Drillbotics – Phase I Design Report

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1 Introduction and objectives

The 2017 drillbotics™ competition marks the first year that a team from the Clausthal University of Technology, Germany is entering the competition. As the time between enrollment and the deadline was very short, intensive efforts have been undertaken by the team to offer a proposal that can compete, and simultaneously serve as a strong base for future participation in the drillbotics™ competition. First and foremost, the team would like to thank the organizers for creating the framework for us students to learn, develop and apply our skills in a multi-disciplinary, team-oriented, highly technical challenge.

Based on the 2017 Drillbotics™ competition guidelines, the objective of this year’s competition is “to design a rig and related equipment to autonomously drill a vertical well as quickly as possible while maintaining borehole quality and integrity of the drilling rig and drillstring.”. The drilling rig will be used to drill a rock sample with the size dimensions of 1x1x2 ft³ (30 x 30 x 60 cm³), containing unknown layers of rock.

Nowadays, drilling automation is gaining more interest from the oil and gas industry, equipment manufactures, and research organizations. Automating the drilling process is considered to offer safety improvements during drilling operations, less drilling time requirements, increasing accuracy in data acquisition, better well placement and quality, and a reduction of the costs. In automated drilling systems, the operating parameters are optimized by acquiring relevant data, assessing the data and adjusting the operating parameters without human interference. Ideally, the automation level reaches tier 3, which describes a stage in which the automation has evolved to decide and act autonomously.

The purpose of this proposal is to showcase a well-conceived design plan of a small scale drilling robot that incorporates important features that are essentially encountered in the field. Emphasis has been placed on implementing different and new ideas and solution in the design that are not commonly or frequently used in the conventional drilling process to fulfill the demands of drilling a fast, vertical trajectory utilizing an automated process that from a human intervention point of view only knows the activation of the start button.
The proposed drilling robot design consist of the following surface components:
A standard framework rig structure, a hoisting system that uses a non-rotating screw jack, a top-drive rotary system, and a circulation system that is powerful enough to operate a downhole percussion drilling tool.
In addition, the downhole tool design incorporates chambers for a battery to power a gyroscope, accelerometer, magnetometer and a Bluetooth device for downhole data acquisition.
Furthermore, a rotary-percussion drilling component is implemented in the bottom-hole assembly (BHA). The percussion drilling concept is offered in this proposal because of its potential to considerably increase the ROP for hard and very hard rock formations and the ability to drill a straight well trajectory without the natural tendency to turn to the right as encountered in conventional rotary drilling systems. In percussion drilling, impact, collision or vibratory shock is used to cut the rock. At this point in the design phase, it is also envisioned that the percussion action of the bit will provide further information about the hardness, fracturing of the rock and changes in formation by interpretation of the collected data that is obtained through the sensory tools placed in the BHA.

The automation system will be employed to improve and optimize drilling performance, avoid human intervention and, at the same time, maintain safety of the operation. The automation is achieved by using downhole and surface sensors and applying suitable control algorithm to self-control the drilling operation. Different types of sensors are included in the design plan to measure rotational speed and torque of the drillstring, weight on bit (WOB), pressure, displacement, and flow rate. Other sensors, located in the BHA measure vibration, magnetic field, orientation, angular velocity and acceleration. This data is transmitted via a Bluetooth device.
The data acquisition process during drilling feed the respective parameters used in the control algorithm that controls the output parameters to influence the drilling process. This part of the control algorithm is implemented on the PLC, which has an embedded real-time kernel and uses the CODESYS runtime environment.
The algorithm that is used to optimize the drilling process in terms of rate of penetration (ROP) is based on the concept of mechanical specific energy for rotary drilling as well as specific energy for percussion drilling. The approximated solution to the equation is derived using a metaheuristic, genetically mutating, algorithm.
Further, the verticality of the wellbore shall be maintained and self-correction can be made in case the drilling trajectory deviates away from the plan.

The constraints, such as the strength of the aluminum drillpipe, are set in the program via an operating envelope as a limiting factor for the drilling automation. The aluminum drillpipe is assumed to be one of the weakest part along the drillstring that will govern the maximum WOB, drillstring rotation speed (RPM), applied torque, and internal pressure. Some dysfunction problems, such as stick slip and vibration, are addressed in the design plan. The corrective action and mitigation plan for each dysfunction problem will be included in the control algorithm to support self-control once enough experience from actual test drilling has been obtained in the testing and optimization phase of the competition.

This proposal consists of several chapter as follows:

1. **Introduction and objectives**
   This chapter sets the objectives of the proposal and project. A brief description of the proposal content is explained herein.

2. **Basic calculations**
   The selection of the basic drilling parameters such as flow rate, drillstring rotation speed, pressure, and other parameters are described here. The selection and determination are based on engineering calculation and design by considering predetermined constraints. The engineering calculation results will be used as basis for the rig design and drilling automation plan.

3. **Rig Design**
   This chapter explains the rig structure design, the machine bed, the hoisting system, the sled, the hydraulic system and the top drive system.

4. **Electrical System**
   This chapter contains the electrical layout, control system, PLC, motor control, control data handling and storage.

5. **BHA Design**
   This chapter explains the bottom hole assembly (BHA) design for the senor unit.

6. **Percussion Drilling**
   The rotary-percussion drilling idea is offered in this proposal in order to improve the ROP in hard formations and furthermore to explore the potential of determining rock properties, such as hardness. The theoretical part is set in this chapter.
7. **Safety Consideration**
   Health and safety (HSE) is a main aim of every drilling operation. This chapter discusses the risks and potential harmful events that could occur during the test. Precaution and mitigation plan are set to prevent undesirable events, including the rig structure design and drilling automation system that can accommodate the safety consideration.

8. **Expense Plan**
   The requirement of equipment and tools and the expenses plan are described in this chapter.

9. **Technical drawings**
   The technical drawings provided are supposed to give an overview and are not provided as construction plans.
2 Calculations

2.1 Data and assumptions

Based on the 2017 Drillbotsics™ competition guideline, some assumptions and basic information for calculation and engineering design are summarized in Tables 1 to 5:

Table 1: Drilling Hole and rock data

<table>
<thead>
<tr>
<th>Drilling hole and rock data</th>
<th>Field unit</th>
<th>Metric unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hole diameter (d_h)</td>
<td>1.125 in</td>
<td>2.86 cm</td>
</tr>
<tr>
<td>Rock strength</td>
<td>2-5 ksi</td>
<td>13.8-34.5 MPa</td>
</tr>
<tr>
<td>Cutting concentration (C_conc)</td>
<td>1.5%</td>
<td>1.5%</td>
</tr>
<tr>
<td>Height of rock</td>
<td>2 ft</td>
<td>0.6 m</td>
</tr>
<tr>
<td>Cutting density (p_s)</td>
<td>22.07 ppg</td>
<td>2650 Kg/m³</td>
</tr>
<tr>
<td>Diameter cutting (d_s)</td>
<td>0.04 in</td>
<td>1.016 mm</td>
</tr>
<tr>
<td>ROP</td>
<td>0.8 ft/hr</td>
<td>0.24 m/hr</td>
</tr>
</tbody>
</table>

Table 2: Drilling fluid data

<table>
<thead>
<tr>
<th>Drilling fluid data</th>
<th>Field unit</th>
<th>Metric unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water viscosity (μ)</td>
<td>1 cp</td>
<td>0.001 (Pa.s)</td>
</tr>
<tr>
<td>Water density (ρ_w)</td>
<td>8.33 ppg</td>
<td>1000 Kg/m³</td>
</tr>
</tbody>
</table>

Table 3: Aluminium drillpipe data

<table>
<thead>
<tr>
<th>Aluminium drillpipe data</th>
<th>Field unit</th>
<th>Metric unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ultimate Tensile Strength</td>
<td>15594 psi</td>
<td>110 MPa</td>
</tr>
<tr>
<td>Yield strength (Ys)</td>
<td>13779 psi</td>
<td>95 MPa</td>
</tr>
<tr>
<td>Modulus of elasticity (E)</td>
<td>10x10^6 psi</td>
<td>69 GPa</td>
</tr>
<tr>
<td>Weight</td>
<td>4.7 lb/ft</td>
<td>0.07 Kg/m</td>
</tr>
<tr>
<td>Outside diameter (dp)</td>
<td>0.375 in</td>
<td>9.53 mm</td>
</tr>
<tr>
<td>Outside radius (r_o)</td>
<td>0.1875 in</td>
<td>4.76 mm</td>
</tr>
<tr>
<td>Inside diameter (id_p)</td>
<td>0.305 in</td>
<td>7.75 mm</td>
</tr>
</tbody>
</table>
13

