Modeling of the reciprocating, pneumatic impact hammer

William Allan Bloxsom
University of Nevada, Las Vegas

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MODELING OF THE RECIPROCATING, PNEUMATIC IMPACT HAMMER

by

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Bachelor of Science in Industrial Education
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A dissertation submitted in partial fulfillment of the requirements for the

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Graduate College
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August 2003

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ABSTRACT

Modeling of the Reciprocating, Pneumatic Impact Hammer

by

William A. Bloxsom

Dr. Douglas D. Reynolds, Examination Committee Chair
Professor of Mechanical Engineering
University of Nevada, Las Vegas

The motion of the reciprocating, pneumatic impact tool, the air-driven piston inside the tool, the chisel mounted into the tool, and the specifically designed single degree-of-freedom spring-damper-mass test fixture were modeled with a MATLAB computer code. The three tools modeled and evaluated experimentally were the Ingersoll-Rand IR-121 impact tool, the Sears Craftsman Medium Duty impact tool, and the ATSCO No.2 impact tool.

The computer model of the accelerations of the tool and the test fixture mass were compared to the experimental data obtained on the actual test fixture by the three tools modeled. The correlation between the experimental data and the modeled data is high in both the time domain and the frequency domain.

The computer model is modified to include an air spring vibration attenuation mechanism. The model is used to tune the attenuation device as well as predict the force produced by the chisel into the work piece and the vibration through the tool into the hand-arm of the operator. The computer program allows other attenuation methods to be modeled and evaluated prior to actual construction.
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<tr>
<th>SYMBOL</th>
<th>DESCRIPTION</th>
<th>UNIT</th>
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<tr>
<td>a₁</td>
<td>acceleration of the piston</td>
<td>m/s²</td>
</tr>
<tr>
<td>a₂</td>
<td>acceleration of the tool</td>
<td>m/s²</td>
</tr>
<tr>
<td>a₃</td>
<td>acceleration of the chisel</td>
<td>m/s²</td>
</tr>
<tr>
<td>a₄</td>
<td>acceleration of the single degree-of-freedom mass</td>
<td>m/s²</td>
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<tr>
<td>F_CM</td>
<td>co-linear force between chisel and tool</td>
<td>N</td>
</tr>
<tr>
<td>F_CT</td>
<td>co-linear force between chisel and mass</td>
<td>N</td>
</tr>
<tr>
<td>F_OP</td>
<td>axial force applied by operator to the tool</td>
<td>N</td>
</tr>
<tr>
<td>F_P</td>
<td>Force from air pressure over area of surface</td>
<td>N</td>
</tr>
<tr>
<td>g</td>
<td>gravity constant</td>
<td>m/s²</td>
</tr>
<tr>
<td>m₁</td>
<td>mass of the piston</td>
<td>kg</td>
</tr>
<tr>
<td>m₂</td>
<td>mass of the tool</td>
<td>kg</td>
</tr>
<tr>
<td>m₃</td>
<td>mass of the chisel</td>
<td>kg</td>
</tr>
<tr>
<td>m₄</td>
<td>mass of the single degree-of-freedom mass</td>
<td>kg</td>
</tr>
<tr>
<td>v₁</td>
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<td>m/s</td>
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<tr>
<td>v₂</td>
<td>velocity of the tool</td>
<td>m/s</td>
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<td>v₂_new</td>
<td>post-impact velocity of the tool in momentum equations</td>
<td>m/s</td>
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<td>v₂_old</td>
<td>pre-impact velocity of the tool in momentum equations</td>
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<td>$v_3$</td>
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<td>m/s</td>
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<td>$v_{3\text{-old}}$</td>
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<td>m/s</td>
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<td>$v_4$</td>
<td>velocity of the single degree-of-freedom mass</td>
<td>m/s</td>
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<tr>
<td>$v_{4\text{-new}}$</td>
<td>post-impact velocity of the s-d-o-f mass in momentum eq.</td>
<td>m/s</td>
</tr>
<tr>
<td>$v_{4\text{-old}}$</td>
<td>pre-impact velocity of the s-d-o-f mass in momentum eq.</td>
<td>m/s</td>
</tr>
<tr>
<td>$x_1$</td>
<td>displacement of the piston from zero position</td>
<td>m</td>
</tr>
<tr>
<td>$x_2$</td>
<td>displacement of the tool from zero position</td>
<td>m</td>
</tr>
<tr>
<td>$x_3$</td>
<td>displacement of the chisel from zero position</td>
<td>m</td>
</tr>
<tr>
<td>$x_4$</td>
<td>displacement of the s-d-o-f mass from zero position</td>
<td>m</td>
</tr>
<tr>
<td>$\dot{x}_1$</td>
<td>velocity of the piston</td>
<td>m/s</td>
</tr>
<tr>
<td>$\dot{x}_2$</td>
<td>velocity of the piston</td>
<td>m/s</td>
</tr>
<tr>
<td>$\dot{x}_3$</td>
<td>velocity of the piston</td>
<td>m/s</td>
</tr>
<tr>
<td>$\dot{x}_4$</td>
<td>velocity of the piston</td>
<td>m/s</td>
</tr>
<tr>
<td>$\ddot{x}_1$</td>
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<td>m/s²</td>
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<tr>
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<td>acceleration of the piston</td>
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<td>$\ddot{x}_4$</td>
<td>acceleration of the piston</td>
<td>m/s²</td>
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ACKNOWLEDGEMENTS

It is to my wife, Barbara, that I am the most indebted. Without her support, encouragement, and willingness to endure long separations this effort might have fallen short. Her ruthlessness with a red pen in editing this document elevated my prose to its current level. May this represent the end of our being apart.

My daughters, Katy and Ashley, have spent long periods of time without their father. I appreciate their patience and understanding.

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CHAPTER 1

INTRODUCTION

Health and Safety

The pneumatic impact tool has many industrial and commercial applications. It is available in a wide variety of sizes and shapes. There are small and inexpensive models that are readily available for home shop use. There are sizes appropriate for industrial applications such as pin driving and bearing insertion during manufacturing assembly. Still other industrial sizes and configurations are used in metal processing for the removal of flashing on a casting and the shaping of large castings such as ship propellers. The larger tools are used to bore holes in rock at mining operations. The most readily visible application is the use on highway construction projects to break up asphalt or concrete or to compact soil or substrate prior to overlaying a paving material. All of these tools have a common element: compressed air is used to force a chisel or some other device to impact a surface to produce the desired end result.

The pneumatic impact tool has been in use for more than a century. The shape of the tool has evolved. The original massive size and bulk has been reduced. The use of new and better materials has made them lighter, smaller, and more durable. The breadth of application has increased. In some
applications, electric impact devices have replaced their pneumatic counterparts. The use of pneumatic tools, because of their inexpensive power source and rugged design, has remained the standard in many industries. Through this long evolution, the basic principle of operation of the pneumatic impact tool has remained virtually the same since their inception.

More than eighty years ago, it was found that there was a link between the long term use of vibrating hand-held tools and the disabling, progressive and irreversible circulatory and nerve degeneration maladies in the digits and limbs of the tool operators. The first power tools to provide this type of continuous, high amplitude vibration were the pneumatic tools typically used in the mining and metalworking industries. Among the diseases associated with exposure to high levels of vibration are hand-arm vibration syndrome (white finger disease or Raynard's Syndrome), carpal tunnel syndrome, tendonitis, tenosynovitis, osteoarthritis, decalcification, de Quervain's disease, and Dupuytren's contracture [1].

In the decades following this initial association, medical research has confirmed the link [2]. The occupational safety organizations have used the prevailing medical information to establish guidelines that set maximum limits on the daily duration of human exposure to various levels of vibration. The amplitude of vibration of the tools can be quantified. The limits can be translated to safe use of the tools by tool type. The level of vibration attributable to reciprocating, pneumatic hand-held impact tools is high.
The American Conference of Governmental Industrial Hygienists (ACGIH) have established a time limit for levels of vibration exposure (Table 1). The times do not reflect the continuous exposure but rather the aggregate daily use permitted [3].

In June of 2002 the European Union issued a new Human Vibration Directive [4]. The scope of the Directive is to establish "minimum requirements for the protection of workers from risks to their health and safety arising or likely to arise from exposure to mechanical vibration." The Directive indicates the daily hand-arm "exposure limit value standardized to an eight-hour reference period shall be 5 m/s² (square root of the sum of the squares of the frequency weighted acceleration values from the orthogonal axes as per ISO standard 5349)." The European Union also directs employers to seek out "replacement equipment designed to reduce the exposure to mechanical vibration."

Since its discovery there have been many attempts to attenuate the vibration of the tool handle to reduce the amplitude of the vibration transmitted to the

<table>
<thead>
<tr>
<th>Maximum Time Exposure</th>
<th>Maximum Vibration Amplitude</th>
</tr>
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<tbody>
<tr>
<td>Less than 8 hours</td>
<td>4 m/s²</td>
</tr>
<tr>
<td>Less than 4 hours</td>
<td>6 m/s²</td>
</tr>
<tr>
<td>Less than 2 hours</td>
<td>8 m/s²</td>
</tr>
<tr>
<td>Less than 1 hour</td>
<td>12 m/s²</td>
</tr>
</tbody>
</table>

Table 1: ACGIH recommended exposure limits for hand vibration
operator. Most of the methods have employed devices that are added to the
tool. Most have been unsuccessful. The few attempts to modify the operation of
the tool itself have either not been effective in reducing vibration or have reduced
tool handle vibration at the expense of sacrificing tool efficiency. Many devices
have been granted patents by the United States Patent Office [5-34]. Despite
their ability to be patented, the ability of most of these devices to attenuate
handle vibration is suspect. The primary evidentiary factor is the fact that not
many of those devices are actually in use.

There have been several technical papers written on the subject of hand-arm
vibration in general and with specific reference to various types of tools. Both
active and passive vibration control has been addressed [35-63].

In part, the employee is partially responsible. There are many jobs that
commonly use a hand-held pneumatic impact device where the employee is
either paid by the piece-completed or expected to maintain a pre-determined
level of production. The degenerative diseases associated with the exposure to
hand-arm vibration have no immediate symptoms [64-72]. In fact, some
conditions will not have progressed to a disabling state for decades after long-
term use has commenced. The worker is willing to trade speed and efficiency
today for something that is only a concept tomorrow. Any modification to the
exterior of the tools that does work is frequently treated as an impediment to the
effective operation of the tool and damaged, destroyed, or discarded in an
attempt to increase production. Tools in which the vibration has been attenuated
by decreasing the effectiveness are quickly discarded. Cherng and Chen [73]
report that a rivet hammer with active damping did not receive commercial acceptance because the hammer “has very light hitting force and it bounced too much.”

The environment for the tool is another difficulty for a practical solution. These tools typically work in mines or coarse metal working environments. The tools are routinely dropped. They need to be resistant to distortion or breaking caused by those impacts. The tools must be impervious to the dirt and metal filings that are abundant in the air and surfaces in which they work and lay idle. As a result, the pneumatic reciprocating tool is sturdy, heavy, and unadorned with ancillary items, much as it was one hundred years ago.

Modeling

The modeling of the pneumatic tools, while unique, is straightforward and only moderately complex. The single degree-of-freedom model has multiple components that constantly move independently until there is an impact to alter their velocities and directions of travel. The impacts must be modeled without regard to any preconceived sequence but rather based on the positions of the components at any time increment. The modeled chisel has periodic contact with the piston, the tool, and the single degree-of-freedom moveable test fixture, the relative positions and velocities of which must be tested at each time step for applicable impact analysis. The role of the compressed air power source is to clearly propel the piston downward into the chisel but also to assure the piston's
return to the top of the cylinder and to prevent an impact with the top of the
cylinder.

The results of the modeling are revealing. The model provides information
about the acceleration of the chisel that is impractical and almost impossible to
glean experimentally. The correlation between the moving entities and their
impacts to the various levels of acceleration peaks in the motion of the tool is
exposed. The sources of the vibration in the tool that are directed to the operator
are apparent in the analysis of the modeled output.

The knowledge provided by the computer model is the first step in creating a
viable and effective method to attenuate the vibration transmitted to the operator.
Any method contemplated can be incorporated into the program to gauge its
effectiveness at reducing the peak levels of vibration as well as the effect it has
on the efficiency of the cutting chisel.

Several published papers on vibration and impacts were reviewed [74-75].
Many book authors that dedicated sections of their books to the discussion of the
nature of impacts and their effect on oscillatory motion [76-85]. Typically, the
impacts were a spatially consistent termination of the unforced vibration cycle
that would cause the oscillating body to reverse direction and continue on its
harmonic journey until the next collision. The impacts were addressed with the
convolution theorem or Laplace transforms. The remainder of the unforced cycle
was initiated as an initial value problem. No references were located that
addressed the rapid sequence of impacts occurring almost without regard to the
positions or velocities of the impacting bodies.
The reciprocating impact tool has numerous impacts involving many components. The impacts are neither spatially consistent nor do the components possess intuitively predictable velocities during those impacts. The impacts occur with great frequency. There is a collision between different parts of the tool at virtually every time step. Additionally, there are spatial requirements that the tool must establish for each set of initial loading conditions in order to achieve steady state. In order to be available for the impacts, the moving parts must remain in or return to spatial "windows" during the course of piston impact cycle.

In addressing the resolution of this modeling, attempts were made to construct algorithms based on the convolution theorem and Laplace transforms. Numerical methods such as Runge-Kutta were tried. The problem was also converted from the time-domain to the frequency-domain and analyzed with a Fourier analysis. None of these methods were successful. The discontinuities in the displacement, the velocities, and the accelerations of the different mass components of the system made the numerical evaluation impossible. The rapid succession of impacts, often with only a single time increment between collisions, provided little time for an initial value response to put the oscillating body into any harmonic motion. The frequency analysis drove the system in such a way that the changing frequency function was actually providing too much frequency information and dominated the solution.

The method that produced a model that appears to most closely represent the experimentally obtainable acceleration signals is explained in Chapter 3. The time domain test acceleration values captured were in the frequency range of 0
to 800 Hz with data recorded at the rate of 1600 lines per second. The time
domain model used discrete time steps of 0.0005 seconds or 2000 time steps
per second. This equates to a sampling rate of 2000 Hz. The Nyquist frequency
is 1000 Hz with the possibility of questionable value in the highest 20% due to
aliasing [86]. The effective result is a frequency domain correlation of the
modeled time signal in a range of 0 to 800 Hz.

The International Standards Organization (ISO) has several documents that
directly address the measurement of hand-arm vibration [87-95]. One of those
standards is ISO 5349 “Mechanical vibration – Guidelines for the measurement
and the assessment of human exposure to hand-transmitted vibration” that
establishes a method for evaluating the vibration in the handle of a powered
hand tool [87]. The standard uses a weighted scale to evaluate either the octave
band or third octave band root-mean-square (rms) acceleration values (or
acceleration levels) in the frequency range of 5 Hz to 1425 Hz (6.3 Hz to 1250 Hz
center frequency third octave bands).

The weighting formula is such that acceleration values above 16 Hz are
attenuated at the rate of 6 dB per octave. If rms acceleration values are used,
the third octave band accelerations above the 16 Hz center frequency third
octave band are multiplied by scalar factors to reduce their contribution to the
computed total acceleration. For example, acceleration magnitude in the third
octave band with a center frequency of 20 Hz is multiplied by 0.20 while the
acceleration magnitude in the third octave band with a center frequency of 630
Hz is multiplied by 0.025. The higher frequency third octave bands (with 800 Hz,
1000 Hz, and 1250 Hz center frequencies) are multiplied by even smaller scalars.

Although the correlation of the frequency information for the model and the test data is available only in the range of third octave bands with center frequencies of 6.3 Hz to 800 Hz, and will not comply with the guidelines of ISO 5349, the frequency data within this text is valid for comparison purposes. The weighted acceleration values are computed in accordance with ISO 5349 with the exception that the 800 Hz, 1000 Hz, and 1250 Hz. third octave frequency bands are not used. The frequency range used is 5.7 Hz. to 708 Hz.

The Text

Chapter 2 addresses the functioning of pneumatic impact tools in general and the specific operation of the three models used. A description of the components within the tool is provided.

Chapter 3 describes the test fixture used. The instrumentation and testing is discussed.

Chapter 4 describes in detail the computer code that models the operation of the impact tools. The cycling of the piston and the associated air pressure profiles that control it are presented. The permutations of all the impacts are presented with the associated free body diagrams. The mathematics that creates the digital model is discussed.

Chapter 5 presents a general discussion of the comparison between the data collected during the tests and the data generated by the computer model.
Specific input values and comparisons are provided for each of the three impact tools considered.

Chapter 6 is dedicated to the attenuation of the vibration in the model with a description of the methods considered and the effect of those methods.

Chapter 7 is a conclusion.
CHAPTER 2

THE RECIPROCATING, PNEUMATIC IMPACT TOOL

Tool Operation

The pneumatic impact tool has a piston that moves axially, forward and backward, in a cylinder as its primary operating mechanism. It is this piston that provides the impact to the chisel, or other attached device, to do the task required. The piston is moved toward the chisel by air pressure from a compressed air source. The piston is returned to the top of the cylinder with air pressure as well as the momentum transfer from the chisel impact. The high-pressure air is responsible for initiating operation of the tool and for maintaining its continued operation. The other features of the tool are designed to facilitate the operation of the piston.

The high-pressure air is supplied to the tool from some source, typically a standard one- or two-stage air compressor. The compressed air source and the associated air-lines must be of sufficient size to provide both the pressure and the volume needed to maintain operation of the tool. The high-pressure air line is coupled to the tool through a coupling on the tool that is generally located on or near the handle of the tool. The smaller tools frequently have the handle hollowed into two cavities separated by a valve. The compressed air is imported into one of the cavities as soon as the source is linked to the tool. The second
cavity is not pressurized until the spool valve is operated to allow passage of the air from the first cavity. The valve is controlled by the tool operator and is usually activated through the use of a trigger on the handgrip of the tool. The linear motion of the trigger makes the spool valve the valve of choice in this application. Continued depression of the trigger provides a continuous flow of air to the tool. Partial depression of the valve also provides the high-pressure air to the remainder of the tool but with a volume insufficient to operate the tool or to maintain steady state operation.

The cylinder that provides the range of motion for the piston has several ports to facilitate the insertion and evacuation of the compressed air. There is an intake port at the top of the cylinder and near the bottom of the cylinder to allow compressed air to access each end of the piston. There are exhaust ports located near the center of the longitudinal orientation of the cylinder. These ports permit the exhausting of the compressed air to the atmosphere and effectively terminate the acceleration of the piston due to the force of the line-pressure air. The sequencing of the exhaust ports is controlled simply by the location of the piston within the cylinder. A two-position valve outside the cylinder controls the airflow to the intake ports.

