An Axisymmetric Finite Element Solution for Elastic Wave Propagation Through Threaded Connections

by

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AN AXISYMMETRIC FINITE ELEMENT SOLUTION FOR ELASTIC WAVE PROPAGATION THROUGH THREADED CONNECTIONS

by

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(Abstract)

An axisymmetric finite element solution method is developed for axial wave propagation through a series of threaded connections in rock drills. A piston impacts axially on a string of rods held together by threaded joints and the wave propagates through these joints before reaching the bit. The energy lost in the joints limits the maximum effective depth of the drill.

Several computational techniques are used to efficiently model the problem. Non-reflecting boundaries are used to numerically absorb the waves as they exit a joint. The stored waves are then reinitiated into the next joint eliminating modeling of the entire assembly of rods. The preload in the threads is modeled by shrinking the threaded sleeve onto the rods. A new dynamic relaxation damping scheme is used which starts with an undamped model and then increases the damping until the solution converges. This method converges more rapidly than the standard constant damping.
Influences of friction, initial preload and wave shape on transmission efficiency are studied. Friction had little influence on energy loss in the joints. Initial preload had a more pronounced effect with higher joint preloads resulting in higher transmission efficiencies. An idealized wave was not a good approximation of the input wave.

The changes in the duration, initial slope and peak stress of the wave as it passes through a joint were found to agree with the experimental data. The magnitude of the energy lost, however, was consistently lower than the experimental data.
Acknowledgment

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Introduction

Rock Drilling

Percussive drills are in common use producing holes in rocks, which are then filled with an explosive and blasted. The rock is removed for various reasons. Many times, the rock itself is an end product while other times it is removed to mine an ore, or simply for excavation as part of a construction project.

There are three basic types of percussion rock drills which fall into two categories. The first category is the “top hammer” where the percussion takes place outside the hole. A piston is accelerated and impacts axially on a rod thereby initiating a stress wave propagating along a string of connected rods towards a rock bit where a portion of the wave propagates into the rock and produces the useful work of fracturing the rock. The hammer also independently rotates the drill string to index the bit between each successive percussive blow. The second category of drills is the “down hole hammer” where, as the name implies, the percussion takes place at the bottom of the hole. This type of hammer has the advantage of efficient energy transfer to the rock because the piston impacts directly on the bit. However, it has the disadvantages of running on compressed air, which is an inefficient method of delivering energy, and of having the expensive percussive actuator located in the hole where a cave-in can result in the loss of the hammer.
Top hammers come in two types, air percussion and hydraulic percussion. The two types of actuators vary in overall efficiency because of the relative inefficiency in compressing air as compared to pumping oil. However, they also have a significant difference in that the low pressure available with compressed air requires a piston with a very large shoulder area to accelerate it to the necessary speed (generally 10 m/s) while the relatively high hydraulic pressures available need very small shoulder areas on the hydraulic pistons. The air pistons are therefore vastly different in shape as compared to the hydraulic pistons (see Figure 1). The two types of pistons, because of these geometric differences, have two distinctive shapes of stress waves propagating toward the bit.

The physical design of the drilling machine dictates the maximum continuous length of the drill extension rods. For holes deeper than one rod length (typically 3.6 meters) additional rods must be coupled to the first rod. Then, when the final depth has been reached, the joints are loosened by generating tensile wave reflections off of the bit in a free end condition [1] and the rods are removed and placed back in the storage rack.

Top hole drills are currently limited to an effective range of about 25 meters (based on 3.6 m extension rods). This range limitation is due almost entirely to the energy loss in the joints between each drill rod. These joints consist of the two threaded rod ends held firmly together by a threaded sleeve (see Figure 2). It is the transfer of energy through these threaded joints that is the focus of the present research. Energy content in this context usually means the productive energy for the breakage of rock and is defined as energy contained in a contiguous compression pulse. Energies propagating
in radial modes, in pulses with tensile stresses or in lagging pulses do not contribute to the breakage of rock and therefore are not included in the calculation of the productive energy or simply energy through the joint.

Over the past hundred years, many thread profiles have been tried with various degrees of success [2]. Before the commercial introduction of hydraulic drills, an ISO defined rope thread had become the common thread of choice. After hydraulic drills began to overtake the market, the rope thread was too difficult to rattle loose with the slim hydraulic pistons and a new “T” thread gradually became the standard. The evolution of joint designs as well as piston shapes has resulted in significantly improved energy transfer through the joints. Twenty years ago a steady state temperature on the outside surface of the couplings of 370° C due to energy losses in the joint was considered common [2]. Today, however, with hydraulic drills and the “T” thread, temperatures around the joint are generally below 120° C.

**Previous Investigations**

Many people in the last 40 years have studied drill string joints both experimentally and analytically to gain a better understanding of the energy transmission properties of the joints [3-9]. Most of the experimental data available was taken with air hammers and relatively little experimental data with hydraulic hammers is found in the open literature.
Experimental Studies

In 1955 Clausing studied the loss of drilling rate due to extension rods by drilling one hundred feet with a percussive air hammer using both 1.2 m rods and 6 m rods [3]. His results are shown below in Table 1.

<table>
<thead>
<tr>
<th>Number of Joints</th>
<th>Loss of Drilling Rate from Previous Rod</th>
<th>Loss of Drilling Rate from Initial Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>6 m rods</td>
<td>1.2 m rods</td>
</tr>
<tr>
<td>1</td>
<td>18%</td>
<td>18%</td>
</tr>
<tr>
<td>2</td>
<td>12%</td>
<td>12%</td>
</tr>
<tr>
<td>3</td>
<td>6.7%</td>
<td>6.7%</td>
</tr>
<tr>
<td>4</td>
<td>3.5%</td>
<td>6%</td>
</tr>
<tr>
<td>5</td>
<td>na</td>
<td>5%</td>
</tr>
<tr>
<td>6</td>
<td>na</td>
<td>3.4%</td>
</tr>
<tr>
<td>7</td>
<td>na</td>
<td>3.5%</td>
</tr>
<tr>
<td>8</td>
<td>na</td>
<td>3.5%</td>
</tr>
<tr>
<td>10</td>
<td>na</td>
<td>3.5%</td>
</tr>
<tr>
<td>14</td>
<td>na</td>
<td>na</td>
</tr>
<tr>
<td>18</td>
<td>na</td>
<td>na</td>
</tr>
</tbody>
</table>

na = data not available

One of the earliest published studies of energy loss in the joints was by Takaoka, Hayamizu and Misawa in 1958 [9]. In the paper three types of joints; tapered, buttress, and upset, are considered. A pendulum hammer was used to initiate an elastic pulse into two rods connected by a threaded joint. Strain gages recorded the wave before the joint. The total energy loss in the joint was computed as the difference in the initial pulse energy and the final pulse energy after it had propagated through the joint, reflected off of the free end and then propagated back through the joint as a tensile wave. The authors
point out that this calculation sums both the energy dissipated by the transmission of the compressive pulse and the energy dissipated by the returning tensile pulse. Clearly, the loss in the threaded joints is not guaranteed to be the same for a tensile and compressive pulse so simply dividing the sum by two is not completely accurate. The authors also considered two specific modes of energy loss. First, the energy lost by reflection was calculated as the energy of the first reflected pulse expressed as a percentage of the initial pulse energy. Second, frictional losses in the joint itself are calculated as the difference in the energy entering the joint (initial pulse energy minus energy in the first reflected pulse) and the energy exiting the joint. The reported results are shown below in Table 2.

