Universitetet i Stavanger

FACULTY OF SCIENCE AND TECHNOLOGY
Master Thesis

Study program/specialization: Master's in Petroleum Technology/ Natural Gas	**Spring 2018** Open/~~Confidential~~
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Title of master's thesis: Dynamic modelling of ram BOP	
Credits (ECTS): 30	
Keywords: Ram BOP Dynamic modelling Control system BOP Hydraulics Fluid mechanics	Number of pages: 63 + Supplemental material/other: 4 Stavanger, 15/06/2018

Acknowledgements

It has been a pleasure working with my supervisor at University of Stavanger, Homam Nikpey. I thank him for the support throughout the work of my Thesis. The regular meetings have helped me greatly in keeping a high moral through the thesis and have polished my skills as a professional.

I would like to express my profound gratitude to my external supervisor at MHWirth, Kristian Gjerstad. It has been a great learning curve for me sitting together with him in the same office and all the meaningful conversations we have had during lunch breaks. The master thesis would have been impossible to complete without his efforts and extensive knowledge of thermodynamics.

I would also like to acknowledge the team at MHWirth Stavanger especially Ahmad Mirhaj and Ehsan Peymani for providing continuous support throughout the thesis study.

Finally, I would like to thank my parents for providing me with continuous encouragement throughout the research process and writing of the thesis. This accomplishment would not have been possible without them.

Stavanger, June 2017

Muhammad Umar Nadeem

Abstract

As the closing time of Blowout Preventer rams is safety critical, this thesis proposes a dynamic model to calculate the closing time of a ram Blowout Preventer. The thesis work includes guidelines to recreate such a model that is applicable for most blowout preventer control system designs. The model can also be modified to recreate various scenarios and equivalent models for shear ram, blind and/or pipe rams. The main focus of the study is to generate a concept and establish ordinary differential equations. These differential equations define the changes in pressure and flowrate, as the position and velocity of the ram varies during the ram Blowout Preventer operations. The concept is then modelled to test and investigate the accuracy and reliability of the results.

The model is implemented in Matlab and constitutes of both hydraulic actuators and ram mechanics. The calculations in MATLAB are done using Euler's integration and second order derivatives. The model assumes a specific pressure at the master regulator valve supplied by the accumulator bank and simulations are performed to test if the pressure support is enough to complete the actuation process. The calculations account for the friction losses through the control system, hydraulic actuators and the interface between shear ram and the annulus. Two different closing operations in a Blowout Preventer are put into focus i.e. when there is a drill pipe present in the well bore (scenario 1) and there is no drill pipe where the ram moves freely (scenario 2).

The dimensions of the equipment used for the present study are taken from various published studies. The model is designed in conjunction with API16D and NORSOK standards. The model in the present study is developed in accordance to the boundary conditions of scenario 1 and the results obtained show a good agreement with the published studies. However, the present model exhibits few discrepancies for scenario 2 when compared with the published results. This also concludes that the model established for a scenario is not applicable for other scenarios.

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1 Introduction

The thesis work focuses mainly on development of a concept to investigate the closing time of a ram BOP. The physical model of the closing operation of a ram BOP depends on the pressure support from the accumulator which drives the fluid through the hydraulic circuit into the hydraulic cylinder. This complete operation has been combined with the interface between the ram BOP and the wellbore to investigate closing time of the BOP. A set of ordinary differential equations are proposed which define the pressure requirement for adequate performance of a ram BOP.

A dynamic model using MATLAB has been coded and tested in this thesis project to calculate the closing time of the ram BOP. The modelling approach is based mainly on fluid dynamics, hydraulic mechanics and dynamic modelling. The fundamental building block of the model is conservation of momentum and conservation of mass based on liquid flow in pipes. The principles have been modified to meet the requirements of the investigation procedure in case of a ram BOP. The modifications include addition of pressure losses in the hydraulic circuit as well as the mechanical losses in the actuation process. These additional barriers affect the closing time of the ram BOP and thus need to be investigated to examine the effective closing time. Several literature studies have been used to acquire several parameters and guidelines. The model includes general assumptions for simplification of the model.

Ordinary differential equations have been established to correlate the pressure changes with time and speed of the ram motion. Additional friction and pressure losses have been added in the power circuit as well as the hydraulic actuator to compensate for any reduction in response and closing time of the ram. The model designed in MATLAB calculates the closing time of the BOP rams, the velocity of the ram, the flowrates in different components and the pressure drop across assigned positions. This provides a clear output data in the form of graphs which are easy to interpret.

The thesis work consists of a literature study on BOP modes of operation and the theory surrounding the criteria used in the model. Modelling guidelines have been provided in later sections for recreation of the model. The results from the simulations are also included and later discussed. The conclusion section consists of the findings in the project work and arguments around accuracy of the results.

1.1 State of the Art

The BOP design is dependent upon several regulations and standards which affect the operational capabilities of the system. The design must not only comply with the standards and regulations (API, July 2016; NORSOK, Dec 2012) but also perform appropriately under working boundary conditions. The working boundary conditions consist mainly of the closing time of the BOP and enough hydraulic pressure available for complete shutdown of the well. The ram BOP is put under investigation using dynamic modelling, to confirm the closing time and the pressure requirements.

The common practices in the industry involves calculations around volumetric capacities of the accumulators and obtaining the response time of the ram using discharged volume calculations. The results are then fine-tuned to compensate for any miscalculations or numerical errors. The final stage of BOP design involves factory acceptance test (FAT). The response time of the BOP is verified using FAT test first at the manufacturing plant and later at the installation site by site acceptance test (SAT). This however, as described in the study (W. E. Services, 2004) doesn't guarantee that the ram BOP performs adequately.

The model in this paper enables to mitigate the risk around the underperformance of a ram BOP before installation process. The risk can be reduced by creating various scenarios in a simulator and assessing the performance of the equipment. Verification of adequate closing time can be done beforehand to avoid any time delays. The BOP control system modelling also enhances the cost effectiveness and improves the safety standards during the drilling operations.

The model is designed for shear ram in a ram BOP but slight modifications to the dynamic model are required for the same model to be used for pipe rams, blind rams or variable bore rams. This process can also be helpful during design selection as well as comparison of different designs. The model can be beneficial when designing simulators for training purposes.

Earlier developments have made it necessary to impose stricter regulations around the design of the BOP after the Deepwater horizon oil spill in the Gulf of Mexico (Baugh, Vozniak, & Schmidt, 2011). Due to these recent developments, the oil and gas industry has become more willing to invest in expensive simulators in order to minimize the risk around BOP design.

1.2 Problem Statement

The closing time of the ram BOP is safety critical in the oil and gas industry. It is therefore beneficial to find simulation models for BOP design, before installations at the operation site. This thesis is designed to determine:

- Is it possible to define the closing operation of ram BOP using simple equations?
- How much time is required to close ram BOP for scenario 1?
- Is it possible to use the same model for scenario 1 in scenario 2?
- Is it possible to use the model for scenario 1 in other scenarios?
- What friction and pressure forces affect the closing time of the ram BOP?
- Is the closing time of ram BOP according to API and NORSOK standards?

1.3 Objectives

The objectives of the thesis project are as follows

- Literature study on BOP modes of operation
- Effects of mechanical and pressure forces on BOP ram motion
- Hydraulic actuators and their interaction with the rams
- Friction losses in hydraulic circuit
- Frictions losses in hydraulic actuators and ram-annulus interface
- A representation of a simplified hydraulic power circuit
- Establish ordinary differential equation for hydraulic cylinder
- Establish ordinary differential equation for flow in pipe (hydraulic circuit)
- Construct a dynamic model in Matlab for simulation purposes

Extensive efforts have been done to complete all the objectives listed above with high accuracy. The results from the model and the program model can be recreated to further study the BOP ram operations.

2 Literature review

This chapter is dedicated to the literature study of blowout preventer and the control system.

2.1 Technical Background

The blowout preventer (BOP) is a main well control device used in the oil industry during exploration drilling and intervention operations. It acts as a main barrier in event of an unwanted flow of hydrocarbons into the bore well, usually known as kick. The kick can cause several sophistications in the drilling operations. Under normal operations, the primary well control is hydrostatic mud pressure, however in case of a kick, the formation pressure exceeds the hydrostatic mud pressure resulting in hydrocarbon flow into the annulus. The BOP is then used as last resort to close and seal the well in event of uncontrolled flow. The BOP designs can be divided into mainly surface BOP and subsea BOP.

The BOP consists of several valve lines in order to add or withdraw fluid volumes from the system. The kill-line in BOP is used to pump fluids into the annulus and the choke-line is used to bleed off pressure. The BOP is designed to carry out the following measures (Davorian, 2013):

- Closing the top of the hole
- Allow the release of the fluids
- Allow continued pumping into the hole
- Controlled volumes to be withdrawn from the system
- Movement of the inner string of the pipe

The BOPs can be arranged in different configurations and are known as BOP stacks. The configurations are specific to each operation and are based mainly on working pressures in the well. Configuration codes can be assigned to a BOP stack with pressure ratings, bore size and different BOP components like spools, rams and annular preventers. The BOP stack can either be positioned on the X-mas tree or directly mounted on the well head. A simple BOP stack configuration is illustrated in figure 2-1 as follows:

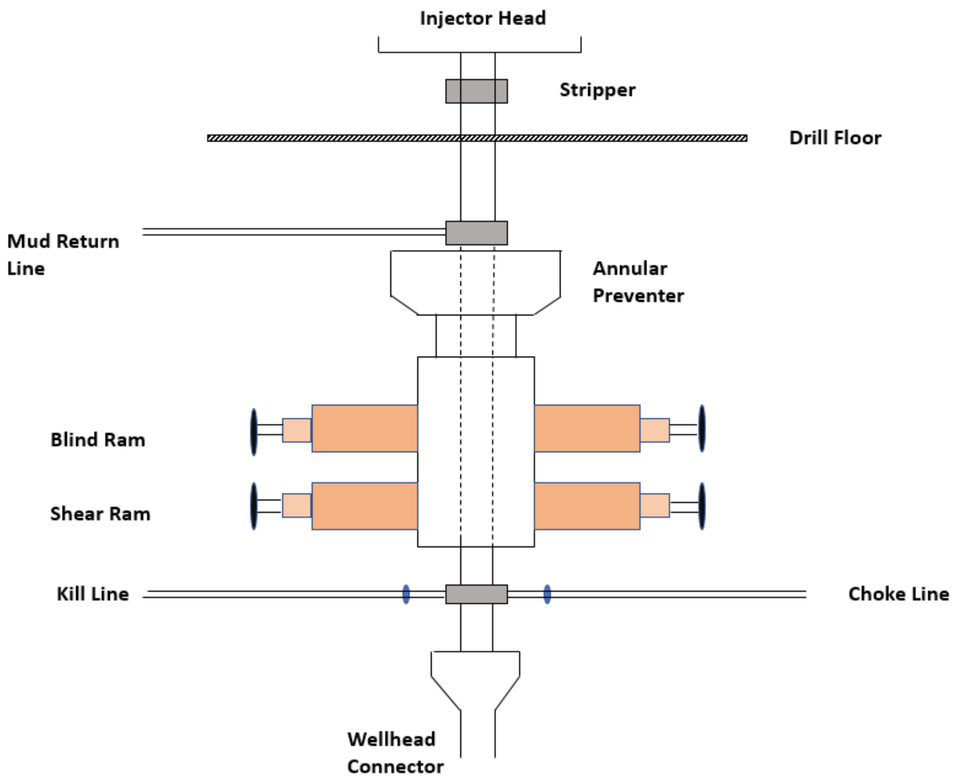

Figure 2-1: Blowout Preventer Configuration

There are several BOP designs available but can mainly be divided into two categories; annular preventer BOP and ram BOP.

The annular preventer BOP uses rubber to close around the drill pipe and secure the annulus around the drill pipe. The sealing around the bore is done using large polymers rings that are reinforced into steel. The rubber seals are mechanically squeezed inward either to seal around pipe or open hole.(Schlumberger, 2018a). The hydraulic cylinders provide a lift force upwards which pushes the rubber gradually into the aperture of wellbore until complete closure. The annular preventer has an advantage for sealing on a variety of pipe sizes. The annular BOP has a far greater reliability when sealing around the tubular than an open hole. Most BOP stacks have at least one annular preventer BOP placed on top of the BOP stack. In the event of uncontrolled flow, the annular BOP is the first step in blowout prevention.

The ram BOP consists of two adjacent rams placed around the borehole and large hydraulic cylinders force the rams to meet each other in the middle to seal the wellbore. Ram BOP are used to quickly seal the top of the well in event of a kick (Schlumberger, 2018b). There are several types of ram BOPs used in the industry today i.e. blind rams, pipe rams, variable bore rams and shear rams.

- Blind rams have a flat opening in the mating surfaces and are used to seal a wellbore in the absence of a DP.
- Pipe rams have a semi-circular opening in the mating surfaces corresponding to the size of the DP and are used to secure the wellbore around the DP. The ram design should be specific to the drill pipe used during drilling operations.
- Variable bore rams have an adjustable semi-circular opening in the mating surfaces and are used to secure the wellbore around the DP. This ram type can seal around a wide range of drill pipes.
- Shear rams have an edged-cutting surface in the mating surfaces and are used to shear the drill pipe and seal the wellbore. Shear rams are used as a last resort to regain pressure control of the well as it limits the future options while drilling.

The ram BOP acts as the most important barrier before a blowout and the closing timing is of critical importance which is investigated in the thesis project. The main focus of the investigation is on the control system and actuation mechanics. Choke-line and kill-line are not in scope of this thesis.

2.2 Control Systems

BOP control systems provide the hydraulic pressure support required during operations like annular preventer activation or ram preventer actuations. The main components of the control systems include: accumulator unit, auxiliary control unit, control lines, master control panel and control valves. The offshore drilling operations use remotely operated vehicles as a support to the control system. The hydraulic circuit for a BOP as shown in figure 2-2:

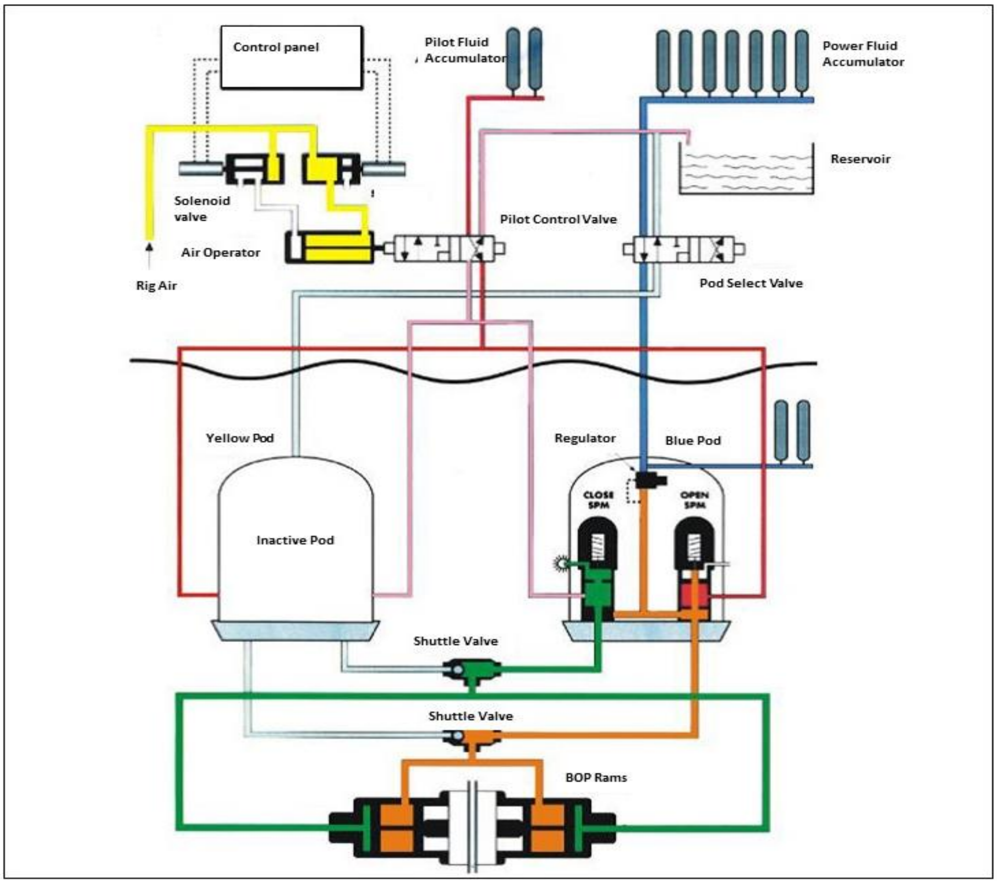

Figure 2-2: Hydraulic Circuit BOP

The primary pressure source in the hydraulic circuit are the nitrogen filled accumulators which regulate pressure in the circuit. The nitrogen gas expansion forces the hydraulic fluid toward the hydraulic actuator. This additional pressure moves the piston in the hydraulic cylinder towards the closing position.

Once a close command is given by the operator, the associated solenoid valve opens to allow pressured air to pass through. The pressurized air actuates pilot valve into close position. This opens the circuit for hydraulic fluid from the accumulators to flow into the control pods. Pressure is regulated using a master regulator valve before fluid entering the control pod. There are two separate control pods, used for redundancy namely as the Blue pod and the Yellow pod. During normal operations, only one pod is active. The control pods contain two separate SPM valves which are connected to open and close-lines of the hydraulic cylinder. SPM valve is a three-way open valve which directs the hydraulic fluid to the hydraulic cylinders. The opening or closing command from the operator decide which way the hydraulic fluid travels form the SPM valve. The hydraulic fluid travels from the SPM valves onto the shuttle valves. Shuttle valve is used to direct the flow of the fluid either to close or open the ram BOP (W.C.Goins, 1983). A detailed explanation of the power circuit can be seen in figure 2-2.

