Experimental and theoretical investigation of thermoacoustic oscillations in natural gas metering stations

Andreas Brümmer, Roland Edlerherr
Technische Universität Dortmund, Faculty of Mechanical Engineering, Chair of Fluidics
Leonhard-Euler-Straße 5, 44227 Dortmund, Germany
Phone: 0049-231-755-5720, FAX: 0049-231-755-5722, E-Mail: andreas.bruemmer@tu-dortmund.de

Johann Lenz
Kötter Consulting Engineers
Bonifatiusstraße 400, 48432 Rheine, Germany
Phone: 0049-5971-9710-47, FAX: 0049-5971-9710-43, E-Mail: lenz@koetter-consulting.com

Abstract: In natural gas flow metering and pressure regulation stations flowmeters, heat exchangers and control valves are usually connected in series. Especially in case of two and more measuring bars sometimes at minor flow rates untypical pipe vibration together with flow metering faults are observed. Based on field investigations the dependencies between the pipe vibration level, the gas pulsation inside the pipe and the operating conditions of the heat exchanger are analysed. It turns out that with increasing heat flow rates into the natural gas the pulsation and hence the metering faults as well as the pipe vibrations are amplified.

In order to understand the physical dependencies of this phenomenon besides experiment a theoretical study is performed. The root cause of the vibration turns out to be thermoacoustic instability. According to the Rayleigh-Criterion gas pulsations are amplified, if heat will be given to the gas in the moment of greatest condensation. Based on detailed theoretical investigations by means of a method of characteristics the physical dependencies are analysed. Finally potential solutions to avoid this vibration problem at natural gas metering stations are introduced.

Keywords: thermoacoustic oscillations, method of characteristics, pipe vibration, flow metering faults, heat exchanger

1. Introduction and problem description

Occasionally, pipe vibrations are detected in natural gas pressure regulation and metering stations, which operators can perceive both visually, as structural vibrations at pipes and assemblies, and acoustically. Such vibrations occur, in particular, in stations with at least two adjoining regulating lines with heat exchangers, with a small volume flow flowing through one line and with the other line being unused. In addition, the concurrently arising volume flow pulsation can influence the measuring behaviour of the upstream flowmeters. This problem shall be shown more precisely on the basis of a surveyed gas pressure regulation and metering station. Figure 1 aims to give an overview of the pipe routing with different measuring points for recording of the vibration and pressure pulsation situation of the studied natural gas pressure regulation and metering station. Starting from the high-pressure side, the gas flows first through the filter separator (FS) and then through the gas flowmeter with the vortex flowmeter (VFM) being connected in series with the turbine flowmeter (TFM). Afterwards, operating and stand-by lines with heat exchangers (HE), safety valves (SV) and the actual pressure regulators (PR) towards the low-pressure side were following. Both structural vibrations and, in certain operating ranges, significant deviations in the measuring behaviour of the vortex gas meter and the turbine flowmeter occurred in the station.