<table>
<thead>
<tr>
<th>Joint Type</th>
<th>Energy Loss due to Reflection</th>
<th>Energy Loss due to Friction</th>
<th>Total Energy Loss Measured (round trip)</th>
</tr>
</thead>
<tbody>
<tr>
<td>None</td>
<td>0%</td>
<td>0%</td>
<td>10%</td>
</tr>
<tr>
<td>Tapered</td>
<td>4%</td>
<td>3.5%</td>
<td>17.5%</td>
</tr>
<tr>
<td>Buttress</td>
<td>3%</td>
<td>5.5%</td>
<td>18.5%</td>
</tr>
<tr>
<td>Upset</td>
<td>2%</td>
<td>16.5%</td>
<td>28.5%</td>
</tr>
</tbody>
</table>

Recently, Fu and Land [5] studied the effects of two different wave forms on the energy transmission from the piston into the rock. Both pistons were from commercial hydraulic hammers and were both long and slender. Experimental data was taken with strain gages configured to cancel bending on each of six rods. The data shows the same loss of energy in the joints for both wave forms and the same type of changes to the waves by successive passes through the joints. As the wave passed through successive
joints the length of the wave and its rise time increased, while the peak stress and initial slope decreased (initial slope was defined as the peak stress divided by the time from the start of the wave to the peak stress). The quantitative results are listed below in Table 3 and the wave forms are illustrated in Figure 3.

Table 3. Experimental Results of Fu and Land [5]

<table>
<thead>
<tr>
<th>Number of Joints</th>
<th>Energy Loss</th>
<th>Peak Stress</th>
<th>Slope of Wave Front</th>
<th>Blow Duration</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0 %</td>
<td>265 MPa</td>
<td>1621 MPa/ms</td>
<td>0.395 ms</td>
</tr>
<tr>
<td>3</td>
<td>18%</td>
<td>235 MPa</td>
<td>1103 MPa/ms</td>
<td>0.420 ms</td>
</tr>
<tr>
<td>5</td>
<td>22%</td>
<td>215 MPa</td>
<td>896 MPa/ms</td>
<td>0.480 ms</td>
</tr>
</tbody>
</table>

**Analytical Studies**

In 1956 Sullivan et al. [8] studied the factors involved in the loss of drilling rate in long holes. Only energy reflected at the joint was calculated. It was reported to be 10% per joint based on the change in cross sectional area and the ratio of pulse length to the joint length typical of air hammers at the time. No consideration was given to any changes in transmission efficiency due to shape changes in the wave as it passed through successive joints.

Fu and Paul [6] published a method of characteristics for one dimensional analysis of the drilling process in 1970 which, using the change in mass per unit length at the joint, accounted for losses due to reflections at the joint. For the piston and joint studied in the present work, the energy loss calculated using this technique is listed below in Table 4 and the wave forms are illustrated in Figure 4.
Table 4. Results Computed using the Algorithm of Fu and Paul [6]

<table>
<thead>
<tr>
<th>Number of Joints</th>
<th>Energy Loss from Energy in the First Rod</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>14%</td>
</tr>
<tr>
<td>5</td>
<td>20%</td>
</tr>
</tbody>
</table>

Lundberg [7] has treated the joints analytically as (a) a constant change in cross section and (b) a rigid body of concentrated mass. He defines the efficiency $\eta$ by

$$\eta = \frac{\int \sigma_i^2 dt}{\int \sigma_i^2 dt}$$  
(1)

where $\sigma_i$ is the initial energy and $\sigma_t$ is the energy transmitted through the joint, and derives expressions for the efficiency of the joint based on its geometry and the length of a rectangular pulse. For the elastic joint with a rectangular pulse equation (1) becomes

$$\eta = 1 - 2p^2 \frac{(1 - p^{2m})}{(1 - p^m)}$$  
(2)

where $p = (r-1)/(r+1)$; $r$ being the ratio of joint area to rod area, and $m$ the pulse length expressed as an integer multiple of the joint length. For the rigid mass model and a rectangular pulse

$$\eta = \frac{4m}{(r - 1)}$$  
(3)
where the rigid joint mass is defined as the mass of the coupling sleeve only (without the rods within the sleeve). He also presents numerical solutions of the multiple joint problem with the rigid mass assumption. These results show that for multiple joints the efficiency of each successive joint increases thereby implying that the shape of the wave is being transformed into a more efficient form with each pass through a joint. He explains this by postulating that the joint acts as a low pass filter removing the high frequency components as the wave passes through it. Data interpolated from his graph of efficiency for the case \( m=6 \) and \( r=3.4 \) are shown below in Table 5.

<table>
<thead>
<tr>
<th>Number of Joints</th>
<th>Energy loss from an initial rectangular pulse</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>9%</td>
</tr>
<tr>
<td>2</td>
<td>15%</td>
</tr>
<tr>
<td>5</td>
<td>25%</td>
</tr>
<tr>
<td>10</td>
<td>34%</td>
</tr>
</tbody>
</table>

**Discussion and Comparison of Experimental and Analytical Results**

Note that except for Takaoka et al. [9] all of the experimental results presented above base the initial energy calculation (or initial drilling rate) on the measured energy (or rate) while drilling with the first extension rod. Therefore the baseline is set after the wave has already passed through one threaded joint. Consequently, if the joints indeed act as a filter as Lundberg [7] suggests, the total loss of productive energy in the joints from the initial piston kinetic energy is even more dramatic than the data suggests at first glance. The graph in Figure 5 shows the experimental and analytical results discussed
above. It should be noted that it is normal practice to assume that the drilling rate is directly proportional to the product of the energy content of the wave and the percussive frequency.

In comparing the data in Figure 5, it is important to distinguish between the wave forms being studied. Sullivan [8] and Lundberg [7] studied rectangular pulses with lumped mass joints. Clasuring [3] studied air hammers with rope thread joints and Fu and Land [5] studied hydraulic hammers with T thread joints. It is generally well established that the energy in a wave generated with a long and slender hydraulic piston is more efficiently transmitted than that in a wave from an air hammer when drilling with several extension rods.

