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Damping of Low Frequency Pressure Oscillation

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**ABSTRACT**

A hydraulic line between a hydraulic pump and an actuator comes under much stress when a pulsatile actuator is used. It causes a huge pressure oscillation in the hydraulic system and thus decreases reliability. For example in a rock drill transferable hydraulic power is tens of kilowatts, so efficiency is also significant. The operating frequency of the rock drill is usually between 30 and 60 Hz. The frequency of the pressure oscillation is then very low compared for example to pressure oscillation of a hydraulic pump.

The aim of this thesis is study pressure dampers when the damped frequency is low and when they work as a part of a wider system. After theoretical inspection, pressure dampers were simulated and then different dampers were tested with a pure sine wave pressure oscillation and then with the real rock drill. Different sizes of accumulators, a Helmholtz resonator, a T-pipe and an inline suppressor were used in tests.

Dimension theories of dampers worked even though the operating frequency was low. The simulation models used operated well enough so that they can be utilised for a definition of the pressure damper.

The accumulator damped pressure oscillation of the rock drill well when it was installed near the hydraulic main line but it didn’t damp pressure oscillation of the rock drill as expected when it was tuned by the “natural frequency” method. The T-pipe was easy to tune and the damping capacity was promising even with the rock drill. The Helmholtz-resonator damped well, but the size of the damper was too big for mobile hydraulic applications. The inline-suppressor didn’t damp pressure oscillation between 30-60Hz much but it damped high pressure oscillation (1200Hz) well.
PREFACE

This work was carried out at the Institute of Hydraulics and Automation (IHA) at Tampere University of Technology during the years 1999-2006. The study was done within the Power Engineering Graduate School (PEGS) programme, funded partly by the Ministry of Education.

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NOMENCLATURE

a speed of sound [m/s]

A_{12}, A_{34} areas of connection pipe [m^2]

A_3 area of connection of Helmholz resonator [m^2]

B_e effective bulk modulus [Pa]

C_a hydraulic capacitance [m^3/Pa]

D diameter of fitting [m]

D_{12}, D_{34} diameters of connection pipe [m]

E modulus of elasticity [Pa]

f natural frequency [1/s]

L length [m]

L_a hydraulic inductance [kg/m^4]

L_1 length of first supply hose [m]

L_2 length of second supply hose [m]

L_{12}, L_{34} length of connection pipes [m]

L_3 length of connection of Helmholz resonator [m]

L_T length of T-pipe [m]

n odd number (1, 3, 5...) [-]

p_0 pre-charge pressure of accumulator [Pa]

p_1 lower limit of pressure of accumulator [Pa]

p_2 upper limit of pressure of accumulator [Pa]

p_a pressure in accumulator [Pa]

p_B pressure of branch fitting [Pa]

Q flow rate [m^3/s]

Q_B flow rate of branch fitting [m^3/s]

Q_{st} steady-state flow rate [m^3/s]

s Laplace operator

u amplitude of flow ripple [m^3/s]

t time [s]

t_{lowerlimit} lower limit of integral [s]

T_{upperlimit} upper limit of integral [s]
V     volume                     [m$^3$]
V$_0$ volume of nitrogen at pre-charge pressure [m$^3$]
V$_1$ volume of nitrogen at steady-state pressure of systes [m$^3$]
V$_2$ volume of nitrogen during dynamic process [m$^3$]
V$_a$ volume of nitrogen in accumulator [m$^3$]
V$_{Hi}$ volume of chamber of Helmholz resonator [m$^3$]
$\Delta$V change in effective volume of accumulator [m$^3$]
x$_1$ distance between actuator and damper [m]
x$_2$ distance between actuator and sensor [m]
y$_1$ length of connection of damper [m]

$\kappa$ polytrophic exponent [-]
$\omega$ natural angular velocity [rad/s]
$\rho$ density of oil [kg/m$^3$]

BSL Best Straight Line definition
FS full scale
NS nominal size
1 INTRODUCTION

There can be a single waterhammer (pressure peak) or continuous pressure oscillation in the hydraulic line. A single waterhammer arises when a column of fluid flowing in a line stops suddenly due to a valve closure or when a hydraulic actuator stops suddenly [Merrit -67]. Continuous pressure oscillation in a line forms if the flow at the end or at the forepart of the line varies. If the flow varies at the forepart of the line the reason is usually the hydraulic pump [Viersma -80]. The flow can be varied also at the end of the line if a hydraulic actuator works periodically. This study concentrates on this last case.

1.1 Research problem

The main problem in a hydraulic system with a pulsating actuator is the reliability of the supply line arrangement. Usually the weakest link in the supply line arrangement is the hose. Stress forms in the hose because

- the hose shakes and bloats due to the pressure and mass flow oscillation
  - the movement stresses the structure of the hose
  - the hose warms [Ijas -01b].
    - warming due to external abrasion
    - warming due to bloating and vibration of the hose structure
- there is continuous pressure oscillation in the hydraulic line (many frequencies at the same time)
  - pressure oscillation stresses the structure of the hose and fittings
- there is continuous static power flow through the hydraulic line
  - continuous pressure
  - continuous internal flow (friction of mass flow)
- there are many external loads
  - oscillations from the boom, from other hydraulic drives, from the main engine, etc
  - temperature of the environment
  - dust and falling stones
  - rotation and sticking of the hose
  - the hose must bend at the joints of the boom
When the pressure oscillates and the power level is several dozen kW, the hydraulic hose is at an extreme limit. It is not possible to double or triple the hydraulic power using this kind of traditional hose system. If it is desired to increase the flow rate, bigger hoses are needed. The maximum working pressure is lower in bigger nominal-size hoses. It is also possible to increase power by raising the pressure level. However, a high-pressure hose of big nominal size is very expensive and stiff. The fittings can be tighter with hoses having a higher pressure level and this tightness causes extra pressure loss in the hose arrangement [Ijas -01a].

The hydraulic hose can be replaced by a pipe, but a pipe is not trouble-free either. If the actuator, for example the rock drill, is at the end of a boom, there are many joints which need hoses in any case. A hose works as a damping element, whereas a pipe is very stiff. The higher pressure oscillation is conducted from the actuator to the pump when a pipe is used [Jääskelä - 02]. This is not a desirable property, because the lifetime of a hydraulic pump can be shortened.

With full regard to above, the research problem is to find a pressure damper which is reliable, cheap and small. It must operate well enough at low frequencies.

1.2 Objectives of the thesis

This study utilizes earlier studies in which different dampers were simulated and measured as a single component. The definitions of studied dampers were obtained by using simply definition equations from earlier studies. In this study the damper is a part of a wider system.

Objectives of this thesis are

- to study how pressure dampers operate and how dimension theories work at low frequencies as usually they are used at higher frequencies
- to study is the operation of pressure damper sensitive to form of pressure wave (sine wave or irregular wave)
- to clarify what kind of damper is best for rock drilling machines (reliability, damping capacity, size, price)

Pressure oscillation and related problems of reliability are commonly known and accepted factors in rock drilling. If the supply hose breaks once per week, the user does not see any
problem in the drilling system. The situation is the same in many applications where the load oscillates considerably. However, better reliability and a constant supply pressure is always desired feature.

Novelties of this study are shown in Figure 1-1. Typically the source of the pressure oscillation is a hydraulic pump and then amplitude of pressure oscillation is quite low, the frequency is higher than 60 Hz and the pressure wave is more or less ordinary. It is also noteworthy that the transferable hydraulic power is several dozen kilowatts when a rock drill is used.

![Diagram showing pressure oscillations and hydraulic actuator](image)

Figure 1-1 Novelties of this study.

1.3 Research methods

The study of the damper has been divided into four parts:

- Dimensioning of the studied damper at low frequencies using the existing definition equation
- Simulation of the studied damper. The purpose of simulations was to ensure that the operating band of the dampers was sufficient and that the dimensioning of the dampers was correct.
- Experimental simulation of the studied damper using a servo valve which imitates a pulsating hydraulic actuator. The frequency and amplitude of the pressure oscillation
and the operating pressure could be varied when the servo valve was used. The aims of
the experimental simulations were to study the effect of pressure level on damping and
operating band.

