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OPTIMISATION OF FLOW MEASUREMENTS IN A PULSATING FLOW -EXPERIENCES FROM FIELD MEASUREMENTS-

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1 ABSTRACT

Analyses of unsteady flow in pipe systems are normally applied in case periodic pulsations are to be expected as a result of well known sources like reciprocating compressors or other positive displacement machinery. The API 618 Standard for reciprocating compressors recommends such a pulsation and mechanical response analysis to prevent unacceptable vibration levels and cyclic stresses in the piping and connected instrumentation. However the impact of pulsating flow on flowmeter accuracy is not considered and criteria for allowable pulsations at flowmeters in relation to metering errors are not specified in the present edition of the standard.
Though it is well known that pulsations may have a considerable impact on traditional techniques like orifice metering, turbine and vortex flowmeters, moreover recent investigations have shown that also ultrasonic flowmeters may be influenced by pulsations.
Flowmeter manufacturers do not often refer to uncertainties as a result of unsteady flow, nor do they define criteria with respect to allowable pulsation amplitudes or frequencies for their flowmeters. Amplitude thresholds can be defined for differential pressure type flowmeters and turbine flow meters without reference to pulsation frequency. In the case of a vortex flowmeter the pulsation frequency relative to the vortex frequency is a much more important parameter than the velocity pulsation amplitude. The impact on flowmeters differs considerably dependent on the measuring technique and the amplitude and frequency of pulsations.

Little information is available so far on actual flow pulsations occurring in flow metering stations due to excitation by compressors or induced by flow due to vortex shedding at T-joints, reducers or valves. This paper shows the results of three cases in which pulsation measurements on-site have been performed and analysed for different flow metering stations with and without compressors.

The first case describes the pulsating flow caused by a number of parallel-operating reciprocating compressors. We investigated the impact of modifications in the piping on the pulsation levels at the flow meters by means of our 1D-simulation software package PULSIM. The actual pulsation levels, determined by on-site measurements in the unmodified lay out, show that there is a considerable impact on a 12-inch turbine and vortex flowmeter, placed in series on the suction inlet line.

In the second case on-site measurements prove that high frequency pulsations caused by centrifugal compressors are damped effectively at a relatively short distance from the compressor station dependent on the geometry of the piping. The on-site measurements convinced the parties involved that a flow metering station could be located at the same site as the compressors.
The layout of the flow metering station, the gas flow velocity and density are important parameters in the occurrence of flow induced pulsations.
The impact of piping geometry and gas properties is illustrated on experiences described in case studies 2 and 3.
The cases described in this paper show that 1D-simulation can be an effective tool in predicting pulsation levels in flow metering stations and in optimising the lay out to obtain minimum flow and pressure pulsations in order to minimise the uncertainty in flowmeter readings. On the other hand a

continuous effort in flowmeter design and signal processing is necessary to improve flowmeter accuracy in case of disturbances like pulsations and vibrations.

2 AN OVERVIEW OF PULSATION IMPACT ON FLOWMETERS

Systematic errors caused by a pulsating flow can be positive or negative and are in general related to flow or velocity pulsation amplitude and frequency. In contrast to the criterion of *pressure* pulsation as specified in API 618 for pipe systems in relation to pulsation forces, the *flow or velocity* pulsation amplitude (and frequency) determines the error in reading.
For purely sinusoidal pulsations the systematic error for orifices and turbine flowmeters can be quantified in direct relation with flow pulsation amplitude and frequency. This aspect should be taken into account in a pulsation analysis according API 618 and errors in reading can be estimated based on the flow pulsations, calculated in the analyses at the flowmeter location.
In ISO/TR 33313 the threshold for differential pressure type flowmeters is defined as $U'_{rms}/U_{mean} \leq 0.05$ and for turbine flowmeters $U'_{rms}/U_{mean} \leq 0.035$, corresponding to a systematic error less than respectively +0.125 % for orifice metering and +0.1% for a turbine flow meter.
In those case were considerable pulsation levels are calculated the study may reveal that the flowmeter location should be altered or that additional measures are necessary to dampen pulsations to achieve acceptable levels by means of a pulsation damper, additional friction, pipe modifications or a combination of these. In most cases the pulsations caused by compressors are periodic, but not necessarily sinusoidal, so that the relation between pulsation levels and misreading cannot be quantified accurately.
An overview of the impact of pulsations on different metering techniques and references to standards and published literature is shown in the table below:

Flowmeter Technique	**Origin of Systematic Error**	**Standard for Flowmeter**	**Standard Pulsation Impact**	**Criterion in ISO/ TR 3313**	**References**
Dp:Orifice, Nozzle and Venturi	Square-root error and gauge line errors	ISO 5167 AGA rep. 3 (API 2530)	ISO/TR 3313	U_{all}= 5 % rms	1,2
Turbine	Inertia of the rotor and fluid	ISO 9951	ISO/TR 3313	U_{all}= 3.5 % rms	3,4,5
Ultrasonic	Aliasing error	ISO TR 12765 AGA rep.9	-	-	8,9
Vortex	Lock-in	ISO TR 12764	ISO/TR 3313	-	6,7
Coriolis	Lock-in	ISO 10790	-	-	10,11,12
Electro Magnetic	Unknown	ISO	-	-	

