BS ISO 10767-1:2015

BSI Standards Publication

Hydraulic fluid power — Determination of pressure ripple levels generated in systems and components

Part 1: Method for determining source flow ripple and source impedance of pumps

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National foreword

This British Standard is the UK implementation of ISO 10767-1:2015. It supersedes BS ISO 10767-1:1996 which is withdrawn.

Attention is drawn to the fact that, during the development of this International Standard, the UK committee voted against its approval as an International Standard.

The UK committee is of the opinion that the method is not proven and needs to be tested by industry. The committee would have preferred an intermediate stage where both ISO 10767-1:1996 and ISO 10767-1:2015 are available while new the version is tried and tested, as the committee views the 1996 edition to be more robust.

The UK participation in its preparation was entrusted by Technical Committee MCE/18, Fluid power systems and components, to Subcommittee MCE/18/-/8, Product testing.

A list of organizations represented on this subcommittee can be obtained on request to its secretary.

This publication does not purport to include all the necessary provisions of a contract. Users are responsible for its correct application.

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Compliance with a British Standard cannot confer immunity from legal obligations.

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Date	Text affected

INTERNATIONAL STANDARD

ISO 10767-1

Second edition
2015-10-01

Hydraulic fluid power — Determination of pressure ripple levels generated in systems and components —

Part 1: Method for determining source flow ripple and source impedance of pumps

Transmissions hydrauliques — Détermination des niveaux d'onde de pression engendrés dans les circuits et composants —

Partie 1: Méthode de détermination de l'onde de flux de la source et de l'impédance de la source des pompes

Reference number
ISO 10767-1:2015(E)

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

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For an explanation on the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the WTO principles in the Technical Barriers to Trade (TBT) see the following URL: Foreword - Supplementary information

The committee responsible for this document is ISO/TC 131, *Fluid power systems*, Subcommittee SC 8, *Product testing*.

This second edition cancels and replaces the first edition (ISO 10767-1:1996), which has been technically revised.

ISO 10767 consists of the following parts, under the general title *Hydraulic fluid power — Determination of pressure ripple levels generated in systems and components*:

— *Part 1: Precision method for pumps*

— *Part 2: Simplified method for pumps*

— *Part 3: Method for motors*

Introduction

The first edition of ISO 10767-1, published in 1996, was developed with a view to provide means for measurement (experimental determination) of the set of two characteristic values consisting of source flow ripple Q_S and source impedance Z_S of hydraulic pumps giving rise to pressure ripple (fluid born vibration) in the hydraulic power circuit., measurement of these two values for a given ripple source is extremely important for design and development of low noise pumps and hydraulic power systems, and for this reason, there is a valid need for such an international standard to experimental measurement of source flow ripple Qs and source impedance Z_S.

However, as discussed in the paragraph below, the so-called "secondary source method" presented in the first edition requires a very complex test system as well as signal processing technique, making its implementation highly difficult; because of this, no country except for the UK, the proposer, has yet adopted ISO 10767-1 as a national standard.

The difficulty can be explained as follows.

To determine the two characteristic values of the source flow ripple, Q_S, and source impedance, Z_S, a secondary ripple source is located in the test circuit to generate wide range ripples in the test system. Frequency characteristics of Z_S, arising from the secondary source, are first determined, followed by measurement of Q_S of the test pump on the basis of the test pump itself. This means that measurement of the harmonics of the pressure ripple is made with both the test pump and the secondary source in operation. As the result, the measurement accuracy of the harmonic component of the test pump deteriorates significantly as we come close to harmonic frequency level, where differences between the harmonic frequency of the test pump ripple and that of the secondary source become small. To deal with the problem, very complicated signal processing such as compensation is performed, but its practical effect is quite limited. In addition, the standard specifies use of a rotary valve for the secondary source of wide range (50 Hz ~ 4k Hz) ripples, but there is no provision as to the design and frequency characteristics.

These problems arise from the requirement for the secondary source, whereas the method proposed by Weddfelt[2] and Kojima[3] allows measurement of delivery ripple characteristics (Q_S) and the internal source (Z_S) on the sole basis of pressure ripple generated by the test pump. This makes the test system quite simple and allows superior accuracy to be achieved without complex processing of signals. The method according to the approaches of Weddfelt and Kojima, respectively, is the same in principle, the only difference between the two being the arrangement of the piping. The present proposal represents the method according to Kojima,[3] while annexing that of Weddfelt[2] for the purpose of reference.

Hydraulic fluid power — Determination of pressure ripple levels generated in systems and components —

Part 1:
Method for determining source flow ripple and source impedance of pumps

1 Scope

This part of ISO 10767 establishes a test procedure for measuring the source flow ripple and source impedance of positive-displacement hydraulic pumps. It is applicable to all types of positive-displacement pumps operating under steady-state conditions, irrespective of size, provided that the pumping frequency is in the range from 50 Hz to 400Hz.

Source flow ripple causes fluid borne vibration (pressure ripple) and then airborne noise from hydraulic systems. This procedure covers a frequency range and pressure range that have been found to cause many circuits to emit airborne noise which presents a major difficulty in design of hydraulic fluid power systems. Once the source flow ripple and source impedance of hydraulic fluid power pump are known, the pressure ripple generated by the pump in the fluid power system can be calculated by computer simulation using the known ripple propagation characteristics of the system components. As such, this part of ISO 10767 allows the design of low noise fluid power systems to be realized by establishing a uniform procedure for measuring and reporting the source flow ripple and the source impedance characteristics of hydraulic fluid power pumps.

In this part of ISO 10767, calculation is made for blocked acoustic pressure ripple as an example of the pressure ripple. An explanation of the methodology and theoretical basis for this test procedure is given in Annex B. The test procedure is referred to here as the *two pressures/two systems method*. Ratings are obtained as follows:

a) source flow ripple (in the standard "Norton" model) amplitude, in cubic meter per second[m^3/s], and phase, in degree, over 10 individual harmonics of pumping frequency;

b) source flow ripple (in the modified model) amplitude, in cubic meter per second [m^3/s], and phase, in degree, over 10 individual harmonics of pumping frequency; and its time history wave form,

c) source impedance amplitude, in Newton second per meter to the power of five [$(Ns)/m^5$]., and phase, in degree, over 10 individual harmonics of pumping frequency;

d) blocked acoustic pressure ripple, in MPa (1 MPa = 10^6 Pa) or in bar (1 bar = 10^5 Pa), over 10 individual harmonics of pumping frequency; and the RMS average of the pressure ripple harmonic f_1 to f_{10}.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 5598, *Fluid power systems and components — Vocabulary*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 5598 and the following apply.

3.1
source flow ripple
fluctuating component of flow-rate generated within the pump, which is independent of the characteristics of the connected circuit

Note 1 to entry: Since there exist the following two definitions of the *pump* source flow ripple, it shall be used with distinct discrimination:

— source flow ripple in the standard "Norton" model, Q_S, is the source flow ripple implicitly assumed to be generated at the pump outlet, as shown in Figure 1 a);

— source flow ripple in the "modified" model, Q_S*, is the source flow ripple assumed to be generated at the inner end of the discharge flow line, as shown in Figure 1 b).

Note 2 to entry: The theoretical pump source flow ripple which is calculated from computer simulation using the dimensions and configuration of the pump, physical properties of the fluid and operating conditions corresponds to the pump *flow ripple* (3.2) in the modified model, Q_S*.

3.2
flow ripple
fluctuating component of flow-rate of the hydraulic fluid, caused by interaction of *source flow ripple* (3.1) with the system

3.3
pressure ripple
fluctuating component of pressure in the hydraulic fluid, caused by interaction of the *source flow ripple* (3.1) with the system

3.4
blocked acoustic pressure ripple
pressure ripple (3.3) that would be generated at the pump discharge port when fluid is discharged into a circuit of infinite *impedance* (3.5)

3.5
impedance
complex ratio of the *pressure ripple* (3.3) to the *flow ripple* (3.2) occurring at a given point in a hydraulic system and at a given frequency

3.6
source impedance
impedance (3.5) of a pump at the discharge port in the standard "Norton" model

3.7
harmonic
sinusoidal component of the *pressure ripple* (3.3) or *flow ripple* (3.2) occurring at an integer multiple of the *pumping frequency* (3.8)

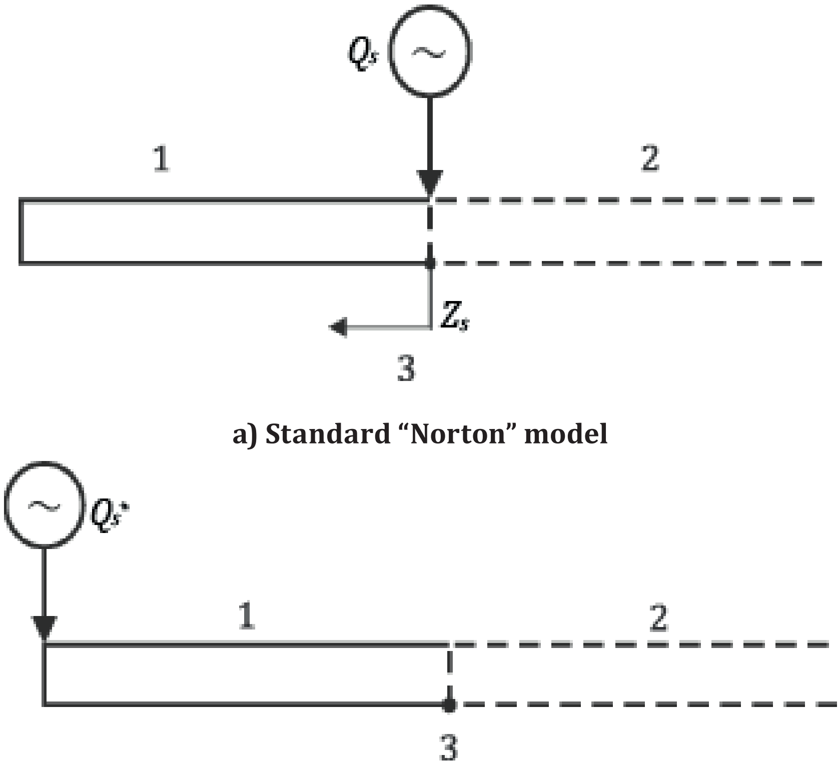
Note 1 to entry: A harmonic can be represented by its amplitude and phase, or, alternatively, by its real and imaginary components, provided that in this part of ISO 10767 the real and imaginary components are used in the arithmetic calculations.

3.8
pumping frequency
frequency given by the product of the *shaft rotational frequency* (3.9) and the number of pumping elements on that shaft

Note 1 to entry: It is expressed in hertz.