- Testing of the studied damper using a rock drill. The situation was more practical with
  the rock drill. The frequency of the pressure oscillation was a function of pressure and
  the pressure amplitude was uneven. In this case there were also several disturbances
  (mechanical and hydraulic vibrations) from the rock drill and a power unit.

If it is desired to study the hose-rock drill combination, the fluid-filled hose is an essential part
of this system. There have been few studies in this research area and according to those studies
[Drew -97, Drew -98, Longmore -97] a good simulation model of a fluid-filled hose is very
complicated and is always a semi-empirical model. Parameters of an impedance matrix must be
measured for every hose.

On the other hand, good results have been reached using analytical pipe models with hoses,
even though the flow was turbulent [Leino -01]. In this study, the pipe models are constructed
using the modal approximation method [Mäkinen -00]. There are several references in which
the correlation of the modal approximation method of Mäkinen has been excellent [Kajaste -
99a, Kajaste -99b, Leino -01, Haikio -00].

The measurement of absolute pressure oscillation is very difficult [Koivula -01]. The result is a
function of the length of the measurement hose, of the volume of the measurement fitting, of the
sensor type, etc. Therefore several measurements were done and these were compared with
each other. All comparisons were made using a similar test installation and sequence.

Two damped frequencies, 33 Hz and 50 Hz, were used. 33 Hz oscillation is a typical idle
frequency of the rock drill and 50 Hz oscillation is a typical operating frequency of the rock
drill. 50 Hz is more important in a real work but testing with the real rock drill drilling is
difficult. Testing with 33 Hz oscillation is easy with an idle rock drill and the results are
sufficiently comparable.
1.4 Restrictions on scope of work

This study concentrates on the hydraulic line between the hydraulic pump and the hydraulic actuator. The hydraulic actuator and the hydraulic pump are not the object of this study. It is not possible to vary the length of the hydraulic line. This study does not address how the total efficiency of the actuator changes when the pressure oscillation in the supply line has been damped.

A potential damper must be usable for serial products. This means that the damper must be cheap, light, small and easy to maintain.

1.5 Contents of thesis

This thesis contains seven chapters and two appendices. The contents of the chapters are briefly discussed below.

Chapter 1 explains the problem of the hydraulic supply line when the actuator is pulsating significantly.

Chapter 2 reviews the subject of pressure damping.

Chapter 3 introduces the principle of rock drilling and studies the properties of a supply hose. The simulated and measured pressure response of a supply hose is presented.

Divergent –type dampers are studied in Chapter 4. The accumulator, the Helmholtz resonator and the T-pipe are simulated and measured.

Chapter 5 studies the inline suppressor.

In Chapter 6 the main results of this work are discussed. Also a measurement result is given for a situation where the accumulator and the inline suppressor are installed in the rock drill system at the same time.
The main results of this study are summed up in Chapter 7 and some conclusions regarding future work are also given.

All measurement components are collected in appendix A, and in appendix B the simulation models are presented.

### 2 STATE OF THE ART IN PRESSURE DAMPING

Pressure oscillation and its consequences, i.e. noise and reliability problems, have been studied widely in hydraulics. Characteristic of these studies has been the aim to damp quite high frequencies, which produce a noise. Generally, the source of the pressure oscillation and of the noise is a hydraulic pump. A good summary of the noise problem in fluid power is found in reference Edge -99.

The classification of dampers is according to that of Esser [Esser -96, page 32]. Dampers have been divided into three types:

- reflection and interference dampers
- absorption dampers
- active dampers

Reflection and interference dampers have been divided into two types:

- divergent-type dampers
- serial dampers

This study concentrates on reflection and interference dampers because absorption dampers are not suitable for low frequencies [Esser -96, Table 4-1] and active dampers are expensive [Esser -96, Table 4-1].
2.1 Divergent type dampers

2.1.1 Accumulators

Probably accumulator-type dampers are those in most general use. The operation principle is that the accumulator absorbs the pressure oscillations in a compressed gas. Part of the storage energy is restored back to the hydraulic system and another part is converted to a heat loss [Mordas -94]. A consequence of this is one of the problems of the accumulator: the accumulator warms. Due to the warming and the moving of the membrane of the accumulator, the accumulator requires maintenance. Depending on the application, the required space might be a problem. The accumulator is installed so that it is branched away from the hydraulic mainline and in this case it also needs satisfactory support.

Viersma [Viersma -80] noticed that the connection line is an essential part when the properties of the accumulator are defined. Kajaste [Kajaste -99a] derived the nonlinear model of an accumulator with friction. The throat connection is also defined more exactly. Edge [Edge -91] defined the impedance characteristic of accumulators and mentioned that capacitive effect is dominant at low frequencies, while inductive effect is dominant at high frequencies. If fluid-borne noise analysis is done, the impedance characteristics can be modelled as a pure inductance. Larsson [Larsson 1987a] simulated the damping characteristics of accumulators and also an accumulator-ECA-damper combination.

In the reference [Garbacik -95] it is mentioned that the highest possible damping frequency is 100 Hz in a typical commercially available hydro-pneumatic accumulator and that a frequency range higher than 100Hz is only possible with specially designed accumulators. Ortwig [Ortwig -99] mentioned that the best operating band for accumulators is 50-300 Hz. At higher frequencies the limiting factors are inertia of the membrane and narrowness of the connection port. If an accumulator is planned for a damp pressure oscillation, there is a wide, special-design connection port (a so-called pressure damper).

2.1.2 Helmholtz resonator

The working band of the Helmholz resonator is slightly wider than that of the T-pipe [Larsson -87a, Viersma -80, Washio -01]. The main problem, as with the T-pipe, is the big size of the damper when low frequencies (30 - 60Hz) are damped. If better attenuation is required, two resonators can be installed in the hydraulic system [Washio -01].
2.1.3 T-pipe
The T-pipe damps odd multiples of quarter wave, so there are many operating bands. The damping capacity is good, but the operating band is quite narrow [Viersma -80]. The length of the T-pipe becomes long at low frequencies. Esser [Esser -96] proved that larger diameter of the T-pipe gives rise to a larger operating band (usually diameter is same as nominal size of the main line). From the basic T-pipe new T-filters have been developed in which there are special restrictors inside the T-pipe [Ichiyanagi -99] or there are several T-pipes with different lengths and position: the so-called π-shaped filter [Washio -01]. In this case the working band is wider.

2.1.4 Mass-spring damper
A different kind of mass-spring damper is used also for the divergent-type damper. Characteristic of this kind of damper is that the damping capacity is good but the operating band is quite narrow [Mikota -00]. Probably external excitation and movement of the actuator disrupt the operation of the mass-spring damper.

2.2 Serial dampers

2.2.1 Expansion chamber attenuator (ECA)
Like the T-pipe and Helmholtz resonator, the ECA damper is also very simple structurally; an extra chamber in line. There are studies on ECA attenuators consisting of one chamber in line [Larsson -87a, Ortwig -99] or one chamber with a mode cancellation pipe [Larsson -87a], and also on an attenuator with two chambers [Larsson -87a, Larsson -87b]. Larsson states that if the disturbance is wide-banded, an expansion chamber attenuator, preferably with two chambers, is an excellent choice to damp pressure oscillation. Obviously pressure loss is higher with two chambers or with mode cancellation pipe attenuators than with one chamber attenuator.

The operating band of an ECA is widest compared with the Helmholtz resonator and T-pipe. Maybe an ECA is the most interesting damper from the point of view of trying to damp pressure oscillation of a hydraulic pump. An ECA is at its best at high frequencies, not at frequencies less than 100 Hz. For example, when Ortwig measured an ECA damper of length 66 cm and diameter 10cm, its lowest damping frequency was 600 Hz.
2.2.2 Inline suppressor

For higher frequencies there are also accumulators of the tube-model type. This means that a membrane is like a tube and the oil flows through the accumulator [Hunt -96, Wilkes -98, Kajaste -01]. This kind of inline suppressor works at higher frequencies than a standard accumulator [Kajaste -01] and that is why one manufacturer has termed this accumulator a noise suppressor [Wilkes -98].