Table 4: Stabilizer/downhole BHA data

<table>
<thead>
<tr>
<th>Stabilizer/downhole BHA data</th>
<th>Field unit</th>
<th>Metric unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside diameter (d_{ol})</td>
<td>0.8 in</td>
<td>20.3 mm</td>
</tr>
<tr>
<td>Inside diameter (id_{ol})</td>
<td>0.6 in</td>
<td>15.2 mm</td>
</tr>
<tr>
<td>Wall thickness (t)</td>
<td>0.1 in</td>
<td>2.54 mm</td>
</tr>
<tr>
<td>Length (L_{dp})</td>
<td>3.5 in</td>
<td>8.9 cm</td>
</tr>
<tr>
<td>Roughness</td>
<td>0.0006 in</td>
<td>0.0152 mm</td>
</tr>
</tbody>
</table>

Table 5: Bit data

<table>
<thead>
<tr>
<th>Bit data [DSATS provided]</th>
<th>Field unit</th>
<th>Metric unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bit diameter</td>
<td>1.125&quot;</td>
<td>28.6 mm</td>
</tr>
<tr>
<td>Nozzle diameter</td>
<td>0.093 in</td>
<td>2.35 mm</td>
</tr>
<tr>
<td>Discharge coefficient</td>
<td>0.95</td>
<td>0.95</td>
</tr>
</tbody>
</table>

2.2 Limit calculations

The aluminium drill pipe can be considered one of the weakest part in the entire string next to a connection. The following calculated values for buckling, burst and torsion maxima’s, are based on the scenario that no other forces are acting on the component for the calculation. The maximum values are good indicators, aiding in the selection of the appropriate equipment and parameters for the drilling robot, such as the maximum WOB for the hoisting system, maximum torque value for the top drive motor, and the feasibility of high pump pressures for the percussion system.
2.2.1 Buckling limit calculation

Buckling is characterized by a lateral deformation or failure of a structural member subjected to high axial compressive stress, where the compressive stress at the point of failure is less than the yield strength that the material can withstand. The critical buckling load limit of the aluminium drill pipe is calculated by the following Euler equation (assuming both of the pipe ends are pinned) [1]:

First the moment of inertia, $I$, is determined:

$$ I = \frac{\pi}{64} (d_p^4 - id_p^4) = \frac{\pi}{64} (0.375^4 - 0.305^4) = 0.000546 \text{in}^4 \quad (2.3 \times 10^{-10} \text{m}^4) $$

Where:

- $d_p$: Outside diameter of the drillpipe
- $id_p$: Inside diameter of the drillpipe

Then the critical buckling load, $P_{cr}$, is calculated:

$$ P_{bcr} = \frac{\pi^2 \times E \times I}{(K \times L)^2} = \frac{\pi^2 \times 10000000 \times 0.000546}{(1 \times 36)^2} = 41.6 \text{ lb f} \quad (185.1 \text{ N or 18.9 kg}) $$

Where:

- $P_{bcr}$: Critical buckling load
- $E$: Modulus elasticity of the aluminium drill pipe
- $I$: Area moment of inertia
- $L$: Length of the column
- $K$: Column effective length factor

Based on the scenarios of buckling failure, there are several recommendations in respect to the effective length factor ($K$). The variation of effective length factor is used to estimate the buckling load limit.
**Figure 1. Design value K variation [2, 1]**

<table>
<thead>
<tr>
<th>K variation</th>
<th>Buckling load limit (lbf)</th>
<th>Buckling load limit (N)</th>
<th>Buckling load limit (Kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>166.3</td>
<td>740.3</td>
<td>75.4</td>
</tr>
<tr>
<td>0.7</td>
<td>84.8</td>
<td>377.7</td>
<td>38.5</td>
</tr>
<tr>
<td>1</td>
<td>41.6</td>
<td>185.1</td>
<td>18.9</td>
</tr>
<tr>
<td>2</td>
<td>10.4</td>
<td>46.3</td>
<td>4.7</td>
</tr>
</tbody>
</table>

### Table 6: Buckling limit according to K variation

#### 2.2.2 Burst limit calculation

Assuming the yield strength, of the aluminium drill pipe is 13779 psi (95 MPa) and a safety factor 1.5, the burst limit, $P_{burst}$, of the aluminium drill pipe can be estimated by following equation (Barlow equation) [3]:

$$P_{burst} = \frac{2 * Y_p * t}{d_p * S_f} = \frac{2 * 13779 * 0.035}{0.375 * 1.5} = 1714.7 \text{ psi (118.2 bar)}$$

Where,

$Y_p$: Yield strength of the drillpipe

$t$: Wall thickness of the drillpipe


2.2.3 Torsional limit calculation

Assuming the maximum yield stress of the aluminium drill pipe is 13779 psi (95 MPa), the maximum limit of torque, \( T_{max} \), of the aluminium drill pipe can be estimated by following equation:

\[
T_{max} = \frac{\pi}{16} \sigma_{max} \left( \frac{d_p^4 - id_p^4}{d_p} \right) = \frac{\pi}{16} \times 13779 \times \left( \frac{0.375^4 - 0.305^4}{0.375} \right)
\]

\[
= 72.9 \text{ in.} \text{lb} \text{f} \quad (8.2 \text{ N.m})
\]

Where,
\( d_p \): Outside diameter of the drillpipe
\( id_p \): Inside diameter of the drillpipe
\( \sigma_{max} \): Yield strength of the drillpipe

2.2.4 Stress envelope with van Mises

As mentioned above, the maximum allowable stresses were calculated without considering the other loads during drilling. A program was written with Matlab™ to consider the combinations of stress states that might occur in the aluminium drillpipe during regular, vertical drilling as a function of the drilling parameters: pump pressure, WOB, and Torque. Figure 2 shows the stresses acting in a thick-wall cylinder (The Lame’s solution defines the magnitude of the stresses).

Figure 2. Stresses acting in a thick-wall cylinder [1]
The considered stresses are:

1. Axial stress, $\sigma_a$, (only compression due to WOB is considered in this case even though other possibilities exist):

$$\sigma_a = \frac{WOB}{A}$$

2. Radial stress, $\sigma_r$, and tangential stress $\sigma_t$ (due to a pressure difference $\Delta p$, across the cylindrical shell of the drill pipe) [4]. For the radial stress, $\sigma_r$:

$$\sigma_r = \frac{P_o * r_o^2 - P_i * r_i^2}{r_o^2 - r_i^2} + \frac{r_o^2 * r_i^2}{(r_o^2 - r_i^2) * r^2} * (P_i - P_o)$$

The following formula is used to calculate the tangential, $\sigma_t$:

$$\sigma_t = \frac{P_o * r_o^2 - P_i * r_i^2}{r_o^2 - r_i^2} - \frac{r_o^2 * r_i^2}{(r_o^2 - r_i^2) * r^2} * (P_i - P_o)$$

The radial and tangential stresses will be at the highest value at the internal wall of the drill pipe ($r=r_i$) [1]. The following formula is used to calculate the shear stress, $\tau$, due to torque, $T$:

$$\tau = \frac{T * r}{J}$$

Where the polar moment of inertia, $J$, is defined as:

$$J = \frac{\pi}{32} (d_p^4 - id_p^4) = \frac{\pi}{32} (0.375^4 - 0.305^4) = 0.0010919in^4 = 4.5 * 10^{-10} m^4$$

Where:

- $A$: Area of steel drillpipe (OD-ID)
- $P_o$: External pressure (atmospheric pressure, considered as no external pressure)
- $P_i$: Internal pressure (estimated from standpipe pressure)
- $r$: Radius of the drillpipe
- $r_o$: Outside radius of the drillpipe
To investigate the drillpipe failure mechanism, the well proven concept of the von Mises criteria, $V_m$, is applied [1]. The combination of stresses during drilling should not exceed the yield strength of the material, otherwise failure occurs.

\[
V_m = \sqrt{\frac{1}{2} \left( (\sigma_t - \sigma_r)^2 + (\sigma_t - \sigma_a)^2 + (\sigma_a - \sigma_r)^2 + 3 \tau^2 \right)} < Y_s
\]

\[
\tau = \sqrt{\frac{1}{3} \left( 2 Y_s^2 - (\sigma_t - \sigma_r)^2 - (\sigma_t - \sigma_a)^2 - (\sigma_a - \sigma_r)^2 \right)}
\]

$V_m$: Von Mises failure criteria  
$\sigma_t$: Tangential stress  
$\sigma_r$: Radial stress  
$\sigma_a$: Axial stress  
$Y_s$: Yield strength  
$\tau$: Shear stress

Based on the calculated values with Matlab, the maximum allowed torsion is 65.5 in.lbf (7.40 Nm). The shear stress caused by the torque is:

\[
\tau = \frac{T \cdot r_p}{J} = \frac{65.5 \times 0.1875}{0.0010919} = 11247.6 \text{ psi} \quad (775.5 \text{ bar or } 77.55 \text{ Mpa})
\]

Where,
$T$: Torque
The safety factor, $SF$, of the drillpipe due to torsional stress can be estimated by the following calculation:

$$SF = \frac{Y_s}{\tau} = \frac{13779}{11247.6} = 1.23$$

The safety factor is considered to be sufficient for the requirements of drilling operation. During the testing phase further observation can be made about this assumption.

