99.6	100.2	100.2	99.9
$p_{Out_{Norm}}$	132.1	95.0	101.0	78.0
$p_{A_{Norm}}$	100.3	100.0	106.7	106.2
$p_{B_{Norm}}$	98.8	99.5	99.4	99.3
$p_{Ctrl_{Norm}}$	104.2	106.4	108.6	109.5
$\Delta u_{H_{Norm}}$	82.6	109.9	114.8	124.2
$\ddot{u}_{H_{Norm}}$	41.8	66.4	76.1	74.2
$\sigma_{H_{Norm}}$	46.5	50.4	50.0	57.5
$\sigma_{Tool_{Norm}}$	98.1	95.2	99.8	100.4

Figure 14: The time limits t_1 and t_2 used in the calculation of the impact energy from the stress signal in the rig tool.

material, defined as

$$c_0 = \sqrt{\frac{E_{Steel}}{\rho_{Steel}}}, \quad (11)$$

and where the time interval $[t_1, t_2]$ for the integration of the stress curve is shown in Fig. 14. E_{Steel} and ρ_{Steel} are the elastic modulus and the density of the steel, respectively.

In general a very good agreement to the experiments was found for the validation, and the used simulation approach is able to generate responses very similar to the conducted experiments. The simulated responses of the piston and the control valve are very similar to the measured responses, see Fig. 13, where the curves are almost on top of each other, which implies that the simulation model to a large extent represents the real physical mechanisms in the hammer. The comparison also shows that total agreement of the impact frequency is achieved due to the tuning of the restrictor, which is to be expected. However, some minor deviations could be noticed that can be related to the modelling of the fluid system and the boundary conditions of the FE-model, see Paper III. In the results from Case 12, 21 and 22, the same behaviour as for Case 11 could be noticed both for the piston position and the pressure in Cavity A, see Fig. 15.

Figure 15: Comparison between the simulation results and measurements for the different running conditions. Showing the piston displacement for a) Case 12, c) Case 21 and e) Case 22 respectively, and b), d) and f) the pressure in Cavity A for the corresponding running conditions.

6.3.1 Parameter Study

A parameter study was also performed in order to investigate the responses from the simulation model when changing a design parameter. This study involved the following parameters: the operating pressure, which is the time average value of the inlet pressure, $\bar{p}_{In}$, and the restrictor diameter that affects the input flow $\bar{Q}_{In}$ and the impact frequency $\bar{f}$. The responses evaluated were input and output power, P_{Input} and P_{Output} respectively, that were calculated as follows

$$P_{Input} = \bar{p}_{In} \bar{Q}_{In} , \quad (12)$$

$$P_{Output} = \bar{W} \bar{f} , \quad (13)$$

Figure 16: Parameter study.

In Fig. 16 the normalised values of the input and output power are shown, which were normalised with respect to the input and output power from the experimental data of Case 11 respectively.

The simulated responses from parameter changes compares well to the corresponding experimental results, which implies that the simulation approach is able to correct represent a parameter change of the model.

Review of Appended Papers

Paper I

A co-simulation method for system level simulation of fluid-structure couplings in fluid power systems

A co-simulation method is presented for fluid power machinery, that is based on a 1D system simulation model representing the fluid system, and a 3D finite element model representing the structural components. The method was implemented using two well-known simulation tools, Hopsan and LS-DYNA, and a co-simulation interface was realised by the use of the Functional Mock-up Interface standard. The TLM technique is used in Hopsan for the system simulations, and the related major issues were presented and discussed. The advantages for using the TLM technique for the co-simulation are also presented, and the background, the challenges and the details of this implementation are discussed. One section treating the TLM theory related to the co-simulation interface is given. A simple fluid power model, where the main mechanisms for a hydraulic percussion unit are represented, was developed in order to verify the co-simulation approach against a reference model in the stand-alone version of Hopsan. The approach was verified using two different cases, with and without the modelling of the impact force. The verification indicates that correct and stable results are obtained, and that the high frequency excitations are resolved. A third case was also analysed where the structural parts were modelled using linear elastic material properties, and also with relevant contacts, with the aim to demonstrate the potential of this method when full 3D FE-results are available from the simulation.

