Direct measurement of blood pressure
by liquid-filled catheter manometer systems

by

J. L. C. Plasman and C. M. M. Timmers
DIRECT MEASUREMENT OF BLOOD PRESSURE BY LIQUID-FILLED CATHETER MANOMETER SYSTEMS

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EUT Report 81-E-121
ISBN 90-6144-121-8

Eindhoven
July 1981
Summary.

The use of liquid filled catheter-manometer systems is commonly accepted as a method to measure blood pressures. The reproduction of the blood pressure waveform is however strongly related to the quality of this measuring system.

This study will show that a hydraulic system should be used with short, stiff and wide bore (φ about 1.8 mm) lines which contain no or hardly any air bubbles and have a minimum of connections.

It is also shown that the amplitude versus frequency characteristic and the phase versus frequency characteristic of hydraulic systems with relatively low undamped resonance frequency $f_n$ (about 40 Hz) and low damping ratio $\rho$ (about 0.15) can be markedly improved by using electrical filtering techniques.

Plasman, J.L.C. and C.M.M. Timmers
DIRECT MEASUREMENT OF BLOOD PRESSURE BY LIQUID-FILLED CATHETER MANOMETER SYSTEMS.
Department of Electrical Engineering, Eindhoven University of Technology, 1981.
EUT Report 81-E-121

This work was done as part of the study of the authors in the Group Medical Electrical Engineering, Department of Electrical Engineering, Eindhoven University of Technology, under supervision of Prof.dr.ir. J.E.W. Beneken and ir. J.A. Blom.
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1. Introduction

The basis of this study is the need of the authors to get more detailed information about the pressure monitoring system present at the thorax O.R. in the Leiden University Hospital. For their thesis studies the authors use several clinically measured blood pressures, which preferably should be highly reliable. As clinical measurements tend to be less accurate than laboratory measurements, the questions to be answered are:

- How good or bad are these blood pressure signals?
- What are the influences of disturbances such as air bubbles?
- What can be done within the clinical environment to improve the quality of the blood pressure signals?

The reasonableness of these questions is not totally imaginary as may be illustrated by fig. 1.1., which is a computer simulation of the distortion of a blood pressure waveform by a specific second order measurement system.

This study can be divided into three parts. Chapters 2, 3 and 4 deal with the theoretical analysis of catheter manometer systems. Chapters 5, 6 and 7 treat the analysis of the measurements from the catheter manometer systems, whereas chapter 8 contains the conclusions and the recommendations for the user of catheter manometer systems.
fig. 11. Computer simulation of the distortion of a blood pressure waveform by a specific second order measurement system.

(Used by permission from Olson, lit. 8)
2. How should a pressure monitoring system perform?

The object of physiological pressure measurement is to record pressure variations faithfully, usually in analog form. This involves a transducer capable of changing pressure signals into, in general, electrical signals. The transducer will need an amplifier and some type of recording and/or displaying instrument. The total system should behave such that distortion of pressure information is within acceptable limits. The distortion may be divided into that related to:

- static or very slowly changing pressures.
- fast changing pressures (dynamic pressures).

2.1. Static pressure requirements.

In general the transducer system should fulfil the following requirements, when static pressures are applied:

- the amount of drift - mostly caused by temperature changes - should be small in terms of pressure and in relation to the final monitoring requirements.
- the system should be linear with a constant gain.
- the system must show minimal hysteresis.

2.2. Dynamic pressure requirements.

For the purpose of discussion it will be convenient to have in mind a particular pressure signal, e.g. the arterial pressure waveform. The general principles of monitoring as they will be formulated for the arterial pressure wave may be readily applied to other waveforms.

Suppose that we have an arterial pressure that repeats itself exactly with each heart beat at a regular heart rate. Fourier theory shows that the waveform can be considered as a mean level together with the sum of a series of sine waves of appropriate amplitude and phase. The lowest frequency sine wave will be that of the heart rate and the other frequencies will be integral multiples of this frequency.
In fig. 2.1. a harmonic analysis is made of a pressure curve in the carotid artery (curve a). Synthesis of the first six harmonics produces curve b. The table gives the relative values for the amplitudes.

![Fig. 2.1. Fourier analysis of the blood pressure curve in the carotid artery.](Used by permission from Peura, lit. 10)

The initial assumption, that the waveform is exactly periodic, is necessary to be able to interpret the results of the Fourier analysis precisely, the basic ideas however are not affected by lack of periodicity. The way in which the spectral content is derived though, may have to be modified.

For a perfect reproduction there should be no amplitude or phase distortion by the reproducing system, with one exception: if there is no disturbance of the relative amplitudes of the frequency components, but their phases are shifted in proportion to frequency, then synthesis from the components still produces the original waveform although it will be delayed in time, dependent on the phase shift.

In practice it is unnecessary to avoid amplitude and phase distortion at all frequencies. As the frequency rises the amplitude of the components will decrease in magnitude until it becomes indistinguishable from the effects of biological and instrumental noise.
There are two problems in this approach.

1. There is no definite boundary above which biological pressure variations may be ignored. In the literature the following recommendations are made.

<table>
<thead>
<tr>
<th>Author</th>
<th>Year</th>
<th>Recommendation</th>
</tr>
</thead>
<tbody>
<tr>
<td>A.T. Hansen</td>
<td>1949</td>
<td>The system should have a flat amplitude versus frequency characteristic (a.v.f.) till above the 10(^{th}) harmonic of the pulse rate.</td>
</tr>
<tr>
<td>D.L. Fry et al.</td>
<td>1957</td>
<td>Same as Hansen.</td>
</tr>
<tr>
<td>D.A. McDonald</td>
<td>1960</td>
<td>For very precise measurements a flat a.v.f. till the 15(^{th}) harmonic of the pulse rate is required.</td>
</tr>
<tr>
<td>R.R. Vierhout and A.J.H. Vendrik</td>
<td>1961</td>
<td>A flat a.v.f. with an error of 10% at the upper freq. which is generally five times the pulse rate is required.</td>
</tr>
<tr>
<td>L.A. Geddes</td>
<td>1970</td>
<td>Same as Hansen. But for highly reliable response a flat a.v.f. till beyond the 15(^{th}) harmonic of the pulse rate is required.</td>
</tr>
<tr>
<td>I.T Gabe</td>
<td>1972</td>
<td>Same as Hansen.</td>
</tr>
</tbody>
</table>

Conclusion: For a good response a monitoring system is required with a flat a.v.f. till above the 10\(^{th}\) harmonic of the pulse rate and for precise measurements the a.v.f. should be flat till beyond the 15\(^{th}\) harmonic.

