Development of a vibration-absorbing handle for rock drills

by J.P.D. Strydom*, P.S. Heyns* and J.L. van Niekerk†

Synopsis

The transmission of excessive vibration to the human body can cause physical stress that may result in crippling pain or permanent disability. Rock drills are known to transmit high levels of vibration that are difficult to attenuate through conventional approaches to vibration isolation. However, it can be shown that once the ISO frequency weighting has been applied, most of the vibratory effect of the rock drill's motion is concentrated at the drill's operating frequency. This observation opens the way for utilizing a concept known as vibration absorption, aimed at attenuating the rock drill's vibration. In essence this entails using the inertia of a secondary vibratory system to counteract the motion of the handle. This article documents an investigation into the use of such an absorber for attenuating rock drill vibration and demonstrates the feasibility of such an approach. A mathematical model of such an absorbing handle was developed and used for design studies. Based on these investigations an experimental prototype model was manufactured and tested. A significant reduction in vibration levels could be achieved.

Introduction

Vibration has long been recognized as a possible source of injury to the human body. Vibration-induced white finger was first recognized in the limestone quarries of Bedford, Indiana as early as 1890-1900 (Taylor *et al.*[1]; Taylor[2]) and was subsequently reported in Italy in 1911 (Griffin[3]). Since then pneumatic drills have replaced the hammers and chisels used in mining operations. Between the two World Wars, electrically driven grinders replaced hand grinding and in the 1950s, chain saws powered by internal combustion engines replaced hand saws and axes in forestry. Such technical advances led to an increase in hand-arm vibration injuries world-wide where such tools were used.

Various epidemiological studies among miners and stoneworkers have confirmed this trend. Such studies were conducted in many countries, including the United States of America (Taylor *et al.*[1]; Wasserman *et al.*[4]), Canada (Narini *et al.*[5]), Britain (Rodgers *et al.*[6]), Sweden (Hedlund[7]), Italy (Bovenzi *et al.*[8]; Bovenzi *et al.*[9]), Korea (Moon *et al.*[10]) and Japan (Matsumoto[11]; Sakakibara *et al.*[12]). Some of these studies reported an extremely high prevalence of vibration-induced white finger among the operators (Griffin[13]).

A recent South African study (Van Niekerk *et al.*[14]) confirmed that the levels of vibration on rock drills are high enough to cause an enhanced level of risk of vibration-induced disorders in a significant proportion of operators. Despite a trend towards mounted drilling there are still many applications where hand held drilling is appropriate. With 60 000 of these drills presently in operation in South African mines, there is a need for rock drill handles that can considerably reduce the levels of vibration to which rock drill operators are exposed.

Conventional approaches to reducing the vibration levels at the handle of a tool, such as the use of vibration isolators, result in handles with a high mass and low stiffness that hamper the handling of the drill. Therefore an alternative approach is required.

In the present study, the concept of using a tuned vibration absorber was explored and the feasibility of such a solution is demonstrated.

Rock drill vibration

Acceleration measurements on rock drill handles indicated extremely high levels of vibration, up to ten times the level that other developed countries generally regard as the point where action should be taken. Since the human body is not equally sensitive to vibrations at all frequencies, ISO 5349[15] suggests a frequency-weighting curve that provides for zero attenuation up to 16 Hz and

* Dynamic Systems Group, Department of Mechanical and Aeronautical Engineering, University of Pretoria, South Africa.

† Department of Mechanical Engineering, University of Stellenbosch, South Africa.

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then reduces the gain by 6 dB/decade. This emphasizes the importance of the low frequency content of the vibration signal and suggests that vibrations at frequencies exceeding about 1 kHz do not significantly contribute to vibration-related disorders. Moreover, it is well known that higher frequencies can easily be attenuated by taking conventional passive measures, such as the use of gloves or cushioned handle grips.