2.3 Standards

The BOP must meet certain requirements set by the authorities before it can be put into operation. The BOP must also pass through certain manufacturer tests before it is issued out. The API and NORSOK regulations listed determine the standards and regulations for the ram BOP. The scope of the thesis consists of only ram BOP, therefore specifications around the ram BOP in API and NORSOK standards are put into focus.

The API standards define the response time of ram BOP to be 30 seconds. This includes the response time from the accumulators to complete seal off for each ram(API, 2012) (API, July 2016). API regulation for the annular preventer BOP states that the closing time to be within 30 seconds and for bore size of BOP greater than 18.75 inches to be within 45 seconds.(API, July 2016)

The NORSOK standards define the closing time of the Ram BOP and annular preventers. The closing time of the ram BOP from the actuation process to the complete seal off the well bore is 30 seconds. For annular preventers BOP, the response time is 45 seconds for preventers with bore size exceeding 20inches.(NORSOK, Dec 2012)

The BOP also must pass the FAT test from the manufacturing unit to be passed out to the operations.

The main focus in this study will be the response time of the ram BOP from the actuation process to the complete seal off the well. There is no definite procedure prescribed by the manufacturers for calculating closing time of the BOP. The standards and regulations mentioned above are taken into consideration while designing the proposed model.

3 Theory

This chapter includes all the relevant theory required to model a BOP control system and hydraulic actuator.

3.1 Fluid Mechanics

The approach taken in this thesis around modelling is the transient dynamic analysis. This technique is used to determine the dynamic response of a structure under the action of any general time dependent loads. The method uses system matrices to calculate the transient responses of the system. This method allows to include all types of nonlinearities and all outputs are calculated in a single pass. The process consists of performing a discretization in space from which a system of ordinary differential equation in time is obtained and then discretized in time.

The mechanics of a hydraulic system in the modelling section are based upon conservation of momentum and Newton's second law of motion. The calculations in the control system are based upon the conservation of mass and the continuity equation.

3.1.1 Conservation of momentum

The continuity equation can be applied for volume calculations in a hydraulic cylinder. The continuity equation along with the Newton's second law are used to derive the following equations (Basniev et al., 2012; Hager, 2010; Munson, 2009). In a closed vent (hydraulic circuit where the velocity and density of the liquid is uniform in space (but varies in time), the momentum balance can be expressed so that,

$$\frac{dv_{Li}}{dt} = \frac{1}{m_{Li}} \sum F \tag{1}$$

Which gives,

$$\frac{dQ_{Li}}{dt} = \frac{A_{CsCv}}{m_{Li}} \left[F_p + F_f + F_g\right] \tag{2}$$

Or,

$$\frac{dQ_{Li}}{dt} = \frac{A_{CsCv}}{\rho_{Li} V_{Li}} \left[A_{CsCv} \Delta P_{Li} + A_{SuCv} \tau_w - \rho_{Li} V_{Li}\, g \cos\theta\right] \tag{3}$$

Where A_{CsCv} and A_{SuCv} are the areas of the cross-section and surface of the CV respectively. τ_w is the shear stress between the liquid and the wall of the duct, θ is the inclination of the duct from vertical, and ΔP_{Li} is the difference between the pressure in current CV and the pressure in the downstream CV (i.e., $P_{Li(i)} - P_{Li(i-1)}$). It must be noted that the fluid is considered to be of a non-Newtonian nature i.e. steady and incompressible.

3.1.2 Conservation of mass

The flow in a closed vent (hydraulic circuit) is addressed here where the conservation law for one-dimensional flow(Basniev et al., 2012; Hager, 2010; Munson, 2009) through a duct is expressed so that:

$$\frac{dm_{Li}}{dt} = q_{LiIn} - q_{LiOut} \tag{4}$$

$$\frac{d(\rho_{Li}V_{Li})}{dt} = \rho_{LiIn}\, Q_{LiIn} - \rho_{LiOut}\, Q_{LiOut} \tag{5}$$

Expanding the derivative and reorganizing gives:

$$\frac{d\rho_{Li}}{dt} = \frac{1}{V_{Li}}\left(\rho_{LiIn}\, Q_{LiIn} - \rho_{LiOut}\, Q_{LiOut} - \rho_{Li}\frac{dV_{Li}}{dt}\right) \tag{6}$$

By assuming isothermal conditions, and applying a linearized equation of state for the liquid, we obtain the simplified relationship between density and pressure:

$$\rho_{Li} = \rho_0 + \frac{\rho_0}{\beta}(P_{Li} - P_0) \tag{7}$$

Derivation of Eq. 7 gives,

$$\frac{dP_{Li}}{dt} = \frac{\beta}{\rho_0}\frac{d\rho_{Li}}{dt}, \tag{8}$$

By combining Eq. 6 and Eq. 8 the conservation of mass can be expressed with pressure as state variable instead of density:

$$\frac{dP_{Li}}{dt} = \frac{\beta}{\rho_0 V_{Li}}\left(\rho_{LiIn}\, Q_{LiIn} - \rho_{LiOut}\, Q_{LiOut} - \rho_{Li}\frac{dV_{Li}}{dt}\right) \tag{9}$$

Properties of the flow out of the closed vent are given by the averaged properties within the closed vent, i.e., we have $\rho_{Li} = \rho_{LiOut}$ and $Q_{Li} = Q_{LiOut}$. This gives our basis equation for liquid continuity,

$$\frac{dP_{Li}}{dt} = \frac{\beta}{\rho_0 V_{Li}}\left(\rho_{LiIn}\, Q_{LiIn} - \rho_{Li}\, Q_{Li} - \rho_{Li}\frac{dV_{Li}}{dt}\right) \tag{10}$$

Or

$$\frac{dP_{Li}}{dt} = \frac{\beta}{\rho_0 V_{Li}}\left(q_{LiIn} - \rho_{Li}\, Q_{Li} - \rho_{Li}\frac{dV_{Li}}{dt}\right) \tag{11}$$

Where q_{LiIn} is the mass flow rate. Note that variable density is maintained since ρ_{Li} is dependent of the state variable P_{Li} according to Eq. 7.

3.2 Pressure losses

The motion of fluids in a hydraulic system produce loss in total energy of the system. The loss in energy is due to the frictional forces acting against the direction of motion. These losses can be a considered a sum of internal friction between the fluid particles, as well as the friction between the fluid particles and environment. These losses can cause a delay in response times of the ram BOP and thus need to be investigated and added in the model for more accurate results. These losses in the hydraulic circuit are considered as pressure losses in this chapter.

3.2.1 Pipe friction

The fluid flowing through a pipe experiences friction between the walls of the pipe and the fluid particles. The liquid particles collide with the internal surface of pipe and thus experience a reduction in velocity which occurs along the surface of the wall. The pipe friction is dependent upon many factors like fluid velocity, fluid properties, and the diameter of the pipe and the relative roughness of the internal surface of the pipe.

The pressure loss caused by friction in a pipe depends mainly on the wall shear stress between the fluid and pipe internal surface. The fluid properties that affect the wall shear stress are the density and viscosity of the fluid. The other factors that cause a pressure drop are the velocity, the pipe length and the measure of the roughness of the pipe wall. The factors affecting the pressure drop can be summed up as follows:

$$\Delta P = F(V, D, L, \varepsilon, \mu, \rho)$$

The head loss due to pipe friction in turbulent flow can be calculated from Darcy-Welsbach equation (Munson, 2009) and is stated as follows:

$$K_{pf} = f\frac{l}{D}\frac{V^2}{2g} \tag{12}$$

Where f is the Darcy friction factor and D is the internal diameter of the pipe.

The Pressure drop due to pipe friction can be written as:

$$\Delta P_{pipe} = \rho g h = \ f\frac{l}{D}\frac{\rho V^2}{2} \tag{13}$$

The above equation is valid for turbulent flow in non-compressible liquid flow in pipes. The friction factor in equation (13) depends upon the flow regime of the liquid and the relative roughness of the pipe wall. The flow regime of the fluid is determined using the Reynolds number. Reynolds number is defined as a dimensionless parameter and is the ratio of inertia to viscous effects in the flow. The Reynold number can be calculated using the equation:

$$Re = \frac{\rho V D}{\mu} \tag{14}$$

Where V is the average velocity in the pipe.

The relative roughness of the pipe wall can be calculated using equation below:

$$\frac{\varepsilon}{d} \tag{15}$$

The values of the Reynolds number are then used in conjunction with the relative roughness to calculate the friction factor. Moody chart is used for friction factor estimations by using the relative roughness and Reynolds number. The Moody chart can be seen in the figure 3-1.

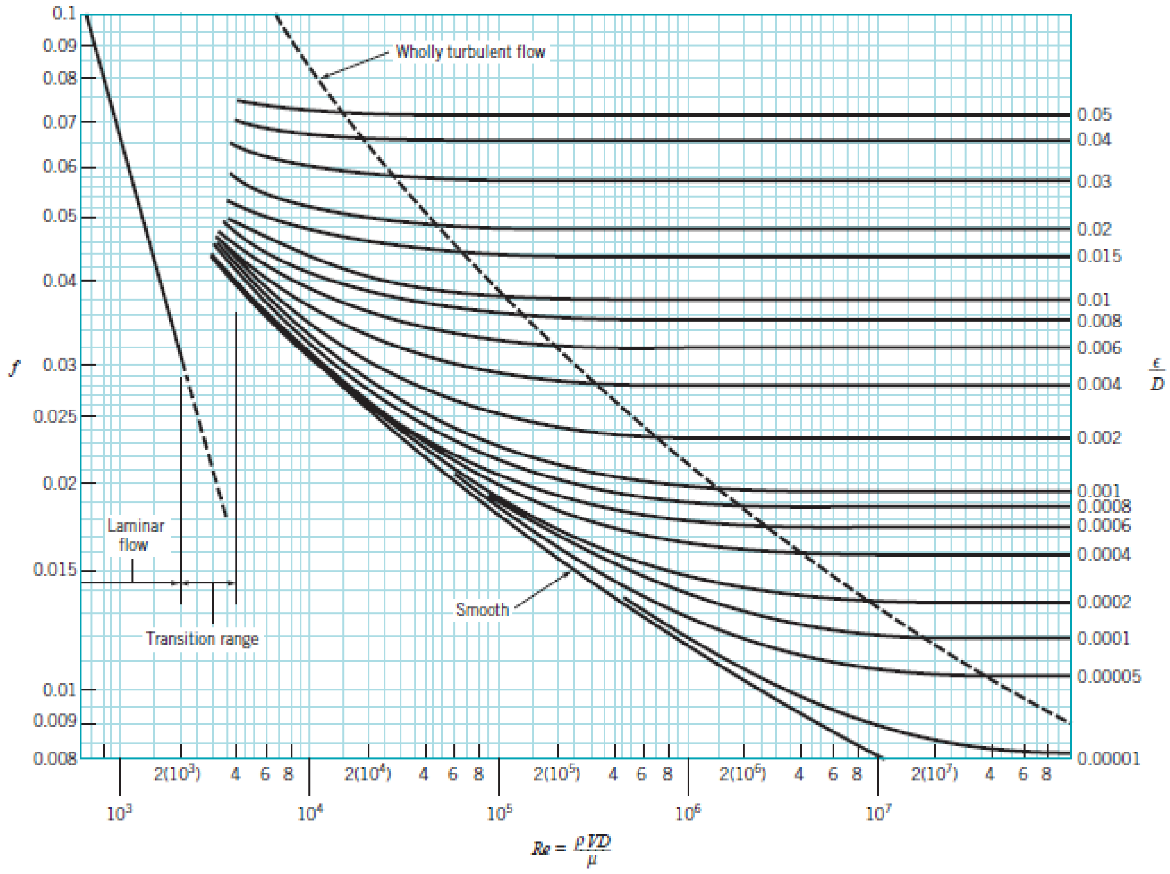

Figure 3-1: Moody chart

The determination of the Reynolds number is important to understand as it determines which flow regime is dominating. For general engineering purposes, the flow in a straight pipe is considered to be laminar if the ***Re*** is < 2100, while for ***Re*** > 2100 the flow is likely to be turbulent .(Munson, 2009).

For laminar flow with ***Re*** < 2100, the friction factor can be calculated using the Darcy friction factor. It should be noted that the roughness of pipe wall has no effect on the friction factor for laminar flow.

$$f = \frac{64}{Re} \tag{16}$$

Considering the hydraulic actuator in a BOP, the flow has a high velocity as low viscosity fluid is preferred in hydraulics. The flow can be considered as turbulent flow. The friction factor in turbulent flow can be calculated using the Colebrook equation (Rennels & Hudson, 2012) (17):

$$\frac{1}{\sqrt{f}} = -2log_{10}\left(\frac{\varepsilon}{3.7D_h} + \frac{2.51}{Re\sqrt{f}}\right) \tag{17}$$

Due to implicit nature of this equation, it must be solved iteratively. Halland equation as provided below can be used to Darcy friction factor explicitly when Reynolds number is between 2300 and 4000(Hager, 2010):

$$\frac{1}{\sqrt{f}} = -1.8log_{10}\left(\frac{\left(\frac{\varepsilon}{D_h}\right)^{1.1}}{3.7} + \frac{6.9}{Re}\right) \tag{18}$$

Where $\frac{\varepsilon}{D_h}$ is the relative roughness of the pipe calculated using roughness of pipe divided by the inner diameter of the pipe.

3.2.2 Valve Friction

The valves are generally used in pipelines to regulate the flow rate either by closing the valve or changing the geometry of the valve. The flow resistance due to presence of the valve makes a sizable portion of the resistance in the system. The loss coefficient denoted as K_v cause a pressure drop in the system and varies with the type of valve used. The loss coefficient is dependent upon the geometry of the component considered and depends slightly upon the fluid properties.

$$K_v = func.(geometry, Re)$$

However, for a turbulent flow regime where the Reynolds number is high and the flow through the component is dominated by the inertia effects rather than viscous effects, the geometry the only reason for minor losses. The loss coefficient(Rennels & Hudson, 2012) for a valve is defined as:

$$K_v = \frac{h_{l\,minor}}{\left(\frac{V^2}{2g}\right)} \tag{19}$$

The pressure drop due to the valve friction can be written as:

$$\Delta P_v = \rho g h = K_v \frac{\rho V^2}{2} \tag{20}$$

3.2.3 Pipe Diameter Friction

Many pipes contain various transition segments in which the diameter changes from one size to another. These changes can occur abruptly or smoothly through some type of change in area. These changes in the pipe diameter produce turbulence in the fluid flow. This turbulence in the flow contributes to the minor pressure loss. In an entry or exit point of the pipe, the fluid changes the flow speed which doesn't return to its initial value(Hager, 2010). The change in flow speed results in kinetic energy that is partially lost due to viscous dissipation of the fluid particles.

In a situation where the fluid has a broader entry point but a smaller exit point as illustrated in figure 3-2, the fluid may accelerate very efficiently but doesn't deaccelerate at the same rate. Some kinetic energy is partially lost, and a small pressure drop occurs in the system.

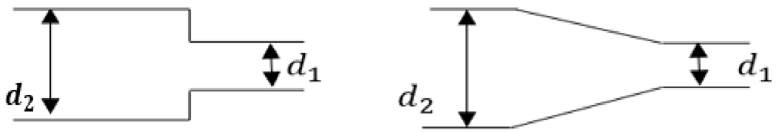

Figure 3-2: Sudden and gradual contraction

This situation explained above in figure 3-2 is of sudden contraction and the loss coefficient for sudden contraction (Rennels & Hudson, 2012) can be written as

$$K_{PD} = \frac{h_l}{\frac{V^2}{2g}} \tag{21}$$

It should be noted that the type of contraction depends upon the transition segment which determines the flow velocity of the fluid. The sudden contraction situation has a higher fluid velocity as compared to gradual expansion where the fluid velocity is relatively slower. Therefor the pressure drop in sudden expansion is relatively higher than gradual expansion(Hager, 2010). The pressure drop due to contraction can be written as:

$$\Delta P_{PD} = \rho g h = K_{PD} \frac{\rho V^2}{2} \tag{22}$$

In a situation where the fluid has a smaller entry point and a larger exit point as illustrated in the figure 3-3, the situation can be considered as sudden or gradual expansion. In case of a gradual expansion, the fluid leaves the smaller pipe and initially forms a jet-type structure as it enters the larger pipe. After the exit, the jet stream disperses across the pipe and fully developed flow becomes established again. During this some of the kinetic energy is lost resulting in pressure losses.

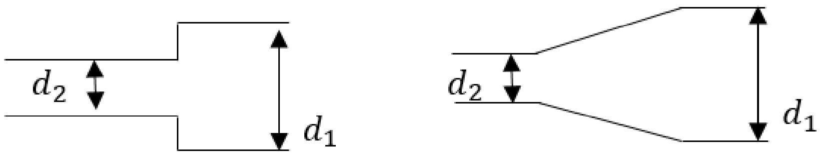

Figure 3-3: Sudden and gradual expansion

The loss coefficient for sudden or gradual expansion are dependent upon the relative speed in entry and exit points. The equation for sudden expansion (Rennels & Hudson, 2012) becomes:

$$K_{PD} = \left(1 - \frac{D_2}{D_1}\right)^2 \tag{23}$$

Where D_2 and $D_1$are the area of entry and exit points of the fluid respectively. It should be noticed that the loss coefficient for gradual expansion are greater than the corresponding contraction loss coefficient. The pressure drop due to expansion can then be written as:

$$\Delta P_{PD} = \rho g h = \left(1 - \frac{A_1}{A_2}\right)^2 \frac{\rho V^2}{2} \tag{24}$$

3.2.4 Bends

Bends in pipes produce a greater head loss than if the pipe is straight. The losses are due to the separated region of flow near the inside of the bend and the swirling secondary flow at the boundary walls. This occurs due of the imbalance of the centripetal forces as a result of curvature of the pipe centreline (Hager, 2010). The loss coefficient for bends are also dependent upon the flow regime of the fluid i.e. laminar or turbulent. The head loss of a pipe bend(Rennels & Hudson, 2012) can be calculated from the following equation:

$$K_b = \frac{h_b}{\frac{V^2}{2g}} \tag{25}$$

The bends also create extra pipe friction as bend and elbows create an extra addition of pipe length. This addition of pipe length depends upon the curvature and radius of the pipe. This length has been added in the pipe friction section of modelling. The head pressure loss from bends can be written as:

$$P_b = \rho g h_b = K_b \frac{\rho V^2}{2} \tag{26}$$

3.2.5 Junctions and fittings

The hydraulic system contains many fitting and junctions for the choice of flow direction. The loss coefficient for these components depends mainly on the shape of the component. The Reynolds number has weak impact on the pressure drop through these components. Thus, the loss coefficients depend upon the curvature angle and whether the pipe joints are threaded or flanged. The manufacturing process relies more on cost reduction rather than for reduction of head losses. The head loss for Junctions (Rennels & Hudson, 2012) or fitting can be expressed as:

$$K_J = \frac{h_j}{\frac{V^2}{2g}} \tag{27}$$

Where K_J is the loss coefficient for junction, fitting or filters.