2.3 Properties of hoses

2.3.1 Analytical models of pipe

The first studies on unsteady flow were done in the middle of the nineteen century, so the pipe model has a long history. Flow processes are governed by an equation of motion and an equation of continuity. Their solution leads to an equation for pulse wave propagation of disturbances in flow and pressure throughout piping system [Wylie-93]. Wylie classifies the method of analysis of unsteady flow as follows:

1. Arithmetic waterhammer
2. Graphical waterhammer
3. Characteristics method
4. Algebraic method
5. Implicit method
6. Linear analyzing methods
7. Other methods

The arithmetic waterhammer calculates the steady-state situation plus water hammer effects at a certain time and place. The graphical waterhammer takes friction into account by a correction. This method had been used after the arithmetic waterhammer method between 1930 and 1960 [Wylie -93].

The characteristics method converts the two partial differential equations of motion and continuity into four differential equations. These equations are then expressed in finite difference form. This method is usually the basis of pipe models of fluid power nowadays.
Algebraic equations are used when two or several pipes are in series. The arithmetic waterhammer is the basis. Although the implicit method is intended for unsteady free surface flow calculations, it has also been used in other applications. The linear analyzing method linearizes the friction term and drops other nonlinear terms in the equations of motion. There are also several other methods for calculating a certain situation.

As mentioned, the characteristics method is the most popular method in use in fluid power pipe models. In this method the basis of the pipe model is two-dimensional viscous flow, where the frequency dependence of friction is taken into account using the instantaneous velocities and the past changes in calculations [Zielke -68].

There are also several models which are based on the modal approximation method. The modal approximation method is the further developed version of the method of characteristics. The model of Mäkinen [Mäkinen -00] is based on a two-dimensional viscous model where the dynamics of the pipe are approximated using rational transfer functions. When the model is a four-pole representation type, it is easy to connect to other hydraulic components. Three different pipemodels are available depending on the boundary conditions of the hose. The PQ-model is for a situation when pressure at the input end of the hose and the flow rate at the output end of the hose are known. In the P-model, pressures are known and in the Q-model flow rates are known as initial values. According to Kajaste [Kajaste -99b] model of Mäkinen works excellent if the “viscosity factor” is low. In this case the viscosity factor was low (calculated in the appendix B).

2.3.2 Properties of hose structure

The travel of the pressure ripple through the hose is a complex phenomenon. An axial, a torsional and a lateral force are developed in the hose structure [Drew -97, Drew -98, Longmore -91b]. To study the pressure ripple in the hose, an impedance matrix of the hose must be determined. Then it is possible to study flows and pressures by 2*2 impedance matrix [Drew -98] and flows, pressures, forces and velocities by 4*4 matrix [Drew -97] as a function of the frequency.

In practice, the parameters from which the impedance matrix is formed are difficult to change. In some cases the length of the hose can be varied, which significantly changes the pressure
amplitude at a certain frequency [Drew -95]. At least must be concerned that the nominal frequency of the hydraulic line and the frequency of the source of the pressure oscillation are not near each other [Hunt -96, Mannesmann Rexroth -88]. Different hose structure leads to a different impedance matrix [Longmore -91b] and different hoses transmit vibration in a different way [Longmore -91a].

In the reference [Longmore -97] the effect of the hose lining on the impedance matrix is studied. The material properties are derived from measurements of an impedance matrix relating pressure and axial wall compression to flow and axial wall velocity at each end of the hose. [Muto -98] concentrated on studying properties of viscoelastic wall hoses. The aim was use a viscoelastic hose to damp pressure pulsation.

A certain kind of suppressor is obtained when a spiral steel strip is installed inside the hose. This kind of tuner is used to minimize the pump-generated pressure ripple of power steering [Drew -98, Drew -97].

2.4 Patents

There are several different registered patents on pulsation dampers, suppressors and attenuators. Perhaps the reason for all these patents is that when there is for example an absorption element (membrane), a throttling device (orifice), or a change of flow direction, these work as a damper. They damp more or less, but can cause also extra pressure loss in the system.

The patent [Sugimura -79] concerns a hydraulic accumulator in which the connection port is specially designed for pulsations. [Pursell -99], [Steffes -99] and [O-Boegel -95] are patents concerning the membranes of accumulators (shapes or structures). Patents [Weber -00] and [Machinesney -93] concern accumulators in which there are no traditional membranes and in which the divider elements are steel bellows tubes. In the patent [Backe -95] a traditional membrane and bellows tube are in the same damper.
The patent [Alven -85] is in principle one or more T-pipes in series with an extra connection at the end of the pipes. The patent [Hansen -80] is almost an inline suppressor, except that the outer diameter is not constant (conical shape) and the inner chamber is divided into several parts.

Usually there is some kind of pressure damper in a fuel injection system. In the patent [Kaneko -96] the significance of the connection element can be seen. There, membranes are connected to the system by a very wide connection fitting. The pressure surge is a problem also in many other applications. For example in a dairy, milk or yoghurt surges in a pipe as it flows. A solution for this is offered in the patent [Tetra Pak -97], where part of the pipe is very flexible hose. This “membrane” expands when needed and then stabilizes the flow.

2.5 Summary

If it is desired to study damping of pressure oscillation between 30-60 Hz, only the accumulator seems to offer possibilities. There are also a few special damper types which can be operated at such a low frequency [Mikota -00, Tetra Pak -79]. Usually the best operating band is at a considerably higher frequency.

As mentioned, almost all dampers are designed to damp pressure oscillation of a hydraulic pump. Usually the damped oscillation is a few hundred Hertz and the amplitude is quite low. Such a frequency means that the size of the damper required is quite small. The situation is different when pressure oscillation is between 30-60 Hz. If the source of the oscillation is the hydraulic actuator, the damper must be installed near it and usually there is not much free space.

It is possible to damp pressure oscillation using a pressure damper but there are also other ways to affect pressure oscillation. The fluid affects the sensitivity of oscillation, especially the concentration of free air. Also the pipe system, the supporting of the pipe and the fittings used affect the oscillation [Kwong -98].
3 INTRODUCTION OF CASE – ROCK DRILLING

3.1 Principles of rock drilling

Rock breaks due to the load of an impact on a drill bit. When impact loading occurs using a button bit, the rock becomes rock powder or the stress causes deformation and local compaction in the rock, which leads to stress-initiated cracks. The rock breakage efficiency is a function of the load on the bit (percussion and feed force), bit geometry, bit rotation and flushing.

![Figure 3-1. Mechanism of rock breakage in percussion drilling [Puhakka -97].](image)

3.1.1 Top-hammer drilling [Puhakka -97]

The principle of top-hammer drilling is shown in Figure 3-2. The rock drill can be hydraulic or pneumatic powered. There are four basic functions of percussion drilling:

- percussion to indent the button
- feed to ensure bit-rock contact
- rotation to index bit indentation
- flushing to remove cutting and cool the tool
As the name “top-hammer” implies, the impact force goes through the drill steel. The impact force crushes rock and part of it is reflected back from the bit. Hydraulic drilling has a typical impact frequency of between 33-75 Hz. Percussion power is a function of pressure and oil flow. Percussion power can be dozens of kilowatts, depending on the drill steel size. There is a certain maximum capacity to transmit kinetic energy in steel, which restricts the maximum percussion power. Percussion is a compromise of penetration rate and drill steel economy.
Figure 3-3. Percussion dynamics [Puhakka -97].

Figure 3-4 shows the main parts of the hydraulic rock drill. The piston moves back and forth and generates the impact force. The distributing valve controls flow while accumulators store energy, damp pressure ripple and prevent cavitation.
3.1.2 Rotary percussive drilling

Percussion force is used to indent the tool in the rock in top-hammer drilling. When rotary percussive drilling is used, feed force is used to indent the tool in the rock. This means that the feed force must be very high. Rotary percussive drilling is most applicable in soft or semi-soft formations.