2. The monitoring system in our case consists out of an electrical and a hydraulic system. The electrical system can easily be developed with a flat a.v.f. till 100 or 200 Hz. The hydraulic system however is restrained to a far lower frequency. Bruner (lit. 1, pg. 129) for example states that a hydraulic pressure line is
restrained to 10 to 20 Hz.

For clinicians phase shift in the frequency components might be of little concern, especially when they consider measurements in the periphery. Phase shift however can disturb the waveform in such a way that derived values such as dP/dt, systolic and diastolic pressures become unreliable. So phase distortion should preferably be avoided (with the exception as mentioned on page 7).

Measurement of the derivative of the pressure signal increases the bandwidth requirements, because differentiation of a sinusoidal harmonic increases the amplitude of that component by a factor proportional to its frequency. Noise at higher frequencies can become very large on differentiation. The original signal should be as free of noise as possible, e.g. by the use of a low-pass filter with a cut-off frequency such that the derivative of the signal is not significantly affected. The bandwidth required may be estimated by Fourier analysis of the time derivative signal. As Gersh et al. (lit. 5. pg. 38) have shown the a.v.f. characteristics of the monitoring system used must remain flat to within 5% up to the 20th harmonic.
3. The hydraulic catheter manometer system.

The wave velocity of a pressure wave in a hydraulic catheter manometer system as in fig. 3.1. is about 500m/sec (lit. 3. pg. 23 and 29). The highest frequency component to be transmitted by the line will normally not be higher then 50 Hz. Hansen (lit. 6. pg. 27) states that he never has measured frequencies higher than 50 Hz in the central part of the aorta in a human being. So the wavelength of the pressure wave will be at least 10m. A catheter manometer system does not normally exceed a length of 2m and therefore this system can not be dealt with in terms of transmission line theory.

fig. 3.1. Blockdiagram of a catheter manometer system.

Vierhout (lit. 11. pg. 73) has shown that a π - filter is an acceptable comparison of the catheter. The analogous electrical circuit of a catheter manometer system then looks like fig. 3.2.

The liquid resistance $R_c$ is due to friction between molecules flowing through the catheter:

$$ R_c = \frac{P}{F} \quad 3.1. $$

where : $P =$ pressure difference across the segment in Pa.
$F =$ flow rate through the segment in m$^3$/sec

Using Poiseuille's equation for laminar flow 3.1. becomes:

$$ R_c = 8\eta l/r^4 \quad 3.2.$$
fig. 3.2. Electrical analogon of a catheter manometer system.

where:

\[ \eta = \text{liquid viscosity} \quad \text{in Pa sec.} \]
\[ l = \text{length of the catheter} \quad \text{in m.} \]
\[ r = \text{radius of the catheter} \quad \text{in m.} \]

The liquid inertance \( L_c \) is due primarily to the mass of the liquid:

\[ L_c = \rho (\frac{d^2}{dt^2})^{-1} = \frac{\rho l}{\pi r^2} \tag{3.3} \]

where:

\[ \rho = \text{density of the fluid} \quad \text{in kg/m}^3. \]

The compliance \( C_c \) is due to the elasticity of the catheter wall and the compressibility of the fluid:

\[ C_c = \frac{\partial V}{\partial P} \text{cath} + \frac{\partial V}{\partial P} \text{liquid} \tag{3.4} \]

where:

\[ V = \text{volume of the catheter} \quad \text{in m}^3. \]

As \( \frac{\partial V}{\partial P} \text{liquid} \ll \frac{\partial V}{\partial P} \text{cath} \) equation 3.4 simplifies to:

\[ C_c = \frac{\partial V}{\partial P} \text{cath} \tag{3.5} \]

The compliance \( C_c \) is equally divided over the parallel connections of the segment. The inertance and resistance of the transducer are assumed to be negligible with respect to \( R_c \) and \( L_c \).
This analogon represents a single degree of freedom system. The transfer function of such a system is completely determined by two phenomena: the resonance frequency of the undamped system:

\[ f_n = \frac{1}{2\pi \sqrt{L(C_m + jC_c)}} \quad \text{(3.6)} \]

or using 3.2. and 3.3.

\[ f_n = \frac{r}{\pi \sqrt{\frac{1(C_m + jC_c)}}} \quad \text{(3.7)} \]

and the damping factor \( \beta \) defined as the ratio of the system damping and the critical damping:

\[ \beta = \frac{R}{2} \sqrt{\frac{C_m + jC_c}{L}} \quad \text{(3.8)} \]

or using 3.2. and 3.3.

\[ \beta = \frac{4\pi}{r} \sqrt{\frac{1(C_m + jC_c)}{\pi \rho}} \quad \text{(3.9)} \]

The derivation of 3.6. and 3.8. is shown in Appendix A. When these two phenomena are determined the a.v.f. and the phase versus frequency (p.v.f.) characteristics of the catheter

(fig. 3.3. Amplitude response to a sinusoidal pressure of unit amplitude. Curves are shown for different values of \( \beta \).

(Used by permission from Gabe, lit. 3)
manometer system can be determined using fig. 3.3. and fig. 3.4. For many purposes a \( \phi \) of 0.64 is desirable. The error in the amplitude is then less than 2% up to 67 per cent of the natural frequency and the phase-lag is tolerably linear over this range.