The two-position valve in these tools is a small, solid disk. The disk is very lightweight. The disk is operated in a small cylindrical cavity, the height of which determines the amplitude of motion of the valve. The motion of the valve disk is measured in millimeters. The valve disk chest is ported so that there is line pressure air on both sides of the disk. The valve chest is also ported to each of
the intake ports on the piston cylinder. Pressurization of the upper port on the piston cylinder is associated with the valve disk positioned at the top of the valve chest cavity. The lower end of the cylinder is pressurized when the valve disk is in the extreme bottom position of the valve chest cavity. The position of the piston within the cylinder also determines the position of the valve disk within the valve chest.

As the piston moves axially within the cylinder one of the exhaust ports is generally uncovered and open to the atmosphere. The other exhaust port is closed and allows pressurization within that half of the cylinder. When the exhaust port is covered and the cylinder pressurized, the corresponding side of the disk valve is also pressurized. The side of the valve that is responsible for pressurizing the cylinder is exposed to the entire valve chest cavity and the line air pressure acts over its entire surface. As the valve disk is forced against the end of the valve chest by the air pressure over its entire area, the line pressure present on the opposite side of the valve is restricted to a much smaller surface area due to the geometry of the upper and lower faces of the valve chest cavity. The valve is pressed against the small, pressurized port on the valve chest and effectively seals off that port. When the pressurized end of the piston passes an exhaust port the air pressure drops to an ambient level. The air pressure on the full face of the valve also drops to an ambient level. The line pressure operating on a small area on the back of the valve disk can now move the disk to the opposite end of the valve chest allowing pressurized air to flow to the opposite end of the piston within the cylinder. This process is repeated when the piston
moves in the reverse direction and passes the exhaust port associated with the
termination of the piston's acceleration in that direction. The valve disk translates
from one end of the valve chest to the other as frequently as the piston changes
direction but not in tune with the piston's reversals in direction.

The line air pressure and the associated valve operation are responsible for
accelerating the piston through the cylinder to its impact with the chisel. The line
air pressure also provides deceleration for the piston to cushion its motion and
control its displacement after the impact with the chisel. These controls are a
function of the location of the exhaust ports on the cylinder wall. The air pressure
is also responsible for starting the motion of the piston without regard to the
location of the piston and without depending on the impulse from the chisel to
supply velocity to the piston.

When the piston impacts the chisel, the motion of the piston is reversed. The
piston acquires a large velocity toward the top of the cylinder. Under these
conditions the air pressure has little effect on the upward motion of the piston.
As the piston nears the top of the cylinder, the line air pressure is introduced to
the top of the piston to decelerate the piston. It is possible for the piston to strike
the top of the cylinder and again have the direction of its motion reversed due to
the impact. Air pressure is, however, the primary method employed to slow the
piston as it moves upward toward the top of the cylinder and to accelerate the
piston downward toward impact with the chisel. This cycle is repeated many
times per second.
The cylinder is bored into the barrel of the tool. The barrel is designed to withstand the abuses of the operating environment of the tool. It is substantial enough to contain the cylinder as well as the smaller bored diameters that act as conduits for the pressurized air. The exhaust ports are vented to the atmosphere through the outside diameter of the barrel. The vented air is diffused and redirected away from the operator by a deflector shield placed around the barrel.

Tool Anatomy

The following pneumatic impact hammers were tested:

1. Ingersoll-Rand IR-121 impact hammer (Fig. 1)
2. Sears Craftsman Medium duty impact hammer (Fig. 2)
3. ATSCO No. 2 impact hammer (Fig. 3)
Figure 2: Sears Craftsman Medium duty impact hammer

Figure 3: ATSCO No. 2 impact hammer
All three tools had similar designs and operating components. The design and component operation is typical of many pneumatic impact tools. The variation in components is due to a scaling factor to accommodate the size of the tool rather than a different method of operation. In each of the tools, the operator has control over only the spool valve that permits pressurized air into the interior of the tool. The sequencing of the internal disk valve and the cyclic operation of the tool occur without operator intervention. The operator also provides a force through the tool and the chisel to the work piece that may affect the cycle rate of the tool. In the single degree-of-freedom testing and modeling that force is applied vertically downward through the tool and the chisel to coincide with the single degree-of-freedom of the moving mass.

The handle of the Ingersoll-Rand IR-121 and the Sears medium duty impact hammers is designed as a piston grip type handle. The pistol grip handle of the Ingersoll-Rand is aluminum. The pistol grip of the Sear medium duty impact tool is plastic. The ATSCO No. 2 is a much larger tool and has a handle cast into the tool. The handle portion of the ATSCO No. 2 is at the end of the tool in line with and perpendicular to the longitudinal axis of the tool. Despite the difference in exterior appearance, the interiors of the handles have a similar function.

The terminal end of each handle is drilled and tapped to accommodate a pipefitting. The Ingersoll-Rand IR-121 and the Sears medium duty are designed to take a one-fourth inch pipe thread. The ATSCO No. 2 is made to insert a fitting with a three-eighths inch pipe thread. The chamber inside the handle that is accessed by the threaded port is vented to the remainder of the tool by a port.
controlled by a spool valve actuated by a trigger. The trigger is a spring-loaded button that requires the application of a positive pressure for the tool to operate. When the trigger is depressed the valve is opened. The open spool valve allows the compressed air to enter the upper portion, or remainder, of the handle. The upper portion of the handle is a cylinder that is perpendicular to the part of the handle gripped by the operator. The cylindrical portion of the handle is bored out along its centerline. The boring is of multiple diameters on a common centerline. The outer diameter of the barrel at the end with the bored cylinder is threaded to allow the joining of the barrel of the tool to the handle that is threaded on an interior diameter. The deeper diameters allow for the fitting and captivation of the valve assembly that is inserted between the air reservoir in the upper portion of the handle and the ports in the cylinder within the barrel of the tool. The bored cavity is deep. The threaded portion designed to fasten the barrel to the tool handle extends approximately four centimeters.

The valve assembly that fits into the bored diameters of the tool handle is composed of four parts (Fig. 4 and Fig. 5):

1. The valve chest lid
2. The valve spacer
3. The valve disk
4. The valve chest bottom
Figure 4: Valve chest assembly

Figure 5: Exploded view of valve chest components
From the rear of the tool (i.e., the end closer to the handle) the first component is the valve chest lid. This cylindrical part is ported to the plenum in the upper portion of the handle. It allows the compressed air to enter the valve assembly. It has airways that connect to the valve chest bottom through the valve spacer so that both sides of the valve disk can be pressurized. When the valve disk is in the lower position, the porting in the valve permits line air pressure to the bottom end of the piston (i.e., the side farthest from the back of the cylinder) (Fig. 6).

Figure 6: Cross-section of Ingersoll-Rand model IR-121 valve in lower position to permit line pressure air to side ports in valve assembly to force piston up
Adjacent to the valve chest lid are the valve spacer and the valve disk (Fig. 7). The valve disk is a solid cylinder about the size of a small coin. The valve spacer resembles a large washer in that it is cylindrical with a large hole in the middle. It is approximately twice as thick as the valve disk. The valve spacer also has several smaller holes through its depth from end to end. The large hole is a slip fit for the valve disk. The smaller holes accommodate the requirement to allow air to move from the rear of the valve assembly (the valve chest lid) to the front side of the valve disk, the valve chest bottom, and to the cylinder. These smaller holes are aligned with air passages in the valve chest lid and the valve chest bottom. The valve disk moves axially within the confines of the thickness of the valve spacer.

Figure 7: Cross-section of Ingersoll-Rand model IR-121 valve in upper position to permit line pressure air to center port in valve assembly to force piston down.
The fourth piece of the valve assembly is the valve chest bottom. This piece is machined to provide the porting that is responsible for the motion of the valve and the piston. When the valve disk is at the top of the valve assembly, the line pressure is directed to the top of the piston accelerating it downward (Figure 7). This piece is also the cap on the top of the piston cylinder bore. The valve chest bottom provides a limit on the rearward travel of the piston.

The valve assembly is held together by the compression offered by the machined surface on the inside of the cavity in the upper portion of the handle and the top of the piston cylinder (barrel) when the later is threaded and tightened into the handle. The barrel must be tightened sufficiently so that the valve assembly is tightly compressed. Air leaks within the valve assembly degrade the efficiency of the valve and can deny successful operation.

In addition to the axial compression of the valve assembly, the valve chest lid, valve spacer, and the valve chest bottom are aligned with dowel pins. The dowel pins are also extended into the barrel to maintain the orientation of the valve assembly ports with the ports on the barrel. The pins are arranged in such a way as to preclude inversion or rotation of the components. The pins are not a press fit.

The airflow through the valve assembly is depicted in Fig. 8 and Fig. 9. Note that the position of the valve disk within the valve assembly corresponds to the end of the cylinder being charged with line pressure air.
Figure 8: Section view of Ingersoll-Rand IR-121 depicting the air flow from the plenum to the bottom of the piston

Figure 9: Section view of Ingersoll-Rand IR-121 depicting the airflow from the plenum to the top of the piston
The barrels of the tools provide the largest portion of the total weight of the tool. The barrel is a cylinder of several outside diameters with a cylindrical cavity of several different diameters that traverses the entire axial length of the barrel (Fig. 10, Fig. 11, Fig. 12). The barrels are made of steel to withstand the harsh exterior environment and the hundreds of thousands of cycles of the piston moving up and down in the bore of the barrel. Additionally, the barrels serve to provide support and orientation for the chisel that collides with the barrel scores of times per second when in operation.

Figure 10: Exterior views of the Ingersoll-Rand IR121 barrel showing the cylinder bore and the porting at the valve end.
Figure 11: Exterior views of the barrel of the Sears Medium Duty Impact tool
Figure 12: Views of the barrel of the ATSCO No. 2 impact tool
The wall between the outboard and the inboard diameters contains several passages that are oriented in an axial direction. The passages terminate at various locations into the interior bore of the barrel and are responsible for either routing line pressure air to the bottom of the piston or discharging air within the cylinder to the atmosphere (Fig. 13, Fig. 14, Fig. 15). The ATSCO No. 2 barrel has more ports than the other two tools. The higher number of passages used to charge the bottom of the cylinder are indicative of a faster rise to line pressure at the bottom of the piston (Fig. 12).

The rear outside diameter of the barrel has external threads that mate to the threads within the upper portion of the handle. The bottom of the barrel has a coarser external thread which allows the attachment of a retaining spring to be attached to the tool to captivate the chisel.

Figure 13: Section view of the Sears Medium Duty Impact tool showing the piston, the forward pressurization port and one of the exhaust ports
Figure 14: Section view of Ingersoll-Rand IR-121 depicting the piston in the cylinder with the exhaust porting to the right of the piston and the forward pressurization port on the left.
Figure 15: Section view of the ATSCO No. 2 impact tool with exhaust ports and forward pressurization port visible

The interior diameter of the barrel is a clearance fit to the outside diameter of the piston. That diameter is from the top of the barrel through almost all its length. The diameter is then coned down to a smaller diameter (approximately 10.2 millimeters or 0.402 inches) that supports and maintains the axial orientation of the chisel. From the bottom end of the barrel, the 10.2 millimeter diameter hole is chamfered to a large conical surface to assist in the insertion of the chisel in the tool. The relationship between the bottom end of the barrel and the bottom of the bored cylinder is standardized. That standardization allows generic chisels to fit in a broad range of similarly sized tools of various manufacturers.

The last two items are not integral to the operation of the tool but rather included as a safety item. The chisel retaining spring threaded onto the front of the barrel has a portion that captures the flange on the chisel and holds it loosely during its excitation. It prevents an unrestrained chisel from being “shot” from the end of the impact tool. The last item is a piece of rolled spring steel fashioned like a conical nozzle. The spring steel is not joined to form a continuous cylinder.
but is split so it remains in place by compression against the exterior wall of the barrel. The deflector fits at the junction of the barrel and the handle and is designed to disperse the air exhausted by the tool away from the hands of the operator.
CHAPTER 3

EXPERIMENTAL DATA

Test Stand

The collection of acceleration data from the three tools to be modeled required the use of several components. The tools themselves were fitted to accept a 3/8 inch quick disconnect air fitting. The flexible air hose was upgraded from 1/4 inch to 3/8 inch in order to supply a volume in excess of that needed to operate the tool.

The tests needed uniform impacting of the work piece over an extended period of time. The Ingersoll-Rand IR-121 impact tool and the Sears Medium Duty tool used a “pin driver” with a 0.401 inch shaft (Fig. 16). A larger diameter chisel was truncated to provide a blunt end (Fig. 17). That blunted chisel was used with the ATSCO No. 2 impact tool. Both impact devices were impacted perpendicularly into a steel plate. The deformation to the steel plate was mild but much less severe than would have been the case with a cutting edge on the chisel. The blunt end permitted future modeling to use a coefficient of restitution much closer to unity.
The blunt face on the impacting device did exacerbate the bounce of the tool on the work surface. The bounce made maintaining a vertical orientation to the tool and keeping the impacting surface over the work piece difficult. That situation was remedied by welding a short section of 1.5 inch by 1.5 inch square tubing with a quarter inch thick wall to a one-inch thick steel bar (Fig. 18). The surface of the steel bar accepted the impact of the tool. The short section of square tubing kept the impacting surface of the chisel from wandering across the
work piece. By inserting and operating the impacting chisel inside the square tubing, it was now possible to keep the tool and chisel vertical and aligned with the single degree-of-freedom of the moving mass. The tool operator was now able to maintain a consistent downward load on the tool.

The device that made this entire testing procedure possible was the large single degree-of-freedom mass that provided a platform for the testing of the tool. The device was designed and built specifically to permit the testing of impact devices where the acceleration levels of the impacting chisels were themselves too high to be measured.

The acceleration associated with the impacting chisel is high enough to destroy small accelerometers in a short period of time. Accelerometers large enough to measure the acceleration and survive the task were actually modifying the parameters of the chisel with their own mass and the required mounting assembly. Direct measurement along the principal line of action is also difficult.
The single degree-of-freedom mass needed to be substantial enough to withstand repeated testing with high impact devices. It had to be massive enough to have a resonance frequency well below the operating frequencies of the tools to be tested. It had to move freely along the desired degree of freedom.

Figures 19 and 20 show the test fixture that was used for the impact tool tests. The base of the test fixture was constructed by laminating one-inch and three-quarter inch thick steel plates to an aggregate thickness of 101.6 mm (4 inches). The plates are held together with four one half inch socket head cap screws threaded into tapped holes in the bottom plate. Alignment of the plates is maintained with four three-quarter inch shoulder bolts that are also threaded into tapped holes in the bottom plate. Plate alignment is also enhanced with the four one inch diameter polished and case hardened steel posts inserted through all but the bottom plate. Each shaft is locked in place with two socket head cap screws that act as setscrews. The shaft locking cap screws are threaded into tapped holes on the side of the base block. The center of the top surface of the base is milled away to provide an opening approximately four inches by four inches to a depth of approximately 3.25 inches. The milled cavity permits the wires connecting the accelerometer to the test instrument to be routed to circumvent the motion of the moving mechanical components of the test fixture. The milled cavity is accessible to the exterior of the base via a milled void in one of the substrate base layers. Overall the massive, fixed base is milled square to a size of approximately 16 inches by 12.5 inches. The one-inch diameter rods
Figure 19: Exploded view of the major components of the single degree-of-freedom test fixture
extend nine inches above the surface of the four-inch thick base. The mass of the stationary base is approximately 80 kilograms.

The moveable, single degree-of-freedom mass attached to the base is also an aggregate of laminated steel plates with a thickness of three and one-half inches. The overall size of the test mass is 12 inches by 12-3/4 inches. The alignment of the test mass is maintained with four three-quarter inch shoulder bolts. There are four linear bearings arranged in a square pattern on nine-inch centers to mate with the four steel shafts rigidly mounted in the fixed base. The top of the base is drilled and tapped with 5/16 inch NC threads in a rectangular pattern with 2.5 inch and 5 inch bolt centers. These holes allow for the addition of plates and fixtures to facilitate the testing without destroying the test mass itself. The underside of the test mass is drilled and tapped to accept the threads.
on the impact shock accelerometer to be used. The tapped hole is in the center of the block. Recessed grease fittings are also threaded into counter-bored holes in the side of the test mass to allow for the lubrication of the linear bearings.

The base mass and the test mass are separated by a nine inch diameter air-filled inner tube from a small tire. The inflated inner tube acts as an air spring against which the test mass may move vertically along its single degree of motion.

Markle investigated the properties of the single degree-of-freedom test fixture [96]. He measured the mass of the stationary base at 91 kilograms (approximately 200 pounds) and the mass of the moveable test block at 67.4 kilograms (approximately 150 pounds). With dynamic testing he calculated that the resonance frequency of the test fixture was 5.6 Hz (35 radians/second) and the damping coefficient to be 587 Newton-seconds / meter.

Instrumentation

The vibration signatures of the tool handle and the mass associated with the test fixture were monitored and recorded with Brue & Kjaer Portable PULSE system coupled with a desktop computer. The accelerometers were mounted to the objects to be monitored. The accelerometers were connected via cable to the PCB model 480D06 Power Units. The Power Units were connected to the Type 3560C Module in the Brue & Kjaer Portable Pulse System on the
computer by a mini-to-BNC cable. Figure 21 shows the interconnection of the components.

Each of the accelerometers was calibrated using the PCB model 394C06 calibrator (serial number 1566) in conjunction with the internal calibration circuitry mode in the PULSE hardware. The calibration signal is an acceleration of 9.8 m/s\(^2\) at 159.2 hertz (Hz). The two accelerometers calibrated and used were:

a. Shock accelerometer  Model 350b04  s/n 6202  
Factory calibration at  0.886 mV/g (0.090 mV/ m/s\(^2\))

b. Shock accelerometer  Model 350b04  s/n 5086  
Factory calibration at  0.901 mV/g (0.092 mV/ m/s\(^2\))
The shock accelerometer (s/n 5086) was affixed to the underside of a large (67.4 kg) moveable mass. The accelerometer was mounted so that it was aligned with the axial motion of the mass. The accelerometer was mounted to the underside of the mass by securing the threaded stud on the accelerometer into the tapped hole in the center of the mass (Fig. 19). The accelerometer cable was routed through the center of the inner tube and through cavities in the large base to the power unit.

The other shock accelerometer (s/n 6202) was mounted to the back of the handle of the chipping hammer (Fig. 22). The threaded stud on the accelerometer was mated into a small block of ¼ inch thick aluminum that was drilled and tapped. The aluminum block was epoxied to the rear of the impact tool so that the accelerometer was coaxial with the line of action of the tool. The accelerometer is further contained by black electrical tape that is wrapped around

![Figure 22: Typical accelerometer mounting to tool](image-url)
the handle and the accelerometer-mounting block to secure it from loosening due
to the vibration and to protect it from operator contact. The position of the shock
accelerometer on the handle is such that when the tool is held in a vertical
position to impact the moveable test mass, the axis of operation coincides with
the mass-mounted shock accelerometer.