In applying ISO 5349 to typical measured handle acceleration signals, it is clear that the single most important contributor to vibration-related disorders is the vibration at the rock drill's operating, or percussion frequency (typically between 25 and 40 Hz). This is illustrated in Figure 1 for the conventional basicentric co-ordinate system depicted in Figure 2. It is also clear that the vibration is most significant in the direction that the drill operates.

This observation suggests the possibility of using a vibration absorber tuned to the drill's operating frequency, to reduce the vibration levels at the handle. A vibration absorber is essentially a secondary spring-mass system, which is fitted to the original device and tuned so that the new system forces a node, or region of low vibration, at the position where the vibration needs to be minimized. In this case this position will be the handle of the rock drill. The system utilizes the inertia of the absorber to force the handle to remain at rest at one specific frequency of isolation, which should coincide with the percussion frequency of the drill.

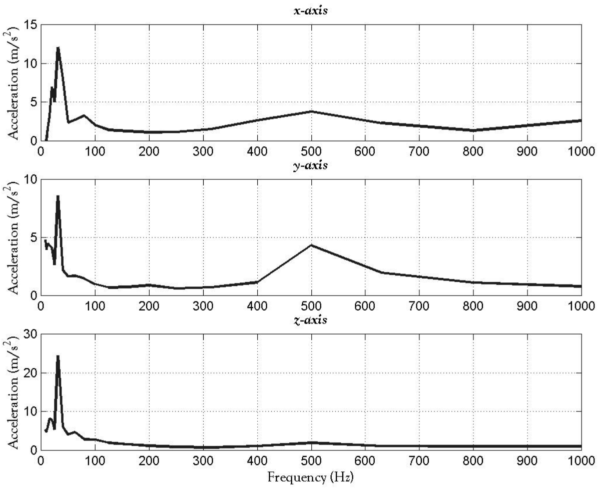

Figure 1—Frequency-weighted acceleration measured on a typical rock drill

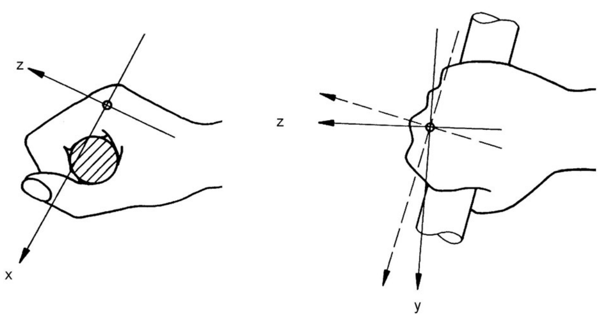

Figure 2—Basicentric co-ordinate system for the hand

Stiff vibration absorbers

A disadvantage of the conventional tuned mass absorber is that in order to get the isolation frequency down to the typical operating frequencies of a rock drill, the isolation mass has to be large, or the stiffness very low. This has little advantage compared to conventional passive isolation.

Flannelly[16] investigated the Dynamic Anti-resonant Vibration Isolator (DAVI) for the purposes of isolating a helicopter's main rotor. The DAVI functions on a lever arm principle to attenuate vibration levels at an isolation frequency, without compromising stiffness. However, such a configuration is not suitable for a rock drill because of the mechanical lever arms and bearings utilized in the DAVI.

More recently NASA (Halwes[17]) developed a hydraulic equivalent of the DAVI system known as LIVE (Liquid Inertia Vibration Eliminator). Figure 3 shows a further development of this concept.

The mass m_1 of which the vibration needs to be reduced, is connected to a moving port in a hydraulic assembly. Rubber elements connect the port to a housing. These act as sealing elements and also provide structural stiffness to the system. Any low-viscosity fluid can be used as the hydraulic fluid, but a heavy liquid gives a more compact system.

When relative movement is initiated between the handle and the housing, the working fluid is forced through the port, and acts as a counterweight to reduce the vibration levels. The area ratio of the housing to port diameter introduces a lever effect and plays a major role in calculating the frequency of maximum displacement and the absorption frequency.