3.9
shaft rotational frequency
frequency (in hertz) given by the shaft rotational speed (in revolutions per minute) divided by 60

Note 1 to entry: Since the calculations in Clause 8 are all carried out using SI unit, all variables and constants shall be expressed in SI units, except for reporting of the end results.



b) Modified model

Key

1 discharge passageway
2 discharge line
3 pump exit

Figure 1 — Modelling of pump pulsation source

4 Instrumentation

4.1 Static measurements

The instruments used to measure

a) shaft rotational speed,

b) mean pressure,

c) mean discharge flow-rate, and

d) fluid temperature

shall have an accuracy throughout each test within the limits specified in Table 1.

NOTE The percentage limits are the of the value of the quantity being measured and not the maximum test values or the maximum reading of the instrument.

Table 1 — Permissible errors of static measurements

Shaft rotational frequency %	Mean flow %	Mean pressure %	Temperature °C
±0,5	±2,0	±2,0	±2,0

4.2 Dynamic measurements

The instruments used for measurement of pressure ripple shall have the following characteristics:

a) resonant frequency ≥ 30 kHz;

b) linearity ≤ ± 1 %.

The instruments need not respond to steady-state pressure. It can be advantageous to filter out any steady-state signal component by using a high-pass filter. This filter shall not introduce additional amplitude or phase error exceeding 1 % or 2°, respectively, at the pumping frequency.

4.3 Frequency analysis of pressure ripple

A suitable instrument shall be used to measure the harmonic amplitude and phase (or its real and imaginary components) of pressure ripple, for individual harmonics of the pumping frequency up to 3,5 kHz. The instrument shall be capable of measuring the pressure ripple from two pressure transducers simultaneously. The respective two pressure ripple signals of system 1 and system 2 shall be sampled in an instrument using external trigger signal obtained from a fixed reference on the pump shaft.

This instrument shall have the following accuracy and resolution for harmonic measurements over the frequency range from 50 Hz to 4 000 Hz:

a) amplitude within the range of ±1 %;

b) phase within the range of ±1°;

c) frequency within the range of ±0,5 %.

This can be achieved using a common type analysing recorder and then carrying out the spectral analyses by calculating discrete Fourier transforms (DFTs) of the time history data on a post processing digital computer. Annex B contains a tutorial explanation of this frequency analysis method.

NOTE In order to improve the accuracy of Fourier transformation, pump speed shall be adjusted minutely while observing the monitor of the analysing recorder so that the higher (e.g. 10th) harmonic amplitude peak appears nearly at the assigned higher (e.g. 10th) harmonic frequency (i.e. in case of f_1 being 225 Hz, f_{10} = 2,25 kHz) of the pumping frequency.

5 Pump installation

5.1 General

The pump shall be installed in the attitude recommended by the manufacture and mounted in such a manner that the response of the mounting-to-pump vibration is minimized.

5.2 Drive vibration

The electric motor and associated drive coupling shall not generate torsional vibration in the pump shaft. If necessary, the pump and the driving unit shall be isolated from each other to eliminate vibration generated by the electric motor.

5.3 Reference signal

A means of producing a reference signal relative to the pump shaft rotation shall be included, as one of essential elements in measurement according to this part of ISO 10767. The signal shall be an electrical pulse occurring once per revolution, with sharply defined rising and falling edges. This signal is used as an external trigger signal of analysing recorder, as well as for measurement of the shaft rotational speed. A magnetic gap detector (or a photo sensor) a satisfactory means of providing the required characteristics of reference signal mentioned above.

6 Test conditions and setting

6.1 General

Pump shaft speed, mean discharge pressure and fluid temperature are set to the values of required test conditions. These operating conditions shall be maintained throughout each test within the limits specified in Table 2.

Table 2 — Permissible variations in test conditions

Test parameter	Permissible variation
Mean flow	±2,0 %
Mean pressure	±2,0 %
Shaft rotational speed	±0,5 %
Temperature	±2,0 °C

6.2 Mean flow

Mean flow is measured by the positive-displacement type flow meter installed on the outlet line of loading valve 2.

6.3 Mean discharge pressure

Mean discharge pressure shall be measured electrically using a piezoresistance type transducer or a strain gauge type transducer mounted in the adapter before loading valve 1.

A bourdon type pressure gauge shall not be used for measurement of the mean discharge pressure.

6.4 Pump shaft speed

Pump shaft speed is measured by the magnetic gap detector (or photo sensor) installed on the pump shaft. Shaft rotational frequency (Hz) is given by the shaft rotational speed (rev/min) divided by 60.

6.5 Fluid temperature

Temperature of the fluid shall be that measured at the pump inlet.

6.6 Fluid property

Density, viscosity and bulk modulus of the test fluid shall be known to an accuracy within the limits specified in Table 3.

NOTE The percentage limits are of the error of the estimated quantity to the real value

Table 3 — Required accuracy of fluid property data

Property	Required accuracy
Density	±2,0 %
Viscosity	±5,0 %
Bulk modulus	±5,0 %

7 Test rig

7.1 General

The test rig shall be installed as shown in Figure 2. The test rig shall include all fluid filters, fluid coolers, reservoir, loading valves and any ancillary pumps required to meet operating conditions of the hydraulic pump. Specific features are described in 7.2 to 7.10.

7.2 Test pump

The test pump shall be installed in the "as-delivered" condition.

7.3 Test fluid

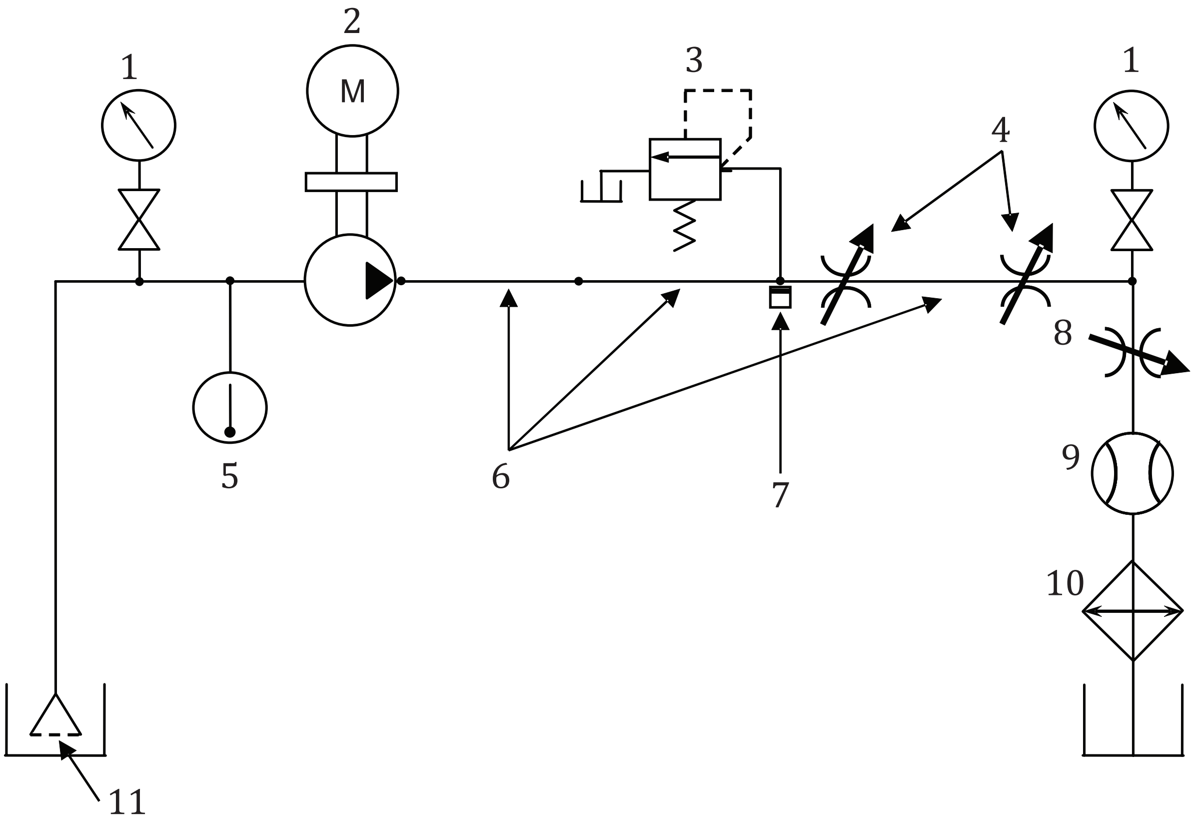
Type of the test hydraulic fluid and the quality of filtration shall be in accordance with the pump manufacturer's recommendations.

7.4 Inlet line

Internal diameter of the inlet line to the pump shall be in accordance with the pump manufacturer's recommendations. Care shall be exercised when assembling the inlet line to prevent air leakage into the circuit. The supply pressure shall be in accordance with the pump manufacturer's recommendations and, if necessary, a boost pump shall be used. If a boost pump is used, the pressure and flow ripple of the boost pump shall be taken into account, so that they do not affect the test results.

7.5 Inlet pressure gauge (for static pressure)

The inlet pressure gauge of Bourdon tube type shall be mounted at the same height as the inlet fitting. Otherwise, the gauge shall be calibrated for any height difference therefrom.