The Ingersoll-Rand IR-121 and Sears medium duty pneumatic tools were
tested using the pin driver. The ATSCO No. 2 impact tool used a standard chisel
with the end blunted. The tool was operated with the pin driver (or blunted
chisel) inside the cavity created by the small section of square tubing mounted on
the test fixture (Fig. 13).

A display of the PULSE testing screen is shown (Fig. 22). The settings are
shown. The screen contains eight graphs. There were four graphs to display the
input information from each of the two channels. Channel 1 was the tool and
channel 2 was the moveable mass fixture. The graphs are labeled:

"Autospectrum"

The magnitude of the accelerometer signals were displayed on a frequency
spectrum. The graph plots acceleration in m/s^2 versus Hertz.

"Right" ("Left")

The signal directly from the sensor accelerometers plotted as m/s^2 versus
time in seconds.

"Right Vel" ("Left Vel")

The accelerometer signal was pre-processed and integrated by the software
to produce a magnitude for the velocity in m/s versus time in seconds.
Figure 23: Pulse software user screen during data collection
"Right Disp" ("Left Disp")

The accelerometer signal was pre-processed and double integrated by the software to produce a magnitude for the displacement in meters (m). The data was displayed versus time in seconds.

The frequency range of interest was set at 800 Hz. The necessary detail was achieved with 1600 lines of resolution. The frequency increment between captured measurements is 0.5 Hz (800 divided by 1600). The time frame required to accomplish those measurements was 2 seconds and was determined by the software. The higher the number of lines of resolution and the lower the frequency range, the more time is required to collect one complete set of data. In the case of a noisy, vibrating tool, consideration must be given to the tool operator and the other inhabitants of the space. The time span was kept short so that at least ten complete sets of data were applied to the average.

The data were saved as a comma-delimited data files that could be displayed in computer spreadsheet software, such as Microsoft EXCEL. The ASCII files were directly readable into MATLAB data processing programs for graphic displays with enhanced operator control of the graph parameters. The graphs of the accelerations of the tool and the mass as well as the frequency spectrums of the acceleration signals from the MATLAB processing program are included in Chapter 4. An exemplary MATLAB program to process and graph the test data collected by the B&K Pulse system is attached as Appendix C.
CHAPTER 4

THE COMPUTER MODELING

The reciprocating, pneumatic impact tool has two internal moving parts: the valve disk and the piston. The entire system consists of the impact tool, the chisel acting as an output for the tool, and the large single degree-of-freedom mass that receives the impulses from the chisel. All of the moving parts are collinear and aligned with the longitudinal axis of the pneumatic tool.

The mass of the valve disk is small in relation to the mass of the entire tool or to the moving piston. The valve disk translates back-and-forth over a small distance along the longitudinal axis of the tool to direct line-pressure air to either the top or the bottom of the piston. This model does not include the motion of the valve disk as its contribution to any momentum transfer would be very small (Table 2). The mass of the disk is retained as part of the total weight of the tool.

The entire system is oriented so that the longitudinal axis of the tool and the chisel, as well as the single degree-of-freedom mass, are vertical. The linearity and the orientation of the system preclude the necessity of computing gravitational components or other off axis values into another set of orthogonal components.

The three factors that influence the motion of the piston are the influence of gravity, the impulsive force due to the collision with the chisel, and the line air
<table>
<thead>
<tr>
<th>Specific Tool:</th>
<th>mass of valve disk (grams)</th>
<th>mass of piston (grams)</th>
<th>mass of entire tool (grams)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ingersoll-Rand IR-121</td>
<td>2.2</td>
<td>93.4</td>
<td>1570.5</td>
</tr>
<tr>
<td>Craftsman Medium Duty</td>
<td>0.4</td>
<td>93.1</td>
<td>1336.2</td>
</tr>
<tr>
<td>ATSCO #2 (serial number 0101)</td>
<td>6.8</td>
<td>307</td>
<td>6519.7</td>
</tr>
</tbody>
</table>

Table 2: Comparative masses of moving internal components

pressure introduced into the cylinder. The acceleration due to gravity is constant regardless of the direction of motion of the piston. The change in velocity due to the impulsive force occurs once per cycle at the bottom of the piston path. The pressurization of the cylinder is dependant on the porting of each specific tool and the pressure profile in the input code that attempts to emulate the cylinder air pressure over multiple impact cycles.

Piston Modeling

The longitudinal axis of the cylinder is vertical. The top of the cylinder is the end that is closer to the air valve, the air source and the handle. The bottom of the cylinder has a hole in its center to allow for the insertion of the tool-end of the chisel. The tool-end of the chisel extends into the cylinder so that it effectively precludes the piston from impacting the bottom of the cylinder bore. Along the side of the cylinder wall are a series of ports. The ports fall into two categories. There are ports that allow the introduction of line pressure air into the bottom
Figure 24: Exhaust ports in Typical tool cylinder

portion of the cylinder. There are ports that permit the venting of the pressurized air on alternating ends of the piston to the atmosphere (Fig. 24). As the motion of the piston opens an exhaust port, not only is the pressurized air vented to the atmosphere terminating the associated acceleration, but also the valve disk is cycled by the pressure drop. Air pressure to the top of the piston is supplied directly through the valve.

These exhaust ports are arranged as two sets (Fig. 25, 36, and 27). There are exhaust ports that terminate the line pressure as the piston is moving from the top toward the bottom. The second set of exhaust ports vent the pressurized air present as the piston moves from the bottom to the top. The location of the ports not only determine where in the cycle the air pressurization responsible for the acceleration of the piston is terminated but also determine the extent of the deceleration of the piston due to the pressurization of the cylinder in opposition to the motion of the piston.
Figure 25: Sectional view of the Ingersoll-Rand IR-121 impact tool showing the locations and dimensions of the exhaust ports.

Figure 26: Sectional view of the Sears Medium Duty impact tool showing the locations and dimensions of the exhaust ports.
Figure 27: Sectional view of the ATSCO NO. 2 impact tool showing the locations and dimensions of the exhaust ports

The acceleration and deceleration of the piston due to the line air pressure in the cylinder are a function of the position of the piston and not a function of time. (Consequently, the variations in the pressure are determined in the computer code by the location and the direction of travel of the piston.) In order to accommodate the three impact tools in a single code, the numeric values of the piston length, the exhaust port locations, and the distance from the top of the cylinder to the intruding chisel end were assigned as variables during the interactive, operator selection portion of the code. By accounting for the variations in each tool, the cycling of each piston can be modeled specifically.

The MATLAB computer code defines the air pressure in the cylinder as a function of location. The distance segments are smaller near the exhaust ports, not due to the release of pressurized air to the atmosphere, but due to the rapid rate of pressurization on the opposite end of the piston. The change in pressure
from ambient to line air pressure and then back to ambient is not assumed to occur instantaneously. *(The duration of the rise or decline in pressure is not measured in time directly but rather as a result of position.)* The time equivalent of that positional change is small because the range of tools investigated cycle between thirty-five and eighty times per second. For each cycle, there is pressurization at one end of the cylinder, a return to ambient pressure, a pressurization of the other end of the cylinder, and subsequent venting to an ambient level.

Tables 3, 4, and 5 list the piston and cylinder parameters for each of the three tools used as well as the forces due to the line air pressure applied over the cross-sectional area of the piston. Table 3 and Fig. 28, Fig. 29, and Fig. 30 are specific to the Ingersoll-Rand IR-121 impact tool.

The air pressure is depicted graphically for the downward and the upward piston movement of each of the tools in Figures 28 through 36. The locations of the exhaust ports are fixed in the tools. In the Ingersoll-Rand IR-121 impact tool and the Sears Impact tool, one set of exhaust ports is uncovered at all times. In the ATSCO No. 2 impact tool, the piston covers both sets of exhaust ports for a short duration as it traverses through the cylinder. The Sears medium duty impact tool uses two concentric diameters on its piston to change the relationship between the piston’s location and exhaust port opening.

Essentially, the Sears piston does not utilize its entire length as a bearing surface on which to slide within the cylinder. The stepped down diameter in the Sears piston is necessary to allow the cylinder porting to control the alternating
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<th>Absolute piston travel (in mm from top of cylinder)</th>
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<th>Bottom of lower exhaust port (in mm from top of cylinder minus the length of the piston barrel)</th>
<th>Min. piston impact distance (in mm from top of cylinder minus the length of the piston barrel)</th>
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Note: Maximum air pressure is 62.1 N/cm² (90 psi). Maximum force is 176 N.

Table 3: Chart of relationships between piston location and force due to air pressure for Ingersoll-Rand IR-121
Figure 28: Air pressure profile with piston moving down in Ingersoll-Rand tool
(note: pressure values indicate side of cylinder charged)

Figure 29: Air pressure profile with piston moving up in Ingersoll-Rand tool
(note: pressure values indicate side of cylinder charged)
Figure 30: Graphic air pressure profile for Ingersoll-Rand IR-121 tool with representative depiction of corresponding piston position within cylinder.
**SEARS MEDIUM DUTY IMPACT TOOL**

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<th>Normalized piston stroke</th>
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<th>Absolute piston travel (in mm from top of cylinder)</th>
<th>Top of upper exhaust port (in mm from top of cylinder)</th>
<th>Bottom of lower exhaust port (in mm from top of cylinder minus the length of the piston barrel)</th>
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Note: Maximum air pressure is 62.1 N/cm² (90 psi).

Maximum force is 176 N.

Table 4: Chart of relationships between piston location and force due to air pressure for Sears Medium Duty
Figure 31: Air pressure profile with piston moving down in Sears Medium Duty (note: pressure values indicate side of cylinder charged)

Figure 32: Air pressure profile with piston moving up in Sears Medium Duty (note: pressure values indicate side of cylinder charged)
Figure 33: Graphic air pressure profile for Sears Medium Duty tool with representative depiction of corresponding piston position within cylinder.
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<th>Normalized piston stroke</th>
<th>Normalized Force (pressure x area)</th>
<th>Absolute piston travel (in mm from top of cylinder)</th>
<th>Top of upper exhaust port (in mm from top of cylinder)</th>
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Note: Maximum air pressure is 62.1 N/cm² (90 psi). Maximum force is 382 N.

Table 5: Chart of relationships between piston location and force due to air pressure for ATSCO No. 2
Figure 34: Air pressure profile with piston moving down in ATSCO tool (note: pressure values indicate side of cylinder charged)

Figure 35: Air pressure profile with piston moving up in ATSCO tool (note: pressure values indicate side of cylinder charged)
Figure 36: Graphic air pressure profile for ATSCO N0.2 tool with representative depiction of corresponding piston position within the cylinder.
### Table 6: Displacement of Ingersoll-Rand and Sears Medium Duty pistons within the cylinder and the corresponding air pressure for one cycle

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<th>Piston Displacement (mm)</th>
<th>Pressure (N/cm²)</th>
<th>Pressure (psi)</th>
<th>Piston Displacement (mm)</th>
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positive air pressure while permitting the piston to reach the bottom of the cylinder.

The ATSCO No. 2 impact tool has multiple exhaust ports as well as multiple ports on the lower end of the cylinder (the impact side of the piston) so that the line air pressure is conveyed to that end of the piston very rapidly.

The tuning of the computer program required that the incrementing of the air pressure be adjusted. The program fails to reach steady state if the piston impacts the top of the cylinder and reverses velocity impulsively. The air pressure incrementing had to remain within the realm of possibility. The air pressure had to accelerate the piston to a velocity to initiate the cycle of work in the chisel. It also had to slow the piston and reverse its direction to prevent impact with the top of the cylinder.

Table 6 provides the time steps in a piston cycle from one impact with the chisel to the next as well as the corresponding piston displacement within the cylinder and the air pressure acting on the piston.

The computer model calculated a momentum transfer when the piston impacted the top of the cylinder. In every case, an impact by the piston into the top of the cylinder resulted in a large downward velocity of the piston. The piston translated through the length of the cylinder prior to the chisel being in position to be impacted. The piston consequently impacted the bottom of the cylinder only to be directed upward at a high velocity. In the course of two or three collisions with the ends of the cylinder, the system became unstable and ceased to function properly.
The forces exerted on the piston as it moves within the cylinder determine its motion (Fig. 37). The piston experiences the force due to gravity and the force from the line air pressure on the end surface of the piston. The collision between the piston and the chisel produces an impulsive force on both entities for that instant in time. The forces sum to produce acceleration. The acceleration is used in conjunction with the velocity from the previous time step to calculate the velocity at the current time step. From the velocity and the acceleration, the displacement is calculated. All of these values are done using Newtonian mechanics.

\[
\begin{align*}
a_i &= \frac{F}{m} & \text{EQ. 4-1} \\
v_i &= (a_i)(\Delta t) + v_{i-1} & \text{EQ. 4-2}
\end{align*}
\]
\[ d_t = \frac{1}{2} (a_t)(\Delta t)^2 + (v_t)(\Delta t) + d_{t-1} \]  

EQ. 4-3

The downward motion of the piston is terminated with its collision with the top of the chisel. The piston impacts the top of the chisel at a consistent steady state rate with a nominal 6.2 Pascals (Pa) (62 N/cm² or 90 p.s.i.) line air pressure of between 35 and 80 times per second, depending on the tool. Over the life of the tool the impacts do not significantly distort the impacting surfaces of either the piston or the chisel. It is therefore assumed that the collision is completely elastic and the coefficient of restitution is one. The chisel, at the time of impact with the piston, may be moving either up or down as shown in Fig. 38.

The post collision velocity of the tool and the chisel is determined using the conservation of linear momentum in conjunction with a restitution coefficient of one. From the basic statement of linear momentum

\[ m_1v_{1-old} + m_3v_{3-old} = m_1v_{1-new} + m_3v_{3-new} \]  

EQ. 4-4

and the expression for the coefficient of restitution

\[ e = 1 = (v_{3-new} - v_{3-old})/(v_{1-old} - v_{3-old}) \]  

EQ. 4-5

the values of the post impact velocities for the piston (associated with mass 1) and the chisel (associated with mass 3) are:

\[ v_{1-new} = [v_{1-old}(m_1 - m_3) + v_{3-old}(2m_3)]/(m_1 + m_3) \]  

EQ. 4-6

\[ v_{3-new} = [v_{3-old}(m_3 - m_1) + v_{1-old}(2m_1)]/(m_1 + m_3) \]  

EQ. 4-7
Figure 38: Velocities of pin-driver and piston at impact:

Impact type "A": chisel moving toward piston at impact
Impact type "B": chisel moving away from piston at impact

where:

\[ m_1 = \text{the mass of the piston} \]
\[ m_3 = \text{the mass of the chisel} \]
\[ v_{1\text{-new}} = \text{the post-collision velocity of the piston} \]
\[ v_{3\text{-new}} = \text{the post-collision velocity of the chisel} \]
\[ v_{1\text{-old}} = \text{the pre-collision velocity of the piston} \]
\[ v_{3\text{-old}} = \text{the pre-collision velocity of the chisel} \]
From the computed velocity of the piston the acceleration rate and the displacement are calculated.

\[ a_i = \frac{(v_i - v_{i-1})}{(\Delta t)} \]  
\[ d_i = \frac{1}{2}(a_i)(\Delta t)^2 + (v_i)(\Delta t) + d_{i-1} \]

The MATLAB computer code also accommodates the possibility of a collision between the piston and the top of the cylinder at the end of the upward stroke. Again, the coefficient of restitution and linear momentum are used to express the post collision velocities. The post collision velocities of the piston (associated with mass 1) and the tool (associated with mass 2) are:

\[ v_{1\text{-new}} = \frac{[v_{1\text{-old}}(m_1 - m_2) + v_{2\text{-old}}(2m_2)]}{(m_1 + m_2)} \]  
\[ v_{2\text{-new}} = \frac{[v_{2\text{-old}}(m_2 - m_1) + v_{1\text{-old}}(2m_1)]}{(m_1 + m_2)} \]

where:

- \( m_1 \) = the mass of the piston
- \( m_2 \) = the mass of the chisel
- \( v_1 \) = the post-collision velocity of the piston
- \( v_3 \) = the post collision velocity of the chisel
- \( v_{1\text{-old}} \) = the pre-collision velocity of the piston
- \( v_{3\text{-old}} \) = the pre-collision velocity of the chisel

The cycle rate of each of the pistons in each of the tools is not predetermined in any way. The total distance traveled in each cycle, the duration of the air pressure, the pre-impact velocities, and the ratio of the masses involved determines the motion of each piston.

The chisel impacts the single degree-of-freedom mass that is limited to vertical motion (Fig. 20). The mass is isolated from the fixed base by a large air bladder that acts as both a spring and a damper. The spring coefficient of the air

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bladder has been determined to be 85042 N/m (95). The range of dynamic motion of the mass is small (less than 0.01 meters) and, consequently, the stiffness is considered to be linear. The static deflection of the air bladder due only to the weight of the mass at rest is used as the initial, zero point of this model. The damping available from the bladder may not be linear with the impulsive loads. (It is, however, linearized and used as a tuning variable in the model.)

The tool has constant mass. There are no internal mechanisms within the tool to dissipate energy. There are no mechanical springs or any type of air suspension system. The tool is, however, held by a human operator, the hand-arm-shoulder of whom interacts with the overall mass of the tool. The hand-arm-shoulder applies a constant vertical, downward force to the tool. In the static condition prior to operation, that force is transmitted through the tool, through the chisel, and into the mass. Additionally, the weight of the tool, the piston, and the chisel combine to create an increased load on the air bladder. This load plus the operator applied force deflects the air bladder suspension system of the mass from its at rest position in proportion to the magnitude of the load and the force.

The vertical load statically applied to the air bladder is not constant under the dynamic operation. The force applied by the operator is modeled as being constant and always applied downward through the dynamic cycling of the system. The hand-arm-shoulder of the operator is modeled as a one directional spring. The hand-arm-tool relationship is depicted graphically in Fig. 39.
The hand-arm-shoulder is oriented so that the forearm is collinear with the longitudinal axis of the tool. The arm system is assumed to add downward force when the tool moves vertically upward. The arm is modeled to respond as a compressing spring when the tool moves upward. The downward force applied by the hand-arm-shoulder system is then a combination of the operator initiated force plus the spring force that is equal to the displacement times some spring constant. Prior studies have attempted to evaluated the spring constants associated with the hand-arm system. Multi-degree-of-freedom models have been created. Based on the published work on those models a value of 525,000
N/m [97-103] was established for the stiffness of the hand-arm system. That value was used as a starting point for the computer model.