The required electric motor power for top drive can be estimated based on the torsional limit of the aluminium drill pipe. Assuming the maximum RPM is 250 RPM and the motor efficiency is 80%, the estimated power of electric motor requirement is:

$$Motor\ power = \frac{T \times RPM}{63025 \times Efficiency} = \frac{65.5 \times 250}{63025 \times 80\%} = 0.325\ HP \quad (0.24\ kW)$$

*Efficiency*: Motor efficiency
2.3 Cutting transport calculation

The following section sets the flow rate calculation required for drilling operation. The minimum flow rate of the mud must be greater than the sum of the slip velocity, $v_{\text{slip}}$, and cutting velocity, $v_{\text{cutting}}$, as expressed in following equation [5]:

$$v_{\text{mud}} = v_{\text{cutting}} + v_{\text{slip}}$$

The cutting velocity is calculated by following equation:

$$v_{\text{cutting}} = \frac{\text{ROP}}{60 \left[ 1 - \frac{d_p}{d_h} \right] C_{\text{conc}}} = \frac{0.8}{60 \left[ 1 - \frac{0.8}{1.125} \right] 1.5} = 0.02 \frac{ft}{s} \quad (0.06 \frac{m}{s})$$

Where,

$d_p$: Outside diameter of the stabilizer

$d_h$: Diameter of the borehole

$C_{\text{conc}}$: Cutting concentration

The slip velocity can be calculated by Moore’s [5] correlation for a vertical well (Figure 4).
The slip velocity of a small spherical particle settling (slipping) through a Newtonian fluid under laminar flow condition, \(v_{\text{slip}}\) is given by Stoke’s law:

\[
v_{\text{slip}} = \frac{138 \times (\rho_s - \rho_f) \times d_s^2}{\mu_a} = 138 \times \frac{(22.07 - 8.33) \times 0.04^2}{1} = 3.03 \text{ ft/s} \times \left(\frac{0.93 \text{ m}}{s}\right)
\]

Where,
- \(d_s\): Diameter of cutting
- \(\rho_s\): Density of the cutting solid
- \(\rho_f\): Density of the drilling fluid (density of water)
- \(\mu_a\): Viscosity of the drilling fluid (viscosity of water)

However, determining the Reynolds number, \(Re\), shows that the Reynolds’ value is greater 300:

\[
Re = \frac{928 \times \rho_f \times v_{\text{slip}} \times d_s}{\mu_a} = \frac{928 \times 8.33 \times 3.03 \times 0.04}{1} = 938.4 \text{ (Turbulent)}
\]
Where, 

\( v_{slip} \): Slip velocity from previous calculation

According to Moore’s correlation, for \( Re > 300 \), the friction factor, \( f \), is taken as 1.54, then the slip velocity is calculated again:

\[
v_{slip} = f \sqrt{\frac{d_s (\rho_s - \rho_f)}{\rho_f}} = 1.54 \sqrt{0.04 \left( \frac{22.07 - 8.33}{8.33} \right)} = 0.396 \frac{ft}{s} \quad (0.12 \frac{m}{s})
\]

\[
abs(v_{slip} - v_{slips}) = |0.396 - 3.03| = 2.634 > 0.001
\]

Since the difference is higher than 0.001 (Figure 2), the iteration is performed to calculate the slip velocity. A program was written in Matlab to perform the iteration for calculating the slip velocity. The final slip velocity was determined to be 0.469 ft/s (0.143 m/s). Consequently the water flow rate must be greater than:

\[
v_{mud} = v_{cutting} + v_{slip} = 0.02 + 0.469 = 0.471 \frac{ft}{s} \quad (0.144 \frac{m}{s})
\]

A flow rate of 6 gpm (22.7 Lpm) is considered to be suitable as a hole cleaning requirement. The annular velocity, \( v_a \), is calculated by following formula:

\[
v_a = \frac{Q}{2.448 * (d_h^2 - d_a^2)} = \frac{6}{2.448 * (1.125^2 - 0.8^2)} = 3.92 \frac{ft}{s} \quad (1.19 \frac{m}{s})
\]

For the transport ratio a value of above 50% is empirically recommended to achieve good hole-cleaning during drilling. The transport ratio is calculated as follows and a value of 88 percent determined:

\[
Transport \ ratio = \frac{v_a - v_{mud}}{v_a} * 100 = \frac{3.92 - 0.471}{3.92} * 100 = 88\%
\]

2.4 Pressure loss calculation

The following section sets the calculation for conventional, rotary drilling pump requirements with the bit dimensions provided by DSATS. Assuming the Newtonian
Fluid (water) flow inside the drill string, the pressure loss inside drill pipe is calculated as follows. First the velocity of the fluid in the drill pipe, \( v_d \), is determined:

\[
v_d = \frac{Q}{2.448 \times (id_p^2)} = \frac{6}{2.448 \times (0.305^2)} = 26.35 \frac{ft}{s} \left( \frac{m}{s} \right)
\]

Based on that, the Reynolds number is determined:

\[
Re = \frac{928 \times \rho \times v_d \times id_p}{\mu_a} = \frac{928 \times 8.33 \times 26.35 \times 0.305}{1} = 62120 \ (Turbulent)
\]

Then, the ratio of the roughness of the pipe divided by the inner diameter of the pipe \( \varepsilon/id_p \) is calculated to determine the friction factor on the Fanning chart [6]:

\[
\frac{\varepsilon}{id_p} = 0.0006 = 0.00197
\]

Where,

\( \varepsilon \): Roughness of the drill pipe

---

*Figure 5. Fanning chart; friction factors for turbulent flow in circular pipes [6]*
Based on Fanning chart (Figure 5), the friction factor, $f$, is approximately 0.0073. The pressure loss inside the drill pipe, $P_s$, is calculated as follows:

$$P_s = \frac{f \cdot \rho_f \cdot v_d^2 \cdot L_{string}}{25.8 \cdot id_p} = \frac{0.0073 \cdot 8.33 \cdot 26.35^2 \cdot 4.18}{25.8 \cdot 0.305} = 22.1 \text{ psi (1.52 bar)}$$

Where,

- $f$: Fanning friction factor
- $L_{string}$: Length of the drillstring

There are two nozzles at the bit, the total area of nozzle, $A_n$, is calculated as follows:

$$A_n = \frac{\pi}{4} \cdot d_n^2 = \frac{\pi}{2} \cdot 0.093^2 = 0.013 \text{ in}^2 \quad (8.67 \text{ mm}^2)$$

Where,

- $d_n$: Diameter of the nozzles

Then, the pressure loss at the bit, $P_{bit}$, can be estimated by following calculation [6]:

$$P_{bit} = \frac{Q^2 \cdot \rho_f}{12031 \cdot A_n^2} = \frac{6^2 \cdot 8.33}{12031 \cdot 0.013^2} = 137.9 \text{ psi (9.5 bar)}$$

Where,

- $Q$: Flow rate of drilling fluid
- $A_n$: Total area of the nozzles

The jet impact force, $F_j$ of the bit is:

$$F_j = 0.01823 \cdot C_d \cdot Q \cdot \sqrt{\rho_f P_{bit}} = 0.01823 \cdot 0.95 \cdot 6 \cdot \sqrt{8.33 \cdot 137.9}$$

$$= 3.5 \text{ lbf (1.6 kg)}$$

Where,

- $C_d$: Discharge coefficient
- $P_{bit}$: Pressure loss at the bit

The jet velocity of the bit, $v_{bit}$ is:

$$v_{bit} = \frac{Q}{A_n} = \frac{6 \cdot 144}{7.48 \cdot 60 \cdot 0.013} = 143.2 \frac{ft}{s} \quad (43.6 \frac{m}{s})$$
Further, the pressure loss in annulus, $P_a$ is calculated as follows [6]:

The annular velocity is 3.92 ft/s (1.19 m/s) or 235 ft/min (71.65 m/min).