Table 2.1 Overview of pulsation impact on various flowmeters

3 PULSATION IMPACT ON A TURBINE AND VORTEX FLOWMETER CLOSE TO A RECIPROCATING COMPRESSOR

A station for natural gas storage and transport is supplied with five similar reciprocating compressors with a speed variation between 650 and 800 rpm. The station is extended with an additional reciprocating compressor with a variable speed between 600 and 1000 rpm to obtain the maximum flow of approximately 60.000 Nm3/hr. The gas is compressed from a suction pressure between 2600

and 7100 kPa to be stored in an underground storage at a discharge pressure between 4500 and 8600 kPa.
A pulsation analyses according to the API618 standard is required, which should also include an analyses of the pulsation impact on the flow metering station located in the suction piping. This study should reveal what measures are necessary to limit the maximum pulsations caused by parallel operating compressors. A schematic layout of the gas compression and metering station is shown in fig. 3.1

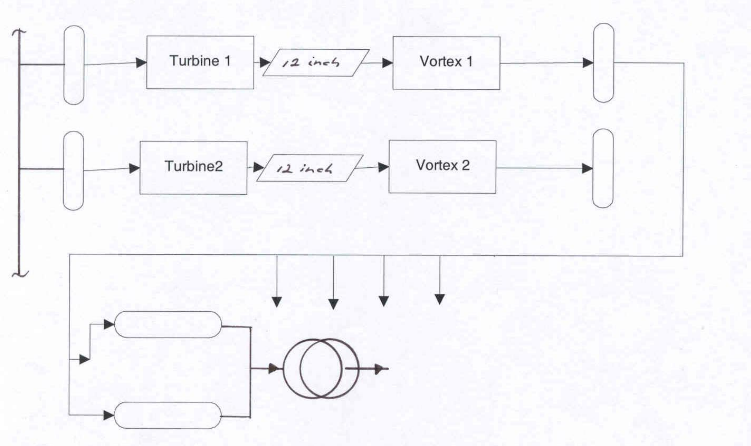

Fig.3.1 Schematic lay-out of compressor and metering station

The pulsation analyses are performed with our simulation package PULSIM, developed for the calculation of plane-wave propagation in pipe systems and fluid machinery. The 1-D approach has shown accurate results for low-frequency pulsations in pipe-systems restricted to a frequency $f < 0.586\ c/D$, in which c is the speed of sound in the gas and D is the pipe diameter.

The pulsation analyses show that considerable pressure pulsations, above API618 limits, and corresponding pulsation induced vibration forces are caused by acoustic resonances between the compressors, mainly on suction side. As a result of the parallel operating compressors, running at different phase and/or speed, beating pulsations will excite the pipe system. The maximum amplitude of the beating pulsation is found by adding the individual pulsation amplitude caused by each compressor.
The flow pulsations at the flow metering station, upstream of the compressors on suction piping are varying between 20 % peak-to-peak for one compressor up to 100 % peak-to-peak for 5 compressors in parallel. This flow pulsation level will cause a considerable systematic error at the turbine and possibly also at the vortex flowmeter. The pulsation amplitude and frequency determine the error in the turbine flowmeter reading, which is due to rotor inertia. A best estimate based on the theory of Bonner and Lee [3,4] is that the turbine metering error ranges from +0.5 to +10%.
The analyses reveal that pulsations cannot be damped effectively by simple means, such as individual orifice plates at the pulsation dampers, without causing excessive pressure losses at increasing flow. An additional damper volume, size 2.5 m length and diameter 0.65 m, is recommended to reduce the flow pulsation at the flowmeters to 2% pp for each individual compressor.
The new-installed compressor could be supplied with an acoustical filter, which dampens pulsation levels successfully over the entire operating envelope. Originally the filter consists of a two-chamber damper with two cylinders on each side of the compressor, which reduces the pressure pulsations within the API 618 criteria. Further reduction is required as to reduce the maximum flow pulsation at the flow meter, which is still 10 % pp. A secondary damper volume, size 2.5 x 0.55m, has been recommended to obtain a level of approximately 2 % pp.

An overview of the reduction in pulsation levels obtained with different recommendations is shown in the table below.

Modification	Maximum flow pulsations, U in % pp	Estimated systematic error In turbine meter
Original lay-out	110 % pp	< 10 %
Modification 1 with orifice plates at suction dampers (dP= 1%)	50 % pp	< 5 %
Additional volume bottle of 0.8 m3 at each compressor plus orifice plates and modified piping for new compressor	11.5 % pp	< 0.5 %
Modification 2 with control valve on each suction line with a pressure drop of 5 %	30 % pp	< 1.0 %

Fig. 3.2 Table showing the simulation results of various modifications

In addition to the model analyses we have been requested by the operator to perform on-site pulsation measurements to be able to determine actual levels and corresponding systematic errors at the flow meters.
Pressure pulsation measurements have been performed on 5 different locations on the 12 inch metering section (see fig. 3.3) at various conditions with1 up to 5 compressors running.