In this system it can be shown that the displacement transmissibility T_r, which is the ratio of the handle vibration to the drill vibration, can be given by

$$T_r = \frac{X_1}{X_3} = \frac{k_1 + ic_1\omega - M_{eq3}\omega^2}{k_1 + ic_1\omega - M_{eq1}\omega^2}$$

where

$$M_{eq1} = m_1 + \left(\frac{b-a}{a}\right)^2 m_2$$

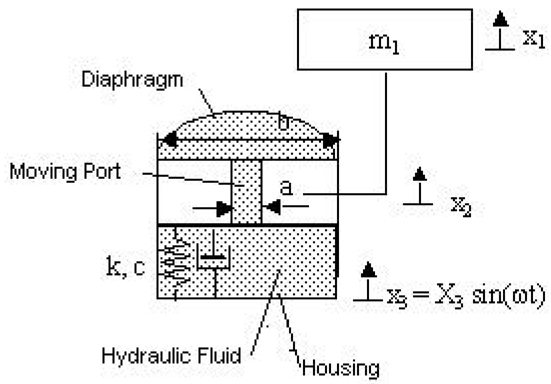

Figure 3—Diaphragm-type absorber

$$M_{eq3} = \left(\frac{b(b-a)}{a^2}\right)m_2$$

and k_1 is the effective stiffness of the absorber, c_1 the effective viscous damping coefficient, m_1 is the effective handle and hand mass, m_2 the mass of the working fluid in the port and a and b are the port and housing diameters of the absorber system. v is the system operating frequency in rad/s.

For parameters typical of a rock drill application (given in Table I), T_r can be plotted as shown in Figure 4. It is clear that this particular system has been tuned for minimum transmissibility at about 33 Hz. It is also clear that absorbers such as this display a resonance condition at another frequency. This is an important feature of all such systems. It is also evident that the frequency of the maximum displacement ratio is lower than the frequency of the minimum displacement ratio (isolation frequency ratio). This is addressed again below.

Design studies

Various sensitivity studies have been conducted to investigate the effect of design parameters on the minimum displacement transmissibility at the isolation frequency. One of the major challenges in designing an absorber of this nature is to minimize the internal damping of the system. It is clear from Figure 5, which is typical of these sensitivity plots, that small increases in the system's damping factor will significantly increase vibration transmissibility at the isolation frequency. Apart from choosing a working fluid with a favourable dynamic viscosity, particular care has to be taken in designing the flow path to reduce turbulence. Obviously an increase in the damping will reduce the resonance effect, which is beneficial, but this is less important because the design should ideally be tuned so that the isolation range coincides with that part of the spectrum where the excitation is most significant.

Another important factor influencing the transmissibility of the absorber, is the ratio g of the frequency of maximum transmissibility to the isolation frequency. Figure 6 depicts the transmissibility of the absorber versus g as a function of the ratio of frequency (v) to the frequency of maximum transmissibility (v_{MT}).

Figure 6 indicates that the minimum transmissibility will decrease as g decreases. Although there are various parameters influencing g, the most important of these are the port and housing diameters. Figure 7 illustrates how these two parameters affect g. The g ratio approaches 1 as the ratio of the housing diameter to the port diameter increases. It also indicates that it is difficult to move the isolation frequency lower than the frequency of maximum transmissibility.

The most important characteristic of the present absorber is that the isolation frequency can be very low, even for a

Table I

Design parameters

Design parameter	Design value
Effective stiffness k_1	100 kN/m
Effective damping c_1	100 Ns/m
Handle and hand mass m_1	2.5 kg
Fluid mass m_2	0.01 kg
Area ratio	20.25