\[
P_a = \frac{1.4327 \times 10^{-7} \times \rho_f \times L_{rock} \times v_a^2}{(d_h - d_a)} = \frac{1.4327 \times 10^{-7} \times 8.33 \times 2 \times 1.97 \times 235^2}{(1.125 - 0.8)} = 0.4 \text{ psi} \quad (0.028 \text{ bar})
\]

Where,

$L_{rock}$: Length of the rock sample

The total downhole pressure loss, $P_{downhole}$ is:

\[
P_{downhole} = P_s + P_{bit} + P_a = 22.1 + 137.9 + 0.4 = 160.4 \text{ psi} \quad (11.1 \text{ bar})
\]

The total standpipe pressure requirement is the downhole pressure loss added by the atmospheric pressure, 14.7 psi:

\[
\text{Standpipe pressure} = P_{downhole} + 14.7 = 175.06 \text{ psi} \quad (12.1 \text{ bar})
\]

It is assumed that the pump will be connected with the hose line (made from rubber material) to the standpipe with roughness 0.0006 in. The pressure loss in the hose, $P_h$ is calculated by following the same schematic shown above:

\[
v_h = \frac{Q}{2.448 \times (id_h)^{2.5}} = \frac{6}{2.448 \times (0.5)^{2.5}} = 9.8 \frac{ft}{s} \quad (3 \frac{m}{s})
\]

\[
Re = \frac{928 \times \rho_f \times v_h \times id_h}{\mu_a} = \frac{928 \times 8.33 \times 9.8 \times 0.5}{1} = 37893 \text{ (Turbulent)}
\]

\[
\frac{\varepsilon}{id_h} = \frac{0.0006}{0.5} = 0.0012
\]

Where,

$\varepsilon$: Roughness of the rubber hose

$id_h$: Internal diameter of the rubber hose
Based on Fanning chart (Figure 5), the friction factor, $f$, is approximately 0.007. The pressure loss inside the hose, $P_h$ is:

$$P_h = \frac{f \cdot \rho \cdot v^2 \cdot H_{rig}}{25.8 \cdot id_h} = \frac{0.007 \cdot 8.33 \cdot 9.8^2 \cdot 7.5}{25.8 \cdot 0.5} = 3.28 \text{ psi} \quad (0.23 \text{ bar})$$

Where,

$v_h$: Annular velocity of the drilling fluid inside rubber hose
$H_{rig}$: The height of the rig

The total pressure loss along the entire system is:

$$Total \ pressure \ loss = P_{downhole} + P_h = 160.4 + 3.28 = 163.7 \text{ psi} \quad (11.3 \text{ bar})$$

Where,

$P_h$: Pressure loss inside rubber hose

The total pressure loss with additional atmospheric pressure is 178.4 psi (12.3 bar).

To estimate the pump horse power requirement, Bernoulli’s equation is used:

$$P_{pump} = P_{loss} + P_{atm} + \frac{\rho_f \cdot \Delta v^2}{2} + \rho_f \cdot g \cdot \Delta h$$

$$= 163.7 + 14.7 + \frac{8.33 \cdot 7.48 \cdot (9.8^2 - 3.92^2)}{2 \cdot 32.174 \cdot 144} + \frac{8.33 \cdot 7.48 \cdot 32.174 \cdot (7.5)}{32.174 \cdot 144}$$

$$= 182.2 \text{ psi} \quad (12.6 \text{ bar})$$

$$P_{pump} = \frac{P \cdot Q}{1714} = \frac{182.2 + 6}{1714} = 0.64 \text{ HP} \quad (0.48 \text{ kW})$$

Where:

$P_{loss}$: Pressure loss inside the circulation system
$P_{atm}$: Atmospheric pressure
$\Delta h$: The height of the rig, $H_{rig}$
$g$: Earth gravitation, 32.174 ft/s$^2$
$P$: Total pressure required
Conversion from ft³ to gallon: 7.48
Conversion from ft² to in²: 144

It is assumed that the efficiency of the pump is 85%, therefore the requirement of the horse power pump is 0.75 HP (0.56 kW) for conventional rotary drilling. The following Table 7 shows the variation of the pump power requirement according to the flow rate variation:

Table 7: Pump power requirement rotary drilling

<table>
<thead>
<tr>
<th>Flow rate variation (gpm)</th>
<th>Flow rate variation (Lpm)</th>
<th>Transport ratio (%)</th>
<th>Pump Pressure (bar)</th>
<th>Pump (HP)</th>
<th>Pump (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>11.4</td>
<td>76%</td>
<td>4.1</td>
<td>0.12</td>
<td>0.09</td>
</tr>
<tr>
<td>4</td>
<td>15.1</td>
<td>82%</td>
<td>6.3</td>
<td>0.25</td>
<td>0.19</td>
</tr>
<tr>
<td>5</td>
<td>18.9</td>
<td>86%</td>
<td>9.1</td>
<td>0.45</td>
<td>0.34</td>
</tr>
<tr>
<td>6</td>
<td>22.7</td>
<td>88%</td>
<td>12.6</td>
<td>0.75</td>
<td>0.56</td>
</tr>
</tbody>
</table>

2.5 Summary calculation results

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Calculated Result</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Field Units</td>
</tr>
<tr>
<td>Critical buckling load</td>
<td>$P_{bc}$</td>
<td>41.6 lbf</td>
</tr>
<tr>
<td>Burst limit</td>
<td>$P_{burst}$</td>
<td>1714.7 psi</td>
</tr>
<tr>
<td>Torque</td>
<td>$T$</td>
<td>65.5 in. lb.f</td>
</tr>
<tr>
<td>Flow rate</td>
<td>$Q$</td>
<td>6 gpm</td>
</tr>
<tr>
<td>Pump pressure</td>
<td>$P_{pump}$</td>
<td>182.2 psi</td>
</tr>
<tr>
<td>Pump horse power</td>
<td>$HPP$</td>
<td>0.75 HP</td>
</tr>
</tbody>
</table>
3 Rig design

The mast is designed using a standard framework system. The framework uses standard, commercially available, aluminium profiles that can easily be disassembled for transportation. The internal dimensions of the mast are: height 2200 mm, width: 450 mm, depth: 600 mm.

During the design the following considerations were paramount:

1. Securely hold the machine bed (marked in blue, Figure 6) in place which contains hoisting system and top drive).

2. Safety function; to provide a frame that is easily covered by polycarbonate plates to prevent access to pinch points and to provide a barrier to contain debris/ fluid in case of (pipe) failure during operation.

3. An energy chain carrier that holds the electrical power and shielded control cables as well as the hydraulic power (mud line).

4. A bottom plate, on which the frame is mounted, is used in order to benefit from the weight of the rock sample (approx. 286.6 lbs (130 kg)) which acts to stabilize the structure.

5. A 17.71 in x 27.5 in (45 cm x 70 cm) wide opening which allows easy placement of the rock sample(s) within the structure and the usage of a tub for the collection and return of the drilling fluid.

6. The framework supports the installation of screw jacks to clamp the rock sample in place. The internal dimension of the mast are: height 2200 mm, width: 450 mm, depth: 600 mm.

Figure 6: Proposed drilling rig design
3.1 Machine bed

The machine bed is designed as a rigid frame that contains the guide rails and non-rotating spindle for the hoisting system. Brackets at an angle of 45° stiffen the corners of the machine bed to ensure the guide rails are parallel to each other at all times. The guide rails (Figure 7) direct the movement of the sled in linear, vertical direction. The sled-guides are made from maintenance-free, self-lubricating high-performance plastics. The lubricant is incorporated into the bearing material, making the bearing materials suitable for dry-running conditions. Therefore, the sled and guide system is virtually maintenance free as it is resistant to dirt, dust and moisture.

3.2 Hoisting system

3.2.1 Spindle installation

The worm wheel (Figure 8) is the mechanism that is provided with a female thread and converts the rotational movement of the hoisting motor into an axial movement of the non-rotating spindle. The spindle is attached to the machine bed and on the upper part of the machine bed and connected to a load cell in order to obtain the WOB. The bottom connection of the spindle is connected to a floating bearing with one degree of freedom of movement in the z-direction.

In this case, the hoisting motor and worm wheel is mounted on the backside of the sled to cause the linear movement of the sled.

