MODELING AND SIMULATION OF A HYDRAULIC DRILL FOR CONTROL SYSTEM DESIGN PURPOSES

Suvi Peltokangas
Automation and Hydraulic Engineering
Faculty of Engineering Sciences
Tampere University of Technology
Korkeakoulunkatu 10, Tampere, Finland
Email: suvi.peltokangas@tut.fi

Sirpa Launis
Markus Saarela
Sandvik Mining and Construction Oy
Tampere, 33330, Finland
sirpa.launis@sandvik.com
markus.saarela@sandvik.com

Jouni Mattila
Automation and Hydraulic Engineering
Faculty of Engineering Sciences
Tampere University of Technology
Korkeakoulunkatu 10, Tampere, Finland
jouni.mattila@tut.fi

ABSTRACT
In rock excavation processes, hydraulic rotary-percussive drilling is used for drilling and blasting in both surface and underground drilling operations. A hydraulic percussive drilling system is composed of percussion, rotation, feed, and flushing functions. In this paper, we detail the interaction of feed and rotation functions using a rock model. The feed actuator is a cylinder drive and a hydraulic motor actuator rotates the drill bit. The feed is force controlled and rotation is torque controlled by a feed reduction valve acting on the pressure compensator of the mobile hydraulic proportional directional control valve. In addition, in this work an individual load sensing variable displacement pump is used for both hydraulic functions. A suitable rock model is developed and verified against a measurement set. The inputs of the rock model are percussion drill flow rate, percussion pressure, feed force, and rotation torque, and the outputs are drill bit penetration rate and rotational speed. The modeling work is carried out to enable intelligent rock drilling control system development for changing rock conditions. The simulation results obtained verify that the simple rock model emulates various rock characteristics ranging from extremely hard rock like granite to softer minerals and that the changes in drilling parameters were as expected.

NOMENCLATURE

\( A_T \) Area of the cylinder plunger side A
\( A_B \) Area of the cylinder plunger side B
\( B_{eff} \) Effective bulk modulus
\( B_H \) Bulk modulus of the pressure line
\( C_C \) Flow coefficient of the pressure compensator
\( C_M \) Motor leakage coefficient
\( C_P \) Pump leakage coefficient
\( F_C \) Coulomb friction
\( F_S \) Stiction force
\( F_{fr} \) Friction force
\( K_C \) Normalized spring stiffness of the pressure compensator spring
\( K_r \) Flow coefficient of a flow channel
\( L \) Cylinder stroke
\( n_M \) Rotational speed of the motor
\( n_P \) Rotational speed of the pump
\( P_A \) Pressure at load side A
\( P_B \) Pressure at load side B
\( P_C \) Pressure compensated pressure
\( P_{LS} \) Load pressure
\( P_{perc} \) Percussion pressure
\( P_{RS} \) Requested supply pressure
\( P_S \) Supply pressure
\( P_T \) Tank pressure
\( P_{TR} \) Pressure limit for flow type determination through an orifice
\( P_0 \) Pre-compression pressure of the compensator spring
\( Q_A \) Flow to load side A from the proportional directional valve
slow dynamics, in the order of few hertz.

feed and rotation affecting the bit-rock interaction have relatively
ing regenerative reflected forces. This can be achieved by con-
forces should be effectively transmitted to the rock without caus-
travel through the drill rod to the drill bit. Generated impulsive
actions. These impulsive forces generate rapid stress waves that
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20 years [2–11]. Most studies have concentrated on rock me-
puter, a relatively simple simulation model for hydraulic rotary-
percussive drilling is constructed where only feed and rotation
affects the achieved penetration rate, is used for the rock model.
Flushing is assumed to be satisfactory. The goal is to achieve a
drilling simulator for testing different control system designs for
various rock conditions.
This paper is organized as follows. In the first section, mathe-
ematical models of each hydraulic drill subsystem are presented.
The second section focuses on modeling rock based on a specific
data set and comparing simulated data to measured data. In the
third section, simulation results from the combined drill and rock
model are presented. Finally, there is a short discussion about the
model and conclusions are drawn.

INTRODUCTION

According to [1], hydraulic percussive drilling rigs were in-
troduced to the market in the early 1970s. High power rock drills
doubled rock drilling capacities compared to previously popular
pneumatic rock drills. Hydraulic rock drilling has led to enhance-
ments in drilling accuracy and automation.

