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Master of Science Thesis
Modelling safety critical BOP closing times for arbitrary BOP control systems

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June 3, 2012

Abstract

There are presently no recommended practises, defined by standards, for closing time calculation of Blow Out Preventers (BOPs), when designing BOP control systems.

As the closing time is a safety critical design parameter for BOPs and BOP control systems, this paper proposes a model for calculating this value. A guideline is included in order to recreate an equivalent model, applicable for most BOP control system designs.

Models are constructed and compared to an equivalent prototype system, comparing closing times and pressure trends throughout the closing function. The model assumes that the BOP control system has hydraulic pressure supplied from an accumulator bank, where the interfacing BOP is a blind/blind shear ram. The models consider closing operations for both situations of drill pipe (DP) being either present or not in the bore.

The modelling is done primarily using simple hydro mechanics and thermal mechanics in conjunction with API16D. All data sources, as far as practical, are published and per-reviewed literature.

Results from the model are promising for BOP closing times without DP in the bore. Inaccuracies were found when comparing to a BOP with DP present in bore.

However the model does require some preliminary information about the BOP in order to give accurate predictions. In addition the model is not applicable for systems using rotating machinery as pressure source or where there are several independent pressure sources.

Preface

Objectives of the project are as follows

- Model and calibrate model of a BOP control system, interfacing a blind/blind shear ram BOP.
- Describe a modelling guideline for modelling closing times for BOPs interfacing an arbitrary BOP control system.

Due to the lack of guidelines, there has been a practice in the industry to design according to standards considering pressure and volumetric accumulator constraints. After the system is constructed, the performance of the BOP control unit is verified.

The paper contains a model of a BOP control system interfacing a blind/ blind shear BOP. The BOP is modelled both with and without a DP present in the bore. The model is compared to a prototype of an equivalent system and calibrated in order to fit prototype results. Following this, there is a description of how the modelling process may be repeated, in order to model other system configurations.

Øystein Angelskår Haga __________________________
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Notation
α: Angle [^{o}]
ΔP: Pressure drop [Pa]
ρ: Density [kg/m^3]
ϵ: Roughness [m]
μ: Dynamic viscosity [Pa s]
ν: Kinematic viscosity [m^2/s]
γ: Adiabatic index [$-$]
ξ: Loss factor [$-$]
ψ: Specific gravity [$-$]
A: Area [m^2]
A_v: Valve orifice area [m^2]
b: Constant [$-$]
BV: Accumulator bottle volume [m^3]
C: Constant [$-$]
C_v: Coefficient of velocity [$-$]
C_q: Orifice flow coefficient [$-$]
d: Inner diameter [m]
F: Force [N]
f: Friction factor [$-$]
FVR: Functional volume requirement [m^3]
g: Gravity [m/s^2]
h: Static pressure height [m]
k: Resistance coefficient [$-$]
k': Function of diffuser angle [°]
K_v: Flow factor [$-$] L: Length [m]
n: Polytrophic index [$-$]
P: Pressure [Pa]
Q: Volume flow rate [m^3/s]
Q_m: Actual flow rate [m^3/s]
r: Radius of bend [m]
Re: Reynolds number [$-$]
t: Time [s]
V: Volume [m^3]
$\dot{V}$: Volumetric flow rate [m^3/h]
v: Velocity [m/s]
v_j: Jet flow velocity [m/s]
VE: Volumetric efficiency [$-$]
x: X-axis value [$-$]
y: Y-axis value [$-$]

Abbreviations
API: American Petroleum Institute
BOP: Blow Out Preventer
DP: Drill Pipe
FT: Flow Transmitter
FV: Functional Volume
FVR: Functional Volume Required
ID: Internal Diameter
LF: Loss Factor
NORSOK: Norsk Sokkels Konkuranseposisjon
PLC: Programmable Logic Controller
P&ID: Piping and Instrumentation Diagram/Drawing
PT: Pressure Transmitter

Unit conversions used
1 Inch = 0.0254 Meter
1 psi = 6895 Pascal
1 US gallon = 0.003785 m^3
1 l = 0.001 m^3
1 bar = 100 000 Pa
1 cSt = 10^{-6} m^2/s
1 lbm/ft^3 = 16.02 kg/m^3
1 BTU/lbm R = 0.0002388 J/kg K

Contents

1 Introduction

The models described in this paper are based on fluid mechanics, hydraulics, thermal mechanics, solid mechanics and measurements. Several assumptions are made in order to simplify the model, as well as to conform to the requirements set in place by industry standards.

The fundamental building block of the model is the Bernoulli equation. From this the model is expanded as the system to be modelled increases in detail, with additions such as pressure losses, hydrostatic or kinematic pressure contributions.

With the model, the closing (or opening) time of a Blow Out Preventer (BOP) may be estimated for a particular BOP control system configuration. This is achieved by splitting the closing (or opening) operation into many small pressure[1] intervals (this is often referred to as *pressure stages* in this paper). For each interval the volumetric flow rate is found from the equilibrium of the Bernoulli equation. From the flow rate the time to complete each interval is calculated. These intervals are totalled to obtain the full completion time.

The remainder of the introduction will describe most important facts about BOPs and BOP control systems. In addition a brief summary of the requirements of the BOP control system is given with regard to standards. Following the introduction is a section with a short "state of the art" description, as well as a larger section describing the theory that has been applied within the modelling. After the introduction there is a section about the models, calibration, prototype and results obtained. Then a section dedicated to the modelling process, describing how to set up an equivalent model. Finally there will be a discussion followed by conclusions.

1.1 BOP

In order to create a double barrier between hydrocarbons and the environment, there arise several occasions where BOPs are required to be utilised. A BOP is a large device that is primarily used to close and seal a well bore in the event of an un-controlled flow, for example if a "kick" occurs.

BOPs are usually stacked on top of each other when utilised in the field. These are known as BOP stacks. The arrangement of BOPs is decided by the rated working pressures in the well [24, p. 6]. Dependent on what work is to be done, and the solution used for the well, the BOP stack may be positioned either on the wellhead or on the X-mass tree.

There are many BOP designs available, but these are often divided into two main categories, the annular preventer and the ram BOP.

The annular preventer BOP seals the bore with large polymer rings that are reinforced with steel. When activated, the polymer ring is forced into

[1] Changes in accumulator pressure corresponding to a volume discharged.

the bore of the BOP by hydraulic pressure, enabling it to seal around any pipe sizes or an empty bore.

There are several types of ram BOPs in use, mainly fixed pipe, variable bore, blind, blind/blind shear and casing rams.
The fixed pipe ram has specific bore diameter that, when closed, seals around a specific pipe size preventing flow around pipe.
The variable bore pipe ram is similar to the fixed pipe ram, but differs in that there is an elastic packing in the bore. This packing allows the variable bore ram to close around a larger range of pipe sizes.
Blind rams are designed to seal the BOP bore when there is no pipe inside. Blind/blind shear rams are designed to seal the BOP bore even if there is a DP inside. This is made possible by fitting rams with cutting edges, thus allowing the shearing of the pipe before closing and then sealing the bore. Primary focus of this paper will be on the blind/blind shear rams, which is described in the models (both with and without DP in bore) and prototype sections.
Casing shear rams are similar to the Blind/blind shear rams, but are designed to shear casings in the BOP bore. However these rams may not be designed to seal the well bore from fluid flow [25, p. 6].

1.2 BOP control system

The BOP control system[2] supplies the BOP stack with hydraulic power during both normal and emergency operation. The primary components in a BOP control system are: accumulators, pumps, control valves and reservoir [24, p. 23].

Since the BOP control system is important for the operational safety, it is designed with independent components for redundancy, where a minimum of two independent hydraulic pressure control circuits shall be provided [25, p. 16]. With redundancy, no single failure mode of any one component would cause failure of the whole system. The BOP control system shall also be located in a safe area such that its ease of operation, in case of an emergency is not compromised [24, p. 29].

Standard requirements for BOP control systems primarily focus on being able to supply a minimum amount of liquid volume, with a specific minimum pressure, within a minimum time constraint [25, 27, 5, 24].

During normal operation the primary hydraulic power source for the BOP stack is supplied by the BOP control system accumulators. The pumps' primary purpose is to recharge the accumulators. Although the pumps are not the primary pressure source during normal operations, they are still required to be able to drive the BOP functions should the accumulators become non-operational. Note that BOP control system pumps are outside

[2]Sometimes termed Koomey, a trademark owned by NOV

the scope of this paper.

In order to control the BOP, the BOP controller has a series of pneumatic piloted hydraulic valves that may be manually overridden. When hydraulic energy is required, valves in the BOP control system open so that hydraulic fluid and pressure is discharged from accumulators to the BOP stack, activating required functions.

The hydraulic fluid, usually a mixture of water and glycol, is circulated in an open system with a closed loop, where the used hydraulic fluid is circulated back to the hydraulic fluid storage tank.

1.3 Standards

In order to assure a minimum level of safety with regards to operations and design, there are developed standards. However the standards are not intended to inhibit the development of technology or equipment improvements [24, p. iii].

Requirements for the BOP accumulator response time differ slightly between the standards used in this paper[3].

The response time of a BOP control system shall according to *API* standards be within:
A surface BOP should be capable of closing each ram BOP and annular preventer within 30 seconds from activation to completion. For annular preventers of size 18 3/4 inches nominal bore and above should not exceed 45 seconds [25, 24, p. 20, p. 27].

The response time of a BOP control system shall be according to *NORSOK* standards be within:
The response time refers to the time elapsed from function activation until BOP function is in closed position [27, 5, p. 33,p. 31-32]. Maximum response time on a surface BOP is 30 seconds[27, 5, p. 33,p. 31-32], whilst for annular preventers exceeding 20 inches a 45 second response time is acceptable[27, p. 33]. Stripper ram maximum response time for snubbing is 5 seconds [5, p. 31-32].

There are also recommendations to the accumulator capacity in the system. As the volumetric capacity is not the focus of the paper this will not be further discussed.

[3]The standards considered in this paper are NORSOK and API standards, listed in references

2 State of the art

In the design of BOP control systems there are several standards that dictate what the operational capabilities of these systems shall be [25, 27, 5, 24]. As mentioned, the requirements revolve around the ability to supply the required volume whilst having sufficient amounts of hydraulic pressure at the completion of a function, within specific time constraints.

Within industry today, the common practice is the pressure and volumetric capacities are calculated and designed according to the standard requirements. Closing times are verified first at the Factory Acceptance Test (FAT) after construction and again after installation at location. Carrying out design this way enables the pressure and volumetric limited requirements for the accumulators to be verified early in the design process. However on the other hand the capability of the BOP control system to perform BOP function within the required time constraints is not verified until after the BOP control system is constructed and installed at location.

This design process creates a risk that the BOP control system, when constructed, potentially underperforms creating additional costs or unnecessary delays late in the project. This risk may be mitigated at an early stage, by verifying that the closing time is adequate.

There is also an issue of closing time being very short, the fast rate of movement on mechanical parts inside the BOP stack may cause unnecessary mechanical wear or that seals do not perform adequately.

The BOP control system design may be fine-tuned by estimating the expected closing time of a BOP with a model. This would help mitigate the risks of added costs during design and construction, as well as future costs for the BOP stack operator. Other benefits of modelling the BOP control system performance, is that it may be used in the design selection process. This allows the designer to compare design and component selections, in addition to the impact on the end product.

Earlier arguments for not establishing models for estimating the performance of new BOP control system designs, have been the low investment risk of these systems when compared to other investments in the industry. In comparison to a pipeline investment that involves very high risks, the willingness to invest in expensive simulations in order to assure that the investment performs as expected is much greater.

3 Theory

This section will describe all relevant theory needed in order to model a BOP control system interfacing a blind/blind shear BOP.

3.1 Fluid mechanics

In order to describe the mechanics in a hydraulic system, the Bernoulli equation becomes essential. The Bernoulli equation is often displayed in the form seen in equation 1. This equation describes the equilibrium between pressures at two different points in a system. The two different points are differentiated in the equation by the indexes 1 and 2 respectively.

$$P_1 + \rho g h_1 + \frac{\rho v_1^2}{2} = P_2 + \rho g h_2 + \frac{\rho v_2^2}{2} \tag{1}$$

Where P_1 and P_2 is the pressure in the liquid, ρ is the density of the liquid, h_1 and h_2 is the vertical height, v_1 and v_2 is the fluid velocity at the two respective points.

There are some assumptions in equation 1 [13, p. 133]:

- Fluid is incompressible. The density of the fluid is constant.
- Viscous effects are negligible.
- Steady flow.
- The equation applies along a streamline
- No energy is added or removed along the streamline.

In order to fit the requirements of the model, equation 1 needs to be expanded in order to account for pressure losses in the system. These pressure losses are accounted for by the addition of the parameter P_{loss} which will be discussed further in section 3.2. When pressure losses are added to the Bernoulli equation we get equation 2.

$$P_1 + \rho g h_1 + \frac{\rho v_1^2}{2} = P_2 + \rho g h_2 + \frac{\rho v_2^2}{2} + P_{loss} \tag{2}$$

3.1.1 Fluid properties

There are three properties of the hydraulic fluids that are important to discuss in the context of hydraulic modelling; density, viscosity and vapour pressure.

- The vapour pressure is of particular interest in hydraulic systems that may experience low pressures, such as on the suction side of pumps. If the pressure falls below the vapour pressure of a liquid, the formation of gas bubbles in the liquid will occur. When the pressure increases again, the bubbles will implode and once again become liquid. This is an undesirable situation, termed cavitation, which has a negative impact on system reliability, noise and efficiency.

- Density describes the mass per unit volume of fluid. For liquids such as those used in hydraulic systems, the density changes little over large pressure ranges and is normally assumed to be constant. The density of liquids is also fairly insensitive to the moderate temperature changes which BOP control systems experience. For gases, on the other hand, the density is very temperature and pressure sensitive, even under moderate changes of either parameter, which may have a significant impact on gas density. How gas density changes with regard to pressure and temperature changes will be further discussed in section 3.3.2.

- Viscosity for liquids is fairly sensitive to temperature changes, potentially becoming high in cold environments [12]. The viscosity describes a fluid's resistance to movement, similar to friction. The more viscous a liquid is, the higher the friction, causing the liquid to "flow slower". Viscosity is termed in two ways, kinematic or dynamic viscosity. Kinematic viscosity will be used in the remainder of this paper. Kinematic viscosity is measured in centi Stokes (cSt) and is related to dynamic viscosity by the following equation below.

  $$\nu = \frac{\mu}{\rho}$$

  Where ν is the kinematic viscosity, and μ is the dynamic viscosity.

Vapour pressure has little impact on the model described in this paper, as focus will be on the high pressure side of the system. It is however a parameter that is worth noting if the model should be expanded to include pumps. The liquid density and viscosity are, on the other hand, important parameters that will be referred to in subsequent sections.

3.2 Pressure losses

When fluid is stationary, all parts of the system at the same height would have the same pressure head. This is not the case for fluids in motion flowing in pipe tubes. When real fluids are in motion there will always be some loss of energy, often expressed as pressure loss. The causes of these losses are friction in the flow producing heat.

3.2.1 Pipe friction

As the fluid flows there will be friction between the pipe wall and the fluid. This friction is present throughout the pipe system and is dependent on the fluid properties, fluid velocity, internal pipe diameter and the roughness of the internal surface of the pipe.

The pressure loss in the pipe, due to pipe friction, is calculated over a length L of pipe. Pressure loss due to pipe friction is found from equation 3 [13, p. 261].

$$h_f = f\frac{Lv^2}{2dg} \tag{3}$$

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

Equation 3 is expressed in pressure head[4], it is for modelling purposes beneficial, to express the losses in pressure (Pascal), by applying equation 4.

$$P = \rho g h \tag{4}$$

Using the relationship described in equation 4, equation 3 may be expressed as in equation 5, giving results in pressure rather than head.

$$P_f = \rho g h_f = f\frac{\rho L v^2}{2d} \tag{5}$$

The friction factor depends on the flow regime of the liquid and the relative roughness of the pipe wall. Typically the value of the friction factor is found by use of a Moody diagram illustrated in figure 1. Using the relative roughness of the pipe wall and the Reynolds number, in conjunction with the Moody diagram the friction factor is derived.

