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COUPLED FLEXIBLE MULTIBODY DYNAMICS AND HYDRAULIC SIMULATION FOR A LOAD DEVICE

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ABSTRACT

Ville-Pekka Saarela: Coupled Flexible Multibody Dynamics and Hydraulic Simulation for a Load Device
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A hydraulic driven load device is actuated by two hydraulic cylinders. The hydraulic cylinders are connected to opposite sides of the rotating frame of the load device. At the start of the work cycle cylinders receive half of the fluid flow generated by the flow source. After a while regenerative loading cycle is used. In regenerative loading one cylinder receives full fluid flow from the flow source and the other cylinder feeds itself. This causes cylinder forces that have different magnitude and opposite direction for each hydraulic cylinder. These forces affect the load devices structure with torsion.

To determine deformations for the structure a coupled flexible multibody dynamics and hydraulics simulation is done. Component Mode Synthesis (CMS) method is used for flexibility of a part of the structure for multibody simulation. CMS method is a linear modal analysis based method. Flexible multibody model is driven by a hydraulic system model. Due to contact non-linearities in load devices structure, joint forces are imported to non-linear dynamic analysis to determine deformations caused by the load cycle.

Results of interest for the hydraulic system are the pressures in different places, cylinder forces and velocities and fluid flow rate in hydraulic pipes. Joint forces caused by the hydraulic system are the primary results which are transferred as boundary conditions for the second simulation that calculates deformations of the structure.

Keywords: Coupled, Simulation, Hydraulics, Multibody, Dynamics, System simulation

TIIVISTELMÄ

Ville-Pekka Saarela: Kytketty Joustava Monikappaledynamiikka ja Hydrauliikka Simulointi Kuormalaitteelle
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Hydrauliikalla toimivaa kuormalaitetta käytetään kahdella hydrauliikkasylinterillä. Hydrauliikkasylinterit ovat kiinnitetty vastakkaisille puolille kääntyvään runko-osaan kuormalaitteessa. Aluksi laitetta käytetään normaalilla nopeudella, jolloin molempiin sylintereihin virtaa puolet nestevirtauksesta. Hetken kuluttua käytetään regeneratiivista kiertoa, jossa toiseen sylinteriin ohjataan täysi nestevirtaus ja toinen sylinteri syöttää itseään. Tästä aiheutuu eri kokoiset ja vastakkaissuuntaiset sylinterivoimat. Nämä voimat aiheuttavat kuormalaitteen runkoon vääntöä.

Väännön aiheuttamien muodonmuutosten selvittämiseksi diplomityössä tehdään kytketty joustava monikappaledynaamiikka ja hydrauliikkasimulointi. Rakenteen joustavuuden huomioon ottamiseksi käytetään Component Mode Synthesis (CMS) menetelmää. CMS-menetelmä on ominaistaajuusanalyysiin perustuva lineaarinen menetelmä. Joustavaa monikappalemallia ajetaan hydraulisella systeemimallilla. Kuormalaitteen rakenteen epälineaaristen kontaktien takia, nivelvoimat siirretään kytketystä analyysistä epälineaariseen dynaamiseen simulaatioon, jotta rakenteen muodonmuutokset voidaan selvittää.

Tutkimuksen mielenkiintoiset suureet hydrauliikan osalta ovat paineet eri hydrauliikkapiirin paikoissa, sylinterivoimat ja nopeudet sekä virtausnopeudet putkissa. Kytketyn simulaation nivelvoimat siirretään reunaehdoiksi rakenteen dynaamiseen simulointiin, jossa lasketaan rakenteen muodonmuutokset.

Avainsanat: Kytketty, Simulointi, Hydrauliikka, Monikappale, Dynamiikka, Systeemisimulointi

PREFACE

This masters thesis was done as a co-operation with Cargotec Hiab and EDRMedeso. Juuso Nänimäinen from EDRMedeso and Matti Randelin, Mikko Nyström and Erkki Ahola from Hiab were a huge help and support during this process. I would also like to thank professor Reijo Kouhia for guidance during, before and after this thesis.

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Tampere, 1st November 2019

Ville-Pekka Saarela

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LIST OF SYMBOLS AND ABBREVIATIONS

a	hydraulic cylinder acceleration
α	rotation angle of hooklift
A	area
a	hooklifts geometrical constant
b	hooklifts geometrical constant
CMS	component mode synthesis
d	diameter
DOE	design of experiments
DOF	degree of freedom
F_{ext}	hydraulic cylinder external force
$\underline{F}$	force vector
F_{stk}	hydraulic cylinder stick friction force
F_{clb}	hydraulic cylinder Coulomb friction force
F_{vsc}	hydraulic cylinder viscous friction force
F_{frc}	hydraulic cylinder friction force
FEA	finite element analysis
FEM	finite element method
γ	angle used for determining cylinder force components
h	timestep
$\mathbf{I}$	identity matrix
K	slope coefficient of friction force
$\mathbf{K}$	stiffness matrix
L	cylinder lenght
LHV	load holding valve
$\mathbf{M}$	mass matrix
m	mass
$\mathbf{\Phi}_s$	eigenvector matrix
p	pressure
Q	volume flow

RBD	rigid body dynamics
ROM	reduced order model
t	time
$\mathbf{T}$	transformation matrix
$\underline{\ddot{u}}$	acceleration vector
$\underline{\hat{u}}$	nodal displacement vector, reduced coordinates
$\underline{u}$	nodal displacement vector
$\underline{\dot{u}}$	velocity vector
v	volume flow average velocity
$y'(t)$	derivative of time dependant function y
$y(t)$	time dependant function y

1 INTRODUCTION

Subject for this thesis is a load device that can load different types of containers onto a vehicle. That motion is created by a hydraulic system. The main interest of the simulations is to understand what kind of response does the hydraulic system create in the load device. Also the hydraulic system is affected by the load the loading device is handling.

Simulation of the coupling between the loading device and its hydraulic system is done in Ansys Workbench environment. Ansys is an American company that develops and markets different simulation softwares for engineering. Simulation softwares vary from structural analysis software to electrical simulations. The load device is simulated in Rigid Dynamics module of Ansys Workbench and hydraulic system is simulated in Ansys Twin Builder. The two systems are coupled by inserting Rigid Body Dynamics model created in Rigid Dynamics module into the hydraulic system model in Twin Builder.

Flexibility of the load devices structure is done by creating superelements of substuctures in Rigid Dynamics using Component Mode Synthesis (CMS) method. Superelement is a system of finite elements method (FEM) elements. Systems unknowns are reduced into unknowns of the interfaces that the superelement is connected to using natural vibration modes. This method allows reducing the hundreds of thousands unknowns into a few depending on the size of the substucture and mesh of the FEM. CMS method used in Ansys is based on book by R.R Craig (Craig 1981).

1.1 Background and motivation

Workflow using commercial programs for simulating coupled response of mechanical and hydraulic systems was studied in this thesis. The load device is driven by a hydraulic system, which will create different loading conditions for the structure. The behaviour of the hydraulic system is controlled by valves and the mechanical response for those changes is studied. As the load device is actuated by the hydraulic system it will change the response of the hydraulic system depending on position of the structure. For these reasons two way coupling of mechanical and hydraulic domains is needed. The most important change in the hydraulic system is to assign the hydraulic fluid flow from the pump to one cylinder instead of both cylinders. When the fluid flows to hydraulic cylinders equally from the pump, the cylinders are said to actuate with normal velocity. When the fluid flow from the pump is directed to only one of the cylinders the velocity of the cylinders is doubled and it is called regenerative loading cycle.

The load device also known as hooklift consists of different parts that are connected with joints, welds and bolted connections. The hooklift is divided into different frames. The hooklift can be fitted to different main frames and in the figure 1.1 it is connected to a frame of a truck. The truck frame is connected to the wheels of the truck. The hooklift itself is connected to the truck frame from subframe with bolts. Rear frame is supported by a shaft from the rear and in the front subframe has stops where the rear frame rests. Rear frame can rotate during tipping. Tipping means that the whole body frame is rotated and this is used for example during winter when roads are icy and sand grains are used to increase friction. When unloading body frame rear frame is fixed to subframe with rear frame locks. Body frame can have containers or similar structures attached to it. Middle frame is supported by the rear frame and hydraulic cylinders and its rotation is the work cycle this masters thesis is studying. Finally the hook frame is inside the middle frame and it is also actuated by another hydraulic cylinder which enables it to move forwards and backwards.

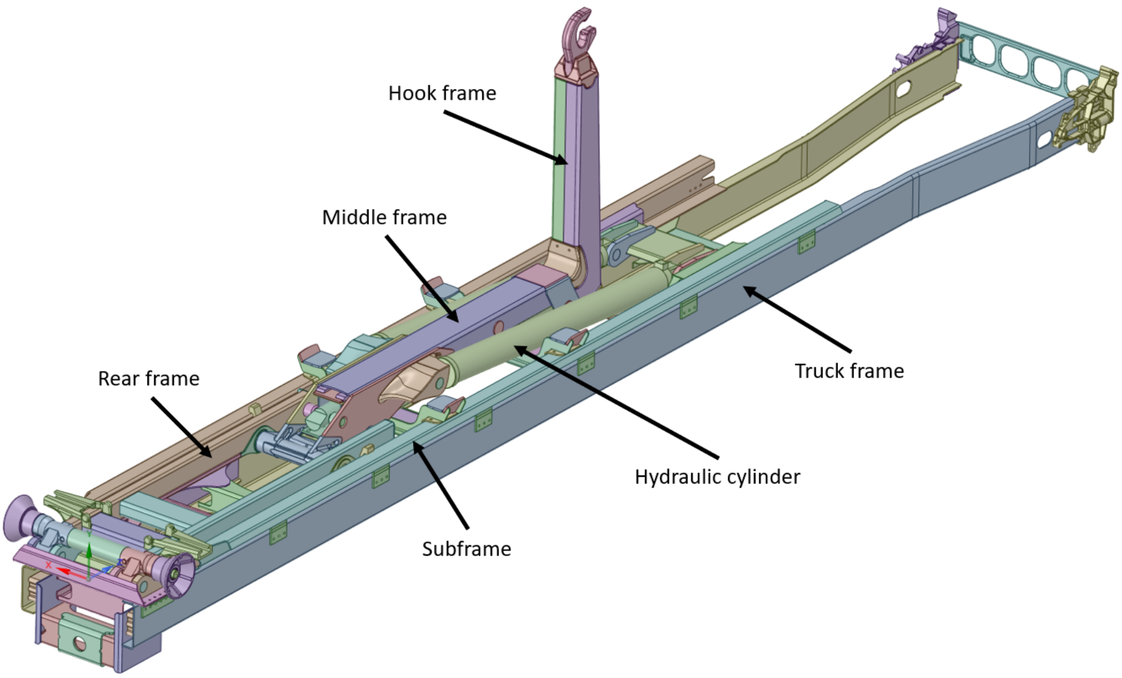

Figure 1.1. *Structure of the load device.*

Regenerative cycle used to create faster loading speed creates cylinder forces of opposite directions and this causes torsion to the frame of the load device. Simulating deformations caused by the torsion is one of the largest motivations for this thesis. Also hydraulic system results are interesting and they are provided by the simulation. Especially how different hydraulic parameters affect the system. However varying these parameters is outside the scope of this masters thesis. Varying hydraulic parameters would be interesting to see how they affect the systems response.

It is challenging to evaluate dynamic forces created by hydraulic cylinders for structural simulation without coupling the two different physical domains. Assumption was that because the external load that the hydraulic system experience changes depending on the position of the frame, the two way coupling is needed. Hydraulic cylinders are actuated by fluid flow which controls the velocity of the hydraulic cylinders. Fluid flow quantity is an input value and the pressure is determined by pressure losses in the system which the fluid flow generates. Pressure difference fluctuates over time because of the fluid flow dependent losses and operating conditions. Because of this it is challenging to define forces needed to move a complex structure without coupling the multibody system and the hydraulic system.

The hydraulic system is challenging to simulate because the components have parameters that are not known or need tests to be determined. Test data is really useful for determining pressure losses of components which can then be used to determine parameters that components have.

2 THEORY

In this chapter methods needed to solve coupled flexible multibody dynamics and hydraulic system are presented. The kinematics of a multibody system is presented in the first section. In the next section CMS method is presented to account flexibility of the structure. Third section introduces theory of hydraulic system and its components. In the final section time integration methods that are used to solve dynamic problems are presented.

Rigid body motion is not explained in this thesis but it is well explained in the literature (Bauchau 2011).

2.1 Multibody analysis

Multibody analysis means analysing the dynamic behaviour of a system of bodies interconnected with joints. Bodies can be rigid or flexible and in addition to joints, springs and dampers can be used for flexibility. In the context of this thesis, joints are the primary focus and flexibility of the structure is created using CMS-method.

