Water Powered Percussive Rock Drilling
Process Analysis, Modelling and Numerical Simulation

GÖRAN TUOMAS

Luleå University of Technology
Department of Civil and Environmental Engineering • Division of Renewable Energy
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Department of Civil and Environmental Engineering
Luleå University of Technology
Luleå, Sweden

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PREFACE

This thesis is presented as a partial fulfilment of the requirements for the degree of Doctor of Philosophy (Ph.D.). The research was carried out at the Division of Renewable Energy, Luleå University of Technology. Financial funding was partly provided by Luleå University of Technology, Technology Link Foundation, The Research Council of Norrbotten, LKAB, and Wassara AB. Their support is gratefully acknowledged.

Many persons and organisations have been of assistance and contributed to this work.

First of all, I want to express my sincere gratitude to my examiner and supervisor, Professor Bo Nordell at Luleå University of Technology, for his guidance throughout the duration of the study. I am greatly indebted for his support, advice, and suggestions to my research work. Professor Anders Sellgren and Professor Shaoquan Kou have also been of great assistance and provided guidance during various parts of the work. Dr. Hongyuan Liu has assisted me with many interesting discussions and comments.

The successful construction of numerous components, equipment, and systems were performed by Martin Edman, Jouko Viiri, Veikko Lipponen, Tage Töyrä, and Lars Johansson at Idé Arctica in Övertorneå, Sweden. They are greatly acknowledged.

Gratitude is also expressed to those working at Wassara AB, especially Lars Öderyd and Roger Lärkmo, who has shown genuine interest and enthusiasm. Also, Ingemar Marklund and Andreas Malmberg, former presidents, are greatly acknowledged.

Other colleagues and friends are also acknowledged for their support in various ways.

Finally, I want to thank my family and especially my parents, Rolf and Else-Maj Tuomas, because their support has been invaluable.

Göran Tuomas

Luleå, October 2004
This thesis is devoted to problems, processes and systems related to the recently developed water powered percussive rock drilling method. The technology, which uses ordinary water to drive down-the-hole hammers, has been used to produce more than 6 million meters of blast holes within the mining industry. The method has several advantages such as low energy consumption, dust free environment and the capability to drill to virtually any depth. A natural disadvantage of this method is the need for relatively large amounts of preferably high quality water to drive the hammer tool, occasionally also leading to waste disposal problems.

To understand the central hammer tool in the system, the function of a 100 mm diameter hammer has been analysed, modelled and simulated. One-dimensional wave propagation theory was used for modelling impacts and axial motions in a drilling system. The rock was assumed to be an elastic-plastic material, where all absorbed energy was used for crush work. Simulation results show good agreement between measured and simulated piston blow frequencies (~60 Hz). A disadvantage with the hammer tool’s function is the discontinuous consumption of water, causing large pressure fluctuations in the feed water line. Measurements indicate peak pressures to be ~3 to 4 times larger than the lowest pressure. Since large pressure variations increase the risk of mechanical damages, a flexible element (pulsation dampener) to reduce the variations was developed. Test bench experiments show pressure fluctuations reduced by up to 40% with the prototype dampener.

For all drilling methods, an efficient rock penetration process is essential for the methods overall competitiveness. The process is also significant for the dynamic behaviour of the water powered hammer tool, since different rock properties have been shown to cause variations in the piston blow frequency. The general bit-rock impact process is therefore discussed and field measurements of the penetration rate during ~115 mm diameter well drilling are presented. A penetration process was also analysed with the assistance of a non-linear explicit FEM code, where the rock material was represented by an established constitutive model. Results show, e.g., the ratio between the indenter’s rebound- and initial kinetic energy to decrease with increased initial energy, where a small part of this initial energy is transmitted by stress waves into the formation.

During the penetration process, crushed rock is flushed away with outlet water from the hammer tool, i.e. used particle contaminated drill water should be recycled when the method is used at locations with limited water access and/or when waste disposal is difficult to accomplish. This has resulted in the development and construction of a prototype mobile cleaning system that makes re-use possible. Hence, this system is described together with measured and simulated results of the unit’s cleaning capacity.

Another phenomenon during the drilling process is the dissipation of a large part of the injected borehole energy into heat. The drill water and the formation will therefore be thermally influenced, providing the possibility to evaluate the ground thermal conductivity with the drill work. This new principle is presented in detail, together with an energy balance equation and a heat transfer analysis during drill work.
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1. INTRODUCTION

1.1 General

The technique of using water instead of air as an energy carrier in down-the-hole (DTH) drill hammers has been known for years. Technical difficulties associated with corrosion, cavitation and wear have, however, slowed down the development and made putting these ideas into practice difficult. This situation began to change in the early 1990s when the Swedish mining industry started to use water powered DTH hammers (Fig. 1.1) for production drilling of blast holes. The use of the hammer tool also meant continuous evaluation of and improvements to the system, considered today an effective and competitive drilling method. To date, the mining industry has drilled more than 6-million meters of blast holes with this water powered system. Some advantages with the method are low energy consumption, a dust free environment and the possibility to drill to virtually any depth. Disadvantages are mainly the necessity to have access to preferably high quality water and the voluminous waste. For instance, an ordinary Ø100 mm hammer tool requires between 0.2-0.4 m³/min of water to achieve a competitive rate of penetration. Since the same water is used for hole flushing, the waste may cause disposal problems and motivate the use of a cleaning system to make re-use possible.

Figure 1.1. Water powered DTH hammers.
1. INTRODUCTION

A method to produce larger boreholes or micro-tunnels with water powered DTH hammers is to combine them in a steel frame (Fig 1.2), usually referred to as a cluster boring machine (CBM). This method has been used by, e.g. The Swedish Nuclear Fuel and Waste Management Co. (SKB), currently in the process of designing the Swedish repository for nuclear waste. An alternative under consideration is to deposit the waste in containers placed in 250-300 meter long drifts with a diameter of 1.85 meter. For evaluation of this principle, SKB used two CBMs to produce a ø1.85 meter drift, with an intermediate size of ø1.44 m.

![Figure 1.2. An ø1850 mm cluster boring machine with 12 water powered ø150 mm DTH hammers and a rear mounted stabiliser. The machine and the drilling method was tested by The Swedish Nuclear Fuel and Waste Management Co. (SKB), currently in the process of designing the Swedish repository for nuclear waste.](image)

1.2 Objectives

The main objectives of this work were to increase the knowledge, determine vital process parameters, and develop models, systems, and processes for water powered DTH hammer drilling. This was realized by information retrieval, prototype development, experimental tests, theoretical analysis, model development, and numerical simulations. The intention was to broaden the applications and create platforms for further system improvements.

1.3 Scope

In this thesis, the general function and characteristics of water powered DTH hammers are initially discussed (Chapter 2), followed by a numerical analysis of a ø100 mm hammer tool. The large pressure variations generated by the hammer are also discussed and a prototype pulsation dampener to reduce the fluctuations is presented.

Chapter 3 is devoted to the rock penetration process, mainly because the penetration rate is of vital importance for the methods overall competitiveness as well as the mechanical rock properties affecting the hammer tool’s dynamic behaviour, such as variations in the piston’s blow frequency. General parameters affecting the penetration
rate are therefore discussed, together with results from an FEM-analysis of a sphere-rock impact process. Field measurements where the penetration rate has been investigated during water powered drilling of ten ø115-ø116 mm boreholes, are also presented.

A big disadvantage that naturally follows the water powered drilling method is the need of preferably high quality water to drive the hammer tool. This means that used particle contaminated drill water should be recycled when using the method at locations with limited water access or when waste disposal is difficult to accomplish or both. This has resulted in the development and construction of a prototype mobile cleaning system, thereby making re-use possible. Chapter 4 describes the system, together with measured and simulated results of the units cleaning capacity.

Finally, Chapter 5 presents an energy balance equation for a borehole. Since a large part of the injected drilling energy dissipates into heat, the drill water and formation will be thermally influenced during drilling. By measuring the injected energy and the temperatures of the drill water entering and leaving the borehole, the heat flow into the formation can be estimated, allowing for the possibility to evaluate the ground thermal conductivity with the drill work. This new principle is also discussed in Chapter 5 together with a heat transfer analysis during drilling.

All information in this thesis pertaining to the water powered DTH hammer drilling system is based on technology used by Wassara AB, Sweden. The work has also focused on Wassara’s ø100 mm hammer (W100), but the principles apply to other sizes as well.
2. WATER POWERED DTH HAMMERS

2.1 Function

During drilling with water powered DTH hammers, the tool is positioned at the front of the borehole while energy is transferred through the drill string in the form of pressurised water, mechanical torque, and a mechanical feed force. The main task of the hammer tool is to convert the potential energy of pressurised water into an oscillating piston movement. Via mechanical impacts, the kinetic energy of the piston is transferred to the bit and finally into the rock (Fig. 2.1). Rock fragmentation occurs at highly pressurised contact zones between the bit buttons and the rock. By rotating the bit, thereby creating new impact positions for the buttons, new rock will be fragmented and the penetration process continues. The debris is flushed away to the outside of the drill string by outlet water from the hammer \( \text{(Paper I)} \).

![Figure 2.1. Main parts in water powered DTH hammers.](image)

Ordinary water powered DTH hammers consist of three moving parts: the drill bit, the piston, and the control valve (Fig. 2.1). The piston and the control valve are made to self-oscillate when a pressure difference between the tools inlet and outlet ports occurs. Typical frequencies range from 30 to 70 Hz, depending on tool size, water pressure, rock properties, etc.

Generally, it is the lighter control valve that controls the flow to and from the heavier piston. When the control valve is in the left position (or actually \( x_S < x_A \) in Fig. 2.1), the piston feed chamber 1 is connected with the outlet channel in the centre of the hammer through the hole in the control valve. A force acting to drive the piston to the left (Fig. 2.1) is then obtained, since chamber 2 is always connected to the high-pressure inlet. The control valve positioned to the right, as seen in Figure 2.1 \( (x_S > x_A) \), will expose piston feed chamber 1 to the high-pressure inlet and the piston will eventually accelerate and move towards the bit. This working principle is possible due to the piston’s displacement area in chamber 1 being larger than that of chamber 2. To make the piston oscillate and generate drill work, the control valve must move back-and-forth in a well synchronised motion with the piston. This is accomplished by using three pilot-pressures to control forces on the control valve. High pressures in pilot inlets 2 and 3 (Fig. 2.1) generate forces on the control valve in a negative x-direction, while a high pressure in the pilot inlet 1 works in the positive direction. In a typical hammer tool, the
displacement area for pilots 1 and 3 is twice that of pilot 2. Together with a constant high pilot pressure, this gives the control valve the positioning logic presented in Table 2.1, where condition C must be active to accelerate the piston towards the bit. Either one of the other conditions, A, B, or D in Table 2.1, will redraw the piston.

**TABLE 2.1**

Positioning logic for the control valve. The value “1” represents a high pressure and “0” represents a low. Parameter \( p_1 \) is the pressure condition in piston chamber 1, since this follows with the positioning condition \( (x_S > x_A) \) of the control valve.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Pilot 1</th>
<th>Pilot 2</th>
<th>Pilot 3</th>
<th>( x_S &gt; x_A )</th>
<th>( p_1 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>0</td>
<td>1</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>B</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>C</td>
<td>1</td>
<td>1</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>D</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

The required synchronisation of the components is accomplished by using the piston to open or close channels that connect pilots 1 and 3 with pressure chambers 1 or 2 (at positions \( x_{B1}, x_{B2}, \) and \( x_C \) in Fig. 2.1). The working function is easily understood when it is initially assumed the piston has just impacted with the bit and the control valve has reached its left end position (at time \( t_1 \) in Fig. 2.2). At this time, condition A is active, since the conditions \( x_{p1} > x_C, x_{p1} > x_{B1}, \) and \( x_{p2} > x_{B2} \) are true, and chamber 1 has no pressure, thereby causing the piston to accelerate away from the bit and increase its speed until time \( t_2 \). Here, condition C becomes active, since conditions \( x_{p2} < x_{B2}, x_{p1} < x_{B1}, \) and \( x_{p1} < x_C \) create a high-pressure in pilot 1, rapidly shifting the control valve’s position towards the right in Fig. 2.1, i.e. the net forces on the piston now act against the bit. The piston will stop at time \( t_5 \), reaching its maximal left end position. This is usually a slight distance away from the tool body to avoid impact, leading to a smooth change of direction and reduced vibrations. After the directional change at \( t_5 \), the same forces remain active, meaning that piston retardation has changed into acceleration towards the bit. At time \( t_6 \), before impact, pilot 3 will rise, since \( x_{p1} > x_C \) accomplishes condition D in Table 2.1. Condition D will only be temporarily active and at time \( t_7 \), condition A will re-activate. The control valve changes its position from right to left (in Fig. 2.1) and a new work cycle begins.

![Figure 2.2](image-url)

*Figure 2.2. Principle displacement and velocity of the piston and the control valve. T is the time between piston-bit impacts.*
2.2 Hammer analysis

Detailed knowledge of the hammer function is valuable for, e.g., predicting system behaviour during different working conditions. The design work may also be improved, since internal hammer tool parameters like port positions, piston design, etc., may be altered virtually, resulting in more efficient tools and lower development costs.

In Paper II, a hammer model was defined (Fig. 2.3) and motion equations are presented together with results from a system simulation with a ø100 mm water powered DTH hammer. The presented model is based on one-dimensional wave propagation theory for the calculation of axial motions and mechanical impacts (LUNDBERG 1980, 1982, 1985). Drill string dynamics are included, since a DTH drill hammer is an integrated part of the drill string. This first approach to realize a water powered hammer model is one-dimensional, with rotational and bending motions in the drill string being omitted. The model also assumes an incompressible fluid and excludes energy dissipation mechanisms like material and fluid damping. The omission of damping means that previously attained vibrations will affect consecutive impacts in multi-impact analyses. The error generated by excluding or including previously generated waves is not investigated and needs further attention.

The formation in the model was assumed to have an elastic-plastic behaviour (LUNDBERG 1985) where all absorbed energy is used for crush work. This is a simplification of the true interaction, where, e.g., water flow, indenter shape, rock fractures, bit rotation, etc., affect performance.

The system model consists of four components (Fig. 2.3). Component 1 represents the rotation head, drill string, and the tool body, component 2 is the control valve, component 3 is the piston, and component 4 is the bit. Each component holds its own force wave functions $N_{pk}(x-ct)$ and $N_{nk}(x+ct)$, where index k represents the component number. The initially wave-free components attain waves from external forces $F_t$, $F_s$, and $F_p$ (Fig. 2.3) that are assumed to act upon the left end of the drill string, control valve, and piston. Force $F_t$ is the thrust force, assumed constant in the analysis. Forces $F_t$ and $F_p$ are pressure induced and defined with assistance from several different logical conditions (see Paper II).
Results from the simulations are, for example, the time-displacement curves of elements in the piston, control valve, shaft, and tool body (Fig. 2.4), which define contact surfaces CS1, CS2, and CS4 (Fig. 2.3). It is shown in Fig. 2.4 how the movements of the control valve are limited between tool body elements in CS1 and CS2, and how the piston impacts the shaft in the bit. The synchronization of the piston and the control valve, vital for an efficient tool function, is also revealed. The time-displacement curve for the tool body and shaft elements (Fig. 2.4) also provides information about the penetration into the formation. However, since the buttons in the simulation are assumed to penetrate the formation at the same positions, a true rate of penetration (ROP) cannot be determined. In reality, torque acting on a drill string rotates the bit, giving new angular and axial positions. This process was not implemented in the current model and requires further attention. Other results from the simulations are the mean piston speed (Fig. 2.5) and mean control valve speed (Fig. 2.6).

Good agreement between simulated and measured time difference between successive piston blows (~0.016 ms as per Fig. 2.4 and Table 3.4) was achieved when the pistons coefficient of restitution was ~0.15 (Fig 2.5). The relation between the coefficient of restitution and the rock properties was not investigated. For a given hammer, the frequency mainly depends on parameters like the hammers feed pressure, drill bit condition, drill bit type and formation stiffness. The impact between the piston and the bit consisted of several bounces with renewed contact, with the maximum contact force being ~680 kN (Fig. 2.7). Energy transfer during the impacts was also evaluated (Fig. 2.8).

The hammer function was also analysed with an explicit non-linear FEM code (LS-DYNA) as a complement to the described analysis. Advantages with commercial FEM codes are the possibility to easily include more degrees of freedom and to change both materials and geometries. Making a coupled fluid-structure analysis is also possible, i.e. an analysis where the water flow in the hammer interacts with the mechanical components. Since coupled calculations usually require extensive computer capacity, a simplified approach was used where the pressure induced loads were defined with the assistance of several different logical conditions (see Paper II). To do this, a user defined load subroutine was required and added into the LS-DYNA code, allowing for nodal positions, velocities, and accelerations at each time step to be obtained and used in functions to calculate various loads in the model. The used nodal data may also be defined as a mean value of data from several nodes, reducing the risk of instability. Hydraulic functions, such as valve opening and closure, are thereby easily simulated. The model set-up in the FEM analyse was similar to the previously described model, yielding well-corresponding results. Work has now been initiated to analyse the drilling process with the drill string rotation included.
2. WATER POWERED DTH HAMMERS

Figure 2.4. Calculated displacement of elements defining contact surfaces CS1, CS2 and CS4.

Figure 2.5. Calculated mean velocity of piston elements.
2. WATER POWERED DTH HAMMERS

Figure 2.6. Calculated mean velocity of control valve elements.

Figure 2.7. Calculated contact force between the piston and the bit.
Figure 2.8. Calculated energy partitioning for the first impact. The bold solid curve represents the total amount of energy delivered into the system, the dashed line is the energy in the piston, and the regular solid line represents the energy delivered into the formation.
2.3 Test bench measurements

The pressure has been measured and recorded at different positions in a ø100 mm water powered DTH hammer. The tests were performed during hammer operation in a test bench (Fig. 2.9 and Paper III) equipped with a PC-based measurement system. Three pressure sensors were installed in the hammer to measure pilot pressures 1 and 3 together with the water feed pressure (see Fig. 2.1). The feed pressure was also measured ~5 meters behind the hammer.

Measurement results show large variations in water feed pressure. During a 0.05 s measuring period, the pressure varied between ~8 to ~23 MPa (Fig. 2.10) when the mean pressure was ~18 MPa. Even greater peak pressures (up to ~33 MPa) were detected in the pilot channels (Fig. 2.11-2.12). Figures 2.11 and 2.12 also reveal, for example, the time when the piston and control valve opens/closes hydraulic ports in the hammer tool.

![Figure 2.10. Feed water pressure measured inside a ø100 mm water powered DTH hammer (thin line) and ~5 meter behind the hammer (thick line). The mean pressure was ~18 MPa.](image-url)
2. WATER POWERED DTH HAMMERS

Figure 2.11. Measured pressure in pilot channel 3 inside a ø100 mm water powered DTH hammer (see Fig. 2.1).

Figure 2.12. Measured pressure in pilot channel 1 inside a ø100 mm water powered DTH hammer (see Fig. 2.1).
2.4 Pulsation dampener

The main reason for the large pressure variations in the feed water is the discontinuous working principle, i.e. the varying flow rate required by the hammer tool. For example, with a ø100 mm hammer tool, the theoretical mean water consumption (without considering leakage) is only ~1/7 of the maximal consumption. The highest consumption, i.e. low feed pressure, is obtained when the piston is at its maximal speed (~10 m/s) just before impact with the bit. When the piston has a negative speed, after impact, water is fed back into the pressure line, thereby increasing the pressure. This behaviour causes many problems such as vibrations in pump packages and increased load on the mechanical components. A method to reduce pressure variations is to install a flexible element (pulsation dampener) into the feed water line, which increases/decreases its volume according to the actual pressure level. The necessary ideal volume variation of a dampener may be estimated by

\[ V_{\text{var}} = \int_0^T (q - \bar{q}) \, dt \]  
(2.1)

where \( V_{\text{var}} \) is the volume variation, \( q \) is the water flow rate, \( \bar{q} \) is the mean flow rate, \( T \) is the cycle time, and \( t \) represents time. Several demands must, however, be fulfilled to achieve an optimal functioning hammer-dampener system. The pulsation dampener must be fast enough to respond to the rapid pressure variations behind the hammer tool as well as be positioned close to the hammer, so that the time delay caused by the pressure propagation time will be short. Other requirements are, e.g., the necessity to endure the difficult environment with heavy vibrations and with the fluctuating pressure.

2.4.1 Prototype test

Experiments were performed with a 600 mm long, 62 mm inner diameter pulsation dampener combined with a water powered ø100 mm hammer. The dampener design (Fig. 2.13) includes an outer steel casing and an inner carbon fiber/epoxy pipe with a rubber sealing. The pressure smoothening effect is obtained by allowing the carbon fiber/epoxy pipe to expand at higher pressures, thereby increasing the centre volume. The rubber sealing is required to prevent water from flowing into micro cracks in the carbon fiber/epoxy pipe, since this would destroy the component. For the ø100 mm hammer tool, a dampener should be able to expand/contract approximately ±10 cm³ (equation (2.1) and Fig. 2.14).

During test bench experiments, the pressure variations with time were recorded with and without the dampener at 10, 12, 14, 16, 18, and 20 MPa mean pressure. Summarized results (Fig. 2.15) show reductions in the variation by up to 40% at 14 MPa feed pressure and ~5 meters behind the dampener. However, the experiments must be followed up with endurance tests to guarantee a long term functioning component.
Figure 2.13. Design principle of a dampener to be inserted into a fluid pipe for reduction of pressure variations. The pulsation dampener was developed by G. Tuomas and the SICOMP company in Öjebyn, Sweden.

Figure 2.14. Estimated ideal volume variation of a pulsation dampener for a ø100 mm water powered hammer tool.
Figure 2.15. Measured pressure variations ~5 m behind a ø100 mm water powered DTH hammer. The two lower lines represent variations with a 600 mm long, ø62 mm pulsation dampener installed. The two upper lines represent variations without a dampener.
3. ROCK PENETRATION

3.1 General

The rate of penetration (ROP) of different drilling methods is closely related to the methods of productivity and competitiveness (BORQUEZ 1981, LILJEKVIST 1995, INGMARSSON 1998). Many studies have therefore been performed to understand the penetration process and optimize the systems (ANDERSSON 1981, KAHRAMAN et al. 2003, KOU 1995, LINDQVIST 1982, LIU 2004, LUNDBERG 1971). For percussive rock drilling, the ROP depends mainly on (i) operational variables, such as rotational speed, thrust force, impact energy, impact frequency, and flushing; (ii) bit design, i.e. number of indenters, indenter positions, indenter shape, flush hole position, flush hole size, etc.; and (iii) the mechanical rock properties.

The influence of rotational speed on the ROP has been studied by, e.g., HARTMAN (1966) and ÖDERYD (2004). Results show the highest ROP to be achieved when the indenter’s new positions are at a critical distance away from the existing craters when the next impact occurs. At the optimal distance, chips are formed with a minimum of energy, thereby increasing the ROP.

The purpose of thrust force is to keep the drill bit in contact with the rock. NORDLUND (1989) studied how the thrust force affects the general performance and concluded that the magnitude must be selected with respect to efficiency, stress level, drill string stability, and bit wear. HUSTRULID and FAIRHURST (1971a, 1971b, 1972a, 1972b) also conducted thrust calculations derived from a detailed theoretical and experimental study of the percussive drilling of rock and developed a general equation for the penetration rate

\[
PR = \frac{E f T}{AS}
\]

where \( PR \) is the penetration rate [m/min], \( E \) is the energy per blow [J], \( f \) is the blow frequency [blow/min], \( T \) is the energy transfer coefficient, \( A \) is the drill hole area [m\(^2\)], and \( S \) is the rocks specific energy [J/m\(^3\)]. Equation (3.1) shows how the penetration rate is proportional to both blow energy and blow frequency, and inversely proportional to the specific energy, \( S \), representing the rock properties. As suggested by MCCARTY (1982), the rocks compressive strength should replace the specific energy parameter in equation (3.1), since specific energy is a difficult parameter to define. MELLOR (1972) suggested that the specific energy may be defined as \( S \approx \sigma_c^2/(2E) \), where \( \sigma_c \) is the rock compressive strength and \( E \) is the Young’s modulus. PROTODYAKONOW (1962) described the coefficient of rock strength (CRS) as a measure of the rocks impact resistance. PAONE et al. (1969) showed that the compressive strength, tensile strength, Shore hardness, and static Young’s modulus correlated well with penetration rates in nine hard, abrasive rocks. Another measure of rock properties to estimate the ROP is the drilling rate index (DRI), an empirical measure based on laboratory experiments of rock samples where brittleness and drillability are tested.
3.2 Indentation analysis

Mechanical rock properties have shown to be important parameters in the dynamic behaviour of water powered DTH hammers. Drilling in elastic high strength rock means that a greater amount of energy rebounds to the piston from the bit, compared with drilling in soft porous formations. Observations reveal the piston blow frequency to vary up to 40% depending on the rock properties (ÖDERYD 2004), indicating that it is possible to roughly classify some rock properties by observing the hammer’s behaviour. This idea is closely related to the principles behind the Schmidt hammer test, for which the rebound number is used to estimate, e.g. the compressive strength (SCHMIDT 1951, CARGILL and SHAKOOR 1990). To approach the problem, Paper IV presents a model and simulations of a simplified indentation process. The purpose here was to estimate the energy partitioning and the mechanical state in a general high-strength rock material during the indentation process. In this study, four different axi-symmetric indentation simulations with the explicit LS-DYNA FEM-code are presented. Common to all of these is the general model setup (Fig. 3.1), where the indenter was modelled as a rigid ø14.5 mm sphere. Material data for a granite rock material were obtained from the literature, and the rock was represented by the Johnson-Holmquist (JH-2) constitutive model (JOHNSON and HOLMQUIST 1994, CRONIN et al. 2003). This enabled the effects of strain-rate hardening and damage to be included in the analyses. The first simulation was quasi-static, since this was required for parameter calibration and for comparison of results against experimental data. The other three analyses were dynamic, where the rigid spheres initial kinetic energies were 4 J, 9 J, and 16 J, corresponding to the mass 0.5 kg and the initial speeds 4 m/s, 6 m/s, and 8 m/s.

The model grid consisted of 0.2 mm quadratic elements within the inner area (Fig 3.1) and 0.4 mm elements in the outer area. To avoid wave reflections from free surfaces, the side and bottom boundaries were defined as non-reflective, thereby reducing the necessary model size and defining a semi-infinite model. This also provided the possibility to determine the amount of elastic stress wave energy passing through the boundaries.

![Axi-symmetric model for simulation of indentation processes.](image-url)
3. ROCK PENETRATION

Results from the quasi-static FEM analysis were compared with semi-empirical data from KOU (1995). Good agreement was achieved between the semi-empirical $p-F$ curves based on a quasi-static simulation (Fig. 3.2). Since data for the critical energy release rate $G$ was unavailable for granite, the value $G=88 \text{ J/m}^2$ was estimated by equation $G=33.769+0.267\sigma_c$, which is based on empirical results by Fong and Nelson (WHITTAKER et al. 1992). Lines for $G=110 \text{ J/m}^2$ and $G=70 \text{ J/m}^2$ are also presented for comparison.

![Figure 3.2. Calculated quasi-static indentation depth–force curve for a ø14.5 mm rigid sphere penetrating 1.5 mm into granite. The dashed lines are the corresponding semi-empirical experimental results for different critical energy release rates, $G$.](image)

In the dynamic indentation analyses, the strain rate hardening behaviour of the rock was shown to affect the solution. Strain rates $\dot{\varepsilon} > 1000 \text{ s}^{-1}$ occurred in all the dynamic analyses, with the largest value $\dot{\varepsilon}=12,814 \text{ s}^{-1}$ occurring at time $t=0.07 \text{ ms}$ in the analysis with 16 J impact energy (Fig. 3.3). The calculated results also indicate differences between the quasi-static and dynamic indentation depth-force curves. The force is slightly higher in the dynamic analysis (Fig. 3.4), i.e. a sign of a stiffer rock response.