**Statement of the Problem**

The present problem is to formulate and find an axisymmetric finite element solution to the axial wave propagation through a series of threaded rock drill joints. It is desired to develop an understanding of the wave interaction with the joint so that future studies could improve the design of the joint or the shape of the wave. Some limitations with the present work are:

- Loss of accuracy from the axisymmetric assumption and from neglecting torsional effects.
• Ability to solve the problem within computer hardware constraints given the relative geometric size of the threads compared to the overall distance of propagation.

• Proper accounting of the effect of initial stresses due to torque on the threaded joints.

Several schemes for dealing with these issues will be investigated and presented. The drill string components selected for this study are Ingersoll-Rand 38mm T-thread rods and couplings. These components were selected as representative of common industrial use and of the studies presented above. The stress wave selected for study is from an Ingersoll-Rand MP317 hydraulic drill piston with an axial impact velocity of 9.8 m/s. The solutions were generated by using a modified version of DYNA2D. DYNA2D is an explicit finite element code developed by the Methods Development Group at Lawrence Livermore National Laboratory for solving non-linear transient problems in solid mechanics [10]. All work presented herein uses four node quadrilateral elements.
Computational Techniques

Non-reflecting Boundary Conditions

As discussed in the introduction, one of the challenges in solving the drill rod coupling problem is the computational size of the problem. The computational constraints arise from the relative magnitude of the physical dimensions involved. The wave studied is approximately 1.5m long while the detailed geometry inside the joint requires elements on the order of 0.1mm. Therefore to analyze successive passes through six joints, millions of elements would be needed in a single model. A model of this size, although solvable, is not practical for parametric studies. Therefore, it was desired to use a method that would numerically “absorb” the wave after it exited the joint and store it for the next successive pass through the joint. This would allow the model to omit the rod section between each joint and solve for only the joint itself for each pass. Standard non-reflecting boundary conditions as implemented in DYNA2D were used for this purpose.

Theory

Non-reflecting boundaries, also referred to in the literature as absorbing boundaries, silent boundaries and transmitting boundaries, are boundary conditions applied to end nodes to simulate an infinite medium. A perfect non-reflecting boundary would absorb all the energy of an impinging wave ensuring that no spurious reflections were induced in the model. Therefore, the boundary conditions are formulated to
minimize any reflections at the boundary [11]. The standard implementation was
developed by Lysmer and Kuhlemeyer [12] in 1969. They proposed a set of linear
dampers applied to each boundary node to absorb both normal and shear stresses. The
damping rate is computed as the product of mass density $\rho$ and wave speed. So the
external tractions applied are

$$\sigma = a\rho \dot{w} V_p$$  \hspace{1cm} (4)

$$\tau = b\rho \dot{u} V_s$$  \hspace{1cm} (5)

where $a$ and $b$ are dimensionless constants, $V_p$ and $V_s$ are the propagation speeds of P
and S-waves respectively, and $\dot{w}$ and $\dot{u}$ are the particle velocities normal and tangent to
the surface respectively. These traction conditions are implemented in DYNA2D as
external nodal forces computed from the nodal velocities at each time step and applied at
the next time step.

To use the non-reflecting boundary as a means for “storing” a wave for later use,
the nodal forces computed to nullify the waves at each time step are written to a file and
then used as the input forces for the next pass through the joint.

**Limitations**

The linear dashpot implementation of the non-reflecting boundary conditions does
not completely absorb all waves impinging on the surface. The efficiency with which the
waves are absorbed is a function of the incident angle and Poisson’s ratio [12]. The
amplitudes of S and P waves reflected at the boundary from an incident shear wave are given by

\[
B = \frac{\left( \frac{b \cos(2\alpha) + s^2 \sin(2\beta)}{-\cos(2\alpha) + a \sin(\beta)} \right) (\sin(2\alpha) - a \cos(\beta)) - (-\cos(2\alpha) - b \sin(\alpha))}{\left( \frac{-b \cos(2\alpha) - s^2 \sin(2\beta)}{-\cos(2\alpha) + a \sin(\beta)} \right) (\sin(2\alpha) + a \cos(\beta)) + (\cos(2\alpha) - b \sin(\alpha))}
\]  

(6)

\[
A = \frac{\sin(2\alpha) - a \cos(\theta) - (\sin(2\alpha) + a \cos(\theta)) B}{a \sin(\theta) - \cos(2\alpha)}
\]  

(7)

where \( \beta \) is the incident angle (with incidence normal to the boundary defined as 90°), \( \theta \) is the angle of the reflected longitudinal wave given by Snell’s law

\[
\cos \theta = \frac{1}{s} \cos \beta,
\]  

(8)

and \( s \) is the material constant defining the ratio of S and P wave propagation speeds computed from Poisson’s ratio \( \nu \)

\[
s = \frac{V_s}{V_p} = \frac{\sqrt{1 - 2\nu}}{\sqrt{2(1 - \nu)}}
\]  

(9)

By considering the transmitted and reflected energies, the relative effectiveness of the boundary conditions can be evaluated. Lysmer and Kuhlemeyer [12] use the ratio of transmitted to reflected energy as the criterion for evaluating a boundary condition’s effectiveness. Impinging energy per unit area per unit time is given by

\[
E_s = \frac{1}{2} \rho \omega^2 A^2 V_s
\]  

(10)
for an S-wave and by

\[ E_i = \frac{1}{2} \rho \omega^2 B^2 V_p \]  

(11)

for a P wave. Since the impingement area of a plane wave is proportional to the sine of

the incident angle, the energy of an incident plane wave with a unit amplitude is

\[ E_i = \frac{1}{2} \rho \omega^2 \sin(\nu) V_s \]  

(12)

for an S wave and

\[ E_i = \frac{1}{2} \rho \omega^2 \sin(\theta) V_p \]  

(13)

for a P wave. The energy of the reflected wave is given by

\[ E_r = \frac{1}{2} \rho \omega^2 \sin(\nu) \left( B_1^2 + B_2^2 \right) V_s \]  

(14)

for an S wave below the critical angle (\( \nu < \nu_{cr} \) for normal incidence defined as 90°)

where \( B_1 \) is the real and \( B_2 \) is the complex magnitude of \( B \), and by

\[ E_r = \frac{1}{2} \rho \omega^2 \sin(\nu) B^2 V_s + \frac{1}{2} \rho \omega^2 \sin(\theta) A^2 V_p \]  

(15)

for an S wave impinging at an angle greater than the critical angle and also for a P wave.

Therefore the three energy ratios are given by
\[
\frac{E_r}{E_i} = A^2 + s \frac{\sin \nu}{\sin \theta} B^2 \quad \text{for a P wave,} \quad (16)
\]

\[
\frac{E_r}{E_i} = B^2 + \frac{\sin \theta}{s \sin \nu} A^2 \quad \text{for an S wave with } (\nu > \nu_{cr}) \text{ and} \quad (17)
\]

\[
\frac{E_r}{E_i} = B_1^2 + B_2^2 \quad \text{for an S wave with } (\nu < \nu_{cr}) \quad (18)
\]

For example, Figure 6 shows that for an impinging P wave, the waves at grazing incidence are completely reflected and waves with an incident angle greater than about 45° are nearly perfectly absorbed. So, the question which must be answered for the present problem is: Do waves in the interior of the axisymmetric rod carry significant energy content along the rod at angles nearing radial propagation? The intuitive answer is that axial wave propagation down a rod would not have significant components bouncing off of the inside and outside surfaces of the hollow rod at high incident angles. However, several test cases were run to check the accuracy of using non-reflecting boundaries to catch and reinitiate waves.