The percussive drilling parameters that affect drilling per-
formance are percussion power, feed force, rotation torque, and
flushing [1]. Percussion output power depends on the pressure
and the flow rate. Feed force keeps the drill bit in contact with the
rock and the bit is rotated between blows to provide fresh rock under the bit buttons. [1]

Percussive drilling has been extensively studied over the past
20 years [2–11]. Most studies have concentrated on rock me-
chanics and the modeling of impulsive forces in rock-bit inter-
actions. These impulsive forces generate rapid stress waves that
travel through the drill rod to the drill bit. Generated impulsive
forces should be effectively transmitted to the rock without caus-
ing regenerative reflected forces. This can be achieved by con-
trolling percussion cylinder stroke length or frequency, which
typically is about 100 Hz. The challenge therefore is that the
indirect control parameters of mobile hydraulic valve controlled
feed and rotation affecting the bit-rock interaction have relatively
slow dynamics, in the order of few hertz.

Rock braking mechanisms under percussive loading have
been studied in various papers. Sazidy et al. [12] modeled rock
as a visco-elasto-plastic material. Saksala et al. presented a nu-
umerical method for dynamic indentation [13]. Their simulation
method included a constitutive model for rock, which was a com-
bination of a viscoplastic consistency model and an isotropic
damage model, and they used a finite element method (FEM)
model for dynamic bit-rock interaction simulation. All modeled
bit-rock interactions were in the time range of a single percus-
sive blow. However, the frequency range of stress waves is in
the order of 10–30 kHz and Pettersson [14], for example, studied
stress-wave propagation and hydraulic control system simulation
in a multi-domain simulation set-up. However, in this study,
the developed simulation model did not take into account high
frequency phenomena, and control system designs developed in
the future will work with frequency ranges that can feasibly be
controlled by mobile hydraulic components. In the present pa-
per, a relatively simple simulation model for hydraulic rotary-
percussive drilling is constructed where only feed and rotation
are considered. However, information about percussion, which
affects the achieved penetration rate, is used for the rock model.
Flushing is assumed to be satisfactory. The goal is to achieve a
drilling simulator for testing different control system designs for
various rock conditions.

This paper is organized as follows. In the first section, mathe-
matical models of each hydraulic drill subsystem are presented.
The second section focuses on modeling rock based on a specific
data set and comparing simulated data to measured data. In the
third section, simulation results from the combined drill and rock
model are presented. Finally, there is a short discussion about the
model and conclusions are drawn.

MODELING DRILL FUNCTIONS

Pettersson et al. modeled the percussion function of a rock
drill as a hydraulic piston and a 4/3 valve. A rock drill cradle is
connected to either a hydraulic cylinder or a motor by pulleys
and cables or chains and sprockets. Here, a hydraulic cylinder is
considered to generate the feed force for drilling.

Both feed and rotation circuits have a pressure compensated
proportional directional control valve (PCPDCV) and a variable
displacement pump, as illustrated in Fig. 1. Figure 2 provides a
detailed diagram of the proportional directional pressure com-
pensated valve. The rotation motor is assumed to be a constant
displacement motor, and the pressure can be controlled utilizing
pressure reducing valves (PRVs).

Axial piston pump

In this simulation, a constant pressure control strategy is as-
sumed for the axial piston pump. The pump produces flow $Q_P$.
shown in Eqn. (1):

$$Q_P = n_P V_P \varepsilon - C_P p_S$$  \hspace{1cm} (1)$$

where $n_P$ is the rotational speed [rad/s], $V_P$ is the pump volumetric displacement [$m^3$/rad], $\varepsilon$ is the pump angle [0...1], $C_P$ is the leakage coefficient [$m^3/(s\cdot Pa)$], and $p_S$ is the pump supply pressure [Pa].

The pump angle is modeled as a function of the pressure difference in Eqn. (2) [15–17]:

$$\varepsilon = \frac{\tau_1}{\tau_2} + 1 (p_{RS} - p_S)$$  \hspace{1cm} (2)$$

where $\tau_1$, and $\tau_2$ are time constants defining the dynamics between the pump angle and the pressure difference.

The derivative of the pump feed pressure is given in Eqn. (3):

$$\dot{p}_s = \frac{B_H}{V_H} (Q_P - Q_{LS} - Q_H)$$  \hspace{1cm} (3)$$

where $B_H$ is the bulk modulus of the pressure line, $V_H$ is the volume of the pressure line, $Q_{LS}$ is the flow going to an actuator, and $Q_H$ is the flow going to a pressure relief valve.