Relative roughness is calculated from the roughness of the pipe inner surface, divided by the inner diameter of the pipe, using equation 6.

$$\frac{\epsilon}{d} \tag{6}$$

Where ϵ is the roughness of the pipe internal surface.

The Reynolds number is a dimensionless ratio that may be found from equation 7. It describes a relationship between the fluid inertia and viscosity. Values of the Reynolds number have shown a correlation to the different flow regimes.

$$Re = \frac{\rho v d}{\mu} = \frac{vd}{\nu} \tag{7}$$

For the purpose of the remaining discussion it is necessary to understand how the Reynolds number may be used to determine the dominating flow regime. In a straight pipe where $Re < 2000$ the flow will likely be laminar, while for $Re \geq 2000$ the flow will likely be turbulent [13, p. 257]. As seen

[4] Height of fluid column

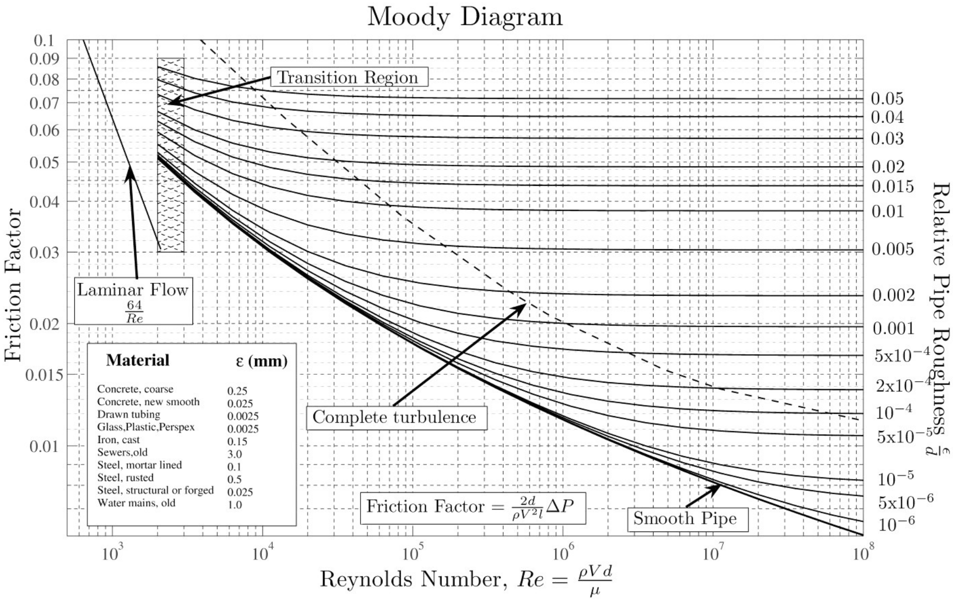


Figure 1: Moody diagram [34]

in figure 1, the friction factor is highly dependent on the Reynolds number and what flow regime is dominating.

For laminar flow ($Re < 2000$) equation 8 may be used [13, p. 256]. As can be seen, the friction factor has a linear relationship to the Reynolds number in the laminar flow region, potentially creating very high friction factors for a flow with low Reynolds number. Note that the friction factor is unaffected by the roughness of the pipe wall when flow is laminar.

$$f = \frac{64}{Re} \tag{8}$$

In a BOP control unit there is usually a high flow velocity, using hydraulic fluids with a low viscosity. Because of this the flow would most often be in the turbulent region. In order to approximate the friction factor for turbulent flow $Re \geq 2000$ the Colebrook equation, equation 9, may be used [13, p. 282].

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\epsilon/d}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) \tag{9}$$

Since Colebrook equation is implicit of $\sqrt{f}$, it has the disadvantage where it needs to be iterated. At the expense of a deviation from Colebrook[5] the Haaland equation, shown in equation 10, may be used instead [13, p. 282].

$$\frac{1}{\sqrt{f}} = -1.8\log\left[\left(\frac{\epsilon/D}{3.7}\right)^{1.11} + \frac{6.9}{Re}\right] \tag{10}$$

[5] $\pm 1.5\%$ for $4000 \leq Re \leq 10^8$

For modelling the Haaland equation will be used, in order to avoid unnecessary iteration and to decrease model complexity.

3.2.2 Minor losses

In this paper all losses that are not pipe friction losses will be considered minor losses.

For long pipe tubes the pressure loss due to minor losses are usually insignificant [13, p. 301]. However for short lengths of pipe these losses may become significant [13, p. 301]. BOP control systems have significant influence from both pipe friction and minor pressure losses. Therefore both these contributions must be assessed.

There are several ways of representing minor losses. The typical methods used are either by using a loss factor (ξ) or by use of equivalent pipe length. Flow coefficient (C_v) or flow factor (K_v) may also be used in order to estimate pressure losses[6].

The loss factor is a constant used to account for the pressure losses in a specific hydraulic component. These are often tabulated values representative for the class of components, or specified in data sheets. Equivalent pipe length is similar to the loss factor approach, but instead of representing the pressure losses as a loss factor, it is represented by a pipe length equivalent to the same pressure loss. Flow coefficient and flow factor are also constants that are similar to the loss factor approach. Being a function of flow rate and pressure drop, these coefficients may be used with some rearrangement of the equations to estimate pressure loss [14]. The flow coefficient and flow factor are equivalent where the flow coefficient is based on the imperial system, while the flow factor is based on the SI system. It is possible to convert between flow coefficient and flow factor by using the relationship

$$K_v = 0.862 C_v$$

Or C_v may be expressed in SI units using [39]

$$C_v = 11.7 Q \sqrt{\frac{\psi}{\Delta P}}$$

Where Q is the volumetric flow rate [m^3/h], ψ is the specific gravity of water[7] [$-$] and ΔP is the pressure drop [kPa].

Bends and elbows Hydraulic systems often have many bends and elbows creating disturbance in the flow thus causing pressure loss [13, p. 312]. Bends

[6]Though typically used to specify the capacity for control valves [38]

[7]Specific gravity is the relative density of the hydraulic fluid in regard to water at 20^o: $\psi = \rho/\rho_w$.

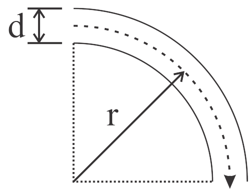


Figure 2: A pipe bend.

and elbows cause turbulence when the flow direction is changed, as well as pipe wall friction from the additional pipe length.

It is worth noting that for bends, use of a loss coefficient is preferably only used for very smooth pipes [13, p. 312]. The head loss a pipe bend or elbow create, may be calculated from equation 11 [13, p. 314].

$$h_b = \xi_b \frac{v^2}{2g} \tag{11}$$

ξ_b is the loss factor for a bend [26]. The magnitude of the loss factor may be tabulated as dependent on both the angle and radius of the bend in relation to the internal pipe diameter [29, 1, 41].

Note that we cannot treat the combinations of different bends using super positioning principle [13, p. 315]. Since the distorted flow may continue to be turbulent up to a 100 pipe diameters downstream [13, p. 312], the pressure losses from contributors in this already distorted flow, will be different than if they were placed in a laminar flow. Knowing this, the super positioning principle will still be applied in the model, creating an additional inaccuracy. This inaccuracy is considered small in comparison to the other pressures and is neglected.

Similarly as for equation 5 the bend head is expressed as pressure loss with equation 12, using the relationship in equation 4.

$$P_b = \rho g h_b = \xi_b \frac{\rho v^2}{2} \tag{12}$$

The same equations used for pipe friction pressure loss, is used when assessing the pressure loss contributions from the additional pipe length. To find the additional pipe length equation 13 may be used.

$$L_b = 2r\pi \frac{\alpha}{360} \tag{13}$$

α is the angle of the bend in degrees and r is the radius of the bend, illustrated in figure 2.

Valves When fluid passes through hydraulic valves we often experience a pressure loss, due to change in fluid flow cross section or directional changes.

The method of assessing the pressure losses in valves is somewhat different from bends, since any additional pipe length is neglected.

The pressure head loss for valves is evaluated using the loss factor approach as in equation 14, using the orifice equation or estimated using flow coefficients/factors.

$$h_v = \xi_v \frac{v^2}{2g} \tag{14}$$

ξ_v is the valve loss coefficient.

The valve head loss is expressed as pressure loss with equation 15.

$$P_v = \rho g h_v = \xi_v \frac{\rho v^2}{2} \tag{15}$$

Another method of calculating pressure losses in valves is to treat them as an orifice using equation 16 [8, p. 93].

$$P_v = \left(\frac{\rho}{2C_q^2 A_v^2} \right) Q_m^2 \tag{16}$$

C_q is the Orifice flow coefficient, A_v is the valve orifice area and Q_m is the flow rate.

As previously mentioned flow coefficient and flow factors are used to compare valves capacities. Considering the flow coefficient defined in equation 17 using SI units [39].

$$C_v = 11.7\dot{V}\sqrt{\frac{\psi}{\Delta P}} \tag{17}$$

By rearranging and utilising some algebra, equation 17 may be expressed as the pressure loss over the valve using flow rate at cubic meters per second.

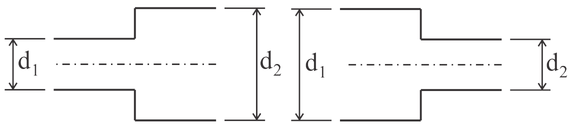
$$P_v = 10^3 \psi \left(\frac{11.7 \frac{Q}{3600}}{C_v} \right)^2 \tag{18}$$

When comparing equation 18 to equation 16 one can see a similarity. This similarity should allow equation 18 to be used as an estimate to the pressure losses in the flow [14].

Since the flow coefficient is readily available from component datasheets, it is equation 18 which is used in modelling throughout the project.

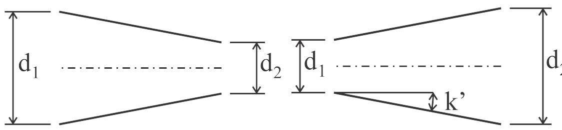
Diameter changes Hydraulic systems may have several pipe diameter changes. In the transition between these diameter changes, there will be pressure losses dependent on the cross section change. These changes are divided into the following four types; sudden contraction, gradual contraction, sudden expansion or a gradual expansion.

If there is a *sudden contraction* there will be a sudden increased pressure at the upstream side of the contraction, resulting in a lowered pressure at



(a) Sudden contraction (b) Sudden expansion

Figure 3: Illustrations of sudden diameter changes in a pipe.



(a) Gradual contraction (b) Gradual expansion

Figure 4: Illustrations of gradual diameter changes in a pipe.

the downstream side accompanied with turbulence [13, p.306]. When there is a contraction, the fluid will experience an increase in flow velocity in addition to turbulence. In order to calculate the head losses due to the sudden contraction equation 19 may be used [13, p.306].

$$h_c = \xi_c \frac{v_2^2}{2g} \tag{19}$$

ξ_c represents a loss coefficient for sudden contraction. The loss factors for sudden contractions are usually tabulated by the ratio of ID change.

In order to reduce the pressure losses in a contraction a *gradual contraction* would be used. At a gradual contraction the pressure loss may be evaluated as a nozzle using equation 20 [13, p. 307].

$$h_c = \xi_n \frac{v_j^2}{2g} \tag{20}$$

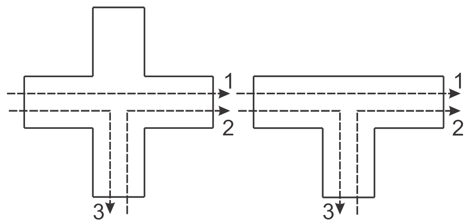
ξ_n is the nozzle loss coefficient and v_j is the velocity of the water jet downstream of the nozzle.

For a *sudden expansion* the pressure loss may be evaluated using equation 21 [13, p. 309].

$$h_c = \frac{\left(\frac{d_2^2}{d_1^2} - 1\right)^2 v_2^2}{2g} \tag{21}$$

d_2 and d_1 are the down and upstream ID of the pipe expansion and v_2 is the downstream fluid velocity.

To reduce pressure losses at an expansion, *gradual expansion* is used. A gradual expansion must be treated as a diffuser using equation 22, assuming



(a) Cross fitting or branch (b) Tee fitting or branch

Figure 5: Flow through fittings or branches.

a conical shape [13, p. 310].

$$h_c = k' \frac{\left(\frac{d_2^2}{d_1^2} - 1\right)^2 v_2^2}{2g} \tag{22}$$

k' is a function of the diffuser angle.

Similarly as for previous equations the head loss from diameter changes can be expressed as pressure with equation 23.

$$P_c = \rho g h_c \tag{23}$$

Fittings In hydraulic systems there are often many fittings and branches similar to those illustrated in figure 5. The term fittings are not limited to the ones illustrated in figure 5, but include many other components in hydraulic systems, such as unions[8]. Head losses from fittings may also be evaluated using loss coefficients as in equation 24. Note that there is often a distinction between the flows through tees and branches shown in figure 5. Where:

1. Represents along run flow.
2. Represent branch in flow.
3. Represents branch out flow.

$$h_F = \xi_F \frac{v^2}{2g} \tag{24}$$

ξ_F is the loss factor for the fitting or branch.

Equation 24 may be expressed in pressure using equation 25.

$$P_F = \rho g h_F = \xi_F \frac{\rho v^2}{2} \tag{25}$$

[8]Fittings sometimes make a distinction between losses caused by components using screw or flange fitting.

Filters As it is an open system the hydraulic filter is a very important component in a BOP control system to ensure the hydraulic fluid cleanliness. If the fluid is not sufficiently clean there will be an increase in the abrasion of hydraulic components reducing the overall system reliability.

The losses in filters are dependent on several factors such as the filter type, fluid viscosity, and flow velocity [9]. The pressure losses are given in pressure loss curves plotted as a pressure loss versus flow rate for a specific pressure level and viscosity. The amount of pressure loss at the filter is dependent on its cleanliness. As the filter gets more dirty pressure loss will increase. If the differential pressure over the filter becomes sufficiently large, a bypass valve will open allowing the fluid to bypass the filter.

Pressure losses from filters will be simplified in the model, where pressure losses will be set as equal to pressure where bypass valve is engaged. This would create a pessimistic estimate for pressure loss during normal operation.

3.2.3 Summation of losses

In most hydraulic systems there are several sources of pressure loss. These losses may be summed up as in equation 26 without too large inaccuracies. For very high accuracies this is not a valid equation because of the turbulence that will persists downstream.

$$P_{loss} = \sum_{i=1}^{n} P_{f,i} + \sum_{j=1}^{n} P_{b,j} + \sum_{k=1}^{n} P_{v,k} + \sum_{l=1}^{n} P_{c,l} + \sum_{\mathrm{m}=1}^{n} P_{F,\mathrm{m}} \tag{26}$$

n is the number of pressure losses for pressure loss i, j, k, l and m.

Collecting all the minor losses under the index m, the equation 26 may be simplified to equation 27

$$P_{loss} = \sum_{i=1}^{n} P_{f,i} + \sum_{i=1}^{n} P_{m,i} \tag{27}$$

Losses are all a function of the velocity of the fluid. Since the volumetric flow rate is constant throughout the system equation 28 can be used to relate the velocity at any point i to the flow rate.

$$Q_2 = A_2 v_2 = A_i v_i \tag{28}$$

Equation 29 is found by rearrangement.

$$v_i = \frac{A_2 v_2}{A_i} \tag{29}$$

Knowing flow rate and cross section and by inserting equation 29 into equation 26, we are able to express the pressure loss at an arbitrary point in the hydraulic using equation 30.

$$P_{loss} = \sum_{i=1}^{n} \frac{P_{f,i} Q_2^2}{A_i^2 v_i^2} + \sum_{j=1}^{n} \frac{P_{m,j} Q_2^2}{A_i^2 v_i^2} \tag{30}$$

3.3 Accumulator theory

Accumulators are used in hydraulic systems in order to store hydraulic power. There are several types of accumulator concepts in use. The standards describe two types, diaphragm and floating piston. By using a diaphragm (illustrated in figure 6) or a floating piston, nitrogen gas is separated from the hydraulic fluid [24, p. 27].