![Graph](image)

**fig. 3.4.** Phase-lag behind a sinusoidal pressure for different values of \( \phi \).
(Used by permission from Gabe, lit. 3)

The electrical analogon in fig. 3.2. changes significantly when air bubbles are introduced into the catheter manometer system. Air bubbles are highly compliant and they decrease the performance of the monitoring system dramatically. Fig.

![Diagram](image)

**fig. 3.5.** Electrical analogon of a catheter manometer system with an air bubble somewhere in the catheter.
3.5. shows the electrical analogon of the catheter with a bubble somewhere in the catheter. When the bubble is situated at the transducer \((L_c = 0 \text{ and } R_c = 0)\) the resonance frequency of the undamped system \(f_n\) and the damping ratio \(\beta\) of the system are:

\[
f_n = \frac{r}{2\pi} \sqrt{\frac{\pi}{\rho\left(1 + \frac{1}{2}C_c + C_b\right)}} \tag{3.10}
\]

and

\[
\beta = \frac{4\rho}{r^3} \sqrt{\frac{1}{\left(1 + \frac{1}{2}C_c + C_b\right)}} \tag{3.11}
\]

Comparison of eq. 3.7. and 3.9. with 3.10. and 3.11. learns that \(f_n\) decreases and \(\beta\) increases when an air bubble is introduced.

When the bubble is situated elsewhere in the system, the system no longer remains a single degree of freedom system and becomes more complex. Generally speaking the influence of the bubble on the performance of the catheter manometer system decreases as the bubble is located further from the manometer, and small bubbles introduce less distortion than large bubbles. Fig. 3.6. shows an example of the distortion.

![Comparison between an undistorted and distorted stepresponse (bubble at transducer).](image)
of the stepresponse measured with an Ailtech transducer connected to a 4 ft polyethylene pressure line and a 20g canula. The bubble is situated at the transducer.
4. Determination of $f_n$ and $\beta$ by transient stepresponse.

The response characteristics of a catheter manometer system can be determined in two ways. The more accurate method, but also more complicated, because it requires special equipment, is to measure the frequency response of the system. A simpler and for our purposes more practical technique involves measuring the transient stepresponse.

The basis of this method is to apply a sudden step input to the catheter and record the resultant damped oscillations of the system. Fig. 4.1. shows a transient response of a P_{50} strain-gauge transducer connected to 4 ft poly-ethylene tubing, i.d. 1.8mm, o.d. 3.1mm and a 19g canula.

![Fig. 4.1. Transient response of P_{50} transducer connected to 4 ft poly-ethylene tubing and a 19g canula. $f_n$ is the natural undamped frequency, $f_d$ the natural damped frequency and $\beta$ is the damping ratio.](image)

The step input is achieved by closing the catheter at the open end with a finger. The system is then pressurised using the intraflo and the finger is withdrawn with a snap. Two problems can occur using this method. Withdrawing the finger can introduce mechanical vibrations into the system. The effects of these vibrations are similar to those due to cathe-
The recording of a transient step with such an artifact will show deformations as in fig. 4.2. Fixation of the open end of the catheter can easily prevent the introduction of such an artifact.

fig. 4.2. Transient step response with artefact due to mechanical vibrations.

The second problem is that the swiftness of the snap influences the quality of the step input. The step input directly measured at the transducer shows a \( \frac{dP}{dt} \) larger than \( 2 \times 10^4 \) mmHg/sec, which is sufficient to be considered as a step input.

The reason the "finger snap" method is used rather than e.g. the method suggested by Gabe (lit. 3. pg. 38) is due to the fact that a series of experiments should be performed in a short time. The outcome of the experiments had to be such that reasonable conclusions could be drawn about the quality of the monitoring system. The finger snap method proved to be a fast, simple, reproducible and reliable means to obtain this goal.

An example of a transient response is shown in fig. 4.3. The damped resonant frequency \( f_d \) can be calculated measuring the time between two consecutive peaks, \( f_d = \frac{1}{t_d} \).
fig. 4.3. Transient response of a second order underdamped system.

The damping factor $\beta$ can be calculated measuring $D_1, D_2, D_3, \ldots$ from successive peaks. The logarithmic decrement $\Delta$ is then calculated

$$\Delta = \ln\left(\frac{D_n}{D_{n+1}}\right)$$ \hspace{1cm} 4.1.

The damping factor is given by

$$\beta = \frac{\Delta}{\sqrt{n^2 + \Delta^2}}$$ \hspace{1cm} 4.2.

The derivation of this method is treated in Appendix B. If the ratio $D_n/D_{n+1}$ is known the damping factor $\beta$ and the ratio of the undamped resonant frequency and the damped resonant frequency $f_n/f_d$ may be derived directly from table I and table II respectively.

When the system is overdamped or nearly overdamped no or hardly any oscillations will occur. The analysis of this system can be made using a method stated by Warburg (lit. 14). The method is based on the fact that the course of the movement to the equilibrium position after a step input, when recorded in relative measures, is dependent only on its degree of damping. The method is discussed in Appendix B.
Table 1

Determination of the damping factor, $\beta$, from the transient response of a single degree of freedom system

<table>
<thead>
<tr>
<th>$D_{11}/D_e$</th>
<th>$\beta$</th>
<th>$D_{11}/D_e$</th>
<th>$\beta$</th>
<th>$D_{11}/D_e$</th>
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<td>0.500</td>
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(Used by permission from Gabe, lit. 3)
Fig. 4.4. gives a response of such a system. The ratio of the times \( \tau_{0.6} \) and \( \tau_{0.1} \) on which the system reaches respectively 0.6P and 0.1P after the step input P is calculated. Using table III the damping ratio can be determined, as well as the natural undamped frequency \( f_n = \frac{\omega_n t_{0.6}}{2\pi t_{0.6}} = \frac{\omega_n t_{0.1}}{2\pi t_{0.1}} \)

\[
\tau_{0.6} = 0.56 \\
\tau_{0.1} = 0.35 \\
\rho = 0.35
\]

fig. 4.4. Response of a nearly overdamped system.
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Table III Determination of the damping factor $\beta$ from the relative time ratio $\tau_{0.6}/\tau_{0.1}$.