**Pressure compensator**

The purpose of a pressure compensated valve is to achieve flow that is proportional to the opening area even though the load pressure changes. The opening of the pressure compensator is given in Eqn. (4) [18]:

$$u_C = \frac{p_{LS} + p_0 + K_C - p_C}{K_C}$$  \hspace{1cm} (4)$$

where $p_{LS}$ is the load pressure [Pa], $p_0$ is the pre-compression pressure of the compensator spring [Pa], $K_C$ is the normalized spring stiffness [Pa], and $p_C$ is the compensated pressure [Pa]. The opening of the pressure compensator is dimensionless and is limited to interval $[0, 1]$.

The orifice flow formula proposed by Ellman and Piché has a smooth transition between laminar and turbulent flow. The flow formula is used for simulating flows through valves in this simulator. The flow through the pressure compensator is given in Eqn. (5):

$$Q_C = \begin{cases} 
\frac{|p_{RS} - p_C| \sqrt{|p_{RS} - p_C| C_{C} u_C}}{C_{C} \left(3 - \frac{|p_{RS} - p_C|}{p_{TR}}\right)} & \text{for } |p_{RS} - p_C| \geq p_{TR} \\
0 & \text{for } |p_{RS} - p_C| < p_{TR}
\end{cases}$$  \hspace{1cm} (5)$$

where $p_{TR}$ is the pressure limit, which determines flow type as either turbulent or laminar, and $C_C$ is the flow coefficient of the valve [19].

The pressure differential $\dot{p}_C$ can be calculated according to Eqn. (6):

$$\dot{p}_C = \frac{B_{eff}}{V_{pc}} (Q_C - Q_{LS})$$  \hspace{1cm} (6)$$

where $B$ is the bulk modulus, $V_{pc}$ is the volume between the pressure compensator and the directional valve, and $Q_{LS}$ is the load flow [18].

The pressure reducing valve controls the pressure $p_{LS}$, which is an input to Eqn. (4). This is done simply using "Lookup Table
where the motor can be used to determine the lapping of the control valve; valves rate limiter blocks in series. The opening position are modeled in Simulink environment using a delay and path defined in Eqn. (7).

Proportional directional valve

Proportional directional valves for both the feed and rotation circuits are 4/3 valves. The flow through a flow channel can be calculated according to Eqn.(7):

\[
Q(p_1,p_2) = \begin{cases} 
|p_1 - p_2| \sqrt{|p_1 - p_2| K_u} & \text{for } |p_1 - p_2| \geq p_{TR} \\
3 - |p_1 - p_2| \frac{p_{TR}}{|p_1 - p_2|} & \text{for } |p_1 - p_2| < p_{TR} 
\end{cases}
\]
(7)

where \(K_u\) is the flow coefficient of the valve, \(u\) is the opening of the valve [0..1], and \(|p_1 - p_2|\) is the pressure difference across the flow path [19]. Basic 4/3 proportional directional valves can be simulated to consist of four different flow paths. Flows to the load are presented in Eqn. (8):

\[
\begin{align*}
Q_A &= Q(p_P,p_A) - Q(p_A,p_T) \\
Q_B &= Q(p_P,p_B) - Q(p_B,p_T)
\end{align*}
\]
(8)

where \(p_P\) is the pressure in the pump side of the valve, \(p_A\) is the pressure at the actuator side \(A\), \(p_B\) is the pressure at the actuator side \(B\), \(p_T\) is the tank pressure, and \(Q(\ldots)\) is a flow through flow path defined in Eqn. (7).

The dynamics between the valve control signal and the spool position are modeled in Simulink environment using a delay and rate limiter blocks in series. The opening \(u\) of each flow channel is a static non-linear function of the spool position. This function can be used to determine the lapping of the control valve; valves can be zero lapped, under lapped, or over lapped.