3.1.3 Down-The-Hole drilling

Down-The-Hole drilling is used if the percussive hammer is installed at the bottom of the hole. Because the piston is in almost direct contact with the drill bit, good efficiency is achieved in the process. A nearly constant penetration rate and good hole accuracy is reached in Down-The-Hole drilling. At its best, Down-The-Hole drilling is big-size drilling (up to 250 mm).
3.2 Properties of supply hose

3.2.1 Role of supply hose

It is clear that the hydraulic line between the hydraulic pump and the main hydraulic actuator is very important. Hydraulic power travels through the hydraulic line to the main function of a machine. All other functions are supplementary functions. If the hydraulic actuator is the rock drill, there is high pressure oscillation in the hydraulic line and hydraulic power can be even greater than 30 kW. The total length of the hydraulic line can be over 20 m and the actuator is installed at the end of the boom. This means that the supply line consists of several different parts (pipes, hoses, fittings and clamps). In this case there are number of fittings and at least 2 different hoses and every fitting causes extra pressure loss. In many times there are narrower fittings when the nominal working pressure level of the hose is higher. The influence of different diameter of the fittings is studied in the reference Ijas -01a.

When the hose vibrates and shakes due to pressure oscillation, the supply line needs special attention [Ijas -01b]. Warming can be prevented by using special abrasion resistant hose type and by attend to fastening of the hose. According to this study [Ijas -01b] the main reason of warming is the abrasion of the moving hose.

3.2.2 Measured pressure oscillation with rock drill

One aim of this work is to study the top-hammer drilling of Sandvik Tamrock and especially as regards the supply line for percussion. As shown in Figure 3-4, the piston moves back and forth during the operation. Even though there are accumulators inside the drill to stabilise pressure oscillation, there is continuous pressure oscillation in the pressure line. The supply line of the rock drill consists of a 15 m NS 1” hose + a 5 m NS ¾” hose. Measured pressure oscillation is shown in Figure 3-5. Pressure is measured near the power inlet of the rock drill.
In this case the rock drill did not actually drill the rock because the drill was idling. Running without a load is more demanding compared with running in a real situation. The pressure oscillation is greater when the piston hits a pad and not the rock. Also the operating frequency is different with idle operating. However, the phenomena are the same in both cases. The average flow rate was about $1.3 \times 10^{-3} \text{ m}^3/\text{s}$ ($80 \text{ l/min}$) and in this way hydraulic power was slightly over 21 kW.

Obviously the pressure oscillation, as Figure 3-5 shows, reduces the reliability of the system. One other result is that hoses shake and the wearing movement strains their structure. The oscillating pressure is conducted to the pump and disturbs the hydraulic system.

Figures 3-5 and 3-6 show that the lowest pressure peak is at the rock drill running frequency (33 Hz) and the second pressure peak is at double frequency of the rock drill (66 Hz). 1200 Hz is a natural frequency of the inlet conduit of the rock drill and this causes pressure peaks around this frequency.
3.2.3 Measured pressure response of supply hose using a servo valve

The pressure response of the hose arrangement was measured using a servo valve as a load. The measurement installation is depicted in Figure 3-7. First the settings of the pump and the servo valve were adjusted so that the pressure was 16 MPa (160 bar) and the flow rate $1.66 \times 10^{-3}$ m$^3$/s (100 l/min). When a steady state was achieved, the servo valve caused pressure oscillation. The servo valve was controlled with a sinusoidal signal. The frequency was changed evenly from zero to one hundred Hertz during 60 s. The amplitude of the excitation signal was $1.17 \times 10^{-4}$ m$^3$/s (7 l/min) and it was constant during measurements.

The distance between the hydraulic pump and a pressure filter (beginning of the supply hose) was 12 m and it was realized using a 2” steel pipe. The volume of the pressure filter was 1 litre.

![Figure 3-6: Power density of pressure oscillation using the rock drill.](image)
Different resonant points can clearly be seen. The places of resonant points are a function of the boundary values of hoses, a function of speed of sound and a function of the lengths of hoses. The hose arrangement works quietly if the resonant point and the working point are not close together.

According to Trainer of Rexroth [Mannesmann Rexroth -88] the resonance frequency of the column of fluid between the source of the excitation and the throttling component can be calculated as follows:
\[ f = \frac{n \cdot a}{4 \cdot L} \]  \hspace{1cm} (1)

\[ n = 1, 3, 5 \ldots \]

If the speed of the sound is 1100 m/s, the calculated resonance frequencies are 13.75 Hz, 41.25 Hz and 68.75 Hz (L = 20 m). It looks that the speed of the sound (1100 m/s) is a good educated guess. Resonance frequencies fit well on the right places.

The formula for definition of the speed of the sound is:

\[ a = \sqrt{\frac{B_e}{\rho}} \]  \hspace{1cm} (2)

The effective bulk modulus is:

\[ B_e = a^2 \cdot \rho \]

\[ = \left(1100 \frac{m}{s}\right)^2 \cdot 890 \frac{kg}{m^3} = 1077 \text{ MPa} \]

The effective bulk modulus is quite high, which indicates that the hose used was very stiff.

### 3.2.4 Simulated pressure response of supply hose

The tested supply line was modelled with a Matlab Simulink simulation program. The pipe model was realized by a modal approximation method [Mäkinen -00] and a simple adjustable orifice functioned as a throttle valve. The input excitation was a sine wave form change of the diameter of the orifice (Appendix B). The frequency was increased evenly from 0 to 100 Hz. The supply line arrangement consisted of three different pieces. First, there was a hydraulic pipe between the hydraulic pump and a pressure filter (see Figure 3-7). The pressure filter was modelled to have a volume 1 litre. The supply hose was modelled as two hoses with an orifice between them.
Figure 3-9. Measured (above) and simulated (down) pressure response of the supply hose at the downstream end of the hose.

The simulation calculates the resonant points in the right places very well. Only high frequency resonant points (over 70Hz) were not totally in the right places. However, the verification is so good that this simulation model can be used as a virtual test environment.
4 DAMPING OF PRESSURE OSCILLATION USING DIVERGENT-TYPE DAMPERS

4.1 Pressure accumulator as a pressure damper

4.1.1 Introduction
General instructions for dimensioning are given so that the hydraulic nominal angular velocity of the accumulator and the connection element can be calculated. In the reference [Viersma -80] the accumulator is calculated by one throat pipe and in the reference [Kajaste -99] the dimensions of connection elements are taken into account more exactly.

Edge [Edge -91] presents the nominal angular velocity equation clearly:

$$\omega = \frac{1}{\sqrt{C_a L_a}}$$  \hspace{1cm} (3)

where

$$C_a = \frac{V_a}{\kappa^* p_a}$$  \hspace{1cm} (4)

Viersma -80 and Edge -91 define inductance $L_a$ as

$$L_a = \rho \frac{L_{12}}{A_{12}}$$  \hspace{1cm} (5)

The form of the inductance of Kajaste -99 is

$$L_a = \rho \left[ \frac{L_{12}}{A_{12}} + \frac{L_{34}}{A_{34}} \right]$$  \hspace{1cm} (6)
Figure 4-1. An accumulator with a connection pipe.

The ratio of pressure and flow rate of the accumulator is (Kajaste -99a)

\[
\frac{Q_B}{p_B} = \frac{C_0 s}{\frac{s^2}{\omega^2} + 1}
\]  

(7)

4.1.2 Simulation of hydraulic system with accumulator

There are many studies in which the properties of the accumulator are simulated as a single component with reflectionless fittings [Viersma -80, Larsson -87a and Kajaste 99a]. In this study the accumulator functions as a part of a wider system. The accumulator is planed to damp pressure ripple which comes from the hydraulic actuator. The simplified hydraulic diagram of the studied system is depicted in Figure 4-2.
Equation 3 defines dimensions of the accumulator for the desired frequency. The aims of the simulation are to study the operating band of the accumulator and also to ensure the correct dimensioning. The simulation model of the supply line is shown in Appendix B and the model of the accumulator is based on Equation 7.