All accumulator sizing calculations will have four conditions of interest: the precharged (condition 0, P_0), charged (condition 1, P_1), discharged to minimum function operating pressure (condition 2, P_2) and fully discharged (condition 3, P_3) [25, p. 9]. Note that in this discussion the indexes represent the accumulator condition and not a point in the system.

Precharged condition is the pressure the gas in the accumulator has before fluid is pumped in. Accumulator bottles need to be pre pressurized in order to push fluid back into the system when required.

Charged condition is the pressure the nitrogen gas has when it is fully charged with fluid. This is where pumps would stop pumping liquid into the accumulator(s).

Minimum operating pressure is that pressure the Nitrogen gas requires in order for the system to perform its functions.

Fully discharged stage is the pressure the Nitrogen gas has when all the liquid inside the accumulator is discharged. In many cases this would typically be the same pressure as the precharge pressure.

3.3.1 Precharge

Precharge is included as this is useful for accumulator calculation. In addition for expanding the model by describing optimal precharge and sizing of accumulators in a BOP control system.

The precharge pressure influences the amount of fluid that can be stored when fully charged and the pressure that will be available as fluid is discharged. The level of pre pressurisation is decided usually by rule of thumb or through design standards.

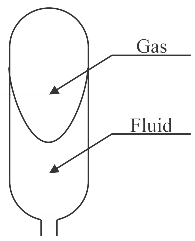


Figure 6: An accumulator with an elastic diaphragm to separate gas and fluid.

Pressure for fully charged accumulator and that of the minimum working pressure are often known for a system design. The precharge pressure is then evaluated using rule of thumb.

The rule of thumb for accumulator precharge is found from equation 31, which is valid for membrane or bladder type accumulators [31, p. 210].

$$P_0 \leq 0.9P_2 \tag{31}$$

Note that the calculated values must be compensated if there is a different working temperature [31, p. 210]

The following relation follows for pressure at precharge [31, p. 210]

$$P_{0,filled} = P_0 \frac{T_{filled}}{T_{working}}$$

The standards recommend optimal precharge pressure according to equation 32 for isotherm ideal gas expansion [25, p. 12]

$$P_0 = \frac{1}{\frac{1.5}{P_2} - \frac{0.5}{P_1}} \tag{32}$$

With regard to isotherm real gas expansion, optimum precharge density is decided from equation 33, while for adiabatic real gas expansion $\rho_0 = \rho_2$ is recommended [25, pp. 13–14].

$$\rho_0 = \frac{1}{\frac{1.4}{\rho_2} - \frac{0.4}{\rho_1}} \tag{33}$$

3.3.2 Expanding gas

Standard requirements for BOP control systems focus on accumulators being able to provide a minimum amount of volume and a minimum amount of pressure when BOP function(s) have been completed. Therefore standards dictate the need to calculate the Volumetric Efficiency (VE) of the Bottle Volumes (BV) in order to find the Functional Volume Required (FVR) for the BOP control system [25]. Where VE is the fraction of liquid in the accumulator bottles that is usable. The FVR may be interpreted as the amount of liquid volume necessary to perform a BOP function.

Since the purpose of the model is slightly different, the full description of calculating VE is not included, but the method of assessing gas expansion will be included.

There are three recommended methods of assessing gas expansion. These are isothermal flow using ideal gas, isothermal flow using real gas and adiabatic flow using real gas [25, p. 10]. The method used is dependent on the rate of discharge and the maximum gas pressure in accumulators [25, p. 10]. Each of these will be discussed.

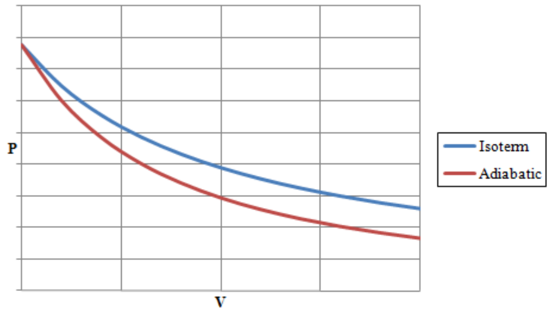


Figure 7: Isothermal and adiabatic expansion for Nitrogen gas.

Isothermal ideal gas may be assumed if the discharge rate from the accumulators is slow. The accumulator gas temperature would then be constant, since heat would diffuse to the gas from the environment. An isothermal gas expansion will have a pressure versus volume curve that look similar to the one illustrated in figure 7.

If the isothermal process does not happen under high pressure (less than 5015 psia [25, p. 12]), the idealisation of isothermal ideal gas can be made. Such an expansion of gas would follow the relation in equation 34 [31, p. 207].

$$PV = C \tag{34}$$

Where P is pressure, V is volume of the gas and C is a constant.

The ideal gas assumptions dependency to pressure, is related to the density of the gas. If the gas density is very low, the distances between the molecules would be so large that the intermolecular interactions can be neglected [32, p. 61].

With *isothermal real gas* assumption accuracy is increased for slow discharge situations. This method shall be used for pressures exceeding 5015 psia, and gives a more accurate result than using ideal gas assumptions [25, p. 13].

By using real gas values we are able to describe properties that cannot be described fully using ideal gas. It is recommended that NIST tables[9] are used to find real gas pressures, based on gas density and temperature [25, p. 13]. An illustration of the behaviour of isothermal gas expansion is shown in figure 7.

Knowing the density change of the real gas in an isothermal expansion, it is possible to use gas tables to find the pressure change.

By using real gas tables the following procedure may be used in order to find each accumulator stage [25].

[9] These may be found at `http://webbook.nist.gov/chemistry/fluid/`

Steps 3 and 4 are used if precharge and further sizing calculations according to standards are performed. With reference to the previously described accumulator conditions we can obtain

1. Gas density in condition 1 from pressure and ambient temperature in condition 1.
2. Gas density in condition 2 from pressure and ambient temperature in condition 2.
3. Calculate optimum precharge (condition 0) density from equation 33.
4. Pressure at condition 0 from density and temperature in condition 0.

Adiabatic real gas is assumed if pressure changes are rapid. It is reasoned that since the process is quick, there is not sufficient time in order for heat to diffuse into the accumulator gas. Consequently the energy in the accumulator during charge or discharge (changing gas density) will be constant, resulting in a change of temperature and pressure but constant entropy. As it is a blind/blind shear BOP that is discussed in this paper, adiabatic real gas expansion is to be considered [25].

Though not fully true, adiabatic assumption is a good approximation for fast fluid discharge from an accumulator, but will be conservative in regard to a real situation.

The adiabatic process may be described with equation 35 [31, p. 207]. An illustration of adiabatic expansion is shown in figure 7.

$$PV^{\gamma} = C \tag{35}$$

γ is the adiabatic index. The adiabatic index is dependent on pressure, temperature and the composition of the compressed medium [8, p. 67]. Nitrogen filled accumulators are often assumed as $\gamma = 1.4$ [35, p. 7].

Standards recommend using gas tables[10] [25] in the following sequence, instead, in order to find the necessary pressure changes for precharge and sizing calculations:

1. Density and entropy in condition 1 from pressure and ambient temperature in condition 1.
2. Density in condition 0 from pressure and ambient temperature in condition 0.
3. Pressure in condition 1 from density and max environmental temperature in condition 1.
4. Density in condition 2 from pressure in condition 2 and entropy in condition 1.

[10]These may be found at `http://webbook.nist.gov/chemistry/fluid/`

5. Pressure in condition 3 from density condition 3 and entropy in condition 1.

6. Density in condition 2 from pressure and entropy for each function in condition 2.

The above mentioned algorithm for calculating gas pressure needs to be altered for modelling purposes. Given the precharge pressure is already decided and the pressure after a volume is discharged from the accumulators is of interest. The suggested method listed below, is found to fit well with dynamic modelling of adiabatic real gas using the Bender model [30]. It is this algorithm that is used to prepare models presented in this paper.

1. Density at condition 0; from pressure and ambient temperature at condition 0

2. Density and entropy at condition 1; from pressure and ambient temperature at condition 1

3. Gas volume at condition 1 is found from the equation $V_0\rho_0 = V_1\rho_1$

4. Gas volume at condition 2' is found from the equation $V_{2'} = V_1 + \mathrm{FVR}_{2'}$

5. Density at condition 2' is found from the equation $V_1\rho_1 = V_{2'}\rho_{2'}$

6. Pressure at condition 2' from density at condition 2' and entropy at condition 1

Where 2' is the system stage after an arbitrary volume is discharged from the accumulators.

Polytrophic real gas should also be mentioned in the discussion. Actual behaviour of gas inside an accumulator during discharge will be somewhere between an isotherm and adiabatic process [31, p. 207]. The reason for this, is that there will always be some diffusion of heat to the gas, causing the gas to have less energy than an isothermal process, but at the same time more than for an adiabatic. Considering polytrophic behaviour, it is possible to avoid over dimensioning with regard to adiabatic processes and under dimensioning with regard to isothermal processes.

A polytrophic process for an ideal gas follows the relation in equation 36 [8, 32, p. 66,p. 276] and is valid for any value of n except $n = 1$ (which is an isothermal process) [8, p. 66].

$$PV^n = C \tag{36}$$

The polytrophic index cannot be accurately predicted [8, p. 67], but is a function of the time used to charge and discharge the accumulator [31, p. 207].

3.4 Regulators

Regulators are components in hydraulic systems that may be used to limit the flow rate or pressure downstream from where they are placed. Focus here will be pressure regulators, where if not differently specified, will be referred to.

In its simplest form the pressure supply downstream of a regulator would follow an equation similar to equation 37. For flow regulators the equation determining the system flow would be similar to equation 37, but volumetric flow would substitute the pressures

$$P_{reg} = \begin{cases} P_{set} & \text{if} \quad P_{set} < P_{acc} - P_{LossBefore} \\ P_{acc} - P_{LossBefore} & \text{if} \quad P_{set} \geq P_{acc} - P_{LossBefore} \end{cases} \tag{37}$$

$P_{LossBefore}$ represents pressure losses (including static and kinematic pressures) upstream of the regulator and P_{set} is the pressure that the regulator regulates to. If there are regulators that are connected in series, the parameter P_{acc} (pressure in the accumulators) may be substituted for the other regulator P_{reg}, where the new P_{reg} is described by an equation equivalent to equation 37.

How pressure downstream of the regulator would experience the changes in pressure supply (assumed here to be accumulators), is illustrated in figure 8. The horizontal line is the pressure downstream before pressure upstream drops below the regulating pressure. The two curves represent the pressure downstream when the pressure upstream drops below the regulating pressure. The solid curved line represents how the curve would look if there are no upstream pressure losses, while the dotted line is the actual pressure curve when pressure losses are also considered.

3.5 BOP back pressure

Back pressure will in this discussion be defined as the pressure experienced at the BOP.

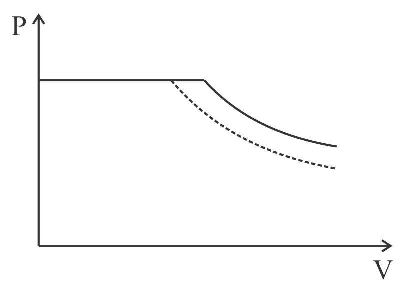


Figure 8: Pressure downstream from regulator.

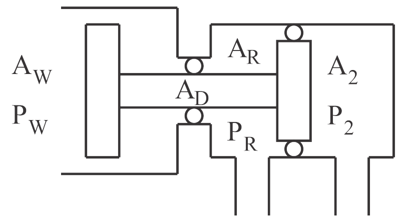


Figure 9: Pressure distributions on a simple hydraulic actuator.

The blind/blind shear rams in a BOP are powered by hydraulic actuators. In order to move the rams they need to be supplied with hydraulic pressure. When moving there are two situations that are of interest:

- When the rams move freely.
- When the blind/blind shear rams move and shear a DP.

3.5.1 Moving BOP rams

In order to move a hydraulic actuator it is necessary for the actuator to have sufficiently high closing pressure to overcome friction, pressure in the return line and pressure in the well. The pressures working on a simple actuator are illustrated in figure 9.

From the figure, the pressure at P_2 is the closing pressure, the pressure at P_R is the pressure in the return line, and P_W is the pressure in the well. In the case of a kick, the pressure P_W may be very high, requiring a significantly higher closing pressure than what would be necessary under normal conditions. This added pressure, necessary to close the BOP, is described by a BOP closing ratio which is the area of the piston operator divided by the area of the ram shaft [25, p. 3] given in equation 38 [11].

$$\text{BOP closing ratio} = \frac{\text{Wellhead Pressure at BOP}}{\text{Hydraulic Pressure Required to Close BOP}} \tag{38}$$

The pressure to move the BOP rams is dependent on the BOP in question. This is because different BOPs have different closing ratios, as well as different losses and sources of friction. Using the illustration in figure 9, the necessary pressure to close the BOP would have to be as in equation 39.

$$P_2 > \frac{P_R(A_R - A_D)}{A_2} + \frac{P_W A_D}{A_2} + \frac{F_{friction}}{A_2} \tag{39}$$

Some BOPs may have tandem actuators (two closing and opening chambers in series). In such cases equation 39 is expanded to account for the extra chamber.

The back pressure from the BOP would be similar to the illustration shown in figure 10. The enumerated points are as follows:

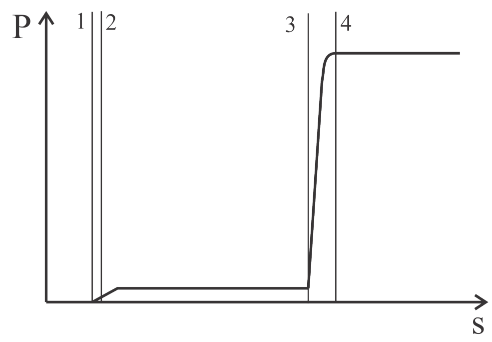


Figure 10: Pressure versus time when closing BOP without DP in bore.

1. Initiation of sequence.
2. Ram starts moving.
3. Ram is fully closed.
4. Ram is fully closed and pressure is stabilized in the system.

3.5.2 Shearing with BOP rams

Pressure behaviour when closing the BOP becomes more complex when there is a DP in the bore. This is because there will be an additional force contributors when the BOP shear rams are shearing the DP.

Shearing of a DP is a fairly complicated process with large strains and plastic behaviour, illustrated in figure 11. This can be simplified to equation 40 by assuming the entire pipe cross section is cut at once, and there is perfectly plastic behaviour where that edge effects are ignored. This would simplify the problem to the one illustrated in figure 12.

$$F_{shear} = \tau_Y 2\pi r t \tag{40}$$

Where τ_Y is the yield stress.

Note that equation 40 is a simplification of the actual problem, as the force needed to shear a pipe in a BOP is dependent on both the BOP design, as well as the drill pipe design and material quality [15, p. 2].