Motor

The simulated motor is a constant displacement motor and external leakage is neglected. Therefore, the volume flow out of the motor \(Q_M\) can be modeled as in Eqn. (9):

\[
Q_M = V_M n_M + C_M \Delta p
\]
(9)

where \(V_M\) is the theoretical volumetric displacement of the motor \([m^3/rad]\), \(n_M\) is the rotational speed of the motor \([rad/s]\), \(\Delta p = p_A - p_B\) is the pressure difference at the motor ports, and \(C_M\) is the motor leakage coefficient \([m^3/(s\cdot Pa)]\). The torque of the motor axle \(T_M\) can be modeled as in Eqn. (10):

\[
T_M = \Delta p V_M
\]
(10)

The pipes from the valve to the motor and from the motor to the valve are modeled as volumes. The pressure differential at the motor ports \(A\) and \(B\) are given in Eqn. (11a) and Eqn. (11b), respectively:

\[
\begin{align*}
\dot{p}_A &= \frac{B_{eff}}{V_a} (Q_A - Q_M) \\
\dot{p}_B &= \frac{B_{eff}}{V_b} (Q_M + Q_B)
\end{align*}
\]
(11)

where \(B_{eff}\) is the effective bulk modulus, \(V_a\) and \(V_b\) are pipe volumes at the motor side \(A\) and \(B\), respectively, and the positive flow direction for \(Q_B\) is from the valve to the motor.

Feed

The double-acting single-rod cylinder (70/50-2700) is modeled as the feed actuator. The pressure differentials on the \(A\) and \(B\) sides of the cylinder are modeled as in Eqn. (12a) and Eqn. (12b):

\[
\begin{align*}
\dot{p}_A &= \frac{B_{eff}}{A_A x} (Q_A - \dot{x} A_A) \\
\dot{p}_B &= \frac{B_{eff}}{A_B (L - x_c) + V_{0B}} (Q_B + \dot{x} c A_B)
\end{align*}
\]
(12)

where \(A_A\) is the area of the plunger side \(A\) and correspondingly, \(A_B\) is the area of side \(B\), \(x_c\) is the position of the plunger, \(L\) is the cylinder stroke, and \(V_{0A}\) and \(V_{0B}\) are dead volumes in sides \(A\) and \(B\).

For the force equation of the cylinder, the cylinder friction is also taken into consideration. Friction force can be modeled according to the steady-state motion relation between velocity and friction force presented by Canudas-de-Wit et al. [20]:

\[
F_{ss}(\dot{x}_c) = F_C \sgn(\dot{x}_c) + (F_S - F_C) e^{-(\dot{x}_c / v_s)^2} \sgn(\dot{x}_c) + \sigma_2 \dot{x}_c
\]
(13)

where \(F_C\) is the Coulomb friction level, \(\dot{x}_c\) is the velocity of the plunger, \(F_S\) is the level of stiction force, \(v_s\) is the Stribeck velocity, and \(\sigma_2\) is the viscous friction parameter.

ROCK MODEL

One commonly used bit-rock interaction model for percussive drilling relates force on the bit to the penetration rate using...
two consecutive phases: loading and unloading. Depouhon et al. modeled bit-rock interaction with the generalized bilinear interaction law. Their computational approach was based on semidiscretization via FEM. Their model did not account for rotation or flushing.

The idea behind the rock model in the present study is that rock can be modeled as a mass-damper system. The formula and the parameters of the rock model are defined experimentally. Measurement data where drill parameters were varied one at a time are used for the rock modeling. Holes were drilled into two kinds of homogeneous granite rock blocks.

Derivation of the model

A mass-damper system can be illustrated as in Eqn. (14):

$$\ddot{x}_c = \frac{1}{m} \left( \sum F - b \dot{x}_c \right)$$

(14)

where \(m\) is the mass, \(\sum F\) is the sum of forces affecting the system, and \(b\) is the viscous friction coefficient.

The penetration rate of modern hydraulic rock drills depends mostly on percussion power. Obviously, percussion power then needs to be included in the penetration rate equation. In addition to percussion power, feed force is also included in the penetration rate equation as the pressure difference between cylinder ports, and torque is calculated based on the pressure difference at the motor ports. A semi-empirical model for cylinder plunger acceleration is given in Eqn. (15):

$$\ddot{x}_c = \frac{c_1(r_{bit})}{m_1} \left( c_1(r_{bit}) g_{p1} Q_{perc} P_{perc} + g_{f1}(r_{bit}) \Delta P_{feed} + g_{t1} T_M + \left( g_{f1} \frac{\Delta P_{feed}}{T_M} \right) - b_1 \dot{x}_c \right)$$

(15)

where \(0.005 \leq T_M \leq 0.024\), \(r_{bit}\) is the radius of the bit, \(Q_{perc}\) is the flow to the percussion device, \(P_{perc}\) is the percussion pressure, \(\Delta P_{feed}\) is the pressure difference at the cylinder ports, and \(m_1, c_1, g_{f1}, g_{t1}, g_{f1}\) are parameters.