Rigid body dynamics is the study of motion of different bodies connected with joints that do not deform, but instead move rigidly in 3D space. These joints restrain the free motion of the bodies. The joints idealize contact between the two bodies. Joints are characterized by the motion that they allow between the two bodies which they connect. For example, a translation joint allows relative motion in one translational direction and constraints two other translations and all three relative rotations.

The primary unknowns are the translation and rotation of each body and the motion in the joints themselves. The output quantities of rigid body dynamics are the forces that develop in the joints and flow through the rigid bodies, as opposed to a structural FEA where the output quantities are strains or stresses.

For the system to be in stable equilibrium, sufficient amount of kinematic constraints are necessary. Sufficient amount of kinematic constraints leads to an unique solution of the system dynamics. Generalized coordinates represent the time history of a particle. For two particles connected with a rigid bar, four Cartesian coordinates define the position of the particles in two-dimensional space. Because of the rigid bar there is a kinematic constraint for the system and only three generalized coordinates are independent. This leads to concept of degrees of freedom (DOF), which can be define as

$$d = n - m, \tag{2.1}$$

where d is the number of degrees of freedom, n is number of generalized coordinates and m is number of kinematic constraints. Two particles with rigid bar in between has four generalized coordinates, x and y Cartesian coordinates for both particles. One kinematic constraint, which is the rigid bar that constraint the particles to always have same distance between them. Using above equation we have three degrees of freedoms for the system. Number of DOFs depends on the system affects how the system behaves. This is covered in detail by (Bauchau 2011, pp. 259-263).

2.2 Flexible multibody analysis

Multibody systems have conventionally been modelled as rigid body systems with springs for the elastic effects of one or more components. A major limitation of these methods is that nonlinear large-deformation, or nonlinear material cannot be incorporated completely into a model.

For transient dynamic analyses, stiff bodies can excite high-frequency modes, resulting in a small time increment in order to obtain a stable solution. Rigid bodies do not excite any frequency modes; therefore, using rigid bodies to represent stiff regions may allow a relatively large time increment.

Finite element method is used to mesh the flexible bodies and this results in many unknowns as each element has several nodes depending on the type of element used. Each node has three unknowns when we consider solid elements. Basics of FEM is covered exhaustively in the literature (Hughes 2000).

Obtaining the flexible response of a body or bodies to a dynamic motion event typically involves solving hundreds or thousands of time points. If a flexible body has many unknowns, a multibody analysis can be time-consuming. To minimize the necessary computing resources, usage of CMS superelements to replace the many thousands of unknowns of the flexible body with tens of unknowns that represent the dynamic response, thereby significantly reducing the required multibody analysis run time.

Component mode synthesis is a technique developed to compute the eigenmodes of very large structures in an efficient way. Idea is to divide the structure to substructures so that the eigenmodes are easily calculated. Each substructure involves two types of unknowns. First are the boundary nodes that also known as master nodes which connect the substructure to another substructure and modal unknowns that represent the internal flexibility (Bauchau 2011).

Modal analysis of the substructures is a common technique to define dynamics of a structure. Natural vibration modes characterize the dynamic behaviour of small perturbations of a system. The modes do not capture any nonlinear behaviour of the system. Examples of nonlinearities are structures changing stiffness with deformation or nonlinear material models like plasticity. Capturing enough of the eigenmodes for the interesting deformations is needed for accurate response. Banerjee discusses mode reduction in detail (Banerjee 2016).

In linear structural analysis, the general form for equations of motion is

$$\mathbf{M}\underline{\ddot{u}} + \mathbf{C}\underline{\dot{u}} + \mathbf{K}\underline{u} = \underline{F}, \tag{2.2}$$

where $\mathbf{K}$ is structural stiffness matrix, $\mathbf{M}$ is structural mass matrix, $\mathbf{C}$ is structural damping matrix and $\underline{F}$ is load vector.

We can partition the matrix equation 2.2 into interface and interior unknowns.

$$\underline{u} = \begin{bmatrix} \underline{u}_m \\ \underline{u}_s \end{bmatrix} \tag{2.3}$$

$$\mathbf{M} = \begin{bmatrix} \mathbf{M}_{mm} & \mathbf{M}_{ms} \\ \mathbf{M}_{sm} & \mathbf{M}_{ss} \end{bmatrix} \tag{2.4}$$

$$\mathbf{C} = \begin{bmatrix} \mathbf{C}_{mm} & \mathbf{C}_{ms} \\ \mathbf{C}_{sm} & \mathbf{C}_{ss} \end{bmatrix} \tag{2.5}$$

$$\mathbf{K} = \begin{bmatrix} \mathbf{K}_{mm} & \mathbf{K}_{ms} \\ \mathbf{K}_{sm} & \mathbf{K}_{ss} \end{bmatrix} \tag{2.6}$$

$$\underline{F} = \begin{bmatrix} \underline{F}_m \\ \underline{F}_s \end{bmatrix}, \tag{2.7}$$

where subscripts m denotes master DOFs defined only on interface nodes and s are all DOFs that are not master DOFs. In a substructuring analysis, the substructure's nodal displacement vector, $\underline{u}$, is represented in terms of reduced coordinates, $\underline{\hat{u}}$, by the coordinate transformation

$$\underline{u} = \mathbf{T}\underline{\hat{u}}, \tag{2.8}$$

where $\mathbf{T}$ is the transformation matrix shown in equation 2.10. Nodal displacement vector $\underline{u}$ can be represented in terms of master DOFs completed by component generalized coordinates

$$\underline{u} = \mathbf{T} \begin{bmatrix} \underline{u}_m \\ \underline{y}_\delta \end{bmatrix} \tag{2.9}$$

For fixed-interface method the transformation matrix

$$\mathbf{T} = \begin{bmatrix} \mathbf{I} & \mathbf{0} \\ \mathbf{G}_{sm} & \mathbf{\Phi}_s \end{bmatrix}, \tag{2.10}$$

where $\mathbf{I}$ is identity matrix, $\mathbf{G}_{sm} = -\mathbf{K}_{ss}^{-1}\mathbf{K}_{sm}$, $\mathbf{0}$ is null matrix and $\mathbf{\Phi}_s$ eigenvectors obtained with interface nodes fixed.

When introducing equation 2.8 to equation 2.2 and multiplying from left with $\mathbf{T}^T$ a reduced equation of motion is obtained

$$\hat{\mathbf{M}}\ddot{\hat{\underline{u}}} + \hat{\mathbf{C}}\dot{\hat{\underline{u}}} + \hat{\mathbf{K}}\hat{\underline{u}} = \hat{\underline{F}}, \tag{2.11}$$

where $\hat{\mathbf{K}} = \mathbf{T}^T\mathbf{K}\mathbf{T}$ is reduced stiffness matrix, $\hat{\mathbf{M}} = \mathbf{T}^T\mathbf{M}\mathbf{T}$ is reduced mass matrix, $\hat{\mathbf{C}} = \mathbf{T}^T\mathbf{C}\mathbf{T}$ is reduced damping matrix and $\hat{\underline{F}} = \mathbf{T}^T\underline{F}$ is reduced load vector. In the reduced system, master DOFs are used to couple the CMS superelement to other elements. The displacements at slave DOFs $\underline{u}_s$ are recovered from the lower part of equation 2.9. Stresses can then be calculated using stress-displacement matrix. Full equations and derivations can be found in (*ANSYS APDL Theory Reference* 2016).

2.3 Hydraulics

In this section basics of hydraulics and hydraulic components that are relevant to this thesis are presented. More information about hydraulics can be found in the literature (Jelali and Kroll 2004).

Jelali (2004, p.2) states that the main advantage of fluid power, is the good ratio between force or torque delivered on the one hand and the actuator weight and size on the other hand.

In figure 2.1 a basic structure of a hydraulic system is presented. Fluid flow is generated by the pump which is rotated by a motor. The system is controlled by a directional valve which can have different kinds of controller that control the position of the valve. Actuating element can be hydraulic cylinder that moves a load depending on which side is the fluid flow from pump coming.

2.3.1 Hydraulic system simulation

In hydraulic system simulation component models of hydraulic components are connected to each other. The component models usually are idealized analytical models which use differential equations to represent behaviour of the component. These components can generate fluid flow (pumps), act as a resistance to flow (orifaces, pipes) or transform fluid flow to mechanical work (actuators). The models are combined into

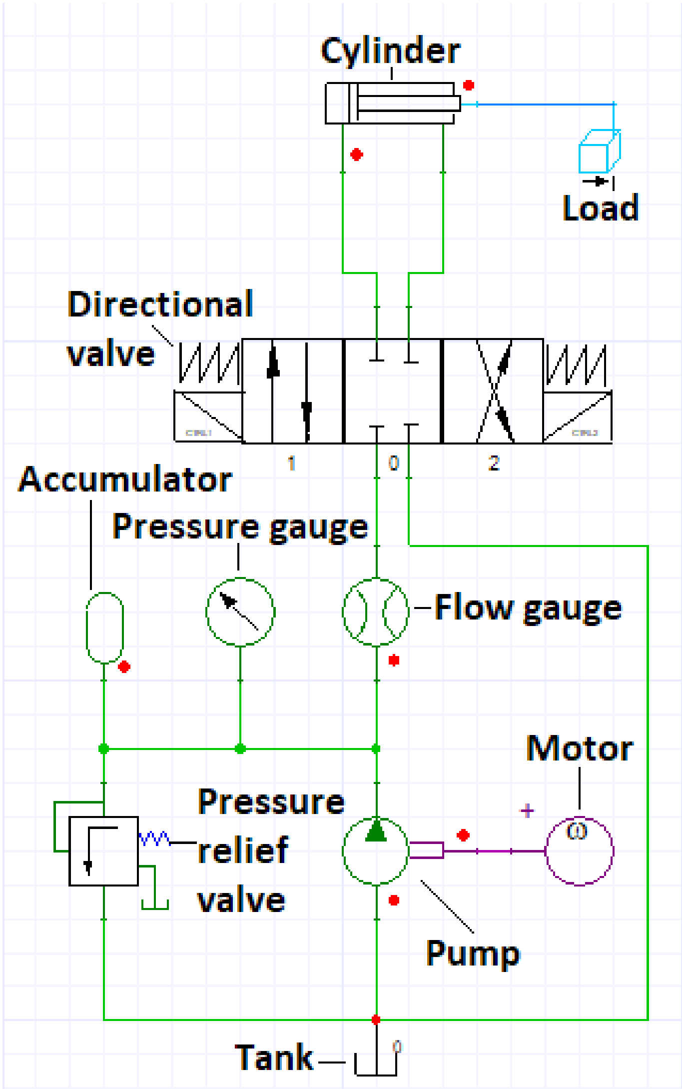

Figure 2.1. *Basic structure of hydraulic systems.*

system of differential equations which is then solved using time integration methods.

In fluid flow domain the state variables are pressure and volume flow. The system components have certain flow resistances which will cause the pressure to drop when fluid flows through the system. Flow resistance is usually a function of fluid flow velocity. Flow source must generate more pressure than what the pressure drop is to generate fluid flow though the system.

Fluid properties include density and bulk modulus which tells how compressible the fluid is. The bulk modulus can be important parameter in systems where the fluid flow suddenly is stopped by a valve or when the fluid has gas which would compress more when pressure changes. In this thesis the oil is thought as relatively incompressible and bulk modulus is quite high.

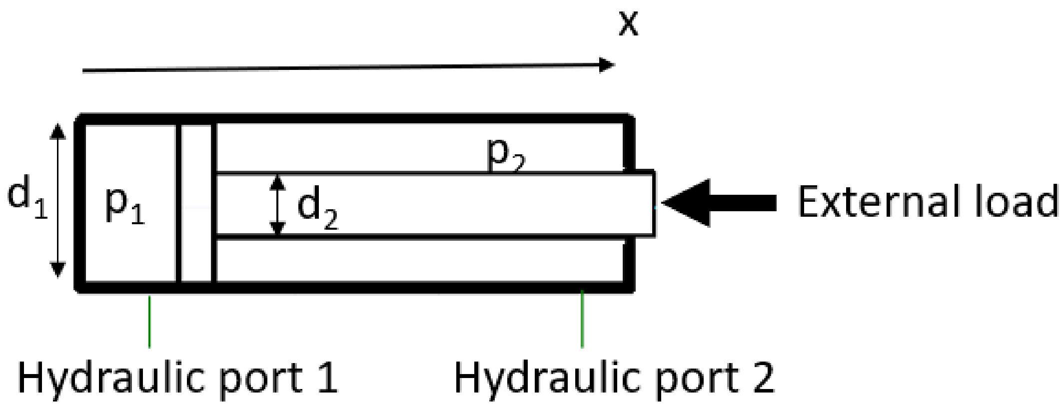

Figure 2.2. *Double-acting asymmetric hydraulic cylinder.*

2.3.2 Hydraulic cylinder

Hydraulic cylinder is a linear actuator that is commonly used in hydraulic systems. Cylinders are used to convert hydraulic power into linear mechanical force or motion. There are different types of hydraulic cylinders. Single-acting cylinder permits the application of hydraulic force only in one direction. Double-acting cylinders allow hydraulic force in both directions.