By expanding equation 39 to include the shearing in equation 40 we will get equation 41.

$$P_2 > \frac{P_R(A_R - A_D)}{A_2} + \frac{P_W A_D}{A_2} + \frac{F_{friction} + F_{shear}}{A_2} \tag{41}$$

The behaviour of the BOP back pressure needs to be considered differently when the BOP is shearing a DP, since the pressure is no longer

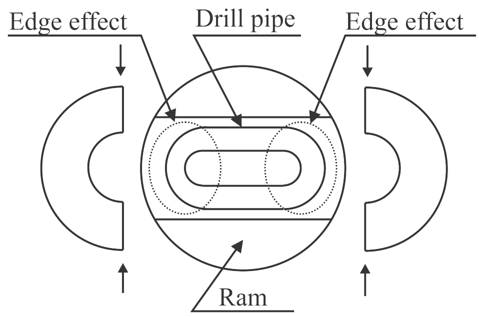


Figure 11: A deformed drill pipe in a BOP during shearing

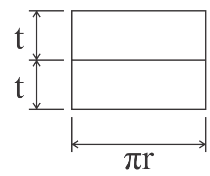


Figure 12: Assumed cross-section that is sheared.

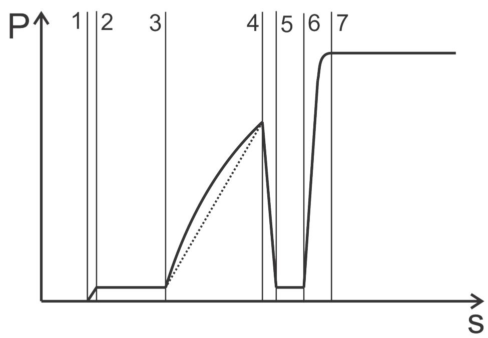


Figure 13: Pressure versus time during shearing of DP.

constant. An illustration of the back pressure exerted by the BOP when shearing a DP is illustrated in figure 13.

The enumerated points in figure 13 are as follows; 1 initiation of sequence, 2 ram start moving, 3 ram touches DP, 4 pressure reaches sufficient pressure to shear pipe, 5 pipe has been cut and ram continues to close, 6 ram is fully closed, 7 ram is fully closed and pressure is stabilized in the system. The dotted line represents a possible linear approximation of pipe buckling and shearing [47].

In the context of modelling the shearing of a DP, equation 42 is proposed. This equation is a linear approximation of the BOP back pressure and is used in the models presented in this paper.

$$P_{BOP} = \begin{cases} P_a & \text{if} \quad V_{dis} < V_a \\ P_a + P_b \frac{V - V_a}{V_b - V_a} & \text{if} \quad V_a \leq V_{dis} \leq V_b \\ P_a & \text{if} \quad V_{dis} > V_b \end{cases} \tag{42}$$

V_{dis} is the volume discharged to the BOP.
V_a is the volume discharge when BOP rams touches the DP.
V_b is the volume discharged when BOP rams have sheared the DP.
P_a is the back pressure from the BOP when the shear rams are moving without resistance.
$P_b = P_{shear} - P_a$ is the required hydraulic pressure needed to shear the pipe minus pressure required to move the actuator.

Using the dimensions illustrated in figure 14, the required volumes may be calculated in the following equations. Note that the parameter Δl may not be readily available. Δl is the sum of the distances that each actuator pistons moves.

$$A_{\text{BOP}} = \frac{\text{FVR}}{\Delta l}$$
$$V_a = A_{\text{BOP}} \left(\frac{\text{OD}_{\text{DP}}}{2} - \frac{\text{ID}_{\text{BOP}}}{2} \right)$$
$$V_b = A_{\text{BOP}} \frac{\text{ID}_{\text{BOP}}}{2}$$

BOP service companies may recommend other methods for calculating the necessary hydraulic pressure to shear a DP.

Two recommended methods of calculating the required hydraulic pressure ($P_2 = P_{shear}$) in order to cut a DP, are dependent on if there is a kick situation or not. For shearing operations where there are no pressure effects, equation 43 should be used. In the case of a shearing operation when there are pressure effects, equation 44 should be used [4].

$$P_{shear} = \frac{C_3 w_n \sigma_y}{C_1} \tag{43}$$

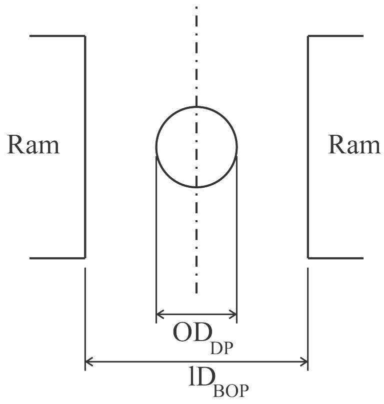


Figure 14: Illustration of a DP inside a BOP

C_3 is the shear ram type/pipe grade constant, which is empirically obtained, w_n is the nominal tubular weight in lbm/ft and C_1 is the BOP/Operator constant and is corresponding to the piston closing area (in^2) [4].

$$P_{shear} = \frac{c_3 w_n \sigma_y + P_w C_2}{C_1} \tag{44}$$

C_2 is a BOP/Operator constant corresponding to the Operator piston rod cross section area

The constants (C_1, C_2, C_3) in equation 43 and 44 are tabulated values supplied by the BOP service provider [4].

3.6 Pressure limited flow rate

The rate liquid is discharged from the accumulator is highly dependent on the pressure losses and differential pressure in the hydraulic system. Rate of discharge is of interest in the later modelling, since this is what determines the time it takes to close a BOP.

Starting with the Bernoulli equation in equation 2 we use the following definitions:

- P_1 represents the pressure in the accumulator (or regulator if present)
- P_2 represents the pressure in the BOP
- Fluid is stationary $v_1 = 0$ in the accumulator

With these assumptions in place and setting $h_2 - h_1 = \Delta h$ we obtain equation 45.

$$P_1 = P_2 + \Delta h \rho g + \frac{\rho v_2^2}{2} + P_{loss} \tag{45}$$

Rearranging the equation to find the fluid velocity at the BOP (point 2) as equation 46.

$$v_2 = \sqrt{\frac{2(P_1 - P_2 - \Delta h \rho g - P_{loss})}{\rho}} \tag{46}$$

By multiplying the velocity of the liquid with the area of the flow, we will get the volumetric flow rate (discharge rate) as in equation 47.

$$Q_2 = v_2 A_2 = A_2 \sqrt{\frac{2(P_1 - P_2 - \Delta h \rho g - P_{loss})}{\rho}} \tag{47}$$

A_2 is the internal pipe area at the BOP.

By inversing equation 47 we get the number of seconds per unit volume. Assuming that the accumulator and actuator pressure depends on the volume discharged. Integrating equation 47 over the volume discharged at point a to b, the time taken to discharge that amount is found. This is illustrated in equation 48.

$$t = \int_{V_a}^{V_b} \frac{1}{Q_2} dV_{dis} \tag{48}$$

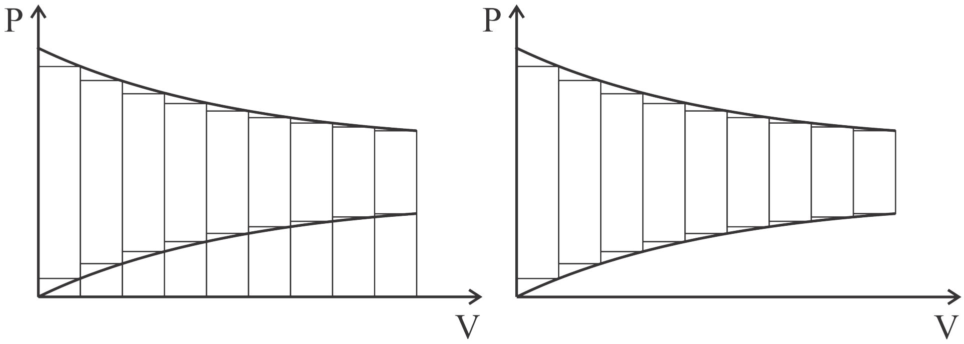
The integral in equation 48 may be difficult to solve analytically for an arbitrary BOP control system and BOP. Using the same principle as equation 48, the model will be similar to a Riemann right sum. Approximating an integral by dividing the function into many smaller intervals, thereby evaluating each and summing up the individual contributions.

These intervals are illustrated in a pressure versus volume discharged diagram in figure 15. Here figure 15a illustrate the calculated pressures for each of the intervals (illustrated by columns). Figure 15b is the calculated differential pressures (pressure source minus back pressure) for each interval. Notice that the differential pressure calculated in this way is slightly lower than actual pressure.

3.7 Sensitivity analysis

All models should be given a sensitivity analysis [3] when modelling real world systems. There will always be uncertainties to the accuracy of the model itself and the outlining parameters and assumptions that might affect it. The uncertainties in the context of a BOP control system would range from system design choices such as pipe lengths, bends, material selection, component selection etc. to those regarding the interconnection of interfacing systems.

Therefore, when modelling a systems performance, it is important to know how parameters influence model results. There may be some parameters that when changed slightly, will cause significant change to the output.



(a) Initial pressures (b) Differential pressure

Figure 15: Pressure intervals that models are divided into. Upper curve is the accumulator pressure, lower curve is the BOP back pressure.

By doing a sensitivity analysis we attempt to assess the relative importance of parameters in the model, in the presence of uncertainties [3]. Knowing the importance or sensitivity of the input parameters, and knowing that all input parameters are uncertain [3], it is possible to assess what parameters influence a BOP control system model the most, thus given the most care to define.

When understanding how much the different parameter in a model impact the end result, possible simplifications to the model may be done. For example parameters in a model that do not impact results may be flagged as irrelevant and neglected in order to avoid unnecessary complexity [3].

It is required from a sensitivity analysis that the concept of importance is defined rigorously before the analysis [3]. In the context of this paper, the closing time of BOP is of importance, thus the parameters that affect the closing times are also of interest.

3.7.1 One-way sensitivity analysis

One-way sensitivity analysis is used in order to find out what parameter has the biggest impact on model outputs. This is carried out by changing each parameter individually, by a predetermined amount and recalculating, noting the results each time [37].

The results from the analysis is then sorted and illustrated in a tornado chart [37]. The tornado chart is used to graphically illustrate the sensitivity of the different parameters.

Higher values depict the more sensitive parameters, and will have a larger impact on the model output if changed. In order to get an accurate model result, it is important to have more accurate input values for the more sensitive parameters. Parameters that are less sensitive, are thus less critical to define accurately in order to obtain an accurate result from the model. It does not, on the other hand, describe the uncertainties related to the parameters [37]

Sensitivity analysis may be further expanded to multi-way sensitivity analysis where two or more parameters are changed simultaneously. This can become very difficult and complex as the number of parameters that are changed increases [37]. In order to simplify such an analysis it is possible only to use an extreme analysis, where only the most extreme optimistic and pessimistic parameter values that are reasonable are used [37].

3.8 Calibration

In a model and/or with measurements, there are two types of errors; random or systematic errors.

The random errors are non-repeatable when comparing a model to a prototype. With regard to the model described in this paper, sources of random errors would be temperature changes or liquid cleanliness changes, etc.

The difficulty with random errors is that they cannot be completely removed since conditions of every prototype test cannot be exactly duplicated. The best way to mitigate the impacts from these uncertainties is via mitigation of sensitive parameters.

The systematic errors are errors present and unchanged with every repeat of the model. With regard to the model in this paper, typical sources of systematic errors would be related to incorrect input values, such as incorrect minor losses, wrong pipe lengths, etc. Systematic errors are easier to mitigate by using more accurate information about the prototype or by calibration.

Calibration is therefore applied to the model in order to mitigate systematic errors. This will be done by simply multiplying a calibration factor with the most sensitive parameter (that has an uncertainty related to it) of the model.

3.9 Polynomial regression

In order to simplify and generalise the model that will be described in this paper, it will be necessary to perform polynomial regression. This in order that a simple function for an arbitrary gas expansion under adiabatic conditions may be described. Polynomial regression is a special case of multiple linear regression models [33, p. 448].

In order to solve a two parameter polynomial regression problem by hand, the following procedure is suggested for n data points, where x is the values along the x-axis, y are values along the y-axis and b are unknown constants

[33, p. 449].

$$\sum_{I=1}^{n} y_I x_I^0 = \sum_{I=1}^{n} x_I^0 x_I^0 b_0 + \sum_{I=1}^{n} x_I^0 x_I^1 b_1 + \sum_{I=1}^{n} x_I^0 x_I^2 b_2$$
$$\sum_{I=1}^{n} y_I x_I^1 = \sum_{I=1}^{n} x_I^1 x_I^0 b_0 + \sum_{I=1}^{n} x_I^1 x_I^1 b_1 + \sum_{I=1}^{n} x_I^1 x_I^2 b_2$$
$$\sum_{I=1}^{n} y_I x_I^2 = \sum_{I=1}^{n} x_I^2 x_I^0 b_0 + \sum_{I=1}^{n} x_I^2 x_I^1 b_1 + \sum_{I=1}^{n} x_I^2 x_I^2 b_2$$

Solve for b_0, b_1 and b_2. When constants are found they will be used in the equation below in order to describe the polynomial function through the data point described [33, p. 449].

$$\hat{y} = b_0 + b_1 x + b_2 x^2$$

4 Modelling

This section will describe the model inputs, results and parameter sensitivity for a BOP control system interfacing a blind/blind shear BOP. The modelling section is divided in to the following subsections:

1. Modelling the BOP control system with a DP inside the BOP.
2. Model is the same BOP control system but without DP present in BOP. In this section the model is also calibrated in order to fit the closing time recorded in the prototype.
3. Model is equivalent to the first model, but with the effects from calibration accounted for.
4. Describes the results from the prototype FAT.
5. A short summary of the results that were found from the models.

When modelling, the ideal is that all theory is established beforehand of the prototype experiments. It is therefore necessary to emphasise that prototype tests were performed and results obtained by the author before the modelling. Some results from the prototype tests have consequently been used in order to create the model[11].

Even though the prototype measurement results where known the model was otherwise unaffected.

4.1 System model shearing a DP

The initial model will describe the closing and shearing of a DP in a blind/blind shear BOP interfacing a BOP control system that supplies power from an accumulator bank. The configuration of the model and the components in the model, are equivalent as those in the prototype discussed in section 4.4.

4.1.1 Inputs

Inputs in the modelled system are listed in table 1, 2, 3 and 4. Where table 1 describes the ambient temperature, liquid used, differential heights, number and size of accumulators and the FVR of the BOP as well as BOP ram travelling distance.

Temperature and accumulator data from table 1 is used in conjunction with table 2 to calculate pressure and liquid volumes in the accumulators for pre-charge, charged and at least two more arbitrarily adiabatic discharge

[11] Pressure at which the pipe was sheared by BOP, and back pressure from BOP when BOP ram is moving.

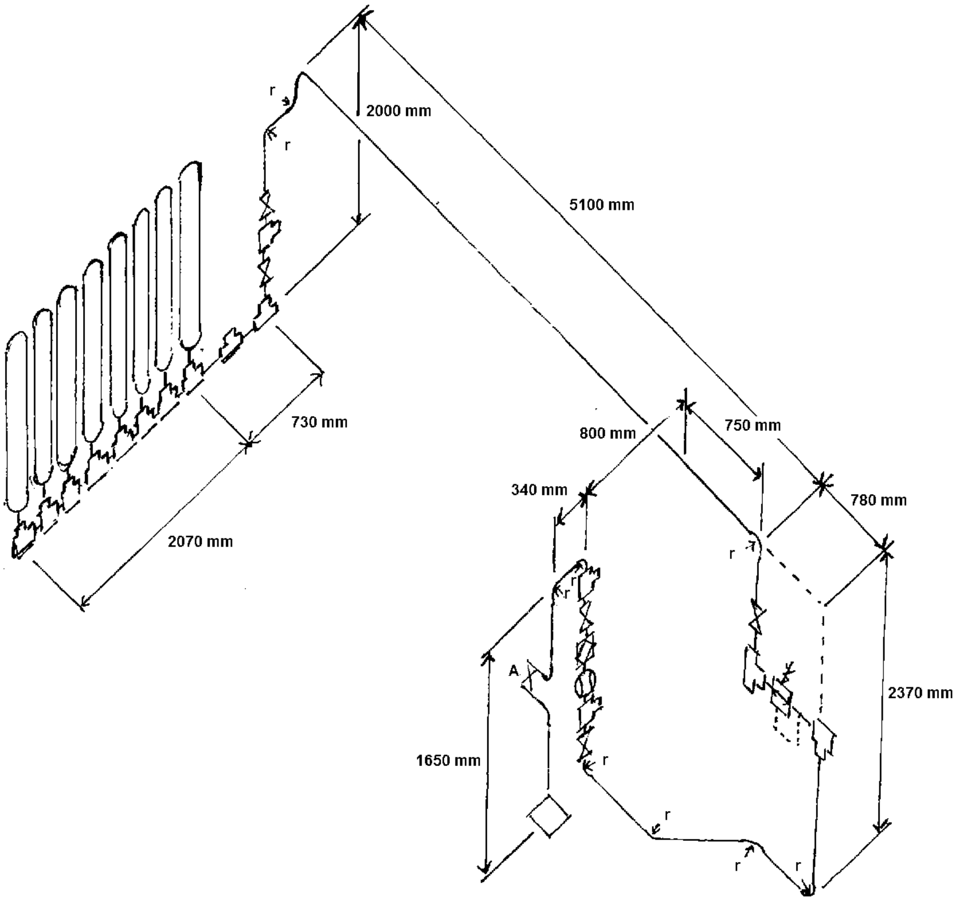


Figure 16: Isometric drawing of BOP control system.

pressure stages. Note that values marked with asterisk (*), in table 2, are found in gas tables.