(Reproduced from Hansen, lit. 6)
5. **Experiments.**

The frequency response of a catheter manometer system was studied by generating a step, at the catheter-tip. The canula was closed with a finger, till the pressure in the system reached a certain level. Then the canula was opened by withdrawing the finger with a snap.

The catheter manometer system existed out of a canula, two or three pressure-lines connected to each other with a stopcock, an intraflow and a manometer (= transducer). The blockdiagram of this experimental composition can be set up as follows:

![Blockdiagram of a catheter manometer system.](image)

**Fig. 5.1.** Blockdiagram of a catheter manometer system.

Canulas of different internal diameter and length were tested (see table VI.), as well as pressure-lines of different stiffness and length (see table V.). During the experiments three different transducers have been used (see table IV.).

The responses of the catheter manometer systems were recorded with a Biograph 82, of Siemens-Clement. The paper-speed was variable from 50 mm/sec. up to 1000 mm/sec.
\[ C_m = (\text{volume displacement}) \]

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<th>[ f_n ] (kHz)</th>
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<tr>
<td>Statham P50</td>
<td>( 4 \times 10^{-4} )</td>
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<td>Hewlett Packard 1290A</td>
<td>( 2 \times 10^{-3} )</td>
<td>&lt;1</td>
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Table IV. Transducers.

Whenever a Statham- or Altech transducer was used, the transducer was connected to the pressure-amplifier 863 of the Mingo gref (the low-pass filter set at 1000 Hz). The HP-transducer was connected to a pressure amplifier of HP, 8805E. (the low-pass filter was set at 850 Hz).

Great care was taken to avoid airbubbles.

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<th>Line</th>
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<td>Cobe</td>
<td>6 ft</td>
<td>3.1 mm</td>
<td>1.8 mm</td>
<td>Polyethylene</td>
<td>L₇</td>
</tr>
</tbody>
</table>

Table V. Pressure-lines.
<table>
<thead>
<tr>
<th>Canula</th>
<th>Length</th>
<th>O.d.</th>
<th>I.d.</th>
<th>Material</th>
<th>Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viggo 14G</td>
<td>54 mm</td>
<td>2.00 mm</td>
<td>1.44 mm</td>
<td>Venflon</td>
<td>C1</td>
</tr>
<tr>
<td>Viggo 17G</td>
<td>54 mm</td>
<td>1.40 mm</td>
<td>0.99 mm</td>
<td>Venflon</td>
<td>C2</td>
</tr>
<tr>
<td>Viggo 18G</td>
<td>54 mm</td>
<td>1.20 mm</td>
<td>0.79 mm</td>
<td>Venflon</td>
<td>C3</td>
</tr>
<tr>
<td>Viggo 19G</td>
<td>40 mm</td>
<td>1.00 mm</td>
<td>0.59 mm</td>
<td>Venflon</td>
<td>C4</td>
</tr>
<tr>
<td>Viggo 21G</td>
<td>32 mm</td>
<td>0.80 mm</td>
<td>0.44 mm</td>
<td>Venflon</td>
<td>C5</td>
</tr>
<tr>
<td>Cathlon 16G</td>
<td>67 mm</td>
<td>1.63 mm</td>
<td>1.26 mm</td>
<td>Teflon</td>
<td>C6</td>
</tr>
<tr>
<td>Cathlon 20G</td>
<td>42 mm</td>
<td>0.99 mm</td>
<td>0.71 mm</td>
<td>Teflon</td>
<td>C7</td>
</tr>
<tr>
<td>Jelco 22G</td>
<td>33 mm</td>
<td>0.79 mm</td>
<td>0.56 mm</td>
<td>Teflon</td>
<td>C8</td>
</tr>
<tr>
<td>Abbocath 22G</td>
<td>35 mm</td>
<td>0.79 mm</td>
<td>0.51 mm</td>
<td>Teflon</td>
<td>C9</td>
</tr>
<tr>
<td>Boehringer 13G</td>
<td>57 mm</td>
<td>2.29 mm</td>
<td>1.62 mm</td>
<td>Teflon</td>
<td>C10</td>
</tr>
<tr>
<td>Boehringer 16G</td>
<td>57 mm</td>
<td>1.65 mm</td>
<td>1.00 mm</td>
<td>Teflon</td>
<td>C11</td>
</tr>
<tr>
<td>Boehringer 17.5G</td>
<td>57 mm</td>
<td>1.37 mm</td>
<td>0.81 mm</td>
<td>Teflon</td>
<td>C12</td>
</tr>
<tr>
<td>Boehringer 18.5G</td>
<td>32 mm</td>
<td>1.14 mm</td>
<td>0.64 mm</td>
<td>Teflon</td>
<td>C13</td>
</tr>
<tr>
<td>Boehringer 22G</td>
<td>32 mm</td>
<td>0.70 mm</td>
<td>0.50 mm</td>
<td>Teflon</td>
<td>C14</td>
</tr>
<tr>
<td>Boehringer 24G</td>
<td>25 mm</td>
<td>0.725 mm</td>
<td>0.45 mm</td>
<td>Teflon</td>
<td>C15</td>
</tr>
</tbody>
</table>

Tablo VI. Canulas.
6. Results.

6.1 Pressure-lines.

Various pressure lines were used in the experiments. The damped natural frequency \( f_d \) decreases with the length of the line. (Eq. 3.6, gives the relation between the natural frequency and length, but for damping coefficients \( c_{pd} \leq 0.2 \), this relation also stands for the damped natural frequency.) Stiff material increases \( f_d \), because the compliance of the line is reduced.