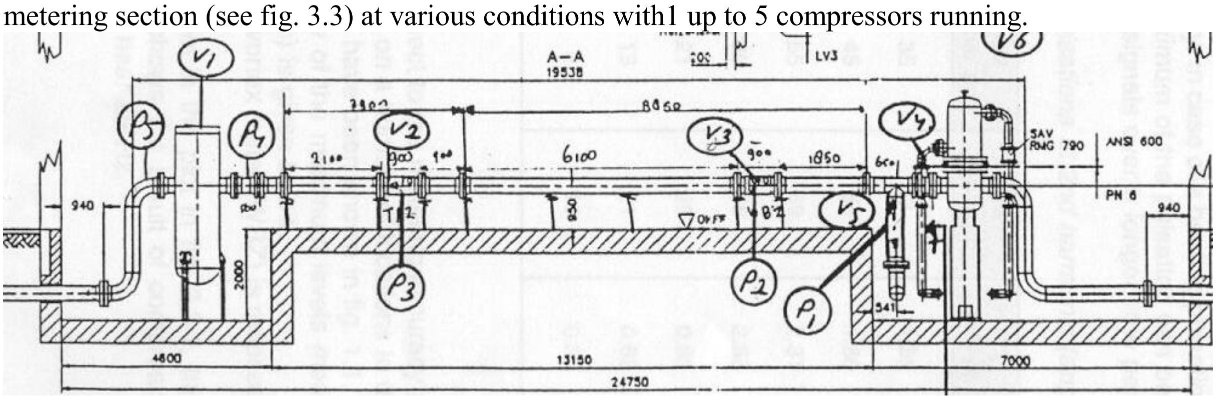

Fig.3.3. Locations for pulsation (P) and vibration measurement (V) at the flow metering station

One of the compressors is varied in speed, whilst the others run at a fixed speed of 750 rpm. Pressure pulsations vary slightly with speed: the maximum level measured is 40 kPa pp, which is approximately 1 % pp of the line pressure of 4300-4500 kPa. The dominant frequency is 25 Hz, which is 2nd harmonic of the compressor speed. This is well in line with the simulation results, which show a 2nd harmonic resonance at 780 rpm.

We have calculated flow pulsations from the measured pressure pulsations by means of the "two-microphone method". The analyses show a maximum flow velocity pulsation of 2.6 m/s pp (0.91 m/s rms) at 25 Hz. This value is measured with 3 compressors running in parallel, with a mean flow velocity in the 12-inch metering line of 6.0 m/s (60.000 Nm3/hr). An example of the (measured) pressure and (calculated) flow pulsation at turbine (P1) and vortex flowmeter (P3) is shown in fig. 3.4

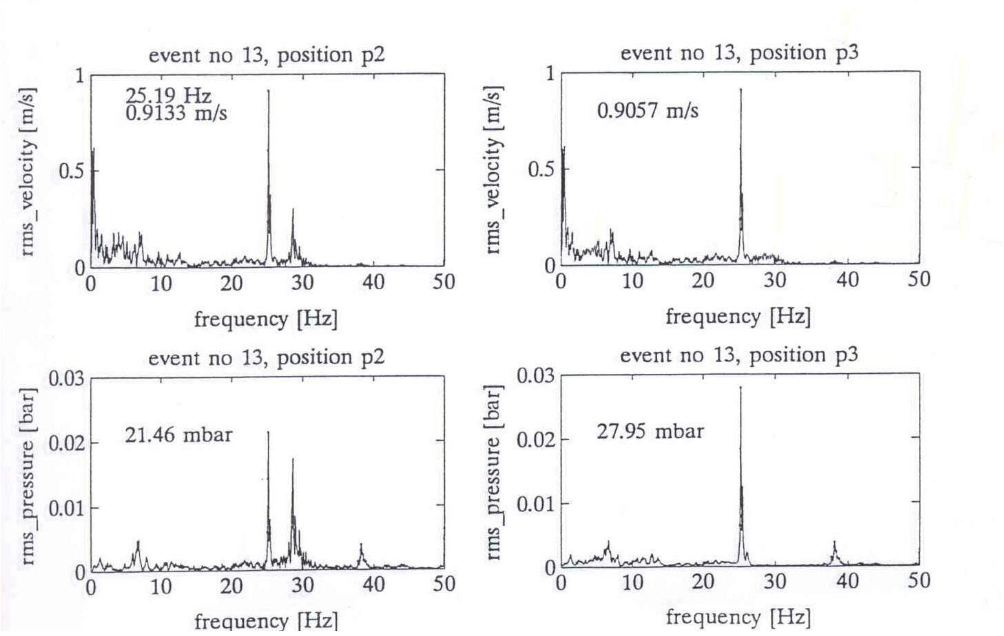

Fig.3.4 Spectra of pressure and flow pulsation at turbine (P3) and vortex flowmeter (P2)

The corresponding systematic error at the turbine flow meter at this flow pulsation level is estimated at + 2.1 %, based on the manufacturer's data and assuming a sinusoidal pulsation.
The systematic error at the vortex flowmeter cannot be determined simply as the relation between flow pulsation and error in reading depends on the individual meter design and the sensor used. Though an earlier investigation of different make vortex flowmeters has shown, that a pulsating flow affects the vortex shedding process, such that the vortex shedding frequency can lock to the pulsation frequency. The strongest lock in occurs, when $f_v/f_p = 0.5$, though lock-in also occurs at $f_v/f_p = 0.25$, .5, 1.0, 1.5,2.0
Actual lock-in has not been noticed at the maximum flow pulsation occurring at 25 Hz of 2.6 m/s peak-to-peak at 6.0 ms/ mean flow, which is 43 % pp.
The relation between vortex frequency and bluff body diameter is presented as:
$f_v = Str*v/D$ in which:

f_v :	vortex frequency in Hz
Str:	Strouhal number 0.3
v:	flow velocity in m/s
d	bluff body diameter 0.3 pipe diameter

According to this expression the vortex frequency f_v for a mean flow velocity of 6.0 m/s is 20 Hz, which is not far from the pulsation frequency of 25 Hz. For the total flow range of 1.500 – 120.000 Nm3/h the vortex frequency for the 12–inch flowmeter varies between 5 and 40 Hz.