The parameters were manually defined based on test measurements. The parameters of the cylinder velocity part of the rock model are shown in Table 1, where the value for \(c_1\) is for a bit with a diameter of 57 millimeters. Parameters \(c_1\) and \(g_{f1}\) depend on the bit radius and need to be adjusted based on measurements.

The experimental rotation acceleration model is given in Eqn. (16):

$$\dot{\omega} = \frac{1}{a_1} \left( a_2(r_{bit}) \left( a_3 + a_4 T_M + \frac{1}{a_5 P_{perc}} + a_6 \Delta P_{feed} \right) - \frac{b_2}{2\pi} \omega \right)$$

(16)

The parameters for the rotational speed model are given in Table 2. The value for parameter \(a_2\) is for a bit diameter of 57 millimeters.

The viscous friction coefficients were fine-tuned for both rocks based on a comparison of the simulated and measured drill velocities. All measurements of the first rock type were compared to the simulated measurements and both simulated and

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**TABLE 1. PARAMETERS OF THE ROCK MODEL.**

<table>
<thead>
<tr>
<th>Variable</th>
<th>First rock type</th>
<th>Second rock type</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(m_1)</td>
<td>5500</td>
<td>5500</td>
<td>kg</td>
</tr>
<tr>
<td>(c_1)</td>
<td>1</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>(g_{p1})</td>
<td>1.5e-5</td>
<td>1.5e-5</td>
<td>(m^{-1}) \cdot s</td>
</tr>
<tr>
<td>(g_{f1})</td>
<td>277e-5</td>
<td>277e-5</td>
<td>(m^2)</td>
</tr>
<tr>
<td>(g_{t1})</td>
<td>-180e-5</td>
<td>-180e-5</td>
<td>(m^{-1})</td>
</tr>
<tr>
<td>(g_{f1})</td>
<td>5/3e-5</td>
<td>5/3e-5</td>
<td>(kg \cdot m^4 \cdot s^{-2})</td>
</tr>
<tr>
<td>(b_1)</td>
<td>4.63e6</td>
<td>4.4e6</td>
<td>(kg \cdot s^{-1})</td>
</tr>
</tbody>
</table>

---

**TABLE 2. PARAMETERS OF THE ROCK ROTATIONAL SPEED MODEL.**

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a_1)</td>
<td>1.8515</td>
<td>kg-m^2</td>
</tr>
<tr>
<td>(a_2)</td>
<td>60</td>
<td>m</td>
</tr>
<tr>
<td>(a_3)</td>
<td>13220.8691</td>
<td>kg \cdot m \cdot s^{-2}</td>
</tr>
<tr>
<td>(a_4)</td>
<td>10^{-4}</td>
<td>m^{-1}</td>
</tr>
<tr>
<td>(a_5)</td>
<td>10^{-5}</td>
<td>kg^{-2} \cdot s^4</td>
</tr>
<tr>
<td>(a_6)</td>
<td>10^{-5}</td>
<td>m^2</td>
</tr>
<tr>
<td>(b_2)</td>
<td>75</td>
<td>kg \cdot m^2 \cdot s^{-1}</td>
</tr>
</tbody>
</table>
measured velocities were normalized. Comparisons for the first and second rock type are shown Fig. 3 and Fig. 4, respectively. The corresponding comparisons for rotational speed are shown in Fig. 5 and Fig. 6, respectively. Colorbar in those comparison plots represents the number of data points inside a tile.

FIGURE 3. TILED HISTOGRAM PLOT OF THE NORMALIZED MEASURED AND SIMULATED DRILL VELOCITIES FOR THE FIRST ROCK.

FIGURE 4. TILED HISTOGRAM PLOT OF THE NORMALIZED MEASURED AND SIMULATED DRILL VELOCITIES FOR THE SECOND ROCK.

FIGURE 5. TILED HISTOGRAM PLOT OF THE NORMALIZED MEASURED AND SIMULATED ROTATIONAL SPEEDS FOR THE FIRST ROCK.

FIGURE 6. TILED HISTOGRAM PLOT OF THE NORMALIZED MEASURED AND SIMULATED ROTATIONAL SPEEDS FOR THE SECOND ROCK.