As expected, damage results from the performed simulations show increased damage with increased initial sphere energy (Fig. 3.5). The side crack causing the formation of chips is also longer in the 16 J (8 m/s) analysis, compared to the other two (Fig. 3.5). The maximum pressure, $p=3,403 \text{ MPa}$, occurred at the sphere-rock contact surface at time $t=0.115 \text{ ms}$ in the 16 J analysis (Fig. 3.6). The influence of the assumed rigid indenter has not been investigated.

Results from energy partitioning analyses (Fig. 3.7) show the kinetic energy of the rigid sphere to be ~46% of the initial energy 4 J, ~40% of 9 J, and 36% of 16 J, after completed impact. The total amount of energy in the model is also varying during the process (Fig. 3.8). The energy reduction in Fig. 3.8 is explained by the fact that energy is transmitted into the rock as elastic stress waves. For the 16 J analysis, 15.21 J are...
present in the model as kinetic and internal energies after the finished impact, i.e. ~5% of the initial energy is transferred into the formation.

Figure 3.3. Calculated effective strain rate in a granite rock material, $0.07 \cdot 10^3$ seconds after impact of a 0.5 kg $\phi$14.5 mm rigid sphere with 8 m/s initial speed. At this time, the largest detected strain rate ($\dot{\varepsilon} = 12,814 \text{ s}^{-1}$) occurred.

Figure 3.4. Calculated indentation depth-force curves for 0.5 kg rigid $\phi$14.5 mm sphere-granite impacts. The spheres initial speed was 4 m/s, 6 m/s, and 8 m/s.
Figure 3.5. Calculated damage in a granite rock material after impact of a ø14.5 mm rigid 0.5 kg sphere. The red zones represents full damage (D=1) while the dark blue zones corresponds to an intact material.

Figure 3.6. Calculated pressure distribution in a granite rock material, $0.115 \cdot 10^{-3}$ seconds after impact of a 0.5 kg ø14.5 mm rigid sphere with initial speed 8 m/s. At this time the largest detected pressure ($p=3,403$ MPa) occurred.
3. ROCK PENETRATION

Figure 3.7. Calculated kinetic energy for a 0.5 kg rigid ø14.5 mm sphere during impact with a granite rock material. The sphere's initial speed was 4 m/s, 6 m/s, and 8 m/s.

Figure 3.8. Variations in the model's total energy during a simulated impact between a rigid ø14.5 mm sphere and a granite rock material. The sphere's mass was 0.5 kg and its initial speed was 8 m/s.
3.3 Field measurements

In the Prinz Sköld mining area, the mining company LKAB in Malmberget, Sweden has drilled several vertical boreholes from the ground with ø100 mm water powered DTH hammers (ÖDERYD 2000). The holes were drilled to perform safety investigations and install rock surveillance systems. Measurements of the penetration rate during this drilling were performed, with the aim to mainly compare the effectiveness of four different ø115-ø116 mm drill bits (Fig. 3.9 and Table 3.1). The formation consists of granite and leptite with 184 MPa and 270 MPa compressive strengths. Ten boreholes were drilled at two locations in close proximity, A and B, where boreholes nos. 1 to 3 were drilled at site A and nos. 4 to 10 were drilled at site B. For all boreholes, the drill hammers feed pressure was 18 MPa at 0-100 m depth and increased thereafter to 19 MPa for compensation of losses. Results from the measurements are presented in Tables 3.2 and 3.3 and Figures 3.10 and 3.11. Illustrated in Fig. 3.11 is an ROP decrease with an increased borehole depth that is probably due to the increased confinement pressure, thus increasing the rock strength. Numerical analysis of penetration at greater depths should therefore include the confinement pressure together with a pressure-hardening rock material model (see section 3.2).

\[\text{TABLE 3.1}\]

<table>
<thead>
<tr>
<th>Bit</th>
<th>Diameter [mm]</th>
<th>Mass [kg]</th>
<th>Indenter type</th>
<th>Indenter size [mm]</th>
<th>No. of indenters</th>
<th>Flush hole size [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seco</td>
<td>ø116</td>
<td>8.7</td>
<td>Ballistic</td>
<td>14</td>
<td>17</td>
<td>2 x ø18</td>
</tr>
<tr>
<td>Sandvik C</td>
<td>ø116</td>
<td>9.0</td>
<td>Ballistic</td>
<td>16</td>
<td>14</td>
<td>2 x ø12.5</td>
</tr>
<tr>
<td>Sandvik P</td>
<td>ø116</td>
<td>9.0</td>
<td>Spherical</td>
<td>14</td>
<td>19</td>
<td>2 x ø16</td>
</tr>
<tr>
<td>Driconeq</td>
<td>ø115</td>
<td>9.3</td>
<td>Spherical</td>
<td>14</td>
<td>14</td>
<td>2 x ø15</td>
</tr>
</tbody>
</table>

\[\text{TABLE 3.2}\]

Mean penetration rate at site A for drilling depths 0-200 m.

<table>
<thead>
<tr>
<th>Borehole No.</th>
<th>Bit</th>
<th>ROP [m/min]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Seco</td>
<td>0.41</td>
</tr>
<tr>
<td>2</td>
<td>Sandvik P</td>
<td>0.45</td>
</tr>
<tr>
<td>3</td>
<td>Driconeq</td>
<td>0.41</td>
</tr>
</tbody>
</table>

\[\text{TABLE 3.3}\]

Mean penetration rate at site B for drilling depths 0-150 m.

<table>
<thead>
<tr>
<th>Borehole No.</th>
<th>Bit</th>
<th>ROP [m/min]</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>Sandvik C</td>
<td>0.54</td>
</tr>
<tr>
<td>5</td>
<td>Driconeq</td>
<td>0.37</td>
</tr>
<tr>
<td>6</td>
<td>Sandvik P</td>
<td>0.33</td>
</tr>
<tr>
<td>7</td>
<td>Sandvik C</td>
<td>0.39</td>
</tr>
<tr>
<td>8</td>
<td>Sandvik P</td>
<td>0.38</td>
</tr>
<tr>
<td>9</td>
<td>Sandvik C</td>
<td>0.47</td>
</tr>
<tr>
<td>10</td>
<td>Sandvik P</td>
<td>0.39</td>
</tr>
</tbody>
</table>
Figure 3.9. A ø115 mm drill bit from Driconeq (left picture) and ø116 mm bits from Sandvik (type P and C) used in the drilling project, together with ø100 mm water powered DTH hammers.

Figure 3.10. Measured ROP during drilling of ten boreholes (Tables 3.2 and 3.3) with four different ø115-ø116 mm bits and ø100 mm water powered DTH hammer.
3. ROCK PENETRATION

A study of the ROP was also performed during upwards directed production drilling of blast holes at the LKAB iron ore mine in Malmberget, Sweden (Table 3.4). The reasons for the large difference in ROP during upward and downward directed drilling are not fully investigated. One explanation seems to be a more efficient removal of crushed material during upwards directed drilling, shown here by the generally larger particle sizes.

**TABLE 3.4**

*Summarized results from upwards directed drilling with water powered DTH hammers.*

<table>
<thead>
<tr>
<th>Hammer:</th>
<th>Wassara W100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bit:</td>
<td>Seco ø116 mm</td>
</tr>
<tr>
<td>Rock:</td>
<td>Iron ore in Malmberget, Sweden.</td>
</tr>
<tr>
<td></td>
<td>Compressive strength ~110 MPa</td>
</tr>
<tr>
<td>Drilling direction:</td>
<td>Upward (inclined)</td>
</tr>
<tr>
<td>Piston frequency:</td>
<td>60 Hz</td>
</tr>
<tr>
<td>Operating pressure:</td>
<td>18 MPa (180 bar)</td>
</tr>
<tr>
<td>Water flow:</td>
<td>190 l/min (new hammer) – 280 l/min (max pump capacity).</td>
</tr>
<tr>
<td>Penetration rate:</td>
<td>0.9 m/min</td>
</tr>
</tbody>
</table>
4. PUMP AND RECIRCULATION SYSTEMS

4.1 General

A disadvantage with water powered DTH hammers is the large flow rate of preferably high quality water required to drive the hammer tool. For instance, an ordinary ø100 mm hammer-tool requires between 0.2-0.4 m³/min to achieve a competitive rate of penetration, meaning that the water should be recycled when this drilling method is used at locations with limited water access or when waste disposal is difficult to accomplish or both (Fig. 4.1).

![Image of a drilling system with recirculation]

Figure 4.1. Principle flow in a drilling system with re-circulation.

The concentration of particles in the drilling water depends mainly on the actual water flow rate, penetration rate, and the density of the drilled rock. Mass concentrations (w/w) between 4-12% are common for rock drilling with an ordinary ø100 mm hammer, corresponding to approximately 13-27 kg/min particle flow, i.e. high-capacity cleaning equipment has to be used. Particle size distribution varies due to a certain number of factors, such as rock properties, drill bit design, and impact energy. An important limiting factor during downwards vertical or inclined drilling is the speed of the flushing water, since this must be faster than the particles settling speed. Particles will otherwise settle in the borehole and be re-crushed by the drill bit until their size is small enough to follow the flow. Particles generated during, for example, ø115 mm well drilling are usually smaller than 1 mm with mass median sizes ($d_{50}$) around 0.1 mm.

For the technique to be successful, the fluid cleaning system must be correctly designed and implemented, since fluid quality directly affects component life. Abrasive particles or aggressive chemical substances in the feed water or both significantly reduce tool life, especially when inexpensive tools made of hardened steel are used. However, it is possible to use tungsten carbide as a tool material, though this increases the cost and this material is normally only used in mud driven tools. For this reason, knowledge of how different water related parameters affect the life of a given tool or material is essential when designing cost effective systems.

Interesting data have been obtained from practical usage of these tools, especially within the mining industry where automated drill-rigs have produced millions of meters of ø115 mm blast holes. Results from water-analysis and data of the corresponding tool-life show the time between repairs corresponding about 1,500 drilling meters in iron ore when the feed-water contains max. 0.02% w/w solids. The mean penetration rate during these drillings was ~0.9 m/minute, giving a total of approximately 6 million piston blows between repairs, since the piston blow frequency is about 60 Hz. Other experiments have shown a drastically reduced life by large amounts of solids in the feed-water. For example, life less than 100 drill-meters have been measured when the feed-water contained about 0.5% w/w solids (ÖDERYD 2004).
4.2 Prototype system

A complete mobile prototype pump- and recirculation unit (Fig. 4.2) to be used with water powered DTH hammers was constructed (Paper V). The unit includes all components required for efficient drilling, i.e. systems for both pressurising drilling fluid and particle-fluid separation to enable recycling. In this system (Fig. 4.3 and Table 4.1), a diesel engine driven plunger pump (P4) pressurises the water used for driving the hammer tool and for flushing the borehole. The particle contaminated drill water is returned for cleaning before re-use in the system. The cleaning process is based on a lamella thickener (T2 and T3), a flocculation system (T1 and P1), and a hydro-cyclone unit.

Figure 4.2. The prototype pump and recirculation system, designed by G. Tuomas and constructed by Idé Arctica in Övertorneå, Sweden.

Figure 4.3. Simplified flow scheme of the prototype pump and recirculation system. Design G. Tuomas.
The lamella thickener in the system is of cross-flow type, leading to a horizontal flow between inclined lamellas. Particles settle onto the lamella, slide towards the centre of the unit, and eventually reach the bottom of the tank. A horizontal feeder transports the sediment towards the end of the settling unit, where a screw pump feeds the waste out of the system. Efficiency of the sedimentation process is significantly improved by adding a flocculent to the incoming slurry flow. These substances gather individual fine and colloidal particles into clumps (flocks) that are easier to settle out. In addition, particle-fluid separation can be achieved with hydro-cyclones, by using them as an alternative to flocculation. The hydro-cyclone unit has a \( d_{50} \) cut-point below 5 µm (particles with density 2750 kg/m\(^3\) in water), is designed for a 0.3 m\(^3\)/min flow, and consists of 60 ø10 mm hydro-cyclones.

**TABLE 4.1**

*System specification.*

<table>
<thead>
<tr>
<th><strong>Drilling fluid:</strong></th>
<th>Water</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>High-pressure pump (P4):</strong></td>
<td>3-cylinder plunger pump Flow (2” plungers): max 286 litres/minute at max 260 bar</td>
</tr>
<tr>
<td><strong>Diesel engine:</strong></td>
<td>12-liter V6 with turbo and intercooler</td>
</tr>
<tr>
<td><strong>Mixing tank (T2):</strong></td>
<td>Volume ~0.5 m(^3).</td>
</tr>
<tr>
<td><strong>Lamella thickener (T3):</strong></td>
<td>Cross-flow gravity settler, waste discharge with screw conveyor and pump. Volume ~4.5 m(^3).</td>
</tr>
<tr>
<td><strong>Drilling fluid tank (T4):</strong></td>
<td>Volume ~1.9 m(^3).</td>
</tr>
<tr>
<td><strong>Flocculation system:</strong></td>
<td>Tank size (T1): ~0.3 m(^3). Pump (P1) flow rate: max 0.78 l/min (proportional to incoming slurry flow).</td>
</tr>
<tr>
<td><strong>Hydro-cyclones:</strong></td>
<td>Feed flow rate: 0.3 m(^3)/min. ( d_{50} )~5 µm.</td>
</tr>
<tr>
<td><strong>Installation:</strong></td>
<td>Container</td>
</tr>
<tr>
<td><strong>Size (L×W×H):</strong></td>
<td>7820×2438×2591 mm</td>
</tr>
<tr>
<td><strong>Net weight:</strong></td>
<td>~10 tonnes</td>
</tr>
</tbody>
</table>
4.2.1 System analysis

In Paper VI, a numerical model to simulate particle flows in a drilling process with the prototype system is presented (Fig. 4.4). Mathematical expressions for significant components were derived and the model was implemented within the Matlab Simulink™ math package for numerical simulations. In the model, the hammer tool block (Fig. 4.4) was defined to add spherical particles with a density of 2,750 kg/m³ to the initially clean system. The size distribution of the generated particles was assumed constant and resulted from laboratory analysis of a drill water sample taken during typical rock drilling on ~100 meters depth with a ø100 mm hammer tool. The mixing tank (T2) and drilling fluid tank (T4) in the model are described by differential equations that relate the tank’s total particle mass (concentration) with time. An in- and outflow of particle contaminated water was considered and both tanks were assumed to be ideally mixed. The lamella thickener (T3) in the prototype system is designed for a horizontal slurry flow between inclined lamellas. A separation efficiency curve for the unit was derived by analysing particle trajectories between the horizontal lamellas. Particles settling against the lamellas were assumed to end up in the underflow (reject), while the rest passed through the unit in the overflow (accept). The derived efficiency curve was assumed constant during the analyses and the underflow was defined to hold ~50% w/w solids. Since the flow in the ideal lamella thickener is laminar without mixing, the thickeners volume was assumed to cause a time delay before incoming particles were reported in the overflow or underflow. Besides the lamella thickener, the prototype system also consists of a hydro-cyclone unit for fluid cleaning. To evaluate the cyclones separation efficiency curve, an empirical model given by Plitt (WILLS 1997) was used, where the efficiency was related to, for instance, the hydro-cyclones design, the flow, and the particle density.

![Figure 4.4. Model for simulation of particle flows in the prototype system.](image-url)
Results from the simulations represent water powered DTH hammer drilling of a Ø115 mm borehole with a 200 m final depth. The penetration rate was 0.6 m/min and drilling was assumed to be discontinuous (5 min drilling / 1 min pause), due to drill pipe insertions. This approach was necessary, since the hydro-cyclone unit performs continuous work. Three different system setups were simulated and presented. Simulation A treats fluid cleaning with only the lamella thickener. In simulation B, fluid cleaning was accomplished by both the lamella thickener and the hydro-cyclones, with the hydro-cyclone underflow being disposed of. Simulation C is similar simulation B, except that the hydro-cyclone underflow is re-used in the system.

Important results from the simulations are the total particle volume sent to the hammer tool and the external water consumption, due to the hammer life being intimately related to the presence of abrasive particles in the flow and the required low consumption of water (and thereby waste flow) for efficient use of the system. Results presented in Table 4.2 indicate the mean concentration of solids in the feed to the hammer tool to be approximately 0.44% w/w when the lamella thickener is used for particle-fluid separation (simulation A). Particle flow is further reduced by ~80 % when hydro-cyclones are used as a complement (simulation B) and the underflow is disposed of. When underflow from the hydro-cyclone unit is re-used (simulation C), reduced particle flow is about 50%. The simulations are valid for the case of no flocculent in the flow. Figure 4.5 describes the volume concentration of solids in the flow to the hammer tool, during drilling of a 200 meter deep borehole. The particle size distribution curves (at t=24,000 sec. in Fig. 4.5) are presented in Figure 4.6.

<table>
<thead>
<tr>
<th>TABLE 4.2</th>
<th>Results from numerical simulations.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hammer fluid consumption:</td>
<td>Simulation A</td>
</tr>
<tr>
<td>Total particle volume to hammer tool:</td>
<td>70 m³</td>
</tr>
<tr>
<td>Mean volume concentration / mass concentration solids in the hammer feed water:</td>
<td>0.16% / 0.44%</td>
</tr>
<tr>
<td>Total underflow volume from lamella thickener:</td>
<td>7.7 m³</td>
</tr>
<tr>
<td>Total particle volume from lamella thickener underflow:</td>
<td>2.1 m³</td>
</tr>
<tr>
<td>Water consumption:</td>
<td>7.7 m³</td>
</tr>
</tbody>
</table>
4. PUMP AND RECIRCULATION SYSTEMS

Figure 4.5. Results from numerical simulations describing the volume concentrations of solids in the flow to the hammer tool. Curves A, B, and C result respectively from simulation runs A, B, and C (see Table 4.2).

Figure 4.6. Simulation results describing the particle size distribution curves at $t=24,000$ seconds (Fig. 4.5) in the flow to the hammer tool. The area under the curves represents the volume concentration of solids in the flow (see Table 4.2).
4. PUMP AND RECIRCULATION SYSTEMS

4.2.2 Field measurements

The mining company, LKAB in Malmberget, Sweden, has drilled several boreholes for safety investigations and installation of rock surveillance systems throughout the year 2001. The purpose of this work was to measure rock thickness and monitor movements. The latter is a sign of instability and is therefore a hazard for the population and the surrounding environment. The holes were drilled using water powered DTH hammers together with the above described prototype system. Recycling was used for approximately 200 meters of drilling, and the capacity of the lamella thickener was studied. Conditions for this drill work are presented in Tables 4.3 and 4.4. Laboratory results are shown in Table 4.5 and Figure 4.7.

**TABLE 4.3**
Operating conditions during field experiments.

<table>
<thead>
<tr>
<th>Drilling orientation:</th>
<th>Vertical</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter of bore hole:</td>
<td>0.115 meter</td>
</tr>
<tr>
<td>Rock type:</td>
<td>Mainly leptite</td>
</tr>
<tr>
<td>Rock compressive strength:</td>
<td>~270 MPa</td>
</tr>
<tr>
<td>Drilling tool:</td>
<td>Wassara W100</td>
</tr>
<tr>
<td>Drill bit:</td>
<td>Wassara</td>
</tr>
<tr>
<td>Drilling tool, pressure:</td>
<td>18 MPa</td>
</tr>
<tr>
<td>Drilling tool, flow:</td>
<td>~0.23 m³/minute</td>
</tr>
<tr>
<td>Drilling fluid:</td>
<td>Water</td>
</tr>
<tr>
<td>Slurry flow rate:</td>
<td>Intermittently ~0.21 m³/minute</td>
</tr>
<tr>
<td>Flow time / pause time:</td>
<td>~5 min / ~1 min</td>
</tr>
<tr>
<td>Penetration rate:</td>
<td>~0.6 m/minute</td>
</tr>
<tr>
<td>Flocculent:</td>
<td>Non-ionic, NF104</td>
</tr>
<tr>
<td>Concentration of flocculent:</td>
<td>~ 8 ppm</td>
</tr>
<tr>
<td>Length of drill pipes:</td>
<td>3 meter</td>
</tr>
</tbody>
</table>

**TABLE 4.4**
Time and positions for samples.

<table>
<thead>
<tr>
<th>Fluid sample no.</th>
<th>Position</th>
<th>Time [minutes]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Lamella thickener overflow</td>
<td>30</td>
</tr>
<tr>
<td>2</td>
<td>Lamella thickener overflow</td>
<td>60</td>
</tr>
<tr>
<td>3</td>
<td>Lamella thickener overflow</td>
<td>90</td>
</tr>
<tr>
<td>4</td>
<td>After hammer tool, before addition of flocculent.</td>
<td>0</td>
</tr>
</tbody>
</table>
TABLE 4.5
Laboratory results from flow samples.

<table>
<thead>
<tr>
<th>Flow sample number</th>
<th>Mass fraction solids</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 – overflow</td>
<td>0.05 %</td>
</tr>
<tr>
<td>2 – overflow</td>
<td>0.10 %</td>
</tr>
<tr>
<td>3 – overflow</td>
<td>0.05 %</td>
</tr>
<tr>
<td>4 - feed</td>
<td>8.4 %</td>
</tr>
</tbody>
</table>

Figure 4.7. Laboratory results of particle distributions in fluid samples. Samples 1 is taken from the lamella thickener overflow at time 30 min, sample 2 for 60 min, and sample 3 for 90 min. Curve 4 describes the feed at t=0 and before the addition of flocculent. The flow rate was ~0.21 m³/min.
5. ENERGY FLOW DURING DRILLING

5.1 Energy balance

For a control volume that represents a borehole with a drill string (Fig. 5.1), the energy balance equation is written as

$$\int (W_1 - W_2 - W_3) dt - \Delta Q_{cv} = 0$$

where $W_1$ represents the injected energy flow, $W_2$ is energy flowing from the control volume, $W_3$ represents the energy flow in the form of heat going into the formation, $t$ represents time, and $\Delta Q_{cv}$ is the change of internal energy inside the control volume.

![Figure 5.1. Energy flow through a control volume during drilling. Here, $W_1$ represents the injected energy, $W_2$ is energy leaving the volume, and $W_3$ is heat energy transferred to the formation. $Q_{cv}$ represents the internal energy inside the control volume.](image)

The energy flow $W_1$ represents the injected energy flow during drilling with an incompressible drilling fluid, and is defined as:

$$W_1 = p_1 q_1 + M \omega + F v + q_1 \rho_1 c_v T_1$$

where $p_1$ is the inlet fluid pressure, $q_1$ is the inlet volume fluid flow rate, $M$ is the mechanical torque acting on the drill string, $\omega$ is the drill string’s angular velocity, $F$ is the feed force and $v$ is the corresponding speed, $\rho_1$ is the inlet fluid density, $c_v$ is the heat capacity, and $T_1$ is the liquid’s inlet temperature.

Parameter $W_2$ (in eq. (5.1)) represents the energy flowing from the control volume (besides $W_3$), and is here defined as

$$W_2 = p_2 q_2 + q_2 \rho_2 c_v T_2 + W_A + W_B$$

(5.3)
where $p_2$ is the outlet fluid pressure, $q_2$ is the outlet volume fluid flow rate, $\rho_2$ is the outlet fluid density, $c_v$ is the heat capacity, $T_2$ is the liquid’s outlet temperature, $W_A$ is heat energy flow through the control volumes upper boundary, and $W_B$ is the energy flow not dissipating into heat, e.g. new particles surface energy. Energy losses due to propagating elastic stress waves should also be included in parameter $W_B$, if the dissipation into heat doesn’t occur near the borehole. Parameters $W_A$ and $W_B$ thereby replaces parameter $W_C$ in eq. (3), paper VII, where only the heat conduction in the drill string was considered.

Parameter $W_3$ in equation (5.1) is defined as the heat energy flow into the formation. By integrating $W_3$ with time, the heat energy having reached the formation may be written as

$$ Q_3 = \int W_3 \cdot dt = \int \rho c_v T \cdot dV - \int \rho c_v T_0 \cdot dV $$

where $\rho$ is the rock density, $c_v$ is the rock’s heat capacity, $T$ is the rock temperature, $T_0$ is the initial undisturbed rock temperature, and $V$ is the affected volume.

### 5.2 Evaluation of ground thermal conductivity

By determining the heat flow during drilling ($W_3$ in Fig. 5.1), the ground’s thermal conductivity is possible to estimate (Paper VII). This data is significant for, e.g., the design and performance of Borehole Heat Exchangers (BHE). Today, conductivity data is usually obtained by either laboratory analysis of ground samples (rock cores) or in-situ measurement with a thermal response test system (GEHLIN 2002). During a thermal response test, heat is injected or extracted to or from a borehole, from where the resulting temperature response is used to evaluate the ground thermal conductivity and thermal resistance of the BHE. A low thermal conductivity is, for instance, indicated by a more rapid temperature change of the heat carrier fluid. For the proposed integrated method in Paper VII, the heat energy released during drilling corresponds to the energy injected in a conventional thermal response test. This new technology would have several advantages compared to the conventional method. Ground conductivity values could be continuously estimated along the borehole, i.e. values are obtained through the formation. This quality could be used, for example, during production drilling in a mine to instantly detect lithological boundaries, possibly resulting in increased ore extraction efficiency. The quality of Borehole Thermal Energy Storage (BTES) systems would increase if the systems could be dynamically designed to optimize the required number or length of the boreholes while drilling, since the energy storage capacity would be recognized and verified. The new integrated method would also supply data for all boreholes, while only a few boreholes are evaluated in conventional thermal response tests. Measurements’ being performed in thermally undisturbed formations is another advantage, while existing thermal response tests are done in boreholes previously influenced by heat from drill work.
5.3 Heat transfer analysis

Heat transfer from a water powered DTH hammer to the drill fluid (water) and rock was simulated using the CFD-analysis software Fluent (Paper VII) so as to investigate expected temperatures for the proposed new method to determine the ground thermal conductivity. Continuous drilling of a ø115-mm borehole from 0 to 160 m depth was assumed, while the inlet water (300 l/min) had the same temperature as the undisturbed ground (10ºC). An assumed heat transfer (150 kW) into the fluid was distributed over the DTH-hammer length (~1 meter) at the end of the drill string (borehole bottom), since energy dissipation into heat starts in the hammer through internal leakages, friction, etc. Water losses and changes in water properties due to an increased amount of solids were not considered. The formation was assumed to be an isotropic homogeneous material without ground water flow. The simulation model is axi-symmetric and assumes an unsteady, incompressible flow.

A result of primary interest was the outlet temperature of the drill water, since this reveals the energy flow into the formation. The temperature development of the circulating fluid is presented in Figure 5.2. Here, the energy release near the bottom of the borehole heats up the circulating fluid to a higher temperature than the surrounding formation. The expected outlet water temperature during drilling at different depths is presented in Figures 5.3 and 5.4. As the drill starts to penetrate the rock, thereby exposing the borehole wall to the fluid, heat is transferred into the formation. Figure 5.3 shows how the temperature varies during drilling at different depths and for different ground thermal conductivity values. The values $\lambda=1$, $\lambda=3$, and $\lambda=5$ can be related to sandstone, granite, and magnetite (SUNDBERG 1988 and HOFMEISTER 2001). The temperature difference between the inlet and outlet water diminishes as the ground thermal conductivity increases. Another characteristic of the proposed method is the possible detection of variations in ground thermal conductivity by a change in the outlet water’s temperature gradient, presented in Figure 5.4 for two different formations, each with two different thermal conductivity values. In formation B the thermal conductivity changes from $\lambda=3$ to $\lambda=5$ at 60 meters depth, identified by the discontinuity in the curve’s gradient. According to this paper, this means that an iron ore deposit could be revealed and mapped in an environment of granite simply by ordinary drilling and analysis. Formation A changes from $\lambda=3$ to $\lambda=1$ at 60 meters depth, which could be of interest for, e.g. the construction and design of BTES-systems. The geothermal temperature gradient was not considered in the calculations presented above. To determine its effect on the outlet water temperature, four different formations with geothermal gradients from 0ºC/m to 0.03ºC/m were analysed. Figure 5.5 shows the increase in fluid temperature with increases in borehole length and geothermal gradient.
Figure 5.2. Calculated mean water temperatures at different depths in the inlet and outlet channels. The values are taken at time $t=320$ min (drilling at 160 meters depth) and the ground thermal conductivity is $\lambda=3$ W/(m·K).