Damping coefficient \( \beta \) increases with line-length and stiffer material gives a lower damping coefficient. (Eq. 3.9)

See table VII.

<table>
<thead>
<tr>
<th>Line</th>
<th>( f_d )</th>
<th>( \beta )</th>
<th>Length</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L_2 )</td>
<td>82 Hz</td>
<td>0.110</td>
<td>10 cm</td>
</tr>
<tr>
<td>( L_3 )</td>
<td>52 Hz</td>
<td>0.180</td>
<td>30 cm</td>
</tr>
<tr>
<td>( L_4 )</td>
<td>10 Hz</td>
<td>0.410</td>
<td>180 cm</td>
</tr>
<tr>
<td>( L_6 )</td>
<td>75 Hz</td>
<td>0.080</td>
<td>3 ft</td>
</tr>
<tr>
<td>( L_7 )</td>
<td>37 Hz</td>
<td>0.175</td>
<td>6 ft</td>
</tr>
</tbody>
</table>

Table VII. Natural damped frequency and damping coefficient for several pressure lines. (1 ft = 30.48 cm).

![Diagram](image)

Fig 6.1 Configuration used for this experiment.

(Symbols: See fig. 5.1)

For this experiment a P50-transducer was used.
6.2 **Length.**

The natural damped frequency $f_d$ was inversely related to the square root of the line length for both p.v.c.- and polyethylene tubing. Figure 6.2 represents this relation for lengths from 30 cm up to 180 cm for both types of tubing. The polyethylene line (COBE) is built up by connecting 1 ft and 2 ft lines. The p.v.c.-line had a length of 180 cm, for each other experiment, it was each time shortened with 30 cm till a length of 30 cm remained.

![Graph showing frequency vs. length characteristic for both types of tubing.](image)

**Fig. 6.2.** Frequency vs. length characteristic for both types of tubing.

The damping coefficient is related to the square root of the pressure line length. See fig. 6.3.
Fig. 6.3. Damping coefficient vs. length characteristic for both types of tubing.

Used configuration:

Fig. 6.4. Configuration, with a HP 1290A. (Symbols: See Fig. 5.1)

The damping coefficient is related to the square root of the pressure line length and the damped natural frequency is related to the inverse root of the line length. These results are according to the theory, which suggests such relations.

6.3 Pressure line configuration.

A polyethylene configuration of 4 ft gives a high damped natural frequency $f_d$ and a low damping coefficient $\beta$.

Replacing the 1 ft Cobe-line by a soft Talas (p.v.c) line
of the same length gives an increase in $\beta$ and a decrease in $f_d$. Adding another Talas-line gives a further decrease of $f_d$ and an increase of $\beta$. Changing these two Talas-lines ($l=30$ cm, i.d. = 1.0 mm) by two Connecta-lines ($l=10$ cm, i.d. = 2.1 mm) gives almost the same damped frequency, between certain bounds, and a decrease in the damping coefficient $\beta$.

From the theory follows the relation $f_d \sim r$ and $f_d \sim \frac{1}{\sqrt{r}}$. With this knowledge, it is possible to explain the little change in $f_d$. (see table VII). The damping coefficient $\beta$ is related to the length $l$ as $\beta \sim \sqrt{l}$ and with the internal diameter $r$ as $\beta \sim 1/r^2$. (see eq. 3.9) Both relations give a decrease in the damping coefficient. (see table VIII.)

**configuration:**

![Configuration diagram](image)

**Fig. 6.5. Configuration with T = P50. (Symbols: See Fig. 5.1)**

<table>
<thead>
<tr>
<th>Configuration</th>
<th>$f_d$</th>
<th>$\beta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_3L_6$</td>
<td>40 Hz</td>
<td>0.210</td>
</tr>
<tr>
<td>$L_2L_6L_3$</td>
<td>24 Hz</td>
<td>0.255</td>
</tr>
<tr>
<td>$L_1L_6L_1$</td>
<td>25 Hz</td>
<td>0.145</td>
</tr>
<tr>
<td>$L_6L_5$</td>
<td>61 Hz</td>
<td>0.095</td>
</tr>
</tbody>
</table>

Table VII. Pressure line configuration.

### 6.4 Canulas.

The internal diameter $r$ of the canula is linearly related to the damped natural frequency $f_d$. This relation can only be held for the Boehringer canulas. (see fig. 6.7).

The Viggo canulas, especially the 19G-camula, easily contain airbubbles and have a built in deformation. Too few different Cathlon, Jelco and Cathlon canulas are available to draw any conclusions about these types.
Fig. 6.6. Configuration with T= HP 1290A.
(Symbols: See Fig. 5.1)

Fig. 6.7. Frequency vs. internal diameter for different types of canulas.

6.5 Pressure line configuration with a canula.

The canulas used for these experiments have a natural damped frequency between 60 Hz and 200 Hz (depending on internal diameter and material used.). The natural damped frequency $f_d$ of the whole system decreases with decrease in the internal diameter of the canula.
As long as $r_{\text{can}} \geq r_{\text{el}}$, the decrease in $f_d$ is minimal. For $r_{\text{can}} \leq 0.5$ mm, the decrease for the Boehringer canulas is relatively large. See fig. 6.8.

Fig. 6.8. Frequency vs. internal diameter of the canula for the configuration of fig. 6.10.

The damping coefficient $\beta$ increases for decreasing internal diameter of the canula. See fig. 6.9.

Fig. 6.9. Damping coefficient vs. internal diameter of the canula for the configuration of fig. 6.10.
Configuration:

Fig. 6.10. Configuration used, for the experiments with Boehringer and Viggo canulas. See also fig. 6.8. and 6.9. (Symbols: See Fig. 5.1.)

For the configuration of fig. 6.13., a 3 ft Cobe line and a 30 cm Talas line, the internal diameter $r_{can}$ is of less influence to the damped frequency $f_d$. See fig. 6.11.

![Graph showing frequency vs. internal diameter](image)

Fig. 6.11. Frequency vs. internal diameter of the canula, for several Viggo canulas.