This is partially within the range of pulsation frequencies excited by the compressors running between 600 and 1000 rpm (10 – 16.6 Hz) and exciting mainly 2nd harmonic of compressor speed. It should also be noticed that fluctuation in pulsation and vibration levels occurs if compressors run at different speeds. The beating frequency of the pulsation is determined by the difference in compressor speed, as shown in the example fig. 3.5

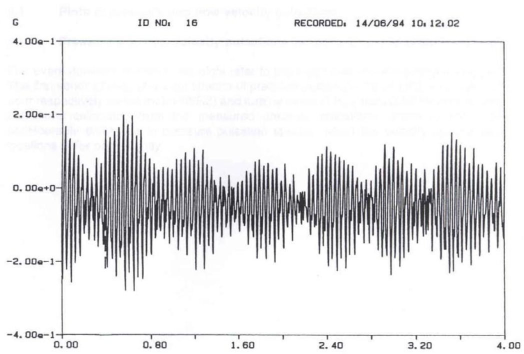

Fig.3.5 Example of a beating pulsation as a function of time (left)

The pulsation errors on vortex flowmeters are found to be mainly negative, especially at lock-in, though also positive errors occur if vortex frequencies are approaching towards pulsation frequencies. The figures of actual flows of the flowmeters are only available as mean-hour flows, showing a deviation between turbine and vortex flowmeter of approximately +1.5 %. This confirms the theoretical positive systematic error of the turbine meter assuming the error in the vortex flowmeter is negative or neutral if no lock-in occurs.
The client will now taken measure to decrease pulsation levels, as resulting from the simulation studies. In this way systematic errors are reduced effectively to a level below 0.5 % over the entire operating envelope of the compressors.

4 IMPACT OF PULSATIONS FROM A TURBOCOMPRESSOR AND ON AN ORIFICE METERING STATION

A natural gas transport station at Olbernhau (Germany) is equipped with two identical centrifugal compressors, operating single or parallel and running in a speed range of 5000-7200 rpm. The gas flow, which varies from 180.000 to 900.000 Nm3/h, is measured on the suction side, line pressure 4200-5000 kPa, via a flow metering station provided with orifice plates.
The flow is measured via 1 up to 5 parallel lines: four 16-inch and one 6-inch metering line dependent on the gas flow. The range of flows used for the 16-inch metering is between 100.000 and 300.000 Nm3/h. A simplified flow scheme of the compressor station is shown in fig. 4.1.

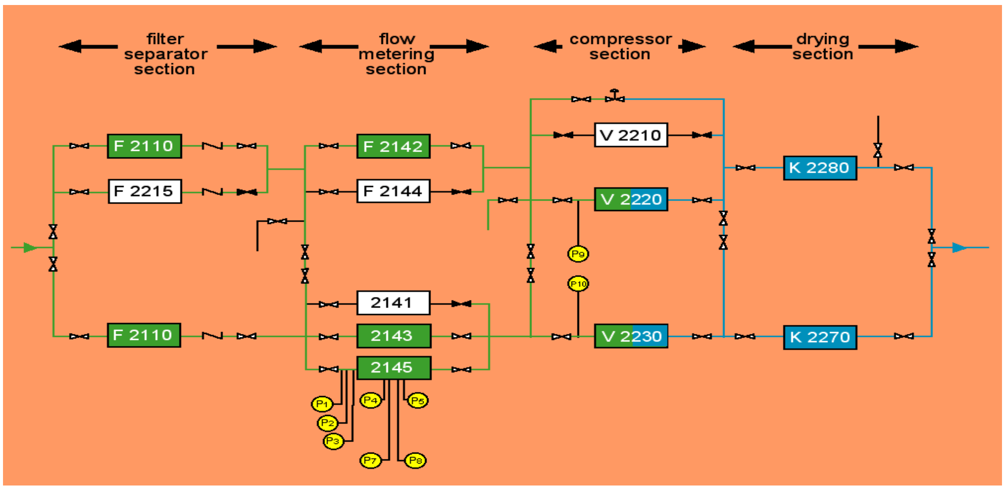

Fig. 4.1 Simplified Flow scheme of compressor and metering station Olbernhau

The purpose of the on-site measurements and analyses is to investigate whether pulsations caused by turbo compressors can have an impact on an orifice flow metering station located nearby. The results are used to establish if a flow metering station can be located at relatively short distance from the compressor(s) for the Yamal pipeline stations. The Olbernhau station is similar to the compressor stations to be located along this pipeline from the Yamal field to Germany and is therefore used for this investigation.
Transducers for pressure pulsation measurements are located on the suction inlet of the turbo compressors and on one out of four16-inch metering line at various locations:
1) P1 up to P6 directly on the 16 inch metering line
2) P7 and P8 close to the dP-transmitter at the end of the gauge lines (length 7 meter; diam. 9 mm)
3) P9 and P10 on suction inlet of the compressors
Some transducer have a small connection line (150-500 mm) whilst other transducers (P1 and P3) on the main lines are flush-mounted.