3.2.2 Spindle Calculation

The design calculation for the spindle are in accordance with the requirements set forth of the selected manufacturer of the spindle components. Table 8 list the properties of 3 different options. NSE 5 SN has been chosen.
Table 8: Spindle component properties and requirements

<table>
<thead>
<tr>
<th>Model - Nozag</th>
<th>NSE 2-SN</th>
<th>NSE 2-SL</th>
<th>NSE 5-SN</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. lifting force [N]</td>
<td>2000</td>
<td>2000</td>
<td>5000</td>
</tr>
<tr>
<td>Translation ratio</td>
<td>5:1</td>
<td>20:1</td>
<td>4:1</td>
</tr>
<tr>
<td>Lift/rotation driveshaft</td>
<td>0.8</td>
<td>0.2</td>
<td>1</td>
</tr>
<tr>
<td>Drive torque [Nm]</td>
<td>F[kN]*0.34+0.21</td>
<td>F[kN]*0.14+0.11</td>
<td>F[kN]*0.45+0.1</td>
</tr>
<tr>
<td>Max. Drive torque [Nm]</td>
<td>2.5</td>
<td>0.8</td>
<td>5.6</td>
</tr>
<tr>
<td>Druchtriebs torque [Nm]</td>
<td>12</td>
<td>12</td>
<td>23</td>
</tr>
<tr>
<td>Max. driveshaft rotation [min⁻¹]</td>
<td>1800</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td>Weight of gearbox [kg]</td>
<td>0.64</td>
<td>0.64</td>
<td>1.06</td>
</tr>
<tr>
<td>Weight of spindle [kg/m]</td>
<td>1.05</td>
<td>1.05</td>
<td>1.58</td>
</tr>
<tr>
<td>Lifting speed at 1500 min⁻¹ [mm/s]</td>
<td>20 mm/s</td>
<td>5 mm/s</td>
<td>25 mm/s</td>
</tr>
<tr>
<td>Time at a length of 1100mm [s]</td>
<td>55</td>
<td>220</td>
<td>44</td>
</tr>
</tbody>
</table>

The drive torque for the gearbox is calculated by:

\[ M_{Ge} = \frac{F(kN) \cdot P_{Sp}(mm)}{2 \cdot \pi \cdot \eta_{Ge} \cdot \eta_{Sp} \cdot i} + M_L \]

The drive rating of the gear is calculated:

\[ P_1 = \frac{P_{Ge}}{\eta_{Ku} n_{Ku}} \]

And the effective power requirement of the motor is determined using:

\[ P_{Ge} = \frac{M_{Ge}(Nm) \cdot n(min^{-1})}{9550} \cdot S \]

Table 9: Spindle calculation results

<table>
<thead>
<tr>
<th>Drive torque of the screw jack</th>
<th>NSE 2-SN</th>
<th>NSE 2-SL</th>
<th>NSE 5-SN</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drive torque [Nm] for one gearbox</td>
<td>M_{Ge}</td>
<td>0.880</td>
<td>0.447</td>
</tr>
<tr>
<td>Lifting load (dynamisch) [kN]</td>
<td>F</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Efficiency of the screw jack (without spindle)</td>
<td>\eta_{Ge}</td>
<td>0.76</td>
<td>0.45</td>
</tr>
<tr>
<td>Efficiency of the spindle</td>
<td>\eta_{Sp}</td>
<td>0.5</td>
<td>0.42</td>
</tr>
<tr>
<td>Pitch of spindle</td>
<td>P_{Sp}</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Ratio</td>
<td>i</td>
<td>5</td>
<td>20</td>
</tr>
<tr>
<td>No-load torque [Nm]</td>
<td>M_L</td>
<td>0.21</td>
<td>0.11</td>
</tr>
<tr>
<td>Drive Rating [kW]</td>
<td>P_{Ge}</td>
<td>0.1797</td>
<td>0.0912</td>
</tr>
<tr>
<td>Drive rating, motor effective [kW]</td>
<td>P_1</td>
<td>0.1797</td>
<td>0.0912</td>
</tr>
<tr>
<td>Efficiency of the coupling</td>
<td>\eta_{Ku}</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Number of couplings</td>
<td>n_{Ku}</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Rotational speed of the motor</td>
<td>n</td>
<td>1500</td>
<td>1500</td>
</tr>
<tr>
<td>Safety factor</td>
<td>S</td>
<td>1.3</td>
<td>1.3</td>
</tr>
</tbody>
</table>
3.3 Hydraulic system

The hydraulic system consists of a centrifugal pump, tank, filter, pressure relief valve, pressure sensor and flow meter. The pump sucks water from a tank via a filter and pumps it into the borehole via the drill rod. The hydraulic system has two main functions. The first main function is the transport of cuttings out of the well. The second main function is to drive the hammer drill. Figure 9 shows the schematic structure of the hydraulic system.

Figure 9: Schematic structure: Hydraulic system

The pressure sensor measures the current line pressure behind the pump. The flow meter measures the flowrate behind the pump. The pressure or the volumetric flow can be regulated via the pump speed. If the two values are to be controlled independently, a controllable throttle valve or a controllable bypass must be installed.

The pressure relief valve is installed for safety. If the line pressure exceeds a permissible limit, the valve releases.
3.3.1 Multistage centrifugal pump

The pump, as displayed in Figure 10 is a multi-stage centrifugal pump from the manufacturer Grundfos. The pump has a total of 36 stages and can reach a discharge height of 165m (16.5bar) at 7 m³/h. The characteristics of the pump are shown in Figure 11.

This type of pump has a high pumping capacity with a small size and is insensitive to abrasive media. The pump motor is operated on a frequency converter, thus the flow rate of the pump can be steplessly regulated.
3.4 Drillpipe connection concept

The previous contests have shown that the connection to and from the drillpipe is a potential source of malfunction. A hole for a setscrew is a huge distortion in the pipe and also the entire moment is transferred in this small area. Thus, a mechanism is required that transfers the torque and WOB over a large area. Therefore the Ringfeder® locking assembly is chosen which reduces this potential weak point in the design (Figure 12). It transfers torsional moments and axial force with friction between element and pipe. The loaded area is much bigger than in a setscrew connection and the shape of the pipe is not disturbed. The idea is to use 2 Ringfeder® locking assemblies to transfer the applied moment and axial force.

To verify the ability of the drillpipe to endure the pressure exerted from the locking assemblies, the formula below is used. It calculates the maximum endurable pressure for a given yield strength. The pressure of the locking assembly induces a compressive load. In this case a yield strength of 240 N/mm² and shape factor, $C_3$, of 0.8 is chosen.

$$p_{zul} = \frac{R_{p0.2}(1 - \left(\frac{d}{D}\right)^2)}{C_3 \times 2} = 50.77 \, N/mm^2$$

Where,

$R_{p0.2}$: Yield Strength $= 240 \, N/mm^2$

$C_3$: Shape factor $= 0.8$

$d$: Internal diameter $= 7.747 \, mm$

$D$: External diameter $= 9.525 \, mm$

The maximum applied pressure of the locking assembly RfN 7061 Stainless Steel is about 37 N/mm². Therefore the pipe is able to endure the load induced by the locking assembly.
4 Electrical system

The electrical system uses, the in Germany common, 3 Phase 223/400 V supply. In the electrical enclosure the main breaker feeds the common bus which supplies power to the AC/AC variable frequency drive (VFC) for the pump motor, an AC/DC converter that powers a common bus at 48 VDC (960W), and a AC/DC converter that powers a 24 VDC (240W) common bus.

The 48 VDC bus supplies power to the brushless DC (BLDC) Driver for the Top Drive motor and to the stepper drive for the spindle motor.

The 24 VDC bus supplies power to the Berghof PLC and its connecting switches and sensors and corresponding amplifiers. Further detail can be obtained from the electrical drawings in Figures 13 to 15. Emergency stop button wired in series, will be placed at strategic points around the drillrobot to be able to shut down the hole system in case of a malfunction.
Figure 13: Electrical schematic 1 of 3
Figure 14: Electrical schematic 2 of 3
4.1 Control system

The control of the drilling robot is performed by an industrial programmable logical computer (PLC). The PLC has an Ethernet, RS485 and an RS232 interface which are used for communication with peripheral devices. Different kinds of communication protocols are available. The motor drivers of the Top-drive and the feeding-device communicate with the PLC via the EtherCAT (Ethernet for Control Automation Technology) protocol. The frequency converter of the mud-pump also communicates with the PLC via the EtherCAT protocol. EtherCAT is a real-time protocol that uses Ethernet which was initiated by the Beckhoff automation. The protocol disclosed in IEC standard 61158 is suitable for hard as well as soft real-time requirements in automation technology. The programming and data visualization is carried out via a conventional PC which is used as a human machine interface (HMI) interface. The PC is connected to the 3-Port switch in the PLC and communicates over the TCP/IP protocol. Together with the CODESYS™ plugin “CODESYS WEB VISIO”, an HMI interface results, which can be operated via a conventional WEB browser.