The hand-arm-shoulder system also provides an energy sink for the motion of the tool in the vibrating mechanical system. The damping is related to the velocity of the movement. No assertion is made as to the linearity of the damping over a range of velocities or impulsive velocity changes. The damping coefficient is used in the computations as a linear value. Based on the published four-degree-of-freedom model [68] a damping coefficient of 545 N·s/m was used as an initial value for the damping coefficient of the hand-arm system. (Using the damping coefficient of the arm system as a tuning variable compensates for the possible non-linear character of the actual damping profile.) This is made possible by the consistency of steady state performance by each of the tools examined.

The MATLAB computer code starts with the user determining which of the three tools, the Ingersoll-Rand IR-121, the Craftsman Medium duty, or the ATSCO #1 impact chisel, are to be processed. Based on the user selection, the appropriate values for the masses of the tool, the chisel, and the piston are input into the code. The operator applied downward force was determined during the collection of test data by having the operator stand on a scale and noting the difference between his nominal weight and the weight displayed while doing the test. The measured downward force was different for each tool; 30 pounds for the Ingersoll-Rand IR-121, 30 pounds for the Sears Medium Duty, and 50 pounds for the ATSCO No. 2. That value, linked to the user's choice, is inserted
into the code. The time increment used to step the program through its motions is defined. The time increment used is 0.0005 seconds (one half of a thousandth of a second) for the Ingersoll-Rand and the Sears impact tools and 0.00075 seconds for the larger ATSCO impact tool.

The coordinate systems used set the single degree-of-freedom motion to be the x-direction with the positive direction up. The use of the absolute lengths of the chisel and the tool are avoided by setting a zero point on each of the three entities (tool, chisel, and s-d-o-f mass) based on the static equilibrium position of the s-d-o-f mass prior to the external loading. The top surface of the moveable s-d-o-f mass, prior to any force exerted on it, is set at a coordinate value of zero. The coordinate for the top of the chisel is set at zero based on the bottom of the chisel resting on the mass without exerting any load on the mass. The zero for the tool is based on the chisel at its previously defined location fully inserted in the tool cylinder. (Fig. 40)

The motion of the piston within the cylinder is distinct from the coordinates of displacement for the tool, the chisel, and the s-d-o-f mass, which are all related. The two systems are related by establishing the top of the cylinder, which is fixed within the tool, as zero in the local coordinate system. If the chisel and tool displacements are zero, there will be contact with the piston at the minimum piston-chisel contact distance. If there is separation between the tool and the chisel, the impact will occur at a piston displacement that corresponds to that change in distance. The unchanging length of the cylinder and the constant
zero position for piston within cylinder

top of piston determines location of piston

static equilibrium for mass supporting only its own mass

zero position for tool

zero position for chisel

No load on tool

+ X

Figure 40: Zero positions for the entities and the coordinate system orientation

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length of that portion of the chisel allowed into the tool always relate the position of the tool and the chisel and, through the chisel, the tool and the s-d-o-f mass.

The piston is assumed by the MATLAB computer code to be moving relative to the top of the cylinder, it is not affected by the motion of the tool. The tool has a range of motion measured in single digit millimeters while the piston travels a distance of more than 5 centimeters within the cylinder each half cycle. The relative magnitude of the two motions coupled with the rate of cyclical motion of the tool and the fact that its displacement is close to its starting point when the piston again strikes the chisel has made the rational for not adjusting the displacement of the piston relative to the global coordinate scheme seem appropriate.

With the program initiated, iteration through the time steps is begun. The piston is arbitrarily started from a spatial value of zero that equates to the top of the cylinder. The piston is limited from that point to motion in the downward or negative x-direction. The piston takes many time steps to reach an impact point. During those initial time iterations, a step force is applied to the mass to move it down (again in the negative x-direction). The step force is the sum of the weights of the tool, the chisel, and the piston as well as the operator applied force. The step force compresses the air bladder and establishes the value of the spring force in the positive x-direction acting on the system. The individual coordinate displacements for the tool and the chisel are also adjusted to reflect their new lower positions. The starting point is established for the initial impact between
the piston and the chisel. Tool, chisel, and s-d-o-f mass velocities and accelerations are set to zero prior to the first impact by the piston.

There are several permutations for the spatial relationship of each of the four moving bodies at any instant in time. The forces exerted on the bodies will differ depending on their relation in space to the other bodies.

Initial Impact

The initial impact between the piston and the chisel occurs when the tool, the chisel and the s-d-o-f mass are all in contact with each other and their initial spatial coordinate value is established by the operator applied force and the weights of the tool, chisel, and piston imposing themselves on the s-d-o-f mass to deflect the air bladder downward. The free-body diagram for this permutation is shown in Fig. 41. The force balances for all of the free body diagrams are resolved using d’Alembert’s principle:

\[ \sum F_{m1} = 0 = -m_1 \ddot{x}_1 - m_1 g + F_p \quad \text{EQ. 4-12} \]
\[ \ddot{x}_1 = \left( F_p + F_{\text{IMP}} \right) / m_2 - g \quad \text{EQ. 4-13} \]
\[ \sum F_{m2} = 0 = -m_2 \ddot{x}_2 - m_2 g + F_{CT} - F_p - F_{OP} \quad \text{EQ. 4-14} \]
\[ \ddot{x}_2 = \left( F_{CT} - F_p - F_{OP} \right) / m_2 - g \quad \text{EQ. 4-15} \]
\[ \sum F_{m3} = 0 = -m_3 \ddot{x}_3 - m_3 g + F_{CM} - F_{CT} - F_{\text{IMP}} \quad \text{EQ. 4-16} \]
\[ \ddot{x}_3 = \left( F_{CM} - F_{CT} - F_{\text{IMP}} \right) / m_3 - g \quad \text{EQ. 4-17} \]
\[ \sum F_{m4} = 0 = -m_4 \ddot{x}_4 - m_4 g + k_4 (-x_4) - F_{CM} \quad \text{EQ. 4-18} \]
\[ \ddot{x}_4 = \left( k_4 (-x_4) - F_{CM} \right) / m_4 - g \quad \text{EQ. 4-19} \]
Figure 41: Free body diagram; piston's initial impact into the chisel with all entities in contact
where:

\[ F_p = \text{Force due to air pressure x area} \]
\[ F_{\text{IMP}} = \text{Force from momentum transfer} \]
\[ F_{\text{OP}} = \text{Force from tool operator} \]
\[ F_{\text{CT}} = \text{Colinear force between chisel and tool} \]
\[ F_{\text{CM}} = \text{Colinear force between chisel and s-d-o-f mass} \]

Motion is initiated in the system by the piston when the piston impacts the top of the chisel. There is a momentum transfer and a new velocity for both the piston and the chisel. The equations evaluating the new velocities are Eq. (4-6) and Eq. (4-7). As a result of the momentum transfer the piston instantly changes from having a negative velocity to a positive velocity. The piston moves toward the top of the cylinder to initiate another cycle. The chisel acquires a large velocity in the negative direction but does not display any displacement. The tool and the mass also remain stationary during this time step.

In the next time step, the chisel, which has had no displacement but has a velocity, "impacts" the s-d-o-f mass (Fig. 42). The effectiveness of the tool-chisel system is dependant on the collision between the chisel and the mass not being purely elastic. As a result, the coefficient of restitution equation, needed with the linear momentum equation to solve for the two unknown resultant velocities (EQ. 4-20 and EQ. 4-21), does not have a coefficient of restitution, “e”, equal to one. (The coefficient of restitution is left in the equation as a variable to be used to tune the output of the system.) In fact, the blunted chisel face and the pin-driver do inflict permanent deformation to the steel surface at the bottom of the square tubing. The coefficient of restitution, while clearly not equal to unity, is still very

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Figure 42: Free body diagram; chisel's initial impact into the s-d-o-f mass with all entities in contact
high. The coefficient of restitution for the Ingersoll-Rand IR-121 and the Sears
Medium Duty impact tools with the pin driver was set at 0.97 while the more
substantial ATSCO No. 2 impact tool used a coefficient of restitution of 0.95.

The equations of motion for the four entities during the first chisel to s-d-o-f
mass impact, as depicted in Fig. 42, are:

\[ \sum F_{m1} = 0 = -m_1 \ddot{x}_1 - m_1 g + F_p \]
\[ \ddot{x}_1 = F_p / m_2 - g \]
\[ \sum F_{m2} = 0 = -m_2 \ddot{x}_2 - m_2 g + F_{CT} - F_p - F_{OP} \]
\[ \ddot{x}_2 = (F_{CT} - F_p - F_{OP}) / m_2 - g \]
\[ \sum F_{m3} = 0 = -m_3 \ddot{x}_3 - m_3 g + F_{CM} - F_{CT} + F_{IMP} \]
\[ \ddot{x}_3 = (F_{CM} - F_{CT} + F_{IMP}) / m_3 - g \]
\[ \sum F_{m4} = 0 = -m_4 \ddot{x}_4 - m_4 g + k_4 (-x_4) - F_{CM} - F_{IMP} \]
\[ \ddot{x}_4 = (k_4 (-x_4) - F_{CM} - F_{IMP}) / m_4 - g \]

Equation (4-28) is the linear momentum equation used to solve for the value
of the post-impact velocity of the chisel (subscript 3). Equation (4-29) is
the momentum equation used to compute the post-impact velocity of the mass
(subscript 4).

\[ v_{3,\text{new}} = [v_{3,\text{old}} (m_3 - e \cdot m_4) + v_{4,\text{old}} (1+ e)(m_4)] / (m_3 + m_4) \]
\[ v_{4,\text{new}} = [v_{4,\text{old}} (m_4 - e \cdot m_3) + v_{3,\text{old}} (1+ e)(m_3)] / (m_3 + m_4) \]

In order to calculate the forced response of the damped, spring-mass system,
the resonance frequencies of the system, \( \omega_n \), and operating frequency of the
system, \( \omega \), must be known. The resonance frequency is determined by taking
the square root of the quotient of the mass of the system divided by the stiffness of the system. In the case of the moving mass test fixture, the s-d-o-f mass is the mass of the block with all of the testing fixtures attached and the stiffness is the value assigned for the stiffness of the rubber inner tube that is the air spring. For the tool, the total mass was the mass of the tool plus a value included for the mass of the attached spring (i.e., the arm of the operator). The resonance frequency of the tool was then calculated by finding the square root of the quotient of the assigned mass divided by the stiffness. The units for the calculated circular frequency are radians/second.

The operating frequency is not calculated until the system is operating. To overcome this deficiency, the MATLAB computer code initially uses the approximated tool-operating rate as gleaned from the experimental data. The initial information is converted from frequency in cycles/second (Hz) to frequency in radians/second. During the operation of the tool, the time step for each piston-chisel impact is collected. At each impact, the number of time steps since the last impact is calculated. With the number of time steps between impacts and the number of time steps in a second of time known, the frequency can be calculated in cycles per second (Hz). The frequency is then converted to radians per second.

The time step with the initial chisel-mass impact imparts a velocity to the s-d-o-f mass that is computed into an impulsive force. The impulsive force is summed to the other forces present on the s-d-o-f mass at the instant of impact to provide a total force acting on the s-d-o-f mass. The s-d-o-f mass is a system.
with a spring (the air bladder) and a damper attached. As a result, the simple Newtonian relation between linear displacement, velocity and acceleration will not apply. The particular solution to the force response of a damped system is given by:

\[
F(t)_{\text{d-new}} = k_4 \frac{\cos(\omega \cdot t + \phi)}{\sqrt{(1-r^2)^2 + (2 \cdot \xi \cdot r)^2}} + x_{4\text{-old}} \quad \text{EQ. 4-30}
\]

where:

\( F(t)_{\text{d-new}} \) = the applied force at the current time step

\( k_4 \) = the stiffness coefficient of the air bladder

\( \omega \) = the operating frequency of the system

\( \xi \) = the damping ratio of the air bladder

\( r \) = the frequency ratio, the operating frequency divided by the natural frequency

The velocity and the acceleration for the s-d-o-f mass can be calculated by differentiating the displacement expression. Those equations are:

\[
v(t)_{\text{d-new}} = \frac{-F(t)_{\text{d-new}}}{k_4} \cdot \frac{\sin(\omega \cdot t + \phi)}{\sqrt{(1-r^2)^2 + (2 \cdot \xi \cdot r)^2}} \quad \text{EQ. 4-31}
\]

\[
a(t)_{\text{d-new}} = \frac{-F(t)_{\text{d-new}}}{k_4} \cdot \frac{\cos(\omega \cdot t + \phi)}{\sqrt{(1-r^2)^2 + (2 \cdot \xi \cdot r)^2}} \quad \text{EQ. 4-32}
\]

Since the velocity and acceleration values are starting from zero the MATLAB computer code for the acceleration and the velocity of the s-d-o-f mass remain...
exactly as they appear in EQ. (4-31) and EQ. (4-32). The calculated
displacement value is from a non-zero position. The displacement equation is
written into the MATLAB computer coding as:

\[
F(t) \frac{1}{k_4} \cdot \cos(\omega \cdot t + \phi) + x_{4, old}
\]

EQ. 4-33

to account for the non-zero starting point.

This time step has the tool again remaining stationary. The piston continues
its course toward the top of the cylinder and is unaffected by the collisions or
motions of the other entities in the system. The chisel, which has been involved
in two collisions in two consecutive time steps, has acquired an initial velocity in
the negative direction but no displacement and then a velocity in the positive
direction but again with no displacement as it now impacts the tool in the third
time step. Although the magnitude of both velocities is large, the chisel has yet
to undergo any displacement.

In the next time step, the second since the initial piston-chisel impact, the
chisel has a velocity in the positive direction. It has not moved. The chisel
impacts the tool. The impact between the tool and the chisel should not cause
any material deformation in either the tool or the chisel. Both are designed to
undergo hundreds of thousands of impacts without significant damage. Again
the coefficient of restitution, "e", is considered to be one. The post impact
velocities of both the tool and the chisel are computed from the linear momentum
formulation and the coefficient of restitution. The calculated post-impact
velocities are used to compute the impulsive force imposed on the tool and the chisel.

\[ F_{\text{IMP}} = m_{\text{g}} x_{1} \]

\[ F_{\text{CT}} = m_{2} x_{2} \]

\[ F_{\text{P}} = m_{3} x_{3} \]

\[ F_{\text{IMP}} = m_{4} x_{4} \]

\[ k_{4}(-x_{4}) \]

Figure 43: Free body diagram; initial impact of chisel into tool with tool
and chisel remaining in contact

The equations of motion for the four entities during the first chisel to tool impact, as depicted in Fig. 43, are:

\[ \sum F_{m1} = 0 = -m_1 \ddot{x}_1 - m_1 g + F_p \quad \text{EQ. 4-34} \]

\[ \ddot{x}_1 = \frac{F_p}{m_2} - g \quad \text{EQ. 4-35} \]

\[ \sum F_{m2} = 0 = -m_2 \ddot{x}_2 - m_2 g + F_{CT} + F_{IMP} - F_p - F_{OP} \quad \text{EQ. 4-36} \]

\[ \ddot{x}_2 = \frac{(F_{CT} + F_{IMP} - F_p - F_{OP})}{m_2} - g \quad \text{EQ. 4-37} \]

\[ \sum F_{m3} = 0 = -m_3 \ddot{x}_3 - m_3 g - F_{CT} - F_{IMP} \quad \text{EQ. 4-38} \]

\[ \ddot{x}_3 = \frac{(-F_{CT} + F_{IMP})}{m_3} - g \quad \text{EQ. 4-39} \]

\[ \sum F_{m4} = 0 = -m_4 \ddot{x}_4 - m_4 g + k_4(-x_4) \quad \text{EQ. 4-40} \]

\[ \ddot{x}_4 = \frac{(k_4(-x_4))}{m_4} - g \quad \text{EQ. 4-41} \]

The tool has forces that reflect the presence of the operator applied downward load, the effect of gravity on its mass, and the effect the piston-moving air pressure has on the ends of the cylinder. The piston, which was in contact with the tool at the instant of impact, has the same forces on it as the tool plus the force due to its own mass being accelerated by gravity. The impulsive forces are added to the other forces present on each of the entities to produce a total force. The acceleration of the chisel is computed from the summed forces as shown in EQ. (4-42). The velocities for both the tool and the chisel are determined from the momentum evaluation. The chisel is finally able to move.
The displacement of the chisel is calculated from Newtonian mechanics as shown in EQ. (4-43).

\[
\begin{align*}
\sigma_3 &= F_3 / m_3 & \text{EQ. 4-42} \\
x_{3,\text{new}} &= 0.5 \cdot a_{3,\text{new}} \cdot \Delta t^2 + v_{3,\text{new}} \cdot \Delta t + x_{3,\text{old}} & \text{EQ. 4-43}
\end{align*}
\]

The tool has a spring-damper restraint associated with the hand arm system and must be evaluated by finding the particular solution to the force response of a damped system. The equations are similar to those used to determine the displacement, the velocity and the acceleration of the s-d-o-f mass. Those equations are displayed as EQ. (4-30) through EQ. (4-32). The equation actually used in the MATLAB computer code is:

\[
x(t)_{2,\text{new}} = \frac{F(t)_{2,\text{new}} / \sqrt{(1-r^2)^2 + (2 \cdot \xi \cdot r)^2}}{\sqrt{(1-r^2)^2 + (2 \cdot \xi \cdot r)^2}} \cdot \cos(\omega \cdot t + \phi) + x_{2,\text{old}} & \text{EQ. 4-44}
\]

It must be noted that the stiffness \(k\), the force \(F\), the frequency ratio \(r\), and the phase \(\phi\) are particular and distinct for the tool and the s-d-o-f mass. When the computer user selects a tool from the list of three, the values for each of the named variables are computed and assigned a name in the MATLAB computer code. The equation for the displacement of the tool also contains a term to indicate that the initial position is not zero but rather the location that reflects the result of the pre-impact loading.

In the two time increments subsequent to the initial collision between the piston and the tool there have been two additional collisions. The s-d-o-f mass was accelerated downward, the tool was accelerated in the direction dictated by the sign of the net force after the impact, and the chisel was accelerated.
downward. In three time steps the entities have separated in some manner and until the operation of the pneumatic device ceases there is never again an instant in time when the location of the tool, the chisel and the mass can be predicted with certainty. The subsequent computer driven actions and interactions of each of the entities are determined by the relative displacements of the tool, the mass, and the chisel. The permutations of parameters permit five different situations:

A. The chisel is contacting the mass; the tool is not contacting the chisel.

B. The chisel is contacting the tool; the mass is not contacting the chisel.

C. There is no contact between any of the three entities.

D. All three entities are in contact.

E. The piston impacts the chisel

Although the acceleration and velocity of the chisel is assumed to be very high, the small time increments used will allow only one of the above interactions to take place at any time step. This concept is given further credence when it is realized that each of the above permutations requires a collision with the chisel. The chisel is either moving up or down at high velocities. After the initial collisions, the action of the chisel has established a separation between the mass and the tool. The chisel moves back and forth between the two, impacting them sequentially with ever decreasing velocity until the piston again impacts the chisel. The range over which the piston can impact the chisel is modeled as the distance the piston must travel to reach the maximum insertion depth of the chisel and the distance the piston must travel to “bottom-out” on the end of the
cylinder. It is unlikely that the chisel can simultaneously impact (or be impacted) by two entities.