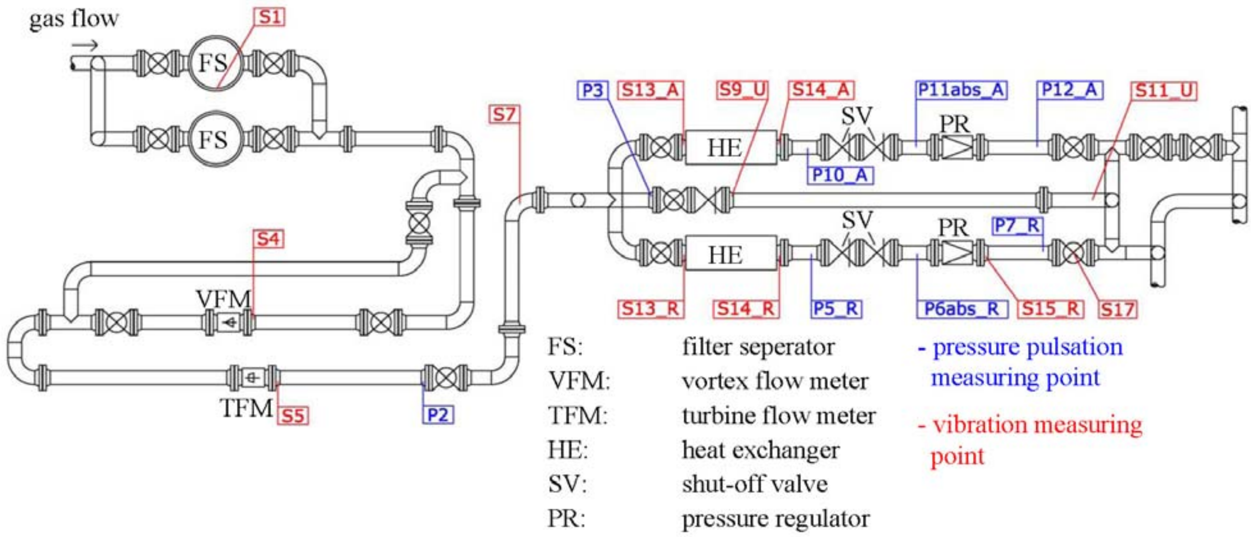

Fig. 1 Overview of the pipe routing with different measuring points of the surveyed natural gas pressure regulation and metering station

In both cases, a direct interrelation can be established between the operating conditions of the heat exchanger, as for example the water temperature, and the vibration of the pipe. The results of the investigations and the dependencies of the quantities are presented below.

1.2 Vibration measurements in the station

The area of the regulating lines housing the heat exchangers was identified as the vibration-exciting component of the station. Fig. 2 shows a plant detail with measuring points. Apart from the operating parameters, pressure pulsations (e.g. P11abs_A and P6abs_R) and structural pulsations (e.g. S9_U) were simultaneously measured at different measuring points. Moreover, parallel to this the gas temperature after the heat exchanger in the operating line as well as the return temperature of the heating water was recorded. The vibration phenomenon observed at the described station occurred exclusively at small flow rates. Upon an increase of the flow rate, the vibrations disappeared again. At a flow rate of 10.000 Nm^3/h[1], the highest vibration velocity of the pipe was determined with an effective vibration velocity of 9 mm/s. These vibrations are excited, in particular, when the water cycle of the heat exchanger is changed over from the closed (no water through the heat exchanger) to the fully open state (max. water flow rate of the heat exchanger). After a short break, pipe vibration sets in (see Fig. 2), with the pressure pulsations correlating with the pipe vibrations. The frequency of the pressure pulsations is about 18 Hz and in phase opposition at both measuring points. With a distance of the pressure regulators of 11 m and a speed of sound of 400 m/s in the natural gas, this frequency and the measured phase position correspond to the first acoustic natural frequency for pipes acoustically closed at both ends. The experiments show furthermore that, with rising water temperature and related higher heat input into the gas, the vibrations are amplified.