4.1 Pressure pulsations at the vane-passing frequency

The pressure pulsations measured near the compressor in-and outlet show a dominant component at the vane passing frequency, which corresponds to the running speed times the number of vanes (15) An example of this measurement is shown in fig.4.2 with a level of 50 kPa (0.5 bar peak) for location P9 at a vane passing frequency of 1530 Hz (compressor speed 6122 rpm). Pulsations at the running speed or higher harmonics thereof are not present in the spectra.

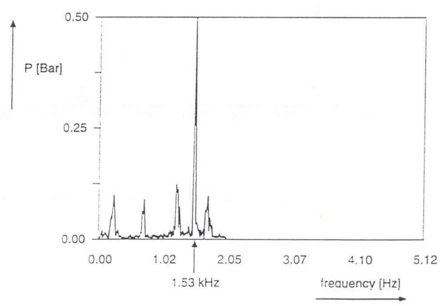

Fig.4.1.2 Pressure pulsation spectrum at compressor outlet – dominant frequency 1530 Hz

Other frequencies shown in the spectrum are 250,750,1250 and 1750 Hz, which correspond to standing waves in the 1 inch connecting line (1/4, 3/4, 5/4 and 7/4 wavelength) to the transducer, which has a length of 350 mm.
The spectra of pressure pulsations in the metering section from P1 up to P8 at the same condition show considerable reduction, over a factor 100, from the vane-passing frequency component to amplitudes < 1 kPa (or 10 mbar). The maximum level is 0.5 kPa ($5*10^{-3}$ bar) at location P2 at 1600 Hz; an overview of the spectra at P2 at different compressor speeds and corresponding vane passing frequencies from 1500-1750 Hz are show in fig. 4.1.2. The flow induced pulsation in the measuring pipe is independent of running speed and remains at 500 Hz. The amplitudes at the pressure transmitter of the orifice (P7 and P8) are even below 0.1 kPa.

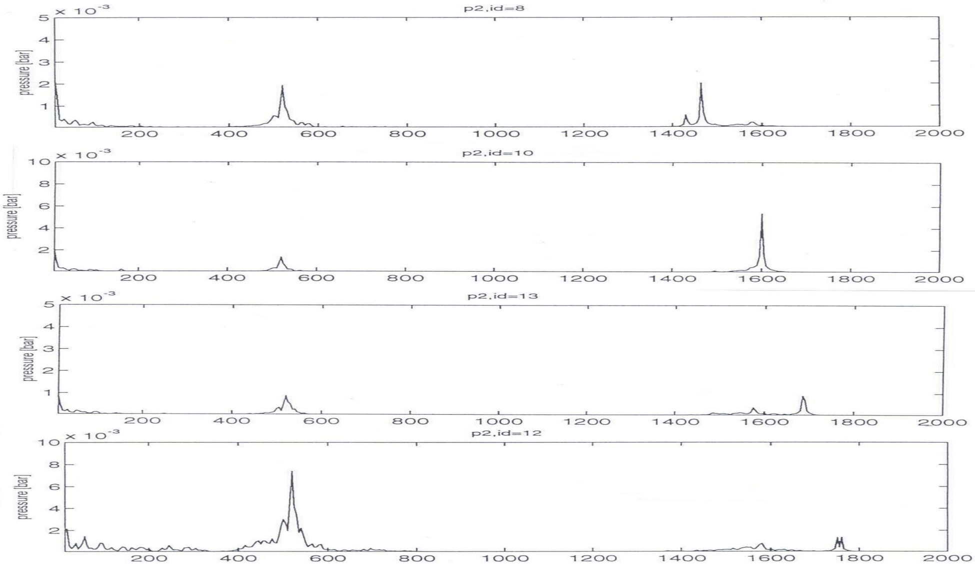

fig.4.1.2. Pressure pulsation spectra at P2 on 16-inch metering line at various compressor speeds

In VDI 3733 the damping for acoustic waves in an ideal gas is expressed per unit of straight pipe length:
Damping equals (0.15 f*T)/(D*P*293) dB per m, in which:

D: line diameter in m
f: frequency in Hz
T: temperature in degrees Kelvin
P: line pressure in Pascal

For a 24 inch line and a frequency of 1500 Hz the damping is approximately 0.004 dB/m. The damping across bends however is dominant as it varies between 0.5 and 0.3 dB, dependent on frequency and bend radius. The flow metering station is at a distance of 150 meter (pipe length) from the compressors and the number of bends, tees and reducers/expanders is 20. The actual damping for the vane passing frequencies measured is at least 40 dB, which is above the calculated level, based on the formula mentioned above showing 10 dB.
We conclude that pulsations caused by the centrifugal compressor are reduced effectively and do not interfere with the orifice flow measurement.