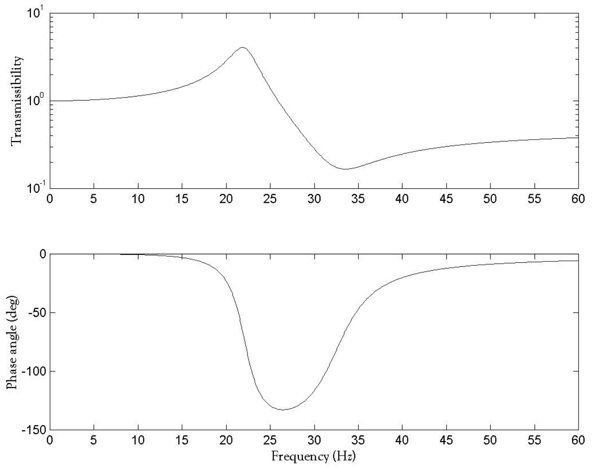

Figure 4—Transmissibility of absorber

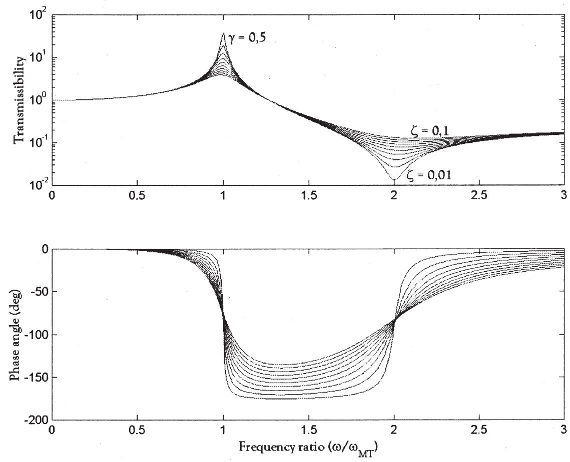

Figure 5—The effect of damping on the transmissibility of the absorber

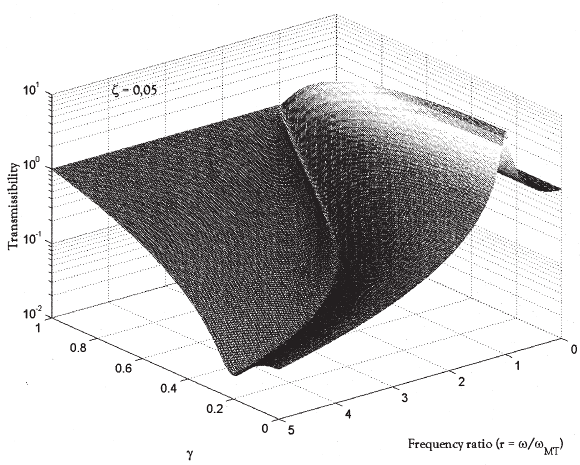

Figure 6—Transmissibility versus g as function of the frequency ratio *r*

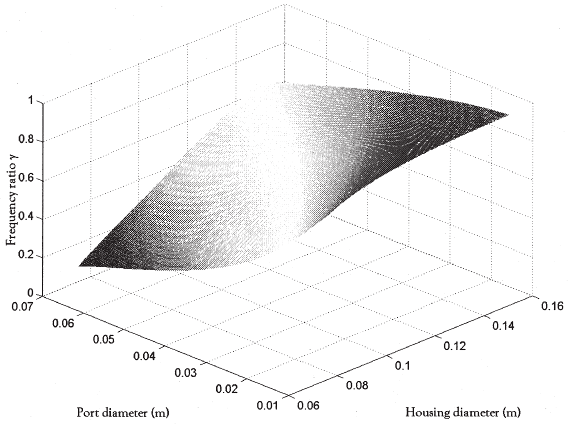

Figure 7—Frequency ratio g as functions of the port and housing diameters

stiff system. Figure 7 indicates, however, that there is an inherent limitation on how low the isolation frequency can be for a certain stiffness level. If the hydraulic lever ratio has to be increased to accommodate a greater stiffness, transmissibility at the isolation frequency will increase, and the absorber will be less effective.

Simulation

A mathematical simulation was done where the response of the mathematical model of the absorber was simulated, using a typical rock drill acceleration signal as measured by Van Niekerk *et al.*[14] as input into the system. This signal is shown in Figure 8.