In systems with a damped natural frequency of 30 Hz or less the internal diameter $r_{can}$ is of less influence.

For this configuration, the damping coefficient $\beta$ also decreases for decreasing internal diameter of the canula. (if $r_{can} < 1.0$ mm) See fig. 6.12.
Fig. 6.12. Damping coefficient vs. internal diameter of the canula.

![Graph showing damping coefficient vs. internal diameter of the canula]

Fig. 6.13. Configuration used, for the experiments with Viggo canulas. See also fig. 6.11. and 6.12. (Symbols: See Fig. 5.1.)

6.6 Remarks.

1. The Ailtech transducer gives the same result as the Statcom transducer. **Small air bubbles** are often introduced, while connecting transducer and intraflo, and are hardly visible in the Ailtech.

2. The Hewlett Packard transducer must be handled with great care. Air bubbles often occur in the dome or between dome and transducer and cannot be flushed out easily.

3. Air bubbles are easily introduced in the canulas made of venflon. These remain in the canula because of its internal construction. (See fig. 6.14.)
Fig. 6.14. Cross-section of a Viggo canula.

A smoother internal construction could avoid this problem. Another problem is the occurrence of burrs in stopcocks and connections. They easily attract air bubbles.
7. Compensating techniques.

A flat frequency range till at least 30 Hz is required in order to get an accurate reproduction of the arterial wave, assuming that the phase-lag is proportional to the frequency.

Catheter manometer systems can usually be described as an undamped second order dynamic system. For instance, one of the experiments done (HP 1290A, 4 ft Cobe and a 18.5G Roehringer) shows a natural frequency

\[
\omega_n = 40 \text{ Hz}
\]

and a damping coefficient of \( \beta = 0.14 \). This would result, according to figure 3.3, in a flat frequency range up to 10 Hz. An ideal system would have a damping coefficient \( \beta_i = 0.64 \), which means a flat frequency range up to \( 0.67 \cdot \omega_n \). In the earlier mentioned experiment, this would be 27 Hz.

Vierhout suggested to introduce damping in the hydraulic system. This could be done by two different methods (lit. 13):
1. Serial damping.
2. Parallel damping.
A third method would be to introduce a low-pass filter in the electrical part of the measuring system.

7.1 Serial damping.

In order to eliminate the "overshoot" in the frequency response, airbubbles could be injected to improve the frequency response. Unfortunately airbubbles not only increase the damping coefficient \( \beta \) but also decrease the natural damped frequency. For example:

**Configuration:**

![Configuration Diagram](image)

Fig. 7.1. Configuration with T= P50. (Symbols: See Fig. 5.1.)

1. Without airbubbles: \( \omega_d = 41.5 \text{ Hz} \)
   \[ \beta = 0.122 \]

2. With airbubbles: \( \omega_d = 28.3 \text{ Hz} \)
   \[ \beta = 0.195 \]
7.2 Parallel damping.

A device called the "acunamic", developed by Sorenson Res. Company, is an example of parallel damping. The device is an adjustable resistor in series with an airbubble and is used to match the impedance of the catheter manometer system.

In the electrical analogon parallel damping is introduced as follows:

![Diagram](image)

Fig. 7.2. The electrical analogon of a catheter manometer system with an acunamic.

In fig. 7.3. and 7.4. the results of 5 different acunamics in the position bypass, 1 till 9, are given.

![Graph](image)

Fig. 7.3. Frequency vs. acunamic position for different acunamics.
Fig. 7.4. Damping coefficient vs. accunamic position for different accunamics.

Configuration:

```
```

Fig. 7.5. Configuration used for experiments with the accunamic, T= HP 1290A. (Symbols: See Fig.5.1.)

Results:

The natural frequency remains the same, between certain bounds. Experiments 4 and 5 give a decrease in the natural frequency only when the accunamic is in use. During these two experiments, there could have been an airbubble somewhere in the accunamic where it does not belong. Damping coefficient changes from 0.15 to 0.35.
Remark: the damping coefficient of the complete system could not be calculated with the acoustic set at position C. (resistance increases with acoustic position number.) Two methods described in the literature (Gabe and Peura) are not suitable in this case. (Lit. 3 and 10).

7.3. Electrical filtering.

A method to improve the amplitude versus frequency characteristic (a.v.f.) of a hydraulic system is to compensate the transfer function with either a passive first order low-pass filter or an active two pole Butterworth low-pass filter, depending on the damping ratio $\beta$.

Calculations indicate (see appendix C.) that proper selection of the type of filter and optimum selection of its 3 dB frequency will flatten the a.v.f. of the total system (hydraulic system and electrical filter) up to at least 75% of the undamped resonance frequency. The calculations have been performed for damping coefficients from 0.05 until 0.50 and they show that the Butterworth filter is best used for $\beta$ between 0.05 and 0.23. For $\beta$ between 0.24 and 0.50 the low-pass filter can better be used.

It can be calculated that the use of an electrical filter markedly improves the bandwidth over which the a.v.f. will be tolerably (within 6%) flat and that the filter reduces the overshoot at the damped resonance frequency for systems with damping ratios higher than 0.10 to an acceptable limit of less than 50%. Overshoot will not occur for damping ratios higher than 0.26.

Curve a in fig. 7.6. shows the a.v.f. of a single degree of freedom system with a damping ratio of 0.25. Curves b and c show the a.v.f. when the characteristic is compensated with a first order low-pass filter or a two pole Butterworth filter respectively.

The use of an electrical filter increases the phase-lag of the total system. However, when this phase-lag is proportional to the frequency, then the waveform will be reproduced by the system with certain delay in time.

Calculations prove that phase-lag is proportional to frequency up to at least 0.75 $f_n$ for systems with damping ratios from 0.05 till 0.30.
Fig. 7.6. Amplitude vs. relative frequency for a catheter manometer system with and without compensation.