At each subsequent time step the MATLAB computer code uses a series of nested conditional statements to determine from the displacements of the previous time step the appropriate mechanics for the current time step. Once the appropriate routine is identified from the permutations offered, the value of a dummy variable is changed to preclude continued evaluation beyond the permutation selected. Only one permutation is evaluated per time step.

For each of the permutations, the applicable forces must be evaluated for each of the entities. Free-body diagrams are used to visualize the equations developed. For each permutation, the acceleration, the velocity, and the displacement are determined from the net forces in effect. The evaluation of the motion of the chisel with no spring or damper is purely Newtonian mechanics. The s-d-o-f mass and the tool require that the particular solution to the forced response of a damped solution be found. The free-body evaluations are depicted in Fig. 44 to Fig. 47.

Permutation "A": Chisel Impacting S-D-O-F Mass

During permutation "A" the chisel is impacting the mass (Fig. 44). The piston continues in its cyclical path where the acceleration and deceleration are controlled by the air pressure applied to either the front or rear piston faces and by the effects of gravity. The tool experiences negative acceleration due primarily to the downward force applied by the operator. Additionally, the tool
experiences accelerative forces due to gravity and to the air pressure applied to
the upper and lower ends of the internal cylinder. The collision between the
chisel and the s-d-o-f mass causes an impulsive velocity change. The impulsive
forces of the collision that act on the s-d-o-f mass and the chisel, respectively,
are:

\[ F_{imp} = m_4 (x_{4\text{-new}} - x_{4\text{-old}})/(\Delta t) \quad \text{EQ. 4-45} \]
\[ F_{imp} = m_3 (x_{3\text{-new}} - x_{3\text{-old}})/(\Delta t) \quad \text{EQ. 4-46} \]

The equations of motion for the four entities from the free body diagram
(Fig.44) are:

\[ \sum F_{m1} = 0 = -m_1 \ddot{x}_1 - m_1 g + F_p \quad \text{EQ. 4-47} \]
\[ \ddot{x}_1 = F_p / m_2 - g \quad \text{EQ. 4-48} \]
\[ \sum F_{m2} = 0 = -m_2 \ddot{x}_2 - m_2 g - F_p - F_{op} - F_k \quad \text{EQ. 4-49} \]
\[ \ddot{x}_2 = (-F_p - F_{op} - F_k)/m_2 - g \quad \text{EQ. 4-50} \]
\[ \sum F_{m3} = 0 = -m_3 \ddot{x}_3 - m_3 g + F_{imp} \quad \text{EQ. 4-51} \]
\[ \ddot{x}_3 = F_{imp} / m_3 - g \quad \text{EQ. 4-52} \]
\[ \sum F_{m4} = 0 = -m_4 \ddot{x}_4 - m_4 g + k_4 (-x_4) - F_{imp} \quad \text{EQ. 4-53} \]
\[ \ddot{x}_4 = (k_4 (-x_4) - F_{imp})/m_4 - g \quad \text{EQ. 4-54} \]

The spring force \( F_k \) acting on the tool is applied in only the negative
(downward) direction. The displacement of the tool in the prior time step is
evaluated. If the displacement is less than the pre-operation, static position,
indicating the tool has moved down, the value of the dummy variable inserted in
that portion of the equation is zero. If the tool has been displaced higher than its
Figure 44: Free body diagram; condition "A", chisel impacting mass
initial position, the dummy variable is set to the difference between the current
displacement and the displacement just prior to the first piston-chisel impact so
that the spring force acts in a negative direction. The concept is that the
hand-arm-shoulder assembly that is applying the downward force input by the
operator will react to being compressed and the resultant spring force will be
equivalent to an addition to the downward force. On the other hand, a negative
displacement will be an extension of the hand-arm-shoulder assembly and
cannot be reacted to quickly enough to reverse the magnitude of the already
applied force. The hand-arm-shoulder is treated like a one-way spring.

Permutation "B": Chisel Impacting Tool

Condition "B" occurs when the chisel and the tool impact (Fig. 45). The chisel
cannot impact the tool and be impacted by the piston at the same time. The time
step is less than one one-thousandth of a second. The chisel does not affect the
single degree-of-freedom mass. The motion of the s-d-o-f mass is a function of
the force from the spring supporting the s-d-o-f mass.

The impulse momentum forces that apply to the tool and the chisel,
respectively, in this permutation are:

\[ F_{imp} = m_2 (x_{2-new} - x_{2-old})/(\Delta t) \]  
\[ F_{imp} = m_3 (x_{3-new} - x_{3-old})/(\Delta t) \]

The equations of motion from the free body diagram (Fig. 45) are:

\[ \sum F_{m1} = 0 = -m_1 \ddot{x}_1 - m_1 g + F_p \]  
\[ \ddot{x}_1 = F_p / m_2 - g \]
Figure 45: Free body diagram; condition "B", chisel impacting tool
\[ \sum F_{m2} = 0 = -m_2 \ddot{x}_2 - m_2 g - F_p - F_{OP} - F_k \quad \text{EQ. 4-59} \]
\[ \ddot{x}_2 = \frac{(-F_p - F_{OP} - F_k)}{m_2 - g} \quad \text{EQ. 4-60} \]
\[ \sum F_{m3} = 0 = -m_3 \ddot{x}_3 - m_3 g - F_{IMP} \quad \text{EQ. 4-61} \]
\[ \ddot{x}_3 = \frac{-F_{IMP}}{m_3 - g} \quad \text{EQ. 4-62} \]
\[ \sum F_{m4} = 0 = -m_4 \ddot{x}_4 - m_4 g + k_4(-x_4) \quad \text{EQ. 4-63} \]
\[ \ddot{x}_4 = \frac{(k_4(-x_4))}{m_4 - g} \quad \text{EQ. 4-64} \]

*Permutation "C": No Collisions*

The third possible combination, "C", of interaction is no interaction at all.

None of the entities has collided with the chisel. The free body diagram, Fig. 46, displays the forces and their orientations. The equations of motion from the free body diagram are:

\[ \sum F_{m1} = 0 = -m_1 \ddot{x}_1 - m_1 g + F_p \quad \text{EQ. 4-65} \]
\[ \ddot{x}_1 = \frac{F_p}{m_2 - g} \quad \text{EQ. 4-66} \]
\[ \sum F_{m2} = 0 = -m_2 \ddot{x}_2 - m_2 g - F_p - F_{OP} - F_k \quad \text{EQ. 4-67} \]
\[ \ddot{x}_2 = \frac{(-F_p - F_{OP} - F_k)}{m_2 - g} \quad \text{EQ. 4-68} \]
\[ \sum F_{m3} = 0 = -m_3 \ddot{x}_3 - m_3 g \quad \text{EQ. 4-69} \]
\[ \ddot{x}_3 = -g \quad \text{EQ. 4-70} \]
\[ \sum F_{m4} = 0 = -m_4 \ddot{x}_4 - m_4 g + k_4(-x_4) \quad \text{EQ. 4-71} \]
\[ \ddot{x}_4 = \frac{(k_4(-x_4))}{m_4 - g} \quad \text{EQ. 4-72} \]
Figure 46: Free body diagram; condition "C", no contact
It is not anticipated that this permutation will be exercised on two or more consecutive iterations as the spacing and the velocities will almost certainly require contact in the space of two time steps.

Permutation "D": No Collisions;
Tool, Chisel, S-D-O-F Mass in Contact

The fourth permutation, "D", has all of the entities in contact with no momentum transfer. The criteria for this state is not only the common displacement of the tool, the s-d-o-f mass, and the chisel but also that the velocity of the three bodies be less than 0.05 meters per second. This situation may never be realized with the tool in operation but would be needed if the line air to operate the tool were to be stopped and the bodies were allowed to return to a static state. Figure 47 displays the free body diagrams for this quasi-static state. The equations of motion are:

\[ \sum F_{m1} = 0 = -m_1 \ddot{x}_1 - m_1 g + F_p \]  
EQ. 4-73

\[ \ddot{x}_1 = \frac{F_p}{m_2} - g \]  
EQ. 4-74

\[ \sum F_{m2} = 0 = -m_2 \ddot{x}_2 - m_2 g + F_{CT} - F_p - F_{OP} - F_k \]  
EQ. 4-75

\[ \ddot{x}_2 = \left( F_{CT} - F_p - F_{OP} - F_k \right) / m_2 - g \]  
EQ. 4-76

\[ \sum F_{m3} = 0 = -m_3 \ddot{x}_3 - m_3 g + F_{CM} - F_{CT} \]  
EQ. 4-77

\[ \ddot{x}_3 = \left( F_{CM} - F_{CT} \right) / m_3 - g \]  
EQ. 4-78

\[ \sum F_{m4} = 0 = -m_4 \ddot{x}_4 - m_4 g + k_4 (-x_4) - F_{CM} \]  
EQ. 4-79

\[ \ddot{x}_4 = \left( k_4 (-x_4) - F_{CM} \right) / m_4 - g \]  
EQ. 4-80
Figure 47: Free body diagram; condition "D", tool, chisel, and mass in contact
The forces are not evaluated independently. The forces are summed for the mass and the displacements for the chisel and the tool are linked to that of the mass. The necessary condition for this state is that there be no momentum transfer. As a result, the velocities and accelerations for the tool and the chisel are set to zero. The forces that control the position of the mass are:

\[ x_4 = \left[ -F_{OP} - 9.81 \cdot (m_1 + m_2 + m_3) \right] / k_4 \]

EQ. 4-81

Permutation "E": Piston Impacting Chisel

The final possible permutation is the impact of the piston and the chisel. The momentum equations and the coefficient of restitution are used to calculate the impulsive force experienced by both the piston and the chisel. This is identical to the evaluation of the very first impact between the piston and the chisel. The second and subsequent piston impacts with the chisel occur at the end of the previous piston cycle, thus the resultant outcome is that the velocity of the tool, chisel, and s-d-o-f mass will have decreased due to the energy dissipating mechanisms in the system. In these subsequent impacts the chisel may possess a small velocity at the time of impact. Thus the momentum equation must contain the velocities of both the piston and the chisel prior to impact. It is also assumed that the entire system is very close to its static state. The displacements, velocities, and accelerations are calculated from the equations of motion and the kinematics calculated in the previous time step.

The evaluation of the motion during this permutation is very similar to the evaluation of the first impact. The chisel will have a post-collision velocity as it
did as a consequence of the very first impact but may or may not have a
displacement depending on its position relative to the s-d-o-f mass. The velocity
of the chisel is so large that the associated displacement would be terminated
prematurely by contact with the mass. As a result, if the relative positions of the
chisel and the mass are close together, the sequential impact between the chisel
and the s-d-o-f mass is included in this time step. Thus, the chisel will move
downward as a result of the piston-chisel impact and then move upward as a
result of the subsequent impact between the chisel and the s-d-o-f mass, all
within one time step.

The total time set for the duration of the simulation is forty seconds or eighty-
thousand 0.0005 second time steps. That length of time is used to assure that
steady state is achieved and that the cycling will continue as intended without
any failure. That time duration will allow a minimum of 1400 cycles for the
ATSCO No. 2 impact tool with a frequency of approximately 35 impacts per
second and 2800 cycles for the Ingersoll-Rand and the Sears Medium Duty with
cycle rates that are near 70 per second.

Every time the piston impacts the chisel a counter is sequenced. The counter
is tied to the time step so that the number of steps between each impact can be
determined. The impact counter is used in the averaging routine. In order to
view and compare the randomly chosen displayed time signal to a composite of
longer duration, the model output is averaged. In this model eight 0.10 second
sequences of impacts are averaged into a single 0.10 second graphic display.
To assure that the system has achieved steady state the averaging routine goes to the first impact after 2000 time increments. The average number of time steps between impacts is identified. The coded routine collects eight such time sequences and strings them together. The process is repeated through the entire 40 seconds of collected data with each subsequent set of acceleration values for the eight impact steps added to the first set of eight impact sets. This continues through the total time of operation until seven sets of eight impacts have been added to the original eight impacts. The newly created vectors containing the summed acceleration values spanning more than one-tenth of a second are divided by eight so that the acceleration values reflect an averaged value. The result is that the tool, s-d-o-f mass and chisel acceleration values for nine-tenths of a second of impacts are collected and averaged to provide a sequence of averaged acceleration signals that span more than one-tenth of a second. The checking process proceeds through the pool of impact cycles in the forward direction only. The acceleration values of the tool, the s-d-o-f mass, and the chisel are then displayed graphically as the average values.

The force-distance-velocity-acceleration relations for the chisel and the piston are straightforward. There are no energy storage or energy dissipation devices. Because the calculated displacement values represent the “distance traveled” during the time increment, those values are added to the positional value calculated in the previous time step to provide a new positional value with regard to the fixed coordinate system. The computed velocity and acceleration values are for the instant in time.
The tool has a constant operator applied force oriented axially in the negative direction. There is a force due to the air pressure on the piston that reverses direction during the course of the cycle of the piston. Additionally, there is an impulsive force that acts on the chisel in a positive direction whenever the chisel and tool collide. The resolution of the forces at each time step is dependent on the sum of those forces during that instant in time. Because the tool is supported by the hand-arm system of the operator, there is an energy storage device (i.e., the stiffness of the hand-arm system) and an energy dissipation device (i.e., the damping coefficient assigned to the hand-arm system). The motion of the tool, that is, its displacement, velocity, and acceleration, is derived from a forced response equation. Since there are no time increments for the tool without a force applied, only the particular solution that resolves the motion due to the force is applied. Velocity and acceleration values are calculated from the first and second time derivatives of the displacement response function.

Analysis of Method

The single degree-of-freedom mass is supported by a spring-damper system. The s-d-o-f mass experiences frequent impacts from the chisel. The change in momentum from those impacts when related to a time increment becomes an applied force. The force is applied in the negative direction. The motion of the s-d-o-f mass must be resolved using the displacement response and the appropriate time derivatives.

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There are time steps when the s-d-o-f mass is not impacted. The traditional method of resolving the displacement, velocity, and the acceleration of the s-d-o-f mass would be to use a displacement response for free vibration with initial conditions (i.e., the displacement and the velocity) and the first and second time derivatives of that equation.

The initial MATLAB computer code modeled the motion of the system uses a forced response resolution at every time step. The displacement of the "spring" associated with the s-d-o-f mass (in meters, m) multiplied by the spring stiffness (in Newtons per meter, N/m) equates to a force. That force is applied to the s-d-o-f mass, the motion of which is resolved with a displacement response to a forced vibration. The advantages are that there is no decision tree involved in deciding which equation to use, there is no recalculating of a phase angle, and the time stepping is not reinitiated.

In the 80,000 (40 seconds) time step computer simulation in the initial MATLAB computer code, several steady state time sequences were checked. The following collisions occur at the stated time steps for the Ingersoll-Rand IR-121 impact tool:

<table>
<thead>
<tr>
<th>Time step</th>
<th>Action occurring</th>
</tr>
</thead>
<tbody>
<tr>
<td>72,001</td>
<td>piston impacts chisel; chisel impacts the s-d-o-f mass</td>
</tr>
<tr>
<td>72,002</td>
<td>chisel impacts the tool</td>
</tr>
<tr>
<td>72,003</td>
<td>chisel impacts s-d-o-f mass</td>
</tr>
<tr>
<td>72,004</td>
<td>chisel impacts the tool</td>
</tr>
<tr>
<td>72,005</td>
<td>chisel impacts s-d-o-f mass</td>
</tr>
</tbody>
</table>
As a result of the chisel-mass impact at time step 72,001 the s-d-o-f mass
acquires the following values:

- displacement (global) -0.006031 m
- velocity 0.002706 m/s
- acceleration 42.139865 m/s²

Using those values the MATLAB computer code computes the following
values at time step 72,002, a “non-collision” time step for the s-d-o-f mass:

- displacement (global) -0.005996 m
- velocity -0.004113 m/s
- acceleration -7.133655 m/s²

The free vibration displacement response with the initial values of
displacement and velocity from time step 72,001 would yield the following values
for time step 72,002:

- displacement (global) -0.006032 m
- velocity -0.000963 m/s
- acceleration -7.315361 m/s²

The difference (free vibration values at time step 72,002 minus the MATLAB
original computer code value of the forced response values at the same time
step) expressed in absolute values (m, m/s, or m/s²) and as a percent (one
hundred times the quantity of the difference just calculated divided by the
MATLAB computer code calculated values) are as follows:

<table>
<thead>
<tr>
<th>Function</th>
<th>Difference</th>
<th>Percent</th>
</tr>
</thead>
<tbody>
<tr>
<td>displacement</td>
<td>-0.000036</td>
<td>0.60 %</td>
</tr>
</tbody>
</table>

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The velocity difference is large. It is used in the calculation of the momentum attributed to the s-d-o-f mass in the s-d-o-f mass-to-chisel impact in the next time step (step 72,002). The following tabulated values represent the values calculated in the MATLAB computer code with a forced response, the calculated values if the initial condition, free vibration response were to be used, and the values computed by the MATLAB computer code if the velocity of the s-d-o-f mass were to be reduced to 35% of their originally calculated value.

<table>
<thead>
<tr>
<th></th>
<th>Force Response</th>
<th>initial conditions</th>
<th>forced response at 0.35 * v4</th>
</tr>
</thead>
<tbody>
<tr>
<td>mass of chisel (kg)</td>
<td>0.120</td>
<td>0.120</td>
<td>0.120</td>
</tr>
<tr>
<td>mass of s-d-o-f mass (kg)</td>
<td>70.0</td>
<td>70.0</td>
<td>70.0</td>
</tr>
<tr>
<td>pre-impact velocity of chisel (m/s)</td>
<td>-6.51199</td>
<td>-6.51199</td>
<td>-6.51199</td>
</tr>
<tr>
<td>pre-impact velocity of s-d-o-f mass (m/s)</td>
<td>-0.004113</td>
<td>-0.000963</td>
<td>-0.0014396</td>
</tr>
<tr>
<td>total momentum (kg·m/s)</td>
<td>-1.069356</td>
<td>-0.848863</td>
<td>-0.882211</td>
</tr>
<tr>
<td>post-impact velocity of chisel (m/s)</td>
<td>6.481489</td>
<td>6.487779</td>
<td>6.486827</td>
</tr>
<tr>
<td>post impact velocity of s-d-o-f mass (m/s)</td>
<td>-0.002639</td>
<td>-0.023248</td>
<td>-0.023723</td>
</tr>
</tbody>
</table>

The 65% reduction of the force response velocity value for the s-d-o-f mass causes the post impact velocity in the next time step to have a difference of less than 0.001 m/s from the post impact velocity were it to be computed with the initial condition, free vibration response.
Other time steps were evaluated in the manner described for the impact related to time step 72,001 as well as the impacts and s-d-o-f mass values related to the impacts of the other two tools are included in Appendix A.