Figure 5.3. Calculated temperature difference between the outlet and inlet water during drilling in formations with thermal conductivity $\lambda=1$, $\lambda=3$, and $\lambda=5$ W/(m·K).
Figure 5.4. Calculated temperature difference between the outlet and inlet water during drilling in two different formations. The formations have piecewise constant thermal conductivity that changes at depths 60 m and 100 m.

Figure 5.5. Calculated temperature difference between the outlet and inlet water during drilling at different depths in formations with thermal conductivity $\lambda=3\ \text{W/(m·K)}$ and varying geothermal temperature gradient.
6. CONCLUSIONS

6.1 Water powered percussive rock drilling

Important results from this research work are concluded as follows.

1. The function of a ø100 mm water powered DTH hammer has been modelled and simulated. Results from the simulations are, for instance, component positions with time, impact forces, and energy partitioning. Drill string rotation was omitted and the bit indenters were assumed to penetrate the formation at the same positions. Therefore, a true rate of penetration cannot be determined. A good agreement between measured and simulated piston blow frequency (~60 Hz) was achieved when the pistons coefficient of restitution was ~0.15. The relation between the coefficient of restitution and the rock properties was not investigated.

2. Pressure measurements in a ø100 mm hammer tool were performed with assistance of an indoor test bench. During hammer operation at ~18 MPa mean pressure, results show pressure variations between ~8 to ~23 MPa, mainly caused by the hammer tool’s working principle that generates a discontinuous water flow. Measurements also show even larger peak pressures in the pilot channels (~33 MPa) and indirectly reveal some information about the components positions with time.

3. Pressure variations increase the risk of damage in general system components. A method to reduce the variations is to install a flexible element (pulsation dampener) into the pressure line. Measurements during testing of a prototype dampener together with a water powered ø100 mm hammer, show reductions in pressure variations by up to 40% at a position ~5 meters behind the hammer tool.

4. A rigid sphere-rock impact process was modelled and simulated with a non-linear explicit FEM program. Rock material data were obtained from the literature and represented by the Johnson-Holmquist constitutive model, which includes effects from damage and strain rate hardening. Results from energy partitioning analyses shows that the kinetic energy of the rigid spheres are ~46% of the initial energy 4 J, ~40% of 9 J, and 36% of 16 J, after completed impact. The total amount of energy in the model also varied during the process, and the final energy reduction is due to energy being transmitted into the rock as stress waves. For the 16 J analysis, 15.21 J are present in the model as kinetic and internal energies after the finished impact, i.e. ~5% of the initial energy is transferred into the rock.

5. Field measurements of the penetration rate during well drilling with a ø100 mm water powered DTH hammer were performed. Four different ø115-ø116 mm drill bits were used and the mean penetration rate was ~0.30 to ~0.58 m/min. Results also show a decreased penetration rate with an increased borehole depth.

6. The main disadvantages of the water powered method are the voluminous waste and the necessity to have access to preferably high quality water. This means that the water should be recycled when this drilling method is used at locations with limited water access or when waste disposal is difficult to accomplish or both. A complete mobile
prototype to be used with water powered DTH hammers has therefore been constructed. The unit includes all components required for efficient drilling, i.e. systems for both pressurising drilling fluid and particle-fluid separation to enable recycling. Results from field measurements of the ingoing lamella thickeners cleaning capacity showed that ~0.05% to ~0.10% mass fraction solids (w/w) were still present in the water after cleaning. The feed initially contained ~8.4% w/w solids and polymers were used in the cleaning process to improve the capacity.

7. A numerical model for simulation of particle flows in the drilling process was developed because of abrasive particles or aggressive chemical substances in the feed water or both significantly reducing hammer life, especially when tools made of more ordinary steel qualities are used. Water cleaning was assumed to take place in the developed prototype pump- and recirculation unit, and the model was implemented within the Matlab Simulink math package. Simulation results indicate that the mean concentration of solids in the feed to the hammer tool is approximately 0.44% w/w when only the lamella thickener is used for particle-fluid separation. Particle flow is further reduced by approximately 80% when hydro-cyclones are used as a complement and the underflow is disposed. When underflow from the hydro-cyclone unit is re-used, the reduction is about 50%.

8. A new idea of how to combine ordinary drilling with determining ground thermal conductivity is presented. In the proposed method, the heat flow into the formation during drilling is suggested to be used in determining the thermal conductivity. This data is of great significance for, e.g., the design and performance of Borehole Heat Exchangers (BHE). Today, conductivity data is usually obtained by either laboratory analysis of ground samples (rock cores) or in-situ measurement with a thermal response test system. During a thermal response test, heat is injected or extracted to or from a borehole and the resulting temperature response is used to evaluate the ground thermal conductivity and thermal resistance of the BHE. For the proposed integrated method, the heat energy released during drilling corresponds to the energy injected in a conventional thermal response test. This new technology would have several advantages compared to the conventional method. Ground conductivity values could be continuously estimated along the borehole, i.e. values are obtained through the formation. For example, this quality could be used during production drilling in a mine to instantly detect lithological boundaries, possibly resulting in increased ore extraction efficiency. Borehole Thermal Energy Storage (BTES) systems could be dynamically designed to optimize the required number or length of the boreholes while drilling. This would increase the quality of the system, since the energy storage capacity would be recognized and verified. The new integrated method would also provide data for all boreholes, while only a few boreholes are evaluated in conventional thermal response tests. Another advantage is that measurements would be performed in thermally undisturbed formations, while existing thermal response tests are done in boreholes previously influenced by heat from drill work.

9. Heat transfer from an operating ø100 mm water powered DTH hammer to the drill fluid (water) and the rock, has been simulated using the CFD-analysis software Fluent. The main reason was to investigate possible drill water temperatures for the proposed new method to determine ground thermal conductivity. A result of primary interest was
the difference in the inlet and outlet temperatures of the drill water, since this reveals the energy flow into the formation. During drilling at 160 metres depth in a formation with thermal conductivity $\lambda = 1$ W/(m·K), approximately 20% of the assumed injected heat energy (150 kW) was transferred into the formation. When the conductivity was $\lambda = 5$ W/(m·K), about 34% was transferred. The temperature difference between the inlet and outlet flow (300 l/min) were in these cases 5.8°C and 4.7°C. Variations in ground thermal conductivity with the depth, was also detected by a discontinuity in the temperature gradient of the outlet water.

6.2 Further research

Some possible further research areas are to

- improve the water powered drilling methods general effectiveness
- study how the hammer tool life may be extended and how the presence of abrasive particles in the feed water affects the tool life.
- study general impact processes and the bit-rock impact process when water is present.
- develop a system for determination of the ground thermal conductivity from drilling data.
REFERENCES


REFERENCES


REFERENCES


PAPER I

Down-Hole Water Driven Hammer Drilling for BTES Applications

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Down-Hole Water Driven Hammer Drilling for BTES Applications

*Göran Tuomas 1, Bo Nordell 2

1 Water Resources Eng., Luleå University of Technology, S-971 87 Luleå, Sweden. Goran.Tuomas@sb.luth.se
2 Water Resources Eng., Luleå University of Technology, S-971 87 Luleå, Sweden, Bo.Nordell@sb.luth.se

KEY-WORDS
Drilling, DTH-hammer, Wassara, BTES, Energy Storage

Abstract
Borehole Thermal Energy Storage (BTES) systems usually require a large number of boreholes. The main part of the construction cost of a BTES system is therefore the drilling cost. More efficient drilling methods would reduce this cost and make BTES systems even more competitive. The objective of this paper was to evaluate a recently developed water driven drilling method (Wassara) as an alternative to conventional drilling systems. Wassara is a down-hole hammer-drilling tool in which water at high pressure drives the hammer. The water hammer drilling concept has several advantages. Experience from the mining industry has proven the method to be considerably more cost effective. The drilling speed is higher compared with air-driven hammers, with 2/3 less energy consumption. The ability of the down-hole water hammer to drill very deep holes in hard rock, even in water rich and fractured environments, is another advantage. The only obvious disadvantage with this technology is that the hammer-tool requires large amounts of water for operation. This is not a problem when drilling close to a lake or a river but to become a general BTES drilling method the drilling water has to be cleaned and re-circulated. This paper summarizes performed work to identify the problems, technical solutions of water handling and technical feasibility of the system.

Introduction
Considerable research is going on throughout the world to develop new efficient drilling techniques and to improve existing ones. This has resulted in several rock-drilling methods of which only a few have proven reliable and cost effective. These commonly used methods are basically top hammer drilling, rotary drilling and down-hole drilling. Each of these methods has advantages and disadvantages. Top hammer drilling can only be used for drilling relatively shallow holes, because of the energy losses when transferring the percussive pulses to larger depths. Rotary drilling is a universal method that can be used for deep drilling and is therefore commonly used in the oil and gas industry. One disadvantage with rotary drilling is the low penetration rate resulting in high production costs. The third commonly used method, down-hole drilling, is based on the air driven down-hole hammer. As the name implies, the percussive work is performed at the bottom of the hole, which is not the case in top hammer drilling. A major disadvantage with the air driven down-hole hammer is the limitations in drilling depth when drilling in water rich rock. The commonly used driving pressure at 2.4 MPa (24 bar)
corresponds to 240 m of water, which thereby is the maximum theoretical drilling depth when water is present. In water rich rock there are problems, however, already after a few meters because of occurring difficulties with hole flushing. The serious disadvantages with the air driven down-hole drilling method, has called for other solutions.

One recently developed drilling method (Wassara) eliminates the limitations of the down-hole hammer, by using water instead of air as drilling fluid. There have, however, been problems and difficulties during the development work. Corrosion, cavitation, and wear, were some of the problems to deal with. It was also necessary to use inexpensive materials to achieve low production cost of the hammers. Continuous efforts have resulted in a number of successful water driven down-hole hammers, now available on the market. Also mud driven hammers have been developed and are now commercially available. The advantage of the method is demonstrated by the fact that the mining industry has used it for drilling of more than 4-million meter blast-holes in hard rock. The method has also proven efficient for directional drilling, geo-thermal drilling, and drilling in the oil and gas industry. Another interesting possibility is to use the water-driven down-hole hammer when constructing Borehole Thermal Energy Storage (BTES) systems. The drilling cost for these systems can occupy about 30-50% of the total investment cost. The possibility to drill deeper holes can be of vital importance for the development of BTES applications.

The Water Driven Down-Hole Hammer

Function

Down-hole drilling is a method where the percussive hammer is positioned at the front of the hole during drilling, with energy supplied through the drill string in the form of pressurized fluid. The purpose of the hammer tool is thereby to convert a portion of this energy into mechanical impacts on the integrated drill bit. The actual rock fragmentation occurs at the high-pressurized contact zones between the buttons of the drill bit and the rock, as a result of the impact energy received from the piston. By rotating the drill bit and thereby creating new impact positions for the buttons, new rock will be fragmented and the penetration process continues. Fragmented rock is flushed away by the outlet water from the hammer flowing upwards to the ground surface on the outside of the drill string. This working principle for the down-hole hammer, are principally the same, regardless what type of drilling fluid that is being used. A complete water driven down-hole hammer system is similar to a system for the air-driven hammer. The main difference is that a high-pressure water pump, usually a plunger-pump, replaces the air-compressor. Another important difference is that a water cleaning system is required, if the water has to be re-circulated for re-use in the system. This is caused by the strong relationship between the hammer life and the quality of the feed water. Some factors of importance for the life are the pH-value, hardness, corrosive properties, and the particle content in the water. A large amount of hard particles in the driving water drastically reduces the life of an ordinary hammer. Wear in the moving parts causes increased internal leakage. An example of this comes from the well-documented drilling data, belonging to the 100 mm down-hole hammer in the Wassara series. A new tool of this type needs about 190 l/min to achieve 18 MPa (180 bar) operating pressure, while a worn out still working hammer, requires the double flow rate at the same operating pressure. Also pure erosion effects can be seen in hammers as a result of heavily contaminated feed water. Hammers with
higher quality and wear resistant materials are usually economically feasible when drilling with feed water containing large amounts of abrasive particles.

**Principal comparison between water and air driven down-hole hammer drilling**

Major differences occur when water is used as drilling fluid in a down-hole hammer instead of air. This is the case even though the hammer-tool itself principally works the same way, regardless of what drilling fluid is used. Some of these principal differences are listed in the following notes;

- **Input power**: Air-driven systems require significantly more energy at the same penetration rate. This is mainly caused by the high energy-losses in air-compressors.
- **Output power**: The water-driven hammer gives about twice as high output power. The main reason is the high percussion rate (usually about 60 Hz).
- **Energy transfer**: Transmitting energy by water-hydraulics can be extremely energy-efficient.
- **Penetration rate**: Though the piston output power to the drill bit is much higher in the water driven hammer, the penetration rate is only slightly higher than air-driven tools with 2.4 MPa (24 bar) working pressure. Water damping and problems with flushing the hole, seem to be the explanation. Drill bits especially designed for the water hammer are being developed.
- **Deep drilling capability**: The air-hammer has a limited drilling depth in water rich rock since the normally used air-pressure of 2.4 MPa (24 bar) corresponds to about 240 meters of water. No theoretical depth limit exists for the water-driven hammer and the tool has successfully performed work at 4300 meters depth.
- **Hammer cost**: The hammer cost is higher for the water-driven tool because of more expensive materials, more hammer parts, and smaller manufacturing series.
- **Environment**: The water hammer is much more environmental friendly. Dust is eliminated and the atmosphere is oil free and without grease residues.
- **Water**: Water is not always freely available. This motivates the use of a water cleaning system for re-circulation and re-use of the water. Waste handling is thereby also achieved because of the de-watering of the drilling waste, which makes it more easily managed.
- **Drill pipes**: The water hammer requires heavier drill pipes due to the higher operating pressure in the water driven hammer. The higher corrosive load from the water may also motivate the use of more corrosive resistant materials. Check valves are also recommended in some applications.
- **Erosion**: Erosion of drill-pipes and hammer casing is significantly reduced when low-velocity water (0.5-1 m/s) is used for flushing, instead of air with recommended velocity between 15-30 m/s.
- **Drilling accuracy**: Since water is used for flushing, erosion is reduced and the use of close fitting stabilizers is more practical. This improves the hole-straightness when compared with conventional air-driven drilling (Nordell B., Fjällström K and Öderyd L, 1998).
Water Handling System

The high water consumption of the down-hole hammer causes two major problems that have to be solved before drilling with the system; access of fresh water and disposal of the particle-polluted wastewater. This is generally not a problem when drilling close to a lake or a river but can cause problems in areas with poor water access or when the wastewater must be cleaned or contained. The basic solution is to use a water cleaning system. This system would clean the wastewater and re-circulate it to the hammer. This would lead to a significantly lower consumption of fresh water and it would also de-water the filtrate for easy handling.

An efficient and low-cost water handling system must therefore be developed to make the water driven hammer drilling competitive to other drilling methods. A principal flow diagram is shown in figure 1 above. Such cleaning system must be mobile, reliable and easy to maintain. It must also have the ability to cope with particles of different size and types, which can occur on drilling locations around the world. Many tests and experiments are therefore being performed to evaluate water-cleaning equipment and the resulting wear in the hammer. The weight concentration of particles in the wastewater depends on the actual water flow rate, penetration rate, and the density of the drilled rock. Concentration can be about 10%, when drilling in rock with a 100-mm Wassara hammer. The total mass load of particles on the cleaning equipment would be about 25 kg/min during effective drilling, which means that high-capacity cleaning equipment has to be used. The particle size distribution also varies with a number of factors. One of these is the drilling direction, where upwards directed drilling generates larger particles compared to downwards directed drilling. Typical particle sizes normally vary between 1-1000 μm with a $d_{50}$-size of about 50 μm.

A natural method to use as a first step in a water cleaning system is the use of a sedimentation tank. Some field tests from well drilling show that the mass concentration of particles can be reduced by 70% with a simple container used as a sedimentation tank. The overflow can then be used in a secondary cleaning unit, for more efficient separation of the smaller particles. The sedimentation tank can be improved with plates for increased area of sedimentation and a feeder for removal of the waste. Producers of sedimentation tanks claim that an efficient unit can reduce the amount of particles down to approximately 0.1% weight share of solids, without the use of any flocculation substances. When chemicals are used, the solid content in the processed water should be lower than 50 mg/l.
Another system of interest as a first separation stage is a vibrating screen (shaker). This is an effective filtrating system for separation of particles from fluids, and a properly designed unit can filtrate the drill water without pre-sedimentation. Disadvantages with the shaker are the continuous service and maintenance need, because of the high wear of the metal weaves that are used as filtrating element. Mesh size in the metal weave would normally be chosen from about 45 µm and upwards.

Both cleaning methods mentioned above can be refined with hydro cyclones as a secondary cleaning unit, for separation of the smaller particles. The use of hydro cyclones is well known and has been used for many years in oil and gas industry for separation of particles from liquids. Cyclones are manufactured in many sizes, but units consisting of several smaller cyclones have better particle separation capability, than units consisting of fewer but larger cyclones. Disadvantages are the relatively high reject flow-rate that causes problems with the waste handling, and the high cost of the effective units consisting of smaller cyclones.

Case studies of a cleaning system containing a simple sedimentation tank combined with high efficient hydro-cyclones, indicate that the system has capability to reduce the solid content to below 0.1% weight share solids. This system is thereby a candidate to be used together with the water driven down-hole hammer.

**Case Studies**

**Production drilling in iron ore mine (upward directed drilling)**

The Wassara 100 mm hammer has today been used for drilling about 4 million meters in mainly iron ore. This has been accomplished with several large automated drilling rigs, under a period of about five years. Mean values from year 1999, based on 200,000 drill meters, are presented in the table below;

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ore hardness:</td>
<td>110 MPa</td>
</tr>
<tr>
<td>Penetration rate:</td>
<td>0.9 m/min</td>
</tr>
<tr>
<td>Hammer life:</td>
<td>~1500 m (limited by pump capacity 280 l/min)</td>
</tr>
<tr>
<td>Piston frequency:</td>
<td>60 Hz</td>
</tr>
<tr>
<td>Operating pressure:</td>
<td>18 MPa (180 bar)</td>
</tr>
<tr>
<td>Water flow</td>
<td>190-280 l/min (190 l/min new hammer, 280 l/min max pump capacity)</td>
</tr>
<tr>
<td>Solid content in water</td>
<td>&lt; 0.02%</td>
</tr>
</tbody>
</table>

**Well-drilling (downward directed drilling)**

The Wassara W100 tool was compared with an air-driven down-hole hammer, with the following result;

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Atlas Copco COP 44</th>
<th>Wassara W100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drilling fluid:</td>
<td>Air</td>
<td>Water</td>
</tr>
<tr>
<td>Operating pressure:</td>
<td>2.0 MPa (20 bar)</td>
<td>18 MPa (180 bar)</td>
</tr>
<tr>
<td>Compressor/pump input power:</td>
<td>110 kW</td>
<td>70 kW</td>
</tr>
<tr>
<td>Power output:</td>
<td>13 kW</td>
<td>25 kW</td>
</tr>
<tr>
<td>Efficiency (before diesel-engine):</td>
<td>12%</td>
<td>36%</td>
</tr>
<tr>
<td>Penetration rate:</td>
<td>0.46 m/min</td>
<td>0.58 m/min</td>
</tr>
<tr>
<td>Input volume (new tools):</td>
<td>205 l/s air</td>
<td>190 l/min water</td>
</tr>
<tr>
<td>Piston frequency:</td>
<td>27 Hz</td>
<td>63 Hz</td>
</tr>
</tbody>
</table>

* Penetration rate for the Wassara W100-tool during downward directed drilling is significantly lower compared with upward directed drilling. The reason seems to be problems with hole flushing, and new drill-bits are therefore under development especially designed for the water driven hammer.
**Down-Hole Water Driven Hammer Drilling for BTES Applications**

BTES systems were developed during the 1980ies and at that time only borehole depths less than about 150 m were considered. It was possible to drill slightly deeper but because of the increasing drilling cost with depth resulted in more shallow BTES systems.

Figure 2 shows the optimum drilling depth as a function of cost increase with depth. In this design optimization the annual storage cost (capital, maintenance, heat loss, and operation) was minimized. Such design optimization shows that extremely high energy cost would result in a storage design that minimized the heat loss. Zero energy cost would result in the minimum construction cost design.

**Conclusion**

Down-hole drilling with water instead of air as drilling fluid, is a new competitive method for production of drill-holes. The use of water eliminates many of the disadvantages connected with air-driven down-hole hammers. One of the main differences is the capability to drill deep holes even in water rich environment. Fast penetration rate together with low energy consumption, are other benefits that has made the method popular in the mining industry. Also working environment is improved because dust is eliminated and the atmosphere is oil-free and without grease residues. The Wassara system is not yet a general BTES drilling method. With an appropriate water handling system, however, that makes it possible to recycle the drilling water, the new down-hole water hammer will fundamentally change possible designs of BTES and boreholes for extraction of cold or heat. A large BTES would previously mean e.g. 600 boreholes to a depth of 120 m. Without limitations in drilling depth and no extra cost for deep drilling a system of 120 boreholes to a depth of 600 m would mean a considerably lower construction cost. The main savings would be a result of less soil drilling and less piping, valves etc above ground.

**Acknowledgement**

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**References**


PAPER II

Dynamic Model of Water Driven Rock Drill Hammer

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**Dynamic Model of Water Driven Rock Drill Hammer**

by Göran Tuomas

Luleå University of Technology, Division of Renewable Energy, S-971 87 Luleå, Sweden.
E-mail: Goran.Tuomas@sb.luth.se

**ABSTRACT**

In this paper, the function and dynamics of water driven down-the-hole (DTH) rock drill hammers have been studied and modelled. Equations of motions are presented together with discussions and results from system simulation with a ø100 mm drill hammer. The presented model is based on one-dimensional wave propagation theory for the calculation of axial motions and mechanical impacts. Drill string dynamics are included, since a DTH drill hammer is an integrated part of the drill string. However, rotational and bending motions in the drill string are omitted. The model assumes an incompressible fluid and excludes energy dissipation mechanisms like material and fluid damping. The formation is assumed to have elastic-plastic behaviour where all absorbed energy is used for crush work. Results from the simulations are e.g. component motions, contact forces, and impact energy transfer. The piston blow frequency, component speeds and component positions were shown to agree well with practical experience.

**1. INTRODUCTION**

Water and mud-driven DTH rock drill hammers are today accepted and used as general-purpose drilling tools. Of the many advantages, the most important are cost-effectiveness and competitive performance. It also offers high penetration rates, low energy consumption, and the possibility to drill to virtually any depth [1]. Development during the last decade, with experiments and theoretical evaluations, has resulted in more efficient designs. Tools from ø50 mm and up are now commercially available and the development is ongoing. The optimisation of systems where motions are involved requires a thorough understanding of their dynamic behaviour. This is also the case for percussive hammer tools, where the energy required for rock penetration is mainly delivered through mechanical impacts. Previous studies have closely analysed this impact process. Lundberg [2] analytically derived equations that describe the efficiency for different drilling systems. For DTH-drilling, he analysed the energy transfer efficiency for when a uniform piston impacts a rigid drill bit, initially in contact with the rock, and assumed that the time between the impacts was long enough to fade out generated vibrations before the next impact occurred. A single piston blow could then be performed to analyse the drilling process. Lundberg later presented schemes, based on one-dimensional wave theory, to evaluate the efficiency for a single piston blow [3-5]. Here, the components were considered as straight bars made of linearly elastic materials with piecewise varying cross-section area, and the rock was modelled as an elastic-plastic material. Nordlund [6] also developed models for simulation of the percussive drilling process and he later analysed the effect of thrust on the performance of the drilling process [7]. Experimental studies to verify the models were presented by Lundberg and Karlsson [8] and Karlsson et. al [9].
In this work, the function and dynamics of water driven drill hammers has been studied, modelled, and simulated. The developed code is based on wave propagation theory, required to accurately model mechanical impacts and motions in the system. Since a DTH hammer tool is integrated as a part of the drill string, the drill string dynamics affect the dynamics of the hammer tool, and vice versa. Drill string dynamics are therefore included in the described model. This first approach to realize a hammer model is one-dimensional and rotational and bending motions in the drill string are omitted. These motions are complex for long strings, where phenomena like resonance, whirling, and contact with the borehole wall become more significant. Yigit and Christoforou [10-12] have done much work within this area. Another characteristic of the developed code is that successive piston impacts may be simulated, meaning that previously attained vibrations will affect consecutive impacts. This contradicts many previous studies that assume that the components are stress free when the impact occurs. Wave attenuation however depends on many parameters, one of them being the piston blow frequency. Time dependant damping should be included in the model because neither way to treat the problem is fully correct. The error generated by excluding or including previously generated waves, which is not investigated in the current study, needs further attention.

There are many reasons why a dynamic model is valuable in studying a hammer-drilling tool, the most important being the advantages gained by predicting system behaviour during different working conditions. Internal hammer tools parameters like port positions, piston design, etc., may be altered virtually, resulting in more efficient tools. Corrections and modifications can therefore be made during the design work before costly field experiments are carried out. The importance of a system model is emphasised further by using the hammer tool as a probe to retrieve geological and geomechanical information, such as hardness, fracture zones, lithological boundaries, etc. The possibility to match blast energy against the formation properties, thereby creating debris that matches the following industrial process is of great interest to the mining industry. Available active output responses from the tool are the drill string vibrations and the fluid’s pressure variations; a technique usually referred to as MWD analysis (Measurement While Drilling). Schunnesson [13-16] has carried out commendable work on MWD analysis.

This paper presents a dynamic model for drilling systems based on water driven DTH hammers. Equations of motion together with discussions and results from a simulation of a typical ø100 mm hammer tool are presented. Simulation results provide the components axial motion, duration of impact, contact forces, energy transfer efficiency, etc.
2. TOOL FUNCTION

The water driven DTH-hammer is positioned in front of the borehole during drilling. Energy is delivered through the drill string in the form of pressurised water, mechanical torque, and a mechanical feed force. The main task of the hammer tool is to convert the potential energy of pressurised water into an oscillating piston movement. Via mechanical impacts, the kinetic energy of the piston is transferred to the drill bit and finally into the rock (Fig. 1). Rock fragmentation occurs at highly pressurised contact zones between the drill bit buttons and the rock. By rotating the drill bit, thereby creating new impact positions for the buttons, new rock will be fragmented and the penetration process continues. The debris is flushed away from the outside of the drill string by outlet water from the hammer.

Ordinary water driven DTH-hammers consist of three moving parts: the drill bit, the piston, and the control valve (Fig. 1). The piston and the control valve are made to self-oscillate when a pressure difference occurs between the tool’s inlet and outlet ports. Typical frequencies range from 30 to 70 Hz, depending on tool size, water pressure, rock hardness, etc.

![Figure 1. Main parts in water powered DTH hammers.](image)

Generally, it is the lighter control valve that controls the flow to and from the heavier piston. When the control valve is in the left position (or actually $x_S < x_A$ in Fig. 1), the piston feed chamber 1 is, through the hole in the control valve, connected with the outlet channel in the centre of the hammer. A force acting to drive the piston to the left (Fig. 1) is then obtained, since chamber 2 is always connected to the high-pressure inlet. A control valve positioned to the right, as seen in Figure 1 ($x_S > x_A$), will expose piston feed chamber 1 to the high-pressure inlet and the piston will eventually accelerate and move towards the drill bit. This working principle is possible since the piston’s displacement area in chamber 1 is larger than that of chamber 2.

To make the piston oscillate and generate drill work, the control valve must move in a back and forth motion, well synchronised with the piston’s motion. This is accomplished by using three pilot-pressures to control forces on the control valve. High pressures in pilot inlets 2 and 3 (Fig. 1) generate forces on the control valve in a negative x-direction, while a high pressure in the pilot inlet 1 works in the positive direction. In a typical hammer tool, the displacement area for pilots 1 and 3 is twice the area of pilot 2. Together with a constant high pilot pressure 2, this gives the control valve positioning logic presented in Table 1, where condition C must be active to
accelerate the piston towards the drill bit. Either one of the other conditions, A, B, or D in Table 1, will redraw the piston.