4.2. Low-frequency pulsations

The measurements on the 16-inch flow metering section and especially those near the pressure transmitter mainly show pulsations in the range below 200 Hz. It is likely that resonances in the gauge lines are excited due to vortex shedding. As a result of gauge line length (L=7 meter) standing waves occur at frequency f = c/4L or odd multiplies thereof, which is 14, 42, 70,…… Hz.
This corresponds to the pressure pulsations measured at P7 and P8 and thus also in pressure difference dP as shown in fig 4.2.1 and fig.4.2.2

Pressure difference
50 mbar

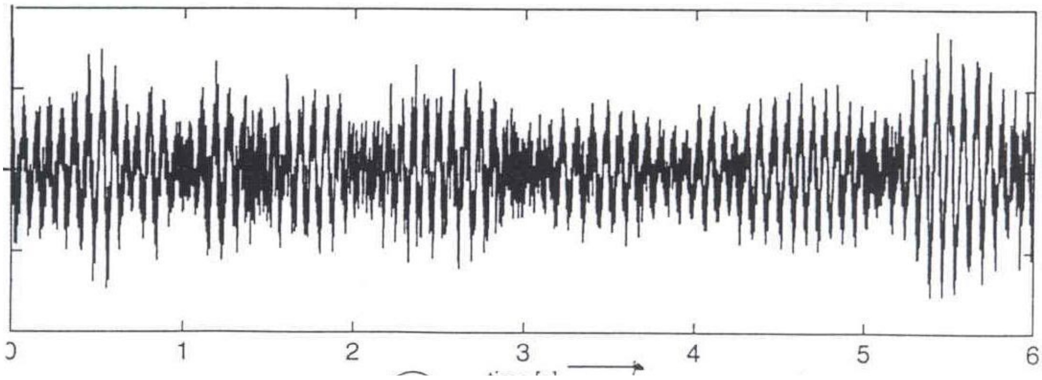

Fig. 4.2.1. Pressure difference P7-P8 as a function of time [s]

The pressure difference P7-P8 shows a considerable dynamic pressure of 33 mbar pp on a static dP of 25.4 mbar at a mean flow of 20.8 m3/s in the metering section.

An overview of the measurement results at various flow conditions is shown in the table in fig.4.3.3. Assuming a slow response dP sensor the time-mean differential pressure will be indicated. The corresponding flowrate than included the square root and temporal inertial effect errors.
The estimated quare-root-error E_T can be expressed as: $E_T = [1 + (U_{rms}/U_{mean})^2]^{0.5} - 1$ [see ISO/TR3313]
If we assume a slow-response sensor the estimated error due to pulsations is less than 0.1 % for almost all conditions, except for the lowest flow conditions 4 and 4a. The flow via metering line 5 at this condition is 75.000 Nm3/h or 20.8 Nm3/s and the maximum measuring error is +5% at condition 4a (hand-operated valve in 6-inch metering section closed and +2% at condition 4 (hand-operated valve in 6-inch metering section open).
During all measurements the metering lines 1,2 and 5 are open, which is not a normal operating condition for the lowest flow of 233.000 Nm3/h over the station. The metering range for a single 16-inch metering ranges from 300.000 to 100.000 Nm3/h, so the total flow can pass via a single 16-inch metering line.
So obviously only at (too) low flows pulsations in the gauge-lines can disturb flowmeter readings due to flow-induced resonances in the gauge lines.
The connecting line-length between primary element (orifice plate) and secondary instrumentation (dP-transmitter) should be restricted to prevent resonance if the length equals a quarter wavelength or odd multiples thereof.

The on-site measurements have shown that the high frequency pulsations caused by a turbo-compressor are damped effectively and will not have an impact on flow metering close to the compressors, assuming the distance is at least a 100 meter.
Considerable low-frequency pulsations did occur at the flow transmitter due to resonances in the gauge-lines, which is excited by vortex shedding. Unsteady vortex shedding can be a strong source of pulsations if the vortex shedding is coupled to an acoustic resonance in the system.
The PULSIM package, in which the pulsation source resulting from the source is incorporated, can be used to analyse a flowmetering station. The package has been applied to analyse flow-induced pulsations in a gas control station and give recommendations for the geometry to omit resonance conditions [14]
Guidelines for the design of flowmeter instrumentation are summarised in the ISO/TR 3313 and cover remarks with respect to both slow-response and fast-response dP-sensors for orifice metering.

5 FLOW-INDUCED PULSATIONS DUE TO VORTEX SHEDDING AND IMPACT ON A TURBINE FLOWMETER

The gas flow metering station concerned consists of two 16-inch streams A and B, each provided with a 16-inch turbine flow meter. TNO Institute of Applied Physics has been requested to perform

pulsation analyses to establish if flow pulsations are present and can have an impact on the accuracy of the flow meter. When the turbine meter is subjected to an unsteady flow the inertia of the rotor can cause the rotor speed to lag behind in an accelerating flow and to exceed it in a decelerating flow. As the impact of a decelerating flow exceeds that of an accelerating flow the error in reading of the flow meter subjected to pulsation will always be positive.
The total flow rate in the metering station varies between 5800 and 11000 m3/h at a line pressure between 1300 and 1600 kPa. Measurement locations are upstream and downstream of the turbine flowmeter and also on the reference pressure tapping as shown in the schematic lay out in fig.5.1. There are no compressors located on-site and there is no gas control or pressure reduction in the station. The only sources are flow-induced pulsations due to vortex shedding at T-joints or at other obstructions in the piping geometry.