**Validation of the Use of Non-reflecting Boundaries to Split the Problem**

As illustrated in Figure 7, for case #1 a piston impacts a 32mm long uniform rod at 10 m/s axial velocity. The far end of the rod was defined with the non-reflecting boundary conditions (nrbc). Stresses were recorded in an interior element 7 mm away from the nrbc. This was taken as the baseline solution. Case #2 was the same impact geometry and velocity, but the bar was 16mm long and again the nrbc were defined at the
far end of the rod. The forces computed for the nrbc at each time step, in case #2, were recorded and used as input forces into case #3. In case #3 the recorded forces were applied to a 16mm bar with an nrbc on the opposite end and the stresses in the interior element equivalent to the one used in case #1 were recorded and compared to the baseline data. This procedure was performed for several mesh densities and several levels of precision in the recorded forces with no discernible differences. A comparison of the two results is shown in Figure 8. The overall wave shape and energy content are the same. Only the high frequency components are modified by the technique. No dispersion effects due to this technique can be directly observed. The test cases were also run with various combinations of the damping constants a and b, but no better agreement was obtained than with a=1 and b=1.

Dynamic Relaxation

Another challenge in the solution of this problem is the presence of initial stress in the joint due to torque on the threads. To simulate this with DYNAC2D, a thermal load is applied to the sleeve to shrink it onto the rods. Dynamic relaxation is used to solve for the initial stress state due to the thermal load before initiation of the axial wave.

The dynamic relaxation technique is used to find a steady state solution using an explicit finite element code. The technique is based on removing the transient portion of the response so that only the steady state solution is left. This steady state solution is then
equivalent to the solution of the corresponding quasistatic problem [13]. Loads are
suddenly applied to the model and then transients are damped out until only the steady
state solution remains. The simplest scheme is to apply linear dampers to each node in
the model. This method converges toward the steady state solution although not
necessarily quickly [13]. When using non-linear material models, care must be taken that
the transients do not exceed appropriate limits, however most of the research presented
herein uses linear material models and this is not a concern. In the case where an elastic
/ plastic material model was used, this problem is discussed in more detail.

There have been many schemes devised to improve the convergence rate of the
dynamic relaxation technique with various degrees of success [13]. In the present
problem, a new scheme is used which is orders of magnitude more efficient than the
simple linear dashpot.

A New Dynamic Relaxation Scheme
The new scheme was developed through numerical experimentation with the
standard implementation. Instead of applying a constant damping rate to each node at
each time step, the new scheme starts with an undamped solution and then gradually
increases the damping rate until the solution converges. This can be written in the form:

\[ c_i = 1 - (1 - c_{i-1})k \]  \hspace{1cm} (19)
where $c_0$ is the damping rate, $k$ is a reduction constant and $i$ is the current iteration block of size $b$. In other words, every $b$ number of iterations, the damping rate is increased by a certain amount governed by $k$.

A Test Case

This relaxation technique has been extensively tested using a cylindrical thermal shrink fit problem and converges to the analytical solution very rapidly. A test mesh for the cylindrical shrink fit problem is shown in Figure 9. The problem consists of a tube with an inner radius of 1mm, an outer radius of 4mm and a hollow sleeve with an inner radius of 4mm and an outer radius of 7mm. The sleeve was defined with a coefficient of thermal expansion ($\alpha$) of 1.2E-05 $(\text{mm}^/{\circ C} \cdot \text{mm})$ and an applied temperature drop of 10$^\circ$ C. A comparison of the numerical solution using the dynamic relaxation technique and an analytical solution for the radial stress distribution through the tube and sleeve is shown in Figure 10. The analytical solution for the radial stress in a thick walled cylinder under internal and external pressure is given in [14] as

$$\sigma_r = \frac{P_i r_i^2 - P_o r_o^2 + \left(\frac{r_i r_o}{r}\right)^2 (P_o - P_i)}{r_o^2 - r_i^2}$$  \hspace{1cm} (20)

where $P_i$ and $P_o$ are the internal and external pressures, $r_i$ and $r_o$ are the inside and outside radii. For the test problem, the interface pressure is given by the radial interference $\delta$ [15]
\[ P = \frac{E \delta}{R} \left( \frac{(r_o^2 - R^2)(R^2 - r_e^2)}{2R^2(r_o^2 - r_e^2)} \right) \]  

(21)

where \( E \) is the modulus of elasticity and \( R \) is the radius of the interface. An approximate value of the radial interference is simply found from the coefficient of thermal expansion.

\[ \delta = R \alpha (\Delta T) \]  

(22)

Here, heat conduction between the sleeve and rod has been neglected as well the effect of the sleeve thickness. So for this problem, equation (20) reduces to

\[ \sigma_r = 8.213 \left( \frac{1}{r_i^2} - 1 \right) \]  

(23)

for the tube and

\[ \sigma_r = 3.733 \left( 1 - \frac{49}{r_i^2} \right) \]  

(24)

for the sleeve. When comparing the numerical and analytical solutions in Figure 10 note that the analytical solution assumes no displacement of the material interface (\( R=4.0 \)), while the converged solution accurately predicts the new radial distance to the interface.

While no blanket statement can be made about the application of this damping scheme to other problems, for the shrink fit problem it works extremely well. Figure 11 illustrates the time history of the radial stress in one element of the cylindrical shrink fit problem with \( k=0.9999 \) and \( b=100 \). As can be seen in Figure 11, the difference between the convergence behavior of the constant and variable damping rate is dramatic.
For all the research presented, this variable damping rate scheme was used to solve for the initial stresses in the threaded joint prior to the initiation of stress waves. By thermally shrinking the threaded sleeve onto the rods and using the dynamic relaxation technique the model simulates the effects of the initial torque on the joint.
Problem Formulation

The present study is organized into two major components:

1. A set of parametric studies of friction, contact algorithm, initial squeeze, attenuation, thread model, plasticity and input wave form using a relatively coarse mesh.

2. A detailed study with a typical set of parameters from above using a fine mesh.

Parametric Studies

Mesh

The parametric studies were carried out with the mesh illustrated in Figure 12. The mesh has three bodies, the left hand rod, the right hand rod, and the sleeve. The mesh statistics are listed below in Table 6. Note that the solution time step is a fraction (determined by the time step scale factor) of the time for a wave to propagate across the smallest element in the mesh.