Table 1. Positioning logic for the control valve. The value “1” represents a high pressure and “0” represents a low. Parameter p1 is the pressure condition in piston chamber 1, which follows the positioning condition \((x_S>x_A)\) of the control valve.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Pilot 1</th>
<th>Pilot 2</th>
<th>Pilot 3</th>
<th>(x_S&gt;x_A, p_1)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>0</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>B</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>C</td>
<td>1</td>
<td>1</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>D</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>0</td>
</tr>
</tbody>
</table>

The required synchronisation of the components is accomplished by using the piston to open or close channels that connects pilots 1 and 3 with pressure chambers 1 or 2 (at positions \(x_{B1}, x_{B2}, \) and \(x_C\) in Fig. 1). The working function is easily understood when it is initially assumed that the piston just impacted with the drill bit and that the control valve has reached its left end position (at time \(t_1\) in Fig. 2). At this time, condition A is active since the conditions \(x_{P1}>x_C, x_{P1}>x_{B1}\), and \(x_{P2}>x_{B2}\) are true, and chamber 1 has no pressure. This will cause the piston to accelerate away from the drill bit and increase its speed until time \(t_2\). Here, condition C becomes active, since conditions \(x_{P2}<x_{B2}, x_{P1}<x_{B1},\) and \(x_{P1}<x_C\) create a high pressure in pilot 1. This will rapidly shift the control valve’s position towards the right in Fig. 1, meaning that the net forces on the piston now act against the drill bit. The piston will stop at time \(t_5\), reaching its maximal left end position. This is usually some distance away from the tool body to avoid impact, leading to a smooth change of direction and reduced vibrations. After the directional change at \(t_5\), the same forces remain active, meaning that the piston retardation has changed into acceleration towards the drill bit. At time \(t_6\), before impact, pilot 3 will go high since \(x_{P1}>x_C\) accomplishes the condition D in Table 1. This will only be temporarily active and at time \(t_7\), condition A will become active again. The control valve changes its position from right to left (in Fig. 1) and a new work cycle begins.

![Fig. 2. Principal displacement and velocity of the piston and the control valve. T is the time between piston-drill bit impacts.](image-url)
3. DYNAMIC MODELLING

3.1 Equations of motion

By assuming the components to be straight bars of linearly elastic material with Young’s modulus $E$, cross-sectional area $A$, and density $\rho$, the one-dimensional wave equation gives the general solution

$$N(x,t)=N_p(x-ct)+N_n(x+ct), \quad v(x,t)=\frac{-N_p(x-ct)+N_n(x+ct)}{Z} \tag{1}$$

where $N(x,t)$ represent the force wave propagating with speed $c = (E/\rho)^{1/2}$ and $v(x,t)$ is the particle velocity. $N_p(x-ct)$ and $N_n(x+ct)$ are the components that propagate in the positive and negative $x$-direction and $Z=AE/c$ is the bar’s characteristic impedance. By finding functions (1), the axial motion for the components is given. The particle displacements $u(x,t)$ are calculated by integrating $v(x,t)$ with respect to time and with suitable initial conditions.

![Axi-symmetric sketch of the components in the hammer model](image)

Figure 3. Axi-symmetric sketch of the components in the hammer model, all with their own force wave functions $N_{pk}(x-ct)$ and $N_{nk}(x+ct)$. CS1 to CS6 are contact surface pairs and $F_t$, $F_s$, and $F_p$ are external forces.

The system model consists of four components (Fig. 3). Component 1 represents the rotation head, drill string, and the tool body, component 2 is the control valve, component 3 is the piston, and component 4 represents the drill bit. Each component holds its own force wave functions $N_{pk}(x-ct)$ and $N_{nk}(x+ct)$, where index $k$ represents the component number. The initially wave free components attain waves from the external forces $F_t$, $F_s$, and $F_p$ (Fig. 3), which are assumed to act upon the left end of the drill string, control valve, and piston. The equation for adding an external force into the left end of a bar with existing waves, is

$$N_p = -F - N_n \text{ for } x = 0 \tag{2}$$

where $F$ is the external load. Waves are assumed to be perfectly reflected when they reach the free end of a component without any interference from the surrounding fluid. This is simply described as

$$N_p = -N_n \text{ at } x = 0, \quad N_n = -N_p \text{ at } x = L \tag{3}$$

where $L$ is the bar length. Another wave behaviour to consider is the fact that waves are partly reflected and transmitted when a section with different characteristic impedances is reached. The equations, derived by setting continuous force and velocity, are
\[ \frac{N_{p1}}{N_p} = \frac{2 Z_1}{(Z + Z_1)} \]  
\[ \frac{N_n}{N_p} = \frac{(-Z + Z_1)}{(Z + Z_1)} \]  
\[ \text{where } N_{p1} \text{ is the transmitted part of the incoming wave } N_p \text{ and } N_n \text{ is the reflected part, } Z \text{ is the bar impedance before change, and } Z_i \text{ is the impedance after change. Incoming waves may also split up into three sub waves (instead of two as in eq. (4)-(5)). In this model it occurs when the surfaces CS1, CS2, CS3, and CS5 are in contact. The equations are} \]
\[ \frac{N_n}{N_p} = \frac{(-Z + Z_1 + Z_2)}{(Z + Z_1 + Z_2)} \]  
\[ \frac{N_{p1}}{N_p} = \frac{2 Z_1}{(Z + Z_1 + Z_2)} \]  
\[ \frac{N_{p2}}{N_p} = \frac{2 Z_2}{(Z + Z_1 + Z_2)} \]  
\[ \text{where } N_p \text{ is the incoming wave in a segment with impedance } Z \text{ and } N_n \text{ is the reflective part, } N_{p1} \text{ is the transmitted wave into a segment with impedance } Z_1 \text{, and } N_{p2} \text{ is the transmitted part into a segment with impedance } Z_2. \text{ Contact within a defined contact surface (Fig. 3) occurs as long as the intermediate forces are compressive (negative), since tensile waves tend to separate the components.} \]

The external forces in the model are \( F_t, F_s, \) and \( F_p. \) The \( F_t \)-force is usually referred to as the thrust force whose purpose is to assure contact between the drill bit and the formation when piston-drill bit impact occurs. The \( F_s \) and \( F_p \) forces only consist of pressure-induced forces, even though other forces such as body forces, flow induced forces, and friction forces also occur. The piston force \( F_p \) is calculated as
\[ F_p = A_1 p_1 - A_2 p_2 - A_3 p_3 \]  
where indexes 1 to 3 denote the chamber number according to Fig 1, \( A \) is the corresponding displacement area, and \( p \) is the pressure. The pressure \( p_1 \) is defined as
\[ p_1 = p_{in} \quad \text{if} \quad x_S > x_A \]  
\[ p_1 = p_{out} \quad \text{if} \quad x_S \leq x_A \]  
where \( p_{in} \) and \( p_{out} \) are the tool’s inlet and outlet pressures, and \( x_S \) and \( x_A \) are axial positions according to Fig. 1. Pressures \( p_2 \) and \( p_3 \) are simply defined as
\[ p_2 = p_{in} \]  
\[ p_3 = p_{out} \]  
The pressure-induced forces that act on the control valve are
\[ F_s = A_{s1} p_{s1} - A_{s2} p_{s2} - A_{s3} p_{s3} + F_{dl} + F_{dr} \]  
where index 1 to 3 denote the pilot number according to Fig 1, \( A \) is the corresponding displacement area, and \( p \) is the pressure. The \( p_{s1} \) pressure is defined as
\[ p_{s1} = p_1 \quad \text{if} \quad x_{P1} > x_{B1} \]  
\[ p_{s1} = p_2 \quad \text{if} \quad x_{P1} \leq x_{B1} \]
where $x_{P1}$ and $x_{B1}$ are axial positions according to Fig. 1. The second of these conditions (eq. (16)) is possible since $x_{P2}-x_{P1} = x_{B2}-x_{B1}$. Pressure $p_{s2}$ is defined as

$$p_{s2} = p_{in}$$

while pressure $p_3$ is based on the conditions

$$p_{s3} = p_1 \quad \text{if} \quad x_{P1} > x_C$$
$$p_{s3} = p_{out} \quad \text{if} \quad x_{P1} \leq x_C$$

Forces $F_{dl}(t)$ and $F_{dr}(t)$ in equation (14), are damping forces that occur when the control valve is close to its end positions. The purpose here is to reduce the control valve’s velocity and stabilize its position close to the wall. This is possible by letting the control valve confine a small volume of fluid into a chamber, with only a small annulus for fluid passage. However, a drawback is that the acceleration will be reduced when the control valve is set to move to the other end. The annulus flow may be regarded as a combined transient Couette-Poiseuille flow, changing from highly turbulent into laminar flow, and vice versa. The pressure difference is calculated with the Darcy-Weisbach friction factor, $f$, and the hydraulic radius $R$ [see e.g. 17]:

$$\Delta p = fLU^2 \rho / (8R)$$

where $\Delta p$ is the pressure difference, $L$ is the annulus length, $U$ is the flow speed, and $\rho$ is the fluid density. $R$ is defined as $R=A/P$ where $A$ is cross sectional area and $P$ is the wet perimeter. The laminar and turbulent friction factor $f$ are given by eqs. (21) and (23), assuming smooth turbulent (Blasius) flow (eq. (23)). Here, the unstable transitional region is simply related linearly to $Re=2000$ and $Re=4000$ (eq. (22)).

$$f = 64 / Re$$

for $Re < 2000$

$$f = 3.87 \cdot 10^{-6} \cdot Re + 2.43 \cdot 10^{-2}$$

for $2000 \leq Re < 4000$

$$f = 0.316 / Re^{0.25}$$

for $Re \geq 4000$

where $Re=U4R/\nu$ is the Reynolds number and $\nu$ is the kinematic viscosity.

The actual damping force acting on the control valve in the left and right end positions are then calculated as

$$F_{dl} = \Delta pA_r \text{sgn}|U|$$

$$F_{dr} = \Delta pA_l \{-\text{sgn}|U|\}$$

where $\Delta p$ is the pressure difference, $A_r$ and $A_l$ are the right and left end damping displacement areas, and $U$ is the control valve velocity.

The drill bit-formation contact surface (CS6 in Fig. 3) is modelled according to the idealized relations outlined in Figure 4. Here, the straight line with slope $k_1$ represents a perfectly elastic relation ($F=k_1 \cdot u$) between the intermediate force $F$ and the displacement $u$ into the formation. Therefore, Decompression in this segment will not permanently deform the rock. The line with slope $k_2$ represents the plastic part where the actual crushing takes place. Decompression reached after line 2 will follow a line with slope $k_l$ down towards the u-axis, with a permanent deformation of the rock material. The accomplished work on the rock is then evaluated as the area enclosed by
the line segments and the u-axis (Fig. 4). The approach is however not straightforward since these idealized relations depend also on many factors. Rock characteristics like compression strength and brittleness are obviously important, as well as drill bit design where the number of buttons, button shape, etc., will affect the curve parameters.

![Figure 4. Idealized force-displacement relation between the bit and the formation.](image)

3.2 Energy equations

The hammer tool’s efficiency is defined as: \( \eta = W_1/W_2 \), where \( W_1 \) is given by

\[
W_1 = \int F \, dx
\]  

(26)

Here, \( F \) is the drill bit-formation force according to Fig. 4 and the work corresponds to the enclosed area. The input energy \( W_2 \) is given by

\[
W_2 = \int (F_t v + pq) \, dt
\]  

(27)

where \( F_t \) is the thrust force, \( v \) is the corresponding axial drill string velocity, \( p \) is the fluid pressure, and \( q \) is the fluid flow rate. The work done by the thrust force and the drill bit-formation force can both be evaluated by:

\[
W_T = \frac{1}{Z} \int (N_p^2 - N_n^2) \, dt
\]  

(28)

where \( W_T \) is the energy passing through a cross section \( x \) during a specified time interval. Equation (28) may also be seen as the difference between the energy transported by the waves in positive and negative directions. The total amount of energy stored in each one of the components at a specific time is:

\[
W_T = \frac{1}{Zc} \int (N_p^2 + N_n^2) \, dx
\]  

(29)

where \( c \) is the wave propagation speed. Since no damping is applied this energy will remain constant if the bar has no interaction with other components or forces.
4. SIMULATION RESULTS AND DISCUSSION

Numerical simulations of equations (1)-(29) with appropriate initial conditions, has been executed for determination of the dynamic behaviour for a drilling system, based on a ø100 mm hammer tool. The used scheme for wave propagation is well known and thoroughly described by Lundberg [2-5]. Other schemes for stress wave propagation, based on the finite difference method, are well described by Lin [18]. Because of the selected scheme, a solution procedure based on a fixed-step solver is required with the time step matched against the element length. This simulation used 10 mm element lengths, giving the time step ~1.93·10⁻⁶ s with the wave propagation speed ~5,172 m/s. The position increment per time step is then ~0.02 mm for 10 m/s component speed, thereby approximately corresponding to the maximum error for contact surface and pressure port overlap. In the simulation, the dominating wavelengths of the applied forces are much larger than the lateral dimensions of the bars, meaning that one-dimensional conditions prevail and that the wave is transmitted without significant distortion of its shape due to dispersion [19]. The error made by using a 1-D instead of 3-D scheme, was not greater than a few percent in a case considered by Lundberg et. al [20].

For the actual simulation, parameters and results have been made dimensionless to simplify interpretation, according to Table 2. Used parameters are presented in Table 3.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Unit</th>
<th>Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time</td>
<td>t* = L_E/c</td>
<td>L_E = Element length, c = Wave propagation speed</td>
</tr>
<tr>
<td>Length</td>
<td>L* = L_H</td>
<td>L_H = Piston length</td>
</tr>
<tr>
<td>Area</td>
<td>A* = A_0</td>
<td>A_0 = Representative piston area</td>
</tr>
<tr>
<td>Velocity</td>
<td>v* = V_0</td>
<td>V_0 = Representative impact speed</td>
</tr>
<tr>
<td>Penetration</td>
<td>u* = 2L_HV_0/c</td>
<td></td>
</tr>
<tr>
<td>Stiffness</td>
<td>k* = A_0E/(2L_H)</td>
<td>E = Young’s modulus</td>
</tr>
<tr>
<td>Force</td>
<td>F* = A_0EV_0/c</td>
<td></td>
</tr>
<tr>
<td>Pressure (stress)</td>
<td>p* = EV_0/c</td>
<td></td>
</tr>
<tr>
<td>Energy</td>
<td>W* = ρA_02L_HV_0²</td>
<td>ρ = Material density</td>
</tr>
<tr>
<td>Kinematic viscosity</td>
<td>u* = L_Ec</td>
<td></td>
</tr>
<tr>
<td>Dynamic viscosity</td>
<td>µ* = A_0EV_0/(L_Ec²)</td>
<td></td>
</tr>
</tbody>
</table>

In Table 2, the unit for time t* is equivalent to the time it takes for an elastic wave with propagation speed c, to propagate through an element with length L_E. This time is thereby equal to the time step used during the simulation. The units for length, area and velocity, are all related to representative piston data in the analysis. The penetration unit u* is based on the fact that an incident wave during an impact, has twice the length of the piston itself (2L_H), thereby generating the displacement u*. The stiffness k*, is also related to this length and to the representative piston area A_0, thereby representing ~half the piston stiffness. The unit for force F* and pressure (stress) p*, and energy W*, are all derived from k*, u* and A*.
Table 3. Parameters used in simulations

<table>
<thead>
<tr>
<th>Unit parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Element length, L_E</td>
<td>0.01 m</td>
</tr>
<tr>
<td>Wave propagation speed, c</td>
<td>$\sqrt{\frac{E}{\rho}}$</td>
</tr>
<tr>
<td>Piston length, L_H</td>
<td>0.49 m</td>
</tr>
<tr>
<td>Representative piston area, A_0</td>
<td>$1.584 \cdot 10^{-3}$ m^2</td>
</tr>
<tr>
<td>Representative impact speed, V_0</td>
<td>10 m/s</td>
</tr>
<tr>
<td>Young’s modulus, E</td>
<td>210 GPa</td>
</tr>
<tr>
<td>Material density, $\rho$</td>
<td>7850 kg/m^3</td>
</tr>
</tbody>
</table>

Dimensionless simulation data:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston area, A_P</td>
<td>0.51-1.64 (varying)</td>
</tr>
<tr>
<td>Control valve length, L_CV</td>
<td>0.18</td>
</tr>
<tr>
<td>Control valve area, A_CV</td>
<td>0.19-0.38 (varying)</td>
</tr>
<tr>
<td>Drill bit length, L_DB</td>
<td>0.57</td>
</tr>
<tr>
<td>Drill bit area, A_DB</td>
<td>1.14-4.17 (varying)</td>
</tr>
<tr>
<td>Drill string length, L_DS</td>
<td>20.4</td>
</tr>
<tr>
<td>Drill string area, A_DS</td>
<td>1.03-123.96 (varying)</td>
</tr>
<tr>
<td>Drill string thrust force, F_T</td>
<td>0.031</td>
</tr>
<tr>
<td>Mean inlet pressure, $p_{in}$</td>
<td>0.037</td>
</tr>
<tr>
<td>Outlet pressure, $p_{out}$</td>
<td>0</td>
</tr>
<tr>
<td>Kinematic viscosity, $\nu$</td>
<td>$1.93 \cdot 10^{-8}$</td>
</tr>
<tr>
<td>Dynamic viscosity, $\mu$</td>
<td>$8.04 \cdot 10^{-8}$</td>
</tr>
<tr>
<td>Formation stiffness, k_1</td>
<td>9.72</td>
</tr>
<tr>
<td>Formation stiffness, k_2</td>
<td>0.97</td>
</tr>
<tr>
<td>Formation threshold force, F_th</td>
<td>0.62</td>
</tr>
</tbody>
</table>

Some results from the simulations are presented in Figures 5 to 13. The basic, most descriptive diagram, Fig. 5, shows the dimensionless time-displacement curves of elements in the piston, control valve, shaft, and tool body, which define contact surfaces CS1, CS2, and CS4 (from Fig. 3). In Fig. 5, it is shown how the control valve’s movements are limited between tool body elements in CS1 and CS2, and how the piston impacts the shaft in the drill bit. The synchronization of the piston and the control valve, vital for an efficient tool function, is also revealed. The tool’s frequency is determined by measuring the time between impacts (see e.g. Fig. 5). Good agreement between simulated and measured time between impacts (~8000 time units [21]) was achieved when the pistons coefficient of restitution was ~0.15 (Fig 6). The relation between the coefficient of restitution and the rock properties has not been investigated in this work. For a given hammer, the frequency mainly depends on parameters like the hammers feed pressure, drill bit condition, drill bit type and formation stiffness.

The time-displacement curve for the tool body and shaft elements, also gives information about the penetration into the formation. However, since the buttons in the simulation are assumed to penetrate the formation at the same positions, a true ROP (Rate of Penetration) cannot be determined. In reality, torque acting on the drill string rotates the drill bit, giving both a new angular and axial position. This process has not been implemented in the current model and needs further attention.
Figure 5. Calculated displacements of elements defining contact surfaces CS1, CS2, and CS4. To simplify interpretation of the diagram, the displacement of the shaft element in CS4 and the tool body element in CS2 (thick lines), are increased to correspond to the maximum initial strike length of the piston and the control valve.

Figure 6. Calculated mean velocity of piston elements.
In Figures 6 and 7, the mean velocities of all piston and control valve elements are presented. The mean velocity ($v_{\text{mean}}$ in Fig. 6 and 7) is calculated by

$$v_{\text{mean}} = \frac{\sum_{k=1}^{N} m_k v_k}{\sum_{k=1}^{N} m_k}$$

(30)

where $m_k$ is the element mass, $v_k$ is the element velocity, and $N$ is the number of elements. The coefficient of restitution, $e$, is from Fig. 6 found to be approximately 0.15 in this simulation run, but changes with, e.g., different rock characteristics. Another interesting detail found in Fig. 6, is that the piston still has a positive acceleration after the impact process. This is because the control valve at this moment has not closed the feed channel to the piston, giving a positive force on the piston. Here, moving the port position that controls the control valve redraw could optimise the situation. Caution must however be taken to avoid an overly fast control valve redrawn, since this undoubtedly will lead to cavitation with subsequent mechanical damages.

The control valve’s impact speed (Fig. 7) is strongly dependant on the damping forces, as described in section 3.1. Optimal functionality is achieved when the control valve bounces without temporarily closing or opening the pressure and flush channel to and from the piston. If the damping is too high, the control valve will be slow when starting from either end positions.

The dimensionless displacement-time relation for the piston and the drill bit, during the first simulated impact (impact 1 in Fig. 5), is shown in detail in Figure 8. Here, the displacement of all piston elements as well as a single shaft element is plotted. The first piston-drill bit dimensionless contact time is approximately 110, which can be compared against the time it takes for a wave to travel twice the piston length ($\approx 100$). The components regain contact after the first hit due to by the drill bits re-bounce from the formation and because the piston still has a positive feed force. Figure 9 shows a Boolean value representing contact (1) or separation (0) between these components during impact 1 (Fig. 5). Here, it can be seen that the first piston blow actually consists of six to seven bounces with renewed contact.
Figure 8. Calculated displacements during impact 1 (Fig. 5) for all piston elements (49 pcs.) and for a single shaft element (thick dashed line) defining contact surface CS4. The shaft element displacement is increased with the strike length of the piston, to simplify reading of the diagram.

Figure 9. Contact logic for contact surface CS4 during impact 1 (in Fig. 5). Value 1 means physical contact with compressive intermediate forces and value 0 means that the surfaces are separated.
The force acting between the piston and the drill bit at contact surface CS4, is shown in Figure 10. The maximum dimensionless force (~1.05) is attained during the first main impact, when penetration is achieved during this simulation run. This is shown in Figure 11 where the curve representing the energy delivered into the formation becomes constant at time ~2750. In principle, new, unworn hemispherical buttons require only a small force to produce a high contact pressure against the formation (in CS6), while worn or cylindrical buttons require higher forces. So, the gain from the secondary contacts between the piston and the drill bit, and thereby drill bit and formation, will vary with the shape of the buttons.

In the energy diagram, Fig. 11, the thick top line represents the total amount of energy delivered into the system, according to equation (27). This energy decreases after impact since the negative piston velocity forces high pressure water back into the pressure line. At time ~3100, it begins increasing once more because water energy is required to feed the piston back again. The dashed line represents the total energy in the piston (equation (29)) and the maximum amount of energy before the impact (mainly kinetic energy) is ~0.34. After the first main impact, the piston still possesses ~0.02 dimensionless units of energy, giving the primary piston-drill bit energy transfer efficiency ~0.94, during the described circumstances. The thin solid line in Figure 11 represents the amount of energy transferred into the formation. The overall efficiency at time ~3100 is about 0.28/0.34≈0.82.

After the first impact, the components begin to vibrate. This is shown in Fig. 12, where the velocity of a single piston element is drawn. In reality, these vibrations will diminish with time, caused by attenuation in the material and wave transfer to the surrounding media. Usually, it is assumed that no vibrations are present in the piston when the next impact occurs, if the time between two successive blows is more than 20 to 30 ms [2]. Since the time between impacts usually is shorter for small water hammers, especially for the control valves, remaining vibrations may influence the
following impact process. However, since no damping is applied in the described model, generated vibrations will fully affect the dynamics of the system, resulting in the calculated efficiency being too high since energy is conserved in the components between successive blows. Other reasons why the efficiency will be too high are that (1) no vibrations are transferred to the surrounding fluid, (2) mechanical friction is not considered, (3) no external damping is applied, (4) internal fluid leakage is not considered, and (5) all energy delivered into the rock is assumed to be consumed for rock crushing.

![Figure 11. Calculated energy partitioning for the first impact. The thick top solid curve represents the total amount of energy delivered into the system, the dashed line is energy in the piston and the thin solid line represents the energy delivered into the formation.](image)
Figure 12. Velocity of a single piston element (closest to CS4).

Figure 13. Velocity of a pipe element, positioned at x=1 meter (element number 100)
The development of this model with the belonging simulation code is the first approach in fulfilling our objective to retrieve characteristic rock information from only hammer response data during drilling (MWD analysis). This is symbolized by Fig. 13, where the velocity of a single pipe element is plotted. The element is positioned 1 m into the drill string; the idea is to compare it with real accelerometer data in this position. The signals, both theoretical and real, would naturally need to be further processed to find hidden information such as frequency composition, RMS-value, energy content, etc. By using this model to produce these values under different input conditions, it should be possible to find relationships to carry out this idea. However, future work is required before reliable results are obtained.
5. CONCLUSION

The function and dynamics of water driven DTH rock drill hammers have been analysed and modelled. Ingoing equations are presented together with discussions and results from a drilling system simulation based on a ø100 mm hammer tool. For example, result shows that the drill hammer’s maximum piston speed is ~10 m/s and that the impact energy transfer efficiency between the piston and drill bit is ~0.94. The impact was also found to actually consist of six to seven bounces with renewed contact, with a maximum contact force ~1.05 (dimensionless). The synchronization of the piston and the control valve, vital for an efficient tool function, is also revealed. Good agreement between simulated and measured time between piston-bit impacts (~8000 time units) was achieved when the pistons coefficient of restitution was ~0.15. The relation between the pistons coefficient of restitution and the rock properties has not been investigated. The fact that actual tool parameters are used in the simulation also indicates that the model agrees fairly well with real drilling data. There are many reasons why a dynamic model is valuable in studying a hammer-drilling tool, the most important being the advantages gained by predicting system behaviour during different working conditions. Internal hammer tools parameters like port positions, piston design, etc., may be altered virtually, resulting in more efficient tools. Corrections and modifications can therefore be made during the design work before costly field experiments are carried out. In a future work, energy dissipation mechanisms and motions from drill bit rotation should be included, to enable efficiency predictions and improve multi-impact analysis.

ACKNOWLEDGEMENT

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REFERENCES

PAPER III

Test Bench for Water Powered DTH Hammers

Technical report, Luleå University of Technology, 2004
Test Bench for Water Powered DTH Hammers

by Göran Tuomas

Luleå University of Technology, Division of Renewable Energy, S-971 87 Luleå, Sweden.
E-mail: Goran.Tuomas@sb.luth.se

ABSTRACT

During development of water powered down-the-hole rock drill hammers, the Swedish company Wassara used full-scaled production drill rigs for hammer tests, which have several disadvantages. The hammers internal pressure levels and component positions have been rather difficult or impossible to measure due to the difficult environment, while tests have sometimes been costly to perform. This situation trigged the development and construction of a test bench that makes indoor experiments possible. The indoor system has all the necessary equipment to drive a water powered DTH hammer and systems to measure and record various hammer data during operation, thereby allowing the possibility to further optimise and improve hammer functions as well as analyse other vital components like pressure accumulators, pressure relief valves, nozzles, etc. This paper describes the test bench and the characteristics of the installed measurement system.

1. INTRODUCTION

1.1 General

The Swedish company Wassara develops and constructs water powered down-the-hole (DTH) rock drill hammers. The hammer is a hydraulic machine with an integrated flow valve, which generally controls the piston’s movements (TUOMAS 2004). During development work, knowledge of the internal working conditions is valuable, since the information may be used for, e.g., optimising and trimming the function. To simplify measurements, an indoor test bench was development and constructed (Fig. 1.1).

The complete test bench system can be divided into five subsystems: (i) the rig with the hammer tool, (ii) the water system, (iii) the oil hydraulic system, (iv) the control system, and (v) the measuring system (Fig. 1.2). The rig with the hammer tool is the central component, including all functions required to simulate real drilling. The tool performs the work as it is rotated and pressed against a shock absorber. A single user can control the whole system, since tool lifting devices are available and the components are manoeuvred from one place. Measuring is accomplished with a PC-based system equipped with a flow meter, an accelerometer, and five pressure sensors. This system will be further developed, though the hammer function is mainly analysed today by measuring the internal pressure levels.