The head pressure loss can be expressed as:

$$P_J = \rho g h = K_J \frac{\rho V^2}{2} \tag{28}$$

3.2.6 Summation of pressure losses

The major and minor losses in a hydraulics system are summed up in the equation 29. It is important to mention that all the components in the hydraulic circuit for ram BOP are included and the loss coefficient are used accordingly.

$$P_{loss} = \sum_{i=1}^{n} P_{pipe} + \sum_{i=1}^{n} P_V + \sum_{i=1}^{n} P_{PD} + \sum_{i=1}^{n} P_B + \sum_{i=1}^{n} P_J \tag{29}$$

The above equation can be simplified to include the major and minor losses in the system as:

$$P_{loss} = \sum_{i=1}^{n} P_{Major} + \sum_{i=1}^{n} P_{Minor} \tag{30}$$

The proposed equation 29 and equation 30 provide a good estimation for the pressures losses from the master regulator to the hydraulic actuator. The schematic P&ID diagram of the power circuit has been shown in the figure 3-4 for easier understanding:

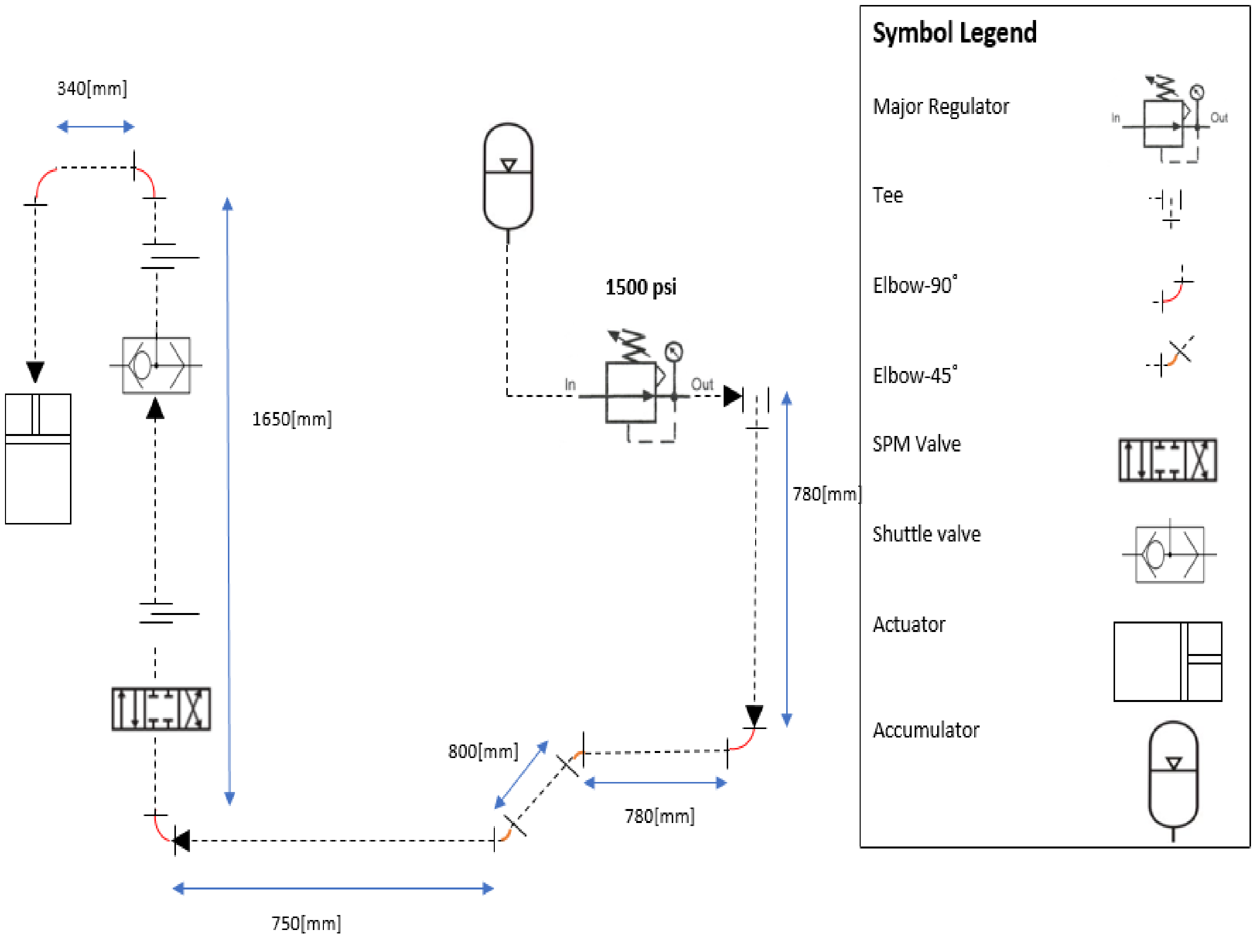

Figure 3-4: Schematics of Ram BOP Hydraulic Circuit

3.3 Frictional Losses

This section will describe the most significant frictional losses in the ram BOP actuation operations. These losses can cause a delay in response times of the ram BOP and thus need to be investigated and added in the model for more accurate results. These losses in the system are considered as friction losses in this section.

3.3.1 Friction cylinder

The shear rams in a BOP are moved using hydraulic cylinders which operate on the principles of Pascal's Law. The main function of the hydraulic cylinders is to transform hydraulic power into mechanical power by means of a translating piston rod. The moving piston is connected to a piston rod which uses the cylinder housing to seal off two pressure chambers. The chambers are connected to the remaining hydraulic system. The piston moves in the direction when hydraulic flow is led to either one of the pressure chambers. The movement requires a certain pressure difference across the piston depending upon the load on the piston it must move.

The movement of shear rams through the annulus will only have hydrostatic pressure exerted by the mud as load but during the shearing of the drill pipe, the load will be the shearing force of the drill pipe.

In addition to the friction losses experienced from the master regulator to the actuators, there would also be mechanical friction in between the piston and the cylinder when the shear ram is moving to secure the annulus and the drill pipe. This can be the solid-solid friction between the moving piston and the wall of the cylinder. In addition, there will also be solid-solid friction between the rod and rod seals which are illustrated in the figure 3-5. These losses are considered as mechanical losses and decrease the overall efficiency of the system.

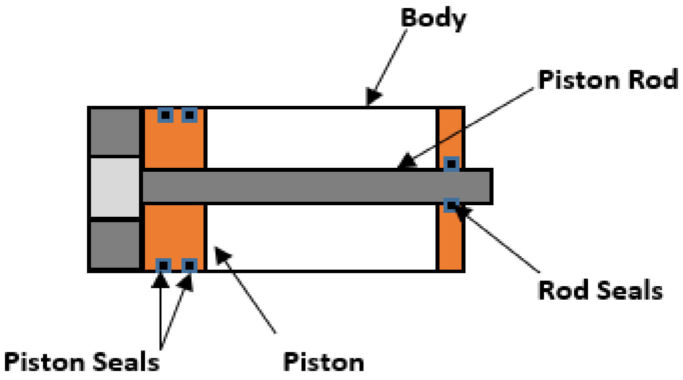

Figure 3-5: Friction sources in a Hydraulic Cylinder

The mechanical system involving a hydraulic actuator depends upon the friction characteristics of the piston and cylinder. The sliding piston seal is lubricated with thin oil film to minimize the frictional forces, but the condition prevailing here is not of a dry neither fully lubricated. The situation becomes more complex when simple models like coulomb friction and viscous friction cannot be used for the calculations, but rather a customized combination of these models.

The coulomb friction is described as friction force of constant magnitude acting in a direction opposite to the direction of motion. The coulomb friction force depends upon the force pressing together two surfaces and the friction coefficient μ which depends upon the stiffness of the material in contact. The coulomb friction can be written as

$$F_c = \mu . F_N \text{ Sign (v)} \quad (31)$$

Where v is the relative velocity of the moving object. The presence of a thin lubrication layer between the piston and the cylinder voids the use of coulomb friction as the only friction source in case of hydraulic cylinders. The nature of friction is neither dry nor completely wet in this case. In addition, the coulomb friction law is not sufficient to model frictional characteristics in case of a dynamic modelling.(Jitendra Yadav, 2015)

The viscous friction model represents the frictional forces proportional to the sliding velocity of the sliding piston. Viscous friction force is linear with respect to the velocity and can be expressed as

$$F_v(v) = \sigma_v v \quad (32)$$

Where σ_v is the viscous friction coefficient.

The model most suitable for dynamics modelling in a hydraulic cylinder is represented by the LuGre model (Muvengei & Kihiu, 2011; Tran & Yanada, 2013). The LuGre model tries to combine coulomb friction, viscous friction and the static friction into one equation. The advantage with the LuGre model is that both pre-sliding and the sliding regimes are described by the same model. The LuGre model expresses the total friction in the system as:

$$F_f = \sigma_0 Z + \sigma_1 \dot{Z} + \sigma_2 V \tag{33}$$

The LuGre model considers the Z parameter as surface asperities (hardness) which undergoes deflection in presence of a lateral force. The value of parameter Z is dependent upon the elastic properties of the manufacturing material and the normal force in the surface. The typical values of Z in hydraulic systems varies from 0 to 40 microns. With the dependency of Z on elastic properties and the normal force, the LuGre model can be written as:

$$F_f = \sigma_0 Z + \sigma_1 v - \frac{\sigma_0 . v . Z}{\left[F_c + (F_s + F_c) e^{-(\frac{v}{v_s})^2}\right]} + \sigma_2 v \tag{34}$$

The significance of the different parameters is described as under:

σ_0: is the average stiffness of the material. The value depends upon the surface properties in contact and the normal load on the surface. The typical values range between 10^4 to 10^6 N/m.

σ_1: defines the micro-damping at some very low viscosities. Many researches take its value as square root of the σ_0 (De Wit, Olsson, Astrom, & Lischinsky, 1995). The typical values range between 200 to 1000 Ns/m.

σ_2: accounts for the viscous friction in the system when the sliding piston is in a moving position. The contribution of the viscous friction is velocity dependent and can be considered negligible in presiding regime. The value ranges between 0.1 to 0.7 Ns/m.

F_c: is the static friction force which corresponds to the coulomb friction.

F_s: is the dynamic friction force. The value of dynamic friction is taken as 50 % higher than the static friction force.

Vs: is the Stribeck velocity.

The kinetic and viscous friction in the hydraulic cylinder is dependent of velocity and change accordingly with the ram movement. However, the Stribeck effect (figure 3-6) is a phenomenon which occurs when friction decreases with the increase in the velocity on a lubricated surface. This applies also to the hydraulic cylinder as the surface between the piston and the pistons seals of lubricated nature. The assumption that must be taken during the modelling is omitting friction jump at zero velocity (Stribeck effect) to avoid numerical errors in the dynamic model.

The hydraulic cylinder systems model described above is based on ideal conditions without any power loss in the hydraulic system. However, the hydraulic cylinder has both volumetric and hydro-mechanical losses which need to be accounted. These losses define the efficiency of the hydraulic system. The volumetric losses can be caused by leakage across the sealing between the piston and the housing. These losses are very small and can be ignored to reduce the complexity of the model. The hydro-mechanical losses however cannot de disregarded which are caused by the mechanical friction between pistol seals and the cylinder housing or piston rod and the cylinder housing.

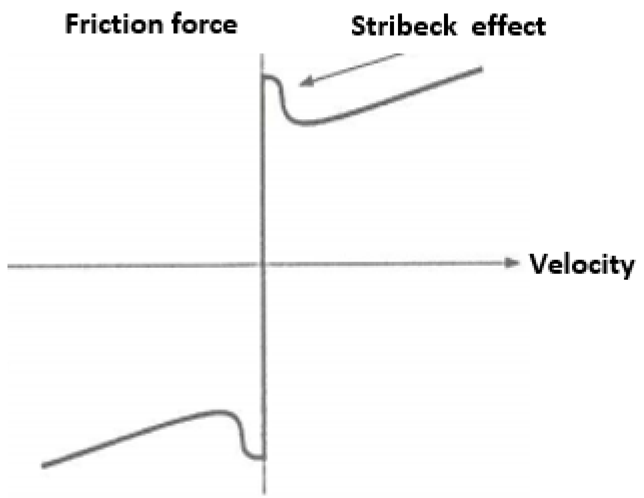

Figure 3-6 Kinetic and Viscous Friction with Stribeck effect

3.3.2 Shear ram Friction

The motion of the shear rams through the annulus is also a factor which needs to be included in the research for more precise calculations. The presence of mud in the annulus would resist to the motion of the shear rams resulting in excess force to be supplied by the whole hydraulic system. In fluid dynamics, drag is a force which acts opposite to the relative motion of any object through a fluid. The resistance to the motion of shear rams in this case is considered as the drag force in this chapter. Drag depends upon several factors like the properties of the fluid, size and speed of the object which can be expresses by means of the drag equation:

$$F_D = \frac{1}{2}\rho v^2 A.C_D \tag{35}$$

Where ρ is the density of the fluid, v is the velocity of the object relative to the fluid, A is the cross-sectional area and $C_D$is the drag coefficient.

The drag coefficient depends upon the shape of the object and the Reynolds number $R_e = \frac{\rho v D}{\mu}$.The dependency of drag co-efficient is mostly conducted experimentally specific to fluid under investigation. Several researches have been made to relate Reynolds number with the drag coefficient and the most suitable for non-Newtonian fluids with a relatively high Reynolds number is the Haider-Levenspiel model (Phillip P. Brown 2003). The Haider-Levenspiel model is as follows:

$$C_D = \frac{24}{R_e}(1 + 0.1806R_e^{0.6459}) + \frac{0.4251}{1 + \frac{6880.95}{R_e}} \tag{36}$$

The drag coefficient C_D can then be calculated by finding the Reynolds number using the velocity of the moving ram, the density of the mud, the cross-sectional diameter of the shear ram and viscosity of the mud.

In addition to the drag force, the movement of ram through the mud in the annulus faces resistive force due to hydrostatic pressure of the mud. This can be calculated with the use Pascal's law. The hydrostatic pressure in the well column right above the ram can be multiplied by the cross-sectional area of the ram in order to find the resistive force against the motion of the rams as represented in equation (37).

$$F_{Well} = P_{hyd}.A_{ram} \tag{37}$$

The drag force due to movement of the ram has very low significance because the ram moves very slowly. Thus, the drag force can be considered as a minor loss. However, the resistance from the hydrostatic pressure in the annulus has been added in the calculation for higher accuracy.

3.3.3 Friction ram-annulus interface

The author has also made efforts in finding out the resistance force experienced by the shear ram body and the door cavity of the BOP connected to the annulus. The illustrative description can be in the figures below (Varco, 2010)below:

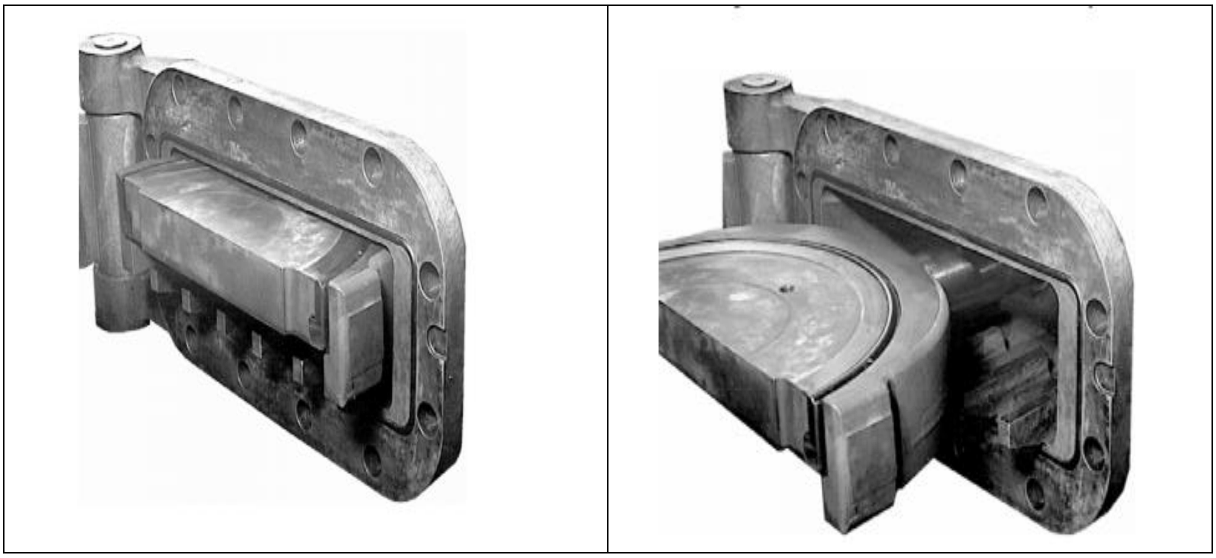

Figure 3-7: Ram BOP door cavity

There could be seen a gap between the door cavity and the ram which explains that there isn't possibly any dry friction between the two surfaces.