Fig 7.7. Phase-lag vs. relative frequency for a catheter manometer system with and without compensation.

(a) = no compensation
(b) = 1st order low-pass compensation
(c) = 2nd order Butterworth compensation
For higher values of $\phi$ the phase-lag is proportional to the frequency up to at least $0.5 f_n$. Fig. 7.7. shows the phase-lag of the system used in fig. 7.6. Curve a is the phase-lag of the hydraulic system. Curve b and c show the phase-lag of the total system after compensation by either a low-pass filter or a two pole Butterworth filter.

Distortion results, when the phase characteristic departs from a straight line. Filtering clearly reduces this type of (phase) distortion.
8. Conclusions and recommendations.

Conclusions.

1. The frequency response of a hydraulic pressure line system is better for shorter lines. It is further improved by using stiff material (polyethylene).

2. Short, stiff lines decrease the damping coefficient. Low damping coefficients can be increased by using a low-pass filter in the electrical part of the pressure monitoring system.

3. Canulas made of teflon are preferred above canulas made of venflon. Little difference between these two kinds of canulas exists for small bore canulas.

4. Air bubbles occur mostly in stopcocks and in connections between a stopcock and a pressure line, a pressure line and a canula. Leakages occur at connections of stopcocks and pressure lines, especially when the connectors are made of soft material.

8.1 Recommendations.

1. The hydraulic system should be filled at a low pressure. This must be done with great care, or else there will be air bubbles trapped in the system, especially near stopcocks. The system must be filled with cold injection fluid.

2. The hydraulic system must be as short as possible, using stiff pressure lines with as few connections as possible.

3. The efficiency of stopcocks should be tested by pressurizing the system and closing the stopcocks while output is recorded. If the pressure falls within a few seconds there is a significant leak. The stopcocks may need tapping.

4. If the pressure system is partly built up and filled with liquid, it should be closed by a stopcock. This prevents the introduction of air bubbles.

5. Coils do not influence the frequency response of the system. But mechanical vibrations do disturb the frequency response of the system.

6. A flush response gives accurate information about the condition of the pressure system.
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Appendix A.

Mathematical analysis of a single degree of freedom system.

The system function of the analog circuit in fig. 3.2 is:

\[ L \cdot (C_m + \frac{i}{C_c}) \cdot \frac{d^2 P}{dt^2} + R \cdot (C_m + \frac{i}{C_c}) \cdot \frac{dP}{dt} + P_o = P_i \]  

(A1)

or

\[ w_n^2 \cdot \frac{d^2 P}{dt^2} + 2\beta \cdot \frac{dP}{dt} + P_o = P_i \]  

(A2)

where

\[ w_n = 2\pi f_n = (L \cdot (C_m + \frac{i}{C_c}))^{-\frac{1}{2}} \]  

(A3)

(\( f_n \) is the resonant frequency of the undamped system.)

and

\[ \beta = \frac{R}{2} \cdot \sqrt{\frac{C_m + \frac{i}{C_c}}{L}} \]  

(A4)

(\( \beta \) is the damping factor.)

The general solution of the associated equation of A.2

\[ w_n^2 \cdot \frac{d^2 P}{dt^2} + 2\beta \cdot \frac{dP}{dt} + P_o = 0 \]  

(A5)

is the complementary function

\[ P_o(t) = A \cdot \exp \left[ \left(-\beta + \sqrt{\beta^2 - 1}\right) \cdot w_n \cdot t \right] + B \cdot \exp \left[ \left(-\beta - \sqrt{\beta^2 - 1}\right) \cdot w_n \cdot t \right] \]  

(A6)

The constants A and B can be calculated from the initial conditions if a particular solution of equation (A2) is known.

Suppose a stepinput \( P_i \):

\[ P_i = P \quad t < 0 \]

\[ P_i = 0 \quad t > 0 \]  

(A7)

is applied.
The initial conditions are:
\[ P_0 = 0 \quad t \leq 0 \]
\[ \frac{dP}{dt} = 0 \quad t \leq 0 \] (A8)

The solution of equation (A2) can have three forms, depending on the damping ratio. These forms are:

**Overdamped** $\beta > 1$:  
\[ P_0(t) = \frac{\beta + \sqrt{\beta^2 - 1}}{2} \cdot P \cdot \exp \left[ \left( -\beta + \sqrt{\beta^2 - 1} \right) \frac{w_n}{2} t \right] 
+ \frac{-\beta - \sqrt{\beta^2 - 1}}{2} \cdot P \cdot \exp \left[ \left( -\beta - \sqrt{\beta^2 - 1} \right) \frac{w_n}{2} t \right] \] (A9)

**Critically damped** $\beta = 1$:  
\[ P_0(t) = (1 + w_n t) \cdot P \cdot \exp(-w_n t) \] (A10)

**Underdamped** $\beta < 1$:  
\[ P_0(t) = \frac{\exp(-w_n t \sqrt{\beta})}{\sqrt{1-\beta^2}} \cdot P \cdot \cos \left( \sqrt{1-\beta^2} \cdot w_n t - \arcsin \beta \right) \] (A11)
Appendix B.

Determination of $\beta$ and $f_n$ from a step response.

Fig. B.1. shows a step response of a second order underdamped system. The frequency of the oscillations in the underdamped response is the damped resonant frequency $f_d$.

$$f_d = \sqrt{1-\beta^2} \cdot f_n$$  \hspace{1cm} (B1)

A maximum in the response curve is reached when the argument of the cosine in Appendix A. equation A.11 is $2n\pi$.