The most important specifics of the system are summarised in Table 1.1.
Figure 1.1. Test bench for water powered DTH hammers

Figure 1.2. Main components in the test bench system

Table 1.1. Test bench data

<table>
<thead>
<tr>
<th><strong>RIG</strong></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Size:</td>
<td>~4500x2000x2000 (height x width x depth)</td>
</tr>
<tr>
<td>Hammer length:</td>
<td>Max. 1500 mm</td>
</tr>
<tr>
<td>Hammer piston blow energy:</td>
<td>Max. ~600 J</td>
</tr>
<tr>
<td>Feed force:</td>
<td>0-20 kN</td>
</tr>
<tr>
<td>Rotational speed:</td>
<td>0-60 rpm</td>
</tr>
<tr>
<td>Shock absorber:</td>
<td>Cylinder with lid, filled with solid steel spheres</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>WATER SYSTEM</strong></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Plunger pump #1:</td>
<td>Prattisoli LH55 (driven by el. motor)</td>
</tr>
<tr>
<td></td>
<td>Max. 300 l/min at 200 bar</td>
</tr>
<tr>
<td>Plunger pump #2:</td>
<td>Hammelmann (driven by el. motor)</td>
</tr>
<tr>
<td></td>
<td>Max. 300 l/min at 200 bar</td>
</tr>
<tr>
<td>Flow regulation:</td>
<td>Pressure controlled by-pass valve.</td>
</tr>
<tr>
<td>Water volume (effective):</td>
<td>~7,2 m³ (3+3+1,2)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>MEASURING SYSTEM</strong></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Hardware:</td>
<td>PC with National Instruments Data Acquisition Board PCI 6035E</td>
</tr>
<tr>
<td>Software:</td>
<td>LABVIEW*</td>
</tr>
<tr>
<td>Sampling speed:</td>
<td>Max. 200 kHz</td>
</tr>
<tr>
<td>Number of channels (max/used):</td>
<td>16 / 7</td>
</tr>
<tr>
<td>Pressure sensors (345 bar):</td>
<td>Sensotec A-105 (4 pieces)</td>
</tr>
<tr>
<td>Pressure sensor (250 bar):</td>
<td>Danfoss MBS 33</td>
</tr>
<tr>
<td>Flow sensor (0-700 l/min):</td>
<td>Krohne M950</td>
</tr>
<tr>
<td>Accelerometer (50g / 80 g)</td>
<td>SKF CMSS2200</td>
</tr>
</tbody>
</table>
2. SYSTEM DESCRIPTION

2.1 Rig

The rig for the hammer tool (Fig. 2.1) is designed by Sandvik, USA. It can hold tools with piston blow energies below ~600 J and a max length of 1,500 mm. The rig is mounted on a steel frame doubling as a water collector, and the whole unit stands on vibration isolators towards the floor. The rotation head is mounted on a sleigh which is guided by steel bars along the rig. The axial displacement is accomplished by a ø50 mm hydraulic cylinder (Fig. 2.2), driven by an external oil pump package, and controlled from the rigs manoeuvring unit. The cylinder’s stroke is doubled by a wheel-steel wire arrangement, thereby giving only half the cylinder force as feed force. The cylinders positive direction (with the larger drive area) feeds the sleigh against the shock absorber. Suitable forces for ø100 mm hammers are between 10-20 kN, corresponding to ~10 to 20 MPa oil pressure.

Two oil-hydraulic engines drive the rotation head (Fig. 2.3). Each has 195 cm³/rev of displacement and are designed for pressures up to ~172 bars. The maximum allowed rotational speed is 288 rpm, making them consume max. ~56 litres oil per minute and engine. A gear arrangement with a ratio of 4.8:1 reduces the speed to a max. 60 rpm on the output shaft. The theoretical output torque (without considering efficiencies) for the rotation head with two engines is:

\[
M_{\text{out}} = \frac{p \cdot d}{\pi} \cdot s
\]

where \( p \) is the pressure, \( d \) is the displacement per revolution and engine, and \( s \) is the gear ratio. With \( p=172 \) bar, \( d=195 \) cm³/rev, and \( s=4.8 \), the max. total output torque is ~5,100 Nm. Both the feed cylinder and the rotation head are controlled from a common manoeuvring unit (Fig. 2.4), installed between the external oil pump package (Fig. 2.5) and the consuming components in the rig.

![Figure 2.1. Sketch of rig with hammer tool](image-url)
Figure 2.2. Feed cylinder arrangement

Figure 2.3. Rotation head with swivel

Figure 2.4. Rig manoeuvring unit

Figure 2.5. Oil-hydraulic pump package.
A problem during testing of the hammer tools is the durability of the test material (rock), only lasting for a few minutes because of the high penetration rate. Using rock also causes other difficulties, e.g. handling problems, debris handling, etc. This motivates the use of a shock absorber (Fig. 2.6) instead of a rock sample.

The absorber is constructed by a ~ φ280 mm steel cylinder, partially filled with solid steel spheres of the type bearing balls. A steel lid is mounted inside the cylinder and on top of the balls to serve as the drill bit’s impact surface. The unit is also equipped with a splash shield (Fig. 2.7) and a coolant system, since internal friction heats up the unit. An important property of the shock absorber is the force response to the drill bit during work. However, this has not yet been evaluated and will need further attention.

Figure 2.6. Sketch of shock absorber

Figure 2.7 Installation of splash shield
2.2 Test room

The complete rig is enclosed in a room (Fig. 2.8) to protect users from splashing water and noise. The room’s frame is constructed of 80-mm square steel bars that are covered from both sides by sound isolating ISOLAMIN®-elements (Fig 2.9). The whole room stands on vibration isolating elements to limit stresses and vibrations on the components and in the building foundation.

Figure 2.8. Drawing of the test room
The test room is also equipped with tool lifting devices to assist the handling required during tool installation and removal, since typical ø100 mm tools can weigh about 65 kg, warranting the avoidance of manual handling. Lifting devices are installed inside and outside of the test room (Fig. 2.10).

Figure 2.10. Lifting devices for the hammer tools
2.3 Water system

The purpose of the water system (Fig. 2.11 to Fig. 2.19) is to deliver filtered pressurised water to the hammer tool and collect used water for re-use in the system. The required components are positioned within the pump and test rooms.

Figure 2.11. The water process scheme

Figure 2.12. Layout drawing of the pump room
Figure 2.13. View of the pump room (view A-A in fig. 2.12).

Figure 2.14. Installation of valves V1 to V4 and the flowmeter head.
Figure 2.15. View of the pump room (view B-B in fig. 2.12).

Figure 2.16. Pump room with tanks T2 and T3 and pump P2.
Two pumps (P3 and P4 in Fig. 2.11) supply the hammer tool with high-pressure water. Each pump has three cylinders and delivers ~300 l/min at 200 bars pressure. They also have integrated gearboxes with ratio ~1:3 and are driven by 110 kW electric motors, running at 1,500-rpm rotational speed. The main difference between the two pumps is that the plungers work horizontally in the Prattisoli pump and vertically in the Hammelmann pump. Both have motors mounted individually on steel frames that stand on vibration isolating elements against the floor (Fig. 2.17 and Fig. 2.18).

Figure 2.17. High-pressure pump units P3 (Hammelmann) and P4 (Prattisoli). Both pumps have integrated gearboxes and are driven by 110-kW electric motors at 1,500-rpm rotational speed.

Figure 2.18. High-pressure pump installations.
The technical installation notes for both the Prattisoli- and Hammelmann pumps state that pressure on the pumps suction sides must be more than ~2 bars during operation, since cavitation has to be avoided due to material damages caused by the resulting high-pressure peaks can be extensive. A centrifugal pump (P2) is therefore installed, raising the pressure above the critical value. The pressure is also constantly watched by a pressure transmitter that trigs the system to halt if the pressure becomes too low.

The flow rate of the outlet water from the plunger pumps is always constant since the pumps have both constant displacements and speeds. By-pass valves (V1 and V3) are therefore used to regulate the flow to the hammer tool. These valves open the feed to the high-pressure line if the pressure becomes lower than the chosen value, and closes the gate if the pressure becomes too high. The by-pass water is fed back into the tank (T2). The principal function is that the valve is force-balanced between high-pressure water at one end and 0-6 bar air pressure at the other. However, the pressure area at the air end is larger, thereby creating suitable matching forces. Controlling the air pressure to the valve will also control the outgoing water pressure.

Before the high-pressure water reaches the hammer tool, it must pass through a valve (V2 or V4) and a swivel. The valves are necessary to avoid unwanted back-flows caused by the parallel pump arrangement and the swivel is required to pass the high-pressure water into the rotation heads shaft from the stationary tube. After being used in the hammer tool, the water is collected into tank T1 and pumped back into tanks T2 and T3.
2.4 Control system

A description of the test bench’s electrical system is not included in this report. Figures 2.20 and 2.21, however, shows the systems general manoeuvring panels.

Figure 2.20. Control panel nears the test room.

Figure 2.21. Manoeuvring panel and cabinet for electric components in the pump room.
3. MEASUREMENT SYSTEM

3.1 General

The measurement system is equipped with five pressure sensors, one accelerometer, and one flow meter. The idea is to: (i) Measure the internal pressure levels in the hammer tool, (ii) detect the piston blow frequency, (iii) measure the rig acceleration, and (iv) measure the tool’s water consumption. A flow-chart with the pressure sensors (PT1 to PT5) and flow meter sensor (FT1) positions is presented in Figure 3.1. The accelerometer is fastened near the shock absorber, to detect rig vibrations.

![Figure 3.1. Flow-chart with sensor positions in the test bench system. PT1 to PT5 are the pressure sensors and FT1 is the flow meter sensor.](image)

The sensor signals must go through several steps before they can be presented to the end user (Fig. 3.2). Since signals are usually of different types (for instance, current and voltage) and have different levels, they must be transformed into a suitable voltage level to be measured because most hardware data acquisition boards are designed for measuring and sampling a voltage signal, usually between 0 to 10 V or -10 to 10 V. A signal conditioning device SC-2345 from National Instruments, is used to accomplish this transformation. This device includes modules of different types, each selected to suit the connected sensor.

![Figure 3.2. Measurement system](image)
After conditioning, the signal is ready to be sampled. The test bench system uses a PC-based data acquisition board, PCI-6035E, from National Instruments for this task. The device has capacity for a 200 kHz sampling with 16-bit resolution on 16 separate channels. Seven channels are used, since the system is equipped with seven sensors. The voltage range for the card is -10 to +10 V, providing the code width (smallest detectable change):

\[
\text{code width} = \frac{20}{2^{16}} \approx 0.3 \text{ mV}
\]

The actual measuring resolution for a specific sensor can then be calculated as:

\[
\Delta s = s_{\text{max}} \cdot \frac{\text{codewidth}}{U_s}
\]

Here, \(\Delta s\) is the resolution, \(s_{\text{max}}\) is the measuring range of the sensor, and \(U_s\) is the sensors (signal conditioning modules) output voltage at \(s_{\text{max}}\).

The software LABVIEW is then used for signal scaling, transformation, analysis, and presentation.

### 3.2 Rig vibration

The purpose of the accelerometer is to detect the tool’s piston blow frequency by measuring the rig acceleration. This information is vital due to the blow frequency being proportional to the output power of the tool. Another idea is that the acceleration level correlates to the hammer tool’s piston blow energy. However, this relation has not yet been proven and has to be further investigated.

The sensor used in the system is an SKF CMSS2200 1-axis accelerometer with sensitivity 100 mV/g and a range up to 80 g (Fig. 3.3). The signal is conditioned in the module SCC-ICP01 that is installed in the SC-2345 unit. This module supplies the necessary excitement current (4 mA), gains, and reads the output voltage. The range is, however, limited to ±5 V, meaning that the maximum registered acceleration is ±50 g. The module also filters the signal with a 0.8 Hz high-pass filter and a 19 kHz low-pass filter.

To reconstruct a continuous time signal from a discrete, the sampling frequency must at least be twice that of the highest time signal frequency (Nyquist sampling theorem). An acceleration signal frequency of, e.g., 1 kHz will require at least a 2 kHz sampling speed to be correctly represented. In practice, sampling speeds five to ten times faster than the highest frequency in the time signal are normally used.

The signal samples obtained from the data acquisition board constitute the time domain representation of the signal. To obtain the frequency content, the signal has to be transformed using the algorithm Discrete Fourier Transform or DFT. If the number of samples is a power of 2, the more efficient FFT-algorithm (Fast Fourier Transform) can be used. The test bench system uses an FFT-algorithm included in the LABVIEW software package to transform the samples into the frequency domain. An important detail to consider is the resolution of the frequency signal, calculated with the equation \(\Delta f = f_s / N\). Here, \(\Delta f\) represents the frequency resolution, \(f_s\) is the sampling rate, and \(N\) is the number of samples.
Table 3.1. Rig vibration measuring data

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensor</td>
<td>SKF CMSS2200</td>
</tr>
<tr>
<td>Type</td>
<td>1-axis accelerometer</td>
</tr>
<tr>
<td>Range (Accelerometer / module)</td>
<td>80g / 50g</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>100 mV/g</td>
</tr>
<tr>
<td>Signal conditioning module</td>
<td>NI SCC-ICP01</td>
</tr>
<tr>
<td>Excitation</td>
<td>4 mA (constant)</td>
</tr>
<tr>
<td>Hardware channel</td>
<td>0</td>
</tr>
<tr>
<td>Max input</td>
<td>±5 V</td>
</tr>
<tr>
<td>Gain</td>
<td>2</td>
</tr>
<tr>
<td>Output from signal cond. module</td>
<td>0 to ±10 V</td>
</tr>
<tr>
<td>Measuring resolution</td>
<td>0.0015 g (~0.015 m/s²)</td>
</tr>
</tbody>
</table>

Figure 3.3, Sensor SKF CMSS 2200
3.3 Water consumption

The water consumption of the hammer tools is measured with an electromagnetic flow meter of type Krohne M950, together with the signal converter Krohne IFC090F/D (Fig. 3.4). The sensor is positioned in the high pressure feed to the hammer tool, since water losses occur in the high-pressure plunger pumps due to a separate flow of cool water. The flow meter generates a 4-20 mA current that represents 0-700-l/min water flow. For signal conditioning, the module SCC-CI20 is used, transforming the 4-20 mA current into a 0-5 V proportional voltage signal that is measured with the DAQ-board. The system uses a minimum 10 kHz sampling rate for the flow meter signal.

![Image of the flow meter installation](image_url)

**Table 3.2. Flow measuring data**

<table>
<thead>
<tr>
<th>Sensor</th>
<th>Krohne M950</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Flow</td>
</tr>
<tr>
<td>Range</td>
<td>0-700 litres / minute</td>
</tr>
<tr>
<td>Deviation</td>
<td>&lt;0.1% within 23-100% range</td>
</tr>
<tr>
<td>Output</td>
<td>4-20 mA</td>
</tr>
<tr>
<td>Excitation (allowed/used)</td>
<td>19-29 V AC or DC / 24 V DC</td>
</tr>
<tr>
<td>Signal conditioning module</td>
<td>NI SCC-CI20</td>
</tr>
<tr>
<td>Hardware channel</td>
<td>1</td>
</tr>
<tr>
<td>Output from signal cond. module</td>
<td>0-5 V</td>
</tr>
<tr>
<td>Measuring resolution</td>
<td>0.042 litres / minute</td>
</tr>
</tbody>
</table>
3.4 Internal tool pressures

Four high frequency pressure sensors, type Sensotec A-105 (PT1 to PT4 in Fig. 3.1), are installed in the test bench system to measure the fast, varying water pressure in different channels of the hammer tool, so that the characteristics and behaviour of the tool can be verified and documented. The A-105 sensor (Fig. 3.5) has the resonance frequency 57 kHz, and may, according to the manufacturer, be used up to 30% of this value (~17 kHz) with acceptable accuracy. No information is, however, available of how big the error levels are at this frequency.

The sensors are of the full-bridge type and measure the strain in the casing during an applied external pressure. The bridges are excited with a 5V DC; at this level it delivers the output voltage ~8.75 mV at 5,000 psi (~345 bar). The power source is an external laboratory device of type TTi EL-183. The manufacturer guarantees a ripple below 1 mV rms, line regulation <0.01% for a 10% input voltage change, and a load regulation <0.01% for a 90% load change.

The bridges low output voltage necessitates the use of an amplifier. The SCC-AI07 signal conditioning module amplifies the signal 200 times, giving the maximum output ~1.75 V at 5 V excitation and at full pressure. The resolution becomes ~0.06 bar, according to equation (2). The SCC-AI07 module also includes a 10 kHz filter that makes it impossible to read signals with higher frequency. A sampling rate higher than 20 kHz (Nyquist sampling theorem) must be used to avoid aliasing and to fully use the capabilities of the sensor and module.

Another important detail to consider is that unwanted noise can be generated in the cable to the sensor. This is especially important if the signal cable is long and noise in form of electromagnetic fields exists. All the signal cables in the test bench system are therefore shielded in an attempt to reduce unwanted signal absorption. To achieve best noise reduction, the shields are usually only grounded at one end of the cable.

<table>
<thead>
<tr>
<th>Sensor: Sensotec A-105</th>
<th>Table 3.3. Hammer pressure measuring data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type: Pressure</td>
<td></td>
</tr>
<tr>
<td>Range: 0-5000 psi (~345 bar)</td>
<td></td>
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<tr>
<td>Resonance frequency: 57 kHz</td>
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</tr>
<tr>
<td>Output: 0-8.8 mV</td>
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</tr>
<tr>
<td>Excitation (Allowed/Used): 5V / 5 V</td>
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</tr>
<tr>
<td>Signal conditioning module: NI SCC-AI07</td>
<td></td>
</tr>
<tr>
<td>Gain: 200</td>
<td></td>
</tr>
<tr>
<td>Hardware channel: 2 to 5</td>
<td></td>
</tr>
<tr>
<td>Output from signal cond. module: ~1.75 V</td>
<td></td>
</tr>
<tr>
<td>Measuring resolution: 0.06 bar</td>
<td></td>
</tr>
</tbody>
</table>
3.5 System pressure

A low-frequency pressure sensor of type Danfoss MBS 33 is installed (PT5 in fig. 3.1) to measure the pressure near the pump. The sensor (Fig. 3.6) is designed for an excitation voltage between 10-30 V DC and outputs 4-20 mA within the measuring range 0-250 bar. The output signal is conditioned in the NI-module SCC-CI20, where the current is transformed into a 0-5 V proportional voltage signal over a 249Ω resistor. Measuring resolution $\Delta p$ is ~0.015 bar, according to equation (2). The reaction time required for the sensor to indicate a pressure raise from 10-90% of full scale is about 4 ms.

Table 3.4. System pressure measuring data

<table>
<thead>
<tr>
<th>Sensor:</th>
<th>DANFOSS MBS 33</th>
</tr>
</thead>
<tbody>
<tr>
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<tr>
<td>Dynamic response:</td>
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<tr>
<td>Output:</td>
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<tr>
<td>Excitation (Allowed/Used):</td>
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<tr>
<td>Hardware channel:</td>
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<td>Output from signal cond. module:</td>
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</tr>
<tr>
<td>Measuring resolution:</td>
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</tbody>
</table>

Figure 3.6, Sensor Danfoss MBS 33
4. CONCLUSION

A system for testing water powered DTH rock drill hammers has been developed and constructed. The unit includes all components required to drive a hammer tool with the tests taking place inside a sound- and waterproof room. Hammers with max. ~1,500 mm length and 600 J piston blow energy can be tested. The hammer tool’s water consumption must also be less than 600 litres/minute while the maximum allowed pressure is 20 MPa. Simultaneous measurements may be accomplished by a PC-based measurement system, equipped with five pressure sensors, one accelerometer, and one flowmeter. The main purpose with the pressure sensors is to give information of internal pressures inside the operating water powered DTH hammers, regarded as essential knowledge for, e.g., trimming and optimising the hammer tool. The accelerometer in the system is used to determine the piston blow frequency and eventually also the piston blow energy. The system may also be used for analysing other vital components like pressure accumulators, pressure relief valves, and nozzles.

REFERENCES

PAPER IV

Rock Penetration by a Spherical Indenter

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Rock Penetration by a Spherical Indenter

by Göran Tuomas

Luleå University of Technology, Division of Renewable Energy, S-971 87 Luleå, Sweden.
E-mail: Goran.Tuomas@sb.luth.se

ABSTRACT

The dynamic behaviour of, for example, water powered down-the-hole rock drill hammers depends on the mechanical rock properties. Drilling in elastic high strength rock may increase the piston blow frequency by up to ~40%, compared with drilling in soft, brittle formations. The purpose of this work was to numerically estimate the energy partitioning and mechanical state in a general high-strength rock material during a 14.5 mm diameter rigid sphere-rock impact process. Material data for a granite rock material were obtained from the literature and the impact process parameters were defined as comparable with the indenter-rock process during percussive ~100 mm diameter rock drilling. Axi-symmetric simulations were carried out using an explicit FEM-code together with the Johnson-Holmquist constitutive model, permitting the effects of dynamics, strain-rate hardening, and damage to be included in the analysis. Results show that approximately 5% of the initial 16 J kinetic energy was transmitted into the formation by stress waves, while 36% was returned back to the indenter. Calculation results were compared with semi-empirical data and showed good agreement.

1. INTRODUCTION

The mechanical properties of rock have shown to be important parameters in the dynamic behaviour of water powered down-the-hole (DTH) rock drill hammers [1]. Compared with drilling in soft porous formations, drilling in more elastic, high strength rock implies a greater amount of energy being returned to the piston from the bit. This phenomenon has been observed at, e.g., the Swedish company LKAB’s iron ore mines, where about 1 million meters of ø115 mm blast holes are annually drilled with water powered DTH hammers. Observations reveal the piston blow frequency to possibly vary up to 40% depending on the rock properties [2], leading to the assumption that it is possible to roughly classify some rock properties with the drill hammer acting as a “sensor”. This idea is thereby closely related to the principles behind the Schmidt hammer test, for which the rebound number is used to estimate e.g. the uniaxial compressive strength (UCS). The bit-rock penetration process is however very complicated to describe and depends on many parameters, such as rock properties, drilling conditions, bit design, impact energy, and impact speed. Much work about the general indentation process, though, has been made through the years. As early as 1881, Hertz analyzed the contact between two elastic bodies, perhaps the start of modern contact analysis. In 1885, Boussinesq analysed the stress field of a point load on a linear elastic half space, today referred to as the Boussinesq field. In more recent years, researchers have been conducting extensive indentation experiments and analyses, see e.g. ref [3]-[8]. The crack growth and force-penetration curve have been studied on different kinds of rocks when subjected to various shapes and sizes of indenters. Most studies assume a static indentation with the indenters being mainly wedges or cones. In
this work, the indentation process for a spherical indenter has been axi-symmetrically modelled and calculated using an explicit FEM-code with the Johnson-Holmquist (JH-2) constitutive model, permitting the effects of dynamics, strain-rate hardening, and damage to be included in the analysis. The process parameters, e.g. the indenter size, energy levels, and impact speeds, were comparable to those occurring during drilling with a ø100 mm water powered DTH hammer. Rock parameters were obtained from the literature and the results were compared with published semi-empirical indentation data. A future objective is to combine a comprehensive indentation model with a model for hammer dynamics to develop a method for a more detailed drilling analysis.

2. SIMULATIONS

Four different axi-symmetric indentation simulations with the explicit LS-DYNA FEM-code are presented in this study. Common to all simulations is the general model setup (Fig. 1), where the indenter was modelled as a rigid ø14.5 mm sphere and the rock (granite) was represented by the JH-2 constitutive model. Fully integrated elements were also used in all analyses to avoid hourglass modes. The first analysis was quasi-static, since this was required for parameter calibration and to compare the results against experimental data. The penetration rate was constant (±1 m/s) and the strain-rate hardening effect was not included. The other three analyses were dynamic where the initial kinetic energies of the rigid spheres were 4 J, 9 J, and 16 J, corresponding to the mass 0.5 kg and the initial speeds 4 m/s, 6 m/s, and 8 m/s. The grid in the model consisted of 0.2 mm quadratic elements within the inner area (Fig. 1) and 0.4 mm elements in the outer area. To avoid wave reflections from free surfaces, the side and bottom boundaries were defined as non-reflective, thereby reducing the necessary model size and defining a semi-infinite model. This also gave the possibility to determine the amount of mechanical wave energy passing through the boundaries.

![Figure 1. Axi-symmetric model for simulation of indentation processes](image-url)
In the JH-2 constitutive model, [9]-[10], the material strength is given by

\[
\sigma = (1 - D)\sigma_i(p, \dot{\varepsilon}) + D\sigma_f(p, \dot{\varepsilon})
\]  \hspace{1cm} (1)

where \(D \in [0,1]\) is the damage parameter, \(\sigma_i\) is the yield strength of the intact material, and \(\sigma_f\) is the yield strength of the damaged material. The general form of the equivalent stress is

\[
\sigma = \sqrt{\left[\left(\sigma_x - \sigma_y\right)^2 + \left(\sigma_y - \sigma_z\right)^2 + \left(\sigma_z - \sigma_x\right)^2 + 6\left(\tau_{xy}^2 + \tau_{xz}^2 + \tau_{yz}^2\right)\right]/2}
\]  \hspace{1cm} (2)

where \(\sigma_x, \sigma_y,\) and \(\sigma_z\) are the normal stresses and \(\tau_{xy}, \tau_{xz},\) and \(\tau_{yz}\) denote the shear stresses. The intact strength and fully damaged strength are given by

\[
\sigma_i(p, \dot{\varepsilon}) = A\sigma_H \left(\frac{\sigma_x + p}{p_H}\right)^N \left(1 + C \ln \left(\frac{\dot{\varepsilon}}{\dot{\varepsilon}_0}\right)\right)
\]  \hspace{1cm} (3)

\[
\sigma_f(p, \dot{\varepsilon}) = B\sigma_H \left(\frac{p}{p_H}\right)^M \left(1 + C \ln \left(\frac{\dot{\varepsilon}}{\dot{\varepsilon}_0}\right)\right)
\]  \hspace{1cm} (4)

where \(A, N, C, B,\) and \(M\) are dimensionless parameters, \(\sigma_H\) and \(p_H\) define a normalisation point on the yield curve, \(\sigma_T\) is the maximum tensile strength, and \(\dot{\varepsilon}_0\) is the threshold strain rate for strain rate effects. Damage \(D\) is calculated by

\[
D = \sum \Delta \varepsilon^p / \varepsilon_f^p
\]  \hspace{1cm} (5)

\[
\varepsilon_f^p = D\left(\frac{\sigma_T + p}{p_H}\right)^{D_2}
\]  \hspace{1cm} (6)

where \(\Delta \varepsilon^p\) is the increase in plastic strain per computational cycle and \(D_1, D_2\) are dimensionless parameters.
2.1 Material model constants

Strength and material data of the analyzed granite are obtained from the published experimental results of Stavrogin and Tarasov [11]. Stavrogin and Tarasov summarized some results from triaxial compression tests in the form of linear relations in the ln(τ)-CST space, where τ is the shear strength and CST is the quota between the main stresses σ₂ and σ₁. The relationships for the actual granite, Fig. 1.18c in Ref. [11], are given by

\[
\ln(\tau_i) \approx 4.5 + 10C_{ST} \\
\ln(\tau_f) \approx 1.8 + 17.7C_{ST}
\]

(7)

(8)

where τᵢ is the intact shear strength and τᵢ is the fully fractured shear strength for CST ∈ [0, 0.35]. A good correlation between the experimental data and JH-2 strength curves was possible to obtain by fitting the model parameters (Fig. 2 and Table 1).