When sealing the annulus, the ram body moves along the walls of the annulus which must be added to the overall friction of the system. Once the ram moves from the initial position along the outer boundary of the annulus, there is kinetic friction between annulus walls and ram body. The normal force has a very small value and the co-efficient of friction between two lubricated steel surfaces has a very small value. The justification to a very small co-efficient of friction is due to the nature of contact between the two surfaces. The presence of mud in the annulus results in wet conditions around the wall surface and thus the nature of friction can be taken as wet. The conclusion here is that this can be considered as a minor loss and doesn't contribute much to the overall friction in the system.

3.3.4 Summation of friction forces

The frictional losses in ram BOP actuation process are summed up in the equation (38). It is important to understand that most significant frictional losses have been added into the calculation but there are always minor losses that are not considered. These frictional losses can be avoided by tuning the model to calculate the overall friction.

$$F_{loss} = \sum_{i=1}^{n} F_{CF} + \sum_{i=1}^{n} F_{D} + \sum_{i=1}^{n} F_{Well} \tag{38}$$

The different friction losses are represented in the figure below:

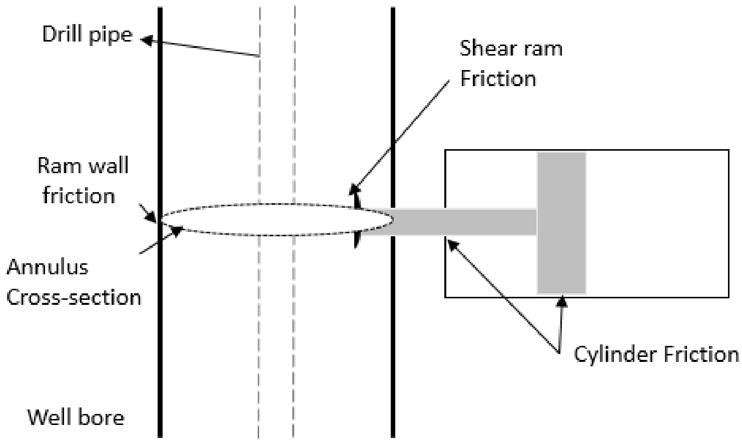

Figure 3-8: Mechanical Friction forces

3.4 Bulk modulus

Any changes in the physical properties of the hydraulic fluid can cause a delay in the dynamic response of the system. The hydraulic fluid experiences a reduction in volume by the ratio of pressure applied and this phenomenon depends on many factors, e.g. pressure, temperature and undissolved air volume. (L. Hružík, 2013). The stiffness of the hydraulic fluid used in hydraulic circuit is of considerable importance as it determines the speed and nature of the dynamic response as well as the static accuracy that can be achieved when positioning loads. (Chapple, 2004). The relationship between changes in volume subjected to change in pressure is referred to as bulk modulus of elasticity of liquid and can be written as:

$$B = \frac{Change\ in\ pressure}{volumetric\ strain} = -\frac{V\Delta P}{\Delta V} \tag{39}$$

For liquid the relationship between pressure and liquid is non-linear and depends upon the isothermal or adiabatic expansion of the fluid.

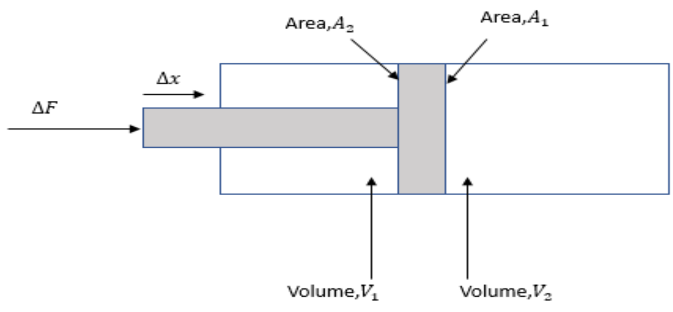

Figure 3-9: Hydraulic Cylinder

The difference between the isothermal and adiabatic oil bulk modulus depends upon the compression speed of the fluid. The adiabatic process involves high speed changes in pressure and in isothermal bulk oil modulus, the process occurs at slow pressure changes (L. Hružík, 2013). The bulk modulus can be subdivided into secant and tangent models. The secant model is more suitable for high pressure changes and the tangent modulus applies more to the small dynamic pressure changes.

The secant modulus of elasticity is defined as:

$$K_s = \Delta p . \frac{V_0}{\Delta V} \tag{40}$$

The tangent modulus of elasticity can be defined by the formula:

$$K_T = V . \frac{dp}{dV} \tag{41}$$

Where V_0 is the initial volume, V is the oil volume after compression,ΔV is the oil volume change and ΔP as the pressure change.

In a double acting hydraulic actuator as used in a ram BOP, the force variation can be written as:

$$\Delta F = -A_1 \Delta P_1 + A_2 \Delta P_2 \tag{42}$$

Using the bulk modulus of elasticity, the force variation can be written as

$$\Delta F = B\left(\frac{A_{1\Delta V_1}}{V_1} - \frac{A_{1\Delta V_1}}{V_2}\right). \tag{43}$$

The area of interest in developing of a BOP hydraulic circuit is the change in the volume of the hydraulic fluid when pressure is supplied by the accumulator. This change in the pressure reduces the volume of the hydraulic fluid and in return increases the response time of the system. As the API standards require the BOP to close the shear rams less than or in 30 seconds, the hydraulic actuation process can be considered as isothermal and tangent. This is due to the fact that isothermal bulk modulus more suitable to the slow compression speed and secant model is more applicable for small pressure changes (Wit, Olsson, Astrom, & Lischinsky, 1995).In hydraulic cylinders, a stiff system is often required meaning a high bulk modulus but in a dynamic model the values of the bulk modulus can be slightly altered to get a stable system and decrease the time step of iterations.

3.5 Accumulators

An accumulator is a storage device which can store and release specific quantity of fluid at the required system pressure (Parr, 2011) .The accumulators are preloaded by either a spring, weight or compressed gas to accumulate energy by volume compensation. It is important to notice that there must be equilibrium between pressures exerted by the fluid that is equal to the counter pressure exerted by the spring, load or the compressed gas(SRL, 2016). This helps in providing large flow rates instantly, reducing the need of large pumps and motors to be used which in turn reduces the installation and operational costs.

The accumulator consists of different zones i.e. fluid zone, gas zone and a separating gas-tight element. As it can be seen in figure 3-10, any increase in pressure of the hydraulic circuit will push the diaphragm inwards and pressurizes the gas. The situation is reversed when the pressure of the hydraulic circuit decreases, causing the gas zone to expand. This expansion of gas forces the fluid into the poppet valve which is the entry point to the hydraulic system

The accumulator bank in a ram BOP are pre-charged and provide the necessary pressure change in the hydraulic circuit to control the ram movement. There are three types of accumulators used in the industry as listed below:

1. Bladder accumulators
2. Piston accumulators
3. Diaphragm accumulators

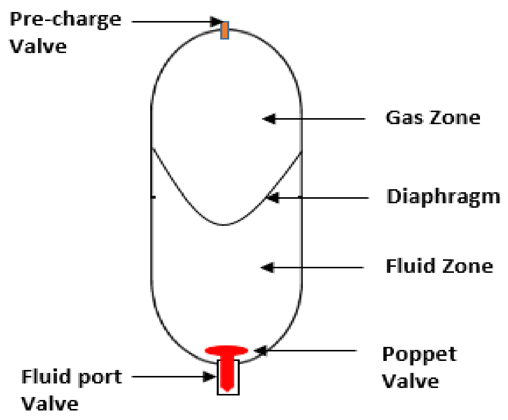

Figure 3-10: Accumulator

All the accumulator types work almost on the same principles. The fluid zone is connected to the hydraulic circuit, and pressure increase in the circuit results in entry of fluid into the accumulator causing compression of the nitrogen gas. If there is any drop in the circuit, the accumulator gas which is relaxed would push the fluid delivery to the circuit.

The advantage with the use of accumulators is that there is no static friction to be overcome as compared to piston seal, and there is no piston mass to be accelerated or deaccelerated. This makes the BOP operation more robust and requires minimal maintenance. Another advantage is the capacity of the accumulator bank which can be increased significantly by adding gas cylinders in series, in case a higher pressure is required.

3.5.1 Pre-Charge

The pre-charge process involves filling the gas side of the accumulator with dry, inert gas such as hydrogen before admitting fluid to the hydraulic side. It is important to charge pre-charge the accumulator under the correct specified pressure as it will determine volume of fluid remaining in the accumulator when the system pressure is at its minimum. Typically, the pre-charge pressure of the accumulator is 90 percent charged of the max capacity.

$$P_0 = 0.9P_1 \tag{44}$$

The accumulator is charged with nitrogen gas due to its inert properties. The accumulator pre-charge can be described in stages(SRL, 2016) as stated below:

Stage 1: accumulator is empty

Stage 2: accumulator is pre-charged

Stage 3: accumulator is charged. The hydraulic fluid flows into accumulator and the system is pressurized

Stage 4: accumulator is fully charged. No more additional

Stage 5: accumulator goes back to charged pressure as stage 3. The system pressures falls and the hydraulic fluid is discharged from the accumulator

Stage 6: accumulator is partially discharged and minimum pressure is reached.

The temperature variations can seriously affect the pre-charge pressure of an accumulator. As the temperature increases, the pre-charge pressure increases as per Charles's law. The temperature variation must be factored in when designing of the accumulator. The temperature and pressure relationship at pre-charge can be written as:

$$P_{filled} = P_0 \frac{T_{filled} + 273}{T_{working} + 273} \tag{45}$$

The minimum system pressure would also determine the size of the accumulators to be used or if there is a need to install a series of cylinders to increase capacity.

3.5.2 Volume Calculation

The compression and expansion inside the accumulator happens according to the Boyle-Mariette law (SRL, 2016) regarding the status change in the gas laws:

$$P_0 V_0^n = P_1 V_1^n = P_2 V_2^n \tag{46}$$

Where $V_0^n$is nitrogen pre-charge volume at pressure P_0 which is the maximum volume of gas which can be stored in the accumulator and it is equal to, or slightly lower than, nominal capacity.

The value of n is called the polytrophic constant which depends upon the heat exchange during the robustness of the operation. If there is heat exchange between environment and the gas at constant temperature due to compression of expansion of the nitrogen gas, the value of n can be taken as 1 and the process can be regarded as isothermal.

However, if the operation is so quick that the no heat exchange occurs, the value of n can be taken to be 1.4 as the condition is adiabatic. The value of the polytrophic constant can also be determined from Boyle-Marriot law if the expansion or compression time is known.

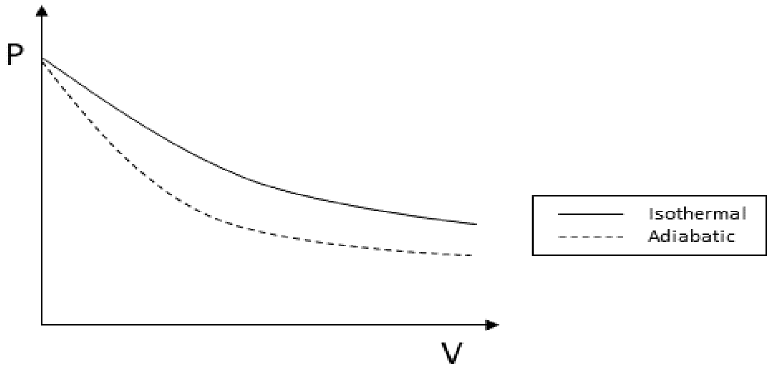

Figure 3-11: Nitrogen Gas expansion

3.6 BOP Pressure

The BOP pressure defined in this section is the collective pressure experienced at the BOP during closing operation of the ram BOP. Hydraulic actuators are used to push the ram BOP towards the well bore to close a BOP. The hydraulic actuators get their pressure support from the accumulators to move the ram BOP. The BOP pressure for scenario 1 and scenario 2 is described in the subsequent sections.

3.6.1 Free ram movement (scenario 1)

The first situation of interest is when there is no DP present in the well. The ram BOP will experience relatively lower BOP pressure as the rams move freely. The second situation can be considered more complex if there is a DP present in the well. In this situation greater BOP pressure is experience as the hydraulic actuators require pressure support to move the free moving ram but also provide the necessary pressure support the shear the DP in case of shear ram BOP.

For the free movement of the ram BOP, it is necessary to have enough pressure support to overcome pressure and frictional losses as explained in chapter 3.2 and 3.3 respectively. A relationship between pressure at the master regulator and the pressure required to close the ram BOP can be written as:

$$P_{Reg.} > \sum_{i=1}^{n} P_{loss} + \sum_{i=1}^{n} F_{loss} \tag{47}$$

Along with the major and minor losses in a hydraulic circuit, it is necessary to include the pressure in the well and pressure working of an actuator during ram movement. Figure provides a simple illustration of pressure distribution in a hydraulic actuator.

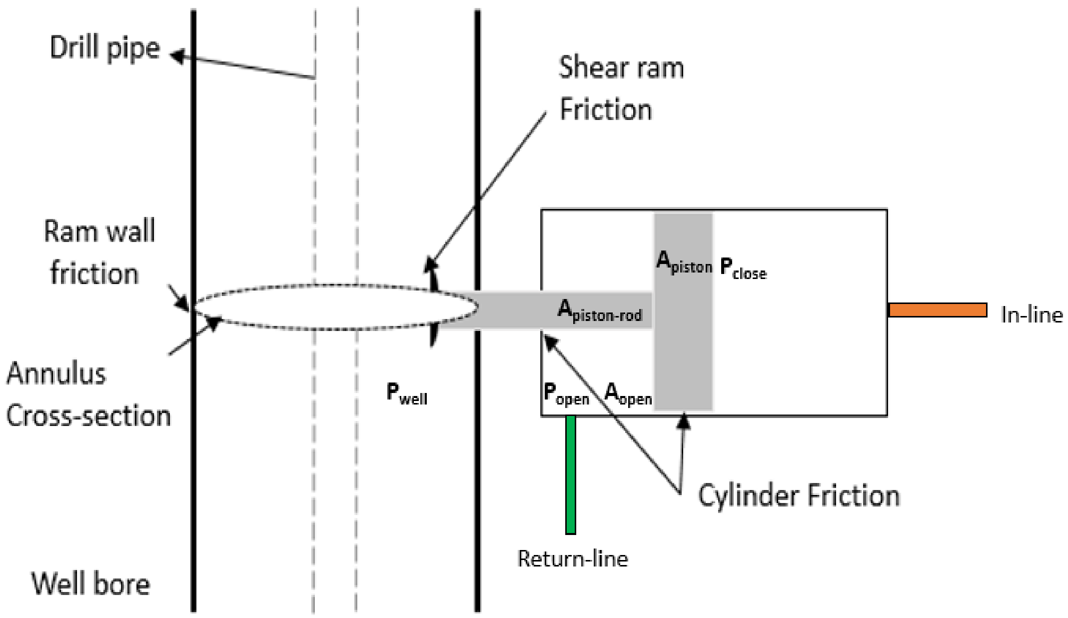

Figure 3-12: Ram BOP-Annulus interface

From figure 3-12, it can be observed that the pressure P_{close} must compensate for pressure in the well P_{well}, pressure at the return-line P_{open} and the friction losses in the mechanical components. The equation (47) can be rewritten as:

$$P_{Reg} > \frac{P_{well} \cdot A_{piston-rod}}{A_{piston}} + \frac{P_{open} \cdot (A_{open} - A_{piston-rod})}{A_{piston}} + \frac{\sum F_{losses}}{A_{piston}} + \sum_{i=1}^{n} P_{loss} \qquad (48)$$

The equation proposed above states that the pressure P_{Reg} needs to be maintained at the master regulator, to successfully close the ram BOP. The pressure in the well P_{well} depends upon the hydrostatic pressure of the mud column and can vary if a higher mud density is used or there is a kick situation. The P_{Reg} must also overcome the pressure losses P_{loss} in the hydraulics circuit. The mechanical losses that occur during the movement of the ram must also be compensated by the $P_{Reg.}$

The pressure difference in P_{open} and P_{close} during the free ram movement can be seen in figure 3-13. The different points represent different position of the ram as explained:

A. Process initiation.

B. Ram movement start.

C. Ram fully extended, annulus secured as pressure builds up.

D. Pressure stabilization as ram fully closed.

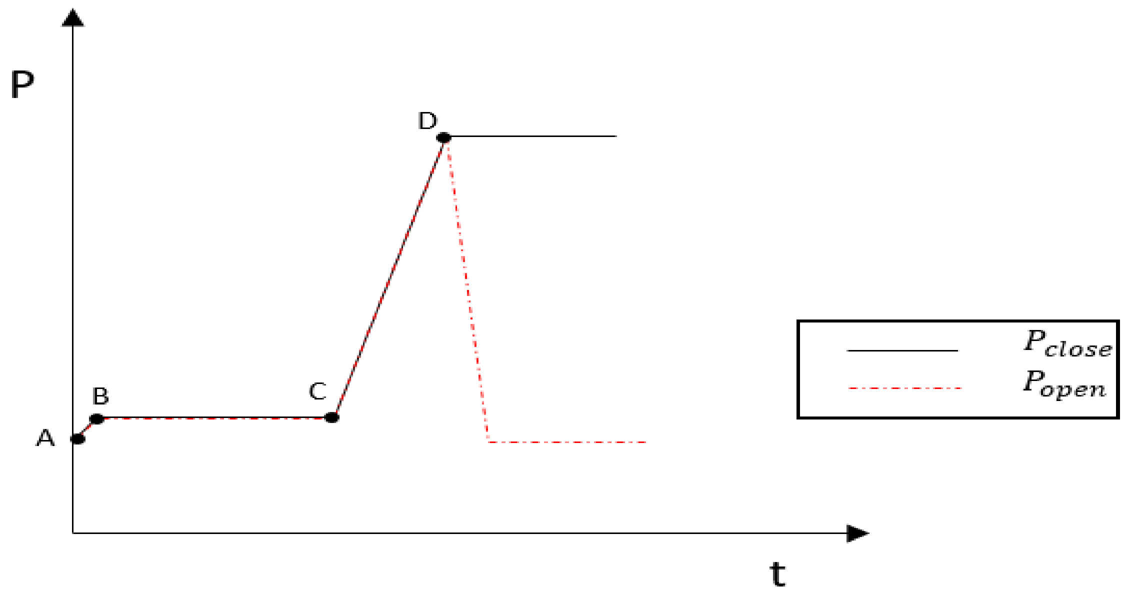

Figure 3-13: Pressure vs time for Scenario 1

The changes in the flow rates at the entry and exit point of the hydraulic actuator for free ram movement can be represented in figure 3-14. The different points represent the various positions of the shear ram as explained below:

A. Process initiation and hydraulic cylinder experiences static friction
B. Ram movement starts as high flow rate provides pressure support
C. Ram nearly fully closed
D. No difference in the flow rate as pressure stabilized and ram fully closed

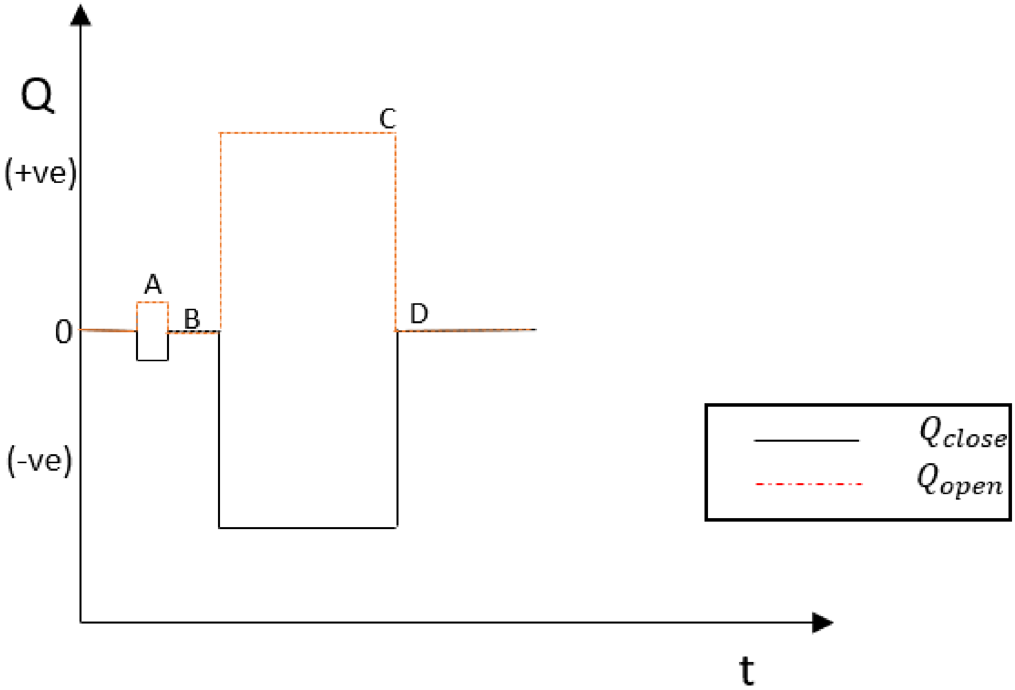

Figure 3-14: Flowrate vs time for Scenario 1

3.6.2 Shear ram movement (scenario 2)

The second situation of interest is when there is a DP present in the wellbore. This would result in higher BOP pressure required to seal the well. The extra pressure required to shear the drill pipe depends mainly on the meta1nllurgical properties of the drill pipe. Furthermore, the shape of the shear ram also affects the in-situ stress required to shear the pipe.

The equation (48) can be modified by addition of the shear force required to shear the drill pipe and can be written as:

$$P_{Reg} > \frac{P_{well} \cdot A_{piston-rod}}{A_{piston}} + \frac{P_{open} \cdot (A_{open} - A_{piston-rod})}{A_{piston}} + \frac{\sum F_{losses} + F_{shear}}{A_{piston}} + \sum_{i=1}^{n} P_{loss} \quad (49)$$

The force required to shear the drill pipe depends upon the yield strength of the drill pipe τ and the cross sectional area of the drill pipe A (Liu, Guo, Guo, Cole, & Sridhar, 2017; Tekin, Choi, Altan, & Adin, 2015).The simplest formula to calculate the shear force can be written as:

$$F_{shear} = \tau . A \quad (50)$$

There are many OEM formulas used in the industry to calculate the shear force required to cut different types of pipes. The most significant OEM formula (Tulimilli et al., 2014) are as follows:

Nominal weight method:

No wellbore pressure: $$P_{shear} = \frac{(N1 * \omega_n * \sigma_{yield})}{A_1} \quad (51)$$

Including well bore pressure: $$P_{shear} = \frac{(N1 * \omega_n * \sigma_{yield}) + (P_w * A_2)}{A_1} \quad (52)$$

Dimensional method:

No wellbore pressure: $$P_{shear} = \frac{(N1 * \sigma_{yield})(OD^2 - ID^2) * 2.92}{A_1} \quad (53)$$

Including well bore pressure: $$P_{shear} = \frac{(N1 * \sigma_{yield})(OD^2 - ID^2) * 2.92 + (P_w * A_2)}{A_1} \quad (54)$$

Where $N1$ is the ram type/pipe grade constant, obtained from laboratory testing and ω_n is the nominal weight of the pipe.

The pressure difference in P_{open} and P_{close} during the ram movement for scenario 2 can be seen in figure 3-15. The different points represent different position of the ram as explained:

A. Process initiation.

B. Ram movement start.

C. Ram reaches DP, high shear force required as pressure builds up.

D. Shear pressure reached.

E. DP sheared, and pressure falls

F. The ram BOP has reached stroke length and both rams attached to each other

G. Pressure stabilization as ram fully closes.

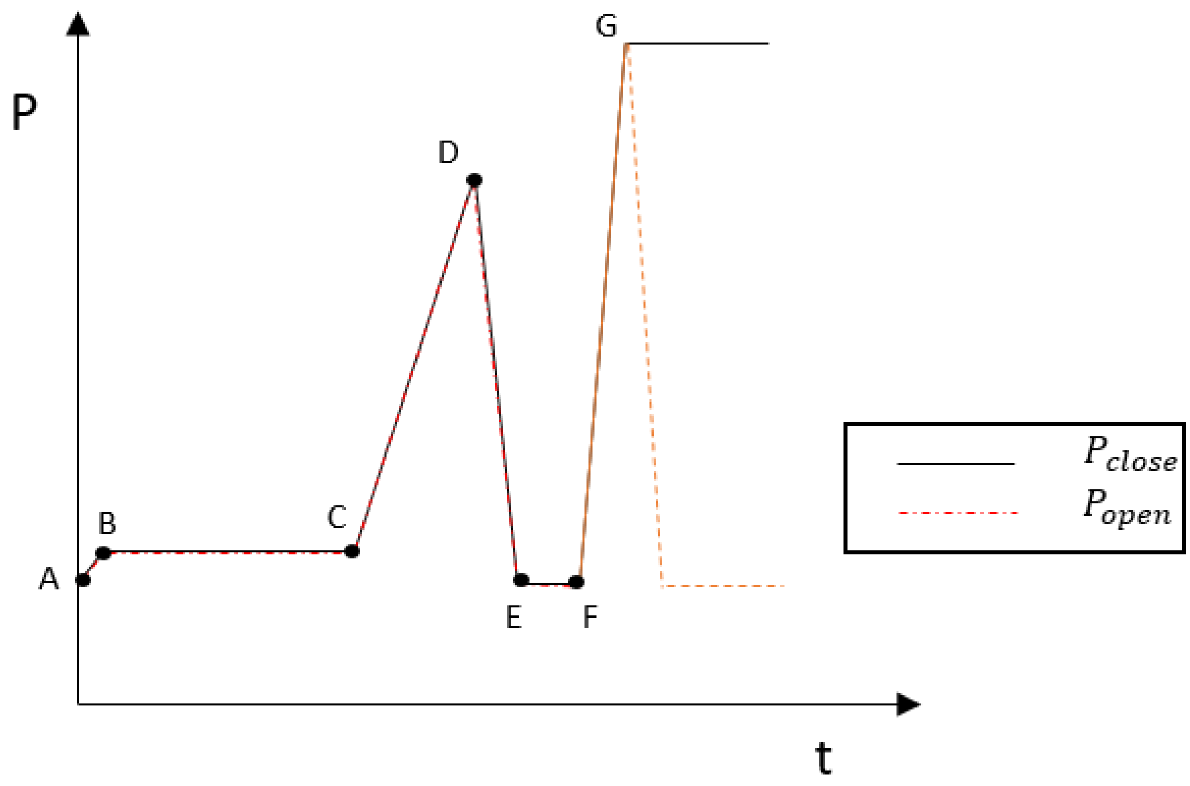

Figure 3-15: Pressure vs time for Scenario 2

The changes in the flow rates at the entry and exit point of the hydraulic actuator for scenario 2 can be represented in figure 3-16. The different points represent the various positions of the shear ram as explained below:

A. Process initiation and hydraulic cylinder experiences static friction

B. Ram movement starts as high flow rate difference provides pressure support

C. Ram reaches DP

D. No difference in the flow rate as ram reaches DP

E. High flowrate difference to provide shear force

F. Rams meet after shearing drill pipe

G. Flow difference zero as pressure stabilized

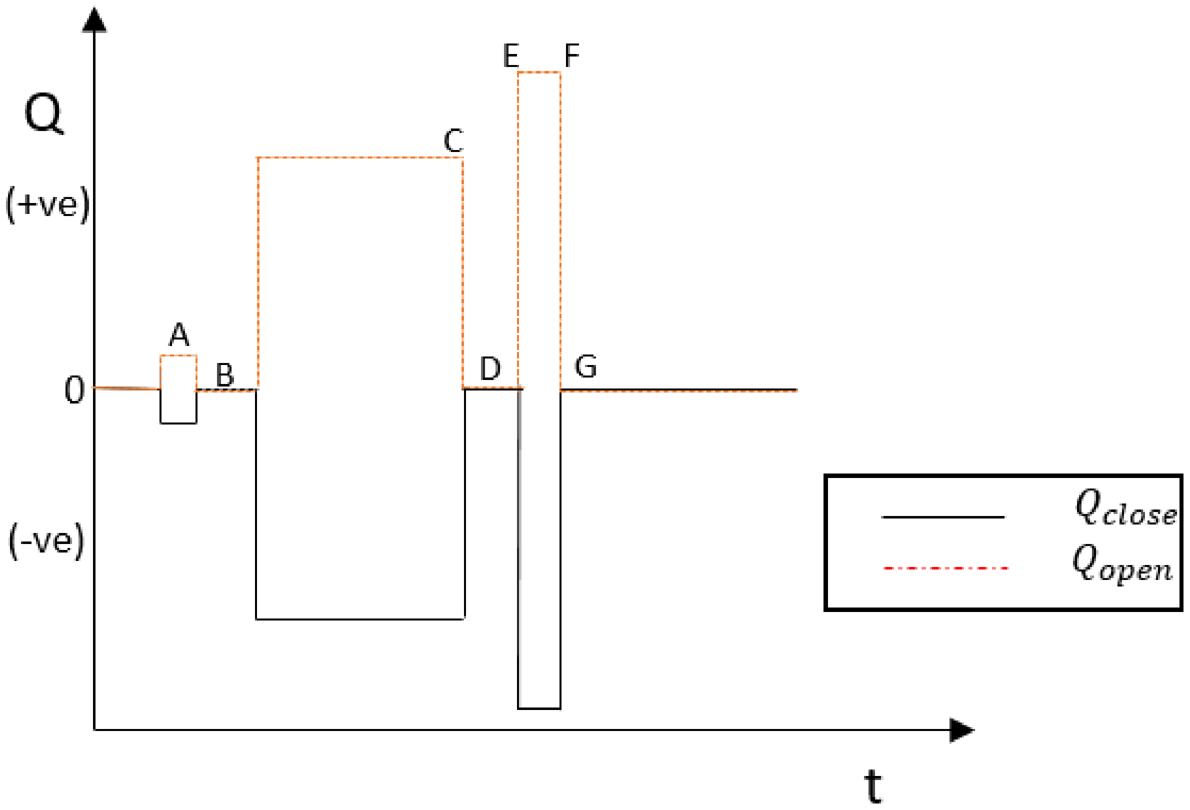

Figure 3-16: Flowrate vs time for Scenario 2

With reference to figure 3-15 and 3-16, the BOP pressure required would be higher if there is a DP present in the well bore. This additional shear force would result in higher pressure build-up and thus the time required to shear the drill pipe would increase. The higher pressure would also result in higher flow rates in the entry and exit points of the hydraulic cylinder in order to create a higher pressure.

4 Modelling

This section explains the inputs, calculations, parameter selection and results from the simulations for ram BOP. The modelling section is divided into the following subsections:

1. The base model
2. ODE for hydraulic actuator
3. ODE for flow in pipe
4. Modelling without a DP in borehole (scenario 1)
5. Modelling with a DP in borehole (scenario 2)

The modelling of the ram BOP has been performed in Matlab with the help of theory explained in chapter 3. It is necessary to emphasize that the model is designed only for the hydraulic circuit and the actuator. The model doesn't include volume discharge calculation for accumulators. The calculations from the ODE are used to estimate the time required to move the shear rams from the start position to complete the seal-off of the well. The pressure at the master regulator is set to be 1500 psi (W.C.Goins, 1983) and the system is tested to see if the pressure is enough to complete the actuation operation. It must be noted that all the pressure losses in the hydraulic line and friction losses in the hydraulic actuation are considered for all calculations.

A simple overview of the hydraulic circuit which is used for modelling can be seen in the figure 4-1. The main pressure source is the hydraulic accumulators which provide a pressure support of 3000psi (W.C.Goins, 1983). The pressure from the accumulator is regulated by the master regulator to a value of 1500 psi which is denoted as $P_{manifold}$. This pressure at the master regulator can be directed to either the annular preventer or the ram BOP depending upon the choice of activation. As the operator gives the command through the control system which ram is to be activated, the pressure flow is directed towards the selected ram BOP. This pressure is then received at a three-way valve denoted as SPM valve. The SPM valve is used to start the closing or opening operation by directing the fluid to respective shuttle valves.

The pressurized hydraulic fluid passes from the SPM valve on to the Shuttle valve. The shuttle valve directs the fluid in such a way that the hydraulic actuator is set either as closing or opening position. If the fluid is passes onto the opening shuttle valve, the piston in the cylinder moves towards closing of the ram. The same procedure is followed for opening of the ram, when fluid is directed onto the shuttle valve to open the ram. The surplus hydraulic fluid is vented into the reservoir.

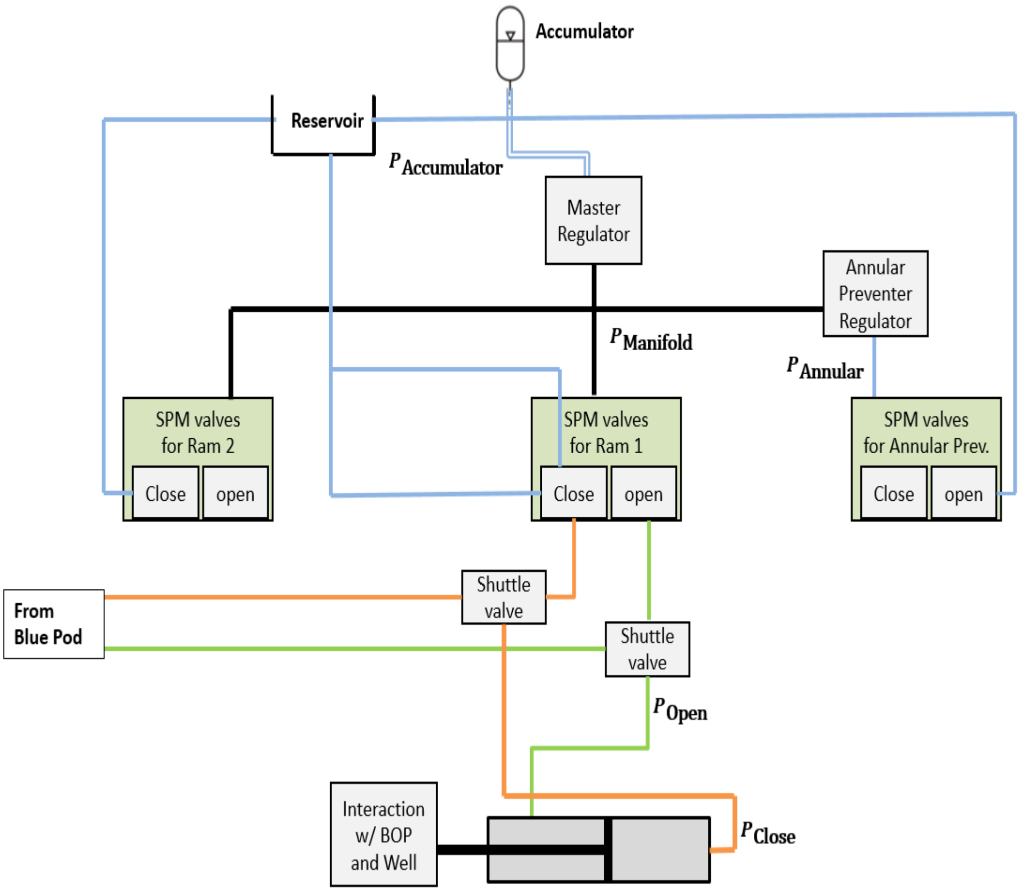

Figure 4-1: Hydraulic Circuit ram BOP

4.1 Base model

The modelling of the ram BOP hydraulic circuit has been performed in Matlab for a subsea BOP. The opening and closing operation has been divided into four different sections for the ease of modelling. Section A defines the flow from the reservoir to the “SPM close valve”. Section B defines the flow from the “SPM close valve” to the hydraulic actuator and includes the frictional losses experienced in the system. Section C defines the flow from the vent-line to another SPM valve which is set into opening position and includes the frictional losses experienced at the return-line. The section D defines the flow from the “SPM open valve” back to the reservoir. This is illustrated in figure 4-2.