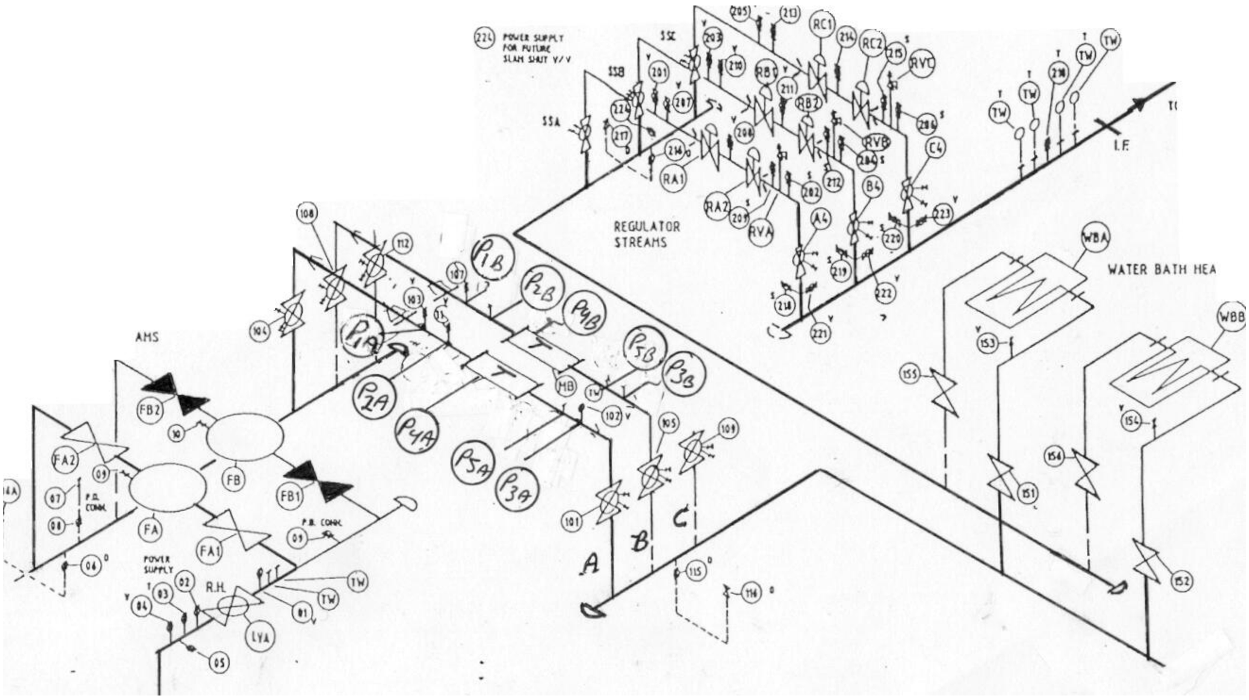

Fig.5.1 Overview of piping lay out in turbine metering station

The pressure pulsations measured on-site are ranging from 2-24 mbar pp, which is rather low: in the order of 0.01 to 0.15 % pp of the line pressure. Main frequencies occur in the range between 0-25 Hz with dominant components at 6, 12 and 16 Hz. An example of the spectrum of the measured pressure and calculated flow pulsation is shown in fig. 5.2.

4.2.3 Pressure pulsations P1, P4 and velocity at P4 for 7-B

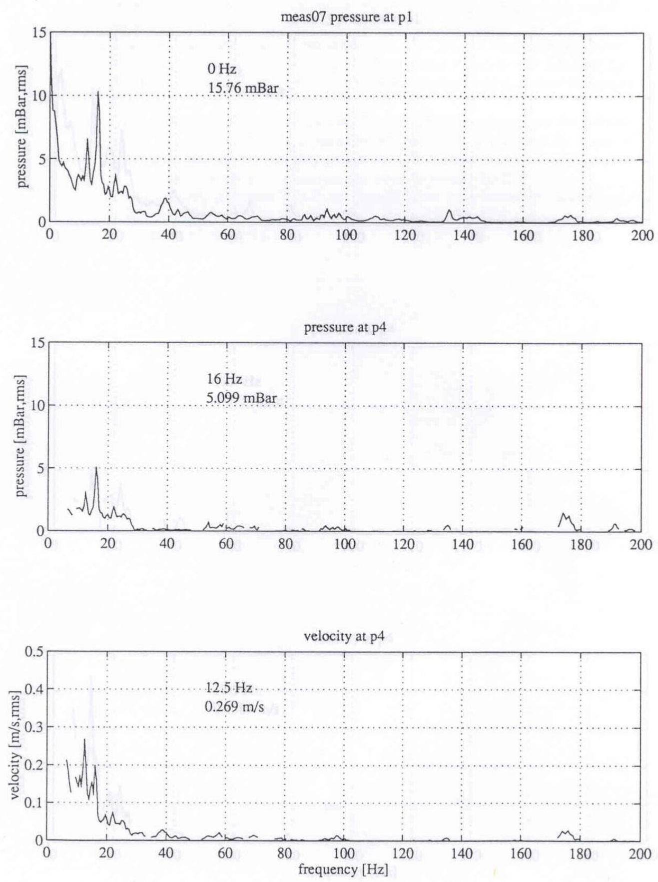

Fig.5.2 Measured pressure pulsations at P1 and P4 are used to calculate the flow pulsation or velocity at P4.