Using a system with a stiffness of 100 kN/m, a handle mass of 1.7 kg, a port diameter of 20 mm, and a housing diameter of 90 mm, it can be shown that the isolation frequency is about 30 Hz, and the transmissibility at that frequency is about 30 per cent. Using this theoretical system in conjunction with the measured acceleration, it is shown that significant reduction of the drill response levels may be achieved through the application of such an absorber (as indicated in Figure 9).

Although the drill acceleration is indeed slightly amplified at approximately 20 Hz, more than half of the drill

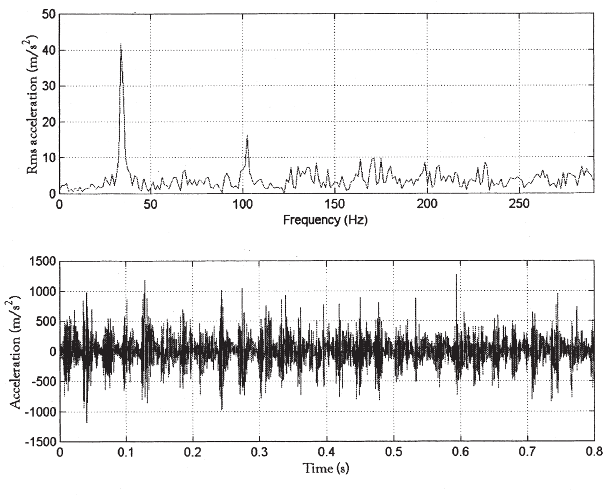

Figure 8—Rock drill vibration input signal

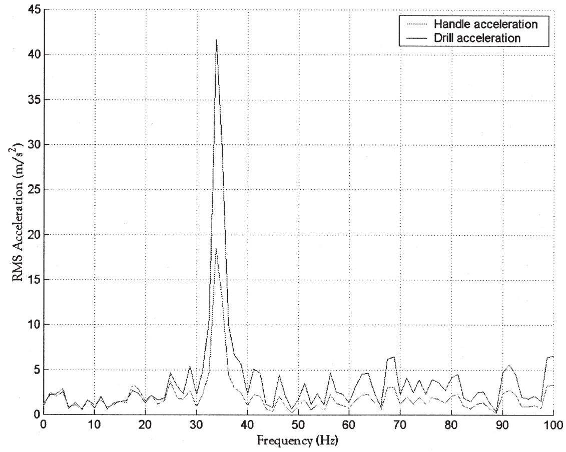

Figure 9—Spectrum of simulated handle response

acceleration has been absorbed at the operating frequency of the rock drill.

Experimental prototype model

Based on these numerical investigations, an experimental prototype model was designed and constructed to prove the viability of using an absorber for the reduction of rock drill vibration at the operating frequency. This design was specifically configured to allow the easy adjustment of parameters such as port diameter, the replacement of parts, etc. for exhaustive comparison with theoretical models. For this reason, no attention was given to lateral stiffness, robustness and other more practical considerations. Figure 10 displays a drawing of this absorber.

Measurement procedures and results

To prove the concept, the displacement transmissibility of the manufactured absorber was measured on a servo-