Key

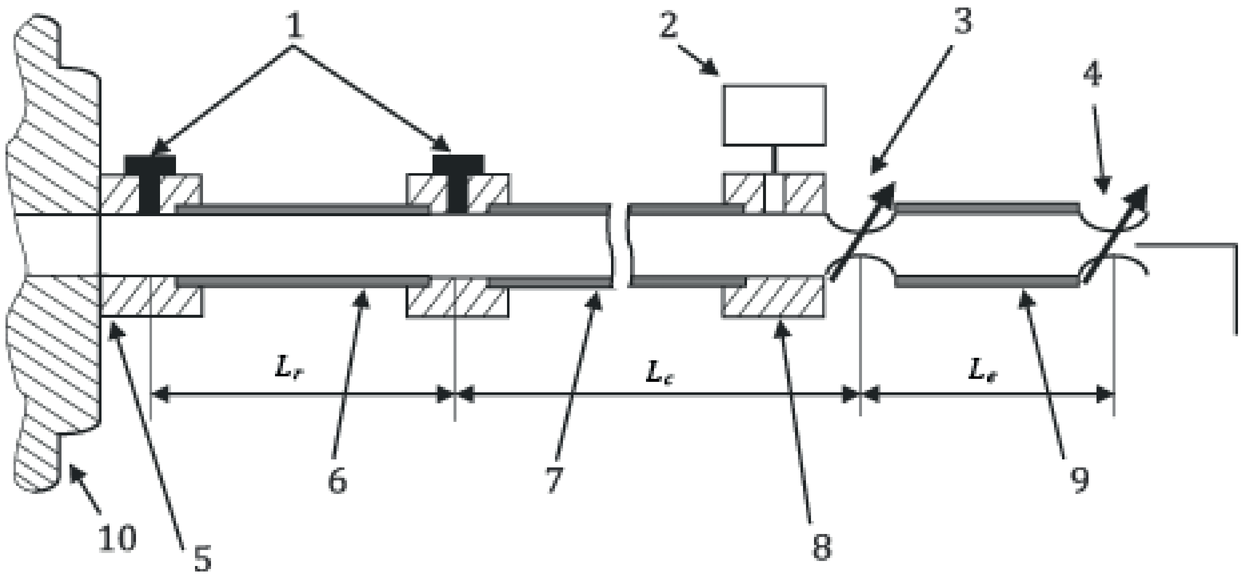
1 Bourdon tube pressure gauge
2 variable speed motor
3 direct operated relief valve
4 loading valves
5 temperature indicator
6 straight rigid pipe (see Figure 3)
7 pressure transducer (for static pressure)
8 back pressure valve
9 flow meter
10 cooler
11 strainer

Figure 2 — General circuit diagram for *two pressures/two systems* method test rig

7.6 Pump discharge line

7.6.1 General

The discharge line shall be as shown in Figure 3. Its functional test section consists of the reference pipe, connecting pipe, two loading valves, extension pipe, safety valve (direct operated relief valve) and the plural number of adaptors connecting these components.



Key

1	piezoelectric pressure transducers	6	reference pipe
2	direct operated relief value	7	connecting pipe
3	loading value 1	8	mounting block for loading value relief valve and pressure transducer
4	loading value 2	9	extension pipe
5	mounting block	10	pump

Figure 3 — Arrangement of discharge pipe

7.6.2 Pump discharge port connection

The adaptor connecting the pump discharge port to the reference pipe shall have an internal diameter identical to that of the reference pipe within the tolerance of not more than 10 % at any point. The adaptor shall be arranged in order to prevent the formation of air pockets in it. The reference pipe shall be mounted in line with the pump discharge port without any change in direction.

7.6.3 Reference pipe

The reference pipe shall be a uniform, rigid and straight metal pipe. Internal diameter of the reference pipe shall be not less than 80 % and not more than 120 % of the diameter of the pump discharge port. The reference pipe shall be supported in such a manner that pipe vibration is minimized. Length of the reference pipe (specifically, a distance between two pressure transducers), L_r , shall be set according to the pumping frequency. When the series of tests involves a given range of pumping frequencies, the length shall be selected in relation to the maximum pumping frequency in that series. For example, assuming the frequency range of interest to be from around 200 Hz to 3,5 kHz, the optimum length of reference pipe is around 150 mm. Details of selection of length of the reference pipe (and the extension pipe) are referred to Annex B.

7.6.4 Connecting pipe

Connecting pipe is the pipe installed subsequent to the reference pipe to eliminate the influence of flow turbulence generated in the vicinity of the loading valve orifice on the pressure ripple of P_1. The length of the connecting pipe shall be around 200 mm ~ 300 mm.

7.6.5 Extension pipe

Extension pipe is the pipe installed between the loading valves 1 and 2 to alter the shape of fluid vibration mode in the reference pipe in system 1 from that in system 2. When the series of tests involves a range of pumping frequencies, the pipe length shall be selected in relation to the maximum pumping frequency in that series. For example, assuming that the frequency range of interest to be from around 200 Hz to 3,5 kHz, optimum length of the extension pipe, L_e, is around 150 mm. Provided that the length of the extension pipe does not directly affect on the calculations of the source flow ripple Q_S and the source impedance Z_S in 8.3 and 8.4. Details in selection of the length of the extension pipe should be referred to Annex B.

For the purpose of changing the fluid vibration mode in the reference pipe in accordance with system 1 and system 2, a pressure vessel with a suitable volume capacity may be used instead of the extension pipe. In such a case, the test procedures given from Clause 8 onwards remain the same.

NOTE It is always correct practice to install the extension pipe behind loading valve 1, with loading valve 2 fully opened for the test of system 1, and with loading valve 1 fully open for the test of system 2.

7.7 Pressure transducer

7.7.1 Dynamic pressure transducer

For the purpose of pressure ripple measurement, a pair of pressure transducers are mounted on the adaptors connected at the both ends of the reference pipe. These transducers are to be fitted so that their diaphragms are flush with the inside wall of the bore in the adaptor. These pressure transducers shall be of piezoelectric type capable of accurate measurements of ripples for the pump drive shaft frequency up to 10 kHz minimum. The part of reference pipe between the two pressure transducers constitutes a measuring section where analysis of the standing wave is done. In this test method, the part from outlet end of pump casing to pressure transducer for P_0 (length 10 mm ~ 15 mm) is considered to be a part of the discharge passageway inside a pump casing.

7.7.2 Static pressure transducer

For the purpose of measurement of the mean discharge pressure, a piezoresistance pressure transducer or strain gauge type pressure transducer is mounted on the adaptor located immediately before loading valve 1.

7.8 Loading valve

Loading of the pump shall be accomplished using a needle valve or equivalent. A valve with free moving parts such as a pressure-relief valve shall not be used for the loading purpose.

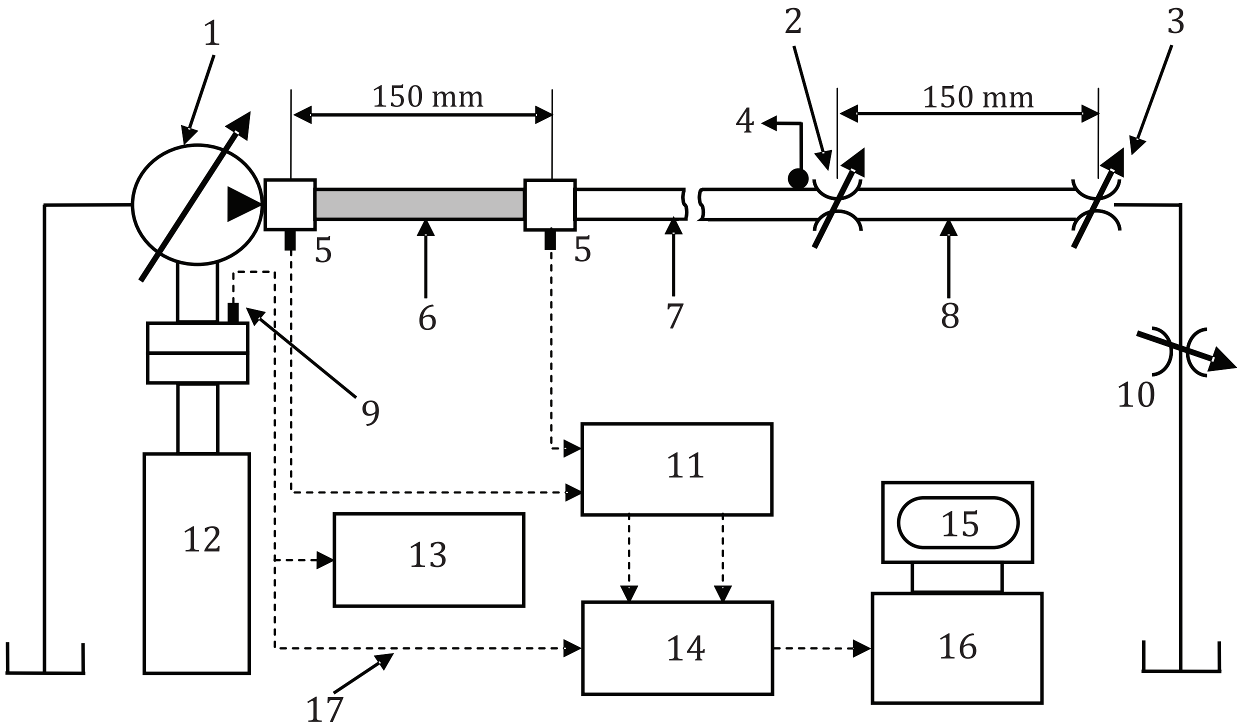
7.9 Back pressure valve

A back pressure valve of needle valve type shall be installed on the outlet flow line of loading valve 2 to prevent cavitation which can occur at the loading valve orifice.

7.10 Safety valve

The test circuit shall have a safety valve (specifically, a direct-operated relief valve) to protect both test equipment and the personnel in attendance from the effects of extreme line pressure. This direct-operated relief valve shall be located as close as possible to the test line to minimize branch circuit interactions. It is desirable that the safety valve is connected directly to the adaptor immediately before loading valve 1. Such a valve shall be set to relieve at a pressure at least 20 % greater than the mean test pressure.

The schematic arrangement of the hydraulic test circuit and that of the measuring system employed for the *two pressures/two systems* method are shown in Figure 4.



Key

1	test pump	10	back pressure valves
2	loading valve 1	11	amplifier
3	loading valve 2	12	variable-speed electric motor
4	safety valve	13	revolution counter
5	PT (piezoelectric pressure transducer)	14	24 bit analysing recorder
6	reference pipe	15	CRT
7	connecting pipe	16	PC for processing
8	extension pipe	17	external trigger signal
9	magnetic gap detector		

Figure 4 — Arrangement of hydraulic test circuit and measuring system for *two pressures/two systems* method

8 Test procedure

8.1 General

Prior to the commencement of a series of the tests, it is necessary to operate the pump with the two loading valve opened for a sufficient period of time to eliminate the air in the system. Then adjust pump drive speed, mean discharge pressure and oil temperature to the values of assigned test condition by means of a variable mechanics of electric motor control loading valve 1 or loading valve 2 and a cooler, respectively.

This measuring method assumes that the speed of sound in test fluid in the reference pipe is known in advance. The speed of sound in the reference pipe shall be determined by the either method described in Annex B.

8.2 Frequency analyses of pressure ripple

Frequency analyses of pressure ripples are to be made in accordance with the following procedure.

a) First, immediately after setting the mean discharge pressure to a desired value by adjusting loading valve 1, the two pressure ripple signals in system 1, $p_0(t)$ and $p_1(t)$, are simultaneously sampled in the frequency analysis instrument (a commercial 24 bit analysing recorder) at the sampling frequency of more than 10,24 kHz. Sampling of the pressure ripple signals in the frequency analysis instrument is triggered by pulse signal obtained from the magnetic gap detector on the pump shaft.