[1] Nm^3/h – standard m^3/h at 0°C (273 K), 1013 mbar

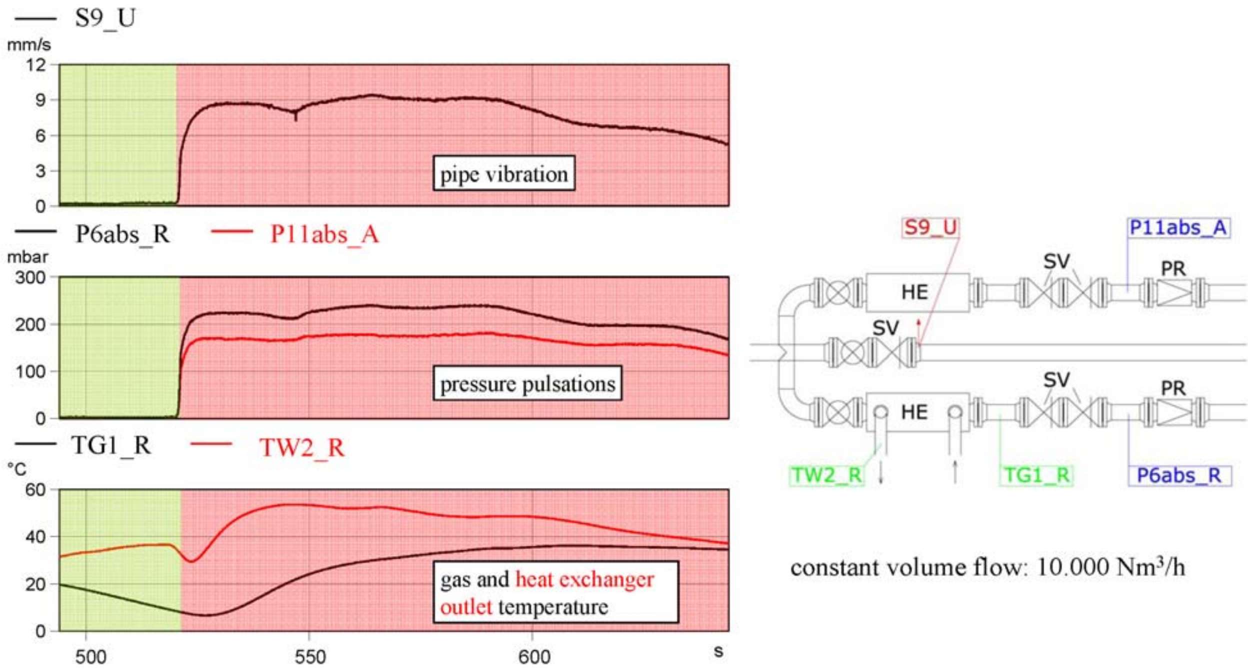

Fig. 2 Plant detail with measuring points in the range of heat exchangers and pressure regulators

1.2 Flow-metering errors

Through the gas column, the described vibrations in the area of heat exchangers and pressure regulators interact also with the measurement section of the station with turbine and vortex flowmeter. The gas vibrations result in metering errors and increased synchronism deviation in the flow-metering devices. Analogously to the measurements described above, the pressure pulsations in the area of the flow-metering devices were also measured.