Table 3 contains all minor losses in the system model based on system P&IDs [22, 23, 17], where flow coefficient are found from the components respective datasheets and loss factors are found from other available sources. Similarly for table 4, pipe and hose lengths are found from general arrangement drawings [20, 21, 18, 19, 46]. It's worth noting that the loss factors in table 3 and roughness values in table 4 use pessimistic values from the sources listed in the appendix.

An isometric drawing of the BOP control system can be seen in figure 16. The drawing is simplified to include the relevant parts of the system (unions not included, hose length drawn as pipe), not including the system interfacing the BOP control system. Note that the components seen in figure 16 in the BOP control system are listed in sequence in table 3.

Note that the pressure loss due to the regulator in table 3 are included

Table 1: Temperature, liquid, accumulator and BOP inputs.

Environment:	
Temperature	293.15 K
Liquid[12]:	
Density	1.12
Viscosity	9 cSt
Static heights[46]:	
Accumulators	0 m
Regulator	0 m
BOP	2 m
Accumulators[30, 45]:	
Precharge	2900 psi
Charged	5000 psi
Volume	50 l
Number	8
BOP[45]:	
FVR (close)	24.5 US gal
Travel distance	320.2 mm[28]

Table 2: Pressure and fluid volumes in accumulators.

	Pressure	Temp.	Density	Entropy	Volume [l]		
Stage	psia	K	lbm/ft^3	BTU/lbm R	Gas	Liq.	Disc.
0	2900	293.15	14.893*	1.2041*	400	0	0
1	5000	293.15	22.469*	1.1538*	265.13	134.87	0
2”	4000	257.45*	20.647*	1.1538	288.53	111.47	23.40
2’	3000	238.40*	18.365*	1.1538	324.38	75.62	59.25
2	2393.9*	224.20*	16.646	1.1538	357.87	42.12	92.75

in the pressure losses downstream of the regulator. Losses upstream and downstream of the regulator are separated by a horizontal line in table 3 and 4. Loss factors (LF) are the pessimistic values from sources used.

Table 4 also accounts for the added lengths from the bends in table 3. It is assumed that all bends are long radius bends with a radius of 6.7 cm [2].

4.1.2 Calculations

In order to use equation 42 the required parameters for the equation are $P_a = 300$ psi, $P_{shear} = 2423.1$ psi and volumetric requirements are calculated as $V_a = 56.554$ l, $V_b = 78.1643$ l [47, 28, 45]. Afterwards pressure versus volume discharged function is created using second order polynomial regression for the stages 1, 2", 2' and 2 in table 2.

The closing of the BOP is then divided into intervals in order that flow rates may be calculated. At each of the intervals, differential pressure is calculated. This calculated differential pressure is afterwards used to iterate a flow rate so that pressure losses are equal to the differential pressure. Knowing the amount of fluid at each interval, the discharge time for each was calculated using the appropriate flow rate. When all of the intervals were calculated the results were summarised and total time found.

All losses were calculated according to the appropriate equations, as well as for the regulator. The pressure loss from the regulator was included in the losses downstream of the regulator.

Initially the model was produced where each of the fourteen stages had its own copy of the system in the spreadsheet. The intervals were selected to be large, but as the differential pressure decreased, the increments were reduced in order to mitigate model errors. Afterwards a macro was written and used in order to make it practical to increase the number of intervals and remove bias.

4.1.3 Results

Initial results by using a model with fourteen intervals are summarised in table 5. From the results it can be seen that the model expects a closing time of approximately 22.5 seconds.

It is found that several pressure stages have a negative differential pressure. This means that the system, if adiabatic, would never be able to shear the DP. This is not to say that the actual system would not be able to shear the pipe. Since pre-charge pressure is higher than the pressure needed to shear the pipe [47, 45, 30], the gas in the accumulators would eventually heat up sufficiently to create the remaining pressure needed to shear the DP.

The volume of the interval that has a negative differential pressure would be approximately ~77-78.2 l liquid discharged from accumulator, corresponding to a volume of ~1.2 l. Should this shearing operation be successful,

Table 3: Minor losses listed along the flow path.

#	Description	Parameter	ID ["]	LF	C_v	Note
1	Acc.tee	Branch in	2	2		P_{acc}
2	Avg.acc.tee	Along run	2	0.9		
3	Avg.acc.tee	Along run	2	0.9		
4	Avg.acc.tee	Along run	2	0.9		
5	Tee	Along run	2	0.9		
6	Tee	Branch out	2	2		
7	Valve	Ball	2		105 [42]	
8	Tee	Along run	2	0.9		
9	Valve	Ball	2		105 [42]	
10	Elbow	R-45^o	2	0.42		
11	Elbow	R-45^o	2	0.42		
12	Union		2	0.8		
13	Union		2	0.8		
14	Elbow	R-90^o	2	1.5		
15	Valve	Ball	2		425 [44]	
16	Tee	Branch out	2	2		
17	Regulator	3000 psi	0.75		2 [10]	
18	Tee	Branch out	1	2		
19	Elbow	R-90^o	1	1.5		
20	Elbow	R-45^o	1	0.42		
21	Elbow	R-45^o	1	0.42		
22	Elbow	R-90^o	1	1.5		
23	Valve	Ball	1		105 [43]	
24	Tee	Long run	1	0.9		
25	FT		1			0.5 bar [36]
26	Filter	Bypass	1			3.4 bar [9]
27	Valve	Ball	1		105 [44]	
28	Tee	Long run	1	0.9		
29	Elbow	R-90^o	1	1.5		
30	Elbow	R-90^o	1	1.5		
31	Valve	Control	1		9.2 [6, 7]	
32	Union		1	0.8		
33	Union		1	0.8		
34	Contraction	BOP inlet	0.75	0.35		

Table 4: Summary of piping lengths in system.

#	Description	ID ["]	Length [mm]	Roughness [mm]
1	Avg. acc. distance	2	1,035	0.03
2	Acc. end horizontal	2	730	0.03
3	Acc. end vertical	2	2,000	0.03
4	Hyd. hose	2	6,000	0.05
5	Corner vertical 1	2	2,370	0.03
6	Add. from bend	2	210	0.03
7	Corner horizontal 1	1	780	0.03
8	Corner horizontal 2	1	780	0.03
9	Extra length back	1	750	0.03
10	Corner horizontal 3	1	800	0.03
11	Corner vertical 2	1	1,650	0.03
12	Valves horizontal	1	340	0.03
13	Valves vertical	1	1,650	0.03
14	Hyd. hose	1	31,000	0.05
15	Add. from bend	1	526	0.03

a temperature increase of 5.35 K to 235.35 K is necessary. Shearing operation may have been completed if the pipe was sheared earlier, when less fluid had been discharged from the accumulators. In this case, the BOPs back pressure curve in figure 17 would have been shifted towards the left where the differential pressure is larger.

The intervals in table 5 are plotted in a pressure versus time graph as in figure 17. Note that the right most axis in figure 17 is the discharged volume from the accumulators.

When performing the same calculations but with a larger number of intervals (262 intervals, equal to 10 psi accumulator pressure change each) the calculated closing time decreases somewhat to 21.44 seconds. A plot from these calculations is illustrated in figure 18. Notice that the model ignores negative differential pressures. The discharge time is therefore optimistic. In situations where calculating a negative differential pressure, the BOP control system may not be able to supply the sufficient amounts of pressure for shearing the DP.

4.1.4 Sensitivity analysis

It is of interest to see how sensitive the model is to changes in the input data. This is important to know when defining the model, in order to know what parameters need to be given extra attention in order to get an accurate model. While also assessing what parameters that may be neglected without effecting model results.

The following parameters are considered in the sensitivity analysis:

Table 5: Calculated system performance with un-calibrated system model

	Time			Pressure		
Stage	Stage	Sum	Volume	Acc.	Reg.	BOP
0	0	0	0	5000	3000	300
1	10.5	10.5	56.1	3100	3000	300
2	1.6	12.1	63.1	2950	2916	1032
3	0.6	12.7	65.5	2900	2870	1300
4	0.7	13.4	67.9	2850	2824	1573
5	0.8	14.2	70.4	2800	2778	1851
6	1.1	15.2	72.9	2750	2734	2134
7	1.7	17.0	75.5	2700	2691	2421
8	0.4	17.4	76.0	2690	2682	2479
9	0.6	18.0	76.5	2680	2675	2537
10	1.4	19.4	77.0	2670	2668	2595
11	0.0	19.4	77.5	2660	NEG	2654
12	0.0	19.4	78.1	2650	NEG	2713
13	0.1	19.5	78.6	2640	2601	300
14	3.0	22.5	92.3	2390	2354	300
n	0	22.5	92.3	2390	2390	2390

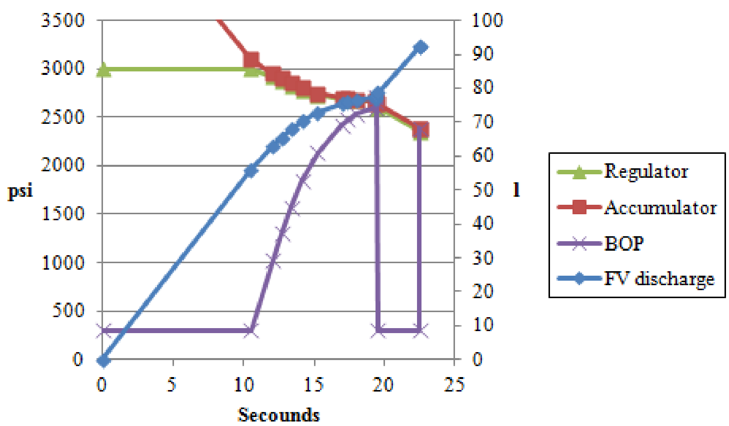
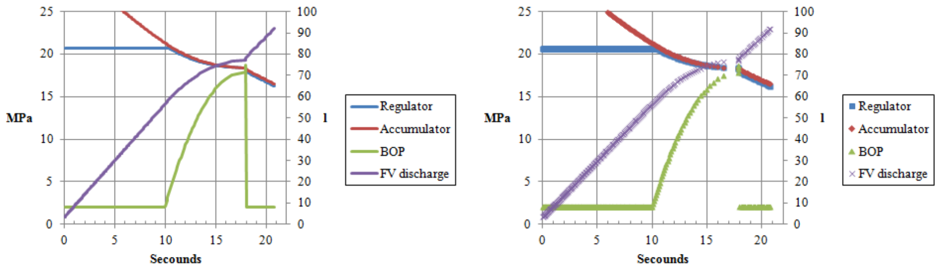


Figure 17: Pressure versus time for un-calibrated model with DP.



(a) Lines between data points (b) Only data points

Figure 18: Pressure calculations using a high number of data points, for an un-calibrated model with DP.

- Total piping and hose lengths are of interest since these values may be uncertain or may change during design. Lengths of piping downstream of the BOP control system may also vary significantly depending on where it is installed.
- ID of piping or hose interfacing the BOP control system is of interest, as these dimensions are dependent on installation location.
- The viscosity of the hydraulic fluid may vary significantly[12] due to variations in temperature.
- Roughness of the internal surfaces of piping and hoses may influence the performance of a BOP control system as well. This is also design dependent, and may change significantly on design selection.
- Minor losses before the regulator are included as they are part of the pressure losses in the system.
- Minor losses after and including the regulator are included as they are part of the pressure losses in the system. Some of the largest individual minor loss contributors are included here as well.
- Changes to when shearing rams touches the DP is not necessarily known by the analyst, and may therefore sometimes be estimated, as well as the differential pressure is a function of this parameter. With regards to "when", in this context, it is the volume that is discharged during closing.
- Pressure needed to shear the DP may vary greatly depending on BOP and the DP. Consequently it will impact the differential pressure during

[12]A temperature change from may change the kinematic viscosity of a liquid from 20 cSt to 9 cSt by varying the temperature from 0 to 20^{o}C.

shearing, with the possibility that the shearing may not be completed at all.

- Pressure needed to move the BOP rams may often be unknown, as well as affecting the differential pressure throughout the closing function.

The sensitivity analysis is performed with a simple one-way sensitivity analysis. Each parameter considered is changed, one at a time, by ±20% and model is recalculated. The relative change in seconds is calculated and ranked in table 6. To illustrate the relative sensitivity of each parameter a Tornado chart is made from the values in table 6 and shown in figure 19.

Note that the shear point would not allow a 20% increase in volume. This would result in a volume where shearing is completed with a FV that is larger than the total FVR for the BOP. A smaller increase was not included, as the only result from this, is a larger number of measurements would be neglected. Also a 20% increase of shear pressure was neglected as this would result in a much larger shear point pressure ($2700 \times 1.2 = 3240$ psi), only resulting in more negative differential pressures that would be neglected.

In table 6, the most sensitive parameter is the minor losses downstream of (and including) the regulator. The pressure losses here are dominated by the regulator and control valve. This shows that it is very important that large minor loss contributors are defined in order to model correct closing times.
The second most sensitive parameter is the ID of the hose between the BOP control system and the BOP stack. The dimension of interfacing system is of high importance for the BOP control system performance.
The shear point and share pressure is also of significant importance, both of which have a large impact on closing time.
Movement pressure of the BOP ram actuators has a less substantial impact on the closing time but should not be neglected. If high accuracy is needed this parameter must be included.
Changing the pipe length ±20%, has little impact on the closing time, but should still be estimated by the analyst in order to get as accurate a model as possible. These values are also fairly easy to acquire from technical drawings and models. The estimation of approximate length of piping and hoses in the hydraulic system may vary considerably more than 20% as the interfacing system may be very large.
Hydraulic fluid, pipe and hose roughness, and minor losses before the regulator have little impact on BOP closing time. However, roughness and viscosity cannot be neglected, since the model is dependent on these values for the pipe friction pressure loss calculations.
The model is insensitive to minor losses upstream of the regulator, showing that small minor losses may practically be neglected from the model. The types of minor losses that are found to be insignificant are fully open ball valves, elbows, unions and tee fittings.