The Sears Medium Duty impact tool, very similar in size to the Ingersoll-Rand IR-121, also indicated that although the displacements and the accelerations for the two methods of computation were comparable the initial condition, free vibration response velocity was only 35% of the force response velocity. This was the same result found for the Ingersoll-Rand IR-121.

The ATSCO No. 2 impact tool had agreement from the two methods of calculation on the computed displacement. The velocity calculated with the forced response method for the ATSCO No. 2 tool was consistently about 35% to 40% less than the velocity determined from the initial condition evaluation. Additionally, the acceleration calculated from the forced response equation was approximately 150% of the acceleration determined from the initial condition equation. As a result, velocity and the acceleration values calculated in the computer code for the "non-collision" time steps for the ATSCO No. 2 impact tool were adjusted by factors of 1.35 and 0.65, respectively. The evaluations are included in Appendix A.

The final version of the MATLAB computer code that models the three impact tools reflects the adjustments due to the differences between the force response and the initial condition calculated values.

The displacement response function used to compute the displacement, the velocity, and the acceleration of the tool and the s-d-o-f mass is for a harmonic
input force. The forces present in the impact tools are not harmonic but impulsive. The importance of the denominator of the frequency response function and the sensitivity of the computed values to its magnitude was of interest. If the denominator was not an important factor in the determination of the computed displacement, velocity, and acceleration values, could it be replaced by an arbitrary scalar value? A sensitivity study was conducted by multiplying the "one over the denominator" (in the computer code) by positive factors from 0.90 to 2.00. The numerical results are included in Appendix B.

None of the modeled tools responded well when "one over the denominator" was multiplied by a value less than 1.0. There was no commonality among the tools when the factor was greater than 1.0 except that all of the computed acceleration magnitudes became larger. The ATSCO No.2 tolerated the increasing values of the multiplication factor the least of the three tools. The Sears Medium Duty also eventually ceased to achieve steady state operation. The Ingersoll-Rand IR-121 had increasing magnitudes through all of the factor incrementing from 1.0 to 2.0 and continued to achieve steady state although the magnitudes became unrealistically large.

It would appear that the "denominator" does provide a numerically significant value for the determination of the displacement, velocity, and acceleration terms. The computer generated displacements, velocities, and accelerations are not changed significantly when "one over the denominator" is multiplied by a factor the values of which are close to 1.0. The computed values do, however, change
significantly enough or the model fails to go to steady state so that applying a multiplication factor with a value too much greater than 1.0 is noticed.

The model is generated in the time domain. The testing of the actual tools provided signals in both the time domain and the frequency spectrum. In order to compare the frequency domain values of the modeled tool to those of the tested tool the time-signal acceleration values had to be converted into the frequency domain. The time signals are recorded in the model and by the testing as "magnitude" values. Those signals are converted into the frequency domain and are then converted to root mean square (rms) values. The algorithms used to convert the acceleration values from the time- to the frequency-domain and then the time domain magnitude values to the rms value are:

\[ (X_s)_{\text{rms}} = \frac{1}{n} \sum_{r=1}^{n} (x_r)_{\text{rms}} \sqrt{2} \]  
\[ (X_s)_{\text{freq}} = \frac{2^{n-1}(r-1)(s-1)}{n} \]  
\[ (X_s)_{\text{freq}} \]  

The graphs of the rms accelerations of the tool and the s-d-o-f mass from the testing and from the model are displayed for comparison in Chapter 5. Additionally, the rms values of the tool and for the s-d-o-f mass are weighted as per the third octave band weighting values from the International Standards Organization standard on "Mechanical vibration – Guidelines for the measurement and assessment of human exposure to hand-transmitted vibration" (ISO 5349) [87] and summed to provide a single number that represents, when compared with similar numbers, a relative vibration. ISO 5349, however, requires that the summation of octave band or third octave band rms weighted...
values include all the bands within the frequency range from 5 to 1500 Hz. The modeling of the impact tools does not produce a great deal of high frequency energy. The weighted rms acceleration values for the tested tools and from the modeled tools are summed for third octave band center frequencies up and including 630. Although the summation is not valid for comparison to summations calculated in accordance with ISO 5349, it is valid for comparison within the scope of this paper.
CHAPTER 5

RESULTS OF THE COMPUTER MODELING

Model Variables

The computer code was written to model three different pneumatic tools to avoid idiosyncratic anomalies that might go undetected if only one tool were modeled. In addition to the various mass values associated with the entities of each tool, the program allows for the insertion of other values used in the computer code. Some of the values are known (air pressure, the force applied by the operator). Other values are assigned initial values based on information available in the literature [97,103] such as the damping coefficients and spring stiffness for the inner tube supporting the s-d-o-f mass and for the hand-arm system. Those values are adjusted in the tuning process. The similarity in size and mass of the Ingersoll-Rand IR-121 and the Sears Medium Duty impact tools dictate that the parameters assigned for those tools be close to identical. The ATSCO No. 2 impact tool was larger. In testing, it required the operator to use two hands and the orientation his body over the tool. The initial deflection and the magnitude of the motion of the s-d-o-f mass by the ATSCO tool are significantly different from the other tools.
and the s-d-o-f mass test fixture is not perfectly elastic as evidenced by the chisel's modification of the surface of the test fixture. That value is set in the tuning process. Table 6 displays the values for the input variables used by the MATLAB computer code.

The last adjustable perimeter is the time increment used. The control of the magnitude of this quantity is needed because the relation between the displacement, velocity, and acceleration in the evaluation of the motion of the piston and the chisel with Newtonian mechanics is dependant on the duration of the time. The viable time increments were identified in the tuning process. The total time for the computer simulation is set to eighty thousand time steps of the selected time increment. In simulations for the two light tools (Ingersoll-Rand and Sears), a time increment of 0.0005 seconds is used. The ATSCO No. 2 had time steps of 0.00075 seconds.

The system of entities will not ever achieve steady state operation unless the piston operates repeatedly within the limits of its parameters. The mass of the piston and the location of the exhaust ports cannot be altered. The MATLAB computer code does allow some discretion in establishing a pressure profile for acceleration and deceleration portions of the stroke. The profiles also had to be within a plausible range for the piston cylinder pressure used.

Early attempts to run the model code revealed that there could be no contact between the piston and the top of the cylinder. This impact generates a
### Input variables for the MATLAB computer code

<table>
<thead>
<tr>
<th>Variable</th>
<th>Ingersoll-Rand IR-121</th>
<th>Sears Medium Duty</th>
<th>ATSCO No. 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spring Stiffness for s-d-o-f mass</td>
<td>85,042</td>
<td>85,042</td>
<td>127,563</td>
</tr>
<tr>
<td>Spring Stiffness for hand-arm</td>
<td>525,000</td>
<td>525,000</td>
<td>525,000</td>
</tr>
<tr>
<td>Damping Coefficient for s-d-o-f mass</td>
<td>900</td>
<td>900</td>
<td>960</td>
</tr>
<tr>
<td>Damping Coefficient for hand-arm</td>
<td>681</td>
<td>681</td>
<td>681</td>
</tr>
<tr>
<td>Multiplier for maximum air pressure</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Downward Force operator applied</td>
<td>30 (133 N)</td>
<td>30 (133 N)</td>
<td>50 (222 N)</td>
</tr>
<tr>
<td>Mass attributed to hand-arm</td>
<td>5</td>
<td>5</td>
<td>12</td>
</tr>
<tr>
<td>Coefficient of restitution piston-chisel</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>Coefficient of restitution chisel-mass</td>
<td>0.97</td>
<td>0.97</td>
<td>0.95</td>
</tr>
<tr>
<td>Coefficient of restitution chisel-tool</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>Velocity of mass reduction factor</td>
<td>0.35</td>
<td>0.35</td>
<td>1.35</td>
</tr>
<tr>
<td>Time increments</td>
<td>0.0005</td>
<td>0.0005</td>
<td>0.00075</td>
</tr>
</tbody>
</table>

Table 7: Input variables in MATLAB computer code

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large, negatively oriented velocity for the piston that is carried into the successive impact with the chisel or the bottom of the cylinder. The cyclical impacts increase in magnitude modifying the cyclical period. Ultimately the piston is simply oscillating between collisions at the top and the bottom of the cylinder with a cyclical rate of more than ten times the tool's operating repetitive rate. An impact with the top of the cylinder is a situation that causes the model to cease to function.

The importance of the pressure profile is not only to accelerate the piston downward toward the chisel but also to decelerate it to a stop to avoid impacting the top of the cylinder. At the bottom end of the piston stroke the air pressure forcing the piston up has little effect on the piston. The air pressure impeding the downward velocity of the piston is insufficient to cause any significant slowing of the piston prior to its impact with the chisel. After the piston-chisel impact, the velocity acquired by the piston is in excess of any change in velocity that would be caused by the air pressure within the cylinder.

The computer-generated profiles of the displacement and the velocity of the piston for each of the three tools modeled are shown in Figures 48 through 50.

The top of the cylinder is the upper boundary for the displacement of the piston and is set at zero. The piston moves in a negative direction toward its impact with the chisel and then returns upward to complete the cycle. The piston displacement at the bottom of the displacement cycle when it impacts the chisel varies from cycle to cycle. The chisel moves independently of the piston. The exact location of the chisel at impact varies and hence the total distance traveled
by the piston is different for each cycle of operation. All of the distances are within the range between minimum and maximum allowable impact locations as determined by the physical constraints of the system. The model does not produce a series of identical cycles but rather adapts to the range of displacements and velocities of the chisel and the piston.

The tool, chisel, and the mass all share a common static displacement prior to the introduction of air pressure to the tool. The displacement curves that start from the beginning of tool operation depict the tool and the mass separating due to the motion of the chisel between them. During operation, the dynamic range of motion of the mass remains consistent.

![Model of Ingersol-Rand IR-121](image)

Figure 48: Displacement and velocity curves for Ingersoll-Rand modeled piston

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Figure 49: Displacement and velocity curves for Sears Medium Duty modeled piston

Figure 50: Displacement and velocity curves for ATSCO #2 modeled piston

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The velocity of the chisel at impact with the piston is a factor in the displacement of the piston. The displacement appears to stabilize within a few tenths of a second after operation begins. The displacement of the mass well into the steady state operation indicates that the displacement of the test fixture is not only a function of the magnitude of the impacts but also, on a larger scale, a result of its resonance frequency.

The magnitude of the change in displacement from static loading to dynamic operation is related to the size of the tool and chisel used. The computer model goes to steady state in less than 0.3 seconds. The range of maximum and

![Model of Ingersoll-Rand IR-121](image)

Figure 51: Modeled displacements of the tool, the chisel, and the mass for Ingersoll-Rand IR-121

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<table>
<thead>
<tr>
<th>(dimensions in mm)</th>
<th>Ingersoll-Rand IR-121</th>
<th>Sears Medium Duty</th>
<th>ATSCO No. 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>static deflection</td>
<td>-2.34</td>
<td>-2.32</td>
<td>-3.21</td>
</tr>
<tr>
<td>maximum dynamic deflection</td>
<td>-6.94</td>
<td>-6.2</td>
<td>-10.87</td>
</tr>
<tr>
<td>minimum dynamic deflection</td>
<td>-5.23</td>
<td>-4.82</td>
<td>-5.26</td>
</tr>
<tr>
<td>range of dynamic motion</td>
<td>1.71</td>
<td>1.38</td>
<td>5.61</td>
</tr>
</tbody>
</table>

Table 8: Static and dynamic deflections of s-d-o-f mass

Minimum displacement values are gleaned from the values stored by the computer code between time steps 1000 (0.5 seconds) and 80,000 (40 seconds). For each of the three tools the values are shown in Table 7.

Figure 51 is a graphic display of the displacements of the piston, the tool, the chisel, and the mass from the initiation of operation of the Ingersoll-Rand IR121 to a time of 0.25 seconds. The change in the stroke length of the chisel is apparent as the separation between the tool and the s-d-o-f mass increases. The displacement exhibited by the mass is the more apparent due to the net forces applied, its weight, and the nature of its restoring forces. These displacements are typical of the three tools tested.

Each of the three tools tested and modeled are discussed separately.
Ingersoll-Rand IR-121 Impact Tool

The Ingersoll-Rand model IR-121 was tested with the pin driver as the chisel-type device. The tool was tested in the moveable mass, single degree-of-freedom test fixture with the square-tubing impacting fixture. The acceleration signals for the mass and the tool were collected in both the time and the frequency domains by the Pulse software.

Figure 52 depicts steady state displacements of the tool, chisel and s-d-o-f mass. It is apparent that the motion of the chisel causes the separation between the tool and the s-d-o-f mass. The peak-to-peak amplitude of motion for the tool is approximately 3 mm. The chisel moves through a range of motion of 7 mm.

![Figure 52: Graphic representation of the displacement of the tool, the chisel, and the mass from the Ingersoll-Rand IR-121 computer model.](image)
The individual, non-averaged acceleration values for the tool, the chisel, and the mass are depicted in Fig. 53. The values are greater for the tool and the mass than the averaged values from the computer model. Of note is the range of the acceleration values for the chisel. The model depicts those acceleration values of ± 20000 m/s². The mass has acceleration peaks less than 40 m/s² while the tool is in the +250 m/s² to -400 m/s² range.

The acceleration values of the s-d-o-f mass, the tool and the chisel were averaged over ten impulse cycles to produce the values depicted by the graphs in Fig. 54. In the model the values of eight cycles define more than one-tenth of

![Graph of acceleration values](image)

Figure 53: Graphic representation of the acceleration of the tool, the chisel, and the mass from the Ingersoll-Rand IR-121 computer model

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a second in time. To the individual acceleration values of each time increment in those eight cycles the acceleration values of seven additional cycles are added. The acceleration values are divided by eight to provide the average acceleration values of the tool, chisel, and mass over one-tenth of a second. The averaged values are lower than the extremes of individual signals. The ranges are displayed graphically in Fig. 54.

Figure 54: Graphic representation of the averaged acceleration of the tool, the chisel, and the mass from the Ingersoll-Rand IR-121 computer model
Figure 55: Acceleration values from testing of Ingersoll-Rand IR-121 tool

Figure 56: Average Acceleration values from model of Ingersoll-Rand IR-121

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The time domain acceleration values of the tool and the mass obtained from the testing of the Ingersoll-Rand model IR-121 are shown in graphic form in Fig. 55. That graph is generated by a MATLAB computer code that reads the digitized Pulse acceleration data from the tests. The averaged time domain acceleration graphs for the modeled tool and s-d-o-f mass based on data generated by the MATLAB computer program are shown in Fig. 56.

The discrete time domain acceleration signals of the Ingersoll-Rand tool collected during the test with the B&K Pulse system and the time domain calculated accelerations from the MATLAB computer model were converted into the frequency domain by a Fast Fourier Transform (FFT) algorithm and converted to root mean square values.

The B&K Pulse system made 4098 discrete measurements during the two seconds of recorded data. The calculation was made based on the testing measurement criteria that operated over a frequency range from 0 Hz to 800 Hz and 2048 discrete time domain data points in one second of time (Fig. 57).

The time domain model data was based on a time step of 0.0005 seconds or 2000 time steps per second.

The modeled tool repetitive operating rate is a function not only of the geometry of the tool but also the motion caused by the various collisions within the system. There is nothing in the MATLAB computer code of the modeled tool that dictates the cyclical rate of the modeled tool (Fig. 58).
Figure 57: Frequency domain acceleration graph from Ingersoll-Rand test

Figure 58: Frequency domain acceleration from Ingersoll-Rand computer model
The weighted third octave band rms acceleration values in the third octave bands of center frequencies 6.3 to 630 Hz were summed for the tool and the s-d-o-f mass with the following results:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tool (from testing; time signal)</td>
<td>18.749 m/s²</td>
</tr>
<tr>
<td>Tool (from testing; frequency signal)</td>
<td>22.193 m/s²</td>
</tr>
<tr>
<td>Tool (modeled)</td>
<td>21.843 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (from testing; time signal)</td>
<td>1.306 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (from testing; frequency signal)</td>
<td>1.744 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (modeled)</td>
<td>1.528 m/s²</td>
</tr>
</tbody>
</table>

Sears Medium Duty Impact Tool

The Sears Medium duty impact tool was tested with the pin driver attached and operated into the square-tubing impacting fixture mounted on the moveable mass. The acceleration signals for the tool and the mass were collected in both the time and the frequency domain. The discrete acceleration values were saved as an ASCII file and utilized by a MATLAB code to generate the graphic displays of the acceleration values.

Figure 59 depicts the steady state displacements of the tool, chisel and the s-d-o-f mass. The amplitude of the motion of the tool was greater than that of the Ingersoll-Rand IR-121 tool.
Figure 59: Graphic representation of the displacement of the tool, the chisel, and the mass from the Sears Medium Duty computer model at steady state.

The modeled action of the components displaced the mass from its static, pre-impact position of minus 2.32 mm to a maximum dynamic deflection depth of minus 6.20 mm. At steady state, the impacting of the chisel into the s-d-o-f mass caused the s-d-o-f mass to oscillate less than ±0.71 mm from a maximum coordinate value of minus 6.20 mm to the minimum coordinate value of 4.82 mm. Figure 59 depicts the vertical displacement of the tool to be approximately 3.5 mm and the chisel to move through a range of motion of approximately 7.0 mm.
Figure 60: Graphic representation of the acceleration of the tool, the chisel, and the mass from the Sears Medium Duty computer model at steady state.

Figure 60 graphically depicts the range of accelerations for the tool, the chisel, and the mass for individual impact cycles. The chisel is operating within maximum values of approximately ±20,000 m/s². The mass has individual acceleration peaks to about 40 m/s². The tool experiences individual peaks around +250 m/s² to -400 m/s². The time frame displayed is for steady state operation. The acceleration profiles in the first few hundredths of a second are different from the remaining displayed signals as the mass and the tool are being separated by the action of the chisel.
Figure 61: Graphic representation of the averaged acceleration of the tool, the chisel, and the mass from the Sears Medium Duty computer model.

As with the Ingersoll-Rand model IR-121 the magnitudes of the averaged acceleration values decreased from those of the individual signals (Fig. 61).

Figure 62 shows the tool and s-d-o-f mass acceleration from the testing of the Sears Medium Duty impact tool. Once again, the graphs represent the values recorded by the B&K Pulse system.

The averaged acceleration magnitudes for the modeled tool and the modeled s-d-o-f mass are shown in figure 63.
Figure 62: Acceleration values from testing of Sears Medium Duty impact tool

Figure 63: Averaged acceleration values from model of Sears Medium Duty tool
Figures 64 and 65 display the weighted rms acceleration in the frequency domain from the actual tool test and from the model, respectively.