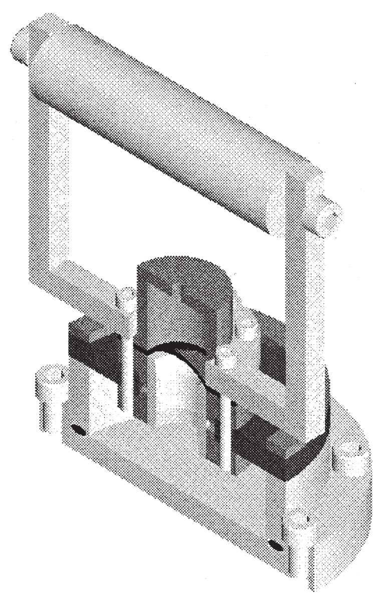

Figure 10—Solid model of experimental prototype model

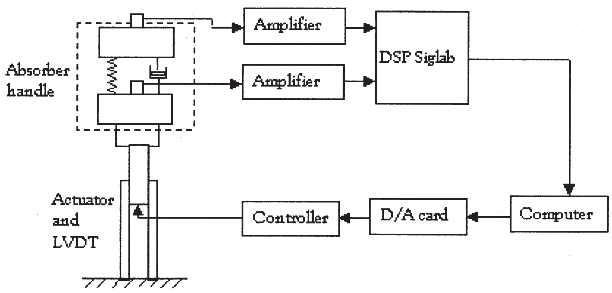

Figure 11—Transmissibility measurement

hydraulic actuator using a random input signal, band-limited between 5 and 90 Hz and with an rms amplitude varying between 0,5 mm and 1,5 mm (Figure 11). Accelerometers were attached to the handle and the absorber base and the transmissibility functions were determined by means of a DSP SigLab FFT analyser. The mass of the handle was modified with a mass representing the effect of a typical operator's hand, using an apparent mass model presented by Griffin[13].

Typical transmissibility measurements are given in Figure 12(a). The first configuration has an isolation frequency of approximately 42 Hz and a port diameter of 20 mm. This gives a transmissibility of 15 per cent. By reducing the port diameter to 12,5 mm, the isolation frequency can be lowered to 26 Hz, as shown in Figure 10. However, this reduces the efficiency of the absorber and increases the transmissibility to 35 per cent

As stated above, the design of an absorber with an isolation frequency lower than the frequency of maximum transmissibility can prove difficult. The fact that the frequency of maximum transmissibility occurs before the frequency range where the vibration is attenuated could be troublesome, especially during the start-up or hole-collaring procedures when the drill operates at frequencies lower than the normal operating frequency.

Another related issue in the design of such an absorber is the fact that the operating frequency of a rock drill may

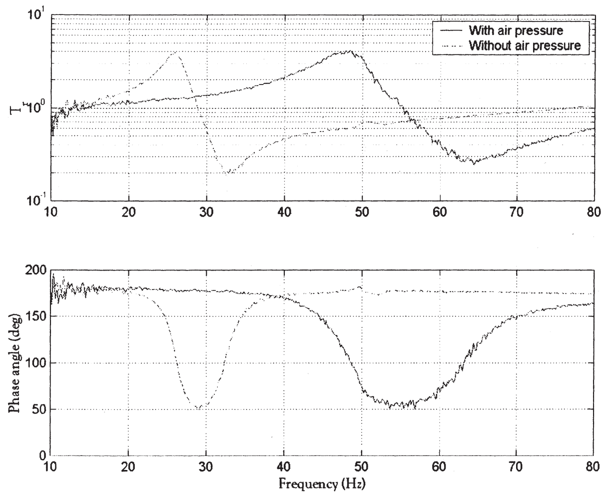

Figure 13—Typical effect of applied air pressure on the measured transmissibility

vary by up to 10 per cent because of variations in air pressure, rock hardness, drill settings and conditions, etc. Although the trough in the transmissibility function is relatively wide at the isolation frequency, as some damping is always present despite efforts to reduce damping as much as possible, the best solution would be to adjust the isolation frequency to fit the new excitation frequency. This can in principle also be accomplished by applying pressure behind the absorber diaphragm and in this way, changing the system characteristics.

This is illustrated in Figure 13 where an arbitrary pneumatic pressure of 40 kPa was applied behind the absorber diaphragm, by way of illustration . Significant changes to the transmissibility function may be observed.