Equation of motion for hydraulic cylinder can derived from figure 2.2. In the figure x is the position of the cylinder, d_1 is diameter of the cylinder, d_2 is the diameter of the rod and F_{ext} is the external force. Hydraulics ports are the ports where fluid is entering and exiting the cylinder. Velocity of the rod is v, positive sign means that the direction of the motion is toward rod end and p_1 is the pressure in the cylinder end, p_2 is the pressure in rod end.

$$ma = p_1A_1 - p_2A_2 + F_{ext} = \frac{\pi p_1 d_1^2}{4} - \frac{\pi p_2}{4}(d_1^2 - d_2^2) + F_{ext}, \tag{2.12}$$

We also need to couple velocity of the rod to the fluid flow.

$$Q_1 = vA_1 = \frac{\pi}{4} v d_1^2, \tag{2.13}$$

Similarly for other side

$$Q_2 = vA_2 = \frac{\pi}{4} v d_2^2, \tag{2.14}$$

We can also include friction term in the equation of motion. Friction term

$$F_{frc} = sign(v)\Big((F_{stk} - F_{clb})e^{-\frac{|v|}{K}} + F_{clb} + K_{vsc}|v|\Big), \tag{2.15}$$

where F_{stk} is sticking friction force, F_{clb} is Coulomb friction force and K_{vsc} is viscous friction and K is slope coefficient.

2.4 Time integration methods

Numerical methods are used to solve the resulting ordinary differential equations. Often it is not possible to solve differential equations analytically and numerical methods have been developed. Numerical methods provide an approximate solution for the equations.

Several methods exists and simulation tools use different time integration methods depending on the dynamics of the problem. Rigid body dynamics usually use explicit time integrations but when flexibility is included with CMS method implicit time integration should be used. System simulation of the hydraulic system also uses implicit methods.

Ansys simulation tools used in this thesis use advanced time integration methods so it is useful to first introduce the most simplest time integration methods for both implicit and explicit.

2.4.1 Explicit time integration methods

Starting from a first order differential equation with initial value

$$y'(t) = f(t, y(t)), y(t_0) = y_0, \tag{2.16}$$

where y is unknown function of time t, f is a known function and y_0 is the initial value. To solve this problem with numerical method we can see that from a point in a curve we can move to the direction of the tangent of that point and find and an approximation of another point on the curve. Mathematically

$$y'(t_n) \approx \frac{y(t_n + h) - y(t)}{h}, \tag{2.17}$$

where h is the time step and y is the function and t is time. We can re-arrange the terms

$$y(t + h) \approx y(t) + hy'(t), \tag{2.18}$$

which can be constructed as

$$y_{n+1} = y_n + hf(t_n, y_n). \tag{2.19}$$

Using this recursive scheme we can solve value of the function. This method is know as forward Euler method named after Leonhard Euler. This method is an explicit method which means that y_{n+1} is defined from known y_n. Small time step will make the approximation more accurate as the tangent of a point is closer to the curve with smaller time step. This also means that approximation generally diverges more with larger time step.

Rigid multibody dynamics uses fourth order Runge-Kutta scheme. Fourth order Runge-

Kutta

$$y_{n+1} = y_n + \frac{1}{6}(k_1 + 2k_2 + 2k_3 + k_4). \tag{2.20}$$

The increments $k_1 = hf(t_n, y_n), k_2 = hf(t_n + \frac{h}{2}, y_n + \frac{k_1}{2}), k_3 = hf(t_n + \frac{h}{2}, y_n + \frac{k_2}{2})$ and $k_4 = hf(t_n + \frac{h}{2}, y_n + \frac{k_1}{3})$ (Wood 1990).

2.4.2 Implicit time integration methods

We get implicit time integration method for the initial value problem if instead of equation 2.17 we use

$$y'(t) \approx \frac{y(t) - y(t-h)}{h}. \tag{2.21}$$

Then we can construct the equation as

$$y'_{n+1} \approx \frac{y_{n+1} - y_n}{h}. \tag{2.22}$$

Solving the recursive scheme we get

$$y_{n+1} = y_n + hf(t_{n+1}, y_{n+1}). \tag{2.23}$$

The difference to equation 2.19 can be seen in the last term. This is called implicit method because term y_{n+1} is on both sides. This equation is usually solved with Newton-Raphson method.

Flexible multibody dynamics solver uses implicit generalized α method. This implicit method is a generalized Hilber-Hughes-Taylor (HHT) method that allows high frequency energy dissipation. Hydraulic system simulation solver uses adaptive trapezoidal time integration method. Adaptivity means that the step size can change and the trapezoid refers to trapezoidal rule which is given by formula

$$y_{n+1} = y_n + \frac{1}{2}h\big(f(t_n, y_n) + f(t_{n+1}, y_{n+1})\big). \tag{2.24}$$

3 COUPLED MULTIBODY DYNAMICS AND HYDRAULIC SYSTEM SIMULATION

Coupled multibody dynamics and hydraulic system simulation means that the hydraulic system and the multibody dynamics are solved simultaneously. This way hydraulic system and multibody system responses affect each other. This is defined as two-way coupling. In one-way coupling first system would only affect the second system but the change in second system would not affect the first system.

In this chapter we discuss multibody and hydraulics systems individually. In first section multibody model is discussed along its components. In the second section hydraulic system is presented. Third section is about reduced order models (ROM) used in this thesis.

3.1 Multibody model

A multibody model is created by connecting components together with joints. Components can have single or multiple parts. A component that has multiple parts is called a multibody part. Multibody parts are useful method for combining multiple parts without the need for manually creating fixed joints. In a rigid system a component is presented as a point mass at the component center of gravity. The point masses are coupled together with constraint equations that tie those coordinates together algebraically. Flexible bodies are modelled so the components share a common node with adjacent elements.

3.1.1 Joints

In a rigid multibody model, joints that are used in this thesis simulation model are fixed, revolute, translational and cylindrical. Fixed joint has zero free DOFs. A revolute joints has one free DOF, rotation over z-axis. Translational joint also has one free DOF, translation in the direction of x-axis. A cylindrical joint has two free DOFs, translation in direction z-axis and rotation over z-axis. Axis can be defined to any local cartesian coordinate system.

One part has six degrees of freedom in three-dimensional space, in Cartesian coordinates x,y and z coordinate and rotation in respect to each axis. For multibody model

$$6m - n < 1, \tag{3.1}$$

where m is number of bodies and n is number of DOFs that the applied joints constrain the system. For example revolute joint restricts five DOFs. If equation 3.1 is true then the system is overconstrained.

When defining joints, equation (3.1) tells if there is a need for redundancy analysis. Redundancy means that there are joints that constrain the same DOF.

In flexible multibody model joints need to be modeled so that they do not overconstrain the flexible part. If joints are overconstraining the structure, joint forces are not transmitted correctly through the superelement. This is where general joints are useful as they allow selecting which DOF's are free and which are fixed.

Generally the interface joints for the superelement needs to be defined as general joints. In these interfaces rotational DOF's can generally be free to allow flexible part to rotate. These changes can also be thought as clearance the real structure has in its joints although clearance could also be modeled and simulated. This however would be computationally more expensive.

Joint stops and locks can be used to restrict relative displacement or rotation of the joint. However it is adviced that these are not defined in the multibody model but in the system model. In this thesis joint stops or lock were not needed to be defined because hydraulic cylinder model has flexible ends defined that restrict the displacement near the ends of the cylinder.

3.2 Hydraulic system model

Hydraulic system model has four main components: a flow source, valves, cylinders and a valve control system. In this thesis the flow source has been idealized as a constant fluid flow generation. The system has two different kind of valves: directional valves and a load holding valve (LHV). Cylinders are asymmetric double-acting actuators. Valve control for directional valves is done by limit switches attached to hydraulic cylinders position. LHV is a functional model and is discussed in more detail later.

3.2.1 Regenerative cycle

The most interesting work cycle is the regenerative cycle. It means that instead of fluid flowing from pump to the one end of the cylinder and then from other side of the cylinder into the tank the cylinder feeds itself from the other end. In coupled two cylinder system this results to faster cylinder velocity but also cylinders have opposite directions of cylinder forces. The cylinder with normal cycle needs to work against the regenera-

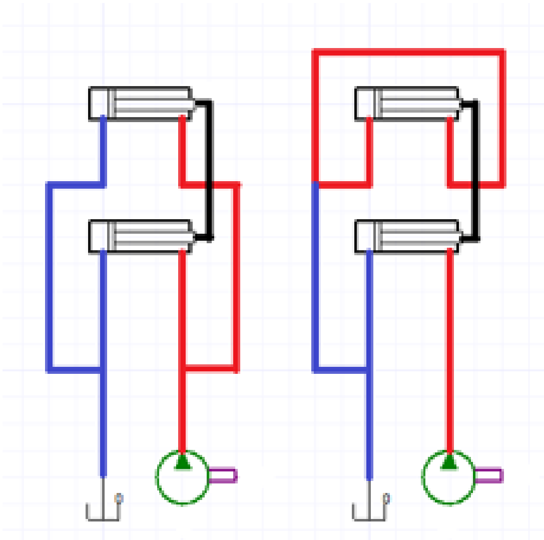

Figure 3.1. *a) Normal cycle b) Regenerative cycle. Red color means fluid flow to cylinders and blue means fluid flow to tank*

tive cycle cylinder which results as torsion to the structure. It can result as relative large displacements and stresses for the structure.

Simulation starts with normal cycle where fluid flow is directed to the rod end of the both cylinders and the cylinders move inwards. In other words each cylinder receives half of the total fluid from the flow source which is directly proportional to cylinder velocity provided that the pressure difference between the cylinder chambers is enough to actuate the loading. After some time regenerative loading start and the fluid flow is directed to one cylinder. The fluid flow from the other cylinders bottom end is allowed to flow in to the rod end and also to the tank.

First cylinder has higher pressure in the rod end and the regenerative cylinder has higher pressure in the bottom end. This results as opposite direction of cylinder forces where regenerative cylinder works against the direction of the motion.

Fluid flow also behaves differently with regenerative loading. In asymmetrical hydraulic cylinder input volume flow is different than output volume flow because of different cross section areas in the piston. When cylinder moves inwards like in the simulation case, output flow is larger than input flow. With regenerative loading one cylinder feeds itself but not all of the output flow can feed the cylinder. The excess flows to the tank. This extra fluid increases pressure losses of the components as pressure loss is a function of the fluid flow.

3.3 Reduced Order Model

Coupled rigid multibody and hydraulic system simulation is fast to compute but cannot provide results of the structures deformation. Reduced order model (ROM) of static structural analysis of the structure would approximate deformations but it does not take account the dynamics of the system.

Because most of the loading cycle is with constant cylinder velocity and therefore ROM model should provide accurate results for constant loading. When loading starts and fast loading is toggled on and off results are not accurate because there is acceleration in the system. This is acceptable because dynamic effects can be analyzed with flexible model.

ROM is generated using different frame angles and cylinder forces as inputs and rear frame displacement in specific points as output. ROM has similar idea as Design of Experiments (DoE). DoE is done by varying different parameters like geometrical properties or material properties. Instead in this case input parameters are varied and the output results are collected. Then a response surface is generated with algorithms that find the best fit with the calculated results.

Response surface can then be part of the system model where the system gives the input values to the ROM and the ROM interpolates the output values. Using this ROM technique we can have approximate values of rear frames deformation with fully rigid multibody model.

Parameters for the generation of the response surface are hooklift frame angle and cylinder forces. For the FEM model force input needs to divided into x and y components which can be done with trigonometry using geometry of the hooklift.

In the figure 3.2 these cylinder forces are show and the relation to position of the hooklifts frame is shown. Figure is simplified for easier demonstration. Using laws of sine and cosine magnitude of the force components can be calculated.