The weighted third octave band rms acceleration values in the third octave bands of center frequencies 6.3 to 630 Hz were summed for the tool and the s-d-o-f mass with the following results:

<table>
<thead>
<tr>
<th>Source</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tool (from testing; time signal)</td>
<td>26.262 m/s²</td>
</tr>
<tr>
<td>Tool (from testing; frequency signal)</td>
<td>28.622 m/s²</td>
</tr>
<tr>
<td>Tool (modeled)</td>
<td>20.033 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (from testing; time signal)</td>
<td>1.716 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (from testing; frequency signal)</td>
<td>2.051 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (modeled)</td>
<td>1.441 m/s²</td>
</tr>
</tbody>
</table>
Figure 64: Frequency domain acceleration for Sears Medium Duty from test

Figure 65: Frequency domain signal from modeled Sears Medium Duty

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ATSCO No. 2 Impact Tool

The ATSCO No. 2 impact tool was tested using the truncated chisel in the impacting fixture on the moveable mass. The acceleration values of the tool and the mass were obtained in both the time and the frequency domains by the B&K Pulse system.

The major differences between the modeling of this tool and the two previously discussed, smaller tools are the operator applied load and the mass attributed to the hand-arm system of the operator. During testing, the operator of this tool held the tool with both hands and oriented the upper part of his body over the line of action of the tool.

Additionally, the damping and the spring stiffness of the inner tube supporting the s-d-o-f mass were increased over those values used for the smaller tools. The stiffness was increased because the range of motion and the maximum displacement of the s-d-o-f mass was much greater. The spring was assumed to offer more resistance to additional motion at the more compressed levels. The damping was increased due to the difference in the magnitude of the impulse to the mass (Table 6).

The truncated chisel did cause considerable damage to the impacting fixture. The coefficient of restitution was lowered to 0.95 to account for the increase in the momentum used in the deformation of the steel fixture.

The individual, non-averaged acceleration values for the tool, the chisel, and the mass are depicted in Fig. 67. The acceleration values for the chisel are in the range of ± 12,000 m/s². That value is considerably lower than the values for
the acceleration of the pin driver used in the Ingersoll-Rand IR-121 and the Sears Medium tool tests. The mass of the truncated chisel is approximately 0.59 kilograms while the pin driver has a mass of approximately 0.12 kilograms. The force associated with the motion of the truncated chisel is approximately fifty percent greater than that of the pin driver even with the substantial reduction in acceleration magnitude.

The average acceleration values are depicted in Fig. 68 and are much lower than the peaks of some of the individual pulses.

Figure 66: Graphic representation of the displacement of the tool, the chisel, and the mass from the ATSCO No. 2 computer model at steady state
Figure 67: Graphic representation of the acceleration of the tool, the chisel, and the mass from the ATSCO No. 2 computer model at steady state

Figure 68: Graphic representation of the averaged acceleration of the tool, the chisel, and the mass from the ATSCO No. 2 impact tool computer model

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Figure 69: Acceleration values from the testing of the ATSCO No. 2 impact tool

Figure 70: Acceleration values from the model of the ATSCO No. 2 impact tool

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Figures 69 and 70 display the acceleration values of the s-d-o-f mass and the tool from the actual testing and the modeled operation, respectively. Figures 71 and 72 display the rms acceleration values of the s-d-o-f mass and the tool in the frequency domain.

The weighted third octave band rms acceleration values in the third octave bands of center frequencies 6.3 to 630 Hz were summed for the tool and the s-d-o-f mass with the following results:

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Tool (from testing; time signal)</td>
<td>21.459 m/s²</td>
</tr>
<tr>
<td>Tool (from testing; frequency signal)</td>
<td>23.233 m/s²</td>
</tr>
<tr>
<td>Tool (modeled)</td>
<td>28.859 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (from testing; time signal)</td>
<td>5.255 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (from testing; frequency signal)</td>
<td>6.259 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (modeled)</td>
<td>5.469 m/s²</td>
</tr>
</tbody>
</table>
Figure 71: Frequency domain acceleration signal for ATSCO No. 2 from test

Figure 72: Frequency domain acceleration signal from modeled ATSCO No. 2
The MATLAB computer program modeled values for the accelerations of the tool and the mass are close to the values attained in the actual testing. The acceleration values of the chisel cannot be easily measured due to their magnitude. The computer model provides some insight to the magnitude of those accelerations:

Ingersoll-Rand model IR121 impact tool chisel accelerations
   Maximum  + 29417 m/s²
   Minimum  - 27667 m/s²

Sears Medium Duty impact tool chisel accelerations
   Maximum  + 29883 m/s²
   Minimum  - 28314 m/s²

ATSCO No. 2 impact tool
   Maximum  + 17921 m/s²
   Minimum  - 16082 m/s²

Chisel Comparison

The MATLAB computer code to model the three impact tools was adjusted to approximate the collected test data. All of the acceleration values were experimentally obtained with a blunted chisel (i.e., the pin-driver or the larger truncated chisel) impacting the mass. Additionally, the impact between the blunted chisel and the single degree-of-freedom mass was assessed to have a high, almost completely elastic, coefficient of restitution. Values for the coefficient of restitution were set at 0.97 for the Ingersoll-Rand IR-121 and the
Sears Medium Duty impact tools and at 0.95 for the larger, heavier ATSCO NO. 2 impact tool.

Test data for the Ingersoll-Rand IR-121 impact tool with an unmodified chisel was obtained. It was clear from the penetration into the work plate mounted on the single degree-of-freedom mass that the sharp edge of the chisel was penetrating the plate and that the coefficient of restitution was no longer almost purely elastic.

The *MATLAB* computer code for the Ingersoll-Rand IR-121 impact tool was unchanged except for the coefficient of restitution and the mass of the chisel. The coefficient of restitution was lowered from 0.97 to 0.88. The mass of the chisel was increased to 0.145 kg. The accelerations values for the tool and the single degree-of-freedom mass acquired from the testing are shown in Fig. 73. The acceleration values from the *MATLAB* computer model are displayed in Fig. 74.

The displacement of the single degree-of-freedom mass that is modeled by the *MATLAB* computer code to replicate the testing with the blunted tool (i.e., the coefficient of restitution is set at 0.97) is shown in Fig. 51. The mass is driven down from its initial (loaded and static) position.

The displacement of the single degree-of-freedom mass with the coefficient of restitution set at 0.88 is depicted in Fig. 75. The mass has nearly the same loaded and static initial position as the simulation with the coefficient of restitution of 0.97. The lower (more plastic) coefficient of restitution model has the single degree-of-freedom mass move to an equilibrium range that is less than the
Figure 73: Ingersoll-Rand test accelerations with sharp chisel

Figure 74: Ingersoll-Rand modeled accelerations with sharp chisel
Figure 75: A graphical representation of the displacements of the tool, the chisel, and the single d-o-f mass from computer model with restitution coefficient of 0.88 range caused by the pin driver. This is due to the reduced magnitude of momentum being transferred to the mass by the chisel as more momentum is converted to generating plastic deformation.

The frequency analysis of the impacts with the chisel reveals some separation in values between the modeled Ingersoll-Rand tool and the test data. The sum of the third octave band energy for the model is approximates that of the tested tool. Quite possibly because of the variation of operator loading due to the magnitude of the impulses being transmitted into the hand-arm system. That same loading variation may also be responsible for the frequency change in the tested tool's repetition rate. Figure 76 is the frequency domain rms
Figure 76: Frequency spectrum of Ingersoll-Rand test with chisel

Figure 77: Frequency spectrum of Ingersoll-Rand model with chisel
acceleration of the test tool converted from the time signal. Figure 77 is the FFT analysis of the rms acceleration of the modeled tool.

The rms acceleration values are as follows:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tool (from testing; time signal)</td>
<td>19.781 m/s²</td>
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<tr>
<td>Tool (from testing; frequency signal)</td>
<td>23.137 m/s²</td>
</tr>
<tr>
<td>Tool (modeled)</td>
<td>17.218 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (from testing; time signal)</td>
<td>1.591 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (from testing; frequency signal)</td>
<td>2.075 m/s²</td>
</tr>
<tr>
<td>S-d-o-f mass (modeled)</td>
<td>1.339 m/s²</td>
</tr>
</tbody>
</table>
CHAPTER 6

CONCLUSION AND SUMMARY

Summary

- The internal operation of the reciprocating, pneumatic impact tool were analyzed and documented.
- The internal motion of the tool is modeled by the computer code.
- The single degree-of-freedom test fixture was constructed to facilitate the testing of the impact hammers.
- The B&K Pulse system was ideal for the testing as it provides the ability to obtain time domain and frequency domain signals from the single degree-of-freedom mass and the tool.
- The interaction of the tool, the chisel, and the single degree-of freedom mass were modeled spatially in the dynamic operation of the tool.
- The computer code imposes no limitations except naturally occurring constraints on the location of the impacts.
- The computer code models the dynamics of individual components either by Newtonian mechanics (those with no springs) or by the particular solution to the displacement response of a harmonic
forcing function where the forcing function is actually an impulsive force.

Conclusions

- It was shown that the piston could not impact the top of the cylinder without chaotic results.
- The MATLAB computer model is general enough to model several different tool models.
- The model allows insight into the motion of the chisel which is difficult to quantify by testing.
- The model provides a reasonable portrayal of the motions involved in the tool and provides a platform for analytical investigation and tool modification without a time consuming and expensive iterative process.
- The model develops a method for the analysis of impacts that are:
  - Rapid in succession.
  - Occurring without regard to consistency of time intervals.
  - Occurring without regard to magnitude or spatial constraints.

Recommendations

- Conduct dynamic pressure testing to further define the modeled pressure profile.
- Model various attenuating devices to suggest effectiveness of the methods.
- Modify the code to have a minimum of 3200 time steps per second to produce a frequency domain analysis that extends to 1600 Hz to allow for correlation to ISO 5349.
- Construct a prototype attenuated, two degree-of-freedom model from the results of the computer model for testing and comparison.
APPENDIX A

CORRELATION BETWEEN INITIAL-CONDITION, FREE VIBRATION RESPONSE AND THE RESPONSE DUE TO A HARMONIC FORCING FUNCTION
<table>
<thead>
<tr>
<th>Time step</th>
<th>Action occurring</th>
</tr>
</thead>
<tbody>
<tr>
<td>72,000</td>
<td>piston impacts chisel</td>
</tr>
<tr>
<td>72,001</td>
<td>chisel impacts the s-d-o-f mass</td>
</tr>
<tr>
<td>72,002</td>
<td>chisel impacts the tool</td>
</tr>
<tr>
<td>72,003</td>
<td>chisel impacts s-d-o-f mass</td>
</tr>
<tr>
<td>72,004</td>
<td>chisel impacts the tool</td>
</tr>
<tr>
<td>72,005</td>
<td>chisel impacts s-d-o-f mass</td>
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</tbody>
</table>

Evaluation 2: (Piston-chisel impact #1551)

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<tr>
<th>Time step</th>
<th>Action occurring</th>
</tr>
</thead>
<tbody>
<tr>
<td>43,821</td>
<td>piston impacts chisel</td>
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<tr>
<td>43,822</td>
<td>chisel impacts the s-d-o-f mass</td>
</tr>
<tr>
<td>43,823</td>
<td>chisel impacts the tool</td>
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<tr>
<td>43,824</td>
<td>chisel impacts s-d-o-f mass</td>
</tr>
<tr>
<td>43,825</td>
<td>chisel impacts the tool</td>
</tr>
<tr>
<td>43,826</td>
<td>chisel impacts s-d-o-f mass</td>
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</table>

Evaluation 3: (Piston-chisel impact #2051)

<table>
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<tr>
<th>Time step</th>
<th>Action occurring</th>
</tr>
</thead>
<tbody>
<tr>
<td>57,911</td>
<td>piston impacts chisel</td>
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<tr>
<td>57,912</td>
<td>chisel impacts the s-d-o-f mass</td>
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<tr>
<td>57,913</td>
<td>chisel impacts the tool</td>
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<tr>
<td>57,914</td>
<td>chisel impacts s-d-o-f mass</td>
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<tr>
<td>57,915</td>
<td>chisel impacts the tool</td>
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<tr>
<td>57,916</td>
<td>chisel impacts s-d-o-f mass</td>
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### Values for single d-o-f mass (Ingersoll-Rand IR-121, first evaluation)

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<th>forced response values</th>
<th>forced response values</th>
<th>initial condition</th>
<th>difference</th>
<th>percent difference</th>
</tr>
</thead>
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<td>time step 72002</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>displacement (m)</td>
<td>-0.006031</td>
<td>-0.005996</td>
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<td>-3.6E-05</td>
<td>0.60</td>
<td></td>
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<tr>
<td>velocity (m/s)</td>
<td>0.002706</td>
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<td>-0.000963</td>
<td>0.00315</td>
<td>-76.59</td>
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<td>acceleration (m/s²)</td>
<td>42.139665</td>
<td>-7.133655</td>
<td>-7.315361</td>
<td>0.18171</td>
<td>2.55</td>
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<td><strong>MOMENTUM CALCULATIONS</strong></td>
<td>forced response (72002)</td>
<td>I-C free vibration (72002)</td>
<td>35% of forced (72002)</td>
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<td>mass of chisel (kg)</td>
<td>0.12</td>
<td>0.12</td>
<td>0.12</td>
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<td></td>
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<td>mass of s-d-o-f mass (kg)</td>
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<td>70.0</td>
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<td>pre-impact chisel vel (m/s)</td>
<td>-6.51199</td>
<td>-6.51199</td>
<td>-6.51199</td>
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</tr>
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<td>pre-impact s-d-o-f mass vel (m/s)</td>
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</tr>
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<td>-0.088221</td>
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<td>coef. of restitution</td>
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<td>0.96</td>
<td>0.96</td>
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</tr>
<tr>
<td>post-impact chisel velocity (m/s)</td>
<td>6.481489</td>
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<td>6.486827</td>
<td></td>
<td></td>
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</tr>
<tr>
<td>post-impact s-d-o-f mass velocity (m/s)</td>
<td>-0.002639</td>
<td>-0.023248</td>
<td>-0.023723</td>
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Values for single d-o-f mass (Ingersoll-Rand IR-121; second evaluation)

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<th>from computer code forced response values</th>
<th>forced response values</th>
<th>initial condition free vibration response</th>
<th>difference</th>
<th>percent difference</th>
</tr>
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<td>velocity (m/s)</td>
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<td>acceleration (m/s²)</td>
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TIME STEP CALCULATIONS

<table>
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<tr>
<th></th>
<th>forced response (43823)</th>
<th>I-C free vibration (43823)</th>
<th>35% of forced (43823)</th>
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<td>mass of chisel (kg)</td>
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<td>mass of s-d-o-f mass (kg)</td>
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<td>total momentum (kg*m/s)</td>
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<td>post-impact chisel velocity (m/s)</td>
<td>4.353149</td>
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### Values for single d-o-f mass (Ingersoll-Rand IR-121; third evaluation)

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<td>displacement (m)</td>
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<td>acceleration (m/s²)</td>
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<tr>
<td>mass of s-d-o-f mass (kg)</td>
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<td>pre-impact chisel vel (m/s)</td>
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<td>total momentum (kg*m/s)</td>
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<td>post-impact chisel velocity (m/s)</td>
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<td>-0.022647</td>
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# Sears Medium Duty

## Evaluation 1: (Piston-chisel impact #1000)

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<tr>
<th>Time step</th>
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<tr>
<td>28,768</td>
<td>chisel impacts the s-d-o-f mass</td>
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<td>28,769</td>
<td>chisel impacts the tool</td>
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<tr>
<td>28,770</td>
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## Evaluation 2: (Piston-chisel impact #1500)

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<tr>
<td>43,155</td>
<td>chisel impacts the s-d-o-f mass</td>
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<tr>
<td>43,156</td>
<td>chisel impacts the tool</td>
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<td>43,157</td>
<td>chisel impacts s-d-o-f mass</td>
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<td>43,158</td>
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## Evaluation 3: (Piston-chisel impact #2000)

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<td>57,546</td>
<td>chisel impacts the s-d-o-f mass</td>
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<tr>
<td>57,547</td>
<td>chisel impacts the tool</td>
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<tr>
<td>57,548</td>
<td>chisel impacts s-d-o-f mass</td>
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<td>57,549</td>
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### Values for single d-o-f mass (Sears Medium Duty; first evaluation)

<table>
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<th>forced response values</th>
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#### TIME STEP CALCULATIONS

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<th>Computer Code</th>
<th>Forced Response</th>
<th>Initial Condition</th>
<th>Difference</th>
<th>Percent</th>
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<tbody>
<tr>
<td>Displacement (m)</td>
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<tr>
<td>Velocity (m/s)</td>
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#### MOMENTUM CALCULATIONS

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<td>Mass of s-d-o-f mass (kg)</td>
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<td>Pre-Impact Chisel Vel (m/s)</td>
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<td>Pre-Impact s-d-o-f Mass Vel (m/s)</td>
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<td>Total Momentum (kg*m/s)</td>
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<td>5.623545</td>
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<td>from computer code</td>
<td>free vibration response</td>
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<tr>
<td>displacement (m)</td>
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<td>velocity (m/s)</td>
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<td>acceleration (m/s²)</td>
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<tr>
<td>mass of s-d-o-f mass (kg)</td>
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<td>pre-impact chisel vel (m/s)</td>
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<td>pre-impact s-d-o-f mass vel (m/s)</td>
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<td>total momentum (kg*m/s)</td>
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<td>coef. of restitution</td>
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<td>0.96</td>
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<td>post-impact chisel velocity (m/s)</td>
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<td>-0.026339</td>
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### Values for single d-o-f mass (Sears Medium Duty, third evaluation)

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<th>Difference</th>
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<tr>
<td>Displacement (m)</td>
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<td>Velocity (m/s)</td>
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<td>Acceleration (m/s²)</td>
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<td>Mass of s-d-o-f mass (kg)</td>
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<td>Total momentum (kg*m/s)</td>
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<td>-0.020690</td>
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### Evaluation 1: (Piston-chisel impact #1000)

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<td>31,726</td>
<td>chisel impacts s-d-o-f mass</td>
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<td>31,727</td>
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### Evaluation 2: (Piston-chisel impact #1500)

<table>
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<td>47,598</td>
<td>chisel impacts the s-d-o-f mass</td>
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<tr>
<td>47,599</td>
<td>chisel impacts the tool</td>
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<tr>
<td>47,600</td>
<td>chisel impacts s-d-o-f mass</td>
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<td>47,601</td>
<td>chisel impacts the tool</td>
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### Evaluation 3: (Piston-chisel impact #2000)