Table 6: Change in closing time when varying parameters

	-20%	Base case	+20%
Minor losses before reg.	0	0	0
Roughness	-0.04	0	0.04
Viscosity	-0.06	0	0
Pipe length	-0.33	0	0.33
Movement pressure	-0.38	0	0.37
Shear point change	-1.53	0	N/A[a]
Shear pressure	-2.12	0	N/A[b]
ID of hose to BOP	2.63	0	-0.83
Minor losses after reg.	-2.35	0	3.38

[a] Shear would not be possible
[b] Outside BOP FVR, shear would not be possible

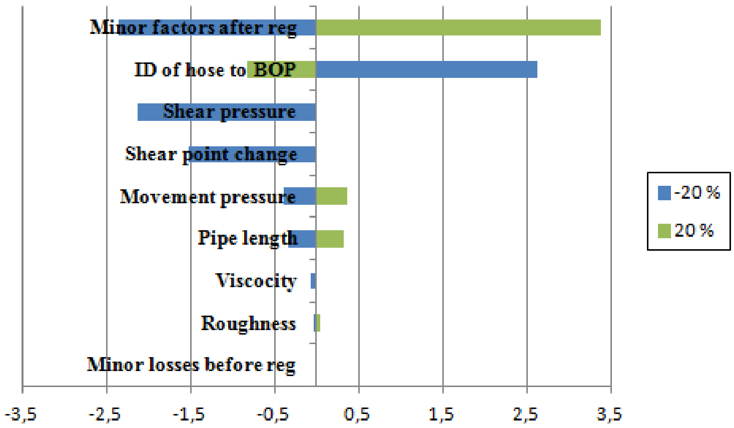


Figure 19: Ranked Tornado chart of relative change in closing time.

4.2 System model without DP and calibration

This section will discuss the process of, and the results found from modelling and calibrating the model without a DP in comparison to a prototype test [45].

4.2.1 Methodology

A model without a DP present in the bore is created since this is the simplest closing operation. Calibration is then carried out by matching the closing time of the model to an equivalent prototype. By doing this the systematic errors will be mitigated, which are not due to DP buckling and shearing. Allowing the mitigation of risk as well as modelling BOP closing times without DP.

From FAT tests the back pressure was observed to be almost constant with a known closing time [45, 47]. From this the model presented in section 4.1 is reused, where only the BOP back pressure is changed to a constant 300 psi throughout the closing operation.

Calibration is performed by simply multiplying a calibration factor with the models most sensitive (and uncertain) parameter. As this parameter is the most sensitive, the likelihood of an error in this parameter is considered more likely.

Magnitude of the calibration factor is selected by running the new model and comparing the time-estimate with the time measured during the prototype test. If there is an error[13] the calibration factor is changed in order to compensate for this error. This is repeated until the time-estimate that the model predicts matches the time measured from the prototype.

The parameters in the model are calibrated by multiplying them with the calibration factor. This is illustrated in the equation below, where γ is the calibration factor

$$\Delta P = P_{\text{stat}} + P_{\text{kin}} + P_{\text{pipe}} + \gamma P_{\text{minor}}$$

The steps used to present a model without a DP and calibrating it are:

1. Establish an equivalent model for the system without DP.
2. Calculate closing time of model.
3. Perform sensitivity analysis.
4. Calibrate model by changing the most sensitive parameter.
5. Calculate closing time of calibrated model.
6. Perform sensitivity analysis of calibrated model.

[13] 0.01 seconds inaccuracy is used.

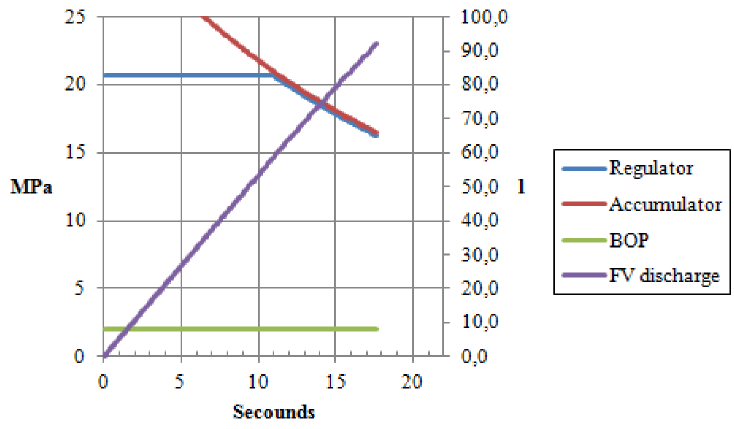


Figure 20: Pressure curves after calibration and without DP.

4.2.2 Results

Without calibration using 262 pressure stages (pressure intervals of 10 psi), an expected closing time of 17.66 sec was found. This is within 1% deviation from prototype value. Sensitivity analysis was performed, and the most sensitive parameter with uncertainty connected to it was chosen to be calibrated. The selected parameter to be calibrated was minor losses in the model. The ID of the hose is not selected for calibration, as the hose ID was not in doubt.

A calibration factor equal to $\gamma = 0.98$ proved to be sufficient.

When re-run with the calibration factor considered, the model estimates a closing time of 17.50 sec.

It should be noted that $\gamma = 0.98$ represents a 2% reduction of the calculated minor losses in the system. As the minor losses represent, at maximum flow rate, 77.3% of total back pressure (including back pressure from BOP) this would constitute a calibration of total back pressure of 2.5%.

$$2.5\% = \frac{1-\gamma}{\frac{\text{Calibrated minor losses}}{\text{Differential pressure}}}$$

A plot of the pressure curves from the model is illustrated in figure 20.

Initially the calibration was performed using only 3 intervals. This was increased to 14 and again later to 262 intervals. It was found that increasing the number of pressure stages had a noticeable impact on results predicted and the calibration factor magnitude. As the number of intervals was increased the amount of calibration needed became less. When moving from 3 to 14 intervals the model results was significant, while moving from 14 to 262 intervals the change in closing time was less pronounced.

Table 7: Relative change in closing time with varying parameters before calibration.

Parameter	-20%	Base case	+20%
Minor losses before reg	0	0	0
Roughness	-0.03	0	0.04
Viscosity	-0.03	0	0.04
Movement pressure	-0.21	0	0.22
Pipe length	-0.25	0	0.25
Minor factors after reg	-1.64	0	1.52
ID of hose to BOP	2.05	0	-0.62

Table 8: Relative change in closing time with varying parameters after calibration.

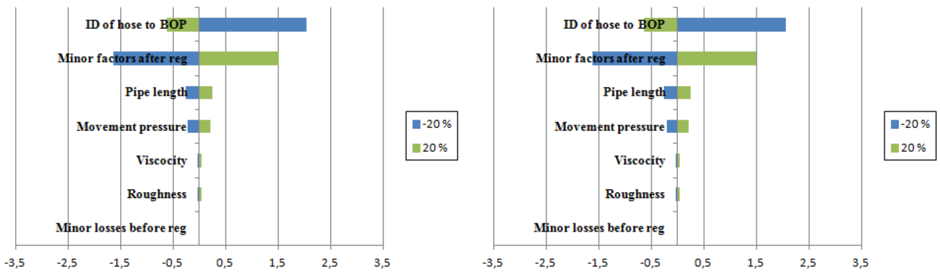
Parameter	-20%	Base case	+20%
Minor losses before reg	0	0	0
Roughness	-0.03	0	0.04
Viscosity	-0.03	0	0.04
Movement pressure	-0.2	0	0.22
Pipe length	-0.25	0	0.26
Minor losses after reg	-1.62	0	1.5
ID of hose to BOP	2.07	0	-0.63

4.2.3 Sensitivity analysis

A sensitivity analysis of the model showed some unexpected results, summarised in table 7 and 8. It is found that the hydraulic hose ID is more sensitive than the minor losses downstream of the regulator. This is contradictory to what was observed in section 4.1. It was expected that the minor losses after regulator would be the most sensitive parameter because of the high flow rate in the model without DP. Due to the higher flow rates it was expected that minor losses would increase more than pipe friction losses, as the friction factor would decrease as the Reynolds number increases giving a smaller friction factor.

The model was also run while neglecting the BOP backpressure completely. This showed, without calibration that run time would be 16.68 seconds, approximately one second less than actual runtime. This however, shows that even if no BOP data is available, or chosen to be neglected, a reasonable estimate for BOP closing time is found. Though being a less accurate optimistic value.

To illustrate the magnitude of the values in table 7 and 8, Tornado charts are made respectively in figure 21a and 21b.



(a) Uncalibrated (b) Calibrated

Figure 21: Ranked Tornado chart of model without DP.

4.3 Calibrated system model shearing a DP

The calibrated model presented in this section is unchanged from the original model described in section 4.1 apart from the addition of a calibration factor from section 4.2.

The use of a calibration factor would only be to mitigate some of the systematic errors related to the closing of the BOP, and does not account for any of the effects related to the shearing operation.

4.3.1 Results after calibration

Using a large amount of intervals (262) the closing time was found to be 21.20 seconds. This is a reduction from 21.44 seconds for the un-calibrated model. The reduction was expected as the pressure losses were reduced by the calibration factor, thus allowing for higher flow rates. On the other hand the results show that as such a small amount of calibration was necessary, there would be little change in model results before and after calibration.

A plot of the pressure curves calculated by the model is illustrated in figure 22. When figure 22 is compared to figure 17 it can be seen that the changes to the pressure curves are unnoticeable.

4.3.2 Sensitivity analysis

With regard to the models sensitivity, the calibration had an impact. Since the calibration factor is $\gamma < 1$, the model sensitivity from the pressure losses (which is calibrated) will be reduced. This is due to the reduction of minor losses, which reduces the parameters relative size in comparison to other pressure loss contributors. This makes the model less sensitive to changes in minor losses, as can be seen in table 9. It needs to be emphasised that the main purpose of the calibration factor is to adjust the model to mitigate systematic errors, not to change model parameter sensitivity.

To illustrate the magnitude of the parameters sensitivity in the model, a Tornado chart is made in figure 23 from the relative time changes in table 9.

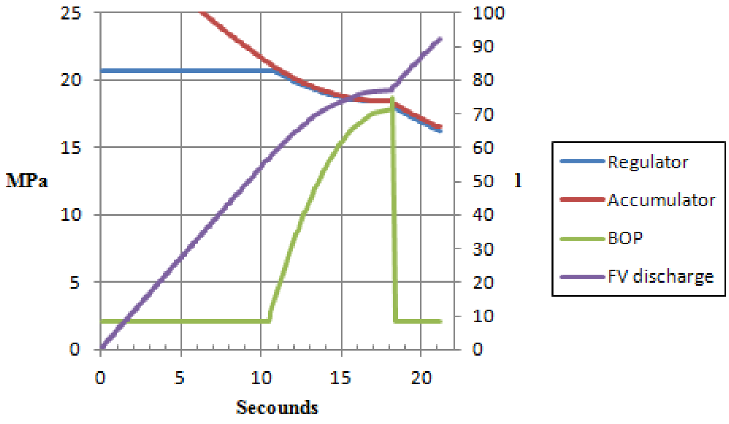


Figure 22: Pressure curves in the calibrated model with DP.

Table 9: Relative change in closing time with varying parameters, related to the calibrated model with DP.

Parameter	-20%	Base case	+20%
Minor losses before reg	0	0	0
Roughness	-0.04	0	0.03
Viscosity	-0.06	0	-0.01
Pipe length	-0.34	0	0.33
Movement pressure	-0.37	0	0.41
Shear point change	-1.47	0	N/A
Shear pressure	-2.05	0	N/A
ID of hose to BOP	2.62	0	-0.83
Minor factors after reg	-2.3	0	2.94

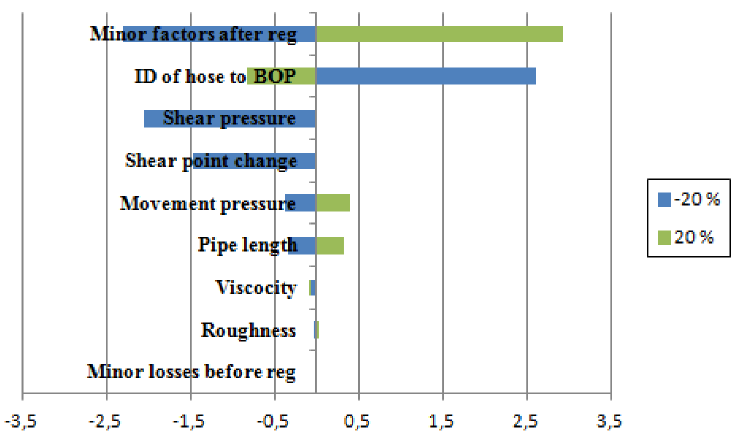


Figure 23: Ranked Tornado chart of a calibrated model with DP.

4.4 Prototype test

In October 2011 an FAT test was performed, the second of two for a BOP control system [47, 45]. This second test was done in order to assess if a blind/blind shear ram BOP would be capable of shearing a specific high quality drill pipe [47] within the limitations of the requirements designated by the standards [25, 27, 5, 24].

During the second FAT pressure readings were logged at several points in the BOP control system and one on the BOP. The pressure reading at the BOP was the back pressure of the BOP.

4.4.1 Setup

During the FAT test the BOP control system accumulators were fully charged and pumps disconnected. The accumulator banks used were interconnected to the control valve rack using hydraulic hoses. From the control valve rack long hydraulic hoses were run to the BOP. A PT was mounted, interfacing both the hydraulic hose and the BOP, in order to record the required pressure readings. These pressure readings were logged by a PLS as a function of pressure versus time.

Pressure readings at several other points in the system were recorded during the FAT. These other PTs were mounted along the lines, as well as on some of the valves. The sampling rates on these PTs were low and not archived.

Inside the BOP a drill pipe was suspended and aligned so that it could be sheared by the shear ram, illustrated in figure 24.

The BOP used is designated; Cameron Double U Ram BOP 21-1/4" 2K. SBR Rams. Large Bore Bonnet with Tandem Booster (LBT). The DP sheared is designated; 5-7/8" 26.3 ppf S135 DP 0.445" WT [47].

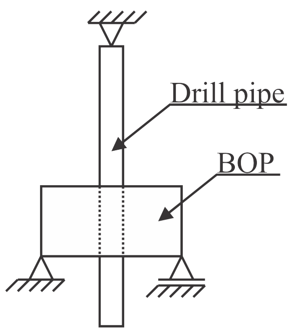


Figure 24: Drill pipe suspended in BOP

4.4.2 Test procedure

The test was initially run twice (closing and opening shear rams twice) without DP with a few minutes intervals [47].

After the second run the DP was placed inside the BOP. The shear test was performed, pressure held, and then rams opened [47]. During the shear test the pressure was supplied by both main accumulator banks[14] as well as the shear boost system[15]. A shuttle valve at the BOP controlled which accumulator bank that supplied fluid to the BOP. A shuttle valve typically contains a small ball that can move inside a cylinder, blocking the inlet of the pressure source with the lowest pressure. As a consequence the pressure source that has the highest pressure at the BOP will be the one that supplies hydraulic pressure and liquid volume.

The pressure measurements were recorded during the whole FAT.

4.4.3 Test results

All time and pressure values are read from the pressure versus time plot, made during the last FAT [47], with the exception of the first FAT [45]. Time "0" represents the point just before the first pressure change is measured, and further time values are relative to this point. Note that pressure readings and time readings are slightly rounded off.

Without drill pipe: When closing, the BOP experiences a back pressure of approximately 300 psi while moving the shear rams. This represents the necessary pressure in order to overcome pressure losses and friction in the BOP. Once moving, the pressure is almost constant throughout the closing cycle. When fully closed the pressure rapidly increases until approximately 3000 psi which is the pressure setting in the regulator on the shear boost system [22].

The pressure levels and the point in time the pressures where read, are listed below for the first closing operation [47]. Enumeration corresponds to the numbering seam in figure 10.

1. Time 0 sec, hydraulic pressure is approx 0 psi.

2. Time 1 sec, hydraulic pressure is approx 300 psi.

3. Time 17.5 sec [45], hydraulic pressure is approx 300 psi.

4. Time $>$17.5 sec, hydraulic pressure is approx 3000 psi.

With drill pipe: When closing the BOP the pressure will also be approximately 300 psi while rams are moving. When the cutting edge comes in

[14] Main accumulator banks are pressure regulated down to 1500 psi.