![Figure 2. Experimental strength curves (dashed) and fitted JH-2 strength curves (solid) for intact (D=0) and fractured (D=1) granite.](image)

The reference point parameters HEL and pᵢ were used to define the point where both strength curves coincide (at CST=0.35), yielding the values HEL=9,200 and pᵢ=5,200 MPa. LS-DYNA calculates the effective stress at the reference point with the equation

\[
\sigma_H = 3(HEL - p_i)/2
\]

(9)

which gives the numerical value σᵢ = 6,000 MPa.
Investigations of how the strain rate influences the yield stress were also presented by Stavrogin and Tarasov [11]. They reported that the uniaxial compressive strength of a granite may increase by about 30% when the strain rate is increased from quasi-static loading at $2 \cdot 10^{-6} \text{ s}^{-1}$ to $2 \cdot 10^{-1} \text{ s}^{-1}$, corresponding to an approximate increase of the strength curves in Fig. 2 of about 10%, since the uniaxial loading curve is inclined. The parameters $C$ and $\varepsilon_0$ in the JH-2 model can therefore be determined to $C=0.0087$ and $\varepsilon_0=2 \cdot 10^{-6}$.

Parameters $D_1$ and $D_2$ are more difficult to estimate and usually require extensive laboratory tests [10]. They greatly influence the simulation results since they are used for calculating the required amount of plastic flow to accomplish a fully damaged material. The equation for $\varepsilon_f^p$ (eq. (6)) is pressure dependent, i.e. the test data should be available in both low and high pressure regions. At low-pressures, the parameters in this work were chosen to give stress-strain curves from one-element triaxial compression simulations comparable to the corresponding triaxial experimental compression curves. According to Fig. 3, the chosen value $\varepsilon_f^p \sim 16 \cdot 10^{-3}$ together with equation (6), where $p=0 \text{ MPa}$, $\sigma_T=17 \text{ MPa}$, and $p_H=5200 \text{ MPa}$, defines a curve in the $D_1-D_2$ space. By choosing a $D_1-D_2$ combination along the curve in Fig. 3 with a larger $D_2$ value, the penetration response will be stiffer since more plastic flow is required to create damage at higher pressures. To find a representative combination, the penetration depth was compared with semi-empirical data (see section 3.1 in this paper) and the values $D_1=0.09$ and $D_2=0.3$ were chosen and found by iterative search. All JH-2 parameters used in the analysis are summarized in Table 1.

Figure 3. Equation (6) plotted in $D_1$-$D_2$ space with parameters $\varepsilon_f^p \sim 16 \cdot 10^{-3}$, $p=0$, $\sigma_T=17 \text{ MPa}$ and $p_H=5200 \text{ MPa}$. 
Table 1. Material- and Johnson-Holmquist model parameters

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<tr>
<th>Parameter</th>
<th>Notation</th>
<th>Value</th>
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<td>Poisson’s ratio</td>
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<td>Third pressure coeff.</td>
<td>K3</td>
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<tr>
<td>Intact strength coefficient</td>
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<td>Strain rate coefficient</td>
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<tr>
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<td>Damage coefficient</td>
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3. RESULTS AND DISCUSSION

3.1 Quasi-static analysis

Results from the quasi-static FEM analysis have been compared with experimental data from Kou [3]. Kou presented the following semi-empirical and semi-theoretical equation for indentation force and depth for a hemispherical indenter

\[
\frac{p}{d} \left( \frac{G}{\sigma_c d} \right) \cdot 10^6 = -1.265 + 2.239 \left( \frac{F}{\sigma_c d^2} \right)
\]  

(10)

where \( p \) is the indentation depth, \( d \) is the indenter diameter, \( G \) is the critical energy release rate, \( \sigma_c \) is the compressive strength of the rock, and \( F \) is the indentation force. The formula is based on dimensional analysis together with results from quasi-static indentation experiments on four different types of rock. Good agreement is achieved between \( p-F \) curves based on equation (10), with the \( p-F \) curve being based on the quasi-static simulation (Fig. 4). Since the critical energy release rate, \( G \), is not presented in ref [11], the value \( G=88 \text{ J/m}^2 \) was estimated by equation \( G=33.769+0.267\sigma_c \), based on empirical results by Fong and Nelson [12]. Lines for \( G=110 \text{ J/m}^2 \) and \( G=70 \text{ J/m}^2 \) are also presented for comparison. The other parameters used in Fig. 4 for equation (10) are \( d=14.5 \text{ mm} \) and \( \sigma_c=180 \text{ MPa} \).

![Figure 4](image-url)

Figure 4. Calculated quasi-static indentation depth–force curve for a \( \phi 14.5 \text{ mm} \) spherical indenter penetrating 1.5 mm into a granite. The dashed lines are the corresponding semi-empirical experimental results according to equation (10).
3.2 Dynamic analyses

Besides the quasi-static indentation analysis, three dynamic analyses with different initial sphere speeds (4 m/s, 6 m/s, and 8 m/s) were performed, corresponding to the initial kinetic sphere energies 4 J, 9 J, and 16 J. In the dynamic indentation analyses, the strain rate hardening behaviour of the rock was shown to affect the solution. Strain rates $\dot{\varepsilon} > 1,000 \text{ s}^{-1}$ occurred in all the dynamic analyses, with the largest detected value $\dot{\varepsilon} = 12,814 \text{ s}^{-1}$ occurring at time $t=0.07$ ms in the analysis with 16 J impact energy. The calculated results also indicate differences between the quasi-static and dynamic indentation depth-force curves. The slightly higher force in the dynamic analysis is a sign of a stiffer rock response. The dip in the curve near the indentation depth $\sim 0.6$ mm in Fig. 4 is absent in Fig. 6, since the damage (fracturing) process progressed more slowly due to a strain-rate hardened rock. As mentioned earlier, the fracturing (damage) process depends strongly on the selection of damage parameters $D_1$ and $D_2$. An easy method to validate calculated damage results is to compare them with crater sizes from simple drop tests. Since the analyzed rock was not available, $D_1$ and $D_2$ were partially found by comparing indentation depth-force curves. However, this indirect method should produce acceptable results, assuming that the other parameters and the constitutive model itself are representative. Damage results from the performed simulations naturally show the increase in damage with the amount of initial sphere energy (Fig. 7). The side crack causing the formation of chips is also longer in the 16 J (8 m/s) analysis, compared with the other two (Fig. 7).

![Figure 5. Calculated effective strain rate in a granite rock material, 0.07·10^{-3} seconds after impact of a 0.5 kg ø14.5 mm rigid sphere with 8 m/s initial speed. At this time the largest detected strain rate ($\dot{\varepsilon} = 12,814 \text{ s}^{-1}$) occurred.](image)
Figure 6. Calculated indentation depth-force curves for 0.5 kg rigid ø14.5 mm sphere-granite impacts. The spheres initial speed was 4 m/s, 6 m/s and 8 m/s.

Figure 7. Calculated damage in a granite rock material after impact of a ø14.5 mm rigid 0.5 kg sphere. The red zones represents full damage (D=1) while the dark blue zones corresponds to an intact material.
The largest detected pressure, \( p=3,403 \text{ MPa} \), occurred at the sphere-rock contact surfaces at time \( t=0.115 \text{ ms} \) in the 16 J analysis (Fig. 8). The influence of the assumedly rigid indenter is not investigated in this study.

**Figure 8.** Calculated pressure distribution in a granite rock material, \( 0.115\cdot10^{-3} \text{ seconds after impact of a 0.5 kg } \varnothing 14.5 \text{ mm rigid sphere with initial speed } 8 \text{ m/s. At this time the largest detected pressure } (p=3,403 \text{ MPa}) \text{ occurred}**

**Figure 9.** Calculated kinetic energy for a 0.5 kg rigid \( \varnothing 14.5 \text{ mm} \) sphere during impact with a granite material. The spheres initial speed was 4 m/s, 6 m/s and 8 m/s.
The energy partitioning during the impact processes is of significance for the behaviour of many different kinds of drilling machines. For example, with water powered DTH hammers, the piston blow frequency may increase by up to ~40% during drilling in more elastic high strength rock, compared with drilling in soft brittle formations. The impact speed of the bit (indenter) also affects the behaviour. Results from this study (Fig. 9) show that the kinetic energy of the rigid sphere are ~46% of the initial energies 4 J, ~40% of 9 J, and 36% of 16 J, after completed impact. The total amount of energy in the model is also varying during the impact process (Fig. 10). Energy being transmitted into the rock as mechanical waves explains the energy reduction in Fig. 10. For the 16 J analysis, 15.21 J are present in the model as kinetic and internal energies after the finished impact, i.e. ~5% of the initial energy is transferred into the rock.

Figure 10. Variations in the models total energy during a simulated impact between a rigid ø14.5 mm sphere and a granite rock material. The spheres mass was 0.5 kg and the initial speed was 8 m/s.
4. CONCLUSION

The energy partitioning and mechanical state during ø14.5 mm rigid sphere-granite impact processes have been numerically estimated. Material data were obtained from the literature and the impact parameters were comparable to conditions occurring during ø100 mm water powered down-the-hole rock drilling. Simulations were performed with the explicit LS-DYNA FEM-code and the rock was modelled using the Johnson-Holmquist constitutive model. Dynamic effects, strain-rate hardening effects, and damage were considered in the analysis.

Results from a quasi-static indentation simulation show good agreement between calculated and semi-empirical indentation depth-force data. Three different dynamic indentation simulations corresponding to the mass 0.5 kg were also performed with initial sphere energies 4 J, 9 J, and 16 J and initial speeds of 4 m/s, 6 m/s, and 8 m/s. Results show high strain-rates ($\dot{\varepsilon} > 10^4$ s$^{-1}$) near the indenter, indicating that strain-rate hardening effects are important. The dynamic indentation depth-force curves show a stiffer penetration resistance when compared with the quasi-static curve, as explained by a hardened rock with less damage. Presented damage results also reveal the characteristic side crack that causes the important formation of chips in the drilling processes. Damage parameters were determined from the triaxial compression test curves and by calibration against experimental indentation depth-force data. Besides the indentation and damage results, the energy partitioning is also important for e.g. rock drilling tools. Results from this study reveal that the kinetic energy of the rigid sphere after completed impacts are ~46% of the initial energies 4 J, ~40% of 9 J, and 36% of 16 J. For the 16 J impact analysis, 5% of the initial energy was transferred into the rock by stress waves.

ACKNOWLEDGEMENT

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REFERENCES

PAPER V

System for Water-Driven Downhole Hammer Drilling

Proceedings, OTC 2001, Offshore Technology Conference,
Houston, Texas, USA, April 30-May 3, 2001
Abstract

Drilling with liquid-driven downhole hammers is a new competitive method for production of boreholes. The system requires large amounts of fluid during operation, which can be difficult and/or expensive to accomplish. One method to reduce the fluid consumption is to process used drilling fluid into a quality acceptable for re-use in the system. This will also make waste less voluminous and more easily handled. Extensive studies have been performed to find cost-effective suitable cleaning methods and corresponding compact mobile equipment. This paper describes the technical solutions and characteristics of the resulting prototype unit, together with a brief discussion about important design criteria.

System design criteria

A fluid driven downhole hammer drilling system basically consists of a drill-rig with related components and a high-pressure pump-package to pressurize the drilling fluid. This is virtually the same as for systems based on the air-driven tools, with the exception that a fluid pump replaces the air-compressor. The most significant difference is that a fluid cleaning system is required if low fluid consumption and an efficient waste handling are required (Fig. 1). For the technique to be used successfully, the fluid cleaning system must be correctly designed and implemented since fluid quality directly affects component life. The following section of the paper explains the principal function and behavior of each block, highlighting important operating parameters.

Fluid-driven hammer-tool. Fluid driven downhole hammers are available from several manufacturers in many different sizes and qualities. Development in this area is rapid, with new tools and application areas emerging all the time. Hammer-tools can be classified into two groups, namely clear-water hammers and mud-hammers. Clear-water tools are normally made of high-quality hardened steel with appropriate surface treatment. This makes them a low-cost alternative, suitable for...
use when fluid properties are known. Mud-hammers are significantly more tolerant of aggressive substances and solids in the fluid. This is usually achieved by using tungsten carbide for the components in contact with the fluid. Since the hammer-tool is the central component in the drilling system, it is important to design the system so that correct operating conditions can be accomplished.

**Operation.** The purpose of a hammer-tool is to convert potential energy in the pressurized fluid into mechanical impacts on the rock via the integrated drill-bit. This results in rock fragmentation in front of the bit which, when rotated creates new impact positions and the fragmentation and penetration process continues. The outlet water from the hammer is used to flush away fragmented rock on the outside of the drill string. For the tool to function as designed, the specific working pressure must be kept constant above a certain critical level in order to give enough energy to the drill-bit to crush the rock. If the pressure is to low the bit will tend to bounce off the rock, which results in a low penetration rate. A drilling system must therefore be capable of delivering constant operating pressure to the tool, even though fluid consumption increases due to increasing internal leakage.

**Clear-water hammer-tools.** The clear-water hammer-tool is, as the name implies, designed for use with water or water-like fluids, without significant amounts of abrasive particles and/or aggressive chemical properties. The working life of this kind of tool is therefore intimately related to the quality of the water. For this reason, knowledge of how different water related parameters affect the life of a given tool or material is of vital importance when designing a cost effective system.

**Solids in feed-water.** To determine a statistically safe relationship between hammer-life and solid content in the feed water, large numbers of experiments, preferably with continuous control of the operating parameters are required. In practice this means that the amount and type of particles, operating conditions, and chemical substances in the water must be measured and/or controlled over the operating life of a large number of hammer-tools. This procedure would have to be repeated for different tools until the effect of all parameters had been determined. The time required and cost of such experiments would be prohibitive which is the main reason why no manufacturer so far has performed this kind of work.

Interesting data can, however, be obtained from the data obtained on an ad hoc basis during the practical use of these tools, especially within the mining industry where automated drill-rigs produced million’s of meters of blast holes (Ref. 1). Results from water-analysis and the corresponding tool-life for the period when the water samples were taken, show that a mean life of a 4-inch hammer-tool is about 1500 meters when the feed-water contains between 0-220 mg/l of solids. The drilling application cited above was in iron ore, with a mean penetration rate of 0.9 m/min. The main particle materials were, as could be expected, iron-ore, but other crystalline rock particles were also present. Other experiments have shown that the life of the tool can be drastically reduced if the feed-water contains large amounts of solids. For example, a life of less than 100 drill-meters has been measured when recycling systems with poor particle separation capacity were used (Ref. 2).

The nature of the particle material also affects the wear rate of a hammer-tool. Soft round particles can more easily pass through the tool, whilst hard sharp particles causes significantly more damage. Especially quartzite particles cause high wear rates and this has also been confirmed during drilling (Ref. 2). Several empirical methods are used for determining the abrasiveness of different kinds of particles. Moh’s hardness and Cerchar Abrasivity Index (CAI), are two of these, and tables describing abrasivity index for different rock types have been published (Ref. 3).

Operational experience (Ref. 2) give that no more than 0.1% weight-share of solids is acceptable for the clear-water hammer-tool if the abrasivity of the particles corresponds to that of gneiss. A general conclusion of this is that the solids content in the feed-water must be low and controlled if clear-water hammer-tools of today are to be used successfully.

**Chemical substances in feed-water.** Aggressive chemical substances in the fluid can significantly reduce the life of hammer-tools and plunger-pumps. One example (Ref. 4) of acceptable fluid quality to a low-cost high-pressure plunger pump is presented in Table 1. The levels presented cannot directly be applied to the hammer-tool itself, since it is a different kind of machine to a high-pressure water pump. The typical effective runtime for an ordinary 4-inch hammer-tool is about 30 hours between repairs, whilst the runtime of a plunger pump is much longer. The time for corrosion to take place is therefore shorter in the hammer-tool, and the acceptable levels of chemical substances in the feed water could, from this point of view, be higher than the recommendations based upon high-pressure water pumps. However, it must be remembered that the hammer-tool is a high precision hydraulic machine with small tolerances, and therefore perhaps more sensitive to corrosion than a plunger-pump. Unfortunately reliable recommendations concerning acceptable levels of chemical substances in the feed water to hammer-tools are unavailable since the area has not been fully investigated yet and will need further attention. The list in table 1 can be used as a guideline until more evaluations have been performed.

**Mud-hammers.** Ordinary water without additives is the most preferable drilling fluid because of its neutral impact on the environment. Other advantages are the low cost, availability, and the low level of complexity associated with ancillary equipment to name but a few. Additives are, however, occasionally required to resolve technical problems encountered during borehole production. Thickening polymers and bentonite are examples of additives used to modify the rheological properties of the water. Bentonite is often used to seal and stabilize a fractured borehole, or to assist transportation of rock particles. This has lead to liquid driven hammer-tools (so called mud-hammers) being developed, which can be used together with bentonite fluids or mud. Since bentonite is a solid material, the mechanical wear in the
hammer-tool is drastically increased when it is used. The solution is to use more wear resistant materials, usually tungsten carbide, in the hammer. Sliding bearings and the surfaces of closely fitted piston-cylinder parts are typical components that are made of tungsten carbide since these areas are exposed to the greatest wear.

The use of mud-hammers is a relatively new technique and more practical experience from their use is required for meaningful evaluation of the tool to be made. Intensive work in this area will almost certainly lead to mud hammers becoming commercially available within the near future. An example of the performance of a 4-inch mud-hammer is given by the results from a recently performed horizontal-drilling test (Ref. 5). The tool was used over a 50 hour period with a re-circulated fluid system. A decanter centrifuge was used to clean the returning fluid before being pumped back to the hammer. After 50 hours the hammer was stripped down and the internal parts made of tungsten carbide were found to be almost unaffected and could be reassembled with only new seals before being used further.

High-pressure system. The purpose of the high-pressure system (Fig. 1) is to pump suitable amounts of drilling fluid to the downhole hammer-tool. This system usually consists of an engine-driven plunger pump and components for pressure- and flow-regulation. The system must be matched to the required operation conditions for the hammer-tool as far as pressure and flow are concerned and must, naturally, also be capable of handling the drilling fluid used.

High-pressure pump. Constant displacement plunger pumps are normally used in the drilling system associated with liquid driven hammer-tools. This kind of pump offers high efficiency, long life and reliability. The pumps are usually selected based upon flow, pressure level, configuration and fluid quality criteria.

Flow and pressure. The flow to be delivered by the pump is basically decided by the consumption of the hammer-tools that will be used during drilling work. Beside this flow, additional capacity must be available if bypass streams through nozzles are used; a technique sometimes used when flushing of the borehole needs to be improved. The operating pressure is also decided by the performance of the hammer-tool. Losses in swivel joints, contractions, drill-pipes, and other system components must also be considered.

Configuration. Plunger pumps are available with several different cylinder arrangements; basically more cylinders give a smoother flow and smaller pressure-variations. Although 5-cylinder pumps are preferable, the most commonly used pumps in drilling systems are equipped with 3-cylinders for cost reasons. Pulsation dampers can be used to reduce pressure variation if it causes significant problems.

Fluid quality. High-pressure plunger pumps built for handling solids in the fluid usually work at low rotational speeds and are equipped with tungsten carbide components in areas where wear is expected. These pumps can cost twice as much or more than ordinary clear-water pumps. The maximum recommended amount of solids in the feed water for a low-cost high-pressure clear-water plunger pump is usually about 50-75 mg/l with a permitted particle size of up to 75-100 µm (Table 1, Ref. 4). These pumps should only be used in combination with efficient cleaning systems and/or if a suitable supply of clean water is available. If this is not the case mud- or slurry-pumps should be used.

Engine. Diesel engines are normally used to drive the high-pressure pump, but electric and hydraulic motors are also used. The main advantage of a diesel engine is that the system is independent of external power sources, which gives the possibility for drilling in remote areas. Electric motors are usually installed in systems designed for smaller flow-rates; typically corresponding to hammer-sizes up to 4 inch. The high efficiency of these motors gives a low usage cost and they are to be recommended if electric power is available. It is also possible to use a diesel-driven generator as a power source.

Flow-regulation. A classic problem to be solved is how to control flow and hence pressure. There are three principle techniques that are commonly used in drilling systems with constant displacement pumps; (1) Regulating the pump speed, (2) flow division via a pressure-relief valve, and (3) bypass flow through nozzles. Regulating the pump speed is theoretically the best method, since no energy-losses occur. The other methods can waste large amounts of energy, which is confirmed by the higher fuel consumption associated with systems based on these techniques. The system components needed for methods (2) and (3) must also be maintained and occasionally replaced, since wear rates can be high if abrasive particles are present in the fluid.

Speed control of the pump is easily accomplished, especially if a diesel engine is used to drive the pump since the engine-speed can be regulated from idle up to full-speed; a speed range which usually corresponds to the allowed speed-range of the pump. To control the speed of an electric motor, a frequency transformer can be used with relatively small energy losses, however, such equipment has a relatively high investment cost.

Drilling fluid recycling. A water driven downhole hammer consumes relatively large amounts of water. An ordinary 4-inch tool uses between 0.2-0.4 m³/min, which is a significant volume of water to find, especially at remote drilling sites. Another practical problem is that used drilling water must be disposed of. Since this water contains particles the environmental impact must not be neglected.

An obvious solution to these problems is to reuse the drilling water in a closed recycling system (Fig. 1). The concentration of solids in a re-circulated drilling fluid will however increase as drilling continues. This will tend to increase wear and the fluid’s rheological behavior will also change. Solids-control equipment must therefore be used to control the amount of solids in the fluid.

Fluid cleaning criteria. Designing an efficient system for removal of solids from drilling fluids requires a detailed
understanding of separation processes and the operating conditions encountered during drilling. The choice between expensive highly efficient systems and less expensive systems with poorer separation must be carefully considered. The presence of solids in the fluid increases wear of the hydraulic components and the overall cost-effectiveness of the drilling system must be considered when selecting cleaning equipment. Some of the most important criteria are presented below:

1. Efficiency of solids removal. An ideal cleaning system would remove all solids from the recycled drilling fluid. This is almost impossible to achieve in practice and a build up of solids is therefore difficult to avoid. Clear-water hammers require high separation efficiency whilst mud-hammers are more tolerant of rock debris in the fluid.

2. Efficiency of liquid conservation. Some cleaning processes loose large amounts of liquid with the extracted solids. This increases the usage cost because fluid consumption increases and waste handling problems become more extensive.

3. Dilution. Adding base-fluid to a fluid system is necessary if cleaning equipment looses liquid. This will further reduce the concentration of solids. Another situation that causes dilution is simply when used fluid is dumped and replaced with new base-fluid.

4. Load. If unchecked, the concentration of solids in a recycled drilling fluid will increase as the penetration process continues, leading to a higher load of solids affecting the hydraulic parts in the system. The cleaning system must be designed to hold the solids concentrations within acceptable limits. It is relatively easy to estimate the mass of the crushed rock leaving the borehole during a specific time. This must, naturally, be based on actual penetration rates, hole-size and rock density. Combining this with the flow-rate of liquid entering the hammer-tool gives the mass-concentration of solids. This is an important parameter when dimensioning a cleaning system.

Cleaning processes can also take advantage of the fact that drilling only takes place about 40-50% of the time, by using a buffer tank before the cleaning system. Sedimentation systems take a natural advantage of the low duty cycle, since a halt in the slurry-flow gives more time for settlement, which reduces solid content in the clarified water.

Besides the concentration of solids, the size of the particles is also of interest when designing cleaning systems. Based upon the analysis of several samples of drilling water, distribution curves of particle-sizes have been determined. Upwards drilling with a 4-inch tool in an iron-ore mine (Ref. 1), typically generated particle sizes below 2 mm, while downwards directed drilling produce smaller particles. Samples from 4-inch well drilled in crystalline rock contained particles below 1 mm, with \( d_{50} = 37 \ \mu \text{m} \), \( d_{90} = 125 \ \mu \text{m} \) and \( d_{95} = 280 \ \mu \text{m} \) (Ref. 2).

5. Fluid type. Another important criteria when designing fluid cleaning systems, is the simple fact that the cleaning process must be matched to the type of drilling fluid used.

Sedimentation processes are, for instance, less suitable when high-density or high-viscosity drilling fluids are used. The risk of filtrating out additives from the fluid should also be considered.

6. Waste handling. Any discharge or waste from the cleaning system must be transported away. If the cleaning system has poor liquid conservation efficiency, the waste will be in the form of a slurry whereas a high efficiency unit will result in a drier waste with no free drilling liquid, which is more easily handled.

7. Maintenance cost. Different types of cleaning processes require different amounts of maintenance and repair. The cost for these additional services must be considered.

8. Size and mobility. Since boreholes have to be produced at many different sites, the drilling and fluid systems should be compact and easily transported. For this reason, cleaning systems are often installed into containers, suitable for freight by ships, railroad or trucks.

9. Power consumption. When drilling in remote areas with mobile systems, electrical power is not always available. It is therefore advantageous if the components for fluid cleaning consume little or no electricity.

Fluid cleaning processes. There are today three cleaning processes commonly used within the drilling industry. These are decanting centrifuges, which have the capability to clean the water to a high level of purity, but have the drawback of relatively high investment and service costs. The second alternative is vibrating screens (shakers) combined with hydro-cyclones. This is a low-cost standard system, well proven within the oil- and gas-industry for mud cleaning in rotary drilling applications. The third possible cleaning process is based on ordinary sedimentation techniques, optionally refined with equipment for flocculation and/or hydro-cyclones for removal of the finest particles. This system is probably best suited for drilling with water without additives.

System for water-driven downhole hammer drilling.
A complete prototype service unit for a water-driven downhole hammer, WASP, has been developed for both experiment and production use (Fig. 2). The unit is designed for re-circulation and includes systems for both fluid cleaning and high-pressure water generation. The equipment is built into a single container for ease of transport and handling.

Requirements. An important task when planning a drilling system is to decide which demands and requirements that are realistic and meaningful. Excessively high demands will make the system expensive, while the opposite usually results in reduced usability and increased service, repair and maintenance costs. System demands must therefore be carefully balanced to achieve a cost-effective system.

The following list presents some particular demands and requirements placed on the WASP system;

1. Water used as drilling fluid.
2. Low water consumption. This entails recycling of used
drilling water.
3. Cleaning / separation based on gravity sedimentation.
4. System for flocculation.
5. Use of hydro-cyclones for optional particle separation.
7. Ability to drive 4" hammer-tools.
8. Diesel engine to drive the high-pressure pump.
9. Simple operation and maintenance.
10. All equipment installed into a single container.
11. Mobile, easily transportable. Freight possible by truck and/or train.

Cleaning system. The WASP cleaning system uses gravity sedimentation for the primary separation of particles from the drilling water (Fig. 3). This process can be improved, by getting the particles to clump together (flocculation) creating larger particles that will settle out more rapidly. In situations where flocculation is inappropriate or ineffective, hydro-cyclones can be used for additional particle separation. These different methods should be sufficient to reduce of the solids content to an acceptable level in most drilling situations.

Lamella thickener. In many ways, gravity sedimentation is a natural choice for use with hammer-tools since gravity settling will continue even if the drill work is halted and the slurry-flow stops. This is even the case when active drilling is being performed, since only 40-50% of the time is used for effective drilling. The suitability of the method has already been evaluated by simple field-tests (Ref. 2). Sedimentation in an open 5 m³ container reduced concentration of solids by 70% when drilling with the 4-inch hammer in crystalline rock. Drilling through sedimentary layers will produce much smaller particles, which are not equally well suited for the sedimentary process. Flocculation will probably be needed in these cases to achieve a satisfactory result.

Principal function. The settling unit used in the WASP system is of cross-flow type, which means that flow is horizontal between inclined lamellas (Fig. 4). The purpose of the lamellas is to increase the sedimentation area, for more efficient settling. Particles settle against the lamella and slide towards the center of the unit and eventually reach the bottom of the tank. A horizontal conveyor then transports the sediment towards the end of the settling unit, where another inclined conveyor transports the waste out of the unit.

Waste handling. As well as transporting the sediment out of the settling unit, this second conveyor also serves to dewater the waste to achieve low water consumption. This is possible because the waste-outlet is positioned so that free water flows back into the tank. The amount of water in the waste will thus be reduced, which simplifies the waste handling since the extracted sediment can simply be piled up beside the container and removed at convenient points in time. The settling unit is also equipped with a pump-outlet if the conveyors are insufficient.