   The pressure ripple analysis instrument shall take samples of the pressure ripple signals over a sufficiently long period of time to provide the required frequency resolution. In this method the sampling time length shall be 0,8 s.

   A 24 bit analysing recorder shall be one with a built-in anti-aliasing filter to reduce the influence of aliasing.

b) Next, immediately after setting the mean discharge pressure by adjusting loading valve 2 to the desired value, two pressure ripple signals in the system 2, $p_0'(t)$ and $p_1'(t)$, are sampled in the frequency analysis instrument in the same manner as in a).

c) Measure the harmonic components of four pressure ripples, P_0, P_1, P_0' and P_1', using a commercial 24 bit analysing recorder and DFT on a PC. Record the individual harmonics of pumping frequency until tenth harmonic or 3,5 kHz (whichever is the lower). Establish the peak amplitude and the phase of each harmonic or its real and imaginary components. Discard any harmonic above tenth harmonic or 3,5 kHz (whichever is the lower).

   Extensive use of complex number is made in this analysis. Therefore, the harmonic components of pressure ripple shall be expressed by complex number as follows:

   $P_{0.i} = \mathrm{Re}\,(P_{0.i}) + j\mathrm{Im}\,(P_{0.i})$

   $P_{1.i} = \mathrm{Re}\,(P_{1.i}) + j\mathrm{Im}\,(P_{1.i})$

   $P_{0.i}' = \mathrm{Re}\,(P_{0.i}') + j\mathrm{Im}\,(P_{0.i}')$

   $P_{1.i}' = \mathrm{Re}\,(P_{1.i}') + j\mathrm{Im}\,(P_{1.i}')$

   where the second suffix i represents the ith harmonic of pressure ripple, and Re(x) and Im(x) the real and imaginary component, respectively.

8.3 Evaluation of source flow ripple, Q_s, in the standard “Norton” model

The ith harmonic component of source flow ripple, $Q_{s.i}$, is calculated using Formula (1) and the values of $P_{0.i}$, $P_{1.i}$, $P_{0.i}'$and $P_{1.i}'$:

$$Q_s = j\frac{1}{Z_c}\frac{P_0P_1' - P_0'P_1}{(P_0 - P_0')\sin(\beta L)}$$

$$\equiv \mathrm{Re}(Q_{s.i}) + j\,\mathrm{Im}(Q_{s.i}) \tag{1}$$

where characteristic impedance, Z_c, and wave propagation coefficient of the reference pipe, β, are given by Formula (B.3) and Formula (B.4) in Annex B.

Harmonic amplitude and phase of Q_s, $|Q_{s.i}|$, and $\angle Q_{s.i}$ are obtained from Formulae (2) and (3):

$$|Q_{s.i}| = \sqrt{\{\mathrm{Re}(Q_{s.i}\}^2 + \{\mathrm{Im}(Q_{s.i}\}^2} \tag{2}$$

$$\angle Q_{s.i} = (180/\pi) \times \tan^{-1}\left\{ \mathrm{Im}(Q_{s.i}) / \mathrm{Re}(Q_{s.i}) \right\} \tag{3}$$

Care should be taken to ensure that the phase lies in the correct quadrant.

8.4 Evaluation of source impedance, Z_s, in the standard "Norton" model

The ith harmonic component of the source impedance, $Z_{s.i}$, is calculated using Formula (4), using the values of $P_{0.i}$, $P_{1.i}$, $P_{0.i}'$ and $P_{1.i}'$:

$$Z_s = jZ_c \frac{(P_0 - P_0')\sin(\beta L)}{P_1 - P_1' - (P_0 - P_0')\cos(\beta L)}$$
$$\equiv \mathrm{Re}(Z_{s.i}) + j\,\mathrm{Im}(Z_{s.i}) \tag{4}$$

The total length of reference pipe after the assembling of test circuit, L_r, to be used in Formula (1) and Formula (4) shall have the accuracy of ±0,5 mm.

NOTE The process of deriving Formulae (1) and (4) is presented in detail in Annex B.

The harmonic amplitude and phase of Z_s, $|Z_{s.i}|$ and $\angle Z_{s.i}$, is obtained from Formula (5) and Formula (6):

$$|Z_{s.i}| = \sqrt{\{\mathrm{Re}(Z_{s.i})\}^2 + \{\mathrm{Im}(Z_{s.i})\}^2} \tag{5}$$

$$\angle Z_{s.i} = (180/\pi) \times \tan^{-1}\left\{ \mathrm{Im}(Z_{s.i}) / \mathrm{Re}(Z_{s.i}) \right\} \tag{6}$$

8.5 Evaluation of source flow ripple, Q_s*, in the modified model

Assuming the discharge flow line to be an equivalent single uniform pipe, having the same characteristic impedance, Z_c, as that of reference pipe, the harmonic component of flow ripple in a modified model, Q_s*, is estimated using the measured pump source flow ripple, Q_s, and source impedance, Z_s, from Formula (7):

$$Q_{s.i}^* = \frac{Z_{s.i}}{\sqrt{Z_{s.i}^2 - Z_c^2}} Q_{s.i}$$
$$\equiv \mathrm{Re}(Q_{s.i}^*) + j\,\mathrm{Im}(Q_{s.i}^*) \tag{7}$$

NOTE 1 The derivation of Formula (7) is discussed in detail in Annex B. The flow ripple in the modified model can be estimated solely from the values of measured Q_s and Z_s, without any other mathematical processes.

NOTE 2 Formula (7) is applicable to the all types of oil hydraulic pumps except those in which the control elements such as flow control valve are equipped in the outlet channel inside the pump (e.g. automotive power steering pump).

The ith harmonic amplitude and phase of Q_s*, $|Q_{s.i}^*|$, and $\angle Q_{s.i}^*$ is calculated from Formula (8) and Formula (9), respectively:

$$|Q_{s.i}^*| = \sqrt{\{\mathrm{Re}(Q_{s.i}^*)\}^2 + \{\mathrm{Im}(Q_{s.i}^*)\}^2} \tag{8}$$

$$\angle Q_{s.i}^* = (180/\pi) \times \tan^{-1}\left\{ \mathrm{Im}(Q_{s.i}^*) / \mathrm{Re}(Q_{s.i}^*) \right\} \tag{9}$$

Furthermore, a close approximation of source flow ripple in the modified model, Q_s*, to time history waveform, $q_s(t)$*, can be obtained by summing up the individual sinusoidal components taking into account their relative phases as explained below. It is sometimes desirable that reconstruction of waveform of the source flow ripples be done in this way. In order to do this, the values of the phases of the source flow ripple are required in addition to values of amplitude.

$$q(t)^* = \sum_{i=1}^{10} \left| Q_{s.i}{}^* \right| \cos\left(2\pi f_i t + \psi_i\right) \quad (10)$$

where

f_i is the ith harmonic frequency of pumping frequency;

t is time;

ψ_i is equal to $\angle Q_{s.i}{}^*$.

8.6 Evaluation of blocked acoustic pressure ripple rating

The harmonic amplitude of the blocked acoustic pressure ripple, $|P_{b.i}|$, is evaluated using the measured source flow ripple, Q_s, and source impedance, Z_s, from Formula (11):

$$\left| p_{b.i} \right| = \left| Z_{s.i} \right| \times \left| Q_{s.i} \right| \quad (11)$$

NOTE The process of derivation of Formula (11) is explained in detail in Annex B.

In addition, Formula (12) is used to obtain the overall RMS of average blocked acoustic pressure ripple amplitude, $|P_{b.RMS}|$, for the integral harmonics of pumping frequency from f_1 to f_{10} or 3,5 kHz (whichever is the lower) for each operating condition.

$$|P_{b\,RMS}| = \sqrt{\frac{|P_{b.1}|^2 + |P_{b.2}|^2 + |P_{b.3}|^2 + \cdots + |P_{b.n}|^2}{2}} \quad (12)$$

NOTE Since all of the equations given in Clause 8 consist of not more than four modes of basic operations, mathematical processes of calculation are relatively easy. Nevertheless, because of the large amount of calculations, data processing is preferably carried out using a digital computer.

9 Test report

9.1 General information and test conditions

General information and test conditions shall be reported on the forms presented in Annex A.

9.2 Test results

The following test results shall be reported in the form of tables (numerical data) or diagrams.

a) Numerical data of harmonic amplitude and phase of the source flow ripple in the standard “Norton” model, $|Q_{s.i}|$ and $\angle Q_{s.i}$, for the individual harmonic of pumping frequency until the tenth harmonic or 3,5 kHz, whichever is the lower (on a form similar to Table A.3, inserted as an example).

b) Amplitude and phase spectra of the source flow ripple in the standard “Norton” model, Q_s, for the individual harmonic of pumping frequency until tenth harmonic or 3,5 kHz, whichever is the lower (see Figure A.1 for an example).

c) Numerical data of harmonic amplitude and phase of the source impedance in the standard “Norton” model, $|Z_{s.i}|$ and $\angle Z_{s.i}$, for the individual harmonic of pumping frequency until the tenth harmonic or 3,5 kHz, whichever is the lower (on a form similar to Table A.3).

d) Amplitude and phase spectra of the source impedance in the standard "Norton" model, $Z_{s,}$, for the individual harmonic of pumping frequency until the tenth harmonic or 3,5 kHz, whichever is the lower (see Figure A.2 for an example).

e) Numerical data of harmonic amplitude and phase of the source flow ripple in the modified model, $|Q_{s.i}*|$ and $\angle Q_{s.i}*$ for the individual harmonic of pumping frequency until the tenth harmonic or 3,5 kHz, whichever is the lower (on a form similar to Table A.3).

f) Amplitude and phase spectra of the source flow ripple in the modified model, Q_s*, for the individual harmonic of pumping frequency until the tenth harmonic or 3,5 kHz, whichever is the lower (see Figure A.3 for an example).

g) Time history waveform of the source flow ripple in the modified model, $q_s(t)*$, for two periods of pumping frequency. (see Figure A.4 for an example).

h) Numerical data of harmonic amplitude of the blocked acoustic pressure ripple, $|P_{b.i}|$, for the individual harmonic of pumping frequency until the tenth harmonic or 3,5 kHz, whichever is the lower (on a form similar to Table A.3).

i) Amplitude spectra of the blocked acoustic pressure ripple, $|P_{b.i}|$, for the individual harmonic of pumping frequency until the tenth harmonic or 3,5 kHz, whichever is the lower (see Figure A.5, for an example).

j) Overall RMS average blocked acoustic pressure amplitude, $|P_{b.RMS}|$, from f_1 to f_{10} or 3,5 kHz, whichever is the lower (on a form similar to Table A.3).