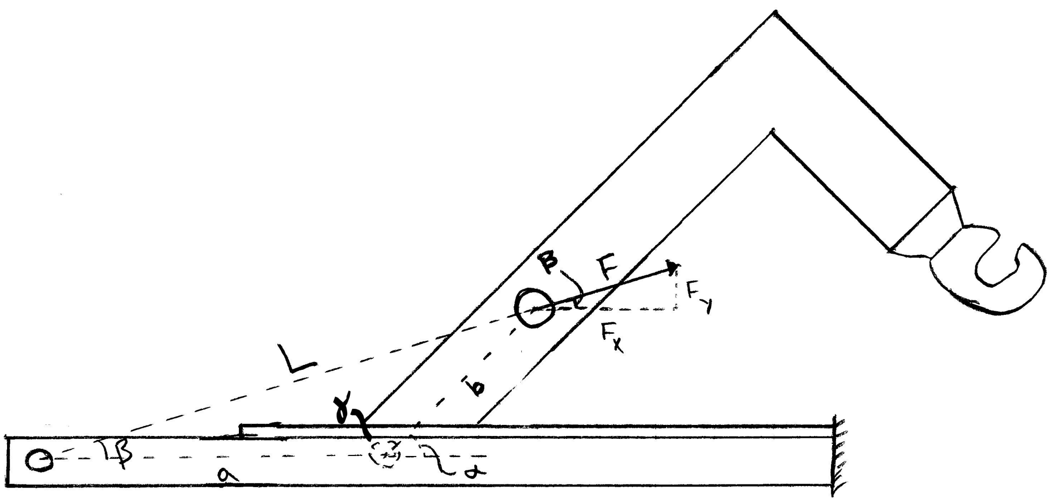

Figure 3.2. *Hand drawn sketch of force components.*

If we simplify the triangle in figure 3.2 to figure 3.3 where β is the unknown angle and L is the unknown distance but a,b and γ are known we can solve both β and L.

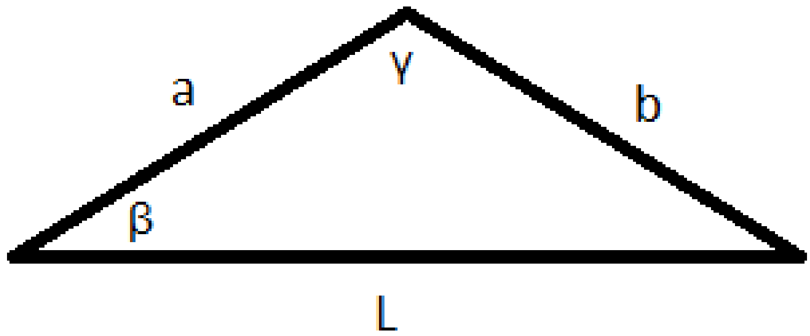

Figure 3.3. *Triangle showing angles and edge lengths for reduced order model.*

Parameters a and b are geometrical distances of the hooklifts frame which we can assume to be constant. Angle γ is

$$\gamma = 180^\circ - \alpha, \tag{3.2}$$

where α is the rotation angle of the hooklift. This formulation could be simplified but it was decided that rotation angle increases when cylinders move inwards. The length L can be solved from the law of cosine

$$L^2 = a^2 + b^2 - 2ab\cos\gamma \tag{3.3}$$

$$\Rightarrow L = \sqrt{a^2 + b^2 - 2ab\cos\gamma}. \tag{3.4}$$

Now we can solve β using the law of sine

$$\frac{b}{\sin\beta} = \frac{L}{\sin\gamma} \tag{3.5}$$

$$\Rightarrow \beta = \arcsin(\frac{b}{L}\sin\gamma). \tag{3.6}$$

When we input L from above equation we get

$$\Rightarrow \beta = \arcsin(\frac{b\sin\gamma}{\sqrt{a^2 + b^2 - 2ab\cos\gamma}}). \tag{3.7}$$

Now the hydraulic cylinder force components are now

$$F_y = F\sin\beta = \frac{Fb\sin\gamma}{\sqrt{a^2 + b^2 - 2ab\cos\gamma}} \tag{3.8}$$

and

$$F_x = F\cos\beta = F\sqrt{1 - \Big(\frac{b\sin\gamma}{\sqrt{a^2 + b^2 - 2ab\cos\gamma}}\Big)^2}. \tag{3.9}$$

$$\Rightarrow F_x = F\sqrt{1 - \frac{b^2 \sin^2 \gamma}{a^2 + b^2 - 2ab\cos\gamma}} \tag{3.10}$$

We can plot the force components as a function of the rotation of the frame. Results can be seen in figure 3.4. Values used to plot the curve are a=3.4, b=1 and the results is normalized with the force F. It can be seen that most of the force generated by the hydraulic cylinder is going to x-axis force component. Also it is good to notice that F_y changes quite a lot relatively and that creates the large displacements in the frame.

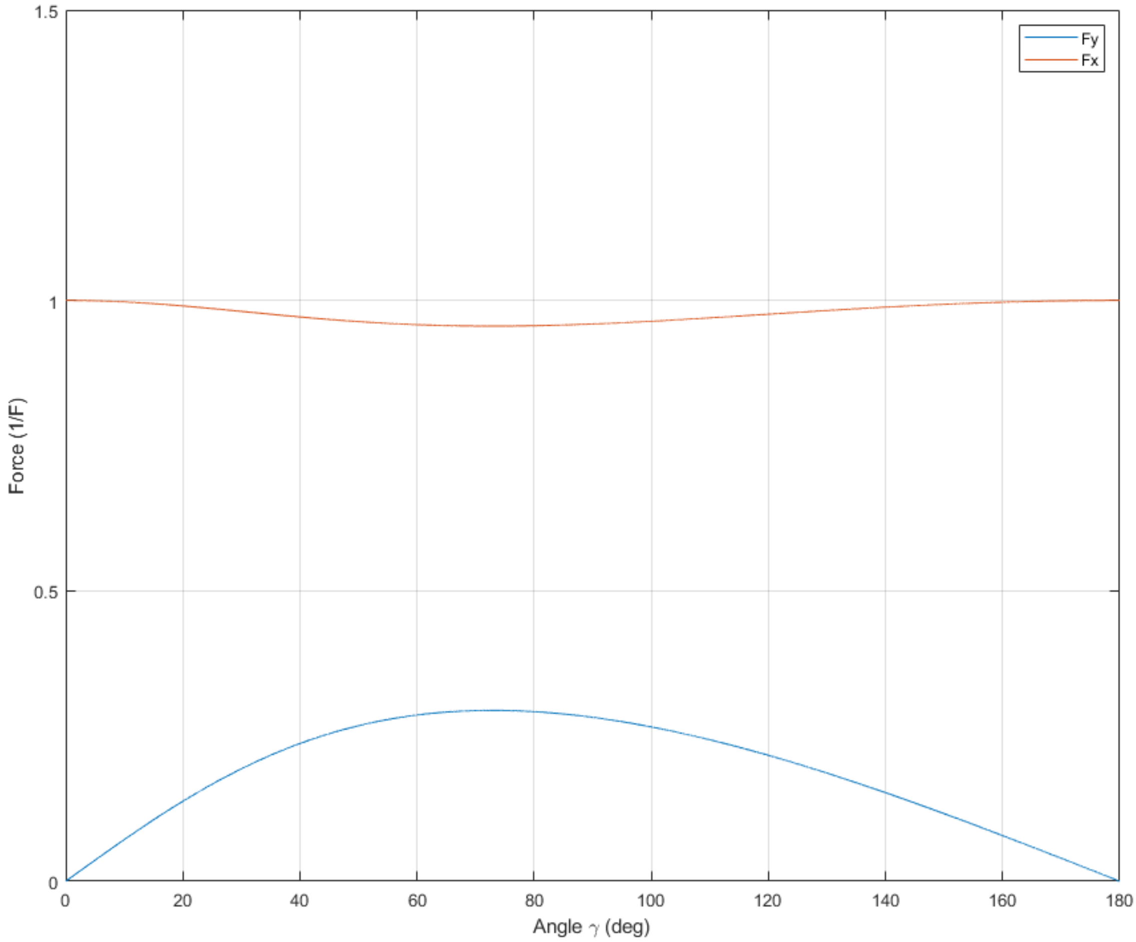

Figure 3.4. *Non-dimensional force components over rotation of the hooklifts frame divided by the total force.*

Now we can describe the force components to the FEM model for a parametric study of displacements with different frame angles and cylinder forces. The parametric study creates a response surface using curve fitting methods. The response surface is a ROM that can be imported to the hydraulic system model. Inputs for the ROM are the hydraulic cylinder force from the hydraulic system model and frame angle from the multibody model.

Response surface gives results from input values almost instantly as it is basically a multidimensional table and it also gives good results if the parametric study includes enough solved design points to define the responses which in this case are the displacements.

4 CREATING THE SIMULATION MODELS

The simulation program environment used in this master's thesis is Ansys Workbench. Ansys Workbench is a simulation environment where solvers for different physical problems can be used and coupled. Inside Ansys Workbench there is a Rigid Dynamics module than can be used to simulate rigid or flexible multibody systems.

ANSYS Twin Builder is a simulation software specialized in system simulation. These systems can be electrical, mechanical or hydraulic for example. They can also be combined. One example is to create a control system for the hydraulic system. Coupling of the multibody system to the hydraulic is created in this environment.

Coupling of the two different physical systems in this thesis is done by inserting the Rigid Body Dynamics model into the hydraulic system model. Hydraulic cylinders are coupled into the RBD model so that the hydraulic cylinders forces are input variables and hydraulic cylinders displacements are output variables.

4.1 Simulation workflows

First two workflows have empty hooklift but in the third workflow lifting of a body frame is done. These workflows are visualized in the figure 4.1.

Three different workflows for the simulations are used to provide results for different areas of interest. Simplest workflow is the rigid multibody system coupled with the hydraulic system. This mainly provides interesting results of the hydraulic systems response to the external force. Rigid multibody system provides joint force results which could be used as boundary condition for static analysis for a specific time step using a submodel. Using joint forces in static analysis was not studied in this master's thesis.

Second workflow has flexible multibody system and hydraulic system coupling. Only the middle frame of the hooklift is defined as flexible, rest of the model is rigid. Flexibility of the parts is created using the CMS-method. For the CMS-method it would be computationally inefficient to account for non-linear contacts that exist in the hooklifts frame. That is the main reason why in the second workflow additional analysis is needed to simulate deformations of the rear frame.

Joint forces from the flexible coupled simulation are imported to a FEM dynamic analysis using a submodel of the hooklift. Submodel is a part of the full model where the parts

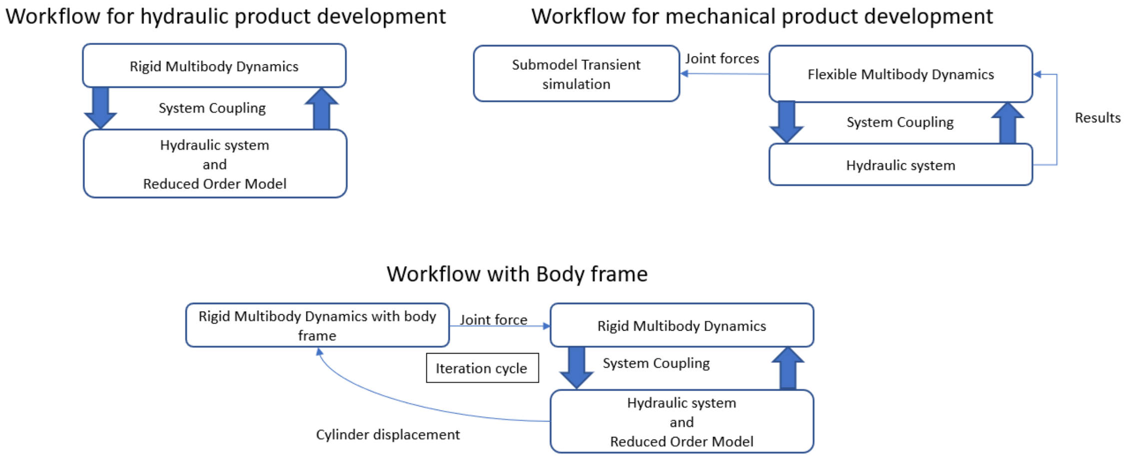

Figure 4.1. *Different simulation workflows.*

that are not included in the model are accounted with their inertia forces in the reaction force of the joints.

Last workflow focuses on the hooklift and a body frame. This is not critical for the displacements of the rear frame because the weight of the body frame actually works against the force created by regenerative cycle. Therefore no flexible analysis is done with the body frame included in the simulation. Hydraulic results are especially interesting because with when a heavy load is moved the pump might not be able to generate enough pressure so that the full speed of cylinders is reached.

4.2 Multibody model

The load device is grouped into subassemblies, where bodies in the subassembly do not move relative to each other. In these subassemblies rigid bodies center of gravity is fixed to each other. In flexible bodies geometrical features that touch each other share FEM nodes at the interface. This method is powerful because there is no need to manually define joints or bonded contacts between bodies that should not move relative to each other. Subassemblies are connected to other subassemblies by joints shown in figure 4.2.