[15] A dedicated function in a BOP control system for closing blind/blind shear rams in a BOP.

contact with the drill pipe there will be a nonlinear[16] rise in pressure. The pressure will increase to a maxima where the pipe was sheared through.

After the pipe is sheared the pressure in the actuator again drops. This time to approximately 360 psi until fully closed. When BOP is fully closed the pressure will again rapidly increase until the pressure reaches accumulator pressure or limiting pressure by regulator

The measured pressure levels as well as the point in time when the pressures are read are listed below. Enumeration corresponds to the numbering scheme in figure 13 on page 23. The actual pressure curve measured from the FAT is illustrated in figure 25.

1. Time 0 sec, hydraulic pressure is approx 0 psi.
2. Time 1 sec, hydraulic pressure is approx 300 psi.
3. Time 12 sec, hydraulic pressure is approx 300 psi.
4. Time 20 sec, hydraulic pressure is approx 2740 psi.
5. Time 21 sec, hydraulic pressure is approx 360 psi.
6. Time 22 sec, hydraulic pressure is approx 400 psi.
7. Time >22 sec, hydraulic pressure is approx 3020 psi.

4.4.4 Summary

The FAT test was a success, shearing DP and sealing within the required time constraint of 30 seconds [25, 27, 5, 24]. The total closing time without pipe was approximately 17.5 seconds, and closing time with DP was approximately 22 seconds. This shows that the closing time increases noticeably when shearing a DP.

The PT at the BOP stack was recorded with a high sampling rate and archived after FAT completion. PT (and FT) modules mounted elsewhere in the hydraulic system were recorded but, after FAT completion, not archived.

It should be noted that the success of the shearing operation was a surprise for the observers, as it was expected that available pressure would not be sufficient to shear the pipe.

[16]It is worth mentioning that though un-linear it is approximately linear with little error when plotted in a pressure versus time diagram as seen in figure 25.

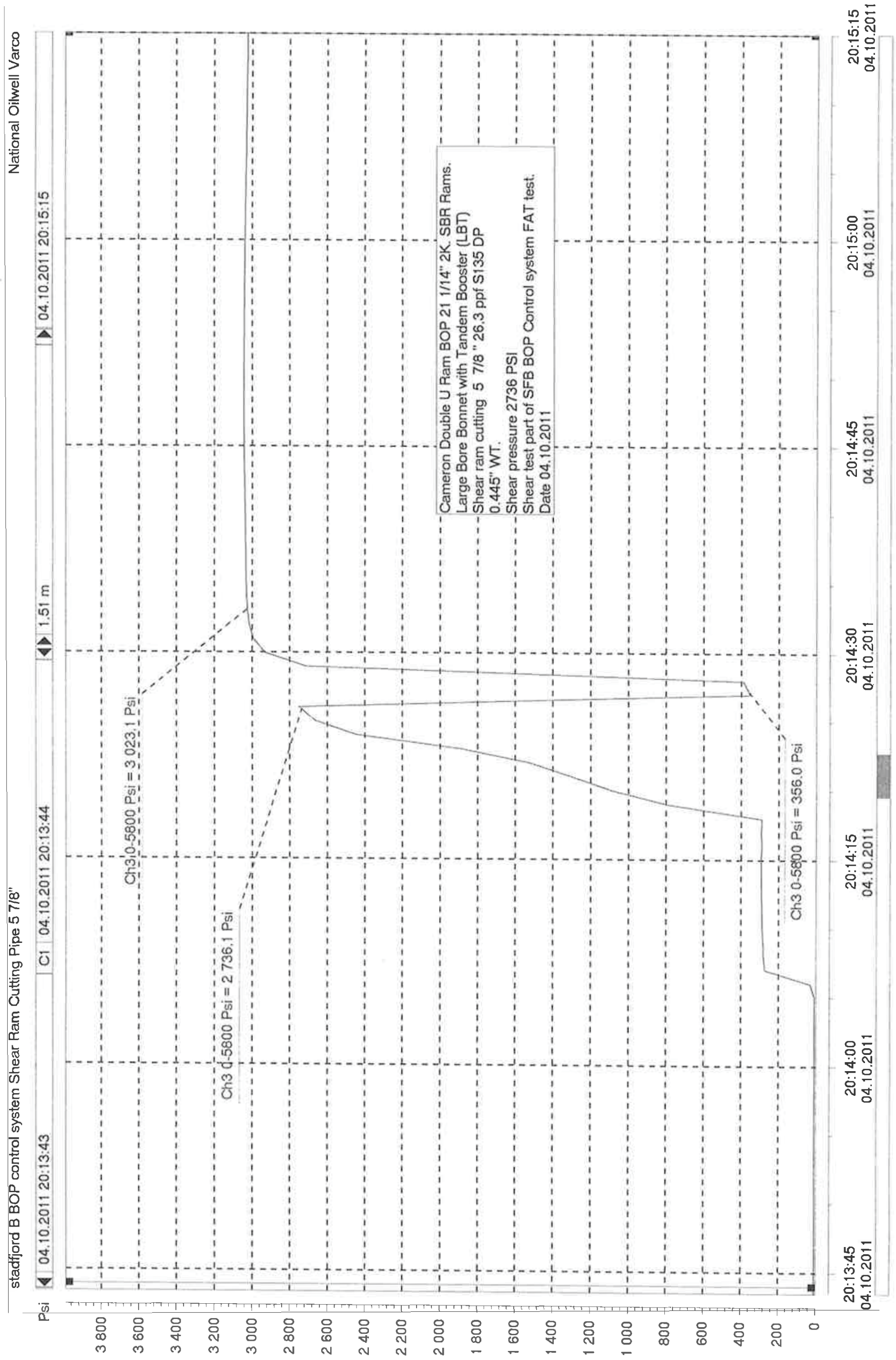


Figure 25: Prototype PT measurements [47].

4.5 Results

In this section the results found from the models and that of the prototype are discussed. As previously mentioned it is important to emphasise that some data (P_a, P_{shear}) was acquired from the prototype test and applied in the models.

4.5.1 Models in comparison to prototype

The initial models used only loss factors to accommodate for the minor losses in the system. This was found to cause significant errors when compared to the prototype test. However when flow coefficients/factors from data sheets of the prototype components were used[17], model estimates became much more accurate.

Model without shearing the DP described in section 4.2 showed, even without calibration, that it was able to estimate the closing times with good accuracy, with an error of less than half a second shorter closing time than prototype. These are considered good estimates that should be sufficient for relevant applications.

Models that account for shearing a DP were less successful, as a result of the negative differential pressure. Even though model and prototype closing times are similar, the model indicates that closing operation would not be successful.

It should also be emphasised again that when an adiabatic expansion of the gas in the accumulators is assumed, the BOP control system would not be able to supply sufficient pressure to shear the DP. Since there was a successful shearing operation in the prototype, actual behaviour of the gas in the accumulator must have experienced a polytrophic expansion. Therefore the assumption of adiabatic behaviour could indicate a too conservative assumption for this application or be seen as a safety factor.

Pressure curves in figure 18 and 22 do not fully match what was measured pressure from the prototype shown in figure 13.

When comparing models shearing DP with prototype, it is evident that more time elapses before the pressure curve rises in the prototype. This indicates that the ram travels further before it touches the DP. This is further emphasised when looking at the pressure curves after the shearing, where the prototype requires a shorter duration before BOP is fully closed than in the model.

In the prototype test the pressure curve was found to be almost linear, whilst in the model it was curved. This indicates that the back pressure from the BOP does not behave linearly, during the contacting of the DP to the shearing is complete, as what is assumed in the model. Instead, the shearing

[17] For low pressure loss components such as fully opened ball valves this is not so critical.

of the DP follows a higher order function, where the back pressure will increase at an increasing rate as the shear point is approached.

This is explained by the behaviour of the DP inside the BOP when it is closing. Initially the DP would start deforming inside the BOP, but as the DP continues to deform the cross section that needs to be sheared will increase in order for the rams to continue to move, resulting in an increasing back pressure.

4.5.2 The effects of calibration

The parameters calibrated are the minor losses in the system. This parameter was selected because of its high sensitivity in addition to the uncertainty regarding the accuracy of parameter values.

Very little calibration was necessary in order to make model fit the prototype results, and may even have been neglected without large added inaccuracies. Even though little calibration is performed, the parameters sensitivity was noticeably affected in model for shearing DP.

4.5.3 Influence from BOP back pressure

Without DP present in the BOP bore, the BOP back pressure had a noticeable effect on closing times. If the BOP back pressure been ignored, it would result in a closing time of 0.82 seconds less than actual. This is approximately 5% less than actual closing time. This indicates that the model is insensitive to BOP back pressure when DP is not present in the BOP bore, and may for rough estimates be ignored.

For closing operations when DP is present, the BOP back pressure should not be ignored, as it is crucial for deciding if closing and shearing is possible. This is further emphasised as the back pressures in such a situation are much larger, resulting in a significantly larger impact on closing times.

Considering a situation where a design depends on a BOP with limited data available, assumptions need to be made. The BOP supplier should be able to supply the analyst with the shearing pressure, as well as the volumetric requirements and the distance the blind/blind shear rams travel in the closing operation. In such a case the back pressure from moving the actuator may be assumed as a reasonable value or ignored completely with only a few seconds error.

Though it should not be used as a design parameter, the ratio between shear stress and hydraulic pressure may be used to indicate the necessary pressure in order to shear a DP for a specific BOP. Using equations 40 and 41 (assuming they are valid) the following may be concluded from the prototype:

- The necessary stress for shearing the DP is calculated to be 930.8 MPa.
- The BOP exhibited a shear stress equal to 515.6 MPa.

- Giving a stress ratio of 1.8, illustrating how beneficial BOP design allows shearing of DP at much lower hydraulic pressure.

5 Modelling guidelines

Modelling of a BOP control system can easily be performed using a spreadsheet with the aid of a simple macro. This section will describe a guide to how a model may be constructed using the spreadsheet Microsoft Excel 2007[18].

The guide is divided into three sections. The first section will describe what tables are necessary in order to construct the model. The second section describes how to setup the models functionality as well as describing the macro code required in order for the model to work. The third section describes how to assess the output information from the model.

Even though the model is meant to be generic, the one presented does have some limitations that should be noted:

- Hydraulic power sources are accumulators
- System must be adiabatic
- There is only one pressure source
- Evaluation is an iterative process
- Application limits model to 1048576 intervals

5.1 Input

At least two spread sheets need to be created, named *Model* and *Result*.

All necessary data required to set up a model are summarised in six (6) tables;

- Model table, table 10, must be placed on a spread sheet named Model in the appropriate cells.
- Regression data, table 11.
- Boundary conditions, table 12.
- Accumulators, table 13.
- Minor losses, table 14.
- Pipe friction losses, table 15.

When entering data it is very important to be vigilant and consistent with units. Mixing units is rarely a good idea, therefore if possible always use a single system of units. Descriptions of relevant cells for each of the tables are as follows:

[18] This should not limit the modelling to this software, but is what is used by the author.

In table 10, relevant cells are:
`B2` is the total FVR by BOP function.
`B3` is the pressure in accumulators when function initiates.
`B4` is the pressure in the accumulators when function is complete. This can be calculated using procedure described in section 3.3.2.
`B5` is the step size per iteration.
`B8` refers to the cell `F4` in table 11.
`B11` refers to the cell that outputs the pressure at each interval (in this case it needs to be converted from psi to Pa in order to be consistent in units).
`B12` is the equation 37. If no regulator is present the cell is equal to `B11`.
`B13` is, the equation 42 or P_a, describing BOP back pressure. Volume discharged, used in equation 42 is found in the cell `B8`.
`B22`, `B28` is the sum of pressure losses from pipe friction in table 15 and `B23`, `B29` is the sum of minor pressure losses in table 14, upstream and downstream of a regulator respectively.
`B24`, `B30`, `B25`, `B31` are the kinetic and static pressure up and downstream of regulator respectively.

In table 11 the referenced cells are:
The range `G2:G6` corresponds to liquid volume discharged.
The range `H2:H6` correspond to the accumulator pressure in table 13 in pressure stages 1, 2", 2' and 2. The reference `D11` in cell `F4` refers to the cell in table 10.

Table 12 contains values necessary for the calculations in table 10, 13, 14 and 15.

Table 13 contains data for the accumulator gas in precharge (Stage 0), charged (Stage 1), discharge (Stage 2, 2', 2"). Note that the values that are marked with an asterisk (*) are found in gas tables[19]. Table values may be found using the algorithm described in section 3.3.2 for adiabatic gas expansion.

Table 14 and 15 show how tables may be arranged in order to calculate pressure losses for different components. Listing pressure loss contributions sequentially, from BOP control system to BOP stack, simplifies the summation of losses into the cells `B22`, `B23`, `B28` and `B29` in table 10. If there is a regulator present, a distinction between contributors before or after regulator should be made. Columns are filled with the required values or equations. Equations needed are all found in section 3.

[19] `http://webbook.nist.gov/chemistry/fluid/`

Table 10: Model setup for a system with a regulator. Example values and some functions inserted.

	A	B	C	D	E
1	Pressure stage				
2	Total FVR	92.74	l		
3	Start pressure	5000.00	psi		
4	End pressure	2400.00	psi		
5	Step size	50.00	psi		
6					
7	FVR before	1.05	l		
8	FV discharge	`=F4`	l		
9	FVR after	`=B2-B8`	l		
10					
11	Pressure accumulator	`=D11*6895`	Pa	2393.90	psi
12	Regulator	16257714	Pa	2356.9	psi
13	BOP	2068500	Pa	300	psi
14	Differential	`=B12-B13`	Pa	2057	psi
15	Pressure loss	`=B32`	Pa	2094	psi
16					
17	Flow rate	0.004717601	m^3/s		
18	Time (to next stage)	`=(B7-B9)*10^-3/B17`	sec		
19					
20	Losses				
21	Before regulator				
22	Pipe	22006	Pa	3.2	psi
23	Minor	51034	Pa	7.4	psi
24	Kinetic	181856	Pa	26.4	psi
25	Static	0	Pa	0.0	psi
26	Sum	`=SUM(B22:B25)`	Pa	37.0	psi
27	After regulator				
28	Pipe	2043740	Pa	296.4	psi
29	Minor	12371332	Pa	1794.2	psi
30	Kinetic	0	Pa	0.0	psi
31	Static	21974.4	Pa	3.2	psi
32	Sum	`=SUM(B28:B31)`	Pa	2093.8	psi

Table 11: Functions for second order regression analysis.

	F
1	`=INDEX(LINEST(G2:G6);H2:H6^{1,2});1)`
2	`=INDEX(LINEST(G2:G6);H2:H6^{1,2});1;2)`
3	`=INDEX(LINEST(G2:G6);H2:H6^{1,2});1;3)`
4	`=(F1*D11^2)+(F2*D11)+F3`

Table 12: Boundary conditions with example values.

Environment:	
Temperature	293.15 K
Liquid:	
Density	9 cSt
Static heights:	
Accumulators	0 m
Regulator	0 m
BOP	2 m
Accumulators:	
Precharge	2900 psi
Charged	5000 psi
Volume	50 l
Number	8
BOP:	
FVR (close)	24.5 US gal
Travel distance	320.2 mm

Table 13: Pressure and fluid volumes in accumulators.

	Pressure	Temp.	Density	Entropy	Volume [l]		
Stage	psia	K	lbm/ft^3	BTU/lbm R	Gas	Liq.	Dis.
0	2900	293.15	14.893*	1.2041*	400	0	0
1	5000	293.15	22.469*	1.1538*	265.13	134.87	0
2"	4000	257.45*	20.647*	1.1538	288.53	111.47	23.40
2'	3000	238.40*	18.365*	1.1538	324.38	75.62	59.25
2	2393.9*	224.20*	16.646	1.1538	357.87	42.12	92.75

Table 14: Table for calculating minor pressure losses.