The sensors in the BHA communicate via Bluetooth with a Bluetooth module, which is connected to the serial RS232 interface of the PLC. All other sensors and actuators are connected directly to the digital and analog inputs and outputs of the PLC. Figure 16 shows the schematic structure of the control system.

![Schematic structure of the control system](image-url)
4.1.1 PLC

The ECC22XX is a modular automation system for industrial control applications (Figure 17). The ECC22XX compact controller incorporates a CODESYS environment for control of automatic processes in low voltage systems. The programming language CODESYS 3.5 (IEC 61131-3) is used for programming. The CODESYS PLC program on the PLC controls all processes within the system and communicates with the HMI via Ethernet TCP/IP. CODESYS is an established company and its software tool can be used for complete automation, control, communication and visualization tasks. All areas are covered by a single tool. Common communication protocols such as EtherCAT and CANopen are integrated. The controller EC Compact in conjunction with CODESYS thus enables the complete automation and visualization of the drilling process.

![Figure 17: Compact controller PLC](image)

The basic data of the EC Compact control are:

- Simple network integration
- Remote access via Internet or Ethernet (CODESYS WEB VISio)
- Integrated 3-port network switch
- 800 MHz ARM CPU
- 256MB RAM / 256MB Flash / 100kB FRAM
- Certification according to EN 61131-2
• Communication protocols CANopen, EtherCAT, Profinet, Modbus
• Real-time clock
• 16 DIO
• 16 DII
• 12 AI
• 6 AO

4.2 Motor control

4.2.1 Top Drive components

For the drive of the Top Drive, a BLDC motor with a rotary encoder from the company Nanotec with 440W at 3000 rpm is used and operated by a fieldbus-capable motor control.

The engine control unit N5L is also manufactured by the manufacturer Nanotec. The motor control communicates with the PLC via the EtherCAT protocol and supports field-oriented closed-loop control. Thanks to the real-time capability and the high transmission rate of up to 100Mbit / s, EtherCAT allows several subscribers to run synchronously and to capture large amounts of data in a short time. Together with the field-oriented control which is parameterized using a software tool from Nanotec, a fast and accurate control of the motor is possible. In addition, time-critical processes, such as the speed control of the motor, are transferable to the motor control and therefore need not be adopted by the PLC.

![Figure 18: Top Drive components](image-url)
4.2.2 Spindle drive components
For the spindle drive, a stepper motor with rotary encoder from the company Nanotec with 1Nm holding torque is used and operated by a fieldbus-capable motor control. The engine control unit SMCIT2 is also manufactured by the manufacturer Nanotec. The motor control communicates with the PLC via the EtherCAT protocol. The integrated software-based current controller together with the microstep resolution ensures optimal running behavior at all speeds.
Thanks to the real-time capability and the high transmission rate of up to 100Mbit / s, EtherCAT allows several subscribers to run synchronously and to capture large amounts of data in a short time. In addition, time-critical processes, such as the speed control of the motor, are transferable to the motor control and therefore need not be adopted by the PLC. The calculated power requirement of the spindle is 0.32 kW.

4.2.3 Pump control components
The pump is driven by an asynchronous motor with a variable frequency converter (VFC). The VFC (MX2) is from the company Omron and has a rated power of 6.0kW. The VFC has also like the BLDC- and Stepperdriver a Fieldbus Interface to communicate with the PLC.
4.3 Sensors

4.3.1 Load cell and amplifier

To measure the WOB an aluminum single point load cell is installed at the spindle (Figure 18). A load cell is a device that converts a mechanical movement into an electrical signal.

Figure 21: Load cell device
In conjunction with an amplifier/transmitter (Figure 22), the electrical signal from the pressure transducer is converted into a proportional 0-10V signal. This 0-10V signal can be further processed by the PLC.

**Figure 22: Amplifier for load cell**

This load cell and amplifier can also be used to measure the torque at the drill rod. With a defined lever arm, the torque on the drill string can be calculated via the measured force.

### 4.3.2 Pressure Sensor

For pressure measurement behind the pump an analogue pressure transducer is used. The pressure transducer has a pressure range between 0-25bar and converts the measured pressure in a proportional electrical 0-10V signal. This 0-10V signal can be further processed by the PLC.

### 4.3.3 Flowmeter

The flowmeter consists of a metal housing, an impeller and a Hall sensor. The Hall sensor emits a pulse per revolution of the impeller. The pulses per time are measured via a counter input on the PLC. The volume flow can be calculated with the amount of water pumped per revolution, which can be taken

**Figure 23: Pressure transducer**
the manufacturer data sheet. The flow meter can be operated up to 16 bar pressure and has a measuring range of 30-5000 l/h.

**Figure 24: Flowmeter**

### 4.3.4 Rotary encoder

The encoder is incorporated with the selected stepper motor and BLDC motor. Please consult the according chapters.

### 4.4 Control system architecture

**Requirements for the control system:**

- Real-time measurement and control
- Fully automated drilling process
- Observation of the drilling process via a suitable panel
- Measurement & data storage
- Downhole monitoring

**Figure 25: Control system architecture (continuation Figure 19)**
4.4.1 Visualization and programming

In the initial phase, the PC is used for programming PLC. Later, the PC is used to handle the computationally extensive tasks. In addition, the PC serves as the (HMI) to control the drillrobot over a graphic user interface (GUI). Please refer to chapter 4.6 for more information.

The GUI will plot the important parameter that are used to influence the concept of mechanical specific energy. These parameters are weight on bit, applied torque, rotational speed, and rate of penetration. Furthermore, data from the downhole sensors, such as the gyroscope, accelerometer, and magnetometer will be interpreted and displayed. During the trial phase, the GUI will be continually optimized to serve the needs as required.

*Figure 26: Control system architecture (continuation from Figure 18)*
4.5 Control algorithm

Figure 27: Genetically mutating algorithm structure
The flow charts in Figure 27 represents a genetic programming approach to optimize the variable drilling parameters $n$ and $WOB$. For a genetic algorithm a fitness function is needed. The goal in the given case is to minimize the mechanical specific energy ($MSE$), therefore better fitness means a lower value.

$$MSE = \frac{P\Delta t}{V}$$

$$MSE = \frac{WOB}{A} + \frac{120 \cdot T \cdot n}{A \cdot ROP} = \text{minimal}$$

Where,

$$T = \frac{\mu \cdot d \cdot WOB}{36}$$

$P$: Power  
$\Delta t$: Time difference  
$V$: Volume of destroyed rock  
$n$: Rotational speed  
$WOB$: Axial force  
$T$: Applied torque  
$ROP$: Rate of penetration

As it is based on genetic algorithms the program has two main phases:

- A random initialization of the population
- An iteration over the steps: fitness evaluation, selection of individuals, and reproduction and mutation

The iteration ends when a termination criterion is reached. The only termination criterion so far is reaching the maximum number of iterations. Calibration runs enable the selection of a termination criterion which implies the quality of the solution.

The algorithm described by the flowchart in Figure 27 contains two input steps before the initialization. The boundaries, denoted in a configuration file or something similar, are needed to set limit for the random initialization, whereas the sensor value input will
be regularly set by the programmable logic controller such that the algorithm will start each run with the newest sensor data.

The fitness evaluation subroutine will iterate over all members of the population to evaluate their individual fitness and find the highest overall fitness. This is used to save the elitist of the old population to the new one which will avoid rediscovery of good solutions.

Now, the first steps to fill a new population have been made. Beside of the elitist, only children of old population members will be part of the new population. The first step is the parent selection – which will favour fitter parents by comparing their higher relative fitness (bestFitness / ownFitness, as bestFitness<= ownFitness) to a random value between 0 and 1. Next a child will be created which will inherit the $n$ parameter randomly from one parent and WOB from the other. For any child there is a small chance of mutation, which will result in either WOB or $n$ being set to a random value.

As soon as the new generation is filled and if the termination criterion is not reached, the iteration will start again by evaluating the fitness of the population. If the termination criterion is reached the best values found will be made available as an output to the PLC so that the PLC can access them for regulation.
After this the whole program as shown in the flowchart will start again to find new best values for the newest sensor data.

4.6 Data handling and storage

The PC and the PLC have different tasks in the control process of the drilling. The PLC will take on the real-time controlling. Its two main tasks are the regulation of $n$ and WOB, according to the values calculated by the PC, and the prevention of catastrophic failures, e.g. by surveying the drilling process and preventing boundary violations (i.e. mechanical operating envelope).