The recorded pressure pulsations are analysed to calculate the corresponding flow pulsations by the two-microphone method (2MM) as the amplitude of the flow-pulsation and frequency determines the error as explained in chapter 2. In the 2MM a coherence analysis is performed: only those frequencies for which the coherence > 0.95 are plotted in fig. 5.2. (So lines of the spectrum are missing if the coherence is less than 0.95)
An overview of some results of the measurements for the various test cases are shown in table 5.2, measurements have been performed for stream A, for stream B only and for stream A and B in parallel. The maximum relative flow pulsation of 0.269 m/s rms or 6.6 % occurs for case 7 with stream A and B in operation at 3330 m3/h each.

Meas.	Density	Flow velocity V_{mean}, m/s		Pressure pulsation, mbar		Flow pulsation, m/s rms		Flow pulsation of dominant frequency in %	
No.	Kg/m3	V_A	V_B	P1	P2	V2	V4	V2	V4
1	10.12	10.03	11.94	1.7	1.7	0.014	-	0.2	-
2	10.22	9.94	11.53	1.9	2.1	0.014	-	0.2	-
5	11.45	6.12	7.24	2.7	1.6	0.068	0.063	2.0	1.9
6	11.45	13.36	closed	3.3	7.1	0.184	0.167	2.5	2.3
7	11.17	6.29	7.36	23.9	9.4	0.250	0.269	6.1	6.6
8	11.17	closed	13.76	11,9	9.3	0.396	0.439	5.2	5.8

We assumed a sinusoidal pulsation, with the dominant frequency in the pulsations at 12.5 Hz and based our error analysis on the approach described in the latest edition of ISO/TR 3313. The

corresponding pulsation error for the turbine meter at 6.6% flow pulsation is + 0.2 %, whilst for all other measured operating conditions the error is lower.
As compressors are not involved vortex shedding along a T-joint of a closed side-branch is most likely the cause of these pulsations.
The piping geometry of the flow metering station is such that standing waves in the low frequency range can be excited by vortices. Side branches to closed valves in stream C (not in use) and also to A/B streams if not in use are potential locations for the occurrence of resonances. Also the closed valve to the filter stream II offers a potential location for vortex shedding.
If we consider the case in which the 2nd filter stream B and the metering section A is closed a configuration with 2 closed side branches is present. Vortex shedding frequencies Fv are determined by Fv = Str V/D in which

Fv : Vortex frequency in Hz
Str: Strouhal number, for this configuration maximum source strength is obtained if Str=0.25 [13]
V: Flow velocity in the main line, m/s

The range of vortex frequencies involved is 5-11 Hz or flow velocities Fv between 6 and 13 m/s, assuming Str=0.25 and D=0.3 meter.
Strong resonance can occur if a vortex frequency coincides with an acoustic resonance in the piping system, such as a standing wave between the closed valves in filter stream and metering section.
For the distances involved (approx. 3 m between valve and header) the lowest standing wave resonance frequencies are approx. 29 Hz for a single closed side-branch (1/4 wave length) and 39 Hz for 2 closed side branches (full wavelength 9 meter).
Dependent on the configuration of the T-joint or the combination of flow direction with the acoustic resonance the vortex shedding can result in a strong source of pulsation or even damping of unsteady flow as described by Peters [13]. Further the effect of edge rounding is significant in determining the source strength and the Strouhal number where the maximum source strength occurs.

It is recommended to omit long side branches with closed valves to prevent excitation of standing resonances by vortex shedding, which can have an strong impact on flow meters located nearby.
In order to be able to predict the behaviour of such a gas metering station under various operating conditions an analysis with the Pulsim package can be performed. Such an analysis includes:

- Building a simulation model of the gas piping involved from inlet header to outlet header, including volumes, control sections, side branches and assuming no reflection in the headers
- Calculation of the acoustic response of different piping configurations, thus enabling us to determine resonance frequencies and standing wave patterns in the piping
- Analysis of flow-induced pulsation sources at T-joints, bends or reducers
- Calculation of pressure and flow pulsation levels and frequencies in the pipe system caused by these FIP sources as a function of flow rate and valve positions
- Recommendations to modify and improve the piping design or to relocate flow meters in order to prevent flow induced pulsations, which could have an impact of the flow meters

6 CONCLUSIONS AND RECOMMENDATIONS

The experiences from field measurements that high frequency pulsations from turbomachinery are damped sufficiently and do not have an impact on flow meters located nearby. The reduction in pressure pulsations as observed in on-site measurements is approximately a factor 100 for the vane-passing frequencies involved.
In case reciprocating compressors are applied strong flow pulsations can occur even if pressure pulsations fulfil the API618 requirements for reciprocating machinery, which are based on structural and compressor integrity.
The systematic error due to low frequency pulsations is determined by the flow pulsation amplitude and independent of frequency when we consider dP devices and turbine flow meters. For vortex flow

meters the relation between pulsation and vortex frequency is much more important than the amplitude, due to lock-in effects. The amplitudes measured on-site are in excess of the criteria for flow pulsations at turbine meters defined in ISO/TR 3313. The impact of pulsations on flow meters close to reciprocating compressor should be evaluated in a pulsation analysis. The next edition of API618 will contain recommendations with respect to the pulsation effects on flow meters.
Low frequency pulsations due to vortex shedding in the main piping can be a strong source of pulsations, which effect the flowmeter accuracy especially in case vortex frequencies coincide with acoustic pipe resonances.
Evaluation of the lay out of a flow metering station in a simulation model is very effective in finding the optimum lay out from pulsation point or to define the operating conditions, which can be run safely.

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