A simple adjustable orifice functioned as a throttle valve, which imitated a pulsating actuator. The input excitation was a sine wave form change of the diameter of the orifice. The frequency was increased evenly from 0 to 100 Hz.
The studied system imitated the rock drill. The supply line was identical to that in the real system and also the operating point was the same. It was attempted to make the flow rate and the pressure (steady state and ripple) the same as for a real rock drill. The damped frequency was 33 Hz, which corresponded to idle operating of the rock drill. The parameters of the accumulator were defined to damp 33 Hz oscillation according to Equations 3, 4 and 6 as follows:

\[
A_{12} = 2.0 \times 10^{-4} \, m^2
\]
\[
A_{34} = 7.9 \times 10^{-5} \, m^2
\]
\[
L_{12} = 0.28 \, m
\]
\[
L_{34} = 0.01 \, m
\]
\[
p_a = 160 \times 10^5 \, Pa
\]
\[
V_a = 0.385 \times 10^{-3} \, m^3
\]
\[
\kappa = 1.4
\]
\[
\rho = 890 \frac{kg}{m^3}
\]

The dimensions of the connection element were chosen and then the accumulator was tuned to damp 33 Hz oscillation by define right pre-charge pressure. The polytrophic exponent was 1.4 because it is a typical value when the process is fast. There are references where polytrophic exponent is 1.5 or even greater [Larsson 1987a, Watton 1995] but the influence of variation is quite small and 1.4 is a good average value for the fast process.
Figure 4-3 shows the simulated pressure oscillation at the end of the supply line. When the system without the accumulator (up part Figure 4-3) and the system with the accumulator (down part Figure 4-3) are compared, the influence of the accumulator can be seen. The operating band of the accumulator is wide enough and it seems promising for damping this kind of low frequency.
4.1.3 Measured operating band of accumulator

The accumulator should work well at frequencies between 30-60 Hz [Kajaste -01]. It was desired to ensure this by testing the operating band of accumulators. Figure 4-4 is a simplified diagram of a test stand. Length x1 was 200 mm, length x2 was 300 mm and length y1 was 50 mm. The test sequence is explained in Chapter 3.2.3.

Figure 4-4. Schematic hydraulic diagram of the test installation.

Figure 4-5 shows the situation where pre-charge pressure has been raised from 8.0 MPa to 20.0 MPa. The nominal size of the accumulator was $0.075 \times 10^{-3}$ m$^3$. The calculated nominal frequency of the tested accumulator was about 170 Hz at pre-charge pressure 9.7 MPa and 150 Hz at pre-charge pressure 13.8 MPa. The length L$_{12}$ was 0.05 m and the length L$_{34}$ was 0.01 m. The area A$_{12}$ was $0.3 \times 10^{-3}$ m$^2$ (D=16 mm) and the area A$_{34}$ was $7.85 \times 10^{-5}$ m$^2$ (D=10 mm).
Figure 4-5. Pressure ripple (peak-to-peak value) as a function of frequency and pre-charge pressure of a 0.75 dl accumulator.

Obviously the nominal size of the accumulator is too small for frequencies under 100 Hz. When the pre-charge pressure is higher than the working pressure of the accumulator, the accumulator is in principle out of operation. The 3-D surface shows well good the acceptable operating band of the accumulator. If the surface is even around the working point, the working point can be varied a little without any noticeable change in damping capacity.

Figure 4-6 shows the same type of test with a 2*0.6*10^{-3} m³ accumulator. The connection of the accumulator was the same as before. The calculated nominal frequency of the accumulator was 60 Hz at pre-charge pressure 10 MPa and 53 Hz at pre-charge pressure 13.8 MPa.
38

Figure 4-6. Pressure ripple (peak-to-peak value) as a function of frequency and pre-charge pressure of a 1.2 l accumulator.

When the accumulator works as a part of a wider system, no clear operating point can be observed, but there is a wide operating area. This means that the accumulator damps well even the working pressure (nominal frequency of the accumulator), or the operating frequency changes. It seems that the accumulator works well at such frequencies, i.e. between 30-60 Hz.

4.1.4 Dimensioning by basic accumulator formula [Ijas -00a]

Usually when the accumulator works as a pressure damper, the dimensioning of the accumulator and the connection pipe is done by calculating the nominal frequency. When the accumulator works as a power reserve, Equation 8 is used. Also in the reference [Hydac] the basis of calculation of the pressure damper is

\[ p_0 \cdot V_0^\kappa = p_1 \cdot V_1^\kappa = p_2 \cdot V_2^\kappa \]  \hspace{1cm} (8)

In this case the accumulator must be installed near the mainline and the connection pipe must be large. The use of this equation requires that the volume causing the oscillations be known. If \( \Delta V = V_1 - V_2 \), the following equation is obtained [Hunt -96 ]:
\[ V_0 = \frac{\Delta V}{\left( \frac{p_0}{p_1} \right)^{\frac{1}{\kappa}} - \left( \frac{p_0}{p_2} \right)^{\frac{1}{\kappa}}} \quad (9) \]

If the absorbed oil volume is not known, it may be possible to integrate from the flow rate diagram. The hatched area in Figure 4-7 shows the oil volume which has to be absorbed. The oil flow equation can be written

\[ Q = Q_{st} + u \cdot \sin(2 \pi f t) \quad (10) \]

\[ Q_{st} = \text{steady-state oil flow} \quad \left[ \frac{m^3}{s} \right] \quad (\text{now } 0.0016 m^3/s) \]

\[ u = \text{amplitude of flow ripple} \quad \left[ \frac{m^3}{s} \right] \quad (\text{now } 0.000117 m^3/s) \]

Figure 4-7. An example of the flow ripple.
If it is desired to calculate the hatched area, the lower limit of the integral \((t_{\text{lower\ limit}})\) is 0 and the upper limit of the integral is
\[
t_{\text{upper\ limit}} = \frac{1}{f \times 2}
\]

The following equation is obtained:

\[
\Delta V = \int_{0}^{1/f \times 2} u \sin(f \times 2\pi t) \, dt = \frac{u}{f \times \pi}
\]  \hspace{1cm} (11)

Equation 9 can be written as

\[
V_0 = \frac{u}{f \times \pi} \left( \frac{p_0}{p_1} \right)^{1/x} - \left( \frac{p_0}{p_2} \right)^{1/x}
\]  \hspace{1cm} (12)

The validity of this equation was tested experimentally. The pressure ripple at the actuator was measured using different nominal sizes of accumulator. The test sequence and the test installation were as in Chapter 3.3. The amplitude of the flow ripple was 0.117*10^{-3} \text{ m}^3/\text{s} (7 l/min), the steady-state flow rate was 1.67 *10^{-3} \text{ m}^3/\text{s} (100 l/min), the working pressure was 16 MPa and the pre-charge pressure was 12 MPa.
The contour line of the pressure ripple at the actuator is presented in Figure 4-8. The accumulator was damped weakly (much contour lines) when the frequency was low or the nominal size was small. The nominal sizes of the tested accumulators were $0.075 \times 10^{-3} \text{ m}^3$, $0.15 \times 10^{-3} \text{ m}^3$, $0.32 \times 10^{-3} \text{ m}^3$, $0.6 \times 10^{-3} \text{ m}^3$, $1.2 \times 10^{-3} \text{ m}^3$ and $1.8 \times 10^{-3} \text{ m}^3$. For example, when the accumulator is $0.2 \times 10^{-3} \text{ m}^3$ it does not become quiet (pressure ripple < 3 bar) until the working frequency is higher than 40 Hz.

If there are no contour lines, it means that the pressure ripple is lower than 3.2 bar and it is more like white noise.