Truck frame was neglected in the multibody simulations because for flexible multibody model only the middle frame is flexible so truck frame does not contribute anything to the model. Actually even the subframe and rear frame could have been excluded if needed for the same reason. In the simulation model subframe and rear frame are connected with fixed joint to simplify the connection. In real life that connection is made with tipping shaft which should be modelled as a revolute joint but there is also the rear frame lock which prevents the rear frame from rotating during unloading and therefore it is valid to use fixed joint. Rear frame and middle frame are connected with revolute joints to allow rotation of middle frame during the loading. All the hydraulic cylinders are modelled with

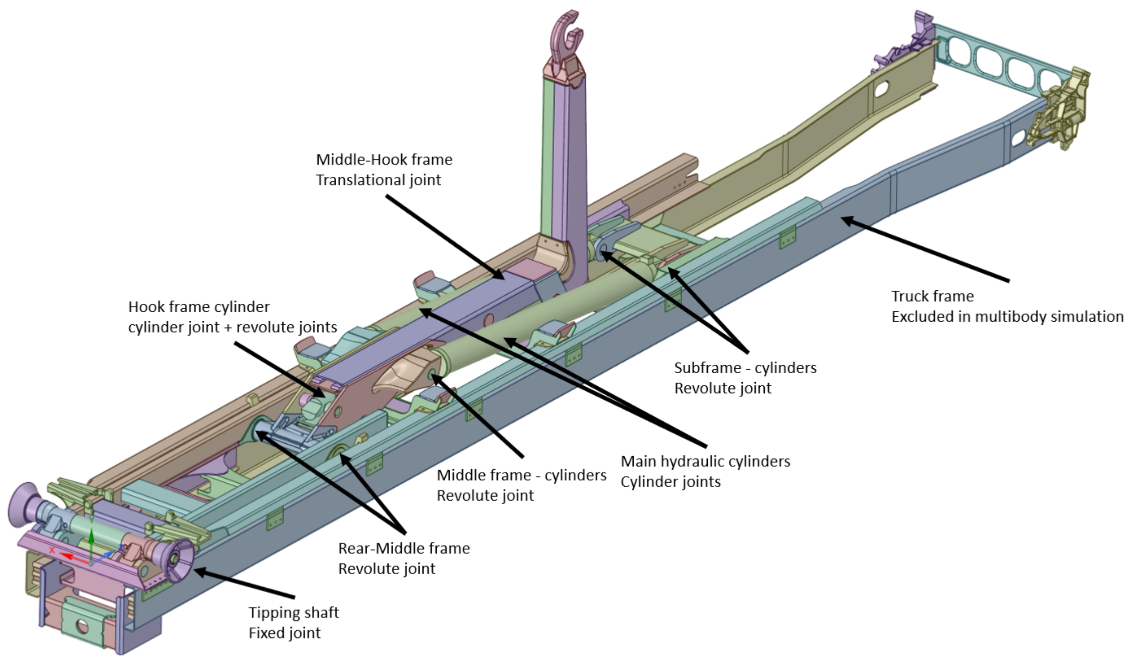

Figure 4.2. *Overview of joints for hooklift used in the multibody model.*

cylindrical joints. The cylinder that moves the hook frame is constrained from moving because it was not studied in this thesis. All the hydraulic cylinders are connected to the structure with revolute joints to allow rotation when cylinders rod is moving.

Care must be taken when defining joints for rigid multibody system. Over- and underconstraining the system can lead to different problems. It is important to have equal amount constrained DOF's from joints as six times the number of bodies minus one. If number of constrained DOF's is higher than the total number of DOF's of bodies the system is overconstrained. If there are more DOF's in the system than joints constrain then the system is underconstrained. Overconstrained multibody system can also give incorrect joint forces.

If there are flexible parts present the system needs extra care when defining joints connecting between rigid and flexible parts. For example defining revolute joint will only allow rotation around one axis on the interface that connects rigid and flexible parts. That would stiffen the flexible structure too much especially in this case where flexible part is connected to rigid frame in several positions. It is needed to allow joint to rotate so that CSM method for flexible body captures the deformation shapes of the real structure.

Finite element mesh in figure 4.3 for flexible multibody dynamics was created in ANSYS Mechanical. Selected meshing method was patch conforming tetrahedrons which first creates a surface mesh for the part and then fills the volume with tetrahedrons. Some parts could be meshed with hexahedrons which are more efficient but require sweepable geometry or large inner volumes of a part. They are more efficient that tetrahedrons because one element fills more volume and therefore total number of elements is smaller. Some simplification were done to the geometry, mainly deleting small details that are not important for the modal shapes.

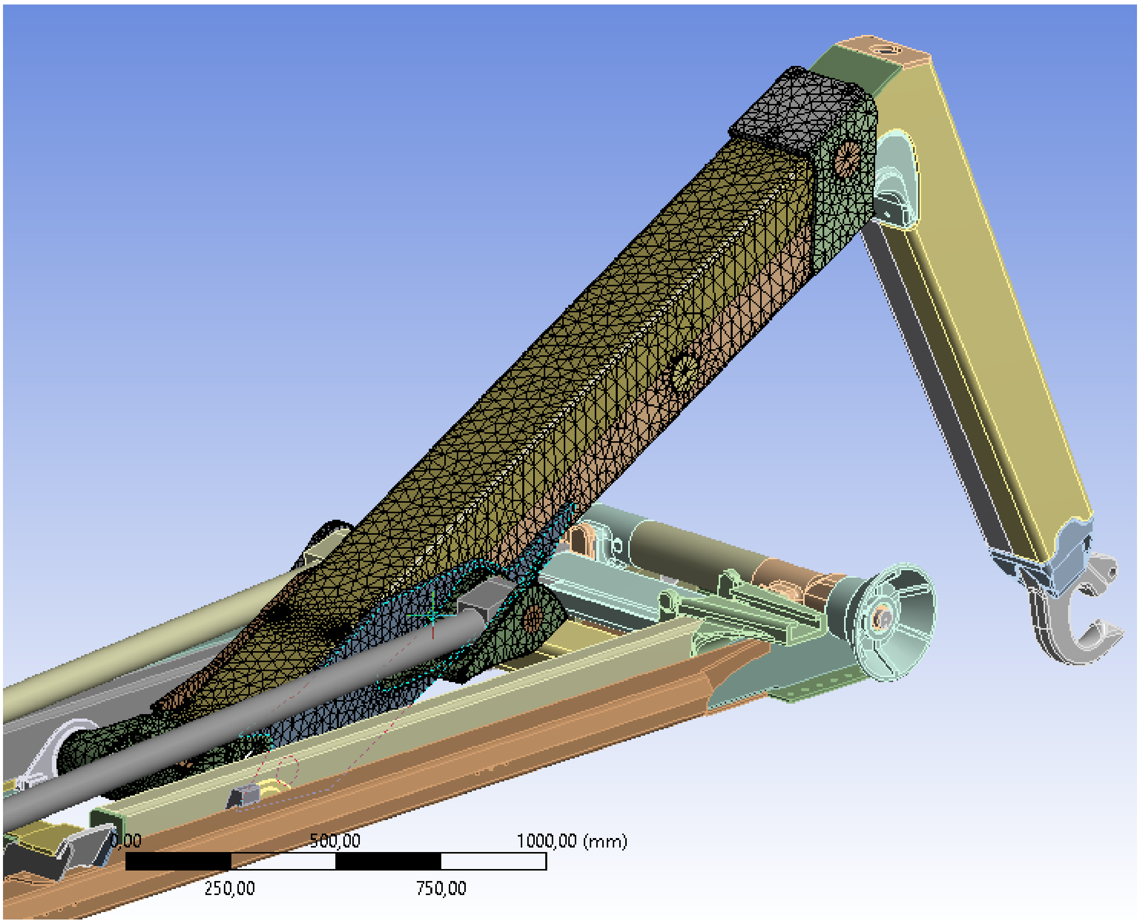

Figure 4.3. *Middle frame FEM mesh for CMS.*

4.3 Hydraulic model

In the simulation, the hydraulic system (figure 4.4) is driven by a constant fluid flow which is controlled by a directional valve and a pressure relief valve. The directional valve is controlled by step control signal which turns on or off in one time step. Also hydraulic cylinders have limit switches which prevent fast speed close to cylinder ends. In the simulation cylinders are moving towards the cap end which means that we are simulating situation where we would load a container onto the truck.

After the directional valve, load holding valve controls when the fluid can through depending on the forces explained in the theory section. Certain pressure level is needed for the LHV for it to open and let the fluid flow to the tank. After the LHV the fluid separates into the two different cylinders in normal loading or all of the fluid flows to cylinder A if fast loading is on.

Load device starts with normal loading until cylinder has moved 10 cm from fully extended position and limits switch controls directional valve to regenerative cycle. Fast loading will continue until cylinder is 10 cm from rod end and normal loading continues. Main direction valve will go into neutral position when cylinder has reach the end.

Most of the components were available in the build-in library of ANSYS Twin Builder. LHV was not present in the library so model for it was created using algebraic blocks and pressure sensors.

4.3.1 Load holding valve model

Load holding valve (LHV) is a component that is meant to ensure safe operating conditions so that the external load does not move the actuators. They are also known as counter-balance valves. Primary function for load-holding valves are load holding, control and safety. Load holding prevents cylinder from drifting, load control provides counter-balance when lowering a load and in case of failure in pipes LHV prevents sudden movement of the actuator.

LHV was not available as a library component so a functional model of it was created using pressure sensors, non linear orifaces and algebraic operators. A LHV consist of three different components, uni-directional valve, pressure relief valve and a pilot piston valve. Uni-directional valve allows fluid to flow freely in one direction and lock the load to a required position. Pressure relief valve holds the loads and limits maximum pressurization. Pilot piston opens the pressure relief valve.

In figure 4.5 forces acting on the LHV are presented. P_o is the pressure setting value, P_1 is the load pressure, P_2 is the counter pressure P_x is the pilot pressure and r_p is the pilot ratio. The opening pressure of the LHV can be then calculated by