10 Identification statement (Reference to this part of ISO 10767)

Use the following statement in test reports, catalogues and sales literature when electing to comply with this part of ISO 10767:

> "Fluid borne noise characteristics of this pump were obtained and presented in accordance with ISO 10767-1, *Hydraulic Fluid Power — Determination of pressure ripple levels generated in systems and components — Part 1: Method for determining source flow ripple and source impedance of pumps*".

Annex A
(normative)

Test forms

See Tables A.1 and A.2.

Table A.1 — General information form

General information	
1) Name and address of pump manufacturer	
2) Reference number for identification of the pump: i) Manufacturer's code/model number ii) Date of manufacture or date code iii) Serial number iv) etc.	
3) Name and address of organization responsible for test on the pump	
4) Date and place of tests	
5) Conformance statement (see Clause 9)	

Table A.2 — Test information form

Test pump description	
1) Type of pump (e.g. axial piston, external gear) including any ancillary equipment	
2) Type of displacement (e.g. fixed or variable)	
3) Pump maximum displacement	
4) Type of displacement controller and setting	
5) Number of pumping elements	
6) Diameter of discharge port, in millimetres	

Description of pump mounting	
7) Type and power of pump drive	
8) Type of pump mounting and details of any vibration isolation treatment	

Test fluid description	
9) Type of fluid	
10) Kinematic viscosity, in centistokes (cSt)	
11) Fluid density, in kilograms per cubic meter	

Table A.2 *(continued)*

Test fluid description	
12) Speed of sound in test fluid or effective isentropic tangent bulk modulus, in bars	

Discharge line description	
13) length of reference pipe	
14) Inner diameter of reference pipe	
15) Length of extension pipe	
l6) Method for changing standing wave mode in reference pipe (extension pipe or pressure vessel ?)	

Instrumentation description	
17) Details of equipment used to monitor pump operating conditions including type, serial number and manufacturer	
18) Details of pressure transducer used for pressure ripple measurement including type, serial number and manufacturer	
19) Details of equipment used for frequency analysis (commercial 24 bit analysing recorder and DFT on PC)	
20) Bandwidth of frequency analyser	

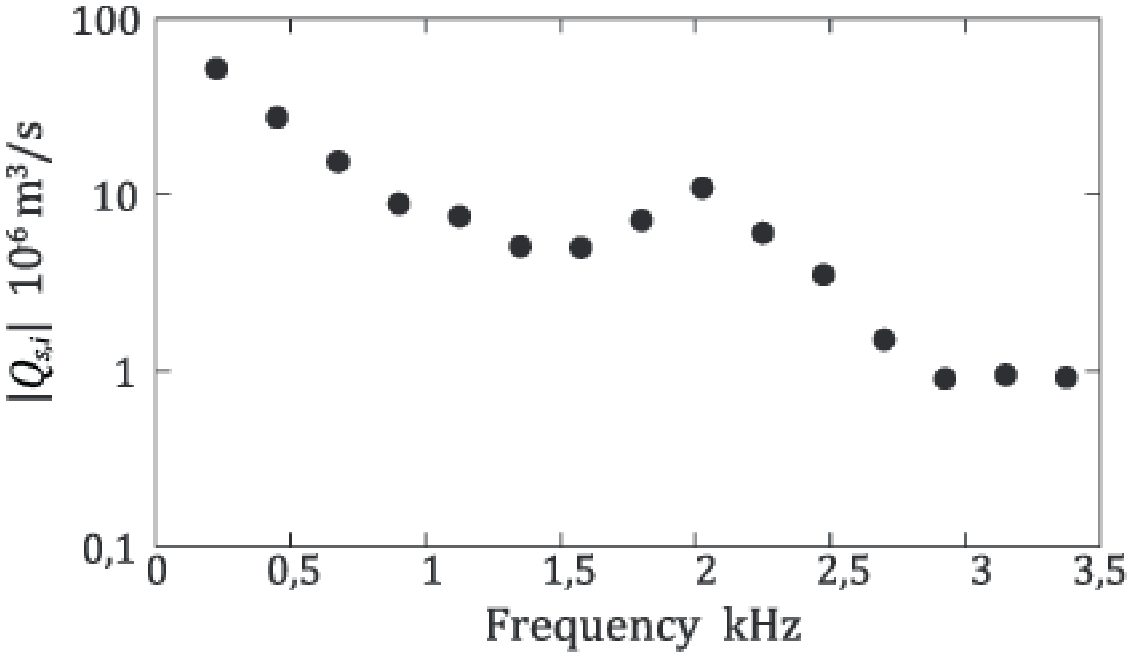
Test conditions	
21) Shaft rotational speed, in revolution per minute	
22) Mean inlet pressure, in bars or megapascals (MPa)	
22) Mean inlet pressure, in bars or MPa	
23) Mean pump delivery flow, in litres per minute	
24) Temperature of fluid at pump inlet, in degrees Celsius	

Table A.3 — Test results (N = 1 500r/min, f_1 = 225Hz, $P_{d.0}$ = 14,0 MPa, Θ = 40°C)

f_i [Hz]	$\|Q_{s.i}\|$ $10^{-6}m^3/s$	$\angle Q_{s.i}$ deg	$\|Z_{s.i}\|$ $10^{10}Pas/m^3$	$\angle Z_{s.i}$ deg	$\|Q_{s.i}^*\|$ $10^{-6}m^3/s$	$\angle Q_{s.i}^*$ deg	$\|P_{b.i}\|$ bar	$\|P_{b.RMS}\|$ bar
f_1 = 225	52,0	30,1	1,021	−90,2	51,9	28,9	5,30	-
f_2 = 450	27,5	−102,5	0,823	−92,5	27,1	−103,5	2,26	-
f_3 = 675	15,5	115,0	0,713	−89,1	12,0	114,0	1,10	-
f_4 = 900	8,95	−85,0	0,523	−85,0	7,01	−86,2	0,47	-
f_5 = 1125	7,55	88,9	0,352	−80,2	4,05	90,2	0,27	-
f_6 = 1350	5,12	−160,5	0,151	−79,0	2,25	−158,1	0,077	-
f_7 = 1575	5,01	−10,1	0,081	−78,0	2,05	−12,0	0,046	-
f_8 = 1800	7,20	150,2	0,052	−65,0	1,52	153,1	0,037	-
f_9 = 2025	11,0	−50,5	0,085	−50,1	1,25	−51,5	0,093	-
f_{10} = 2250	6,10	−70,5	0,151	65,5	1,18	110,2	0,092	4,16
f_{11} = 2475	(3,51)	(85,5)	(0,204)	(70,2)	(1,05)	(−94,5)	(0,072)	
f_{12} = 2700	(1,51)	(−138,5)	(0,405)	(70,5)	(0,82)	(41,5)	(0,061)	
f_{13} = 2925	(0,90)	(−118,0)	(0,605)	(75,0)	(0,85)	(72,1)	(0,054)	
NOTE The values between parentheses are reference values only.								

Table A.3 *(continued)*

f_i [Hz]	$\|Q_{s.i}\|$ $10^{-6}m^3/s$	$\angle Q_{s.i}$ deg	$\|Z_{s.i}\|$ $10^{10}Pas/m^3$	$\angle Z_{s.i}$ deg	$\|Q_{s.i}*\|$ $10^{-6}m^3/s$	$\angle Q_{s.i}*$ deg	$\|P_{b.i}\|$ bar	$\|P_{b.RMS}\|$ bar
f_{14} = 3150	(0,95)	(−45,1)	(0,782)	(90,1)	(0,62)	(135,1)	(0,074)	
f_{15} = 3375	(0,92)	(38,2)	(0,950)	(120,0)	(0,51)	(−142,0)	(0,085)	
NOTE The values between parentheses are reference values only.								



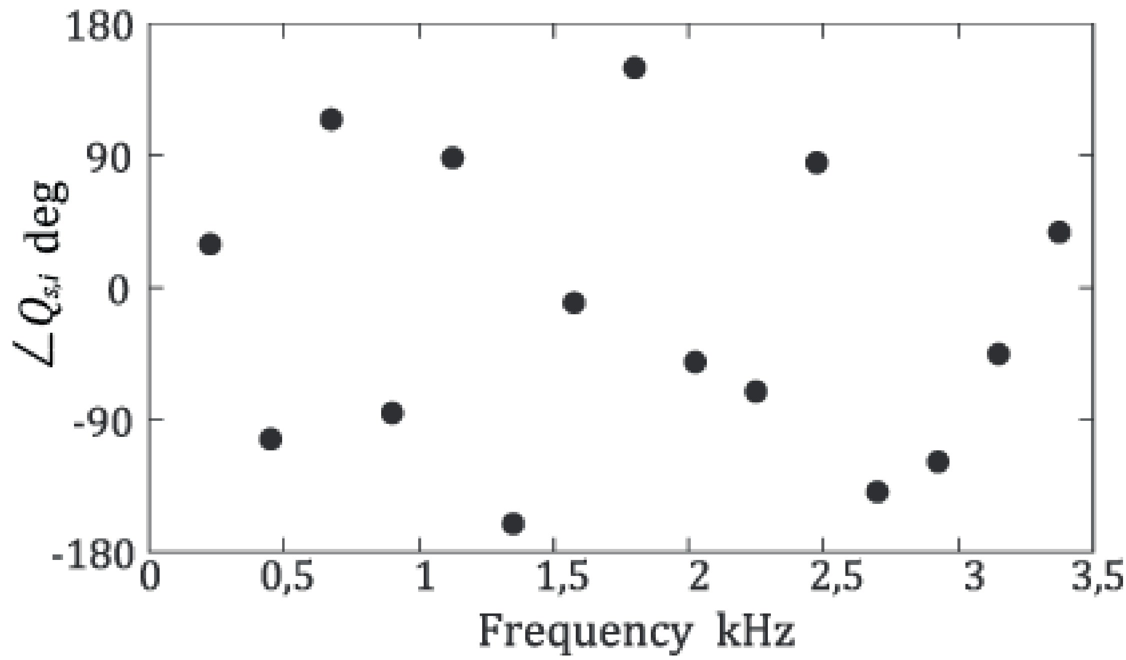
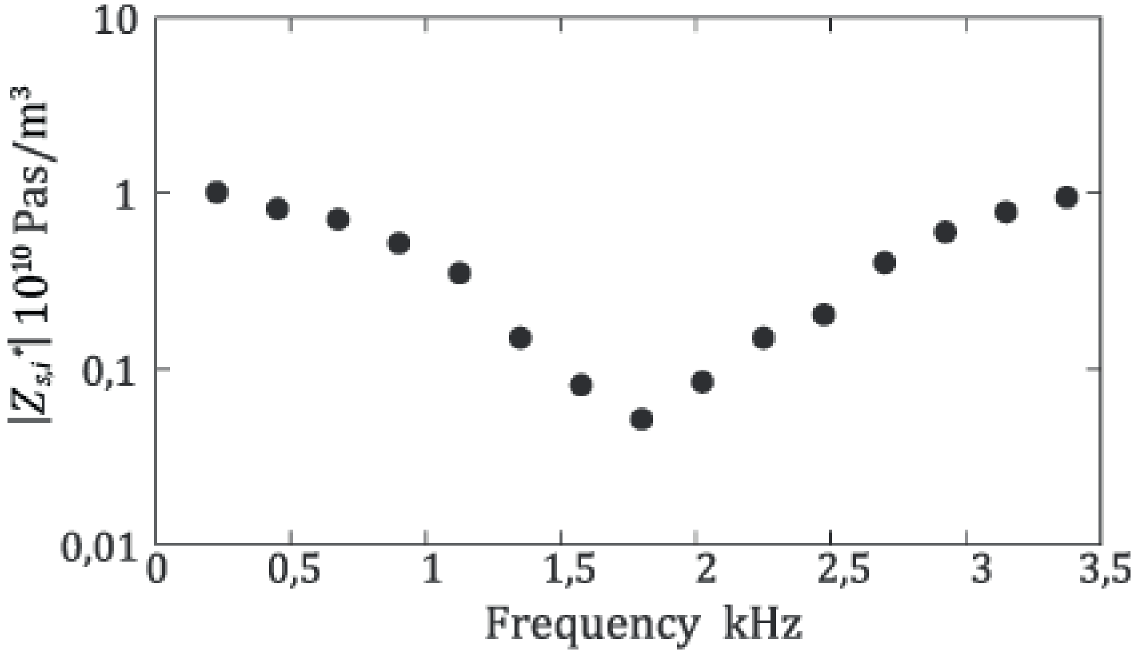