Two different ODEs have been established for flow in pipe and hydraulic actuator. It should be noticed that the hydrostatic pressure is considered during calculations for all sections.

Therefore, the lengths of the different sections are an important parameter which needs to be defined beforehand. The static heights can be changed in the simulations, for the model to be suitable for subsea BOP or surface BOP. The lengths of the different sections used in this simulation are listed in table 4-1. The sections considered here have zero inclination and wall elasticity factors are considered.

Section	A	B	C	D
Length (m)	500	50	500	50
Pipe Diameter (m)	0.05	0.1	0.05	0.1

Table 4-1: Static heights for base model

The assumption taken in the model is that the $P_{manifold}$ provides a constant pressure supply of 1500 Psi. However, this is only possible when there is continuous volume refill

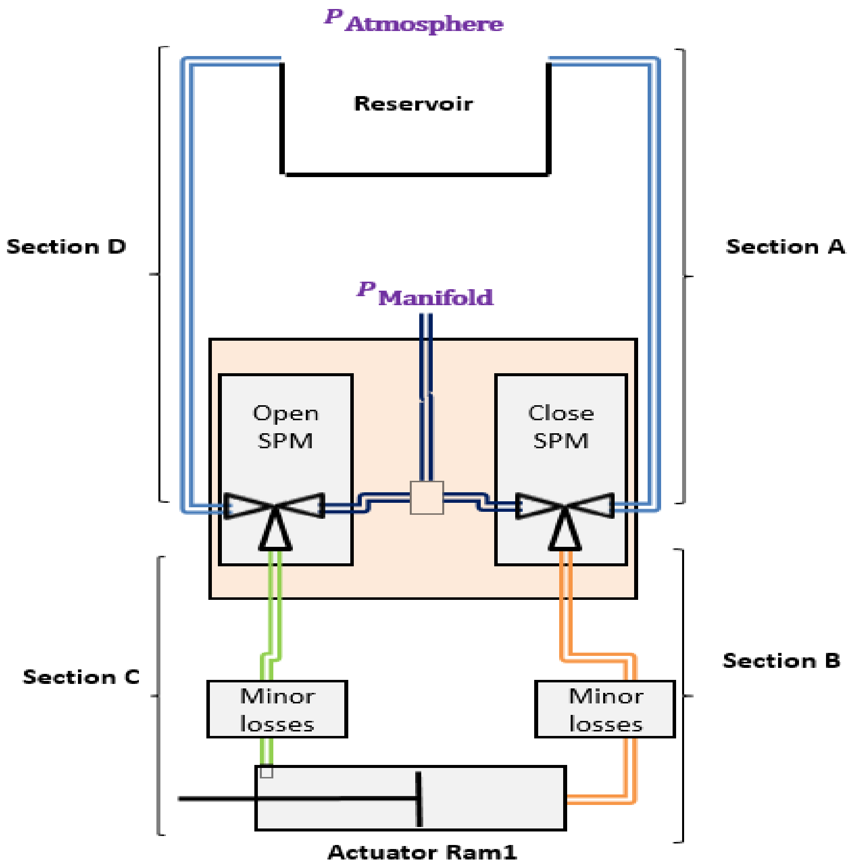

Figure 4-2: Base model

During the closing and opening operation of the hydraulic actuator, flow must be directed in respective direction as it can be seen in the figure 4-3. During the closing operation of the ram, the pressure support is provided by the manifold pressure. The is the pressure at the master regulator which is set as input to the model.

As it can be seen in the closing operation of the ram, the 3-way valve (SPM valve) is opened in such way that the fluid is directed to the closing of the cylinder. This pressurizes the hydraulic line between the SPM valve and the hydraulic actuator. As the ram movement starts, returning fluid from the hydraulic actuator is vented first into the open SPM valve and then back to the reservoir. The SPM valve is opened gradually to give a slow start to fluid flow due to the smaller size of the pipe.

The opening of the ram follows the same procedure as closing of the ram where the only difference becomes the direction of flow for the hydraulic fluid. During the opening of the ram, the $P_{manifold}$ provides the pressure support to the open SPM and the pressure at the return-line pushes the piston in the backward direction.

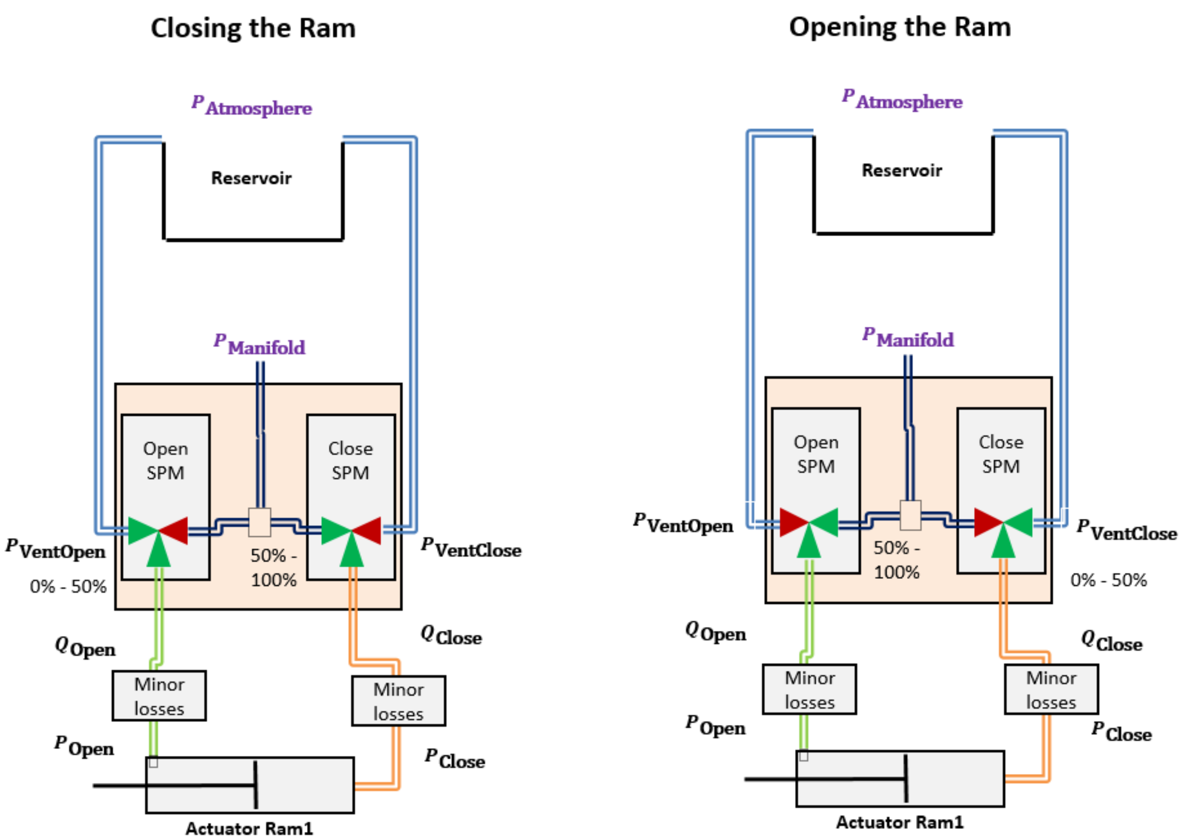

Figure 4-3: Base model for opening and closing of ram BOP

4.2 ODE for Hydraulic Actuator

The ODE for hydraulic actuator is derived using conservation of momentum and the motion of piston in a hydraulic cylinder (with shaft) is modelled from Newton's second law of motion. The hydraulic cylinder if considered as close vent, the momentum balance can be written as:

$$\frac{dv_{Li}}{dt} = \frac{1}{m_{Li}} \sum F \tag{55}$$

The different pressure forces acting on a hydraulic cylinder can be written as:

$$\frac{dv_{Li}}{dt} = \frac{1}{m_{Li}} . [(P_{open}.A_{piston-rod}) + P_{open}.(A_{open} - A_{piston-rod})] \tag{56}$$

The pressure equation proposed in equation (48) can be converted into a force balance equation for a hydraulic actuator. The sum of forces affecting the force balance in the hydraulic actuator as described in chapter 3.3. The equation (56) can be modified by the addition of all the forces affecting the mass balance i.e. cylinder friction, the friction from drag and the friction from well pressure. The new proposed equation becomes:

$$\frac{dv_{Li}}{dt} = \frac{1}{m_{Li}} . [(P_{open}.A_{piston-rod}) + P_{open}.(A_{open} - A_{piston-rod}) + F_{CF} + F_D + F_{Well}] \tag{57}$$

It is important to state that there should be set a condition in the model, where the ram BOP reaches the max stroke-length and doesn't retract backwards after meeting with the adjacent ram. In this model, this force is tackled using a very stiff spring constant force named as F_{Closed}

$$F_{Closed} = -\max(\frac{dx}{dt} - (cylinder\ length * 0.99) * Dummy\ spring\ constant) \tag{58}$$

The F_{Closed} uses a very high dummy spring constant to provide high retardation when the adjacent rams meet each other and do not retract backwards. The change in the position of the ram is denoted by $\frac{dx}{dt}$ which represent the velocity of the ram at a given time.

Another important force to be considered in a hydraulic cylinder is the coulomb damping. Coulomb damping represents the energy dissipation due to the sliding friction of piston. The coulomb law is associated with kinetic and static friction in a hydraulic cylinder and can be represented as:

$$F_{fclose} = -\frac{dx}{dt}.f_c - sign\left(\frac{dx}{dt}\right).f_s \tag{59}$$

where f_c represents the dynamic coulomb friction and f_s represents the static friction in the cylinder. The combined ODE for the hydraulic cylinder can thus be written as:

$$\frac{dv_{Li}}{dt} = \frac{1}{m_{Li}}.\left[\left(P_{open}.A_{piston-rod}\right) + P_{open}.\left(A_{open} - A_{piston-rod}\right) - F_{CF} - F_{Well} - F_{Closed} - F_{fclose} - F_D\right] \tag{60}$$

The equation proposed above simulates the working of a hydraulic cylinder in a ram BOP. Furthermore, different BOP designs have variable dimensions and thus need to be defined for each BOP design. The properties of the hydraulic cylinder for the BOP used in this model are presented in the table below:

	Value	Unit
Length Cylinder	0.5	m
Diameter cylinder	0.3	m
Diameter shaft	0.05	m
Piston Mass	10	kg

Table 4-2: Cylinder parameters

4.3 ODE for flow in pipe

The ODE for flow in control line is derived using conservation of mass for a circular pipe. The change in the flow rate is calculated using the conservation of momentum as explained in section 3.1.1 and can be stated as:

$$\frac{dQ_{Li}}{dt} = \frac{A_{CsCv}}{m_{Li}}\left[F_p + F_f + F_g\right] \tag{61}$$

The above equation can be expanded to:

$$\frac{dQ_{Li}}{dt} = \frac{A_{CsCv}}{\rho_{Li}V_{Li}}\left[A_{CsCv}\Delta P_{Li} + A_{SuCv}\tau_w - \rho_{Li}V_{Li}\, g\cos\theta\right] \tag{62}$$

where F_g represents the gravitational force affecting the fluid flow and the F_f is the friction caused by the wall shear stress for laminar flow. It should be noted that friction due to the wall shear stress in turbulent flow regime is also included in the ODE and the higher value is taken to be conservative in calculations. In addition to the forces mentioned above, the pressure losses described in chapter 3.2 need to be taken under consideration for precision of results. The equation (61) can be expanded by addition of pipe friction and other minor losses. The resulting equation can be written as:

$$\frac{dQ_{Li}}{dt} = \frac{A_{CsCv}}{m_{Li}}\left[F_p + F_f - F_g - F_{Minor} - F_{Pipefriction}\right] \tag{63}$$

The pressure calculation in the Matlab model are performed using conservation of mass principles as explained in section 3.1. The ODE for pressure calculation becomes:

$$\frac{dP_{Li}}{dt} = \frac{\beta}{\rho_0 V_{Li}}\left(q_{LiIn} - \rho_{Li}\, Q_{Li} - \rho_{Li}\frac{dV_{Li}}{dt}\right) \tag{64}$$

Equations (62) and (63) calculate the changes in the flowrate and pressure in comparison with the velocity and position of the ram BOP.

4.4 Model without a DP in borehole (scenario 1)

The pressure required at the major regulator to close the free moving ram BOP is tested using this model. The input for the model are listed in form of tables below. The table 4-3 represents the relevant parameters about the environment (W.C.Goins, 1983) and the fluid properties (O. d. services, 2018) i.e. the Density, viscosity and ambient temperature

Liquid		
Specific Gravity	0.854	g/cm^3
Temperature	293.15	K
Viscosity	17.7	cSt
Bulk modulus	1.83x10^9	N/m^2
Lengths		
Hydraulic Hose	31000	mm
Travel distance Ram	500	mm

Table 4-3: Environment and fluid properties

Table 4-4 consists of the minor loss co-efficient of different components that occur in the hydraulic circuit (Basniev et al., 2012; Munson, 2009).

#	Component	Number of Component	Loss factor	ID [mm]
1	Regulator	1		
2	TEE	3	0.9	1
3	Elbow	6	1.5	1
4	SPM valve	1	17	2
5	Shuttle valve	2	2	2
6	Union	2	0.08	1
7	Expansion	1	0.35	0.75
8	Filter	1		1

Table 4-4: Minor losses co-efficient

Table 4-5 consists of the different pipe and hose lengths (Haga, 2012) used for the pipe friction in the model. It should be noted that the friction losses are included only for the components between the master regulator and the ram BOP. The friction losses between the accumulator and the master regulator are not part of the calculations. A schematic diagram of the pipe lengths can be seen in figure 3-4.

#	Component	Length[mm]	ID[mm]	ε
1	Tee (1) - Elbow (1)	780	1	0.03
2	Elbow (1) - Elbow (2)	780	1	0.03
3	Additional Length elbows	750	1	0.03
4	Elbow (2) - Elbow (3)	800	1	0.03
5	Elbow(3) –SPM valve	1650	1	0.03
6	Shuttle valve-actuator	340	1	0.03
7	SPM valve- Shuttle valve	1650	1	0.03
8	Additional Bend	526	1	0.03
9	Hydraulic hose	31000	1	0.05

Table 4-5: P&ID dimensions

The ODE developed for the calculation of the velocity and the position of the ram BOP is as follows:

$$\frac{dv_{Li}}{dt} = \frac{1}{m_{Li}} \cdot [(P_{open} \cdot A_{piston-rod}) + P_{open} \cdot (A_{open} - A_{piston-rod}) - F_{Closed} - F_{fclose} - F_{CF} - F_D - F_{Well}] \quad (65)$$

4.5 Model with a DP in borehole (scenario 2)

The model used to test the closing time of a ram BOP when a drill pipe is present in the borehole resembles mostly to the model used for scenario 1. The only additional force that would affect the closing time of the shear ram BOP is the shear force required to cut the drill pipe. There are several drill pipes used in the industry which require different shear forces. The most common drill pipe used in the drilling operations is of type S-135. The mechanical properties of the S-135 pipe are given in the table below:

Material grade	S-135
Size	5 ½ inches
Cross-sectional area	5.828 inch^2
Tensile yield	91280 lbf-ft

Table 4-6: Metallurgical properties S-135 Drill pipe

The shear force calculations have been done using the OEM formula described in section 3.6.2. The highest value of shear force is taken to be conservative and used in calculations.

The ODE developed for the calculation of the velocity and the position of the ram BOP is as follows:

$$\frac{dv_{Li}}{dt} = \frac{1}{m_{Li}} . \left[\left(P_{open} . A_{piston-rod} \right) + P_{open} . \left(A_{open} - A_{piston-rod} \right) - F_{Closed} - F_{fclose} - F_{CF} - F_D - F_{Well} - F_{Shear} \right] (66)$$

5 Results

In this chapter, the results obtained from the simulation from two different models have been presented. The first scenario is when there is no DP present in the borehole and the second scenario represents when a DP is present in the borehole. It should be kept in mind that the model is designed in such a way that it can be customized easily for other ram types. The results are presented using graphs, generated by the simulation as can be seen in figure 5-1 and 5-2. A general description of the model outputs is described in this section.

The first graph shows the change in the flow rates against time, at different inlets and outlets of the base model. The flow rates change accordingly with any changes in the pressure.

- The “flow vent close” is the flowrate due to hydrostatic pressure of the liquid column between the reservoir and the SPM close valve.
- The “flow ram close” shows the pressure at the inlet of the hydraulic actuator.
- The “flow ram open” is the pressure at the return-line. The return-line pressure is received at the SPM open valve.
- The “flow vent open” is the flowrate due to hydrostatic pressure of the liquid column between the reservoir and the SPM open valve.

The second graph represents the pressure at SPM valves and the hydraulic actuator against time.

- “Pressure vent close” line represents the pressure at between the SPM close valve and the reservoir. This remains constant during closing of the ram as the pressure support is provided by the accumulators. Since the static height between the reservoir and the close SPM valve is set to 50 meters, the hydraulic column will have a pressure of 50 bar.
- “Pressure ram close” line is the pressure at the inlet of the hydraulic cylinder. This is the pressure $P_{manifold}$ minus the minor and major losses in the hydraulic line.
- “Pressure ram open” line is the pressure at the return-line or the outlet of the cylinder. This is the pressure $P_{manifold}$ in the pressure losses in the line and the frictional losses in the cylinder.
- “Pressure vent open” represents the pressure in the hydraulic line between the “open SPM valve” and the reservoir where the return fluid is added to the reservoir. The pressure value is the combination of hydrostatic pressure and the pressure input from

the return-line. This remains constant during opening of the ram as the pressure support is provided solely by the accumulators.