Applying pressure, the frequency of minimum transmissibility changes to about 60 Hz. Although the transmissibility below the hypothetical drill's operating frequency (in this case about 33 Hz as implied by the transmissibility diagram) is still larger than 1, it is considerably less than that of the absorber without air

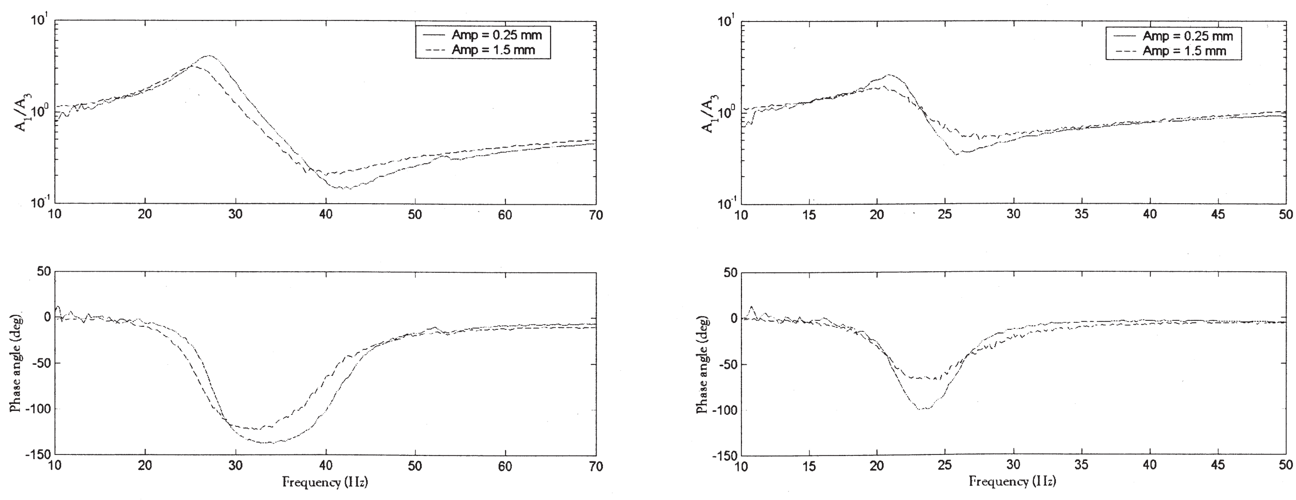

Figure 12—Measured transmissibility

pressure adjustment. Once the collaring is completed, the pressure may be released again to get the advantage of operating at the anti-resonance condition.

Conclusions

It is known that rock drill vibration levels are sufficiently high to increase the risk of vibration-induced disorders in a significant proportion of operators. Since it is difficult to attenuate this vibration, especially at the drill's operating frequency, a new approach is required.

The present study investigated the feasibility of using a tuned vibration absorber to reduce these vibration levels. An attenuator system based on a proper mathematical model of a liquid inertia absorber was designed and the concept was shown to perform well, at least in theory.

A system was then built and tested. In essence, this system performed in accordance with the mathematical model. The verified mathematical model may therefore now be used with confidence to design this type of absorber.

Experimental results indicated that transmissibility could be reduced to between 20 and 40 per cent of the non-attenuated rock drill handle. It has been shown in practice that there is enough stiffness in the system to facilitate the normal operation of the drill. This means that the operator will still have adequate control over the drill but will be subjected to lower levels of vibration.

Resonance might be a problem during collaring and start-up. Since the frequency of maximum transmissibility is difficult to position above the isolation frequency, it was proposed that these conditions could be accounted for by shifting the frequency of maximum transmissibility higher up the frequency spectrum during collaring and start-up. The results have shown that increasing the air pressure behind the diaphragm by about 40 kPa can shift the frequency of maximum transmissibility upwards by about 30 Hz on the frequency spectrum. This resulted in a close to unity transmissibility in the zero to operating frequency band. The absorber may in this way be 'switched off' during the collaring or start-up procedure.

It can be concluded that vibration absorbers offer exciting possibilities for reducing the vibration levels of rock drills and that further research and development should be done to solve the numerous practical problems associated with this technology. These include compact packaging of the absorber, ensuring acceptable robustness and dealing with moments caused by lateral forces applied to the handle.

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