$$P_x = \frac{P_o + P_2 - P_1}{r_p} \tag{4.1}$$

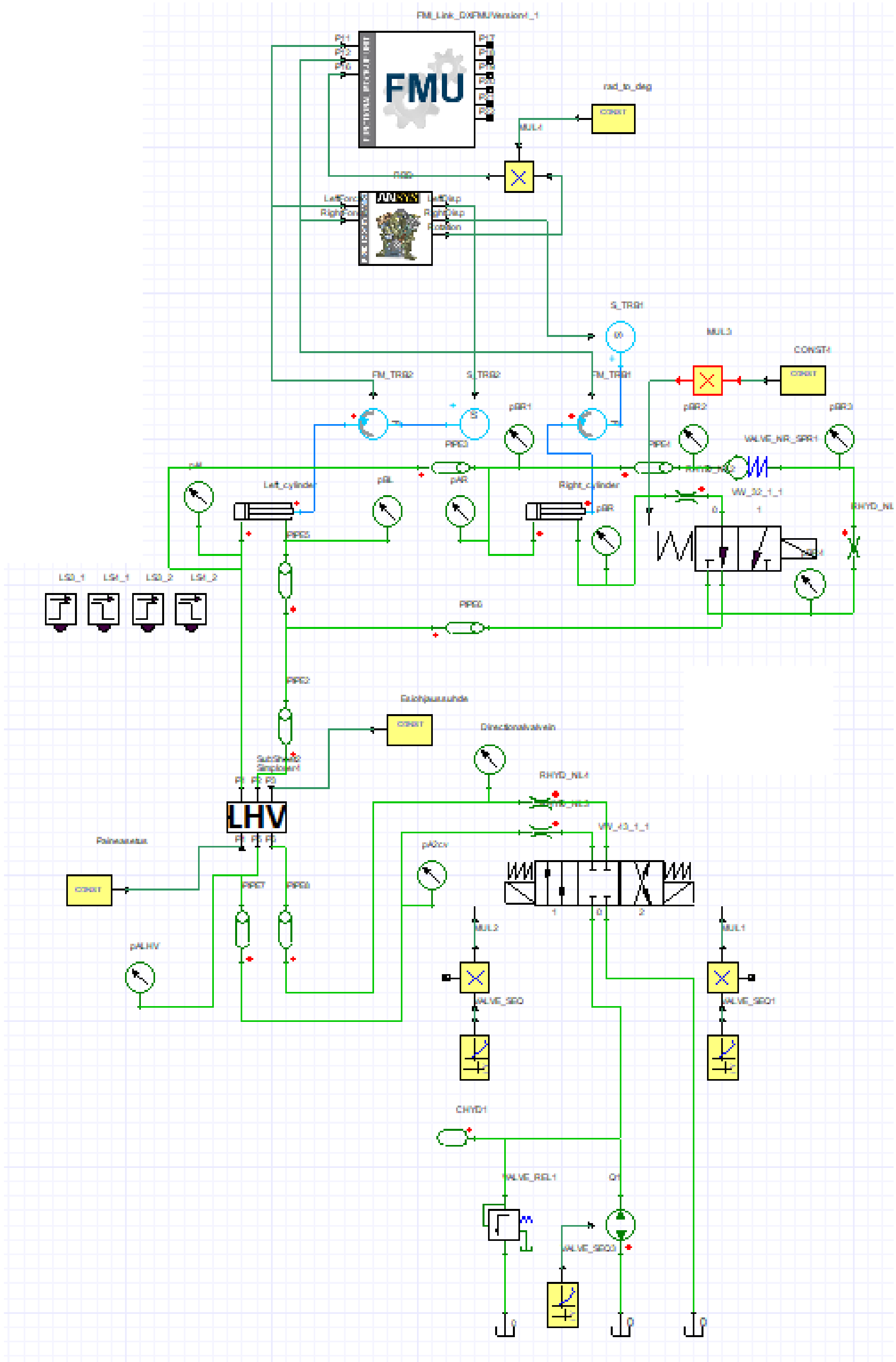

Figure 4.4. *Hydraulic system model in Twin Builder.*

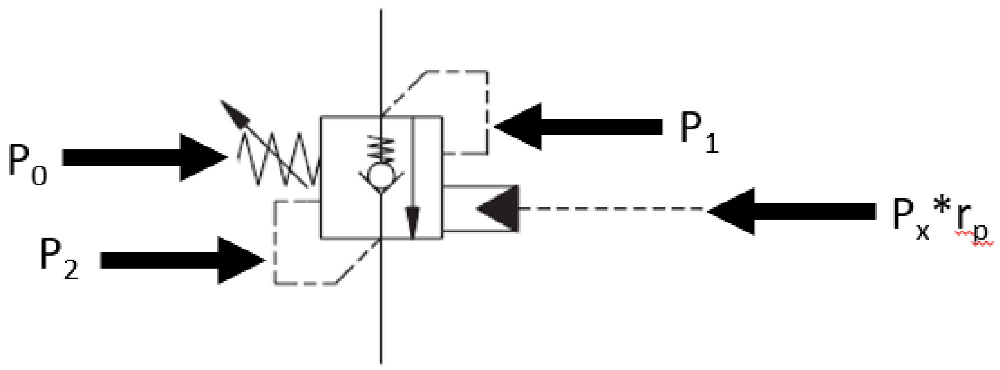

Figure 4.5. *Forces acting on LHV.*

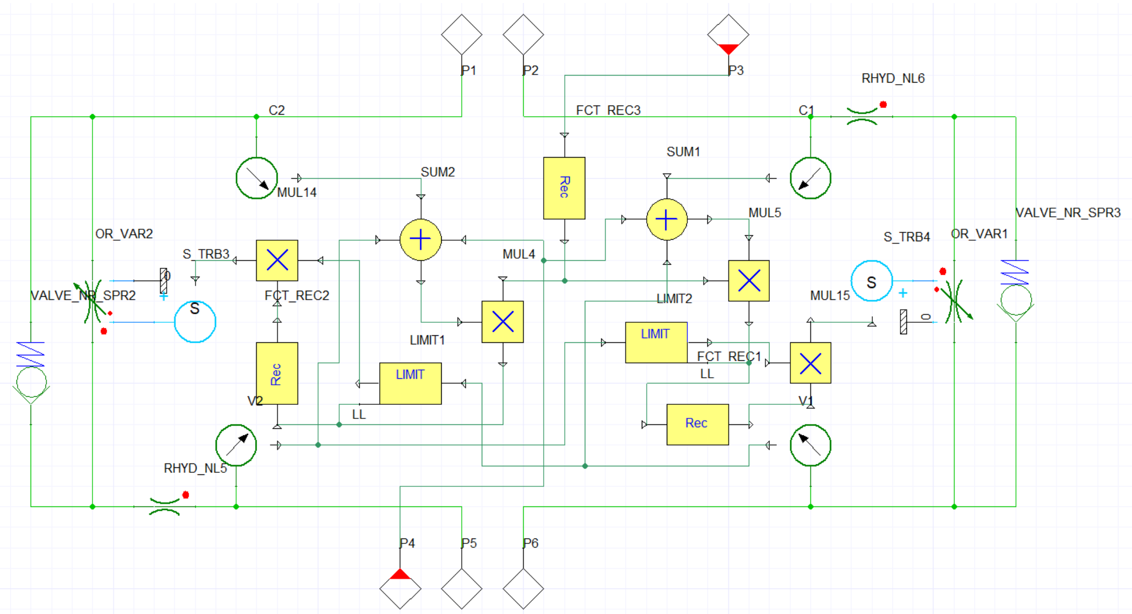

Figure 4.6. *Load holding valve model in Twin Builder.*

5 RESULTS

Results from the coupled simulations are presented in this chapter. First we present the results from the reduced order model, then the results regarding interesting hydraulic quantities and finally from the sub model.

5.1 Reduced order model results

Reduced order model of the hooklift was created using static structural analysis in Ansys Mechanical. Points of interest in the rear frame are shown in figure 5.1. Average displacements of the surfaces shown in the figure 5.2. Response surface inputs are the cylinder forces and the angle position of the middle frame. The response surface outputs average vertical displacements from the area that are shown with the red rectangles in figure 5.1.

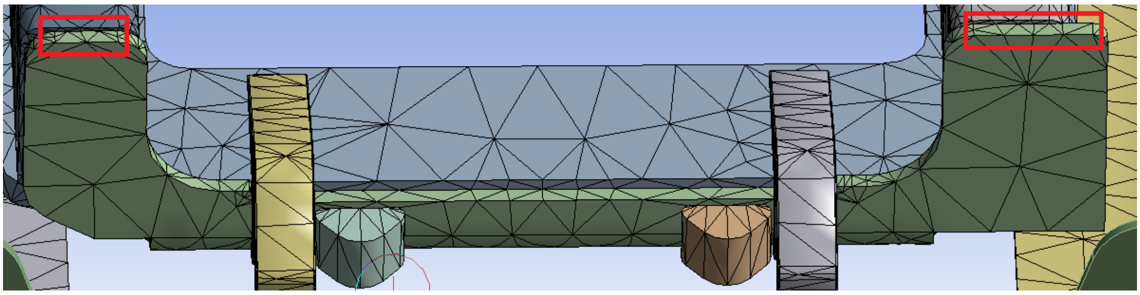

Figure 5.1. *Left and right side FEM nodes used in the response surface are emphasized with red box.*

5.1.1 Rear frame displacements

At the start of the simulation both sides have similar displacements but when the regenerative loading is on, right side moves upwards. Angle of the hooklifts frame affects how the load affects the displacement as the frame rotates over time, which can be seen in figure 5.2. When simulation time is near 14 seconds the regenerative loading is turned off. Then the displacements of right and left side are similar. After regenerative loading is turned on, there is a large negative displacement spike which is unphysical result. The error is a combination of the dynamic effects not accounted in creation of the ROM and the response surface fit. The response surface could have more calculated points for better a fit. Improvements could be done by calculating more DoE points near places where the regenerative cycle is turned on.

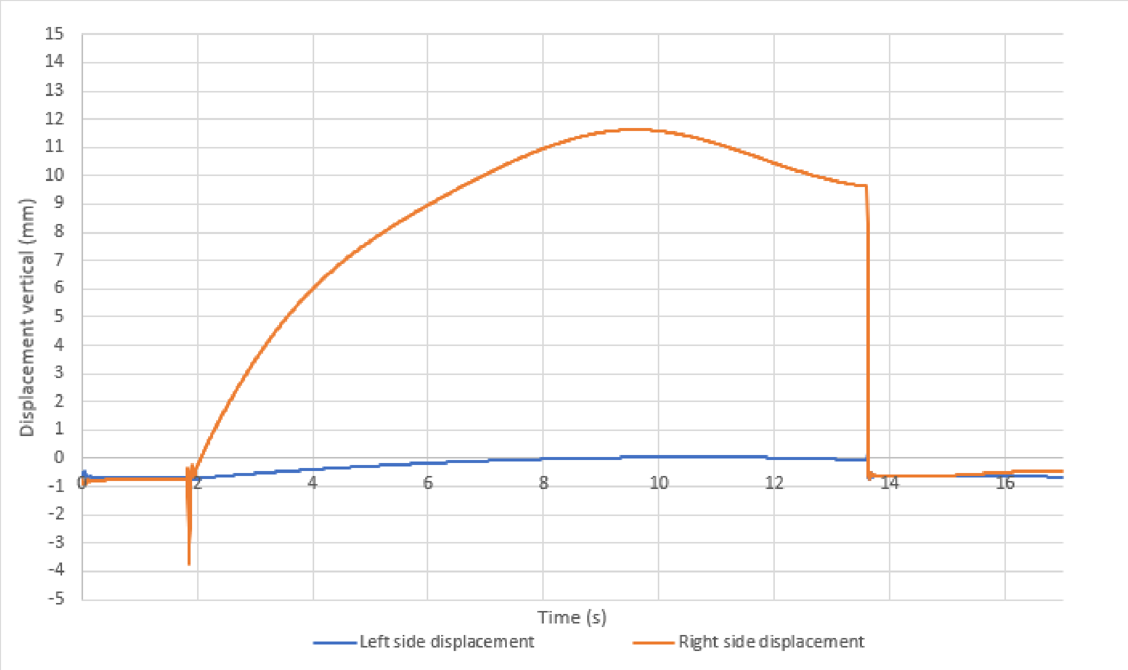

Figure 5.2. *Left and right side displacements of the rear frame using ROM.*

5.2 Hydraulic results

There are many quantities in hydraulic system but the most interesting ones are the pressures in different places and the cylinder forces. Other quantities include flow rates and velocities and the pressure losses.

All results shown have rigid and flexible simulations compared in the same graph. Also results are with empty device. As graphs need to show details they are attached in appendices for a better viewing.

5.2.1 Cylinder force results

Result graph for cylinder forces can be seen in Appendix A. Both rigid and flexible models give similar cylinder forces but there is a slight level difference and the transient effect from acceleration are different. Flexibility of the structure adds dynamic effects compared to a rigid structure.

In the beginning, where directional valve is opened and flow source starts to increase pressure of the fluid, we can see the effect in cylinder force when it quickly starts to increase and eventually stabilizes. Level difference between the left and right cylinder comes from pressure drop of pipes because fluid has to travel longer to reach the left cylinder. When time in the simulation has reached two seconds the regenerative loading is turned on and forces have different direction and the magnitude is much larger compared to normal loading. Sum of the cylinder forces is still similar as before the regenerative loading and only changes because the external force changes. When time in the simulation is near 14 seconds the cylinders are near the end and regenerative loading is turned off. Here the sum of the cylinder force magnitudes is positive which means that the cylinders are working against the external force because the weight of the structure is pressing cylinders inwards.

5.2.2 Pressure results

In appendix B pressure results of different positions in the hydraulic system are presented. Virtual pressure sensors are located in both ports of the directional valve and both ends of both cylinders. These locations give a good view on how does the pressure change over time. Pressure is generated by the pump where it has the highest value. As the fluid flows in the system, the pressure gets smaller for each component it passes.

At start of the simulation pressure starts to rise in all locations. Pressure level needs to rise to a certain level before load holding valve opens and allows fluid to flow. This is seen at the very start. Some differences can be seen in rigid and flexible model as sudden changes deform the structure and that couples back to the hydraulic system. During regenerative loading pressure rises in the pump because of faster flow speed in some parts of the system and because one cylinder works against the motion. In some locations pressure levels drop because fluid flows later to these parts and pressure losses have decreased the pressure.

5.2.3 Fluid flow results

Appendix C shows fluid flow rate in between different locations. Curve named LHV-split means the fluid flow rate between load holding valve and junction before the left cylinder. Curve named Cylinders means the fluid flow rate between ends of cylinders, right-fast between right cylinder and fast loading valve and so on. Fluid flow rates in the graph are absolute so fluid direction is not shown there.

There is only fluid flow from the end of the right cylinder to rod side of the right cylinder during regenerative loading (blue line). Also there is no fluid flow from junction between LHV and the left cylinder rod side to the fast loading valve during normal loading. Fluid flow rates are pretty stable as there are no large transient effects as seen in the pressure and force graphs. However there is a clear numerical error when fast loading is turned off and there is a large spike in the fluid flow rate. This is caused by the idealized valves that are either on or off but in real life they open more slowly and additionally the external load is pushing the cylinder to same direction as the pump.

5.2.4 Revolute joint force results

In appendix D joint forces from revolute joint located in the middle frame can be seen. These are the forces that are exported to transient FEM simulation to determine deformations of the rear frame. Here differences can be seen between rigid and flexible model. There is some level difference between the ROM and flexible model for each side. This is probably related to how the flexible components are constrained in the flexible multibody analysis. It is possible that not all required mode shapes have been captured by the CMS method or that the joints are incorrectly defined. Detailed analysis of the reasons why level difference exists is outside the scope of thesis.