<table>
<thead>
<tr>
<th>Table 6. Mesh Statistics for Parametric Study</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nodes</td>
</tr>
<tr>
<td>Elements</td>
</tr>
<tr>
<td>Number of Slidelines</td>
</tr>
<tr>
<td>Density (g/mm$^3$)</td>
</tr>
<tr>
<td>Young’s Modulus of Elasticity (N/mm$^2$)</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
</tr>
<tr>
<td>Model Solution Time (ms)</td>
</tr>
<tr>
<td>Dynamic Relaxation Reduction Constant</td>
</tr>
<tr>
<td>Dynamic Relaxation Convergence Criteria</td>
</tr>
<tr>
<td>Time Step Scale Factor</td>
</tr>
</tbody>
</table>
Initial Squeeze

To simulate the effects of an initial torque on the joint, the coupling was defined with a linear thermoelastic material model. The material model constants were input such that none of the properties were temperature dependent, but the coefficient of thermal expansion could be used with an applied temperature drop to shrink the sleeve onto the rods thereby preloading the threads. The dynamic relaxation technique discussed above was then used to solve for the initial stress state before the initiation of the wave. The original model used a standard linear thread pitch for the shrink fit. The linear thread pitch model, however, tends to load on the outermost threads as illustrated in Figures 13 and 14. A screw thread, however, loads highest on the innermost threads. Therefore, additional runs were made with a modified tapered thread pitch which tended to load on the first thread as illustrated in Figures 15 and 16.

The commercial rock drill used for this study has 3 N-m of available torque with normal drilling torque approximately 1/3 the maximum. This corresponds to about 27 MPa of interface pressure between the two rods and a range of 0-90 MPa, assuming a coefficient of friction in the threads of 0.1 and ignoring any tightening or loosening of the joint caused by previous blows. Through experimentation with the finite element model, temperature drops were defined that produce interface pressures in the range of study and are detailed in Table 7. Initial squeeze is the first parameter varied in this study.
Table 7. Temperature Drops Used and the Corresponding Interface Pressures

<table>
<thead>
<tr>
<th>Initial Squeeze Level</th>
<th>Sleeve Temp. Drop</th>
<th>Interface Pressure (linear thread)</th>
<th>Interface Pressure (tapered thread)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>na</td>
<td>5 MPa</td>
</tr>
<tr>
<td>3</td>
<td>30° C</td>
<td>na</td>
<td>7 MPa</td>
</tr>
<tr>
<td>5</td>
<td>50° C</td>
<td>0 MPa</td>
<td>35 MPa</td>
</tr>
<tr>
<td>7</td>
<td>70° C</td>
<td>10 MPa</td>
<td>na</td>
</tr>
<tr>
<td>9</td>
<td>90° C</td>
<td>35 MPa</td>
<td>100 MPa</td>
</tr>
<tr>
<td>0</td>
<td>120° C</td>
<td>70 MPa</td>
<td>na</td>
</tr>
</tbody>
</table>

**Friction**

The second parameter varied in the parametric study is the coefficient of friction along the slidelines. Four values of the coefficient of friction were studied based on the normal range reported for dry and lubricated hard steel on hard steel [16]. The friction levels studied are detailed in Table 8. With no friction, DYNA2D allows a choice of kinematic or kinetic slidelines. The kinematic slideline algorithm adjusts the position of nodes to prevent “slave” nodes from passing through a “master” segment, while the kinetic slideline algorithm applies nodal forces in proportion to the depth of interpenetration to try and maintain the contact condition. When modeling friction, only the kinetic algorithm is available [10].
Table 8. Coefficients of Friction and Contact Algorithms Used

<table>
<thead>
<tr>
<th>Friction/Contact Level</th>
<th>Contact Algorithm</th>
<th>Static Coefficient of Friction</th>
<th>Dynamic Coefficient of Friction</th>
</tr>
</thead>
<tbody>
<tr>
<td>03</td>
<td>Kinematic</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>05</td>
<td>Kinetic</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>14</td>
<td>Kinetic</td>
<td>0.01</td>
<td>0.01</td>
</tr>
<tr>
<td>24</td>
<td>Kinetic</td>
<td>0.10</td>
<td>0.10</td>
</tr>
<tr>
<td>34</td>
<td>Kinetic</td>
<td>0.50</td>
<td>0.30</td>
</tr>
</tbody>
</table>

Wave Form

The next parameter studied is the input wave form. The four waves studied are:

1. An idealized wave form.

2. A wave form from the solution of the piston impact problem.

3. A wave form from the experimental data of Fu and Land [5].

4. A square wave.

The waves are shown in Figure 17 except for the experimental wave which is illustrated in Figure 3. The idealized wave form consists of a finite rise time, a gradual linear decay and a finite fall time. The computed wave form was generated using the meshes shown in Figure 18 for the piston and striking bar with an axial impact velocity of 9.8 m/s. Again, a non-reflecting boundary was used as a mechanism to absorb and store the wave for use as input into the threaded joint model. The piston impacts onto what is called a striking bar or shank. This bar is solid at the impact point, has a raised section for the splines, which are used to impart rotation torque, and is hollow from approximately the middle of
the spline area down toward the threaded connection. The striking bar geometry at the threaded end is assumed to be the same as that of the rod ends studied. The experimental wave form is from the experiments of Fu and Land [5] and as with the idealized wave form it is assumed to be constant with respect to the radius of the bar.

**Attenuation**

Two runs were completed to study the effects of attenuation in the rods between the joints. Fairhurst [4] reported 1% attenuation per 3 m of rod, while Takaoka et al. [9] reported 3% energy loss per 3 m. With the non-reflecting boundary scheme being employed to reduce the computational effort, attenuation was modeled as a reduction in the nodal forces stored for the next pass. Since the reported levels of attenuation refer to energy loss and the implementation acts at the nodal force level, the multiplier is adjusted accordingly. The two runs model 1% and 2% attenuation per 3 m respectively. The experimental data of Fu and Land [5] was recorded with 3.6 m rods so the attenuation is 1.2% and 2.4% per rod. Energy is proportional to the square of stress so the attenuation factors applied to the nodal forces were 0.99398 and 0.98793 respectively.

**Plastic Deformation**

To check for energy loss and wave form changes due to localized plastic deformation, a run was completed with an elastic/plastic material model. One important difference in solving this problem is the prevention of plastic deformation during the dynamic relaxation portion of the analysis. The thermal loads must remain low enough to prevent any yielding.
The determination of a set of material properties for this analysis is not straightforward. The surfaces of drill rods and couplings are either carburized or induction hardened to a much higher hardness than the interior. The finite element model used here assumes constant material properties throughout the body, so the problem is to decide whether to use the surface properties, the interior properties, or something in between. The hardness of the surfaces of commercial rods and couplings is about 61 $R_C$ while that of the interior is around 30 $R_C$. This translates to roughly a 2,225 MPa yield stress on the surface and around 950 MPa (43% of the surface value) in the interior. For this particular run, a constant yield stress of 1,670 MPa (75% of the surface value) was selected for study. The choice is somewhat arbitrary, but the highest stresses tend to occur near the surface and the surface properties decay rapidly into the interior properties away from the surface. The tangent modulus was simply selected as 0.01 times the elastic modulus.