The third graph shows the position of the ram against time.

The fourth graph shows the velocity of the moving ram against time. The velocity of the ram is dependent upon the pressure difference as a bigger pressure difference would result in faster movement of the ram BOP.

The fifth graph shows the opening and closing of the SPM valves. The zero value of "open SPM valve" means that the return-line is connected to the vent-line. The value of "close SPM valve" is between 0.5 and 1, which means that the close-line is connected to the $P_{manifold.}$ The values between 0.5 to 1 show the opening percentage of "close SPM valve". It should be noted that the valve opening should be done in a gradual manner to mimic a slow start as the size of the pipe is relatively small.

5.1 Model without a DP in borehole (scenario 1)

The model which represents the closing of the ram when no DP is present in the well bore is called Scenario 1 in the present study. The time taken for scenario 1 can be interpreted from the simulation results presented in figure 5-1. The closing time of the ram BOP without a DP in the wellbore is around 7 seconds when the pressure at the master regulator is set at 1500 psi. It should be noted that all the frictional losses in the hydraulic circuit and the actuator remain same for both the scenarios. The results shown in the graph are for the closing operation of the ram.

The difference in the flowrates at the "ram close line" and "ram open line" can be seen in the first graph, which correspond with the increase in the pressure shown in the second graph. This is the pressure used to close the ram. A greater pressure increase can be seen where the difference in the flowrates of the "ram close line" and "ram open-line" is also high. This difference provides the necessary pressure for the ram BOP to move from starting point to maximum stroke-length of the cylinder. The change in the position of the ram can be seen in the third graph which corresponds to the velocity profile in the fourth graph.

The flowrate for the "flow vent close" represented in the first graph stays constant as there is no differential flowrate. This is confirmed by the pressure profile in the second graph as the pressure stays constant. The "flow vent close" represents the pressure column between the

“SPM close valve” and the reservoir. Since the pressure support during the closing operation of the ram BOP is provided by accumulators, the pressure in the “vent close-line” remains constant. Furthermore, the pressure difference between the “vent open line” and “ram open-line” can be seen moving synchronized. This can be explained by the fact that the difference between both lines is only due to difference in the hydrostatic pressure, which remains the same unless the static height of the reservoir is changed.

At approximately 7 seconds, the pressure decreases to almost zero and the change in the flowrate falls to zero simultaneously. At this point, the maximum stroke length is reached by the ram and no further movement can occur. The “pressure ram close” in the second graph can be seen to stay constant as pressure must be supplied to keep the ram at closing position.

It should be noticed in the fifth graph that the valve signal is opened gradually with the delay of half second. The fluctuations seen in the pressure and flowrate profile can be explained by the dynamic nature of modelling. The fluctuations can be seen to phase out due to the damping factor used in development of ODE for hydraulic cylinder. The slight variations in the velocity profile is due to the boundary condition at maximum stroke length. The boundary condition is set using an additional force F_{Closed} added in ODE for hydraulic cylinder, to create a restriction for rams when maximum stroke length is achieved.

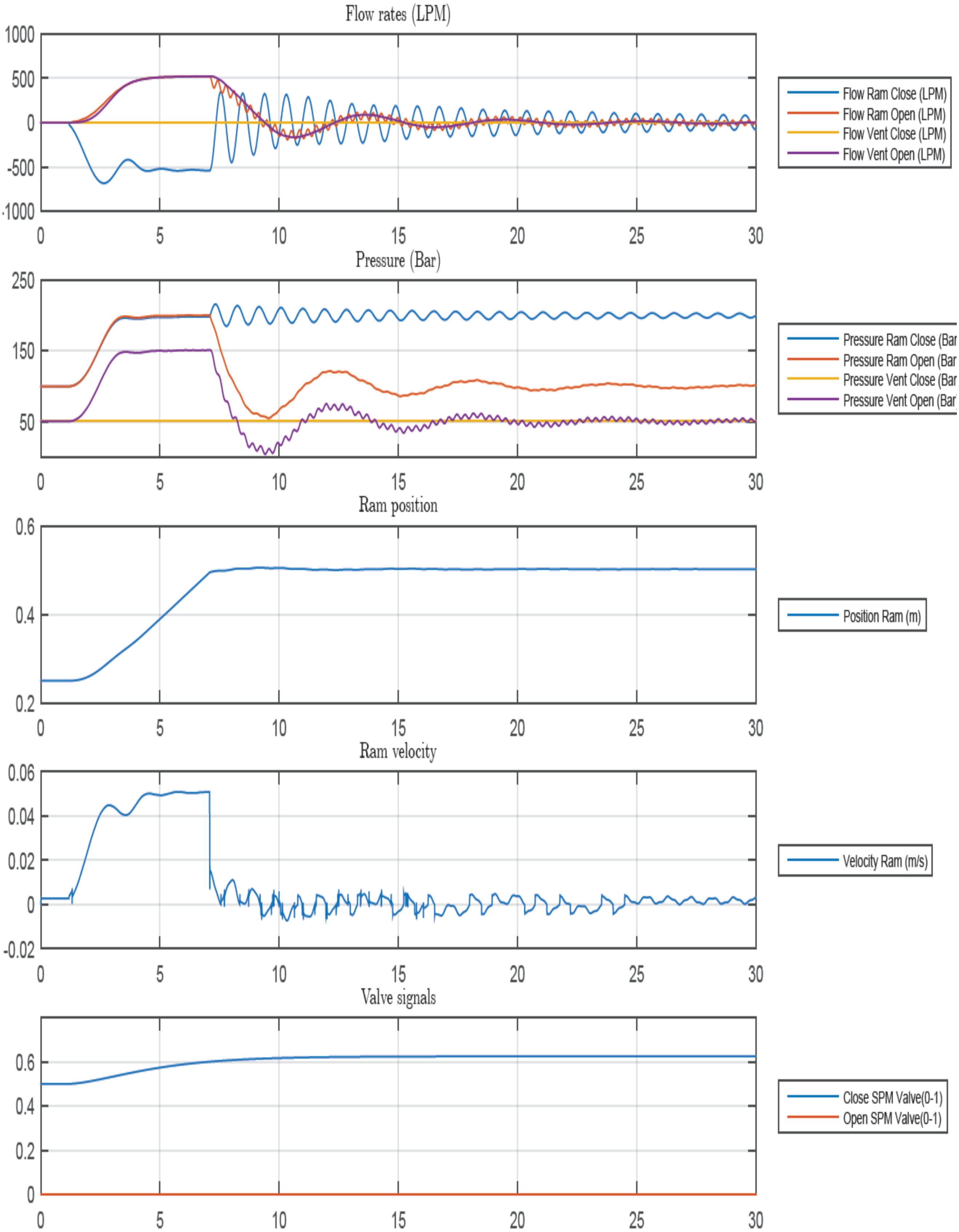

Figure 5-1: Simulation results for Scenario 1

5.2 Model with a shear DP (scenario 2)

The model which represents the closing of the ram when a DP is present in the well bore is called Scenario 2 in the present study. The time taken for closing of ram BOP for scenario 2 is displayed in figure 5-2. The closing time of the ram BOP without a DP in the wellbore is around 20.5 seconds when the pressure at the master regulator is set at 1500 psi.

The difference in the flowrates of the “flow ram close” and “flow ram open” can be seen in the first graph, which corresponds with the increase in the pressure represented in the second graph. The difference in the flow rate is relatively less for scenario 1 as compared to scenario 2. This is due to an additional shear force required to cut the drill pipe. A smaller pressure difference reduces the moving velocity of the ram BOP as seen in the fourth graph. The position of the ram against time can be seen to change at a very slow rate as the ram velocity for scenario 2 is very low.

The flowrate for the “flow vent close” as seen in the first graph remains constants and is justified by the constant “pressure vent close” in the second graph. The value of this pressure remains the same for both the models as there is no changes in the static height of the models. In addition, the pressure difference in the “ram open-line” and “vent open-line” stays constant as it the difference due to the hydrostatic pressure between the reservoir and the “open SPM valve”.

At approximately 20.5 seconds, the pressure decreases to almost zero and the change in the flowrate also falls to zero. At this point the adjacent rams meet each other. As it can be seen, an additional 13 seconds are required until maximum stroke-length is reached if scenario 2 is compared with scenario 1. The relative change in pressure can be seen by comparing both results. The pressure support in scenario 1 is greater than in scenario 2 and thus longer time is required for the ram to reach maximum stroke-length in scenario 2.

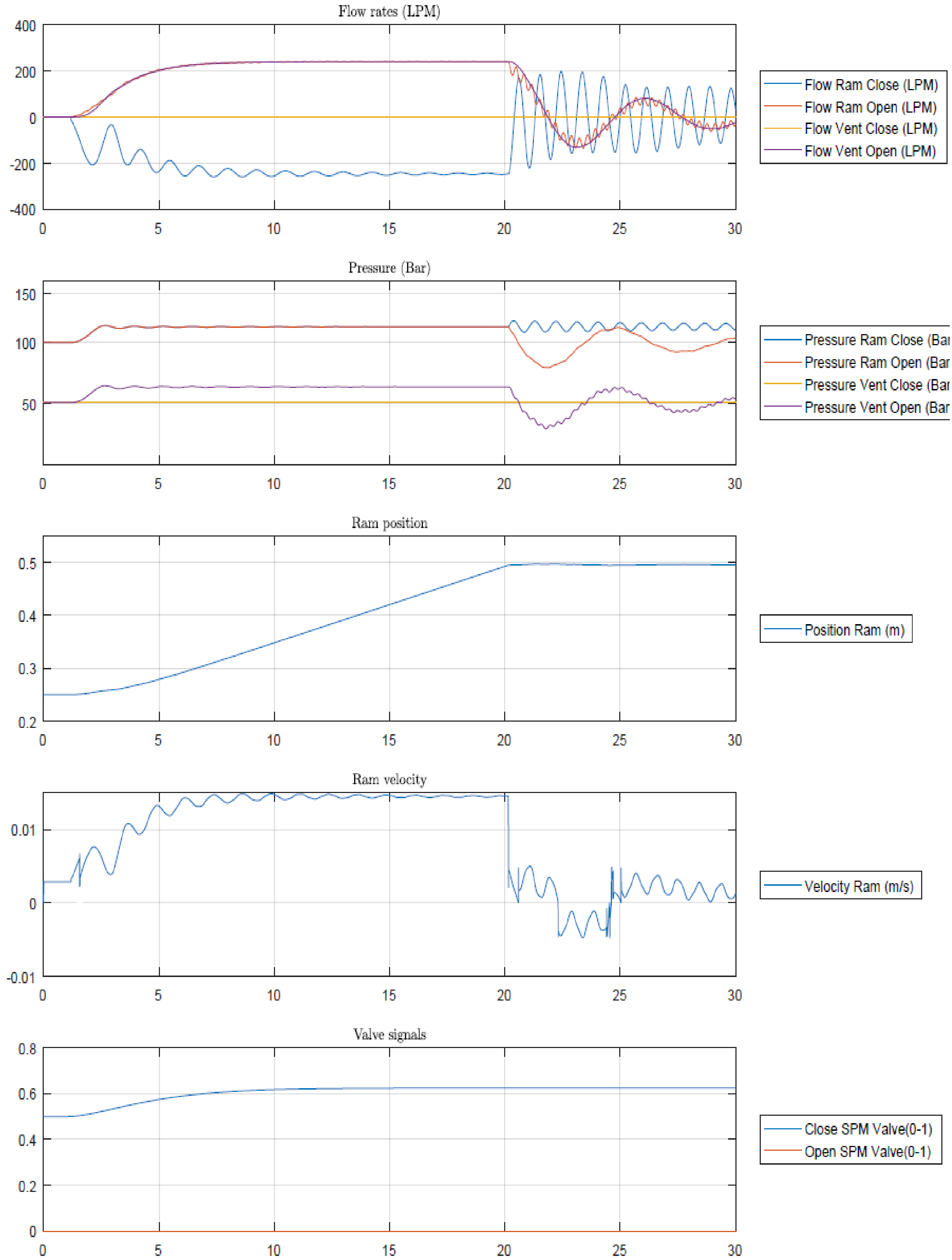

Figure 5-2: Simulation results for scenario 2

6 Discussion

This section covers the discussion of the results obtained from the simulation modelling.

6.1 Accuracy

The accuracy of the model for a ram BOP for scenario 1 is satisfactory and in good agreement with the published study (Haga, 2012) . The pressure and flowrate fluctuations described in figure 3-13 and 3-14 matches the results obtained for scenario 1. The accuracy depends mainly upon the relevant parameters of the BOP design that must be defined correctly. Note that when the pressure at the master regulator is set to be 1500 psi, it takes around 7.2 seconds to close the ram BOP.

The model defined in this thesis is solely developed for scenario 1. To further investigate the applicability of this model, it has also been tested for scenario 2. It is observed that model for scenario 2 has higher error bound. This is mainly due to additional uncertainty from several other parameters (scenario 2) that were not considered since the model is developed for scenario 1. These uncertainties related mostly to the geometry of the ram and the metallurgical properties of the drill pipe. The parameters that affect the accuracy of the model when a DP is present in the borehole are:

- Geometry of shear ram
- Travelling distance of the ram
- BOP opening ratio
- Yield strength of the drill pipe
- Buckling and shearing of DP
- OEM formula used for shear force calculation

The fluctuations in the pressure and flowrate obtained from the published study (plotted in figures 3-15 and 3-16) do not agree well with the determined results in scenario 2. This depicts that the model developed for scenario 1 is not applicable for scenario 2. One way to improve the model can be the addition of another boundary condition into the ODE for scenario 2. The condition must define that the ram movement should continue until it meets the DP. Shear force should also be integrated in the ODE.

The theory established in section 3.6.2 explains the pressure and flowrate profiles for scenario 2. This can be compared with the results for scenario 2 in figure 5-2 and following hypotheses can be made:

- At the point in the graph where the ram hits the DP in the wellbore, pressure should increase to a relatively high value. The pressure should build up until DP is sheared and then once the DP is sheared, pressure should fall to normal level. The pressure should again build up as the adjacent rams encounter each other.
- The flowrate difference should become very small when the rams hit the DP, as the pressure is used only to move the ram freely. The flowrate difference should rise again when adjacent rams meet each other, as high pressure is required to keep the rams in closed position.
- The ram velocity will stay constant until the DP is hit and then fall to zero as the ram stops when contact with DP is made. The velocity should stay zero until the DP is sheared and rise slightly as the ram moves further. The velocity should go back to zero again when the adjacent rams encounter each other.

Another complexity in the dynamic model which must be kept in mind is the frictional and pressure losses that occur in the hydraulic line in scenario 2. Once the ram movement stops as the contact with the DP is made, the pressure and frictional losses should immediately decrease. And if the ram movement stops, the frictional forces become zero as friction is due to the piston movement and fluid flow in the hydraulic circuit. This means that pressure support to the ram will be the same as at the $P_{manifold}$. Therefore, the model developed for scenario 1 requires modifications for it to be applicable for scenario 2.

6.2 Calibration and sensitivity

The parameters which are highly sensitive to the closing time of the BOP are as following:

- Pipe friction
- The cylinder friction
- DP shear force

The shear force required to cut the DP has the highest impact on the closing time. If the model defined for scenario 1 is tested for scenario 2, the closing time increase by more than 200%. Different OEM formulas use different methods (Tulimilli et al., 2014) for calculation of the shear force to cut the drill pipe which has a direct effect on the closing time in scenario 2. Hence, there is uncertainty in the shear force required to cut drill pipe. This hypothesis matches with the studies performed by (Tekin et al., 2015) and (W. E. Services, 2004)

The parameters that have minor effect on the BOP closing time are as following:

- Relative roughness of the internal surface of the pipe walls
- The minor losses in valves
- The drag force on ram motion
- The friction loss between cylinder piston and the ram interface
- The fluid viscosity

6.3 Applicability

The model developed for scenario 1 is fully applicable to calculate the closing time of ram BOP. The pressure at the master regulator set as 1500 psi, is enough to close the ram BOP in time under 30 seconds according to API and NORSOK standards. The model for scenario 1 can be used same for pipe rams, blind rams and VBR rams with slight modifications.

The model is equally applicable for surface BOPs and subsea BOPs. The static height in the model can be modified to match the hydrostatic pressure head at the installed location of the BOP.

6.4 Suggested improvements and challenges

Although the proposed model for scenario 1 has satisfactory accuracy, a few improvements can be done to the model:

- Integration of the pressure variation at the master regulator with the discharged volumes of the accumulator. This helps improve the model by including the overall pressure and frictional losses for the complete circuit. The pressure at the master regulator is assumed to have a constant value, but, the volume refills from the accumulator are necessary to maintain this pressure. This results in fluctuations at the $P_{manifold}$.
- Tuning the model to include the additional losses neglected in the calculations.
- Additional boundary condition to limit stroke length when the ram hits the DP for scenario 2. This way the model for scenario 1 can be easily modified to be used for scenario 2.

- Improvement of the model used for kinetic and viscous friction in the hydraulic cylinder as described in figure 3-6.

The challenges faced during establishment of the ODEs and modelling are listed below:

- The pressure and frictional losses in the hydraulic line depend upon the differential pressure at any time, which varies with the ram movement. This requires calculations on very short time steps. Thus, significant computational resources are required to compute results.
- Acquisition of dimensional data for BOP design
- The dependence of well pressure on density of the mud
- The friction between rams and annulus wall is difficult to calculate, as the nature of friction is neither wet nor dry.
- BOP manufacturers keep the hydraulic circuit P&ID diagrams confidential and data acquisition requires permission.