Size. For gravity sedimentation units, the relationship between performance and size is well understood; with large units being more efficient. A compromise must therefore be made between the space available for installation and the required sedimentation area. Waste-removal from the separating unit may also consume space and must be considered during design. The physical size of the selected unit is about 1 x 2 x 5 meters (width x height x length), when the inclined conveyor is folded into it’s transportation position. The separating unit is therefore positioned along the side of the container for efficient space use. Roof openings are also necessary, both to allow for the foldable conveyor and for maintenance purpose.

Efficiency. Experiments with the separating unit (Ref. 6) have indicated that about 77% of 100 µm particles with specific gravity 2.6, will be removed at a flow-rate of 53 l/s. Efficiencies at lower flow rates have not been determined and will need further attention.

Flocculation. The efficiency of sedimentation processes can be significantly improved by adding flocculants to the incoming slurry flow. These substances gather individual particles into larger clumps (flocs) that settle out more easily in the sedimentation process. This is also one of few practical methods available for separation of fine and colloidal particles from fluids. The use of flocculation is therefore necessary to avoid build up of fine particles in recycled water, which also contribute to wear in the hydraulic system.

Flocculation principle. The working principle for flocculants is based on the fact that all particles have an electrically charged surface. Particles in aqueous suspensions of pH 4 and above generally have a negative surface, while positively charged surfaces mainly occur in strong acid solutions (ref. 7). Electrically charged flocculants added to the liquid use these charges to attract and collect particles into flocs. Today synthetic flocculants, usually polycrylamides, are widely used within the process industry. These can be anionic, non-ionic and cationic in character, and have different molecular weights and charge densities. For negatively charged particles typically found in mining slurry, cationic polyelectrolytes are used, although anionic types might also be feasible.

Cost. The flocculant cost in the WASP is almost negligible since consumption is small for the flow rates in question. A polymer concentration in the slurry of about 1 ppm is usually recommended, which means that only about 70 ml concentrated polymer is consumed during 4-hours effective drilling time with an ordinary 4-inch hammer-tool. This corresponds to about 1$/ day if the drill time efficiency is 50%.

Flocculation system design. A flowchart showing the most important components of the flocculant system is shown in figure 3. The process starts with a 200-liter stainless steel tank (T1) containing a suitable mixture of polymer and water. The tank is equipped with a low-level warning indicator and a blender which runs for one hour after a new mixture has been made. Blending expands the long polymer chains which is necessary to achieve optimal floc creation. This mixture is then pumped via a dosing-pump (P1) through a mixer into the slurry. Floc-growth will then start and continue in the first
compartment of the lamella settling unit which serves as a floc reactor. It is important to achieve turbulent mixing in the slurry for optimal floc creation. The flocculant concentration in the slurry should also be kept constant for optimal floc-growth which means that the flow rate of the incoming slurry must be monitored; WASP uses an inductive flow meter (FT1) for this purpose which is then used by the control system to regulate the dosing-pump to achieve the required concentration. To achieve, say, 1 ppm concentration, a 0.1% polymer-water concentration in T1 followed by a dosing-pump flow rate of 0.1% of the slurry flow rate is required. The actual dose rate can be adjusted to suit actual conditions.

**Laboratory experiments.** Experiments have confirmed and optimized the usage of flocculants on typical drilling water (Ref. 8). The samples tested contained 1400 mg/l solids and different flocculants, mixing-times and settling times were evaluated. Successful results were achieved with a synthetic flocculant (polyacrylamid) and the clarified water contained 14 mg/l suspended solids after 5 minutes of mixing and settling. The recommended concentration for this flocculant was 1 ppm, with mixing and settling times of 2-3 minutes which gave 22 mg/l suspended solids during the experiments.

**Hydro-cyclones.** The clear-water tank (T4) in figure 3 is equipped with hydro-cyclones to provide optional particle separation. This system has been installed primarily for experimental purpose and will enable evaluation of the hammer function with different degrees of particle separation. The idea is also to use the hydro-cyclones as an alternative to flocculation. The hydro-cyclone unit used in WASP is one of the most efficient available with a d_{50} cut-point below 5 µm (particles with s.g. 2.7 in water). The unit is designed for about 0.3 m³/min flow and consists of 60 hydro-cyclones with a diameter of 10 mm. This can be compared with the 75-100 mm cyclones which are typically used in mud-cleaning equipment in the oil- and gas-industry. The advantage of smaller cyclones is their ability to separate finer particles from the fluid, but they also suffer from higher wear and power consumption. The risk for clogging also increases which is the usually reason why these small diameter units are seldom used. The risk for clogging can, however, be reduced if the fluid is screened and the largest particles removed before entering the cyclones. For the 10 mm hydro-cyclones, screening to 0.3 mm is recommended for reliable function. In the WASP unit, this is accomplished by filter F1 (Fig. 3).

The separation capacity of the hydro-cyclone unit has been predicted based on a particle distribution curve, which is typical for drilling fluid from the water-driven downhole hammer (Ref. 9). With 10 % w/w solids with specific gravity 2.65, the hydro-cyclones should remove about 90% of the solids with a water-loss of 15%.

**Turbidity surveillance.** A turbidity meter (XT2) is used to monitor particle content in the outgoing feed-water. This unit detects infrared light reflected back from the suspended particles which gives a measure of turbidity. An increase in turbidity in tank T4, gives the operator an early warning that particle density has increased and that the hydro-cyclone system must be used or the flocculant concentration adjusted or flocculant substance changed. Failure to do so may result in wear of the hammer-tool and plunger pump.

**High-pressure system.** The components in the high-pressure system are mainly the feed pump, safety filter, high-pressure plunger pump with engine, and components for pressure and flow control. These are mounted together on a common steel frame for easier container installation and maintenance. The most important components and their function are described in the sections below.

**Feed pump.** The purpose of the feed-pump (P3) is to deliver pressurized water to the suction side of the high-pressure plunger pump. This is necessary in order to avoid cavitation which can cause damage to the internal parts of the high pressure pump. The lowest recommended pressure on the inlet side is 0.6 bar, a lower pressure (measured with PT7) may be the result of a full or clogged safety filter and will stop the system.

**Safety filter.** The last stage of particle separation is a bag-filter (F2) which prevents large particles from entering the high-pressure pump and down stream components such as the hammer-tool. The generally recommended filter-size is 10 µm, but this must be adjusted for the actual system.

**High-pressure plunger pump.** The high-pressure plunger pump used in WASP is dimensioned for use with a typical 4-inch hammer-tool, which requires a flow-rate between 0.2-0.4 m³/min and a working pressure of approximately 180 bar. Since the pump must cope with pressure losses in swivels, contractions etc. a higher working pressure has been specified.

**Pump configuration.** The chosen pump is a 3-cylinder pump with a plunger diameter of 2.25" and a 5" stroke. The maximum capacity of the pump is 361 l/min at 233 bar, at a maximum allowed pump speed of 370 rpm. In the selected pump, the plunger-size can also be altered in the range 2 - 3 inches, which allows other flow/pressure ratios to be achieved. For example, flow-rates of up to 643 l/min are possible at a lower pressure of 131 bar and the pump can therefore be used together with other types of hammer-tools which require different pressure-flow-ratios.

**Solids in fluid.** The ability of a pump to cope with solids in the drilling fluid is important for the overall cost efficiency of the system. Since the WASP-system is designed to test different separation techniques, the solids concentration will vary and therefore a more tolerant pump was required. The chosen pump uses tungsten carbide plungers, wear resistant valves and a special spring-loaded packing arrangement.

**Engine system.** Selection of a suitable diesel engine is best achieved through discussion with the manufacturers, who often use computer programs developed for this purpose. The most important parameter is the estimated load-time curve, but ambient temperature, cooling system capacity, etc., are also important. A rule of thumb is that a diesel engine should have 30% more power than is required by a given application.

**Size.** The diesel engine used in WASP is dimensioned for driving the 205 bhp plunger pump over its entire working
speed of 175-370 rpm. This corresponds to 892-1887 rpm for the diesel engine which is connected to the pump via a gearbox with a ratio of 5:1:1. No other power-consuming components were installed, even though it would be possible to auxiliary drive pumps etc. using the engine’s integrated hydraulic system or via a belt arrangement.

The engine selected has 12-liter cylinder volume and is equipped with a turbocharger with intercooler. The key dimensioning factor was the torque needed at low-speed and the engine is therefore slightly over-dimensioned at higher speeds. One way to overcome this problem is to limit the minimum speed and to use a bypass nozzle for flow reduction. This would however increase the fuel consumption at low speeds and the method was not implemented in the finished system.

**Clutch.** The gearbox mentioned above includes an integrated electrically controlled multi-plate lamella clutch to allow it be engaged and disengaged from the diesel engine. The reason for this is that practical drilling work is halted during insertion and removal of drill-pipes and it is therefore desirable to be able stop the flow of drilling fluid. The clutch also allows the rotating mass connected to the engine during start-up of the diesel engine to be minimized.

**Environment.** Diesel engines are a major source of pollutants to the environment and implies a health risks for the operator. The engine in WASP fulfils the demands of the European STEP I / USA EPA 1 criteria.

**Flow control.** Flow regulation is accomplished by regulating the pump-speed and by using bypass-flows. With 2.25 inch plungers, the flow can be varied between 171-361 l/min which is suitable when drilling with an ordinary 4” hammer-tool. There are, however, occasions when the flow of less than 175 l/min is required, which is accomplished by using a bypass valve (V15) and thereby reduce the flow to the hammer-tool. A second bypass-stream can also be engaged by opening valve V14.

**Control system.** Industrial processes with a large number of critical operating conditions are difficult to manage without efficient control systems. Automated control together with well-designed interfaces can therefore significantly improve process results and safety. The WASP-system uses an industrial computer (PLC-system) to control the process and equipment. Even though the process is uncomplicated, the advantages of using a PLC-system are many, especially during setup and optimization when the controlling parameters can be easily changed.

**Operator Interface.** The interface between the operator and the machine is another important factor to consider. The system described uses a mobile maneuvering unit consisting of a computer display with an integrated keyboard for control of the process. A complete flow-chart, similar to figure 3, is presented on the display and forms the basis for presentation of process data and control. This gives the operator an excellent overview of both the process function and status. All the significant parameters in the WASP-system can be changed through the PLC-system. The most commonly used controls and process data are however grouped in a single page for efficient usage.

Communication between the interface and the PLC-unit uses a radio-link. This eliminates troublesome and fault-sensitive cables, and also, via an in-built modem, gives opportunity for software-update and troubleshooting over public phone lines.

**Conclusion**  Effective drilling with liquid-driven downhole hammers requires large amounts of fluid for driving the hammer-tool and flushing the borehole. Clear-water hammer-tools require high-quality water whilst mud-hammers are considerably more tolerable against solids and aggressive chemical substances. The described technique reduces the fluid consumption by recycling used drilling fluid. This means an effective (fluid saving) cleaning system to prevent solids-buildup in the fluid, which causes wear in the hammer-tools and other hydraulic components.

The prototype system described in this paper is primarily designed for drilling when water without additives is used as drilling fluid. It consists of cleaning equipment for recycling and a high-pressure system for driving the hammer-tool. The equipment is installed on a single mobile container for easy transport and installation at drilling sites.

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**References**

1. LKAB, SE-983 81 Malmberget, Sweden. Internet: www.lkab.com
4. Fr Ramström AB, Box 23, SE-871 21 Härnösand, Sweden. Internet: www.ramstrom.se
5. Vermeer Manufacturing Company, 2411 Vermeer Road, P.O. Box 200 Pella, Iowa 50219 U.S.A. Internet: www.vermeer.com
6. Huber-Hydropress AB, Box 125, SE-437 22 Lindome, Sweden. Internet: www.hydropress.se
9. Mozley Ltd, Cardrew, Redruth, Cornwall TR15 1SS, UK. Internet: www.mozley.co.uk
Figure 1. Principal flowchart for drilling system with recycling of used drilling fluid.

**TABLE 1 – RECOMMENDED WATER QUALITY FOR LOW-COST HIGH-PRESSURE PLUNGER PUMP.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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</thead>
<tbody>
<tr>
<td>Filtration degree</td>
<td>75 micron</td>
</tr>
<tr>
<td>Max. content solids</td>
<td>150 mg/l</td>
</tr>
<tr>
<td>Total hardness</td>
<td>20° dH</td>
</tr>
<tr>
<td>Max. hardness (CaCO3)</td>
<td>100 mg/l</td>
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<tr>
<td>Max. content iron (Fe)</td>
<td>0.5 mg/l</td>
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<tr>
<td>Max. content chlorine (Cl)</td>
<td>100 ppm</td>
</tr>
<tr>
<td>Max. content sulphate (SO4)</td>
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<tr>
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<tr>
<td>Max. free chlorine</td>
<td>1 mg/l</td>
</tr>
<tr>
<td>pH</td>
<td>6.5-8.5</td>
</tr>
</tbody>
</table>

Figure 2. WASP service unit for the water-driven downhole hammer.
Figure 3. Schematic flow-chart describing the process in WASP. Used drilling water (slurry) is cleaned for re-use in the hammer-tool. The process uses flocculant substances (from tank T1) to improve settlement in sedimentation-tank T3. Particle separation can also be accomplished by hydro-cyclones, working against tank T4. A diesel engine drives the plunger pump (P4) for generation of high-pressure water to the downhole hammer-tool.

Figure 4. Cross-flow lamella thickener with conveyors for grit disposal.
Effective use of water in a system for water driven hammer drilling

Effective use of water in a system for water driven hammer drilling

Göran Tuomas*

Division of Renewable Energy, Luleå University of Technology, SE-97187 Luleå, Sweden

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Abstract

Drilling with water driven down-the-hole (DTH) hammers is a recently developed method for competitive production of boreholes. In order to prevent large amounts of water being used during operation, the drilling fluid is here directly processed into a quality acceptable for reuse. The effectiveness is evaluated in well drilling with a mobile prototype water cleaning and pressurising unit. Especially the presence of abrasive particles in the fluid can drastically reduce tool life and make the method inefficient. The vital significance of this relation has called for detailed studies and a process simulation model for determining particle concentration and size distribution has been developed. This paper describes the model and how it is applied. Simulation results of different system configurations are also presented.

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Keywords: Drilling; DTH; Hammer; Down-the-hole; Particle; Flow; Water; Simulation

1. Introduction

The technique of using water instead of air as an energy carrier to DTH-hammer tools has been known for years. However, technical difficulties associated with corrosion, cavitation and wear have made it difficult and/or costly to put these ideas into practice. This situation began to change in the early 1990s when the Swedish mining company LKAB started to use water driven DTH-hammers for production drilling of blast-holes. The use of the hammer-tool also meant continuous evaluation and improvements of the system, which today is a highly cost-effective and competitive drilling method. Today, more than 5-million meters of blast-holes have been drilled with the water driven hammer tool within the Swedish mining industry.

There are many advantages with this method; the most important are its cost-effectiveness and competitive performance. The technique offers high penetration rates and low energy consumption as well as the possibility to drill to virtually any depth (Tuomas and Nordell, 2000). The working environment is improved since dust is eliminated and the air is free from oil residues. However, one disadvantage is that a large flow rate of preferably high quality water is required to drive the hammer tool. For instance, an ordinary 4-inch hammer-tool requires between 0.2 and 0.4 m$^3$/min to achieve a competitive rate of penetration. This means that the water should be recycled when this drilling method is used in locations with limited water access and/or when waste disposal is difficult to accomplish (Fig. 1).

The concentration of particles in the drilling water depends mainly on the actual water flow rate, penetration rate, and the density of the drilled rock. Mass concentrations ($w$) between 4 and 12% are common for rock drilling with an ordinary 4-inch hammer. This corresponds to approximately 13–27 kg/min particle flow, which means that high-capacity cleaning equipment has to be used. The particle size distribution varies with a certain number of factors. Rock characteristics, drill bit design and impact energy, are some of them. An important limiting factor during vertical or inclined drilling is the speed of the flushing water, since this must be larger than the particles settling speed. Otherwise the particles will settle in the borehole and will be re-crushed by the drill bit until the size is small enough to follow the flow. Particles generated during typical 4-inch well drilling are usually smaller than 1 mm with mass median sizes ($d_{50}$) at approximately 0.1 mm.
For the technique to be successful, the fluid cleaning system must be correctly designed and implemented since fluid quality directly affects component life. Abrasive particles and/or aggressive chemical substances in the feed water significantly reduce tool life, especially when ordinary tools made of hardened steel are used. It is, however, possible to use tungsten carbide as tool material, but this increases the cost and this material is, therefore, normally only used in mud driven tools. For this reason, knowledge of how different water related parameters affect the life of a given tool or material is of vital importance when designing cost effective systems.

Interesting data have been obtained from practical use of these tools, especially within the mining industry where automated drill-rigs produced millions of meters of 4-inch blast holes. Results from water-analysis and data of the corresponding tool-life, show that time between repairs corresponds to approximately 1500 drilling meters in hard rock when the feed water contains maximum 0.02% w/w solids. The mean penetration rate during these drillings was 0.9 m/min, which gives a total of approximately 6 million piston blows between repairs, since the piston blow frequency is about 60 Hz. Other experiments have shown that the life was drastically reduced by large amounts of solids in the feed water. For example, life less than 100 drill-meters have been measured when the feed water contained about 0.5% w/w solids (Öderyd 2001).

To evaluate the possibilities of this system, a complete mobile prototype service unit for use with low-cost clear-water hammers has been constructed (Tuomas 2001). The unit includes all components required for efficient drilling, i.e. systems for both pressurising drilling fluid and particle-fluid separation (by a lamella thickener and a hydro-cyclone unit) to enable recycling. The prototype unit is presently undergoing initial operational tests in order to establish the relation between tool life and particle content in the drilling fluid. System characteristics for the prototype were estimated by simulations with a process model, implemented within the Matlab Simulink math package. Particle size distributions, concentrations, and flows are resolved at strategic locations, which make the model suitable as a tool for optimisation and development of next generation systems.

This paper describes the process in the prototype system and how it is modelled, and discusses the simulated results for different system configurations. In addition, field test data of the lamella thickeners cleaning capacity are also presented.

2. Prototype system description

2.1. General description

The process in the prototype system is described in Fig. 2. A plunger pump (P4) pressurises water, which is used for driving the hammer tool and for flushing the borehole. Particle-contaminated water is returned for cleaning before re-use. The cleaning process is based on a lamella thickener with a flocculation system and a hydro-cyclone unit. The equipment was built into a single container (Fig. 3) for ease of transport and handling. The complete system is described in more details by Tuomas (2001). Table 1 summarises some important specifics of the system.

2.2. Fluid cleaning system

The prototype cleaning system uses gravity sedimentation for primary separation of particles from drilling...
water. The lamella thickener is of cross-flow type, leading to a horizontal flow between inclined lamellas (Fig. 4). Particles settle onto the lamella and slide towards the centre of the unit and eventually reach the bottom of the tank. A horizontal conveyor transports the sediment towards the end of the settling unit, where another inclined conveyor removes the waste out of the system. This second conveyor also serves to dewater the waste in order to achieve low water consumption. The settling unit is equipped with a pump for sediment removal if the conveyors are insufficient. Efficiency of sedimentation processes can be significantly improved by adding a flocculent to the incoming slurry flow. These substances gather individual fine and colloidal particles into clumps (flocks) that settle out more easily. In addition, particle-fluid separation can be achieved with hydro-cyclones. The idea is to use the hydro-cyclones as an alternative to flocculation. The hydro-cyclone unit has a \( d_{50} \) cut-point below 5 \( \mu \)m (particles with density 2750 kg/m\(^3\) in water). It is designed for a 0.3 m\(^3\)/min flow and consists of sixty \( 10 \)-mm hydro-cyclones.

### 3. System process model

A numerical model for simulation of particle flows in the prototype system has been developed. Mathematical expressions for significant components are derived, and the whole model is implemented within the Matlab Simulink™ math package. Results of main interest are

### Table 1

**System specification**

<table>
<thead>
<tr>
<th>Drilling fluid:</th>
<th>Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>High-pressure pump (P4)</td>
<td>3-cylinder plunger pump</td>
</tr>
<tr>
<td>Speed range 175–370 rev./min</td>
<td>Speed range 175–370 rev./min</td>
</tr>
<tr>
<td>Flow (2-inch plungers): max 286 l/min at max 260 bar</td>
<td>Flow (2-inch plungers): max 286 l/min at max 260 bar</td>
</tr>
<tr>
<td>Flow (2.25-inch plungers): max 361 l/min at max 233 bar</td>
<td>Flow (2.25-inch plungers): max 361 l/min at max 233 bar</td>
</tr>
<tr>
<td>Flow (3-inch plungers): max 643 l/min at max 131 bar</td>
<td>Flow (3-inch plungers): max 643 l/min at max 131 bar</td>
</tr>
<tr>
<td>12-l V6 with turbo and intercooler</td>
<td>12-l V6 with turbo and intercooler</td>
</tr>
<tr>
<td>Volume ( \sim 0.5 ) m(^3)</td>
<td>Volume ( \sim 0.5 ) m(^3)</td>
</tr>
<tr>
<td>Cross-flow gravity settler, waste discharge with screw conveyors or pump (see Fig. 4). Volume ( \sim 4.5 ) m(^3)</td>
<td>Cross-flow gravity settler, waste discharge with screw conveyors or pump (see Fig. 4). Volume ( \sim 4.5 ) m(^3)</td>
</tr>
<tr>
<td>Volume ( \sim 1.9 ) m(^3)</td>
<td>Volume ( \sim 1.9 ) m(^3)</td>
</tr>
<tr>
<td>Tank size (T1): ( \sim 0.3 ) m(^3). Pump (P1) flow rate: max 0.78 l/min</td>
<td>Tank size (T1): ( \sim 0.3 ) m(^3). Pump (P1) flow rate: max 0.78 l/min</td>
</tr>
<tr>
<td>(proportional to incoming slurry flow)</td>
<td>(proportional to incoming slurry flow)</td>
</tr>
<tr>
<td>Feed flow rate: 0.3 m(^3)/min. ( d_{50} \sim 5 ) ( \mu )m</td>
<td>Feed flow rate: 0.3 m(^3)/min. ( d_{50} \sim 5 ) ( \mu )m</td>
</tr>
<tr>
<td>Container, freight possible by truck and train</td>
<td>Container, freight possible by truck and train</td>
</tr>
<tr>
<td>7820×2438×2591 mm</td>
<td>7820×2438×2591 mm</td>
</tr>
<tr>
<td>( \sim 10 ) tonnes</td>
<td>( \sim 10 ) tonnes</td>
</tr>
</tbody>
</table>
the time dependent particle size distribution functions $\Phi(s,t)$ and corresponding volume flow rate functions, $q(t)$, at different locations in the system. Fig. 5 shows the principle flow scheme and mathematical descriptions of the different blocks are presented in the following sections.

3.1. Hammer tool

The hammer tool block in the model adds particles to the system. This is mathematically described as:

$$\Phi_{\text{out}}(s,t) = \Phi_{\text{in}}(s,t) + \Phi_h(s)$$  \hspace{1cm} (1)

where $\Phi_{\text{in}}(s,t)$ and $\Phi_{\text{out}}(s,t)$ represent the particle size distributions in the fluid entering and leaving the hammer tool. $\Phi_h(s)$ represent the particles that are generated during drilling. The $\Phi$-functions also represents the volume concentration of particles in the corresponding slurry according to equation:

$$c = \int_0^\infty \Phi(s,t) \, ds = \frac{q_{\text{solids}}}{q}$$  \hspace{1cm} (2)

where $q_{\text{solids}}$ is the volumetric flow rate of solids and $q$ is the flow rate of slurry. $\Phi_h(s)$ in Eq. (1) is calculated as:
where $v$ and $A$ represent the penetration rate and borehole cross-area, respectively, $\Phi_h(s)$ is a time independent function which represents the shape of the particle size distribution curve generated by the hammer tool. The shape used in this study comes from laboratory analysis of a drill water sample, taken during typical rock drilling on ~100 m depth with a 4-inch hammer tool. The shape of $\Phi$ depends on various parameters, such as the actual borehole depth, borehole orientation, flow rate, mineral type and drill bit design as well. The shape of the curve is, however, assumed constant in this model.

The slurry flow rate, $q_{\text{out}}$, from the hammer block is assumed equal to the incoming flow rate, $q_{\text{in}}$.

### 3.2. Mixing tank and drilling fluid tank

In a tank containing a substance with concentration $c$, the changed particle concentration by time is described by a differential equation:

$$ \frac{d(cV)}{dt} = q_{\text{in}1}c_{\text{in}1} + q_{\text{in}2}c_{\text{in}2} + \cdots + q_{\text{in}n}c_{\text{in}n} - c $$

\[ (q_{\text{out}1} + q_{\text{out}2} + \cdots + q_{\text{out}m}) \]

(4)

where $q$ is the flow, $c$ is the concentration at $n$ number of intake- and outlet ports in the tank, $V$ is the volume, which may vary with time. After inserting Eq. (2) into Eq. (4), the equation for a tank with $n$ number of intakes is derived as:

$$ \frac{\partial \Phi}{\partial t} = \frac{1}{V} \left( q_{\text{in}1}(\Phi_{\text{in}1} - \Phi) + q_{\text{in}2}(\Phi_{\text{in}2} - \Phi) + \cdots + q_{\text{in}n}(\Phi_{\text{in}n} - \Phi) \right) $$

(5)

where $\Phi_{\text{in}}$ is the particle size distribution in the fluid entering the tank, $q_{\text{in}}$ is the corresponding fluid flow rate to the tank and $\Phi$ is the particle size distribution in the tank. The model assumes that both the mixing- and drilling fluid tanks are initially filled-up with clear water. The initial condition to Eq. (5) is, therefore, $\Phi(s,0) = 0$. The volume in the drilling fluid tank will steadily decrease during drilling. The reason is that the separation processes in the lamella thickener and hydro cyclones consume fluid during operation. Opening a water intake at a low fluid level, and closing it when the tank is filled solves this problem. The model is designed to work in a similar way. One of the intake flows, $q_{\text{in}}$ in Eq. (5), is changed from zero to a user defined positive value when the tank level $V$ has reached the low limit, and goes back to zero when the upper limit is reached.

### 3.3. Lamella thickener

The lamella thickener (Fig. 4) is designed for a horizontal flow of slurry between inclined lamellas. Particles settle against the lamella and slide towards the centre of the unit and eventually reach the bottom of the tank. Fig. 7 shows some principle particle trajectories between two lamellas during steady flow conditions.

Using symbols in Fig. 7, the critical settling speed for a particle, starting at point $(0,y)$, is calculated by:

$$ v_{\text{cr}} = \frac{yU}{L} $$

(6)

where $v_{\text{cr}}$ is the critical settling speed, $y$ is the path start coordinate, $U$ is the slurry flow speed and $L$ is the lamella length. Particles with settling speed lower than $v_{\text{cr}}$, starting at point $(0,y)$, will go to the overflow (accept), while particles with a higher settling speed will end up in the underflow (reject). The actual terminal settling speed for a spherical particle with diameter $d$ is
Fig. 7. Principle outline of two particle paths in a horizontal lamella thickener. Ideally, all particles larger than \( d_2 \) (belonging to path 2) will go to the underflow. Particles with sizes \( d < d_1 \) and with paths that start above path 1 will go to the overflow.

calculated as:

\[
v = \sqrt{\frac{4d(g \rho_s - \rho_f)}{3 C_d \rho_f}} \tag{7}
\]

where \( d \) is the particle diameter, \( g \) is the acceleration, \( \rho_s \) is the particle density, \( \rho_f \) is the fluid density and \( C_d \) is the form drag coefficient. By setting Eq. (6) equal to Eq. (7) and finding the corresponding particle diameter, the efficiency curve for a lamella thickener according to Fig. 7 is obtained. To solve this, an iteration procedure is required, since \( C_d \) is a function of the particle Reynolds number, \( Re \), which besides the viscosity depends on the particle diameter and the settling speed.