The PC will take on the computationally extensive tasks, such as heuristically finding best values for the parameters $n$ and WOB. Even though the goal is to finish the tasks in a timely manner – as otherwise a deployment in a real time environment does not make sense – the computer can't operate as fast as the PLC. As both need to
communicate with each other the communication must be asynchronous, such that both can finished a started task, but start new iterations with new data.

![Sequence diagram](image-url)

*Figure 28: Sequence diagram*

As shown in the sequence diagram in Figure 28, this asynchronous solution requires a set of initial regulation parameters for the PLC, which will be used until the PC has finished its first iteration to heuristically calculate regulation values. After this, the exchange of values will be done asynchronously. Every newly calculated optimum parameter value will be stored so that the PLC can access the data in the next control iteration.

### 4.6.1 Response time of measurements

The PLC will store the newest sensor values every time after accessing the sensors. These can then be accessed by the PC for the MSE calculation. The minimal response time for the sensor measurements would be 1 ms, but only for one reading. As it is planned to use the offered possibility to receive the average of 10 readings – which would rule out errors – the response time for sensor readings would be 10 ms. A further time overhead must be calculated for the data aggregation in a file, that can be used for data analysis and display as well as for the input values for the algorithm on the PC.
5 BHA Design

Figure 29 shows the proposed concept for the sensor module of the BHA. The electronics module is located as close as possible to the lower end of the string. The drillstring module has a holding cavity which will host a separately build electronics module. The electronics module will be made out of a polymer plastic shell to ensure that the data transmission is not influenced. It will hold the printed circuit board (PCB) which incorporates the power converter, microcontroller, and sensors. An epoxy resin is casted around all components inside the module to seal the components from water and secure it in place. The module is secured to the BHA with screws. The battery power supply is an adjacent chamber which is sealed by a gasket.

![Figure 29: Proposed design of the BHA housing](image)

5.1 Specification for sensors and instrumentation

The idea is to sense the vector of the gravitational field of the earth. With this vector we can calculate the angle between the z-axis and BHA. Using an accelerometer drilling behaviour can be interpreted. In addition, data from a gyroscope is obtained to measure torsional vibration of the drillstring and then analysed.

It is planned to build at least two electronic modules that are easily interchangeable: The first option is a 6 degree of freedom sensor (MPU-6050) Figure 30. It houses an accelerometer and a gyroscope. The communication between the downhole microcontroller and the sensor will be over an FC port. The sensor has an accelerometer with range up to ±16g and the gyroscope range is up to 2000 °/s. The nominal Input Voltage is 3.3V.
The second idea is to sense a magnetic field from two magnets placed next to the rock sample. If the magnitude of the field, the angle of the drillstring, and the z-position are known, the X and Y position can be calculated.

The second option uses 9 degree of freedom sensor (LSM9DS1). It houses an accelerometer, a gyroscope and a magnetometer.

The communication between the downhole microcontroller and the sensor will be over an I²C port. The sensor has an accelerometer range up to ±16g, the gyroscope range is up to 2000 °/s and the magnetometer has a full-scale range of ± 16 gauss. The nominal Input Voltage is 3.3V.

5.2 Concept of downhole sensor assembly

The concept of the downhole sensor is to use a microcontroller which can communicate over Bluetooth with the PLC. It will be powered by single AAA 1.5 V Battery and a DC/DC converter which transforms the voltage to 3.3V, which is suitable for the microcontroller and sensor. The communication between the sensor and microcontroller is either by serial peripheral interface (SPI) or I²C Data Bus. The microcontroller will send a prompt command for each sensor data channel and receive the data. Then the microcontroller will pack the data in an array and send it via Bluetooth to the PLC for further interpretation.

5.2.1 Update time

The roughly estimated update frequency is about 200 Hz. In addition, the sensor offers an interrupt line which can be triggered when a specific sensor channel exceeds
predetermined critical level. This can be used to perform an emergency stop in case the drill bit blocks during the drilling process.

![Communication Schematics](image)

**Figure 31: Communication Schematics**

### 5.3 Microcontroller

The microcontroller will be an Esspressive ESP 32. It has two cores with a clock frequency of 240 MHz. It features an integrated Bluetooth and Wi-Fi transmission unit. The Bluetooth section can operate in Classic and BLE mode. The operating voltage is between 2.2 and 3.6 Volts and has a dimension of only 18mm x 25.5 mm x 3mm. It provides multiple interfaces for sensors like I²C and SPI. Also, this microcontroller has multiple power modes available. It can shut down Bluetooth and Wi-Fi to save power during the data acquisition and to minimize errors due to distortion created by the wireless communication. After the complete data set is collected the Bluetooth section will power up and send the set to the PLC.

### 6 Percussion drilling

Percussion drilling has long been recognized to offer the potential of drilling faster than conventional rotary drilling, particularly in some hard formations such as granite, sandstone, limestone, dolomite, etc. [7]. There are other potential benefits of
percussion drilling: requires lower WOB, results in longer bit life due to less contact
time between the bit and rock, offers less hole deviation, and generates larger cutting
[8]. In conventional rotary drilling, the WOB applied pushes the bit onto the rock and
as the bit rotates, the cutter of the bit shears and cuts the chip of the rock. The WOB
must be high enough so the bit can penetrate the rock. While in percussion drilling,
impact or collision or vibratory shock is used to cut the rock [9]. This is one of the most
powerful forces. Based on the Law of Conservation of Momentum, with high impact
speed and short contact time applied, the bit in percussion drilling can produce much
higher impact force along the direction of the bit movement (Figure 32). The bit will
 crush the rock below the bit and form the cratered zone when the force exceeds the
rock compressive strength. The cratered zone may be much deeper than the actual
depth of bit penetration [10]. In percussion drilling, it is needed to remove the crushed
rock as quickly as possible to avoid the re-crushing failed rocks (percussive energy
does not contribute to rate of penetration).

![Figure 32. Rock crushing process in rotary and percussion drilling [7]](image)

### 6.1 Design of a drilling hammer and drill bit

For the percussion drilling purposes, a special drill bit has to be used.
Improved performance when drilling through hard formations is achieved by setting
front buttons higher than the gage buttons. The elevated front creates a somewhat
recessed hole-bottom pattern that alters the rock-breaking action to achieve improved
performance.
Figure 33: Proposed drill bit for the hammer system

Figure 34: Drill bit with front buttons, suitable for percussive drilling through brittle formations

Our percussion drilling hammer system has a simplified design which consists of a fewer components for a better performance. To avoid excessive stresses in the aluminum pipe the hammer system is incorporated above the drill bit and impacts the bit directly. A hammering action provides short and rapid shocks to pulverize brittle rock material, providing more efficient drilling with less effort. In addition to the hammering process, rotary drilling is used for drilling through softer formation layers.
6.2 Operating modes

The percussion drilling device will be powered by the drilling fluid flow. This design uses the so called direct acting water hit. Figure 36 shows an example of the principle.

**Figure 35: Hammer system design**

**Figure 36: Hammer System in two different operating modes**

The operating mode can be summarized in the in 3 stages

- **Acceleration phase** - the valve is closed and the hammer is accelerated

- **Ballistic phase** – the valve is open and returns to the start position (due to the potential energy stored in the spring) while the hammer continues to travel due to its kinetic energy and hits the anvil which transfers the energy to the drill bit
Return phase – the potential energy stored in the spring transforms into kinetic energy by moving the hammer back into the starting position where the valve closes and the process repeats itself.

6.2.1 Simulation of the percussion drilling device

A design was drawn up to drillbotics™ outlined dimensions and then a model was implemented in Matlab™ Simscape to simulate the performance of the percussion tool for selected pressures and flow rates. Particularly interesting, in terms of the output data, is the achieved energy per hit and the frequency of the percussion action in consideration of the input values for flow rate and pressure.

Figure 37: Percussion tool simulation

For the shown simulation results in Figure 38, the flow rate was 0.667 l/s and the pressure drop was 7.5 bar. The upper graph shows the displacement of the hammer versus time. The middle graph demonstrates the force on the hammer versus time. Thirdly, the lower graphs displays the energy in Joule vs time. This simulation indicates that higher flow rates and pressure across the hammer are desirable. As a result, a new, improved design is currently being developed. In addition, provisions have been made in the mud pump selection to accommodate the high flow rate and pressure requirements.
6.3 Correlation of rock properties

By using a percussion drilling system, as proposed in our design, it is possible to obtain dominant rock properties from measurements of penetration rates [11]. Experimental studies shown that the penetration rates can be directly correlated with the properties of the rock sample. This knowledge is extremely useful for estimation of the type of rock in a simple and direct way without using complicated devices which may fail under drilling conditions or deliver unreliable measurement records.