<table>
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<tr>
<td>63,459</td>
<td>chisel impacts the s-d-o-f mass</td>
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<tr>
<td>63,461</td>
<td>chisel impacts s-d-o-f mass</td>
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<td>63,462</td>
<td>chisel impacts the tool</td>
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Values for single d-o-f mass (ATSCO No. 2; first evaluation)

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<td>displacement (m)</td>
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<tr>
<td>velocity (m/s)</td>
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<td>acceleration (m/s²)</td>
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<td>free vibration (31725)</td>
<td>35% of forced (31725)</td>
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<tr>
<td>mass of s-d-o-f mass (kg)</td>
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<td>-4.586962</td>
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<td>post-impact chisel velocity (m/s)</td>
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APPENDIX B

SENSITIVITY EVALUATION OF HARMONIC FORCING FUNCTION EQUATION FROM MATLAB COMPUTER CODE
The displacement response to a harmonic forcing function on a system with viscous damping is:

\[ y(t) = \frac{F}{k} \frac{\sqrt{(1-r^2)^2 + (2\xi r)^2}}{\sqrt{(1-r^2)^2 + (2\xi r)^2}} e^{i(\omega t - \phi)} \]

where:

\[ r = \frac{\omega}{\omega_n} \]

\[ \xi = \text{the damping ratio} \]

\[ H(\omega) = \frac{e^{i(\omega t - \phi)}}{\sqrt{(1-r^2)^2 + (2\xi r)^2}} \]

Because the applied force in impact tool components is neither constant nor harmonic but rather is impulsive, the importance of the denominator in the frequency response system is checked to verify that it is more significant than an arbitrary scalar. The frequency response function is:

\[ H(\omega) = \frac{e^{i(\omega t - \phi)}}{\sqrt{(1-r^2)^2 + (2\xi r)^2}} \]

The denominator is multiplied by the numeric "FACTOR" in the included charts to produce tool and s-d-o-f mass accelerations. The minimum and maximum values of those vectors in the range of 1000 to 80,000 (i.e., steady state) are included in the charts and are used to determine the sensitivity of the denominator.
### ATSCO No.2 Impact Tool Sensitivity Evaluation

(range of 1000 - 80,000 time steps)

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### Sears Medium Duty Impact Tool Sensitivity Evaluation

(range of 1000 - 80,000 time steps)

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APPENDIX C

MATLAB COMPUTER CODE
TO PROCESS B&K PULSE
TEST DATA
% MATLAB processing PULSE data 03-22-03
% William Bloxsom
%
%A_Run_Tests_032203'
%
% Input EXCEL (.csv) file from PULSE with freq and magnitudes for
% autospectrum magnitude data for two accelerometers
% (tool and base) as well as real values for
% acceleration/time data for both the mass and the tool.
%
% set format for numerical storage
format long;
%
clc; % clear command window
disp(' ')
disp(' The tool model and test to be displayed is:')
disp(' ')
disp(' Tests from 09-10-02')
disp(' 1. Ingersol-Rand IR-121 blunt test number 1')
disp(' 2. blunt test number 2')
disp(' 3. blunt test number 3')
disp(' ')
disp(' 4. Sears-Craftsman Medium duty blunt test number 1')
disp(' 5. blunt test number 2')
disp(' 6. blunt test number 3')
disp(' ')
disp(' 7. 02-28-03 ATSCO #2 Ser. # 0101 blunt test number 1')
disp(' 8. blunt test number 2')
disp(' 9. blunt test number 3')
disp(' ')
disp('Tests from 03-22-03')
disp('10. Ingersol-Rand IR-121 blunt test number 1')
disp('11. blunt test number 2')
disp('12. blunt test number 3')
disp('')
disp('13. chisel test number 1')
disp('14. chisel test number 2')
disp('15. chisel test number 3')
disp('')
disp('16. blunt test at 1600 Hz')
disp('17. chisel test at 1600 Hz')
disp('')
ch1 = input(' the choice is: ');
disp('')
%
%
*******************************************************************************
%
if ch1 == 1;
    tit = ('Ingersol-Rand IR-121 test #1 09-10-02');
    A=dlmread('F:\wbloxsom\Tests 032203\IR-121 1 091002.csv',',',3,0);
end;
if ch1 == 2;
    tit = ('Ingersol-Rand IR-121 test #2 09-10-02');
    A=dlmread('F:\wbloxsom\Tests 032203\IR-121 2 091002.csv',',',3,0);
end;
if ch1 == 3;
    tit = ('Ingersol-Rand IR-121 test #3 09-10-02');
    A=dlmread('F:\wbloxsom\Tests 032203\IR-121 3 091002.csv',',',3,0);
end;
if ch1 == 4;
    tit = ('Sears Medium Duty test #1 09-10-02');
    A=dlmread('F:\wbloxsom\Tests 032203\SEARS 1 091002.csv',;',3,0);
end;
if ch1 == 5;
    tit = ('Sears Medium Duty test #2 09-10-02');
    A=dlmread('F:\wbloxsom\Tests 032203\SEARS 2 091002.csv',;',3,0);
end;
if ch1 == 6;
    tit = ('Sears Medium Duty test #3 09-10-02');
    A=dlmread('F:\wbloxsom\Tests 032203\SEARS 3 091002.csv',;',3,0);
end;
if ch1 == 7;
    tit = ('ATSCO Model 2 test #1 09-10-02');
    A=dlmread('F:\wbloxsom\Tests 022803\ATSCO 1 091002.csv',;',3,0);
end;
if ch1 == 8;
    tit = ('ATSCO Model 2 test #2 09-10-02');
    A=dlmread('F:\wbloxsom\Tests 022803\ATSCO 2 091002.csv',;',3,0);
end;
if ch1 == 9;
    tit = ('ATSCO Model 2 test #3 09-10-02');
    A=dlmread('F:\wbloxsom\Tests 022803\ATSCO 3 091002.csv',;',3,0);
end;
if ch1 == 10;
    tit = ('Ingersol-Rand IR-121 blunt test #1 03-22-03');
    A=dlmread('F:\wbloxsom\Tests 032203\IR121_B_1.csv',;',3,0);
end;
if ch1 == 11;
    tit = ('Ingersol-Rand IR-121 blunt test #2 03-22-03');
    A=dlmread('F:\wbloxsom\Tests 032203\IR121_B_2.csv',;',3,0);
end;
if ch1 == 12;
    tit = ('Ingersol-Rand IR-121 blunt test #3 03-22-03');
    A=dlmread('F:\wbloxsom\Tests 032203\IR121_B_3.csv',';',3,0);
end;
if ch1 == 13;
    tit = ('Ingersol-Rand IR-121 chisel test #1 03-22-03');
    A=dlmread('F:\wbloxsom\Tests 032203\IR121_C_1.csv',';',3,0);
end;
if ch1 == 14;
    tit = ('Ingersol-Rand IR-121 chisel test #2 03-22-03');
    A=dlmread('F:\wbloxsom\Tests 032203\IR121_C_2.csv',';',3,0);
end;
if ch1 == 15;
    tit = ('Ingersol-Rand IR-121 chisel test #3 03-22-03');
    A=dlmread('F:\wbloxsom\Tests 032203\IR121_C_3.csv',';',3,0);
end;
if ch1 == 16;
    tit = ('Ingersol-Rand IR-121 blunt test at 1600 Hz 03-22-03');
    A=dlmread('F:\wbloxsom\Tests 032203\IR121_C_1600.csv',';',3,0);
end;
if ch1 == 17;
    tit = ('Ingersol-Rand IR-121 chisel test at 1600 Hz 03-22-03');
    A=dlmread('F:\wbloxsom\Tests 032203\IR121_C_1600.csv',';',3,0);
end;

% ****************************************************************************
%
% fq = input('frequency span of test data (Hz): 0 to ');
fq = 800;
%
% dfq1 = input('Lines of discrimination ');
dfq1 = 1600;
%
% dfq2 = number of entries in time related column for one second
dfq2 = 2048;
%
% input excel data file into MATLAB
%
% A(i,1) = line counter for frequency data
% A(i,2) = frequency
% A(i,3) = autospectrum magnitude for tool from frequency data
% A(i,4) = autospectrum magnitude for mass from frequency data
% A(i,5) = time record
% A(i,6) = acceleration values for mass from time data
% A(i,7) = acceleration values for tool from time data
%
% ******************************************************************************

% set plot counters

t = [1:1600];
tt = [1:4000];
%
% ******************************************************************************

% figure(1);
subplot(2,1,1);
plot(A(t,2),A(t,3));
grid on;
TITLE ('tit');
XLABEL('freq (Hz)');
YLABEL('Tool Autospectrum mag. (m/s/s)');
%
subplot(2,1,2);
plot(A(t,2),A(t,4));
grid on;
TITLE (tit);
XLABEL('freq (Hz)');
YLABEL('Mass Autospectrum mag. (m/s/s)');

figure(2);
subplot(2,1,1);
plot(A(tt,5),-A(tt,7));
grid on;
TITLE (tit);
XLABEL('time (sec)');
YLABEL('Tool Acceleration (m/s/s)');

subplot(2,1,2);
plot(A(tt,5),A(tt,6));
grid on;
TITLE (tit);
XLABEL('time (sec)');
YLABEL('Mass Acceleration (m/s/s)');

figure(3);
hh = plot(A(tt,5),-A(tt,7),'k',A(tt,5),A(tt,6),'k');
grid on;
set(hh,['LineWidth',{.5;2.5}]);
TITLE (tit);
XLABEL('time (sec)');
YLABEL('acceleration (m/s/s)');
LEGEND('TOOL','MASS');

% end
APPENDIX D

SELECTED MATLAB COMPUTER CODE
FROM MODELING PROGRAM
Initial user screen where choice of tool to be run is made

% Single Degree-of-Freedom, four mass model of pneumatic impact hammer
%
%
% Ingersol-Rand IR-121, Sears-Craftsman Medium duty, ATSCO #2 Serial # 0101
%
% 01-24-2003
% William A. Bloxsom
%
% set numeric formatting for MATLAB code and array storage
% format long g
% max time for run (sec.)
% tmax = 40.0;
%
pi = 3.14159;%
%
clc; % clear command window
%
disp( ' ')
disp( ' ')
disp( ' ')
disp( ' ')
disp( ' ')
disp( ' The tool model to be displayed is:')
disp( ' ')
disp( ' 1. Ingersol-Rand IR-121')
disp( ' ')
disp( ' 2. Sears-Craftsman Medium duty')
disp( ' ')
disp( ' 3. ATSCO #2 Ser. # 0101')
disp( ' ')
disp( ' ')
ch1 = input( ' the choice is : ');
disp( ' ')
%

*******************************************************************************
Code establishes values specific to each tool for computer simulation. The inserted values for the Ingersoll-Rand IR-121 are shown.

```matlab
if ch1 == 1;
    % ********** mass values for Ingersol-Rand IR-121 (kilograms) **********
    tit = ('Model of Ingersol-Rand IR-121');

    m1 = 0.0934;  % m1 is piston
    m2 = 0.9255 + 0.0236 + 0.0022 + 0.0190 + 0.0701 + 0.0598 + 0.3769 + 0.01;
    m3 = 0.1200;  % m3 is the cutting chisel
    m4 = 70.0;    % m4 is the mass of the one d-o-f moving test fixture
    dia = 1.9;    % piston diameter (centimeters)
    Ax = 0.112;   % cylinder length (all in meters)
    Bx = 0.140;   % tool length
    Cx = 0.043;   % chisel to tool mating length
    Dx = 0.046;   % overall piston length
    Ex = 0.038;   % piston barrel length
    Fx = 0.008;   % chamfer + anvil length
    Gx = 0.0015;  % anvil length
    Hx = 0.038;   % barrel sliding length
    Ix = Bx-Cx-Dx; % minimum impact distance
    Jx = Ax-Ex;   % maximum impact distance
    dist2 = 0.044; % distance from rear of cylinder to top of first exhaust port
    dist3 = 0.084; % distance from rear of cylinder to bottom of second exhaust port
    freq = 72;    % initial guess at operating frequency (hertz)

    %
    spgmu1 = 1.0;  % multiplier for spring of mass spring
    spgmu2 = 1;    % multiplier for spring of arm spring
    mult1 = 1.5    % multiplier for damping of mass spring
    mult2 = 1.25   % multiplier for damping of arm spring
    adj = 1.0;     % multiplier for compressing air resistance
    adj2 = 30;     % downward force applied by operator in pounds
    madj = 5;      % addition to m2 for mass of arm
    cor1 = 1;      % piston-chisel coefficient of restitution
    cor2 = .97;    % chisel-mass coefficient of restitution
    cor3 = 1;      % chisel-tool coefficient of restitution
    ipf = .35;     % factor to reduce v4 due to initial-condition correlation
    delt = 0.0005; % time increment for step
end;
%
*****************************************************************************
```

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Root mean square section of code

```matlab
% % root mean square algorithm %
%
%
% lower frequency limits for one third octave bands

tobl(1) = 5;
tobl(2) = 7.1;
tobl(3) = 8.9;
tobl(4) = 11.2;
tobl(5) = 14.1;
tobl(6) = 17.9;
tobl(7) = 22.4;
tobl(8) = 28.2;
tobl(9) = 35.5;
tobl(10) = 44.7;
tobl(11) = 56.2;
tobl(12) = 70.8;
tobl(13) = 89.1;
tobl(14) = 112;
tobl(15) = 141;
tobl(16) = 178;
tobl(17) = 224;
tobl(18) = 282;
tobl(19) = 355;
tobl(20) = 447;
tobl(21) = 562;
tobl(22) = 708;
tobl(23) = 891;
```

% one third octave band center frequencies

tobcf(1) = 6.3;
tobcf(2) = 8;
tobcf(3) = 10;
tobcf(4) = 12.5;
tobcf(5) = 16;
tobcf(6) = 20;
tobcf(7) = 25;
tobcf(8) = 31.5;
tobcf(9) = 40;
tobcf(10) = 50;
tobcf(11) = 63;
tobcf(12) = 80;
tobcf(13) = 100;
tobcf(14) = 125;
tobcf(15) = 160;
tobcf(16) = 200;
tobcf(17) = 250;
tobcf(18) = 315;
tobcf(19) = 400;
tobcf(20) = 500;
tobcf(21) = 630;
tobcf(22) = 800;

% one third octave band magnitude weighting factors from ISO-5349

tobwf(1) = 1;
tobwf(2) = 1;
tobwf(3) = 1;
tobwf(4) = 1;
tobwf(5) = 1;
tobwf(6) = 0.8;
tobwf(7) = 0.63;
tobwf(8) = 0.5;
tobwf(9) = 0.4;
tobwf(10) = 0.3;
tobwf(11) = 0.25;
tobwf(12) = 0.2;
tobwf(13) = 0.16;
tobwf(14) = 0.125;
tobwf(15) = 0.1;
tobwf(16) = 0.08;
tobwf(17) = 0.063;
tobwf(18) = 0.05;
tobwf(19) = 0.04;
tobwf(20) = 0.03;
tobwf(21) = 0.025;
tobwf(22) = 0.02;

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i = 1;
while i < 2050;
    testm(i) = 0;  testt(i) = 0;
    modelm(i) = 0;  modelt(i) = 0;
    i = i + 1;
end;
k = 46000;
l = (-1)^.5;
n = 2000;
if ch1 == 3;
    n = 1333;
end;
m = 1;
while m < (n+1);
    j = 1;
    while j < (n+1);
        modelm(m) = modelm(m) + ( (1/n) * X(j+k,13)...
                             *exp( ((2*pi*i)/n) * (j-1) * (m-1) ) )/(2^5);  %X(j+k,13) BB(j,4)
        modelt(m) = modelt(m) + ( (1/n) * X(j+k,7)...
                             *exp( ((2*pi*i)/n) * (j-1) * (m-1) ) )/(2^5);  % X(j+k,7) BB(j,2)
        j = j + 1;
    end;
m = m + 1;
end;
i = (-1)^.5;
n = 2048;
m = 1;
while m < (n+1);
    j = 1;
    while j < (n+1);
        testm(m) = testm(m) + ( (1/n) * A(j,6)*exp( ((2*pi*i)/n) * (j-1) * (m-1) ) );
        testt(m) = testt(m) + ( (1/n) * A(j,7)*exp( ((2*pi*i)/n) * (j-1) * (m-1) ) );
        j = j + 1;
    end;
m = m + 1;
end;
j = 1;
while j < 24;
    tobtestm(j) = 0;
    tobtestt(j) = 0;
    tobmodelm(j) = 0;
    tobmodelt(j) = 0;
pulsem(j) = 0;
pulset(j) = 0;
    j = j + 1;
end;
\begin{verbatim}
j = 1; k = 1;
while j < 1000;
    if j >= tobl(k + 1);
        k = k + 1;
    end;
    if j >= tobl(k);
        if j < tobl(k+1)
            tobtestm(k) = tobtestm(k) + testm(j);
            tobtestt(k) = tobtestt(k) + testt(j);
            tobmodelm(k) = tobmodelm(k) + modelm(j);
            tobmodelt(k) = tobmodelt(k) + modelt(j);
            pulsem(k) = pulsem(k) + A(j,4);
            pulset(k) = pulset(k) + A(j,3);
        end;
    end;
    if (j + 1) >= (tobl(23));
        j = 100000;
    end;
    j = j + 1;
end;
end;
while j < 23;
    tobtestm(j) = (tobtestm(j) * tobwf(j))^2;
    tobtestt(j) = (tobtestt(j) * tobwf(j))^2;
    tobmodelm(j) = (tobmodelm(j) * tobwf(j))^2;
    tobmodelt(j) = (tobmodelt(j) * tobwf(j))^2;
    pulsem(j) = (pulsem(j) * tobwf(j))^2;
    pulset(j) = (pulset(j) * tobwf(j))^2;
    j = j + 1;
end;
tottestm = 0;    tottestt = 0;    totmodelm = 0;
totmodelt = 0;    totpulsem = 0;    totpulset = 0;
j = 1;
while j < 23;
    tottestm = tottestm + tobtestm(j);
    tottestt = tottestt + tobtestt(j);
    totmodelm = totmodelm + tobmodelm(j);
    totmodelt = totmodelt + tobmodelt(j);
    totpulsem = totpulsem + pulsem(j);
    totpulset = totpulset + pulset(j);
    j = j + 1;
end;
tottestm = abs((tottestm)^(.5))
tottestt = abs((tottestt)^(.5))
totmodelm = abs((totmodelm)^(.5))
totmodelt = abs((totmodelt)^(.5))
totpulsem = abs((totpulsem)^(.5))
totpulset = abs((totpulset)^(.5))
\end{verbatim}
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3. Alaska Department of Labor and Workplace Development, internet, www.labor.state.ak.us


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89. Technical Committee ISO/TC 118, C., pneumatic tools and pneumatic machines, Hand-held portable power tools - Measurement of vibrations


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Modeling of the Reciprocating, Pneumatic Impact Hammer

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