Figure A.1 — Test results of amplitude and phase spectra of source flow ripple Q_S in standard "Norton" model (N = 1 500 r/min, f_1 = 225 Hz, $P_{d.0}$ = 14,0 MPa, Θ = 40 °C) **— Values above tenth harmonic reference values only**



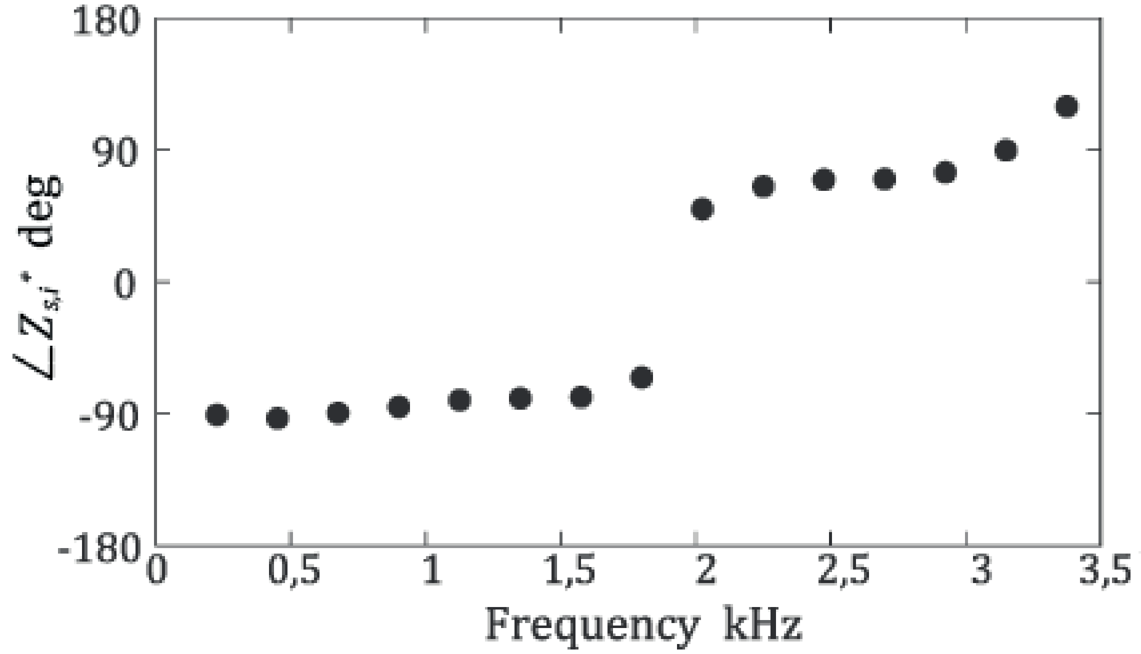
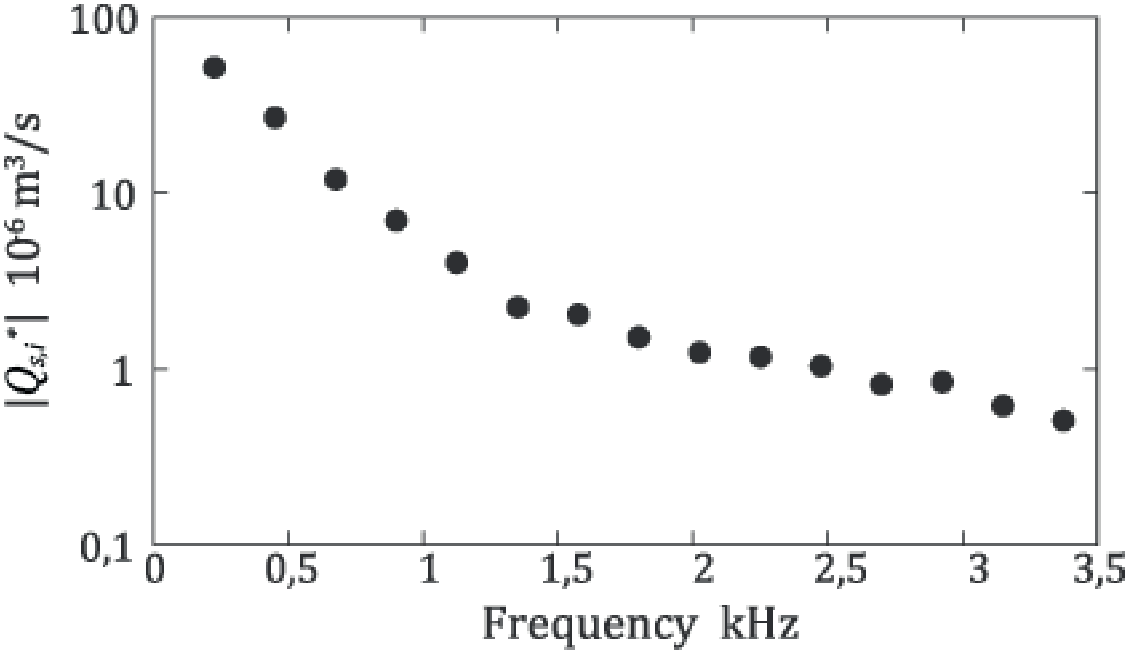


Figure A.2 — Test results of amplitude and phase spectra of source impedance Z_S in standard "Norton" model (N = 1 500 r/min, f_1 = 225 Hz, $P_{d.0}$ = 14,0 MPa, Θ = 40 °C) **— Values above tenth harmonic reference values only**



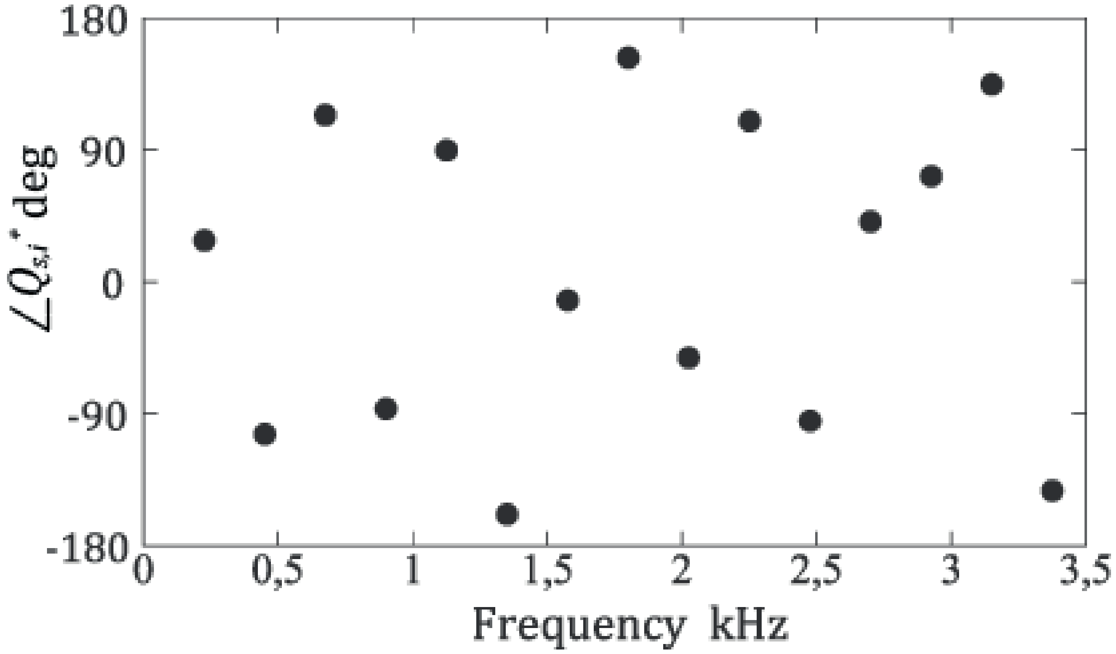


Figure A.3 — Test results of amplitude and phase spectra of source flow ripple Q_S* in modified model (N = 1 500 r/min, f_1 = 225 Hz, $P_{d.0}$ = 14,0 MPa, Θ = 40 °C) **— Values above tenth harmonic reference values only**