#	Desc.	ID	ξ	C_v	A	v	γ	Loss	Extra pipe

Table 15: Table for calculating pipe friction pressure losses.

#	Desc.	ID	Length	ϵ	A	v	Re	$\frac{\epsilon}{\mathrm{ID}}$	$\frac{1}{\sqrt{f}}$	f	γ	Loss

5.2 Calculation

In order to perform calculations with sufficiently high accuracy, a large number of pressure intervals are needed. This is preferably performed using a macro, in order to repeat the large number of repetitive tasks necessary.

The macro below written in Visual Basic, fits a table similar to the one shown in table 10. This table needs to be placed in a sheet named *Model* using the same cells. An unformatted sheet named *Result* also needs to be created if this has not yet been done.

The macro performs the steps necessary to calculate the discharge time for each pressure stage, and places the results in the Result sheet.

```
Sub Model()
'
' Model Macro
'
' Prepare
    Application.ScreenUpdating = False
    Sheets("Result").Select
    Range("A1").Select
    Worksheets("Result").Range("A:Z"). _
        ClearContents
    Sheets("Model").Select
    Range("B2").Select
    Selection.Copy
    Range("B7").Select
    Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
        :=False, Transpose:=False
    Range("B3").Select
    Let Pressure = ActiveCell
    Selection.Copy
    Range("D11").Select
    Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
        :=False, Transpose:=False
    Range("B4").Select
    Let MinPressure = ActiveCell
    Range("B5").Select
    Let ChangePressure = ActiveCell
' Start the work
    Do While Pressure > MinPressure
' Iterate
        Range("B17").Select
        ActiveCell = 0.00001
```

```
        Range("B15").GoalSeek Goal:=Range("B14"),
            ChangingCell:=Range("B17")
' Copy values
        Range("B7:B32").Select
        Selection.Copy
        Sheets("Result").Select
        Selection.PasteSpecial Paste:=xlPasteValues,
            Operation:=xlNone, SkipBlanks _
            :=False, Transpose:=True
        ActiveCell.Offset(1, 0).Range("A1").Select
        Sheets("Model").Select
' Reduce pressure one increment
        Pressure = Pressure - ChangePressure
' Prepare for next iteration
        Range("B9").Select
        Selection.Copy
        Range("B7").Select
        Selection.PasteSpecial Paste:=xlPasteValues,
            Operation:=xlNone, SkipBlanks _
            :=False, Transpose:=True
        Range("D11").Select
        ActiveCell = Pressure
    Loop
' Final iteration
    Range("D11").Select
    ActiveCell = MinPressure
    Range("B17").Select
    ActiveCell = 0.00001
    Range("B15").GoalSeek Goal:=Range("B14"),
        ChangingCell:=Range("B17")
    Range("B7:B32").Select
    Selection.Copy
    Sheets("Result").Select
    Selection.PasteSpecial Paste:=xlPasteValues,
        Operation:=xlNone, SkipBlanks _
        :=False, Transpose:=True
    ActiveCell.Offset(1, 0).Range("A1").Select
    Sheets("Model").Select
    Application.ScreenUpdating = True
End Sub
```

5.3 Assessing model output

When the macro has completed, the results in the cells `B7:B32` in table 10 are listed transpose in the columns `A:Z` on the sheet Results.

For models experiencing large back pressures, there may be some negative differential pressures. These situations must be treated per case, since negative differential pressures means that the BOP control system would not be able to supply sufficient pressure to complete the function.

After running the model, the following codes may be placed in arbitrary cells in order to output the following information:

`=COUNT(Result!A:A)` Counts the number of intervals in the model.

`=SUMIF(Result!L:L;">0")` Totals the positive times, giving the closing time of the BOP.

`=COUNTIF(Result!H:H; "<=0")` If not zero (0) closing operation will be unsuccessful, which is a consequence of a negative differential pressure.

6 Discussion

This section will describe a general discussion of the experiences from modelling BOP control systems interfacing a blind/blind shear BOP.

6.1 Accuracy

It is found that high accuracy may be achieved when modelling BOP control systems when relevant system parameters are described. This is especially evident when modelling the closing of a blind/blind shear ram BOP when no DP is present. Even without calibration, the model is found to have less than 1% deviation from prototype closing time. Note that this closing time is found using back pressure values from the prototype.

If the results from the prototype are neglected from the model (back pressure assumed zero) the deviation from prototype closing time increases to almost 5% deviation from prototype.

Model accuracy is reduced when modelling BOP control systems with a DP in the BOP bore. When applying back pressure data from the prototype, an un-calibrated model has less than 3% deviation from prototype closing time. Inaccuracies are further emphasised since pressure curves in the model and prototype do not match. Also the model does not always agree with the prototype if closing would be successful when DP is present. This phenomenon is explained in section 4.5.1.

For a model with DP present, the BOP back pressures shall not be neglected. In such situations it is necessary to input the expected hydraulic pressure for shearing, in order to give any useable outputs.

6.2 Calibration and sensitivity

The parameters in the model found to be the most sensitive, are the minor losses and the internal diameter of piping/hose interfacing the BOP control system. As there is a lower degree of confidence to the accuracy of the minor losses, these are selected to be calibrated.

As the models are found to fit well with the prototype results, there is a very limited amount of calibration preformed. A reduction of minor losses by 2% proved to be sufficient in order to calibrate the model without DP.

The impact that the calibration had on the closing time of the BOP proved to be limited. Calibration mitigated the 1% inaccuracy in the model without DP, while for model with DP, inaccuracies are increased to less than 4% as closing time was reduced slightly.

Sensitivity of the calibrated parameter in the model is reduced, as the calibration factor is less than 1. This is not the purpose of the calibration, but beneficial for the confidence of the model. If the prototype test had been slightly slower, the calibration would have had a negative impact on parameter sensitivity.

Some parameters are found to have little impact on BOP closing time:

- Minor losses with a small loss factor such as fully open ball valves, which may be neglected.
- Roughness of internal pipe walls may be assumed a value with little impact on the model result.
- Viscosity has only a small impact on the model and may be assumed an approximate value from respective data sheets.

6.3 Applicability

The pressures calculated from the model shows that it is fully applicable for modelling the closing operation of a BOP without a DP when pressure source is an accumulator bank.

When modelling a BOP control system with a DP in the BOP bore, the model becomes less reliable, which is illustrated by the pressure trends found during closing. This is likely due to the incorrect assumptions made when modelling the back pressure from the BOP during buckling and shearing.

The assumption of adiabatic real gas expansion is likely not a fully accurate model of a BOP control system behaviour, since closing would be polytrophic in nature. On the other hand the assumption of adiabatic expansion is fully applicable and recommended practice of design standards [25].

6.4 Suggested improvements and challenges

There are primarily four areas that would benefit from improvements in the model:

- Integrating accumulator sizing calculations into the model.
- Implementing a method for quickly making the model fine-tune BOP control system performance.
- Being able to account for other pressure sources by defining the models intervals in liquid discharged, rather than pressure change.
- It would be beneficial if the model required less user inputs, such as updating key cells, whilst the system being modelled is modified.

The challenges here are primarily centred on:

- Table 2 needs to be updated when accumulators are changed. This may be avoided by using the equation below [8, 35], if neglecting influence from pressure and temperature ($n = 1.4$) [35].

$$V_0 = \frac{\Delta V}{\left(\frac{P_0}{P_1}\right)^{\frac{1}{n}} - \left(\frac{P_0}{P_2}\right)^{\frac{1}{n}}}$$

- Which losses that are up or downstream of regulators which need to be given special consideration.

- BOP back pressures are dependent on the particular BOP and DP used.

- Key cells need to be manually updated when system is changed.

7 Conclusion

A method for modelling BOP control systems is presented in this paper.

The models are found to have a sufficiently high accuracy for possible applications mentioned in Table 16.

When modelling the closing time for blind/blind shear ram BOPs without a DP present in the bore, results are found to be sufficiently accurate for proposed applications even without prototype back pressure measurements. When a DP is present accuracy is reduced, and additional information about BOP back pressures are required in order to produce usable results.

Since model results fit well with the prototype, only a small amount of calibration was preformed which had very little impact on closing times. In future models, where the BOP control systems are well defined (hydraulic components are known), calibration would be unnecessary.

In order to reproduce the models a guideline is included. This guide is applicable for many BOP control system configurations under normal operation. The model and guidelines are formed in accordance to standard requirements [25].

There is a range of applications for modelling BOP control systems, both during design and/or operation which are listed in Table 16.

Table 16: Direct application of modelling.

Design	Operation
-Verifying performance when installed at location. -Verifying performance of final design. -Assessing performance impact from component selection. -Aid in fine tuning of system performance.	-Verifying BOP will perform adequately. -Verifying that the DP is shearable.

Several lessons have been learned during the development of this paper where the most important are:

- The use of loss factors for minor losses without using accurate flow coefficients from component datasheets resulted in very large errors.
- Flow coefficients may be used (in conjunction with equation 17) instead of loss factors to estimate pressure losses.
- Some of the necessary data for BOP back pressures are difficult to acquire without measuring directly.

- Performance of the BOP control system is dependent on the interfacing systems. This gives the operator of the system the added responsibility to define the interfacing systems more accurately in order for BOP control system to perform as expected.

Recommendation for further work would be:

- Improve the BOP back pressure modelling during buckling and shearing of DP.
- Reduction of the amount of user inputs that are necessary.
- Expanding the model to account for several and/or different types of pressure sources.

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A Appendix

This appendix will list values and sources for minor loss coefficients, roughness values, and hydraulic fluid characteristics.

A.1 Minor loss coefficients

In this section a summary of the different tables for minor loss coefficients used in the paper are shown.

Table 17: Loss factors[41]

Fitting	ξ
Unions	0.04
Elbows (bends):	
45^o	0.4
90^o	0.9
180^o	0.9
Tee:	
a long run	0.40
a long branch	1
Globe Valve:	
fully open	10
half open	20
Gate Valve:	
fully open	0.3
half open	5

Table 18: Loss factors [31, pp. 61–63]

Component	ξ
Large radius bend	0.25
Short radius bend	1.2
Tee a long run	0.15
Tee a long branch	1.3
Seat check valve	2.5
Sliding valve	2.8-16

Table 19: Loss factors. Illustrations are not reproduced [1]

Component	ξ
Bend 90^o:	
$r > 4d$	$\xi \simeq 0.2$
$r = d$	$\xi \simeq 0.4$
Bend 180^o	$\xi \simeq 2 \times 90^o$
Branching:	
A long run	$\xi \simeq 0.1$
Branch out	$\xi \simeq 0.9$
Tee:	
A long run	$\xi \simeq 0.4$
Branch in	$\xi \simeq 0.2$
Sudden expansion:	
$\frac{d_2}{d_1} = 1.5$	$\xi = 0.3$
$\frac{d_2}{d_1} = 2$	$\xi = 0.6$
$\frac{d_2}{d_1} = 2.5$	$\xi = 0.7$
$\frac{d_2}{d_1} = 10$	$\xi = 1$
Sudden contraction:	
$\frac{d_2}{d_1} = 1$	$\xi = 0$
$\frac{d_2}{d_1} = 0.8$	$\xi = 0.2$
$\frac{d_2}{d_1} = 0.6$	$\xi = 0.3$
$\frac{d_2}{d_1} = 0.4$	$\xi = 0.4$
Check valve:	
Flap	$\xi \simeq 1 - 0.4$
Seat	$\xi \simeq 8 - 1$
Ball	$\xi \simeq 2 - 0.5$
Standard valve:	
Gate	$\xi \simeq 0.2$
Seat	$\xi \simeq 3$
Ball	$\xi \simeq 0.1$

Table 20: Loss factors [40]

Type of Component or Fitting	Minor Loss Coefficient ξ
Tee, Flanged, Line Flow	0.2
Tee, Threaded, Line Flow	0.9
Tee, Flanged, Branched Flow	1.0
Tee, Threaded , Branch Flow	2.0
Union, Threaded	0.8
Elbow, Flanged Regular 90^o	0.3
Elbow, Threaded Regular 90^o	1.5
Elbow, Threaded Regular 45^o	0.4
Elbow, Flanged Long Radius 90^o	0.2
Elbow, Threaded Long Radius 90^o	0.7
Elbow, Flanged Long Radius 45^o	0.2
Return Bend, Flanged 180^o	0.2
Return Bend, Threaded 180^o	1.5
Globe Valve, Fully Open	10
Angle Valve, Fully Open	2
Gate Valve, Fully Open	0.15
Gate Valve, 1/4 Closed	0.26
Gate Valve, 1/2 Closed	2.1
Gate Valve, 3/4 Closed	17
Swing Check Valve, Forward Flow	2
Ball Valve, Fully Open	0.05
Ball Valve, 1/3 Closed	5.5
Ball Valve, 2/3 Closed	200
Diaphragm Valve, Open	2.3
Diaphragm Valve, Half Open	4.3
Diaphragm Valve, 1/4 Open	21
Water meter	7

Table 21: Loss factors [13]

Component	ξ
Globe valve, wide open	10
Angle valve, wide open	5
Close-return bend	2.2
Through tee side outlet	1.8
Short-radius elbow	0.9
Medium-radius elbow	0.75
Long-radius elbow	0.60
45^o elbow	0.42
Gate valve, wide open	0.19
Gate valve, half open	2.09

A.2 Roughness values

In this section a table is compiled in order to describe roughness values of pipes.

Table 22: Roughness values

	[31, p. 60]	[8]	[16]
Drawn tubing	0.005-0.02	0.0015	0.03
Hydraulic hose	0.02-0.05	0.03	

A.3 Fluid properties

This section describes the hydraulic fluid used in the model as described in table 23

Table 23: Fluid properties [12]

Property	Pelagic GZ Compensator/Tensioner Fluid	Erifon 818 Compensator Fluid
Appearance	Light brown fluid	Green, slightly viscous fluid
Viscosity [cS]@		
-20 ^{o}C	70.0	750
0 ^{o}C	20	125
20 ^{o}C	9.0	
40 ^{o}C	5.8	15
Pour point [^{o}C]	$<$-30	$<$-45
Upper temperature stability limit [^{o}C]	45	60
Flash point	Not applicable	None
pH@ 20 ^{o}C	8.8	9.1
Density@ 20 ^{o}C	1.12	1.08

A.4 Symbol legend

This section describes what the symbols used in the paper summarized in figure 26.

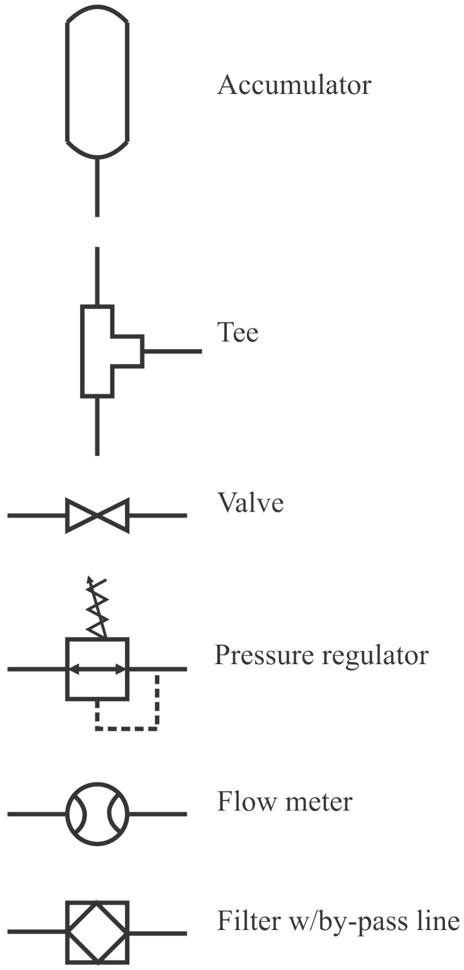

Figure 26: Symbol legend