**Run Coding**

The various runs made in this study are coded for each parameter. The run code consists of five parts. The initial wave form code is followed by an option code, a friction/contact code, then an initial squeeze code and finally the number of passes through the threaded connection.
Table 9. Summary of Input Codes

<table>
<thead>
<tr>
<th>Input Wave Form</th>
<th>Special</th>
<th>Friction/Contact</th>
<th>Initial Squeeze</th>
<th>Pass</th>
</tr>
</thead>
<tbody>
<tr>
<td>i (ideal)</td>
<td>(blank) standard</td>
<td>03</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>a (computed)</td>
<td>attenuation</td>
<td>05</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>f (experimental)</td>
<td>tapered thread</td>
<td>14</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>p (square)</td>
<td>elastic / plastic</td>
<td>24</td>
<td>5</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>34</td>
<td>7</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>44</td>
<td>9</td>
<td>6</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5</td>
<td>0</td>
<td></td>
</tr>
</tbody>
</table>

(See Table 8) (See Table 7)

Run Matrix

The initial matrix of runs consists of idealized wave forms. There are twelve sets of runs in the first matrix as follows:

i035, i037, i039  i055, i057, i059  i245,i247,i249  i345,i347,i349

The cases with level 1 friction were not completed after the results of the friction level 2 cases compared to the frictionless models made it clear that solving for such a low value of friction would not provide any substantial new information. After reviewing the results of this first matrix (discussed in the results chapter) the cases run for the computed wave form were narrowed down to the following six that were considered most beneficial:

a035, a037, a039  a245, a247,a249

Next, the attenuation cases were run to compare results with damping losses in the drill steel. These two runs are coded:
af240 (computed wave form, level 2 friction, level 0 squeeze)

af241 (computed wave form, level 2 friction, level 10 squeeze)

A final set of runs was completed with the tapered thread model discussed above. These included:

it057, at243, at245, it245, ap245

**Detailed Study**

After the initial parametric study was completed, a more detailed run was completed to look at the effect of mesh refinement and to establish a better detailed picture of the wave interaction with the thread contact surfaces. The mesh used is partially illustrated in Figure 19 and the run parameters are detailed in Table 10.

<table>
<thead>
<tr>
<th>Table 10. Parameters for Detailed Study</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nodes</td>
</tr>
<tr>
<td>Elements</td>
</tr>
<tr>
<td>Friction</td>
</tr>
<tr>
<td>Thread Model</td>
</tr>
<tr>
<td>Initial Squeeze</td>
</tr>
<tr>
<td>Attenuation</td>
</tr>
<tr>
<td>Input Wave Form</td>
</tr>
<tr>
<td>Plasticity</td>
</tr>
</tbody>
</table>
Results

Parametric Study

Several trends were established with the cases run in the parametric study as outlined above. Loss of energy in the pulse versus the number of passes through the threaded joint is the primary phenomenon of interest. For this purpose, energy content of the wave is defined as the productive energy for breaking rock under the bit. This is considered to be the energy contained in the first contiguous compressive pulse. Energies in radial modes, in pulses with tensile stresses, or in lagging pulses do not significantly contribute to the breakage of rock and are therefore not included in the calculation of productive energy, hereafter simply referred to as the energy. The results presented below calculate the energy in the compressive pulse based on the axial strain in one selected element (the gage element), thereby assuming a constant radial distribution of energy. This assumption appears valid upon close examination of the results.

Energy loss versus number of passes is presented in two formats. First, energy is normalized with respect to the input energy. Second, energy is normalized by the energy after one pass and plotted as that ratio. The latter format was chosen for comparisons with experimental data which is always referenced to the energy after the first joint due to measurement limitations inherent in the physical design of the hammers.

Several other less precise measures can be used to compare the wave forms after successive passes to the experimental data. One is the peak stress in the wave. This is
simply the maximum axial stress seen by the gage element and is a useful indicator of the maximum rock hardness that can be broken by the wave for a given bit geometry. This is because if the maximum bit loading does not exceed the rock fracture point, the energy content of the wave is reflected back up the drill string without producing work.

Another set of qualitative measures is the blow duration and average initial slope. These are useful in comparing the wave shape changes from one set of conditions to another. These measures are not precise because high frequency components in the wave form tend to make measurement subject to human error. The average initial slope is the initial rate of change in stress per unit time expressed as MPa per ms. In practice it is calculated as the peak stress divided by the time from the arrival of the wave front until the instant of the peak stress. The arrival time can not always be precisely determined because of the presence of high frequency noise. In these results, the initial time was determined manually by inspection of the axial stress versus time curve. The blow duration is simply the time of the first contiguous compressive pulse, or the same time as for the energy calculation. When calculating energy, however, the stress is squared and an imprecise blow duration has very little effect on the energy calculation. However, comparisons of blow duration and initial slope numbers must be made with care due to the influence of the high frequency components discussed above.

Friction

Typical results for the variation of the coefficient of friction at the thread contact zones are shown in Figures 20 and 21. The results in Figure 20 are normalized the energy
in the first rod and in Figure 21 are normalized to the energy in the input pulse (326 J).

As illustrated, the results are virtually the same for frictionless contact and $\mu=0.1$ while $\mu=0.5$ results in a slightly lower overall transmission efficiency. As expected, the highest friction corresponds to the least efficient energy transmission. The total energy lost in the six joints, however, only varies from 35.6% - 37.6% over the range of friction coefficients. One possibility is that the influence of thread friction might be understated in this model because the axisymmetric assumption precludes accounting for any tendency of torsional forces to keep the threads in contact.

**Initial Squeeze**

Typical results for the effect of initial squeeze on the transfer of energy are plotted in Figures 22 and 23 for an idealized wave and in Figures 24 and 25 for a computed wave. As expected, a higher initial squeeze on the thread translated into more efficient energy transmission through the joint. In this case the total variation in energy lost is more significant. For the idealized wave the total energy lost in the six joints varies from 29.9% - 35.6% and for the computed wave the range is 15% - 24.5%. The range of energy lost as a function of initial squeeze is similar for the two wave forms. However the idealized wave was significantly less efficient than the computed wave.

The variation in the peak axial stress nondimensionalized with respect to the peak axial stress of the input wave (250 MPa for the idealized wave and 300 MPa for the computed wave) is shown in Figure 26. As mentioned above, this is an imprecise measure primarily useful for establishing qualitative trends. Note that the peak axial
stress drops anywhere from 5% - 30% over six joints depending on the model. As with the total energy content, peak stress is reduced the most in the models with the lowest initial squeeze. In this case however, it is the computed wave form which loses peak stress faster than the idealized wave form.