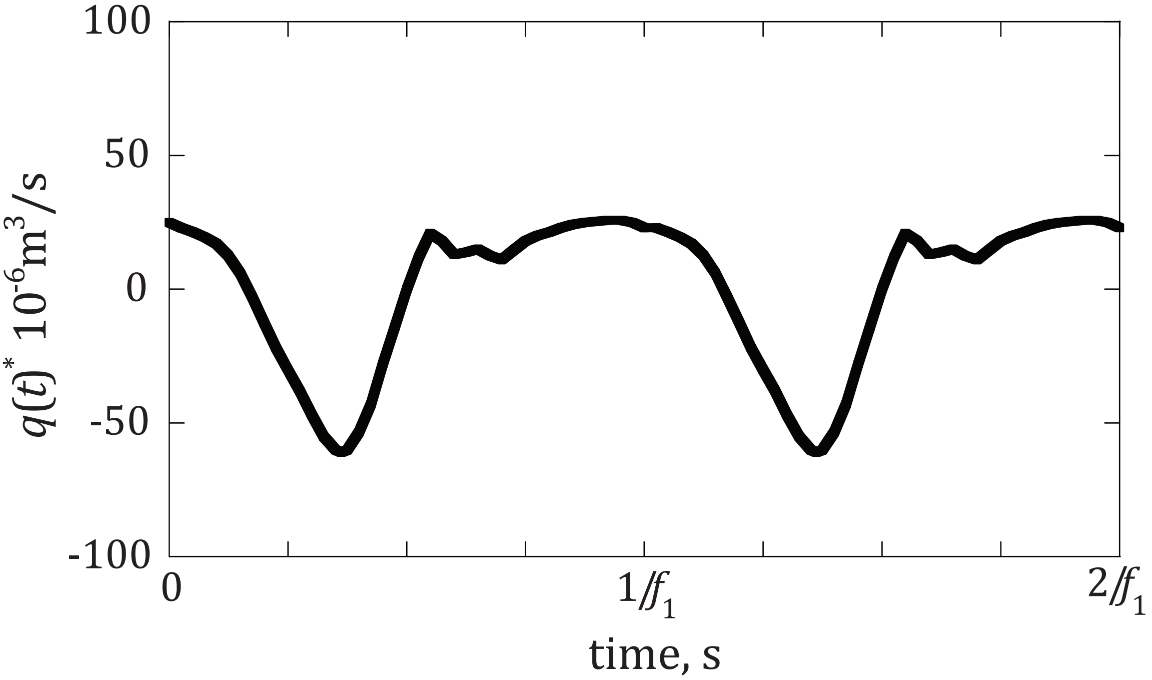


Figure A.4 — Measured time history waveform of the source flow ripple in modified model, $q\ (t)^*$ (N = 1 500 r/min, f_1 = 225 Hz, $P_{d.0}$ = 14,0 MPa, Θ = 40 °C)

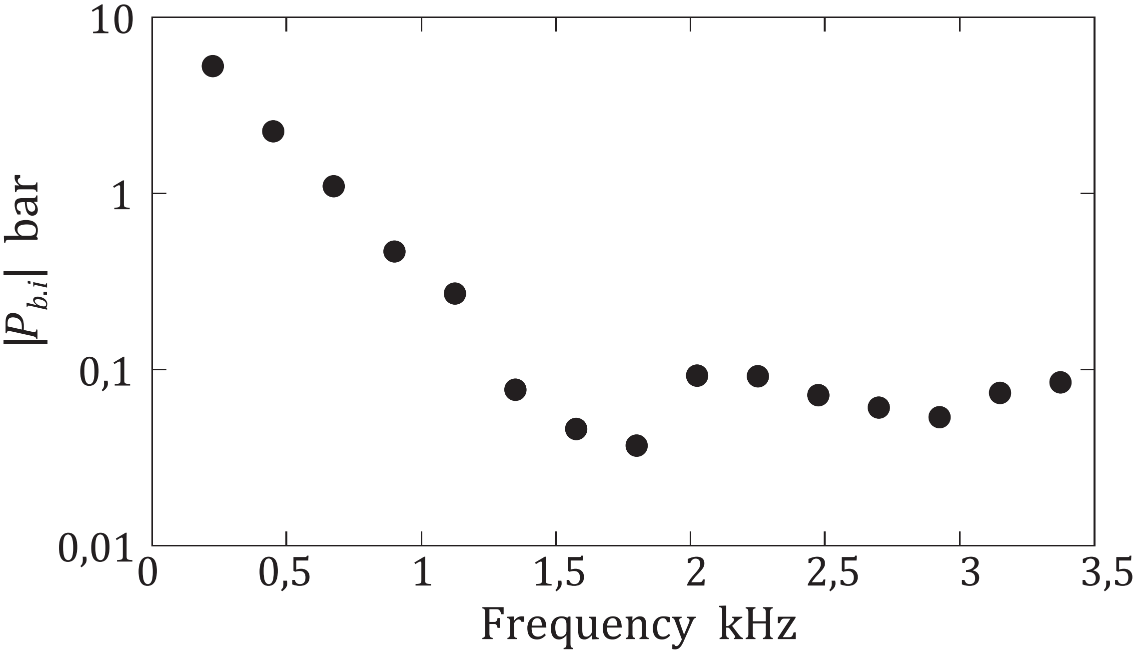


Figure A.5 — Test results of amplitude spectra of the blocked acoustic pressureripple, $|P_{b.i}|$, (N = 1 500 r/min, f_1 = 225 Hz, $P_{d.0}$ = 14,0 MPa,Θ = 40 °C) **— Values above tenth harmonic are reference values only**

Annex B
(informative)

Two pressures/two systems method

B.1 Referential explanation of methodology and theoretical basis for ISO 10767-1 test procedure for measuring pump source flow ripple and impedance

The methodology and the theoretical basis of this measuring method are as follows.

Consider a simple hydraulic circuit shown in Figure B.1 which consists of a test pump, a rigid pipe (reference pipe) of given length L_r, two kinds of connecting pipes of given length L_c (for system (1)) and L_c+L_e (for system 2) and of a loading valve. In such a circuit, the extension pipe of length L_e for system (2) is used for altering the fluid vibration mode (i.e. pattern of the standing wave) in a reference pipe. If P_0, Q_0, and P_1, Q_1 are the pressure ripple and flow ripple at the position of $x = 0$ and $x = L_r$, respectively, in the system (1), and P_0', Q_0' and P_1', Q_1' are those in system 2, then Q_0 and Q_0' can be expressed using P_0 and P_1, and P_0' and P_1' by Formulae (B.1) and (B.2):

$$Q_0 = -j\frac{1}{Z_c}\left\{\cot\left(\beta L_r\right)P_0 - \operatorname{cosec}\left(\beta L_r\right)P_1\right\} \tag{B.1}$$

$$Q_0' = -j\frac{1}{Z_c}\left\{\cot\left(\beta L_r\right)P_0' - \operatorname{cosec}\left(\beta L_r\right)P_1'\right\} \tag{B.2}$$

Characteristic impedance, Z_c, wave propagation coefficient of the reference pipe, β, and complex coefficient representing the effect of unsteady viscous friction, $\xi(\omega)$, are given by Formulae (B.3), (B.4) and (B.5), respectively:

$$Z_c = \frac{pc\xi\left(\omega\right)}{\pi r_0^2} \tag{B.3}$$

$$\beta = \frac{\xi\left(\omega\right)\omega}{c} \tag{B.4}$$

$$\xi\left(\omega\right) \approx 1 + \sqrt{\frac{\nu}{2{r_0}^2\omega}} - j\left(\sqrt{\frac{\nu}{2{r_0}^2\omega}} + \frac{\nu}{{r_0}^2\omega}\right) \tag{B.5}$$

where ω is the angular frequency of the pressure ripple, ν is the kinematic viscosity of the test fluid and r_0 is the inner radius of the reference pipe.

Modelling the pump pulsation source as an imperfect flow source with shunt impedance, as shown in Figure 1 a), Formulae (B.6) and (B.7) can be derived from the continuity law of flow-rate at the pump exit (at $x = 0$):

$$Q_0 = Q_s - \frac{P_0}{Z_s} \tag{B.6}$$

$$Q_0' = Q_s' - \frac{P_0}{Z_s} \tag{B.7}$$

If the phases of pressure ripple, P_0, P_1, P_0' and P_1' are all measured on the basis of an identical angular reference position of the pump shaft, pump flow ripple Q_s in Formula (B.6) can be equal to Q_s' in Formula (B.7), i.e.

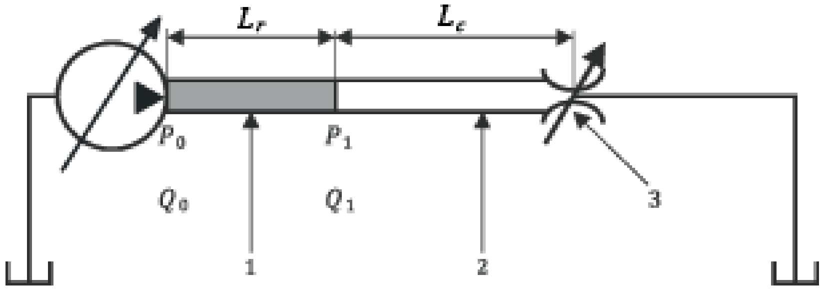
$$Q_s = Q_s' \tag{B.8}$$

Thus, from these premises, the following equations can be derived from Formula (B.1) to Formula (B.8) for evaluation of pump source flow ripple Q_s and source impedance Z_s:

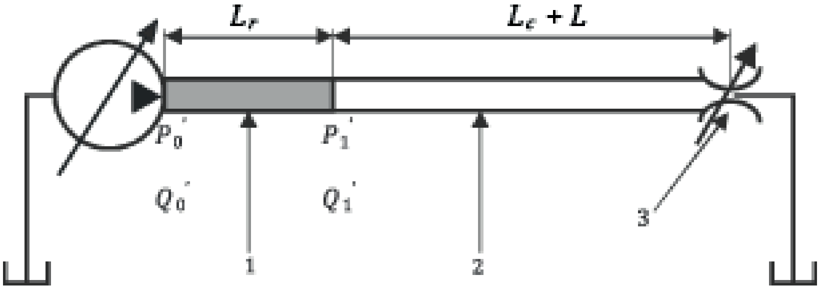
$$Q_{s.i} = j\frac{1}{Z_c}\frac{P_{0.i}P_{1.i}' - P_{0.i}'P_{1.i}}{\left(P_{0.i} - P_{0.i}'\right)\sin\left(\beta L_r\right)} \equiv \mathrm{Re}(Q_{s.i}) + j\,\mathrm{Im}(Q_{s.i}) \tag{B.9}$$

$$Z_{s.i} = jZ_c\frac{\left(P_{0.i} - P_{0.i}'\right)\sin\left(\beta L_r\right)}{P_{1.i} - P_{1.i}' - \left(P_{0.i} - P_{0.i}'\right)\cos\left(\beta L_r\right)} \equiv \mathrm{Re}(Z_{s.i}) + j\,\mathrm{Im}(Z_{s.i}) \tag{B.10}$$

It can be seen from Formula (B.9) and Formula (B.10) that neither the characteristics of the loading valve nor the properties of pipeline with the exception of the reference pipe are included in the calculation for evaluating Q_s and Z_s. That is, the loading valve and extension pipes are used merely for changing the mean discharge pressure and for altering the pattern of the standing wave in a reference pipe, respectively.



a) System 1



b) System 2

Key

1 reference pipe

2 connecting pipe

3 loading valve

Figure B.1 — Hydraulic circuit illustrating methodology and theoretical basis of "two pressures/two systems" technique

B.2 Discrete Fourier transformation

The following assumptions are made for the Fourier transformation of a discrete time signal $x(n\,\tau)$:

a) $x(n\,\tau)$ is to be a discrete time signal, that is to say $x(n\,\tau)$ exists only at the time of n = 0, 1, 2, …N-1, in which N is a positive integer;

b) angular frequency ω is a discrete angular frequency $\omega_{\mathbf{k}}$ and exists at regular intervals dividing by N, the frequency range from DC to sampling angular frequency ω_S, as expressed by

$$\omega_k = \omega_S k/N = 2\pi f_S k/N = 2\pi k/N\tau \tag{B.11}$$

where f_S is a sampling frequency and $\tau(=1/f_S)$ a sampling period, and N is the number of sampling data.