![Figure 4-8. Contour line of pressure ripples when the nominal size of the accumulator and the frequency are changed.](image)

Figure 4-9 shows the calculated minimum accumulator size according to Equation 12. The initial values were the same as before ($u=0.000117 \text{ m}^3/\text{s}$, $Q_{st}=0.0017 \text{ m}^3/\text{s}$, $p_0=12 \text{ MPa}$, $p_1=16 \text{ MPa}$). The aim was to quieten 99% of the pressure ripple so $p_2=1.01 \times 16 \text{ MPa} = 16.16 \text{ MPa}$ and $\kappa=1.4$. 
Equation 12 gives the lower limit of the accumulator size so the selected accumulator must be bigger. Comparison of Figure 4-8 and Figure 4-9 shows, that Equation 12 is quite usable at frequencies below 100 Hz if the flow ripple is measured.

4.1.5 Dimensioning by nominal frequency [Ijas -02a]

Simulations and tests show that the dimension method is most effective when the nominal frequency of the accumulator and the connection pipe is calculated [Edge -91, Kajaste -99a]. This is valid especially when the oscillation and the system are under control.

It was attempted to damp the rock drill pressure oscillation as depicted in Figure 3-5. Equations 3 and 6 define the connection dimensions and the nominal size of the accumulator.
The main oscillation to damp is at frequency 33Hz and the nominal size of the accumulator was 0.6*10^{-3} m^3. Equations 3 and 6 give the following parameters:

\[
\begin{align*}
L_{12} &= 280 \text{ mm} \\
D_{12} &= 16 \text{ mm} \\
L_{34} &= 10 \text{ mm} \\
D_{34} &= 10 \text{ mm}
\end{align*}
\]

When the pre-charge pressure is 8.6 MPa, the nominal frequency of the accumulator is 33 Hz. Length x1 was 200 mm, length x2 was 300 mm and length y1 was 290 mm (L_{12}+L_{34}).

Figure 4-10 is a simplified diagram of a test stand used. There was a steel pipe (length 12m, nominal size 2”) between the flow meter and the supply hoses.
Figure 4-11. Pressure oscillation of the rock drill without accumulator (above) and with the natural-frequency-dimensioned accumulator (down) [Ijas -02a].
Even though the control of the pump was the same as before (Figure 3-5), the pressure level was not exactly the same (Figure 4-11). The reason was probably the hysteresis of the pump control. In any case, the damping was not so good as expected. The main pressure amplitude (33Hz) is slightly lower than originally and the high frequency oscillation isn’t much lower than before. Expectation was that the accumulator should be worked better.

Next the accumulator was installed as near the main line as possible so that a comparison between those two methods could be made. The measurement installation and operating point were the same as before. Length x1 was 200 mm, length x2 was 300 mm and length y1 was 80 mm (L12=70 mm and L34=10 mm). The calculated nominal frequency was 61 Hz.
Figure 4-13. Pressure oscillation of the rock drill without accumulator (above) and when the accumulator is installed near the main line (down) [Ijas -02].
Figure 4-14: Power density of pressure oscillation of the rock drill with the near the main line installed accumulator

Figure 4-13 and 4-14 show that the main pressure amplitude is clearly lower than originally, but oscillation at high frequency is existence (1000Hz – 2000 Hz). It seems that the accumulator works better when it is very close to the hydraulic main line, at least in this case.

4.1.6 Discussion

There are studies which show that the accumulator damps best when it is dimensioned using the “natural frequency” method [Viersma -80, Kajaste -99a]. However, some manufacturers have used the basic accumulator equation, for which the absorbed oil volume is needed as an initial value [Hydac]. In this study, where the pressure oscillation was uneven, the damping was better when the accumulator was installed near the hydraulic main line.

The accumulator near the hydraulic main line is not necessarily the best for all applications, so it is better to use the natural frequency method first. The total damping should be better when the natural frequency is the basis of dimensioning. If damping is not desired, it would be useful to try the “short pipe” method.
When a connection pipe is used, one problem is the installation of the accumulator. When the accumulator is installed at the end of the pipe (length for example 280 mm), it vibrates a lot. The connection line can be realized also by hydraulic hose when there are no mechanical vibrations from the rock drill. In practice the accumulator should be supported well. On the whole, there were problems with the reliability of the accumulators during the tests. During the tests, three membranes of the accumulators broke even though the measurement times were not long. The temperatures had not risen much, so perhaps the high-frequency oscillation was the reason for the problems.

4.2 Helmholtz resonator

4.2.1 Theoretical background

The Helmholtz resonator is a damper which branches away from the main pipeline system. Characteristic is the narrow range of operating frequencies and the requirement for careful dimensioning [Viersma -80, Larsson -87a]. A typical application is damping of hydraulic pump ripple when the frequency is a few hundred hertz.

![Figure 4-15. Schematic picture of Helmholtz resonator.](image-url)
The resonance frequency of the Helmholtz resonator can be evaluated with the following equation [Viersma -80]:

$$f = \frac{a}{2 \pi} \sqrt{\frac{A_3}{L_3 V_h}}$$  \hspace{1cm} (13)

### 4.2.2 Simulation of Helmholtz resonator

The schematic picture of the simulated system is shown in Figure 4-2. Parameters and the simulation model were otherwise the same as in Chapter 4.1.2, except that the accumulator was replaced with a Helmholtz resonator. The damped frequency was 50 Hz. The dimensions of the simulated Helmholtz resonator were

- $A_3 = 7.85 \times 10^{-5} \text{ m}^2$
- $L_3 = 0.42 \text{ m}$
- $V_h = 2.73 \times 10^{-3} \text{ m}^3$

The ratio of pressure and flow rate of the Helmholtz resonator is the same as the accumulator (Viersma -80)

$$\frac{Q_B}{p_B} = \frac{C_a s}{s^2 \omega^2 + 1}$$

The hydraulic capacitance is

$$C_a = \frac{L_3 A_3 + V_H}{E}$$  \hspace{1cm} (14)
Figure 4-16. Simulated pressure oscillation without damper (above) and with the Helmholtz resonator (down).

The frequency response was changed when the Helmholtz resonator was added to the system. (Fig 4-16). The dimensioning is correct because the 50 Hz oscillation is damped.

4.2.3 Measurement of Helmholtz resonator

The aim of the measurements was to measure the operating band of the Helmholtz resonator and its operation as a function of operating pressure. The measurement installation was as in Figure 4.4, except that the accumulator was replaced with the Helmholtz resonator. The measurement sequence was the same as in Chapter 3.2.3. The Helmholtz resonator was tuned to damp 50 Hz oscillation and the dimensions were the same as before (Chapter 4.2.2). Figure 4-
17 shows that the best operating point is at the right place (50 Hz) and the Helmholtz resonator works as was expected.

Figure 4-17. The operation of the Helmholtz resonator as a function of frequency and pressure

4.2.4 Discussion

The Helmholtz resonator is an “old”, commonly used damper type. It is quite easy to tune and the operating band is wide enough for many applications. The best operation point is proportional to pressure, because the speed of sound is a function of bulk modulus, but the effect is minor at a pressure variation such as that used here.

Although the operating band and the damping capacity are sufficient for many applications, there is one problem. If it is desired to damp low frequencies, the size of the Helmholtz resonator becomes too big, especially in a mobile hydraulics application. In tests the length of the connection pipe was 0.42 m and the volume of the chamber was 2.73 litres. This kind of big damper can be difficult to install near the hydraulic actuator. However, the Helmholtz resonator operated well even at this low frequency.
4.3  T-pipe

4.3.1  Theoretical background
A branched pipe, like a T-pipe, is the simplest pressure oscillation damper. The main benefits are cheap price, working reliability and simple structure. If the speed of the sound is known, the T-pipe is simple to dimension [Larsson -87a]. The equation of definition is same as resonance frequency calculations earlier (Equation 1). In this study the number 1 is used as the odd number \( n \) (the length is shortest)

\[
f = \frac{n^*a}{4*L_T}
\]

\( n = 1,3,5... \)

4.3.2  Simulation of T-pipe
Just like before, the schematic picture of the simulated system is shown in Figure 4-2. Parameters and the simulation model were the same as in Chapter 4.1.2, except that the accumulator was replaced with a T-pipe. The T-pipe was modeled as a dead end hose. The damped frequency was 50 Hz and this way the length of the T-pipe becomes 6 m ( \( a=1200 \, \text{m/s} \)).
4.3.3 Measured damping of the T-pipe as a function of working pressure [Ijas -00b]

The aim of the measurements was mostly same as with the Helmholtz resonator, the operating band and the sensitivity of the pressure variation. The test sequence and the measurement stand were the same as before (Figure 4.4) except that accumulator was replaced with the T-pipe. The length of the T-pipe was 6 m ($a = 1200$ m/s, $f = 50$ Hz).
Figure 4-19. The operation of the T-pipe as a function of frequency and pressure.