The variation in average initial slope is illustrated in Figure 27. Because of the precipitous drop in the average initial slope, Figure 27 is plotted on a semi-logarithmic scale. Each model’s slope is normalized to the input slope for that particular model. It is interesting to note that, although the slope of the computed wave form drops approximately 86% in the first pass and the initial slope of the idealized wave form drops 70%, the computed wave started with a 85% higher slope than the idealized wave so after the first pass, the slopes of the two wave forms are quite similar. The initial squeeze does not seem to have a discernible effect on the change in slope within the range studied.

The change in blow duration after each pass is shown in Figure 28. The total length of the blow increases with each pass with the few exceptions attributed to the imprecise nature of the measurement in the presence of high frequency components. The lower initial squeeze tended to have a greater lengthening effect than the higher initial squeeze. The lengthening of the idealized wave form ranged from 87% to 130% while that of the computed wave form ranged from 71% to 77%. Another interesting feature to note is that for both input waves, the minimum lengthening is associated with the middle level of initial squeeze.
The wave forms after 1, 3 and 5 joints are plotted in Figures 29, 30 and 31 for the idealized wave at initial squeeze levels of 0 MPa, 10 MPa and 35 MPa respectively. Besides observing the phenomena already discussed, it is clear from these figures that a portion of the energy is being moved to a tensile “tail” of the wave by internal reflections in each joint.

**Thread Model**

It is difficult to directly compare the efficiency of energy transmission with the standard thread model to that of the tapered thread model because of the influence of initial squeeze. Although the standard model produces interface pressures linearly related to the applied temperature drop, the tapered model has nonlinear characteristics as successive threads engage while the temperature drop increases. Therefore the comparisons are for similar but not exact interface pressures. Figures 32 and 33 show the energy loss from two models with different thread geometry. The overall loss of energy over six joints is 23% in the tapered thread model and 28% in the linear thread model. When the wave forms are examined in Figures 34 and 35, several differences are evident. The wave forms from the linear thread model (Figure 34) have a much higher occurrence of high frequency components than the wave forms from the tapered thread model. The peak stresses are higher in the tapered thread model which accounts for the higher energy, because the blow durations are the same with both models.
**Input Wave Form**

Differences between the idealized and computed input wave forms are partially discussed above along with the initial squeeze effect. Figure 36 illustrates the energy loss for the various input waves. Note that the energy for the experimental wave before the first joint is not available. The results from the experimental and computed waves are in close agreement while those for the idealized and square wave are in good agreement with the main difference being that the square wave loses more energy in the first joint. As seen above the computed and experimental waves are more efficient than the idealized and square waves. Figure 37 illustrates the square wave after 1, 3 and 5 joints. The square wave is modified significantly in the first joint and then follows the normal pattern in the successive joints. Note that after passing through the first joint the initial slopes and blow durations of the square wave are indistinguishable from those of the conventional wave forms.

**Attenuation**

Adding attenuation to any of the models reduced the stress and energy levels after each pass proportionally. No effect was observed on blow duration or average initial slope. No significant new information was obtained from these runs.

**Elastic / Plastic Material Model**

For the yield stress chosen in this model, there was no difference from the equivalent purely elastic model. There was no difference in energy content or wave shape, so it can be assumed that no yield occurred at this level. Modeling the gradient in
material properties from the surface to the interior might provide some additional insight. However, with the highest stresses tending to occur at the surface and with a refined model increasing the yield stress at the surface by one third, it is unlikely to be a productive exercise.

**Comparison with Experimental Data**

Figures 38 - 41 illustrate the results of typical models compared with the experimental results of Fu and Land [5]. Also, the wave forms in Figures 29-31 and 34-35 can be compared with the experimental wave forms in Figure 3. In these figures clearly the general trend of changes to the experimental wave form (in peak stress, initial slope, and duration) is replicated in the finite element models. One distinct difference between the numerical and experiment results is the compressive secondary pulse in the experimental data versus the tensile secondary pulse in the numerical data.

In Figure 38 the energy loss versus number of joints is plotted and all of the models have lower energy losses than the experimental results. The energy loss curve for the computed input wave seems to have a characteristic shape which is more similar to the experimental data than the curve for the idealized input wave. The idealized wave, however, has an overall energy loss closer to the experimental data.

Figure 39 shows the change in average initial slope versus the number of joints. Here, all the models compare reasonably well with the experimental data. In all models the slope after one joint is higher than the experimental data. However after three and
five joints, the slopes are quite similar. As mentioned above, although the computed wave and idealized wave have initial slopes of 7,500 MPa/ms and 13,900 MPa/ms respectively, after one joint they both have initial slopes in the range of 2,000 MPa/ms compared to the experimental result of 1,621 MPa/ms.

Figure 40 illustrates the change in blow duration versus the number of joints. There is almost identical agreement with the computed wave case for an initial squeeze of 10 MPa and good agreement with the trend of most of the computed wave models studied. The blow duration is the measure that most closely matches the experimental data for the parameters studied.

Figure 41 illustrates the peak stress normalized to the peak stress in the first rod versus the number of joints. The data is scattered, but the experimental data falls in the range of the idealized wave model and below the range of the computed wave model. This measure tends to have the worst agreement with experimental data across the models studied.

**Detailed Study**

The mesh used in the detailed study required over 200 hours of CPU time to solve on an SGI R8000 64 bit RISC computer. For this reason, only one case was studied with this mesh. The results, however, are useful in interpreting the results with the coarse mesh.
Figure 42 illustrates the loss of energy versus the number of joints for the fine mesh and the coarse mesh at the same initial squeeze. Both of these cases were run with no friction and 100 MPa of interface pressure. The energy loss in the first joint is significantly higher with the fine mesh than that with the coarse mesh. After the first joint, however, the energy loss is about the same as with the standard model until the fifth and sixth joints where the energy loss is lower than that with the coarse mesh. The net result is an overall energy loss very close for the two models, but with the higher density mesh having a different shape energy loss curve. This curve more closely matches the shape of the experimental data but still the energy lost is not as high as the experimental data.

Figures 43 - 45 compare the wave forms after one, three and five joints for the two models. In Figure 43 the initial slopes of the waves after one joint are quite similar. The peak stress is higher and occurs later with the fine mesh. The blow duration is increased significantly in the fine mesh model after one joint, and the tensile “tail” is much more pronounced in the fine model. Overall, the two waves differ in many small qualitative aspects.

In Figure 44, the comparison after three joints shows the waves from the two models to be somewhat more similar than after one joint. They still have essentially the same initial slope and the peak stresses and durations are similar as well. As in Figure 43, the fine mesh has more high frequency components than the coarse mesh. Also, the tensile portion of the wave is again more prevalent in the fine model.
Figure 45 shows the waves from the two models after five joints. Here, the slopes are again nearly identical along with the peak stress and duration. So after five joints, the waves from the two meshes are very similar even though the intermediate results differ.