This happens:

$$t_n = \frac{2n\pi + \arcsin \beta}{\omega_n \sqrt{1-\beta^2}} \quad (n = 0, 1, 2, \ldots)$$  \hspace{1cm} (B2)

A minimum in the response curve is reached when the argument of the cosine is $\pi + 2m\pi$. This is the case when:

$$t_m = \frac{\pi + 2m\pi + \arcsin \beta}{\omega_m \sqrt{1-\beta^2}} \quad (m = 0, 1, 2, \ldots)$$  \hspace{1cm} (B3)

Substitution of B.2. and B.3. in A.11. permits the calculation of the ratio $D_n/D_{n+1}$ in fig. B.1.

$$\frac{D_n}{D_{n+1}} = \exp (-\beta \sqrt{1-\beta^2})$$  \hspace{1cm} (B4)

Recalculation of B.4. into an explicit expression for gives

$$\ln \left( \frac{D_n}{D_{n+1}} \right) = \frac{\ln(D_n/D_{n+1})}{\sqrt{\pi^2 + \left[ \ln(D_n/D_{n+1}) \right]^2}}$$  \hspace{1cm} (B5)
Substitution of $B.5.$ in $B.1.$ gives an expression for $\frac{f_n}{f_d}$.

$$\frac{f_n}{f_d} = \sqrt{1 + \left[ \frac{\ln(D_n/D_{n+1})}{\pi} \right]^2}$$  \hfill (36)

The relations $B.5.$ and $B.6.$ have been tabulated for different values of $D_n/D_{n+1}$ in table I (pg. 19) and II (pg. 20) respectively.

Another method to determine $\beta$ and $f_n$ is mostly used for overdamped or nearly overdamped systems. Suppose an underdamped system has a $\beta$ of almost 1, then oscillations will hardly occur. The times $t_{0.6}$ and $t_{0.1}$ at which the system reaches 0.6 P and 0.1 P after a step input $P$ at $t_0 = 0$ can be determined. Equation A.11. of Appendix A shows that:

\[
\text{at } t_{0.6}: \quad 0.6 P = \frac{e^{-\beta t_{0.6}}}{\sqrt{1-\beta^2}} \cos(\sqrt{1-\beta^2} \cdot z_{0.6} - \arcsin \beta) \tag{37} \\
\text{in which: } \quad z_{0.6} = w_n t_{0.6} \tag{38}
\]

\[\text{and at } t_{0.1}: \quad 0.1 P = \frac{e^{-\beta t_{0.1}}}{\sqrt{1-\beta^2}} \cos(\sqrt{1-\beta^2} \cdot z_{0.1} - \arcsin \beta) \tag{39} \\
\text{in which: } \quad z_{0.1} = w_n t_{0.1}
\]

For different values of $\beta$ both $z_{0.6}$ and $z_{0.1}$ can be calculated from eq. B.7. and B.9. as well as the ratio $\frac{z_{0.6}}{z_{0.1}} = \frac{t_{0.6}}{t_{0.1}}$. This ratio decreases monotonically as $\beta$ increases. So there is a specific relation between and $\frac{z_{0.6}}{z_{0.1}}$, which can be tabulated as has been done in table III. (pg. 22) From this table $\beta$ can be determined directly if the ratio $\frac{z_{0.6}}{z_{0.1}}$ is known. The undamped resonant frequency can be calculated from B.8. or B.10. if $t_{0.6}$ or $t_{0.1}$ is known.
Appendix C.

Transferfunctions of a catheter manometersystem with and without compensation.

In order to investigate whether an electrical filter, such as a first order low-pass filter or a two-pole Butterworth low-pass filter, will improve the a.v.f. and p.v.f, several simulations are performed on a PDP 11/60.

The transferfunctions of the hydraulic system, a first order low-pass filter and a two pole Butterworth low-pass filter can be stated as follows:

\[ H_n(j\omega) = \frac{1}{1 + j \cdot 2 \beta \left( \frac{\omega}{\omega_n} \right) + \left( \frac{\omega}{\omega_n} \right)^2} \]  
(C1)

\[ H_l(j\omega) = \frac{1}{1 + j \left( \frac{\omega}{\omega_0} \right)} \]  
(C2)

\[ H_b(j\omega) = \frac{1}{1 + j \cdot \sqrt{2} \cdot \left( \frac{\omega}{\omega_0} \right) + \left( \frac{\omega}{\omega_0} \right)^2} \]  
(C3)

The 3 dB frequency of the filter is \( f_0 = \frac{\omega_0}{2\pi} \).

For the calculation of the amplitude versus frequency and phase-lag versus frequency characteristic of the overall system, the hydraulic system followed by the electrical filter used, the transferfunction and phase-lag can be stated as follows:

**Low-pass filter:**

\[ |H_{syst.}| = \sqrt{\frac{1}{1 - \left( \frac{\omega}{\omega_n} \right)^2 + 4 \beta^2 \left( \frac{\omega}{\omega_n} \right)^2}} \]  
(C4)

\[ \arg[H_{syst.}] = -\arctan \left[ \frac{2 \beta \left( \frac{\omega}{\omega_n} \right)}{1 - \left( \frac{\omega}{\omega_n} \right)^2} \right] - \arctan \left[ \frac{\omega}{\omega_0} \right] \]  
(C5)
Butterworth filter:

\[
|H_{\text{syst}}| = \sqrt{\frac{1}{1 - \left(\frac{w}{w_n}\right)^2 + 4\beta^2 \left(\frac{w}{w_n}\right)^2}} \cdot \frac{1}{1 + \left(\frac{w}{w_o}\right)^2}
\]

\[
\arg[H_{\text{syst}}] = -\arctan \left( \frac{2\beta \left(\frac{w}{w_n}\right)}{1 - \left(\frac{w}{w_n}\right)^2} \right) - \arctan \left( \frac{\sqrt{2 - \left(\frac{w}{w_o}\right)^2}}{1 - \left(\frac{w}{w_o}\right)^2} \right)
\]

The relative frequency \( f/f_n = (w/w_n) \) is varied from 0.05 up to 1.20.
The damping coefficient \( \beta \) can be selected between 0.05 and 0.50. The quotient \( \alpha \) of the 3 dB. frequency of the filter and the undamped natural frequency \( f_n \) of the hydraulic system can be set to a value between 0.10 and 1.00.
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