The measurement results validate the statement found in the literature that the frequency band is narrower than with the Helmholtz resonator [Viersma] (refer to Fig 4-17).

4.3.4 Operating of T-pipe with rock drill [Ijas -02b]

Figure 3-5 shows the pressure oscillation of the supply line when the rock drill is running. In Chapters 4.1.3 and 4.2.3 the supply hose (15 m 1”+5 m 3/4”) was not directly next to the hydraulic pump because there was an extra pipe between the pump and the “real” rock drill supply hose. In this case the supply hose was installed with a short hose and a hydraulic block to the hydraulic pump. At first the pressure response of the supply hose was measured using the servo valve as a load (Figure 4-20). The excitation oscillation was raised by 1 Hz steps.

The pressure response of the supply line arrangement is a little different compared to that in Figure 3-8. The lowest resonance frequency is at different place. The boundary condition of the input end of the hose is changed. Also the speed of the sound could be different than before.
Next the speed of the sound has been wanted to know and a 5-metre T-hose was installed in the system. The T-hose was near the servo valve and it branched away from the hydraulic main line. The measurement routine was otherwise the same as before (oil flow, control of servo valve, etc). Figure 4-21 shows that the T-hose has a significant effect on the frequency response. The best damping frequency is at 55 Hz. From this it is possible to calculate the speed of the sound.

\[
a = 4 \times L_p \times f = 4 \times 5 \times 55 \frac{m}{s} = 1100 \frac{m}{s}
\]
When the speed of the sound is known, the T-hose is easy to dimension for the frequency 33 Hz. Figure 4-22 shows the pressure response when the length of the T-hose was 8.3 m. The best damping frequency is, as was expected, at slightly over 30 Hz.
When the T-pipe had been got to the right length to damp 33 Hz oscillation, the servo valve was replaced with the rock drill. When Figures 4-23 and 4-24 are inspected a clear effect is seen. The main pressure oscillation has been decreased and also the high-frequency oscillation is much lower. The use of the long T-hose requires that free air is exhausted from the T-hose. If there is free air in the hose, this causes errors in the calculation.

Figure 4-23. Measured pressure oscillation of the rock drill without damper (above) and with an 8.3 m T-pipe (down).
4.3.5 Damping step change with T-pipe [Ijas -02b]

Few main frequency oscillations can be seen (Figures 3-5) when the rock drill is idling. The first (33 Hz) is at the frequency of the operating frequency (66 Hz is its double frequency) and the second (1200 Hz) is the nominal hydraulic frequency of the inlet chamber of the rock drill. The frequency 1200 Hz pressure oscillation occurs when the control valve causes a step. The use of the T-pipe to dampen this kind of step oscillation is not the traditional use of the T-pipe. The T-pipe changes the acoustic properties of the rock drill.

The length of the T-pipe becomes

\[ L_T = \frac{a}{4 \cdot f} = \frac{1100}{4 \cdot 1200} m = 0.23 \, m \]

The measurement was repeated with a 0.23 m T-pipe and the result can be seen in Figures 4-25 and 4-26. The T-pipe was installed as near the rock drill as possible. The difference is clear; high-frequency oscillation (1200 Hz) is lower with the 0.23 m T-pipe compared to the case with
the original installation. However, the 0.23 m T-pipe damps only 1200 Hz pressure oscillation. The main pressure oscillation (33 Hz) is about the same as originally.

Figure 4-25. Pressure oscillation of the rock drill without damper (above) and with 0.23 m T-pipe (down).
4.3.6 Discussion of T-pipe

Although the damping capacity is not the best and the length of the T-pipe is long when low frequencies are damped, there are a few good features. The damping capacity is good enough and the operating band is wide enough for many applications. The best feature is the easiness of installation of the T-pipe; it is an extra hose in the hydraulic machine. The T-pipe works quite well, although the frequency is low and the pressure amplitude is uneven. The T-pipe slightly damps the vibrations due to step excitation. In future studies it might be interesting to install two T-pipes to the hydraulic system, the first for the 33 Hz oscillation and the second for the 1200 Hz oscillation.
5 DAMPING OF PRESSURE OSCILLATION USING SERIAL DAMPER

5.1 Inline suppressor

5.1.1 Structure of inline suppressor
The inline suppressor is an accumulator with a tubular membrane and the oil flows through the suppressor [Wilkes -98]. It consists of the following parts: gas valve (1), gas volume (2), elastic membrane (3), support pipe with small holes (4) and flow channel (5). The inline suppressor is designed to quieten high frequencies, and hence the name suppressor, which refers to the effect on hydraulic noise.

Figure 5.1 An inline suppressor [Kajaste -01].

5.1.2 Operating band of inline suppressor [Ijas 00a]
The operating band of the inline suppressor was measured using the servo valve as an actuator (Figure 5-2). The test sequence and the installation were the same as Figure 4-4, except that the accumulator was replaced with the suppressor.
The nominal size of the suppressor was 1 1/4”. The size of the inline suppressor is defined according to instruction of manufacturer (hose diameter). The conclusion is that the suppressor works just like a small standard accumulator for low frequencies (see Figure 4-5). It seems that the suppressor works best at a frequency higher than 100 Hz.

5.1.3 Operating of inline suppressor with the rock drill
The inline suppressor was installed in the hydraulic supply line near the rock drill. Installation was as shown in Figure 4-10, except that the accumulator was replaced with the 1 ¼ “ inline suppressor. Figures 5-3 and 5-4 show the influence of the inline suppressor. It damps high frequency oscillation quite well.
Figure 5-3. Pressure oscillation without damper (above) and with inline suppressor (down).
5.1.4 Discussion of inline suppressor

The inline suppressor is mainly a low pass filter. The suppressor damps well high frequencies but passes low frequency oscillation through, which can be seen in the Figure 5-3. Probably the theoretical nominal size of the suppressor, like the nominal size of the accumulator, is too small, because it did not damp low frequency oscillation well. In practice this kind of suppressor is easy in install to the system. As the name indicates, it is installed inline and this means that the suppressor is easy to support. The cooling of the inline suppressor is better than that in an accumulator because of the oil flow through the suppressor.
6 DISCUSSION

The aim of this thesis was to study properties of pressure dampers and this way improves the reliability of the supply line of a pulsating actuator. For example, the reliability of the hydraulics of a rock drill is not good, but users have become accustomed to it. The reason behind the problems is the very demanding actuator. The rock drill causes high pressure oscillation in the system. The hydraulic power required is tens of kilowatts and the length of the hose between the pump and the rock drill is about 20 m.

Because the rock drill is installed at the tip of the boom, the demands placed on the dampers are high. The size and the weight of a damper are limited. Reliability of dampers is essential and a damper should not decrease the efficiency of the system.

Different kinds of dampers were tested with a servo valve and with a real rock drill. The T-pipe seems to be the most promising damper for low frequency. The accumulator damps well and it is worth considering if problems of reliability can be solved. The accumulator needs a very good supporting rack which supports the accumulator and conducts heat to the environment. The inline suppressor damps high frequency oscillation well.

The hydraulic motor, running without a load, damps pressure oscillation. The hydraulic motor has a certain inertia, which is increased by a flywheel. Then inertia of the hydraulic motor absorbs the pressure oscillation. This idea was tested by installing the hydraulic motor in the pipeline and by running the test sequence in the reference [Ijas -00b]. The good damping was achieved on a wide frequency band but the total efficiency and the price were insufficient. The motor was mainly an interface between an unstable and a stable line.
Table 1. Comparison of studied dampers.