Under these conditions, discrete Fourier transformation of $x(n\,\tau)$ can be given by

$$X(i\omega_k) \equiv X(k) = \sum_{n=0}^{N-1} x(n\tau)e^{-j\frac{2\pi nk}{N}}$$
$$\equiv \mathrm{Re}\{X(k)\} + j\,\mathrm{Im}\{X(k)\}, \qquad k = 0, \quad 1, \quad 2, \quad \ldots \mathrm{N}-1 \tag{B.12}$$

Hence, the harmonic amplitude and phase of $X(k)$, $|X(k)|$ and $\angle X(k)$, can be obtained from the following equations, respectively:

$$|X(k)| = \sqrt{[\mathrm{Re}\{X(k)\}]^2 + [\mathrm{Im}\{X(k)\}]^2} \tag{B.13}$$

$$\angle X(k) = (180/\pi) \times \tan^{-1}\left[\mathrm{Im}\{X(k)\} / \mathrm{Re}\{X(k)\}\right] \tag{B.14}$$

NOTE 1 Using a common type 24 bit analysing recorder, and setting the sampling frequency to more than 10,24 kHz and sampling time length to 0,8 s (i.e. in this case, the number of sampling data are more than 8192), resolution power of frequency can be improved to 1,25 Hz.

NOTE 2 Taking into consideration the setting accuracy of the pump rotational speed (i.e. setting error of the pumping frequency) as well as the above-mentioned frequency resolution, it is possible to be considered that a maximum value of $|X(k)|$ in the vicinity of the expected *n*th harmonic frequency of the pressure ripple corresponds to the *n*th harmonic amplitude .

NOTE 3 Accordingly, in the case of using a common type analysing recorder, required setting accuracy of the pump speed (i.e. of the pumping frequency) is not so severe as that in the case of using a commercial FFT analyser (the frequency resolution power of it is usually 12,5 Hz, because the sampling time length is usually set to 0.08 sec), and setting accuracy of around ±1 % is sufficient.

B.3 Determination of length of reference and extension pipes

The lengths of the reference pipe and the extension pipe are determined on the basis of the following considerations. Since sin ($ßL_r$) approximately 0, cos ($ßL_r$) approximately $(-1)^n$, P_0 approximately $(-1)^n P_1$ and P_0' approximately $(-1)^n P_1'$ when the length of the reference pipe is close to a half of the wavelength, i.e. when $L_r = nc/2f$; n = 1, 2, 3 …, both Formula (1) and Formula (4) become indefinite equations of 0/0 independent of the length of extension pipe, L_e, so they are of no use for evaluation of pump source flow ripple nor source impedance. Similarly, since $P_0 \approx P_0'$ and $P_1 \approx P_1'$ when the length of the extension pipe is close to a half wavelength, both Formula (1) and Formula (4) also become indefinite equations of 0/0 independent of the length of the reference pipe, L_r.

Meanwhile, the length of the extension pipe should be chosen so as the change of pattern of the standing wave in the reference pipe to be obtained is as large as possible over the frequency range of interest. Likewise, the length of the reference pipe to be taken should be as long as possible in order to maximize the difference between P_0 and P_1 (and, P_0' and P_1'). Hence, the optimum lengths of these pipes are to be as long as possible within the range of less than a half wavelength at the maximum frequency f_{max} .

For example, considering the frequency range of interest to be from 200 Hz to 3,5 kHz and assuming the speed of sound to be 1 390 m/s, the smallest half wavelength is about 02 m [$c/2f_{max}$ = 1 390/(2 × 3 500) ≈0,2 m]. In this case, assuming the margin for ill condition to be around 75 %, an optimum length of the reference pipe and the extension pipe is found to be around 0,15 m (≈150 mm).

B.4 Determination of speed of sound

Method 1

Values of the speed of sound in the test hydraulic fluid, c_0, and the test fluid mass density, ρ, can be obtained from the manufacturer of the fluid. The speed of sound in the test fluid can be corrected for elasticity of the reference pipe using

$$c = \sqrt{\frac{1}{\frac{1}{{c_0}^2} + \frac{(D_r + h)\rho}{Eh}}} \quad \text{(B.15)}$$

where, ρ is the mass density of the test fluid, E is Young's modulus, D_r the inner diameter and h the wall thickness of the reference pipe.

If a value for the speed of sound in the test fluid, c_0, is not available from the manufacturer of the fluid it can be estimated using the bulk modulus B and mass density ρ of the test fluid from

$$c_0 = \sqrt{\frac{B}{\rho}} \quad \text{(B.16)}$$

Method 2

If the physical properties of the test fluid such as bulk modulus and mass density are not available, speed of sound in the reference pipe is experimentally estimated using the method given in ISO 15086-2.

B.5 Theory of correlation of source flow ripple inside pump

Let P_0 and Q_0 be the pressure ripple and flow ripple at the initial point of the reference pipe (i.e. at the pump exit). From Figure 1 a), the standard "Norton" model representation, the following equation of continuity can be derived:

$$Q_s = \frac{P_0}{Z_s} + Q_0 \quad \text{(B.17)}$$

Let [T] denote the transfer matrix of the discharge passageway of the pump:

$$T = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \quad \text{(B.18)}$$

The relationship between the flow ripple in the modified model, Q_s*, and P_0 and Q_0 is given using the matrix parameters by

$$Q_s{}^* = T_{21} P_0 + T_{22} Q_0 \quad \text{(B19)}$$

On the other hand, since the flow ripple at the inner end of the discharge passageway is assumed to be zero, in the standard "Norton" model representation, P_0 and Q_p are related to each other by

$$0 = T_{21} P_0 + T_{22} Q_p \quad \text{(B.20)}$$

where Q_p is the flow ripple flowing out the discharge passageway.

The source impedance in the standard "Norton" model is defined as the ratio of pressure ripple, P_0, to the flow ripple flowing into discharge passageway, -Q_p, i.e.

$$Z_s = \frac{P_0}{(-Q_p)} \quad \text{(B.21)}$$

From Formula (B.20) and Formula (B.21), Z_s is found to be

$$Z_s = T_{22} / T_{21} \ \text{ or } \ T_{22} = T_{21} Z_s \quad \text{(B.22)}$$

Substituting T_{22} into Formula (B.19) gives

$$Q_s{}^* = T_{21} Z_s \left(\frac{P_0}{Z_s} + Q_0 \right) \quad \text{(B.23)}$$

Combining Formula (B.17) and Formula (B.23), we have

$$Q_s{}^* = T_{21} Z_s Q_s \quad \text{(B.24)}$$

If the discharge flow channel in the pump casing is assumed to be an equivalent single uniform pipe which has the same characteristic impedance, Z_c, as that of the reference pipe, the following equations regarding the transfer matrix parameters can be satisfied:

$$T_{11} T_{22} - T_{12} T_{21} = 1 \quad \text{(B.25)}$$

$$T_{11} = T_{22} \quad \text{(B.26)}$$

$$T_{12} = Z_c{}^2 T_{21} \tag{B.27}$$

From Formula (B.22) and Formula (B.25) to Formula (B.27), can be derived

$$T_{21} = \frac{1}{\sqrt{Z_s^2 - Z_c^2}} \tag{B.28}$$

Hence, from Formula (B.24) and Formula (B.28), the relationship between Q_s, Z_s and Q_s* can be obtained

$$Q_s^* = \frac{Z_s}{\sqrt{Z_s^2 - Z_c^2}} Q_s \tag{B.29}$$

As it can be seen from Formula (B.29), Q_s* can be evaluated directly from only the measured values of Q_s and Z_s without any arithmetic calculations.

The length of a single uniform pipe equivalent to the discharge passageway, L_d, can be estimated generally from the quarter wavelength resonant frequency, f_r, seen clearly in the frequency characteristics of Z_s (see Figure A.2). That is, L_d can be obtained from

$$L_d = \frac{c}{4 f_r} \tag{B.30}$$

Formula (B.22) is expressed as follows using the estimated value of L_d:

$$Z_s = T_{22} / T_{21} = \frac{\cosh(\beta_d L_d)}{\sinh(\beta_d L_d) / Z_c}$$

$$= -j \frac{Z_c}{\tan(\beta_d L_d)} \tag{B.31}$$

Hence, from Formula (B.26) and Formula (B.28), the relationship between Q_s and Q_s* can be derived as follows, using the estimated value of L_d:

$$Q_s{}^* = Q_s \cos(\beta_d L_d) \tag{B.32}$$

Formula (B.32) agrees with the equation originally derived[1] as the relationship between Q_s and Q_s*.

B.6 Blocked acoustic pressure ripple

Using the standard “Norton” model as the pump pulsation source, the pressure ripple at the outlet of the pump (i.e. at the entrance of the transmission line connected to the outlet) can be calculated using Formula (B.33):

$$P_0 = \frac{Z_s Z_e}{Z_s + Z_e} Q_s \tag{B.33}$$

where Z_e is the entry impedance of a transmission line.

It can be seen from Formula (B.33) that if Z_e was infinitely high (equivalent to blocking the outlet port), the pressure ripple that would be developed at the pump outlet port, P_b, is expressed as

$$P_0 \equiv P_b \approx Z_s Q_s \tag{B.34}$$

If necessary, by substituting $Z_e = Z_c$ into Formula (B.33), anechoic pressure ripple, $P_{a.e}$, which would occur at the pump outlet port when discharging into an infinitely long pipe with characteristic impedance Z_c, can also be found using

$$P_{a.e} = \frac{Z_s Z_c}{Z_s + Z_c} Q_s \tag{B.35}$$

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