A commonly used function to describe the efficiency of a particle–fluid separation process is the Rosin-Rammler formula (Crowe et al., 1998):

\[
Y_c = 1 - e^{-\left(\frac{d}{d_{50c}}\right)^m} \tag{8}
\]

Here parameter \( Y_c \) denotes the corrected efficiency and is the probability of a particle with size \( d \) to go to the underflow. The parameter \( d_{50c} \) is the cut size of the corrected grade efficiency curve (or corrected partition curve), and \( m \) is a factor that affects the sharpness of the curve. When the underflow is taken into account, the efficiency \( Y \) is calculated by:

\[
Y = (1 - R)Y_c + R \tag{9}
\]

where \( R \) is the fraction of incoming fluid that goes to the underflow. Several conditions must be fulfilled for Eq. (9) to be useful in estimating the separation efficiency in a lamella thickener. The following simplifications and assumptions have been made to motivate the use of the equation:

i. Even though the slurry flow is discontinuous due to drill pipe installations, the flow is periodic with constant run/stop times. The values of parameters \( d_{50c} \) and \( m \) can thereby be chosen so that the curve \( Y \) represents the mean separation efficiency during a complete period.

Table 2
Parameters and data used in numerical simulations

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Borehole parameters</td>
<td></td>
</tr>
<tr>
<td>Borehole length</td>
<td>200 m</td>
</tr>
<tr>
<td>Borehole diameter</td>
<td>0.115 m</td>
</tr>
<tr>
<td>System parameters</td>
<td></td>
</tr>
<tr>
<td>Max. volume (drilling fluid tank)</td>
<td>1.90 m³</td>
</tr>
<tr>
<td>Min. volume (drilling fluid tank)</td>
<td>0.74 m³</td>
</tr>
<tr>
<td>Mixing tank volume</td>
<td>0.46 m³</td>
</tr>
<tr>
<td>Lamella thickener volume</td>
<td>4.54 m³</td>
</tr>
<tr>
<td>Simulation parameters</td>
<td></td>
</tr>
<tr>
<td>Simulation time</td>
<td>24 000 s</td>
</tr>
<tr>
<td>Cut size ( d_{50c} ), lamella thickener</td>
<td>45 μm</td>
</tr>
<tr>
<td>Sharpness ( m ), lamella thickener</td>
<td>2</td>
</tr>
<tr>
<td>Cut size ( d_{50c} ), hydro-cyclones</td>
<td>3.5 μm</td>
</tr>
<tr>
<td>Sharpness ( m ), hydro-cyclones</td>
<td>1.5</td>
</tr>
<tr>
<td>Solids density</td>
<td>2750 kg/m³</td>
</tr>
<tr>
<td>Fluid density</td>
<td>1000 kg/m³</td>
</tr>
<tr>
<td>Operating conditions</td>
<td></td>
</tr>
<tr>
<td>Penetration rate</td>
<td>0.6 m/min</td>
</tr>
<tr>
<td>Run time/stop time</td>
<td>5 min/1 min</td>
</tr>
<tr>
<td>Pump flow (discontinuous)</td>
<td>0.21 m³/min</td>
</tr>
<tr>
<td>Solids vol. conc. in lamella thickener underflow</td>
<td>~27%</td>
</tr>
<tr>
<td>Vol. fraction to cyclone underflow (analysis B and C)</td>
<td>0.1</td>
</tr>
<tr>
<td>Hydro-cyclone feed flow (analysis B and C)</td>
<td>0.3 m³/min</td>
</tr>
<tr>
<td>Solids conc. in tanks at ( t = 0 )</td>
<td>0</td>
</tr>
<tr>
<td>Fluid volume in tanks at ( t = 0 )</td>
<td>Filled-up</td>
</tr>
</tbody>
</table>
Table 3
Description of simulation run A, B and C

<table>
<thead>
<tr>
<th>Simulation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Particle separation by the lamella thickener, without the hydro-cyclones.</td>
</tr>
<tr>
<td>B</td>
<td>Particle separation by the lamella thickener and the hydro-cyclones. Underflow from the hydro-cyclones are disposed.</td>
</tr>
<tr>
<td>C</td>
<td>Particle separation by the lamella thickener and the hydro-cyclones. Underflow from the hydro-cyclones are fed into the mixing tank.</td>
</tr>
</tbody>
</table>

ii. The flow rate deviation of the slurry is small and does not affect the separation efficiency.

iii. The amount of solids in the slurry is approximately constant and deviations do not affect the separation efficiency.

The actual values of the parameters \( d_{50k} \) and \( m \) are chosen with respect to field data (Fig. 10) and calculations according to Eqs. (6)–(9). \( R \) is adjusted individually in each of the following simulations but is close to 0.1. The reason is that the experience based amount of \( \sim 50\% \) w/w solids in the underflow should be fulfilled. The value for the overflow and underflow are calculated by equations Eqs. (10) and (11):

\[
q_o = q(1-R) \tag{10}
\]

\[
q_u = qR \tag{11}
\]

where index \( o \) and \( u \) denote overflow and underflow, respectively. Particle size distribution curves are calculated by:

\[
\Phi_o(s,t) = \Phi(s,t)Y(s) \frac{1}{R} \tag{12}
\]

\[
\Phi_u(s,t) = \Phi(s,t)[1-Y(s)] \frac{1}{1-R} \tag{13}
\]

where \( \Phi_o(s,t), \Phi_u(s,t) \) and \( \Phi(s,t) \) represent the particle size distributions in the underflow, overflow and the feed. The terms \( (1/R) \) and \( (1/(1-R)) \) are required for the functions to correctly represent the new concentration levels together with the new flow rates.

Another detail to consider is that the volume of the lamella thickener causes a delay of \( \Delta t = v/q \) s before the incoming slurry flow particles are reported in the overflow or underflow. This is the case when the flow is laminar and no mixing occurs. The described model uses a memory buffer to stall the signal \( \Delta t \) s, according to Eq. (14):

\[
\Phi_{ou}(s,t) = \Phi_{in}(s,t-\Delta t) \tag{14}
\]

\[
\Phi_{ou}(s,t) = 0 \quad \text{for} \quad t \leq \Delta t \tag{15}
\]

Eq. (15) implies that the lamella thickener is initially assumed filled with clear water.

3.4. Hydro-cyclones

Several equations for calculation of the separation efficiency for hydro-cyclones have been derived during the past decades (Heiskanen, 1993). The parameters involved are the geometry, operating conditions and fluid characteristics. One model that is often used is the empirical model given by Plitt (Wills, 1997):

\[
d_{50k} = \frac{14.8D_s^{0.46}D_{i}^{1.21}D_{u}^{0.56}c^{0.063}h}{D_u^{0.71}h^{0.38}Q^{0.45}} \left[ \frac{\rho_s}{\rho_f} \right] \tag{16}
\]

In this equation \( D_s = \) hydro-cyclone diameter [cm], \( D_o = \) overflow diameter [cm], \( D_i = \) inlet diameter [cm], \( \phi = \) volumetric fraction of solids in the feed, \( D_u = \) underflow diameter [cm], \( h = \) cyclone height [cm], \( Q = \) feed volume flow rate [m³/h], \( \rho_s = \) solids density [g/cm³] and \( \rho_f = \) fluid density [g/cm³]. The sharpness parameter \( m \) is given as:

\[
m = 2.96e^{-1.58R} \left( \frac{D_i^2h}{Q} \right)^{0.15} \tag{17}
\]
Fig. 8. Results from numerical simulations that describe the volume concentrations of solids in the flow to the hammer tool. Curves A, B and C result, respectively, from simulation runs A, B and C.

By inserting Eq. (16) and Eq. (17) into the Rosin–Rammler formula Eq. (8) and compensating for the underflow Eq. (9), the efficiency curve for the hydrocyclone unit is obtained.

4. Resulting calculations

The described model has been implemented within the Matlab Simulink math-package and three different

Fig. 9. Results from numerical simulations that describes the particle size distribution curves at \( t = 24 \,000 \, s \) (Fig. 8) in the flow to the hammer tool. The area under the curves represents the volume concentration of solids in the flow.
simulation runs representing possible system configurations are presented here. Input parameters and conditions are described in Tables 2 and 3. Results are presented in Table 4 and Figs. 8 and 9.

4.1. Results

Important results from the simulations are the total particle volume sent to the hammer tool and the external water consumption. The reasons are that the hammer life is intimately related to the presence of abrasive particles in the flow and that water consumption (and thereby waste flow) must be low for efficient use of the system. Results presented in Table 4 indicate that the mean concentration of solids in the feed to the hammer tool is approximately 0.44% w/w when the lamella thickener is used for particle–fluid separation (simulation A). Particle flow is reduced by ~80% when hydrocyclones are used as a complement (simulation B) and the underflow is disposed. When underflow from the hydro-cyclone unit is re-used (simulation C), the reduction is about 50%. The simulations are valid for the case of no flocculent in the flow. Fig. 8 describes the solids volume concentration curves in the flow to the hammer tool, during drilling of a 200-m deep borehole. The particle size distribution curves (for t=24 000 s in Fig. 8) are presented in Fig. 9.

5. Conclusion

Drilling with water driven DTH-hammers is a recently developed method for competitive production of boreholes. The technology requires large flow rates of preferably high quality fluid to drive the hammer tool and flush the borehole. One method to reduce the consumption is to process and recycle the used drilling fluid. Studies have been performed to find cost-effective suitable cleaning methods and a mobile prototype unit has been developed. This unit includes components for both pressurising and cleaning drilling fluid to enable recycling and thereby efficient drilling.

A process simulation model was also developed within this project. Simulations determine particle size distributions, concentrations and flows at strategic locations, whereby the system configuration can be optimised. Results indicate that particle flows to the hammer tool is reduced by ~80% when hydro-cyclones are used as a complement to the lamella thickener, and the underflow is disposed. When underflow from the hydro-cyclone unit is re-used, the reduction is about 50%. The simulations are valid for the case of no flocculent in the flow.

Practical prototype experiences and results from numerical simulations will be used in designing next generation systems, leading to even more cost-effective production with further increases in the competitiveness of the drilling method.

Acknowledgments

This work was supported by Technology Link Foundation, The Research Council of Norrbotten and Wassara AB. They are greatly acknowledged.

Appendix A: Field data

The mining company LKAB in Malmberget, Sweden, has during year 2001 produced several boreholes for safety investigations. The distances from the ground down to the mine were measured and rock surveillance systems were installed to monitor movements. The latter is a sign of instability and, therefore, a hazard for the residents and the surrounding environment. The holes were drilled using water driven DTH-hammer tools together with the above described prototype system.
Recycling was used during approximately 200 m of drilling, and the capacity of the lamella thickener was studied. Conditions for this drill work are presented in Tables 5 and 6. Laboratory results are shown in Table 7 and Fig. 10.

References


PAPER VII

Evaluation of ground thermal conductivity from drilling data

Evaluation of ground thermal conductivity from drilling data

G. Tuomasa,*, A.-M. Gustafssona

Division of Renewable Energy, Luleå University of Technology, Luleå SE-971 87, Sweden

Abstract

Knowledge of ground thermal conductivity is necessary for proper design of, e.g., borehole heat exchangers (BHE). Data are usually obtained by either laboratory analysis of ground samples (rock cores) or in situ measurement using a thermal response test system. In a thermal response test, heat is injected or extracted to or from the finished BHE and the resulting temperature response is used to evaluate the ground thermal conductivity and thermal resistance.

In this paper, a new and more efficient way to conduct the thermal response test is suggested. Here, the energy injected during drilling corresponds to the energy injected in a conventional thermal response test. For ordinary drilling methods, energy in the form of pressurised fluid, mechanical torque, and mechanical feed force is injected ($W_1$ in Fig. 1), which all dissipate into heat. Part of the heat stays in the borehole ($Q_{cv}$ in Fig. 1), while the rest leaves the system, mainly with the fluid and through conduction into the formation ($W_2$ and $W_3$ in Fig. 1). By determining these energy flows, the ground thermal conductivity can be estimated.

There are many advantages with this new measurement method. Ground thermal conductivity values would be evaluated continuously along the borehole, i.e. conductivity data are obtained through the formation. This feature could, for example, be used during production drilling in mines to instantly find lithological boundaries, thereby increasing ore extraction efficiency. Energy storage systems could be dynamically designed since the system capacity would be recognized and verified during drilling. Problems with this method occur when the formation contains fractures, and when ground water flow exerts a great influence.

The presented simulation results, where realistic drilling parameters were used, show that the method has the potential to replace the conventional thermal response test method.

Keywords: Ground; Conductivity; Drilling; Borehole; Energy; Thermal response test

Fig. 1. Energy flow through a control volume during drilling. Here $W_1$ represents the injected energy, $W_2$ is the energy leaving the borehole and $W_3$ is the energy transferred to the formation. $Q_{cv}$ represents the internal energy inside the control volume.

*Corresponding author. Tel.: +46-920-492929; fax: +46-920-491697.
E-mail address: goran.tuomas@sb.luth.se (G. Tuomas).

For full length paper see CD-ROM attached.

EVALUATION OF GROUND THERMAL CONDUCTIVITY FROM DRILLING DATA

G.Tuomas¹, A-M. Gustafsson²

¹) Div. of Renewable Energy, Luleå University of Technology, SE-971 87 Luleå, Sweden
goran.tuomas@sb.luth.se

²) Div. of Renewable Energy, Luleå University of Technology, SE-971 87 Luleå, Sweden
anna-maria.gustafsson@sb.luth.se

Abstract: In this paper a new method for evaluating ground thermal conductivity is suggested. The principle is based on ordinary drilling where energy is injected into the borehole in the form of pressurised fluid, mechanical torque, and mechanical feed force, which all dissipate into heat. Part of the heat leaves the borehole with the fluid while the rest is mainly transferred into the formation. By determining the energy flows, ground thermal conductivity can be estimated. This new measurement method would have many advantages. Ground conductivity values would be continuously estimated along the borehole, meaning that values are obtained through the formation. This quality could, for example, be used during production drilling in a mine to instantly detect lithological boundaries, resulting in increased ore extraction efficiency. Energy storage systems could be dynamically designed since the system capacity could be recognized and verified during drilling. Presented simulation results, where realistic drilling parameters were used, show that the method has the potential to be practically applied.

Keywords: Ground; Conductivity; Drilling; Borehole; Energy; Thermal response test

1. INTRODUCTION

Knowledge of ground thermal conductivity is important in many different contexts. For example, with Borehole Thermal Energy Storage (BTES) systems, ground thermal conductivity is directly related to the system capacity and characteristics. BTES-systems are mainly used in buildings for purposes of heating or cooling or both, i.e. extraction of heat or cold from the ground. Such systems vary from a single borehole up to several hundred boreholes. The first system was built in Luleå in 1982, and consisted of 120 boreholes (Nordell, 1994). Further, a system in Stockton, USA consisting of 400 boreholes was constructed (Stiles et al., 1998), and at present, a 600-borehole system is being constructed in Oslo, Norway. The larger the system, the more important it is to know the thermal conductivity value of the ground, thereby avoiding improper dimensioning. Today, conductivity data is usually obtained by either laboratory analysis of ground samples (rock cores) or in-situ measurement with a thermal response test system (Fig. 1). During a thermal response test, heat is injected or extracted to or from a borehole and the resulting temperature response is used to evaluate the ground thermal conductivity and thermal resistance of the Borehole Heat Exchanger (BHE). A low thermal conductivity is, e.g., indicated by a more rapid temperature change of the heat carrier fluid. Mogensen (1983) originally proposed this method. He suggested that a chilled heat carrier fluid should be circulated through a BHE at a constant heat extraction rate, while the outlet fluid temperature was continuously recorded. Later on, the method was developed with a heater instead of a chiller; e.g. Gehlin (2002). This equipment was made mobile to simplify in-situ measurements at different construction sites.

Figure 1. Outline of thermal response test set-up.
Since this method is used in completed boreholes, the resulting thermal properties represent mean values along the borehole. The determined thermal conductivity value also includes the effects of both conductive and convective heat transfers, i.e. an effective (apparent) mean thermal conductivity value is obtained of the ground surrounding the borehole. This value is always higher than the thermal conductivity obtained from a laboratory test of a representative rock sample.

2. THERMAL RESPONSE TEST INTEGRATED TO DRILLING

A new way to carry out a thermal response test during drilling is proposed and described in this paper. Here, the energy released during drilling corresponds to the energy injected in a conventional thermal response test. In most drilling methods, energy comes in the form of pressurised fluid, mechanical torque, and a mechanical feed force. All these energies dissipate into heat. Part of the heat leaves the borehole with the circulating drilling fluid, while the rest is mainly transferred into the formation (Fig. 2). By determining the energy flows, the formation’s thermal conductivity can be evaluated.

This new technology would have several advantages compared to the old method. Ground conductivity values would be continuously estimated along the borehole, i.e. values are obtained through the formation. This quality could, e.g., be used during production drilling in a mine to instantly detect lithological boundaries, resulting in increased ore extraction efficiency. Borehole Thermal Energy Storage (BTES) systems could be dynamically designed to optimize the required number or length of the boreholes while drilling. This would increase the quality of the system since the energy storage capacity would be recognized and verified. The new integrated method would also give data for all boreholes, while only a few boreholes are evaluated in conventional thermal response tests. Another advantage is that measurements would be performed in thermally undisturbed formations, while existing thermal response tests are done in boreholes previously influenced by heat from drill work.

2.1 Energy flow

For a control volume that represents a borehole with a drill string (Fig. 2), the energy balance equation is written as

\[
\int (W_1 - W_2 - W_3)dt - \Delta Q_{cv} = 0
\]

where \( W_1 \) represents the injected energy flow, \( W_2 \) is energy flowing from the control volume, \( W_3 \) represents energy flow in the form of heat going into the formation, \( t \) represents time, and \( \Delta Q_{cv} \) is the change of internal energy inside the control volume.

![Figure 2. Energy flow through a control volume during drilling. Here, \( W_1 \) represents the injected energy, \( W_2 \) is energy leaving the borehole, and \( W_3 \) is energy transferred to the formation. \( Q_{cv} \) represents the internal energy inside the control volume.](image)

The energy flow \( W_1 \) represents ordinary drilling with an incompressible drilling fluid, and is defined as:

\[
W_1 = p_1 q_1 + M \omega + F v + q_1 \rho c \nu T_1
\]

where \( p_1 \) is the inlet fluid pressure, \( q_1 \) is the inlet volume fluid flow rate, \( M \) is the mechanical torque acting on the drill string, \( \omega \) is the drill string’s angular velocity, \( F \) is the feed force, \( v \) is the penetration speed, \( \rho_1 \) is the inlet fluid density, \( c_v \) is the heat capacity, and \( T_1 \) is the liquid’s inlet temperature. Parameter \( W_2 \) (in eq. 1) represents the energy flowing from the control volume (besides \( W_1 \)). \( W_2 \) is here defined as
\[ W_2 = p_2 q_2 + q_2 \rho_2 c_v T_2 + W_C \]  

(3)

where \( p_2 \) is the outlet fluid pressure, \( q_2 \) is the outlet volume fluid flow rate, \( \rho_2 \) is the outlet fluid density, \( c_v \) is the heat capacity, \( T_2 \) is the liquid’s outlet temperature, and \( W_C \) is the energy flow due to vertical heat conduction in the drill string. Parameter \( W_3 \) in eq. 1 is defined as the energy flow into the formation. By integrating \( W_3 \) with time, the energy that has reached the formation may be written as

\[ Q_3 = \int \int \int \int W_3 \cdot dV = \int \int \int \rho c_v T \cdot dV - \int \int \int \rho c_v T_0 \cdot dV \]  

(4)

where \( \rho \) is the rock density, \( c_v \) is the rock’s heat capacity, \( T \) is the rock temperature, \( T_0 \) is the initial undisturbed rock temperature, and \( V \) is the affected volume. The last parameter, \( \Delta Q_{cv} \) in eq. 1, represents the change of internal energy within the control volume.

3. SIMULATION RESULTS AND DISCUSSION

The presented method to determine the ground thermal conductivity was modelled and simulated using the CFD-analysis software Fluent. Parameters from real water driven down-the-hole (DTH) drilling were used to find realistic values of the calculated data. A detailed description of the hammer tool’s function is given by Tuomas (2003). Primary interest was in the drill water’s outlet temperature since this reveals the energy flow into the formation. Continuous drilling of a 115-mm borehole from 0 to 160 m deep was simulated, while the inlet fluid (water) was assumed to have the same temperature as the undisturbed ground (10ºC). The heat transfer (150 kW) into the fluid was distributed over the DTH-hammer length (~1 meter) at the end of the drill string (borehole bottom) because energy dissipation into heat starts in the hammer through internal leakages, friction, etc. Water losses and changes of the properties of water due to an increased amount of solids were not considered, (Tuomas 2003). The formation was assumed to be an isotropic homogenous material, without cracks or ground water flow. The simulation model is an axi-symmetric model of Fig. 2 and assumes an unsteady, incompressible flow. Table 1 summarises the input data and results are presented in Figures 3 to 7.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drilling type</td>
<td>Water driven DTH hammer drilling (Wassara)</td>
</tr>
<tr>
<td>Borehole depth</td>
<td>0 meter at time t=0. Simulation stops at 160 meter (t=320 min).</td>
</tr>
<tr>
<td>Penetration speed</td>
<td>0.5 m/min (continuous)</td>
</tr>
<tr>
<td>Borehole diameter</td>
<td>115 mm</td>
</tr>
<tr>
<td>Released energy</td>
<td>150 kW</td>
</tr>
<tr>
<td>Position of energy release</td>
<td>Uniformly distributed along 0-1 meter of the drill string at the borehole bottom (moving position rel. to the ground)</td>
</tr>
<tr>
<td>Drilling fluid</td>
<td>Water</td>
</tr>
<tr>
<td>Drilling fluid flow</td>
<td>300 litres/min</td>
</tr>
<tr>
<td>Drilling fluid inlet temp.</td>
<td>10ºC</td>
</tr>
<tr>
<td>Undisturbed ground temp.</td>
<td>10ºC (at time t=0)</td>
</tr>
<tr>
<td>Ground geothermal temperature gradient</td>
<td>0 (except for analysis presented in fig. 7)</td>
</tr>
<tr>
<td>Ground thermal conductivity</td>
<td>( \lambda = 1 \text{ W/(m·K)} ) (e.g. sandstone*).</td>
</tr>
<tr>
<td></td>
<td>( \lambda = 3 \text{ W/(m·K)} ) (e.g. granite*).</td>
</tr>
<tr>
<td></td>
<td>( \lambda = 5 \text{ W/(m·K)} ) (e.g. magnetite*). (see figures)</td>
</tr>
<tr>
<td>Ground density</td>
<td>2630 kg/m³</td>
</tr>
<tr>
<td>Ground specific heat</td>
<td>775 J/(kg·K)</td>
</tr>
<tr>
<td>Drill pipe outer diameter</td>
<td>89 mm</td>
</tr>
<tr>
<td>Drill pipe thickness</td>
<td>6.3 mm</td>
</tr>
<tr>
<td>Drill pipe material</td>
<td>Steel</td>
</tr>
<tr>
<td>Borehole surface smoothness</td>
<td>3 mm (profile height)</td>
</tr>
</tbody>
</table>


The temperature development of the circulating fluid is presented in Figure 3. Here, the energy release near the bottom of the borehole, heats up the circulating fluid to a higher temperature than the surrounding formation. This is obvious since the inlet fluid has the same temperature as the undisturbed ground (10ºC). When the fluid flows
upwards, heat will be transferred to both the inlet fluid in the centre of the drill string (see Fig. 2) and to the formation, explaining the temperature decrease of the upwards-flowing fluid.

During drilling, the point of energy release is mainly at the front of the borehole, thereby continuously moving deeper into the ground. Thus, during continuous drilling, the front of this moving heat source is always at approximately the same thermal conditions, even as the ground surrounding the upper parts of the borehole is becoming warmer. The thermal situation along the borehole will thereby develop, as the borehole becomes deeper with time while the ground temperature around the borehole increases. The highest ground temperature occurs at the wall near the bottom of the borehole, though the heating does not reach this far into the formation. The ambient air temperature will influence the temperature close to the ground surface. This leads to the conclusion that the heat flow to the formation is greatest at the bottom of the hole, i.e. where the heat is generated. Figure 4 shows isothermal curves at the position where the formation’s temperature has increased by 1°C (to 11°C) at time t=320 min, corresponding to drilling at 160 meters depth. The different curves represent formations with thermal conductivity $\lambda = 1$, $\lambda = 3$, and $\lambda = 5 \text{ W/(m·K)}$.

The expected outlet water temperature during drilling at different depths is presented in Figures 5 and 6. Initially, the temperature difference between the inlet and outlet water may be estimated with the equation $\Delta T = W/m_c$, where $W$ is the power generated during drilling, $m_c$ is the mass flow of
water, and \( c_w \) is the specific heat of the water. In this simulation, the highest possible temperature difference is \( \approx 7.2^\circ C \), which can be determined from Figures 5 and 6. As the drill starts to penetrate the rock, thereby exposing the borehole wall to the fluid, heat will be transferred into the formation. It can be seen from Figure 5 how the temperature varies during drilling at different depths and for different values on the ground thermal conductivity. The values \( \lambda = 1, \lambda = 3, \) and \( \lambda = 5 \) can be related to sandstone, granite, and magnetite, see Sundberg (1988) and Hofmeister (2001). The temperature difference between the inlet and outlet water diminishes as the ground thermal conductivity increases. The formation will then transfer the heat more effectively, resulting in a lower borehole surface temperature. This lower borehole surface temperature will create a larger heat flow between the upwards-flowing water and the formation, i.e. a lower outlet water temperature.

Figure 6. Temperature difference between the outlet and inlet water during drilling in two different formations. The formations have piecewise constant thermal conductivity, which changes at depths 60 m and 100 m.

Another characteristic of the proposed method is that variations in ground thermal conductivity may be detected by a change in the outlet water’s temperature gradient, presented in Figure 6 for two different formations, each with two different values on the thermal conductivity. Using formation B as an example, the thermal conductivity changes from \( \lambda = 3 \) to \( \lambda = 5 \) at 60 meters depth, which is identified by the discontinuity in the curve’s gradient. This means that an iron ore deposit could be revealed and mapped in an environment of granite, simply by ordinary drilling and analysis, according to this paper. Formation A changes from \( \lambda = 3 \) to \( \lambda = 1 \) at 60 meters depth, which could be of interest, for example, in the construction and design of BTES-systems.

Figure 7. Temperature difference between the outlet and inlet water during drilling at different depths in formations with thermal conductivity \( \lambda = 3 \) W/(m·K) and varying geothermal temperature gradient.

The geothermal temperature gradient was not considered in the calculations presented above. To determine its effect on the outlet water temperature, four different formations with geothermal gradients, from 0°C/m to 0.03°C/m were analysed. Figure 7 shows that the fluid temperature increases with increases in borehole length and geothermal gradient.

4. CONCLUSION

A new method to evaluate the ground’s thermal conductivity is suggested. The idea is to use energies released during ordinary rock drilling for determination of thermal conductivity data. Relevant equations are presented and simulated results are shown for a 115-mm diameter vertical borehole drilling. The power of 150 kW was injected near the bottom of the borehole to a 300 l/min drill water flow. Numerical results show, e.g., that the temperature difference between out-
and in flowing water is 5.8 degrees after drilling a 160-metre borehole, when the formations thermal conductivity is 1 W/(m·K). The temperature difference decreases to 4.7 degrees when the conductivity is 5 W/(m·K). Here, the ground’s temperature was initially equal with the inlet water’s temperature (10ºC) and the geothermal temperature gradient was zero. Results also indicate that changes in the ground’s thermal conductivity value can be detected by changes in the outlet water’s temperature gradient. Curves that show the effect from different geothermal temperature gradients are also presented.

The new method has the potential to become an alternative to the conventional thermal response test method. The advantages would primarily be continuous thermal conductivity data along the borehole, without influence from collectors and borehole filling materials. Other usage areas such as detection of lithological boundaries during production drilling in mines may also prove valuable. For this method to be successful, the inverse problem must however be analysed, i.e. an unknown thermal conductivity must be determined from energy flows in the system. This problem was not analysed in this work and needs further attention.

5. REFERENCES