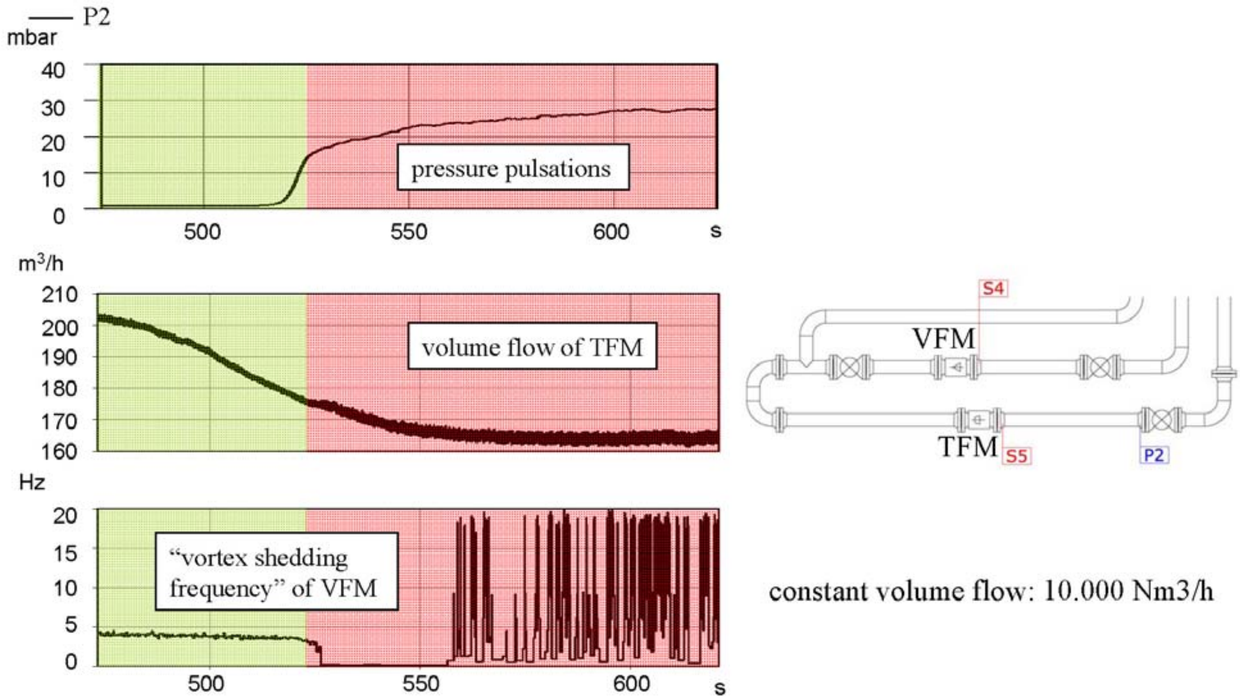

Fig. 3 Plant detail with measuring points in the range of measuring bar

In addition, the vortex shedding frequency at the vortex flowmeter and the HF-impulses at the turbine wheel flowmeter were recorded, which is shown in Fig. 3. The measured values confirm an influence of the volume flow measurement both on the turbine wheel flowmeter and the vortex flowmeter due to the occurring volume flow pulsations. When pulsation sets in, first a failure of the metering system at the vortex flowmeter is to be noticed. After the development of the vibration, the vortex shedding frequency is no longer proportional to the flow rate. It corresponds to half (9 Hz) of the pulsation frequency or to the single pulsation frequency (18 Hz) of the vibration observed. In case of the turbine wheel flowmeter, the spread of the measured values increases directly with the increase in pulsation after the start of the heat exchanger. The spectrum analysis of the HF-pulse signal provides a more detailed examination of the influence on the turbine wheel flowmeter. In steady flow without pulsation influence, the impeller of the turbine wheel flowmeter rotates at constant peripheral speed, which is to be seen in the range between 400 – 520 seconds in Fig. 4. In the pulsating flow with low to medium pulsation frequency, the peripheral speed of the impeller is pulsating, too. In the colour spectrum of the HF-pulse signal, this frequency modulation is visible through side bands next to the carrier frequency (from 520 s). The distance of the side bands in the frequency axis of the blade pass frequency corresponds to the pulsation frequency (18 Hz) of the standing wave in the area of the pressure regulation lines.