5.2.5 Cylinder velocity results

Appendix E shows cylinder velocities for both rigid and flexible simulation. It can be seen that the cylinder velocity stabilizes quickly to the value defined by the fluid flow and usage of static ROM for deformations is valid during these periods. Interesting is the effect from turning regenerative loading on and off where cylinder velocities are separate to opposite direction in flexible model. In rigid model the cylinders are forced to move together and no separation is seen in figure 5.3.

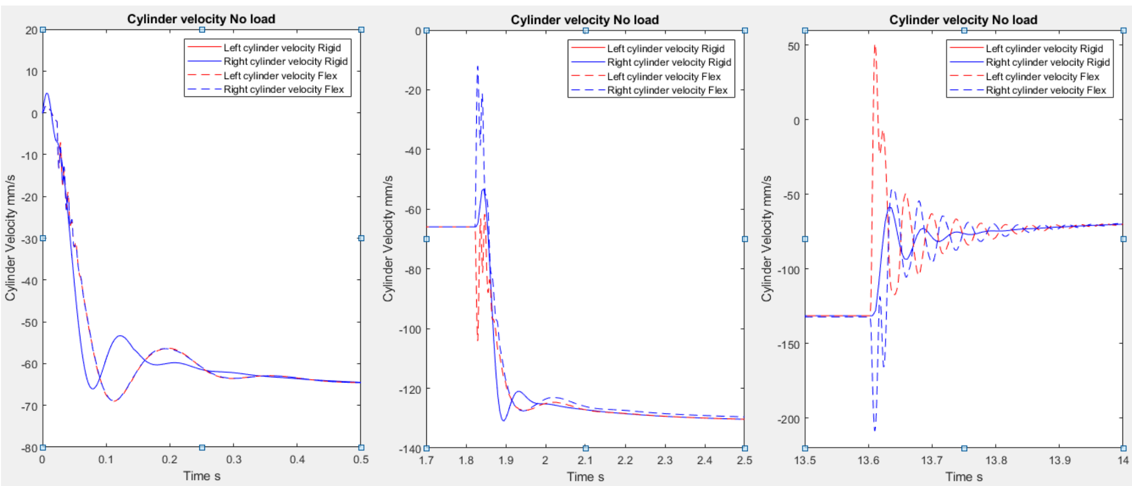

Figure 5.3. *Zoomed graphs of cylinders velocity at the simulation.*

5.3 Using simulation results in dynamic transient analysis

The hooklifts rear frame is a structure which supported from both ends. However in the one end there is a revolute joint connecting it to the subframe and the other end is supported by a stop. The stop restricts the frame from moving past it vertically but it can move up and sideways. This type of constraing needs to be modeled with a non-linear contact and the CMS method is valid only for linear structures. This means that it is not possible use it to determine deformations caused by hydraulic forces using the CMS method.

However we can use the CMS method for the middle frame that is connected to the rear frame with revolute joints. When middle frame is modeled flexible with CMS method, the torsion from regenerative cycle is transferred to the revolute joints. As a result joint forces from the revolute joints can be calculated correctly. Then we can create a substucture model of the load device and import the joint forces to the substructure as a boundary conditions.

Figure 5.4 shows rear frames end plates displacement as a function of time similarly as in figure 5.2 with ROM. There is a quite large difference with maximum displacement, approximately 3 millimeters. Large oscillations are visible after fast loading is turned off and it comes from imported joint forces that are rabidly varying and could be smoothed with larger damping.

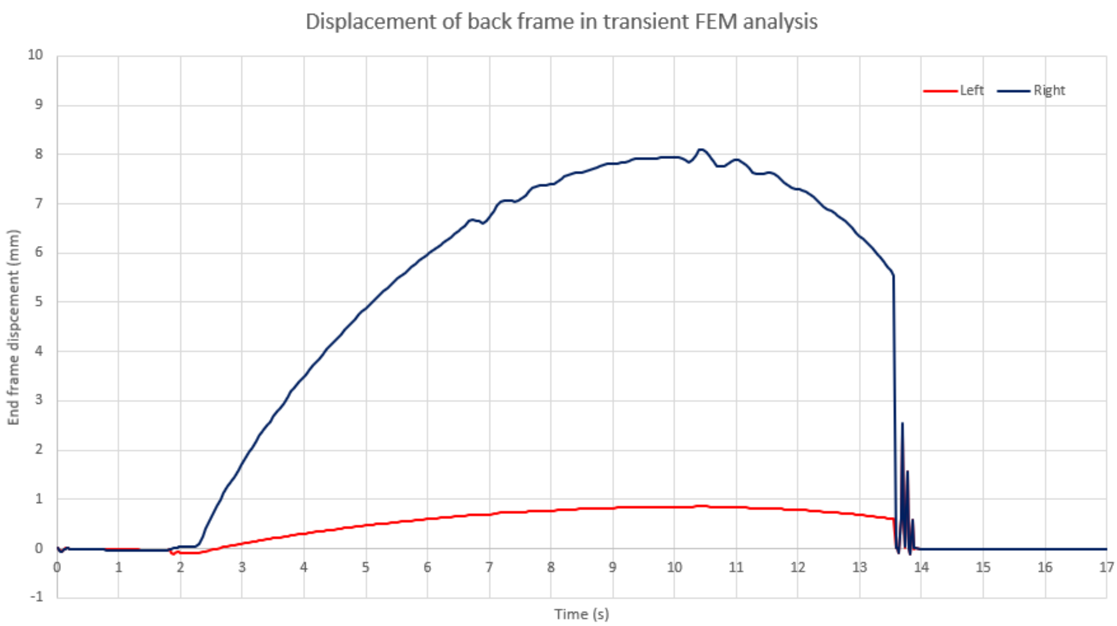

Figure 5.4. *Rear frames end plates deformations in nonlinear transient simulation.*

It can be seen that the cylinder forces of different direction twist the frame resulting right side of the rear frame to lift up while who end of left side is deformed towards the ground. Effect of the springs of the truck was neglected and the frame was rigidly constrained where the springs are located.

6 CONCLUSIONS

Workflow on coupling multibody dynamics with a hydraulic system is presented in this thesis. Multibody system can be rigid or flexible but some restrictions are present. Non-linear mechanics of the structure complicated the workflow for determining the deformations or stresses of the frame but hydraulic simulation can be done with relative ease.

According to test results made by Hiab, the simulation results for hydraulics are behaving correctly but especially the components of the hydraulic system models need calibrating for correct parameters. Biggest uncertainties are the pressure losses of some of the components. This can be done by comparing pressure from virtual sensors to test data from real sensors but that is outside the scope of this thesis.

Future work would include comparing results to test measurements to calibrate hydraulic component parameters. Requirements for this would be several pressure sensors in the hydraulic system that would help to calibrate pressure losses in pipes and other components. One possibility would be to characterize a single component in testbench and accurately determine the pressure loss as a function of flow velocity. This would lead into a component library which could be used in product development. Using different fluid flow speeds it would be possible to calibrate pressure losses as function of fluid flow velocity. In the case of empty device this will affect the transient but not the steady-state situation since the maximum pressure that pump can generate was not exceeded. In simulation where body frame is included with large a load, the maximum pressure can be reached and if the pressure losses are too large the cylinders do not move at correct velocity without calibration of hydraulic parameters.

Also changing the hydraulic system modeling language could be done. Open source modeling language Modelica is very efficient of solving systems of ordinary differential equations and its basic library includes basic hydraulic components and hydraulic fluids. The models used in this masters thesis were not easily customizable. Most of the problems were solver convergence issues during switching position of the valves. If the valves would open relatively slowly from closed to full opening over several time steps the convergence issues and numerical errors would decline. In the valve models used in the thesis, valve ports are either open or closed and switch of the state occurs in one step. Slower switching happens in real valves where all ports can be connected to each other when valve changes position. In Modelica it would have been simple to reduce the flow resistance when valve changes position for smoother transition for example.

Many different workflows was not the initial plan and having to resort to full transient analysis with FEM is computationally time consuming. Initial plan was to have rigid and flexible system with the same workflow and just turning flexibility of parts on or off depending on the interests of the simulation. Rigid multibody dynamics with hydraulics system simulates in couple minutes and the full workflow for flexible simulation takes about 10 hours including the system simulation and full transient FEM simulation. Even with the use of parallel processing of dividing FEM simulation to different physical cores of the CPU, the simulation still takes a lot of time.

Deformation results of the structural analysis can be used to determine how changes in the hydraulic system affect the displacement of the frame during fast loading. The displacement is the result of the different direction of the cylinder forces which comes from the pressure difference inside the hydraulic cylinder. If some change would make the pressure difference smaller then the moment from the cylinder forces is smaller and therefore the displacement in the rear frame is also smaller. Also studying optimal fluid velocity could be done. When the hydraulic system is analysed as a system instead of a single component it is possible to get meaningful insight of to operate it. For this it is necessary to have accurate hydraulic parameters.

Fatigue is also one very important area in steel structures also in the frame of a hooklift. Analysing stress ranges without the coupling between hydraulics and flexible multibody dynamics can be difficult. It can be seen in the simulation results that the system is highly damped and reaches steady-state quite fast but there are large force spikes during switching events of the valves which results to a varying stress which could be analysed with the Rainflow method and provide results of how damaging these stress variations are for the structure.

Further usage of this thesis would be so called digital twins. Physics based models used in product development could be used in predictive maintenance where the models are fed with sensor data from the physical device in the field. The different models could be FEM models that calculate stresses of critical position like welds and that time series stress data could be the used in fatigue calculations. These different models would be packaged as the digital twin that experiences the same loads as the physical twin in the field. The most important values like design life that is left on some of the key components would be shown in a operator dashboard inside the truck. Huge benefit would be that instead of sensor data telling that some component is not working properly or that it is broken, digital twin could predict when that component will most probably exceed its design life based on the loads it the device has experienced and maintenance could be done before damage has happened.