Paper II

System level co-simulation of a control valve and hydraulic cylinder circuit in a hydraulic percussion unit

A further development of the previously proposed co-simulation approach, where multiple fluid-structure couplings can be defined, is presented. In this study a more complex model was set-up, in which two components were defined for co-simulation. This model, which include the couplings for both the main piston and the control valve, represents the real application to a further extent than the simple model of only one cylinder presented in Paper I. Two models were developed and evaluated, one using a simple rigid body

representation, and one more complex with a linear elastic representation of the structural components. The responses from the co-simulation model for the simple case were compared against the outcome from a reference model in Hopsan, which resulted in a good agreement but with a small time shift, which however can be considered negligible for this application. Typical mechanisms of short duration and high amplitude for the hydraulic percussion unit was found to compare well, and can properly be represented by the co-simulation method. The second case, which is more complex, was meant as a demonstration of the method for an industrial application, closer to reality, and also here the high frequency mechanisms are well represented by the co-simulation method.

Paper III

Validation of a co-simulation approach for hydraulic percussion units applied to a hydraulic hammer

In this paper the proposed co-simulation approach has been adopted to simulate the responses of an existing hydraulic hammer product. In order to validate the simulation model, experiments were performed using four different running conditions. The simulation model was developed using the co-simulation method, and the responses were simulated for the corresponding running conditions. The typical mechanisms in the hydraulic hammer generate high frequency and high amplitude excitations, which require a high resolution of the model dynamics from the simulation model. The comparison against experimental data successfully confirms that the simulation model represents the mechanisms in the hydraulic hammer with good agreement, not only on the overall level but also on the detailed component level. Further, a parameter study was performed to investigate the response from the simulation model when changing the operating pressure and the restrictor diameter. The responses were then compared to the corresponding changes from the experiments. The study confirms a correct response from a design parameter change in the simulation model.

Outlook

The work completed so far in this project has been to develop a simulation tool that is able to capture the most important mechanisms in the fluid- and structural systems of the hydraulic percussion unit. A co-simulation approach has been proposed and has successfully been validated against experiments, and can be considered promising as an efficient tool for this application. In this work three different issues have been identified in order to make this method even more efficient.

- In Paper I and II a small time shift between results from the co-simulation model and the reference model was observed, which effect may be important to analyse when using this method in the general case. For this application the effect of the time shift was considered to be negligible, but for the general case it is important to establish a stability criterion and which parameters that are affecting the numerical accuracy.
- The implementation of the co-simulation interface in LS-DYNA used in this project is not supporting Massively Parallel Processing (MPP), but only Shared Memory Processing (SMP), which means that performance will not scale as good with respect to the number of processors as for the MPP solver. To develop MPP support for the co-simulation interface can reduce the simulation times significantly and must be considered important, especially when the size and complexity of the models and also the analysis times tend to increase.
- The support for multiple FMUs should also be considered since the complexity of the simulation models is already high, and is increasing. Today, the number of FMUs is limited to one sub-model only, which means that all connections representing fluid- and structure couplings must be connected to this FMU. When the number of connections are increased a practical limit will be reached due to the limited number of connection points that can be hosted by one FMU, and also the system model that will be very hard to overlook since connection threads are all over the model. A more practical solution would be to use multiple FMUs, one for each component. However, this is not a trivial solution since the communication to the FE-software must be synchronised for all FMUs.

The ambitious goal is to develop a complete simulation tool for the hydraulic hammer, where all important mechanisms are represented, both internal and external. The internal mechanisms can be internal leakage, seizure and cavitation, and the external mechanisms can be wear, vibration and acoustic radiation. Other system related properties are also

important for such a tool, *i.e.* performance, efficiency and productivity, where the latter is a measure for the volume of fractured material.

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Part II

Appended Papers

Appended Papers

The appended papers associated with this thesis have been removed for copyright reasons. For more details about these see:

<http://urn.kb.se/resolve?urn=urn:nbn:se:liu:diva-151018>