Figure 46 illustrates the axial stress in the third joint at 0.08 ms after the wave enters the joint. This corresponds to the time at which about one half of the initial slope has entered the joint. Figures 47 and 48 show the axial stress along a constant radius and a constant axial dimension at time 0.08 ms. These figures provide a view of a typical stress distribution pattern in the joint during the passage of the wave front.
Conclusions

The parametric study provided a strong basis for determining which parameters are important for a useful solution to the axisymmetric drill rod coupling model. The observed effects of the parameters studied are summarized below in Table 11. Clearly the initial squeeze plays a key role in the transmission properties of the rock drill joint and must be modeled with care. In addition, the shape of the wave has a significant effect on the energy transmission efficiency. The results in Figure 36 show that computing an input wave with a two body axisymmetric mesh is sufficiently accurate since results with the computed wave were nearly the same as those with an experimentally recorded wave form input into the model. The idealized wave, however, is probably not a sufficient approximation for drawing conclusions about possible design changes to the joint or piston geometry.

Conversely, friction and plasticity do not seem to play significant enough roles to warrant the extra computational effort. In normal practice they can probably be neglected. It is less clear whether to incorporate a tapered thread model but it should probably be included since it does affect the results and does not impose a large computational penalty.

Refining the mesh changed the shape of the energy loss curve although the overall level of loss for six joints remained essentially constant. The refined mesh used is too fine for practical application, but the coarse mesh could be refined in the thread contact
area. As computer hardware continues to improve, the fine mesh should quickly become feasible.

One interesting item to note is that none of the variables tested had a significant impact on the slope of the wave front. This seems to be strictly governed by the geometry of the joint.

**Table 11. Summary of the Effects of the Parameters Studied**

<table>
<thead>
<tr>
<th></th>
<th>Loss of Energy</th>
<th>Reduction of Peak Stress</th>
<th>Increase in Blow Duration</th>
<th>Lowering of Initial Slope</th>
</tr>
</thead>
<tbody>
<tr>
<td>Increasing Initial Squeeze</td>
<td>Reduces losses</td>
<td>Reduces losses</td>
<td>Reduces lengthening</td>
<td>No effect</td>
</tr>
<tr>
<td>Increasing Friction</td>
<td>Small increase in losses</td>
<td>Small increase in losses</td>
<td>No effect</td>
<td>No effect</td>
</tr>
<tr>
<td>Using Idealized Input Wave</td>
<td>Increases losses</td>
<td>Increases losses</td>
<td>Increases lengthening</td>
<td>Less drop in first joint, then no effect</td>
</tr>
<tr>
<td>Adding Attenuation</td>
<td>Increases losses</td>
<td>Increases losses</td>
<td>No effect</td>
<td>No effect</td>
</tr>
<tr>
<td>Modeling Plasticity</td>
<td>No effect</td>
<td>No effect</td>
<td>No effect</td>
<td>No effect</td>
</tr>
<tr>
<td>Using Tapered Thread Model</td>
<td>Reduces losses</td>
<td>Reduces losses</td>
<td>No effect</td>
<td>No effect</td>
</tr>
<tr>
<td>Refining Mesh</td>
<td>Increases losses in first joint</td>
<td>No effect</td>
<td>Increased in first joint</td>
<td>No effect</td>
</tr>
</tbody>
</table>

The axisymmetric finite element solution method presented is a sufficiently accurate and efficient technique to provide previously unavailable insight into the interaction of wave shape and joint design with the transmission efficiency of a rock drilling system. This technique can be applied to practical rock drill design problems for evaluating the effect of a proposed design change on the joint transmission properties.
Figure 2. Cross Section of a Typical Rock Drill Threaded Joint
Figure 3. Experimental Results from Fu and Land [5]
Figure 4. Numerical Solution Based on the Algorithm of Fu and Paul [6]
Figure 5. Comparison of Various Experimental and Analytical Results
Figure 6. Reflected to Incident Energy Ratio of a P-Wave as a Function of Incident Angle
Figure 7. Non-reflecting Boundary Test Cases
Figure 10. Comparison of Analytical and Numerical Solutions to the Shrink Fit Problem.
Figure 11. Convergence Behavior of the New DR Scheme and the Standard DYNA2D Scheme
Figure 13. Contour Plot of the Initial Axial Stress State with the Linear Thread Model
Figure 14. Axial Stress at r=12.0 mm for the Initial Stress State with the Linear Thread Model
Note: Radial dimension amplified for clarity

Figure 15. Contour Plot of the Initial Axial Stress State with the Tapered Thread Model
Figure 17. Input Wave Forms Used in the Parametric Study
Figure 18. Mesh Used to Solve for the Computed Wave Form
Figure 20. Effect of Friction on the Energy Loss versus Number of Joints
Figure 21. Effect of Friction on the Energy Loss versus Number of Joints
Figure 22. Effect of Initial Squeeze on the Energy Loss versus Number of Joints
Figure 23. Effect of Initial Squeeze on the Energy Loss versus Number of Joints
Figure 24. Effect of Initial Squeeze on the Energy Loss versus Number of Joints
Figure 25. Effect of Initial Squeeze on the Energy Loss versus Number of Joints
Figure 26. Change in Peak Axial Stress versus Number of Joints
Figure 27. Change in Average Initial Slope versus Number of Joints
Figure 28. Change in Blow Duration versus Number of Joints
Figure 30. Wave Forms After 1, 3 and 5 Joints - 10 MPa Squeeze
Figure 31. Wave Forms After 1, 3 and 5 Joints - 35 MPa Squeeze
Figure 32. Effect of Thread Model on the Energy Loss versus Number of Joints
Figure 33. Effect of Thread Model on the Energy Loss versus Number of Joints
Figure 34. Wave Forms After 1, 3, and 5 Joints - Linear Thread Model
Figure 35. Wave Forms After 1, 3, and 5 Joints - Tapered Thread Model
Figure 44. Comparison of Waves After Three Joints
Figure 36. Effect of Wave Form on the Energy Loss versus Number of Joints
Figure 37. Wave Forms After 1, 3, and 5 Joints - Square Input Wave
Figure 38. Comparison of Energy Loss versus Number of Joints with Experimental Data
Figure 39. Comparison of Average Initial Slope with Experimental Data
Figure 40. Comparison of Blow Duration with Experimental Data
Figure 41. Comparison of Normalized Peak Stress with Experimental Data
No Friction, 100 MPa Initial Squeeze

Figure 42. Comparison of Energy Loss versus Number of Joints for Two Meshes
Figure 46. Contour Plot of Axial Stress in the Third Joint at t=0.08ms
Figure 47. Axial Stress in the Third Joint at $r=12.0$ mm and $t=0.08$ ms
Figure 48. Axial Stress in the Third Joint at z=195 mm and t=0.08 ms
References


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