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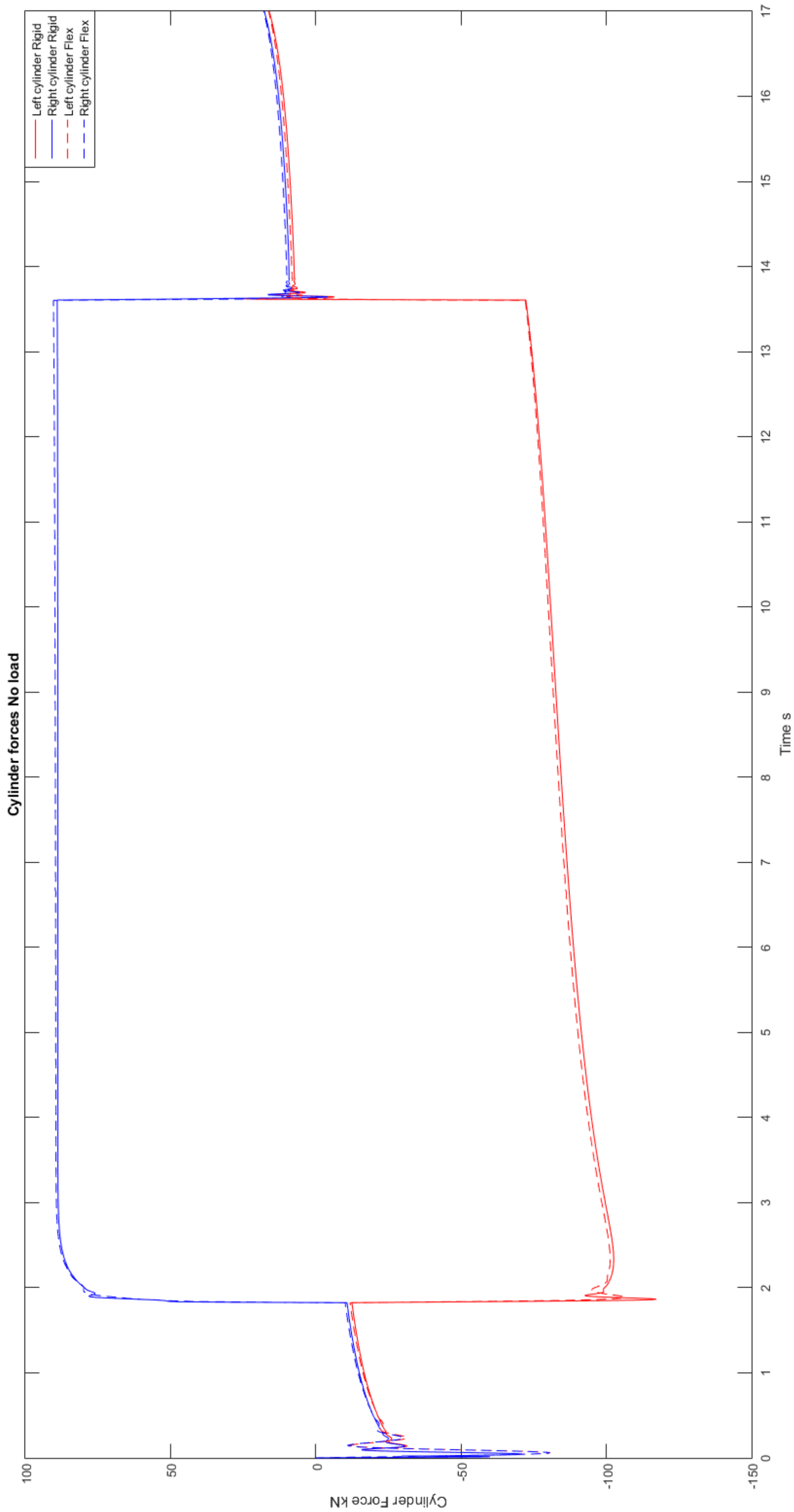
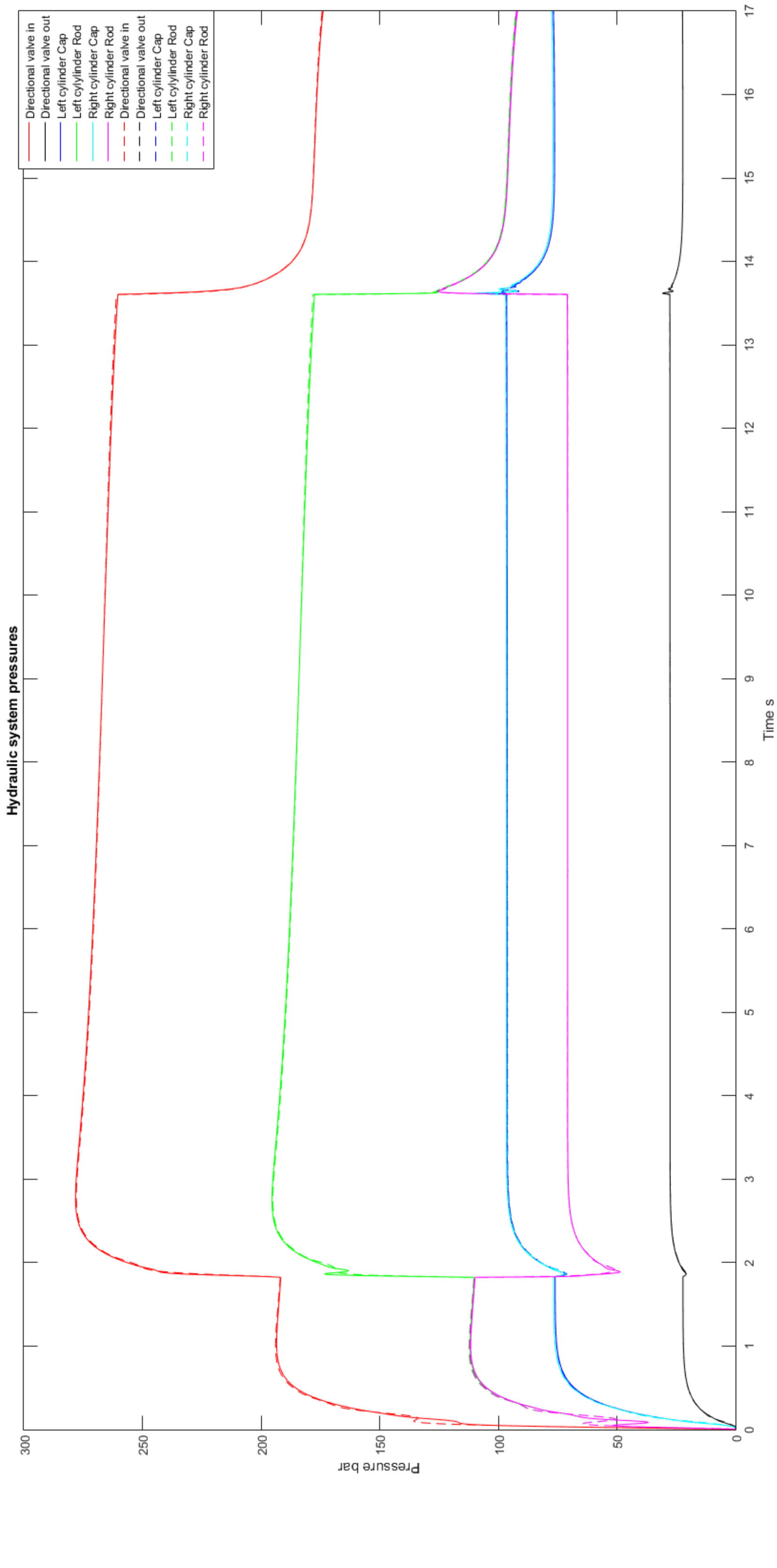
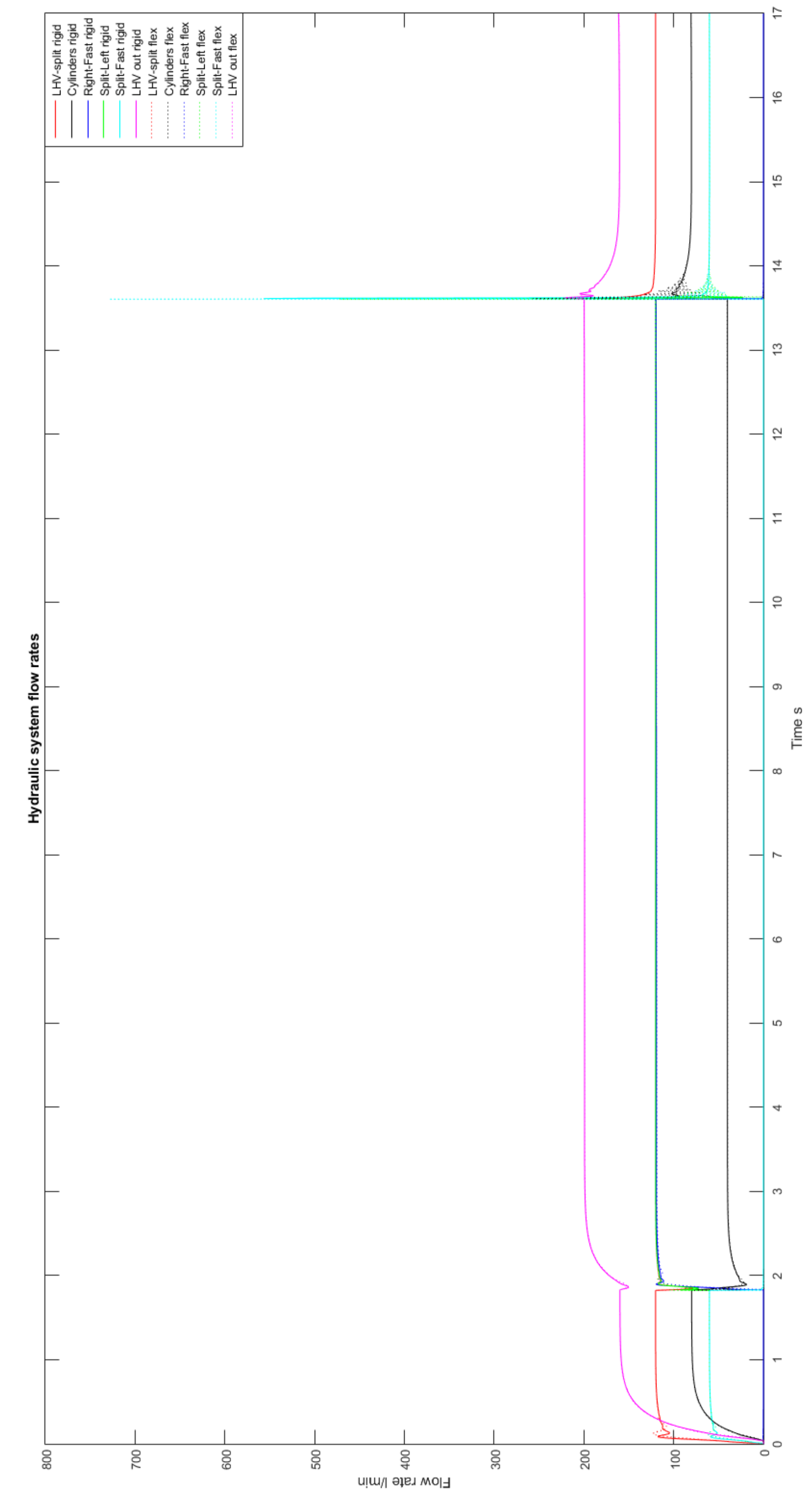
A. CYLINDER FORCE RESULTS



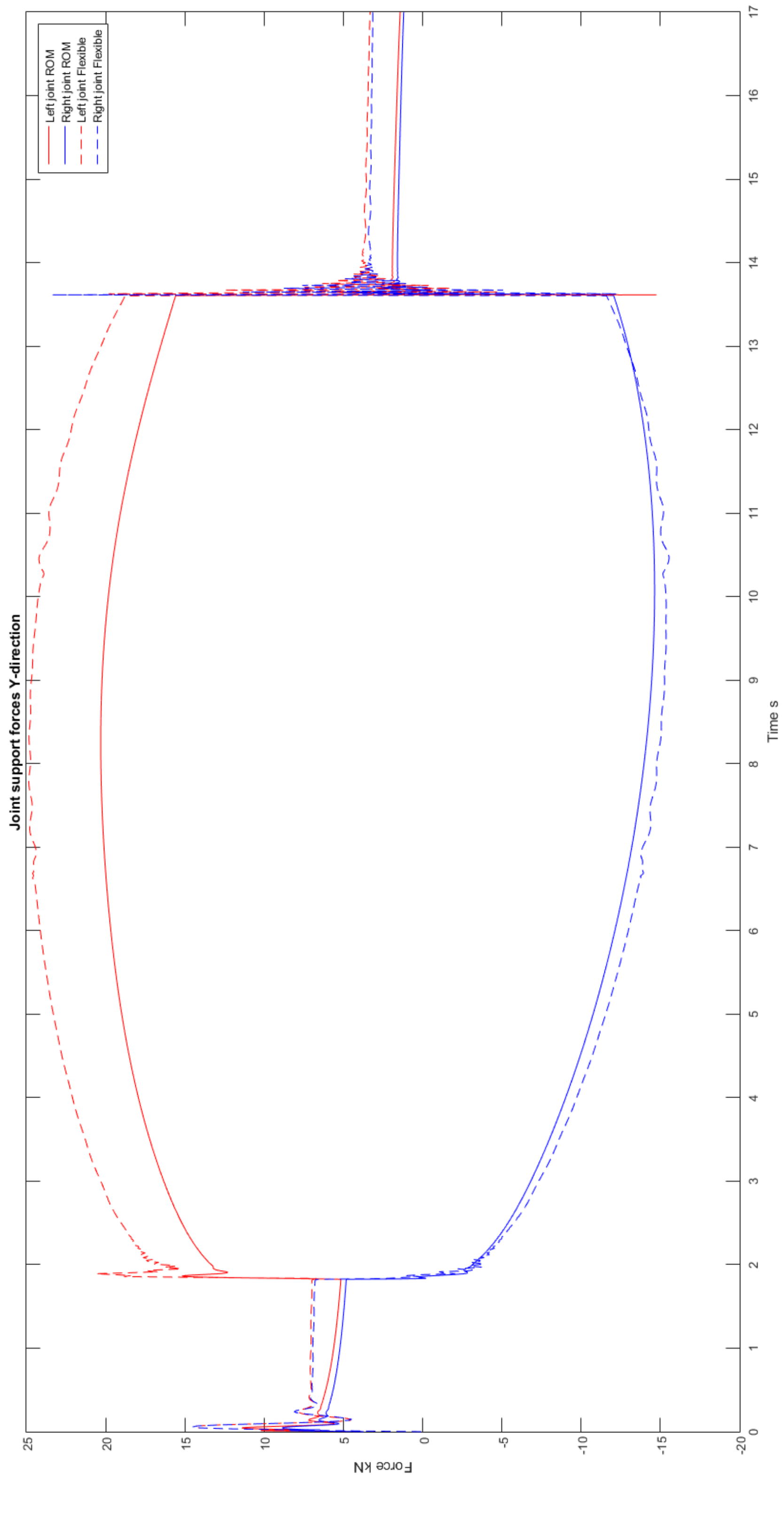
B. PRESSURE RESULTS



C. FLUID FLOW RESULTS



D. JOINT FORCE RESULTS



E. CYLINDER VELOCITY RESULTS