6.5 Future work and findings:

The recommendations for further work would be:

- Integration of accumulator volume discharge calculations with the pressure at the master regulator
- Integration of emergency accumulator bank at $P_{manifold}$ to reduce friction losses
- Expansion of the model for rams of different geometry

The most important lessons learnt which can aid the future work are:

- The dimensional data for specific BOP design is very difficult to acquire and pre-literature information must be consulted
- The frictional losses in the cylinder and the BOP interface are major contributors to the overall losses
- The static heights and the location of the BOP must be pre-defined to include the hydrostatic pressures in the model.
- Simple assumptions must be considered while developing the programming code to avoid numerical errors

7 Conclusion

The closing time of the ram BOP is safety critical in the oil and gas industry. A dynamic model is proposed to calculate the closing time of ram BOP when no DP is present in the well bore (scenario 1). This has been done by establishing ordinary differential equations for hydraulic circuit and the actuator. The following conclusions can be drawn from this study.

- The model in the present study is developed in accordance to the boundary conditions of scenario 1 and the results obtained show a good agreement with the published studies.
- The present model exhibits few discrepancies for scenario 2 when compared with the published results.
- This model for scenario 1 can be used for pipe rams, blind rams and VBR rams with slight modifications.
- The accuracy depends mainly upon the relevant parameters of the BOP design that must be defined correctly.
- When the pressure at the master regulator is set to be 1500 psi, it takes around 7.2 seconds to close the ram BOP. The calculated closing time for scenario 1 is shorter than API and NORSOK standards of 30 seconds.

8 References

1. API. (2012). API Std 53 - Blowout Prevention Equipment Systems for Drilling Wells. (4th edition).
2. API. (July,2016). "Specifications for Control systems for drilling well control equipment and control systems for diverter equipment," *API 16D*.
3. Basniev, K. S., Dmitriev, N. M., Chilingar, G. V., Gorfunkle, M., Mohammed Nejad, A. G., & Basniev, K. S. (2012). *Mechanics of Fluid Flow*. Somerset, UNITED STATES: Wiley.
4. Baugh, B. F., Vozniak, J., & Schmidt, N. (2011). *Post-Macondo BOP Safety Upgrades*. Paper presented at the OTC Brasil, Rio de Janeiro, Brazil.
5. Chapple, P. J. (2004). Hydraulic stiffness. 1-10.
6. Davorian, M. (2013). Well Control Equipment.
7. De Wit, C. C., Olsson, H., Astrom, K. J., & Lischinsky, P. (1995). A new model for control of systems with friction. *IEEE Transactions on Automatic Control, 40*(3), 419-425.
8. Haga, Ø. A. (2012). *Modelling safety critical BOP closing times for arbitrary BOP control systems.* UiS, Stavanger.
9. Hager, W. H. (2010). *Wastewater Hydraulics: Theory and practice* (Second ed.).
10. Januarilham, Y. (2012). Analysis of Component Criticality in the Blowout Preventer.
11. Jitendra Yadav, D. G. A. (2015). MODELING AND SIMULATION OF HYDRAULIC ACTUATOR WITH VISCOUS FRICTION. *International Journal of Advanced Technology in Engineering and Science, 3*(1), 8.
12. L. Hružík, M. V., and A. Bureček. (2013). Evaluation of Bulk Modulus of Oil System with Hydraulic Line. *EPJ Web of Conferences*. doi:10.1051/epjconf/20134501041
13. Liu, Z., Guo, J., Guo, T., Cole, T., & Sridhar, N. (2017). Force prediction in blow-out preventer shearing of drill pipes. *Engineering Failure Analysis, 74*, 159-171.
14. Munson, Y., Okiishi,Hurbsch. (2009). *Fundamentals of Fluid Mechanics* (sixth ed.).
15. Muvengei, M., & Kihiu, J. (2011). Bond graph modeling of inter-actuator interactions in a multi-cylinder hydraulic system. *Int. J. Aerosp. Mech. Eng, 5*(3), 147-156.
16. Muvengei, M., & Kihiu, J. (2011). Bond graph modeling of inter-actuator interactions in a multi-cylinder hydraulic system. *Int. J. Aerosp. Mech. Eng, 5*(3), 147-156.
17. NORSOK. (Dec 2012). NORSOK standards D-001. *third*.
18. Pantokratoras, A. (2016). STEADY FLOW OF A POWER-LAW NON-NEWTONIAN FLUID ACROSS AN UNCONFINED SQUARE CYLINDER. *Journal of Applied Mechanics and Technical Physics, 57*(2), 264-274. doi:DOI: 10.1134/S0021894416020097
19. Parr, E. A. (2011). *Hydraulics and Pneumatics: A Technician's and Engineer's Guide*: Butterworth Heinemann.
20. Phillip P. Brown , D. F. L. (2003). Sphere Drag and Settling Velocity Revisited. *JOURNAL OF ENVIRONMENTAL ENGINEERING © ASCE*. doi:10.1061/~ASCE!0733-9372~2003!129:3~222!
21. Rennels, D. C., & Hudson, H. M. (2012). *Pipe Flow : A Practical and Comprehensive Guide*. Hoboken, UNITED STATES: John Wiley & Sons, Incorporated.
22. Schlumberger. (2018). Schlumberger oilfield glossary: Annular bop. Retrieved from http://www.glossary.oilfield.slb.com/Terms/a/annular_bop.aspx
23. Schlumberger. (2018). Schlumberger oilfield glossary: ram blowout preventer http://www.glossary.oilfield.slb.com/Terms/r/ram_blowout_preventer.aspx
24. services, O. d. (2018). Technical Specification S-135.
25. Services, W. E. (2004). *Shear Ram Capability Study*. Retrieved from

26. SRL, E. I. (2016). Hydropneumatic Bladder Accumulators.
27. Standard.no. (2012). API Std 53 - Blowout Prevention Equipment Systems for Drilling Wells. (4th edition).
28. Tekin, A., Choi, C., Altan, T., & Adin, H. (2015). *Estimation of shear force for blind shear ram blowout preventers* (Vol. 1).
29. Tran, X. B., & Yanada, H. (2013). Dynamic Friction Behaviors of Pneumatic Cylinders. *Intelligent Control and Automation, Vol.04No.02*, 11. doi:10.4236/ica.2013.42022
30. Tulimilli, B. R., Naik, P., Chakraborty, A., Sawant, S., Whooley, A., & Weiss, R. (2014). Experimental and Numerical Investigation of BOP Shear Ram Performance. (45387), V01BT01A057. doi:10.1115/OMAE2014-24714
31. Varco, N. O. (2010). User's Manual LWS ram blowout preventer. 30-40.
32. W.C.Goins, R. S. (1983). *Blowout Prevention* (Second ed.).
33. Wit, C. C. d., Olsson, H., Astrom, K. J., & Lischinsky, P. (1995). A new model for control of systems with friction. *IEEE Transactions on Automatic Control, 40*(3), 419-425. doi:10.1109/9.376053
34. Yadav, J., & Agnihotri, G. MODELING AND SIMULATION OF HYDRAULIC ACTUATOR WITH VISCOUS FRICTION.

9 Appendix

9.1 Technical Specifications

Drill Pipe Specifications:

TECHNICAL SPECIFICATIONS

5-1/2" OD Drill Pipe, S-135, 5-1/2" FH. Conn's.		**New**	**Premium** 80% Remaining Body Wall
DESCRIPTION			
Type		IEU, 21,90 #	
Range		3	
Convensional=welded T-J. / Integral=Monoblock		Convensional	
TUBE DATA			
Material grade		S-135	
Internal plastic coating		TK-34P	
Tube body OD x ID	inch	5,500 4,778	5,355 4,778
Wall thickness, nominal	inch	0,361	0,289
Cross Sectional Area	$inch^2$	5,828	4,592
Polar Sectional Modulus	$inch^3$	14,062	11,042
Tensile yield pipe	lbf	786 800	619 900
	kN	*3 500*	*2 757*
Torsional yield pipe	lbf-ft	91 280	71 680
	kNm	*123,8*	*97,2*
80% Torsional Yield	lbf-ft	73 024	57 344
	kNm	*99,0*	*77,7*
CONNECTIONS DATA			
Connection type		5-1/2" FH. Standard	
Material grade		120 ksi	
Hardbanding		Arnco-300XT	
OD x ID	inch	7,250 3,500	6,719 3,500
B.S.R.	x : 1	2,25	1,60
Tensile yield tooljoint	lbf	1 598 000	1 036 300
	kN	*7 108*	*4 610*
Torsional yield tooljoint	lbf-ft	72 480	46 090
	kNm	*98,3*	*62,5*
Make up torque (Max.)	lbf-ft	43 500	27 700
	kNm	*59,0*	*37,6*
OPERATIONAL DATA			
Tool-joint/Drill-pipe torsional ratio		0,79	1,01
Drift diameter	inch	3,375	
Type of elevator shoulder		18°	
Burst pressure	psi	15 500	14 200
	Mpa	*107*	*98*
Collapse pressure	psi	12 700	7 500
	MPa	*88*	*52*
Adjusted weight	lbs/ft	23,6	
	kg/mtr	*35,1*	
Approx weight each joint	lbs	1 079,7	
	kg	*489,7*	
Capacity	gal/ft	0,89	
	ltr/mtr	*11,08*	
Open end displacement	gal/ft	0,36	
	ltr/mtr	*4,47*	
Closed end displacement	gal/ft	1,25	
	ltr/mtr	*15,55*	
Built In Length (shoulder to shoulder)	ft	45,75	
	mtr.	*13,94*	

Calculated using nominal OD & ID. Safety & Dope friction factor used: 1.0

Figure 9-1: Drill pipe specifications

Fluid Properties:

Property	OB200	Test Method
Viscosity (cSt)		ASTM D445 IP71 ISO 3104
@ -20 °C	137.8	
@ 5 °C	32.8	
@ 20 °C	17.7	
@ 40 °C	9.0	
@ 60 °C	5.4	
@ 80 °C	3.64	
Pour Point	<-30 °C	IP15
Specific Gravity (g cm^{-3})		
@ 5°C	0.863	IP365
@ 20 °C	0.854	
@ 40 °C	0.829	
@ 60 °C	0.822	
@ 80 °C	0.815	
Appearance	Clear, yellow liquid	
Flash Point /°C	>140 °C	ASTM D92 / IP36
Upper Temperature Stability	200 °C	
Cleanliness Level (Minimum)	17/14/12	ISO 4406
	NAS 6	NAS 1638
	6B/6C/6D/6E/6F	SAE AS4059
Shell 4 Ball	Mean Wear Scar Diameter 0.519 mm	IP239/01 1 hour duration, 1475rpm rotation, 30 kgf load
Solubility in Water	Insoluble	
Solubility in Mineral / Crude Oil	Soluble	
Coefficient of Thermal Expansion m^3/m^3 °C	0.00064	
Bulk Modulus N/m^2 (x10^9) (Estimated)	1.83	

Figure 9-2 Hydraulic fluid specifications

Loss-Coefficients:

Component	K_L
a. Elbows	
Regular 90°, flanged	0.3
Regular 90°, threaded	1.5
Long radius 90°, flanged	0.2
Long radius 90°, threaded	0.7
Long radius 45°, flanged	0.2
Regular 45°, threaded	0.4
b. 180° return bends	
180° return bend, flanged	0.2
180° return bend, threaded	1.5
c. Tees	
Line flow, flanged	0.2
Line flow, threaded	0.9
Branch flow, flanged	1.0
Branch flow, threaded	2.0
d. Union, threaded	0.08
***e. Valves**	
Globe, fully open	10
Angle, fully open	2
Gate, fully open	0.15
Gate, $\frac{1}{4}$ closed	0.26
Gate, $\frac{1}{2}$ closed	2.1
Gate, $\frac{3}{4}$ closed	17
Swing check, forward flow	2
Swing check, backward flow	∞
Ball valve, fully open	0.05
Ball valve, $\frac{1}{3}$ closed	5.5
Ball valve, $\frac{2}{3}$ closed	210

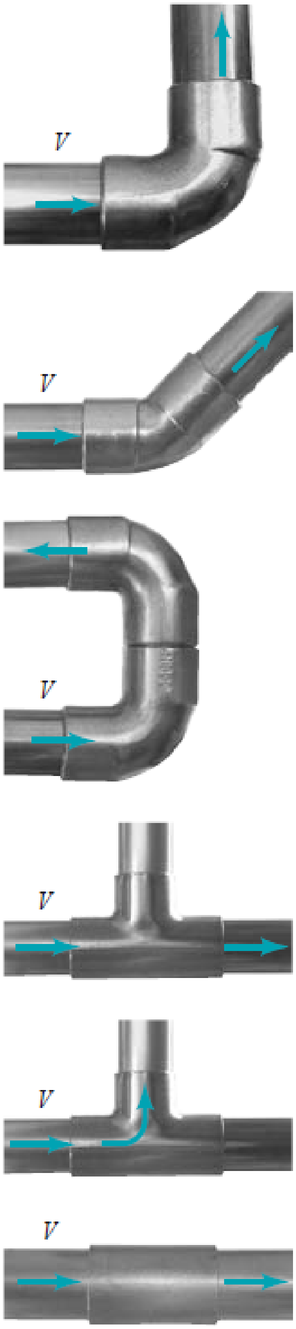

Figure 9-3 Loss Coefficients

Symbol Legend:

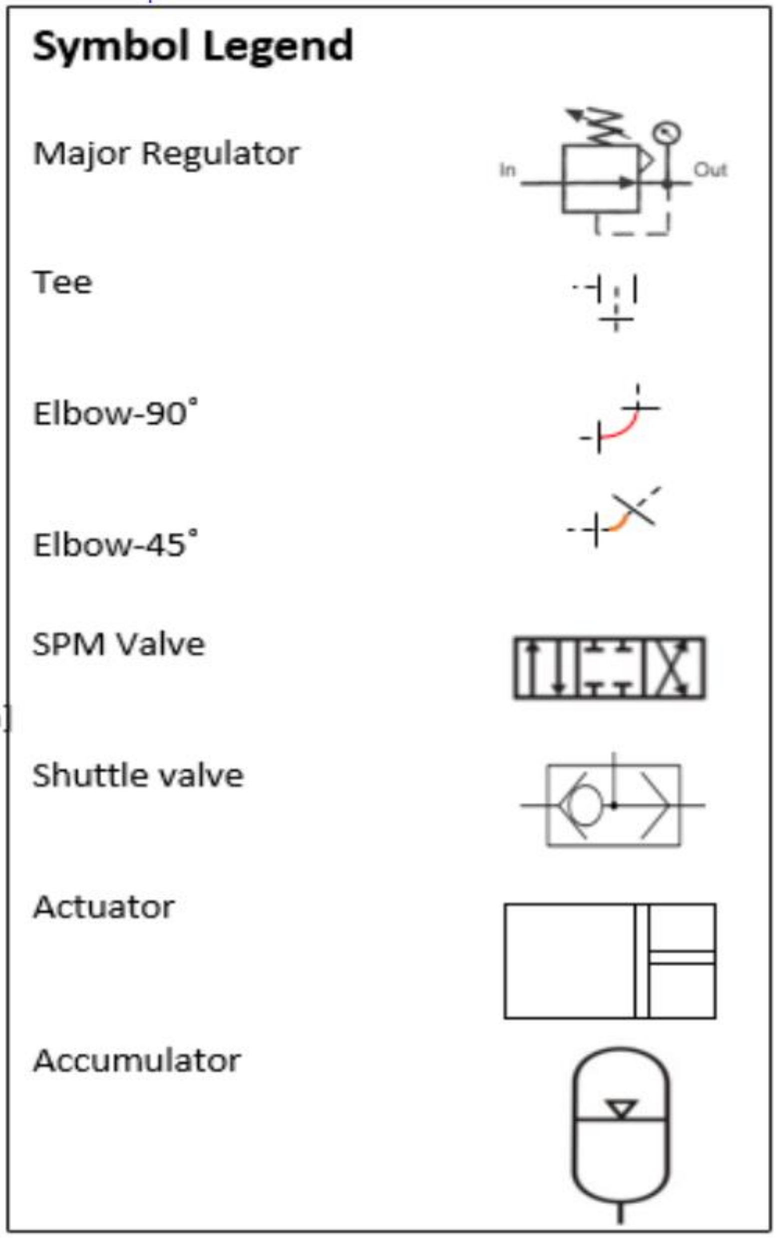

Figure 9-4 Symbol Legend

9.2 List of figures

9.3 List of tables

9.4 Notations

ΔP	Pressure difference	(pa)	V_s	Stribeck velocity
d	Inner diameter	(m)	K_x	Flow factor component
F	Force	(N)	n	Polytrophic index
g	Gravity	(m/s^2)	A_{CsCv}	Cross-sectional area pipe
h	Static height	(m)	A_{SsCv}	Surface area pipe
V	Volume	(m^3)	CV	Closed vent
Q	Volume flow rate	(m^3/s)	τ_w	Wall shear stress
dv/dt	Acceleration of ram	(m^2/s)	β	Bulk modulus
Dx/dt	Velocity of ram	(m/s)	μ	viscosity
Re	Reynolds number		ρ	Density
F_s	Static friction		σ_0	Average stiffness co-efficient
F_c	Coulomb friction		σ_1	Micro-damping co-efficient
C_D	Drag co-efficient		σ_2	Viscous friction co-efficient

Unit Conversion factors:

1 Psi	6895 Pascal	1 bar	100 000 Pa
1 Inch	0.0254 meter	1 cSt	10^{-6} m^2/s
1 Litre	0.001 m^3	1 $lb./ft^3$	16.02 kg/m^3
1 US Gallon	0.003785 m^3		

Abbreviations:

BOP	Blowout Preventer	CV	Closed vent
API	American Petroleum institution	DP	Drill pipe
ID	Internal diameter	LF	Loss factor
P&ID	Piping and instrumentation	ODE	Ordinary differential equation
NORSOK	Norsk Sokkels Konkurranseposisjon	SPM	Sub-plate Mounted Valves
OEM	Original equipment manufacturers	VBR	Variable bore rams