In order to estimate equations which correlate rock properties with measured rate of penetrations several experimental tests were carried out. Results have shown that the strongest correlations with the rate of penetration exhibit uniaxial compressive strength (UCS), tensile strength, point load strength and elastic modulus. Plots of penetration rate as a function of selected rock properties are presented in Figure 39.
Figure 39: Penetration rate vs. UCS and elastic modulus of the rock sample [11]

Plots were constructed by using the method of least-squares regression. For each regression the equation of the best-fit line was determined.

Rock properties have also direct influence on the specific energy, which is a common concept of a rock destruction. When tool geometry, cutter spacing and tool penetration are kept constant, specific energy will be a function of rock parameters. Specific energy, $Se$, can formulated as follows:

$$Se = \frac{\sigma^2}{2E}$$

Where,

$E$: Secant elastic modulus

$\sigma$: Rock compressive strength.

This relation allows for estimation of the rock when the specific energy is known.

The specific energy in terms of the penetration rate can be expressed by

$$ROP = \frac{E_{i}fT_{r}}{ASe}$$

Where,

$E_{i}$: Energy per blow

$f$: Blow frequency

$T_{r}$: Energy transfer rate

$A$: Drill hole area

Dependency of penetration rate and rock properties can be extremely useful in the estimation of the type of rock sample during drilling. Examples of uniaxial compressive strength for different types of rocks are:

Limestone: 123.8 MPa
Dolomite: 68.0 MPa  
Sandstone: 149.2 MPa

For the competition purposes, methods mentioned in this paragraph, are an interesting alternative or support to the electrical downhole devices or expensive measurement while drilling methods. In addition, the team is looking forward to perform vibrational analysis obtained downhole data to see if other correlations can be made.
## 7 Safety considerations and risk analysis

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<thead>
<tr>
<th>Risk name</th>
<th>Risk description</th>
<th>Risk control strategy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drilling fluid (water) leaking</td>
<td>Drilling fluid leaks from hose or circulation system</td>
<td>Use suitable hose and circulation system (e.g. hose with appropriate pressure rating capacity). The circulation system will be checked before and while testing.</td>
</tr>
<tr>
<td>String wobbling</td>
<td>String vibration/wobbing</td>
<td>Start drilling at low RPM then increase gradually after reach a certain depth e.g. 1/4&quot;. Set the limit of the RPM in algorithm.</td>
</tr>
<tr>
<td>Bit walking</td>
<td>Bit drills deviated away from the trajectory plan. This risk is also associated with string wobbling.</td>
<td>Install a guide shoe or riser/casing at rotary table above the rock sample thus limiting the drillstring wobbling. Install downhole sensors, which are accelerometer and gyro, and integrates the acquisition data with algorithm to self-control the drilling trajectory.</td>
</tr>
<tr>
<td>Drillstring stuck</td>
<td>Drillstring stuck due to unclean hole or cutting is not circulated properly from the hole</td>
<td>Apply minimum flow rate according to minimum cutting transport velocity estimation. Apply also minimum drillstring rotation speed (RPM) according to design calculation to help hole cleaning.</td>
</tr>
<tr>
<td>Loss verticality of hole section</td>
<td>Drill directional hole or deviated trajectory away from the plan</td>
<td>Install downhole sensors (accelerometer and gyro) and integrates acquisition data with algorithm program. Apply algorithm program to maintain verticality to self-control string while drilling.</td>
</tr>
<tr>
<td>Drillstring buckling and twist off</td>
<td>The weakest part along the drillstring is aluminum drill pipe. Another potential weak part is connection. Buckling and twist off could occur during the drilling.</td>
<td>Use high strength connection type for all parts of the drillstring, particularly the aluminum drillpipe. Estimate the limit strength of drill pipe then limit the WOB and RPM to avoid the failure according to the design calculation. Set the WOB and RPM limit in the algorithm. In case much vibration occurs, alert (alarm) system will be active thus students can respond.</td>
</tr>
<tr>
<td>Short circuit and/or electrical hazard</td>
<td>The risk is associated with electrical cable or source</td>
<td>Proper handling of electrical cable shall be applied, especially the avoidance from liquid (water). All electrical systems must be set up properly, soldered, installed, and connected to power source carefully and if it is possible enclosed system (use protector) and contained in one place. All of the steps associated</td>
</tr>
<tr>
<td>Mechanical construction risk</td>
<td>During build up and/or testing, risk associated with pinch points, punctures, lacerations, cutting debris, and etc, could cause damage and hazard.</td>
<td>Proper handling of material and pieces during building up the rig structure, particularly it is mandatory to wear the proper HSE equipment according to minimum standard (e.g. safety shoes, safety glasses, gloves, etc). Students will also work from a certain height during building up the rig structure, thus precautions must be taken to prevent injuries. A certain range of zone isolation must be set thus non related person will be allowed to pass the zone. A general working procedure (Standard Operating Procedure/SOP) will be set before rig construction work. The connection of all parts and pieces of rig structure must be checked properly during and after rig construction to confirm the solid rig structure, thus avoiding the loose pieces during the drilling or test. All the sharp edges in rig structure will be covered or protected with protector to avoid human injures.</td>
</tr>
<tr>
<td>Cutting and drilling fluid spill and disposal</td>
<td>During the drilling, cutting and drilling fluid will flow out of the well to the surface and must be disposed properly.</td>
<td>Safety containment will be installed around the rock sample and rig structure (below), thus the cutting and drilling fluid circulation (outflow) can be stored properly. The safety containment will be made from transparent material (e.g. plastic, etc), thus the drilling process can still be observed. Additional storage containment will be installed also to store more cutting and drilling fluid circulation out of the well or rock sample. The cutting and drilling fluid outflow circulation will be recorded and disposed according to regulations (university and state regulations), particularly if there is chemical content. A general procedure will be included together with HSE procedure.</td>
</tr>
<tr>
<td>Hazard during drilling or testing</td>
<td>Accident and hazard during drilling or testing could occur.</td>
<td>The drilling automation (through algorithm and computer control) is set up thus less or even no human intervention is needed during the test. However, supervision during the test is still needed. Before the test or drilling is started, safety meeting (prejob safety meeting) will be held to discuss the procedure and any potential hazard. Emergency shutdown is included in algorithm program and will active in case serious hazard occurs. The emergency shutdown can also be activated manually (intervention from human) in case the self-control program does not work properly. In case there is obstacle or restriction in circulation system that causing increase of the pressure, emergency shutdown system will be activated (included in algorithm) at certain limit of pressure and relief valve is installed to release the pressure. A general testing procedure (Standard Operating Procedure/SOP) will be set before drilling or testing. Minimum HSE equipment standard must be worn (e.g. safety glasses, hearing protection, etc). A certain range of zone isolation must be set thus unauthorized person will not be allowed to pass the isolation zone. A fire extinguisher will be available on location in case fire occurs due to overheating of electronic components or motor, etc. The hoisting system has locking mechanism to stop the movement in case lost control happens.</td>
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<tr>
<td>Communication error</td>
<td>Failure during data acquisition due to communication system error</td>
<td>System restart will be designed, so whenever data is failed to be acquired, by restarting the system, data acquisition can work properly.</td>
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<tr>
<td>Sensors error</td>
<td>Failure during data acquisition due to sensor failure</td>
<td>The sensors will be tested before and after installation. After each test or drilling, maintenance will be performed to confirm whether the sensors can still function properly.</td>
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<tr>
<td>Mobility of the rig</td>
<td>The mobility of rig is incorporated in the design. The rig structure will be placed above the table and wheels will be installed below the table, thus the rig can be mobilized. A brake system will be included in wheels to prevent undesired rig movement during drilling or testing due to vibration. Proper handling system will be installed so the rig can be moved safely and in convenient way. Modular design will be considered and as far as possible applied in the rig construction, such as electrical, computer, pump and disposal storage that can be disassembled or separated. This will make the mobilization and transportation more convenient and safe.</td>
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## 8 Price list

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<td>Saw Cut for Panel Elements Cat. 1</td>
<td>4,99 EUR/pcs</td>
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**Note:** The USD amount is based on a conversion rate of approximately 1.07 EUR to USD.
9 Technical drawings

Figure 40: Technical drawing: Drillrobot

Figure 41: Technical drawing: Machine bed
Figure 42: Technical drawing: Top drive

Figure 43: Technical drawing: Sled
Figure 44: Technical drawing: Percussion drilling tool

Figure 45: Technical drawing: BHA
Figure 46: Technical drawing: Schematic overview power and control
10 Funding plan

The funding is kindly secured by the ITE Engineering GmbH, located in Clausthal-Zellerfeld Germany.

Figure 47: Artistic depiction
11 Bibliography