<table>
<thead>
<tr>
<th>Damper</th>
<th>Maintenance</th>
<th>Damping low frequencies (30-60Hz)</th>
<th>Problems when damping low frequencies</th>
<th>Price</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accumulator</td>
<td>Needs</td>
<td>Excellent</td>
<td>Need maintenance</td>
<td>Medium</td>
</tr>
<tr>
<td>Inline suppressor</td>
<td>Needs</td>
<td>Weak</td>
<td>Price, weak damping</td>
<td>Expensive</td>
</tr>
<tr>
<td>T-pipe</td>
<td>No</td>
<td>Good</td>
<td>Long hose</td>
<td>Very cheap</td>
</tr>
<tr>
<td>Helmholtz resonator</td>
<td>No</td>
<td>Good</td>
<td>Big size</td>
<td>Cheap</td>
</tr>
</tbody>
</table>

It is possible to damp the supply line of the rock drill sufficiently if the price and the size of the damper are not considered. Figures 6-1 and 6-2 give the pressure oscillation when a 0.6 L accumulator and 1 ¼” inline suppressor are used at the same time. Damping is good, but the construction is complicated. The space required is large and there are two membranes, which decreases the reliability of the system.
Figure 6-1. Pressure oscillation of the rock drill without damper (above) and with the accumulator and the inline suppressor (down).
Figure 6-2. Power density of pressure oscillation of the rock drill with the accumulator and the inline suppressor.

The pressure oscillation shakes hoses. Obviously this decreases reliability, but also the possible outer abrasion of the hose raises the temperature. Abrasion can be more dangerous for the hose, depending on the supporting of the hose. The main reason for the warming is abrasion of the hose. If the pressure oscillation cannot be damped, the hose must be supported well. A special abrasion-resistant hose can be used or the hose can be installed inside a hose protector or a sleeve. Usually, when it is desired to increase the hydraulic power, pressure is a factor which is increased, because this permits smaller components to be used. This gives better power density. Unfortunately, the price of hose does not support this trend. Especially, a big (over 1”) high-pressure hose is very expensive. Another undesired feature is stiffness in high-pressure hoses. Also, the inner diameter of the fitting in high-pressure hose can be very tight.

Also, the hydraulic power can be increased by increasing the flow rate. The power density does not become better, but puts less load on hydraulic hoses. Especially if the hydraulic line is composed of several parallel hoses, the load for one hose is lower. The “parallel hose” concept was theoretically studied in the reference Ijas 2002b and it appears that parallel hoses become competitive at power levels over 100 kW.
7 CONCLUSIONS AND FURTHER WORK

Properties of pressure dampers were simulated and experimentally tested. The pulsating actuator was the servo valve or the rock drill.

The main results of the thesis can be summarised in the following statements:

- Dimension equations and simulation models of pressure dampers work quite well even for low frequencies.
- When the pressure oscillation is not of regular sine form the “natural frequency method” is not necessarily the best way to tune the accumulator.
- The T-pipe damps low frequencies well but the length of the T-pipe (or the T-hose) is long. The long hose is not necessary a problem.
- The T-pipe damps the vibrations slightly due to step excitation.
- The inline suppressor damps high frequencies well (1000-2000 Hz).
- The best solution for a rock drill machine is a well supported accumulator installed near the mainline or the T-pipe.

Parts of the tests were done with the real rock drill, but the operating point was not the usual one. In the future, the best of these hydraulic line concepts should be tested in real drilling conditions. The damper used should be tuned at a frequency of about 50 Hz.

Another point for study is the efficiency of the rock drill when the hydraulic line is damped. Does the damper disturb the operation of the rock drill? This can be determined best by measuring the total efficiency of the rock drill from the hydraulic pump to the button bit.

How the reliability changes when the pressure oscillation is damped and when a special abrasion-resistant hose is used can only be determined by long test runs in real conditions. All in all, long test runs are needed where the effect of changes on reliability and efficiency are studied.
References


Appendix A Transducers used in the test systems.

Appendix B Simulation models
## Appendix A

Transducers used in the test systems.

<table>
<thead>
<tr>
<th>Chapters Measured object</th>
<th>Amplifier+sensor</th>
<th>Range, Accuracy, Frequency</th>
<th>Output</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.2.3 Pressure</td>
<td>Druck PTX 1400</td>
<td>0-25 MPa, ±0.25% BSL</td>
<td>4-20 mA</td>
</tr>
<tr>
<td>4.1.3 Pressure</td>
<td>Druck PTX 1400</td>
<td>0-25 MPa, ±0.25% BSL</td>
<td>4-20 mA</td>
</tr>
<tr>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>3.2.3 Flow rate</td>
<td>Volutronic 5+</td>
<td>1-250 l/min, ±0.3 FS</td>
<td>±10 V</td>
</tr>
<tr>
<td></td>
<td>Kracht AS8</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.1.3</td>
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<td>4.1.5</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>5.2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.2.2 Pressure</td>
<td>Kistrel</td>
<td>0-50 MPa, ±0.5 %FS, &gt;50</td>
<td>0-10 V</td>
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<td>4.1.5</td>
<td>4065A500A2</td>
<td>kHz</td>
<td></td>
</tr>
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<td>7</td>
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</tbody>
</table>
Appendix B  Simulation models

Kajaste [-99b] stated that the pipe model of Mäkinen works excellent when the viscosity factor is low enough, for example 0.37 l/s. In this case the viscosity factor was

\[ \alpha = \frac{\nu}{r^2} \]

\[ \alpha = \text{viscosity factor} \left[ \frac{1}{s} \right] \]

\[ \nu = \text{kinematic viscosity} \left[ \frac{m^2}{s} \right] \]

\[ r = \text{radius of the hose} \left[ m \right] \]

\[ \Rightarrow \alpha (NS \ 1") = \frac{46 \times 10^{-6}}{0.0127^2} \frac{1}{s} = 0.29 \frac{1}{s} \]

\[ \alpha (NS \ 3") = 0.51 \frac{1}{s} \]

According to Kajaste [Kajaste -99b] the hose models of Mäkinen should operate well.

Flow rate through the hose was about 1.3\times10^{-3} m^3/s (80 l/min). In that case the Reynolds number is:

\[ \text{Re} = \frac{\nu \cdot d}{\nu} \]

\[ \nu = \text{velocity of flow} \left[ \frac{m}{s} \right] \]

\[ d = \text{diameter of the hose} \left[ m \right] \]

\[ \nu = \text{kinematic viscosity} \left[ \frac{m^2}{s} \right] \]

\[ \Rightarrow \text{Re (NS 1")} = \frac{2.57 \frac{m}{s} \cdot 0.0254 \ m}{46 \times 10^{-6} \frac{m^2}{s}} = 1419 \]

\[ \text{Re (NS 3")} = 1888 \]

Flow is probably laminar in this operating point.
“Valve” was modelled using a turbulent orifice equation. The discharge coefficient was 0.6 and the density of the oil 890 kg/m³. The steady-state opening of the valve was 4.5 mm (circle area) and the amplitude of the excitation was 0.2 mm. The frequency was increased from 0 to 100 Hz.
Figure 3. Parameters used of pipe models.
The accumulator was modelled using Equation 7. The damper was modelled using Equations 7, 13 and 14 in the Helmholtz resonator simulation. The PQ model [Mäkinen -00] with zero flow at the end of the hose operated as the T-pipe damper.
The steel pipe is modelled using the P-pipe model (P-C8) in the Figure 2. The P-pipe model means that the inward oil flow is known and the other end of the pipe is connected to the volume, which was now the pressure filter. The speed of the sound was 1100 m/s and there were eight modes in use in the model.

Q-models (Q-C16, Q-C17) are for situations where oil flow at the both side of the hose are knows. In this case there were 16 modes in use (Figure 6).
Figure 6. Q-C16 model [Mäkinen -00].