Simulation results for the rock model

The measurement set consisted of 38 drilled holes whose durations were approximately 20 to 30 seconds each. From those measurements, 27 holes were drilled in the first granite rock and 11 in the second granite rock.

The rock model inputs were torque, the pressure difference at the cylinder ports, percussion flow, and percussion pressure. Moving average filtering was applied on the torque for reduc-
FIGURE 7. MEASURED VARIABLES, NORMALIZED AND SMOOTHED USING A MOVING AVERAGE FILTER WITH A SPAN OF 0.65% FOR THE FIRST ROCK TYPE.

FIGURE 8. MEASURED AND SIMULATED DRILL VELOCITIES AND ROTATIONAL SPEEDS FOR THE FIRST ROCK TYPE.

FIGURE 9. MEASURED VARIABLES, NORMALIZED AND SMOOTHED USING A MOVING AVERAGE FILTER WITH A SPAN OF 0.65% FOR THE FIRST ROCK TYPE.

FIGURE 10. MEASURED AND SIMULATED DRILL VELOCITIES AND ROTATIONAL SPEEDS FOR THE FIRST ROCK TYPE.

Simulations for the other granite block using the same viscous friction coefficient resulted in similar shape velocity curves, but those curves were below the measured velocity curves. Decreasing the value of the viscous friction coefficient enabled the simulated curves to achieve the same level as the measured curves. One example of drilling into the second granite type is shown in Fig. 12 and the corresponding normalized inputs are shown in Fig. 11. Copyright © 2017 by ASME
FIGURE 11. MEASURED VARIABLES, NORMALIZED AND SMOOTHED USING A MOVING AVERAGE FILTER WITH A SPAN OF 0.65% FOR THE SECOND ROCK TYPE.

FIGURE 12. MEASURED AND SIMULATED DRILL VELOCITIES AND ROTATIONAL SPEED FOR THE SECOND ROCK TYPE.

SIMULATION OF THE WHOLE SYSTEM

In the simulation result shown in Fig. 13, the percussion power was kept constant. In another simulation, shown in Fig. 14, the viscous friction coefficients were decreased by 10% between 15 and 17 seconds; velocity increases. In order to prevent velocity change adequate control actions should be applied. Drilling from hard granite to broken granite without adequate control actions results in an increased penetration rate, which leads to unnecessary high percussion power and over feeding.

This can lead to, for example, breakage of bit buttons, drill string breakage, breakage of the drill shank, and drill hole deviations.

DISCUSSION AND LIMITATIONS

The rock model was verified based on a limited measurement set in Finnish granite hard rock drilling and therefore, the model was not verified for softer rock types such as limestone or fragmented rock. Clearly, more tests are needed to take into account, for example, the wearing of the bit and the effect of increasing the number of rods. Drill bit size and type most likely also have an effect on the model parameters and therefore more tests should be made in order to verify a wider range of experiments. The simulation model was verified in a full power drilling situation in granite and therefore it could not simulate any special situations other than changes in the viscous friction of the rock. However, based on the simulation results, the developed simulator behaves as expected when the condition of the rock changes from hard to fragmented.
CONCLUSION

Encouraging simulation results were obtained by modeling quite complex mobile hydraulic valve and LS-pump driven feed and rotation actuator interaction with the developed rock model. The largest differences between the measured and simulated feed and rotation velocities were in the drilling initialization phase and in the finishing phase. Seldom is rock a homogeneous material and during hard rock drilling, cavities or soft soil zones are frequently encountered. In addition, the propagation of rock cracks changes the rock’s ability to resist drilling. Estimation of the penetration rate is therefore only a rough average. The developed simulator provides an initial step for testing different control systems in changing rock conditions and with different drilling equipment. Simulation models need further development with a wider data set as the presented rock model does not always produce sufficiently accurate feed velocity and rotational speed values. As the rock model is verified only against Finnish granite hard rock, the model most likely needs to be updated for drilling of softer rocks such as limestone. However, changing the viscous friction coefficient of the rock model affects drilling velocity, which is initially enough for the development of a control system. In the future, a wider range of measurements will be made in order to validate the rock model for changing rock conditions. Models of both drill string and percussion circuit will be added to the simulator utilizing, for example, FEM modeling. New control system designs will be implemented and promising design solutions tested using a real-time control scenario with a full-size drilling set-up.

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REFERENCES