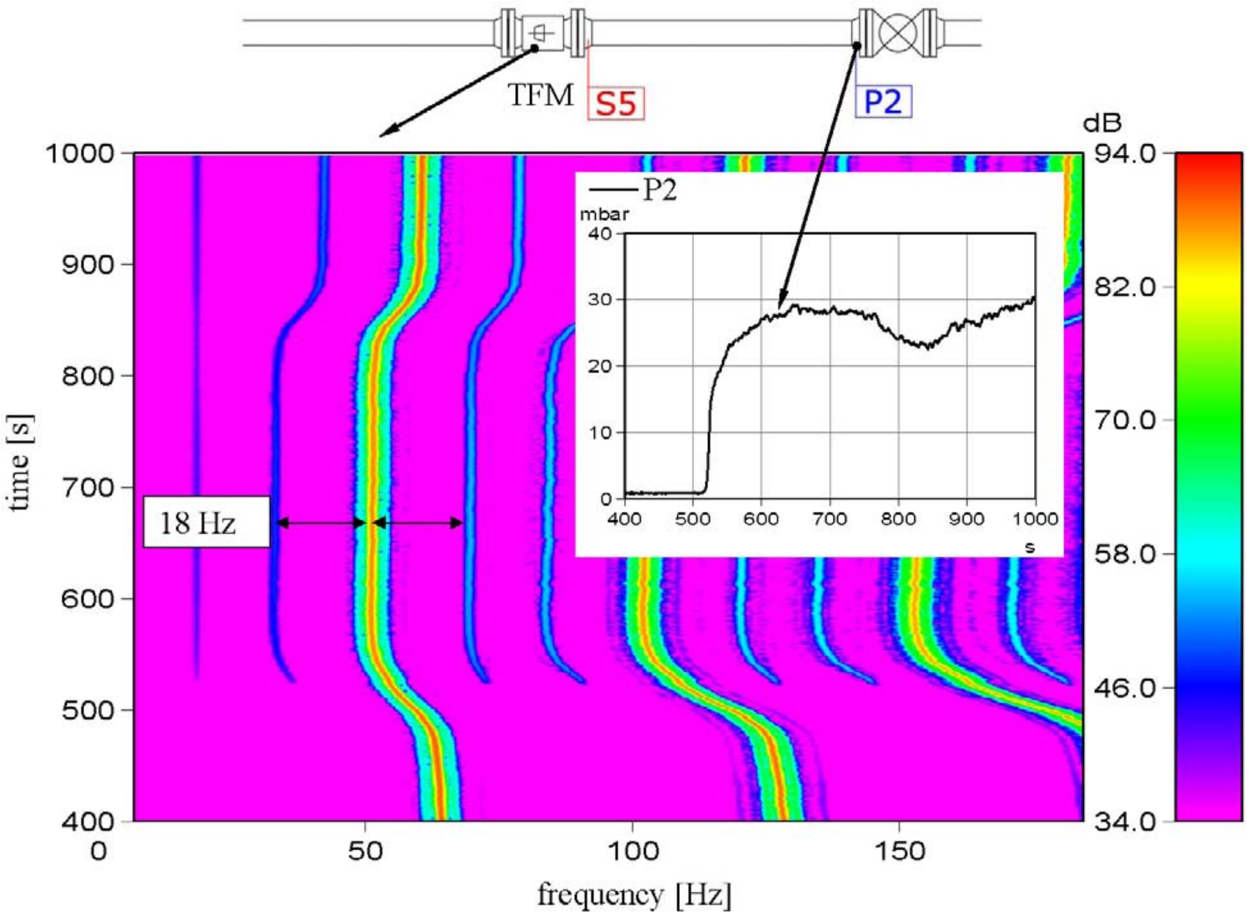

Fig. 4 Spectrum of turbine wheel HF-signal (measurement)

2. Analysis of vibration excitation

The root cause of the phenomenon observed in the station is thermoacoustic instability between the heat input in the natural gas and the gas-column vibration in the pipe. Due to the excitation, the described structural vibrations and flow metering errors occur. The following describes the physical mechanism of action of the vibration problem and illustrates its numerical simulation by means of the method of characteristics. Based on the numerical results, the dependencies arising in the development of thermoacoustic vibrations are discussed. The Rijke tube is used for this purpose as a simplified model for the station.

2.1 Rijke tube

The Rijke tube is a model experiment which allows the simulation of thermoacoustic vibrations. It consists of a horizontally arranged tube with two acoustically open ends where flow is applied by means of a blower. A heat wire which can be positioned at different points in the tube is used as heat source to generate a sound. The frequency of the generated sound is near the first acoustic natural frequency of the tube. Depending on the heat-source position in the tube, the intensity of the sound is amplified or the sound disappears [1].

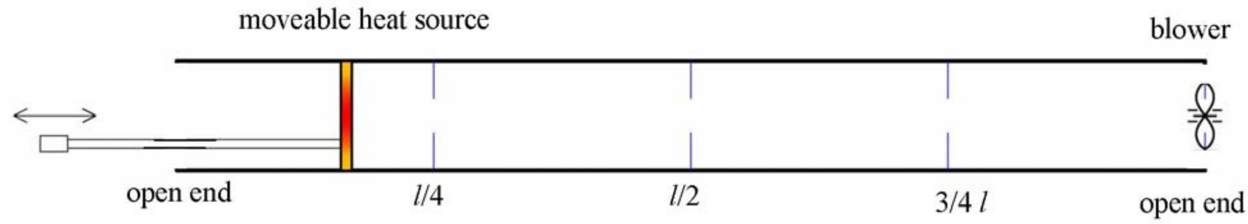

Fig. 5 Schematic of the Rijke tube

The criterion developed by Lord Rayleigh already in 1878 can be used as a precondition for the development of vibrations: "*If heat be given to the air at the moment of greatest condensation, or be taken from it at the moment of greatest rarefaction, the vibration is encouraged. On the other hand, if heat be given at the moment of greatest rarefaction, or abstracted at the moment of greatest condensation, the vibration is discouraged* [2].''

This interrelation can be mathematically described by the Rayleigh integral

$$I = \frac{1}{T}\int_{t}^{t+T} p'(t)\cdot q'(t)dt \qquad (1)$$

Here, p´(t) and q´(t) represent the time fluctuations of pressure and heat input, with their product being integrated by a time period T. Depending on the Rayleigh index I,

$I > 0$ results in an amplification

$I < 0$ results in a dampening

of a disturbance of average flow rates [3].

2.2 Numerical modelling of the Rijke tube using methods of characteristics

For a better understanding of the causes and mechanisms of action in the development of thermoacoustic vibrations, simulation calculations for the Rijke tube were carried out. For this purpose, the Rijke tube is modelled and simulated by means of a one-dimensional method of characteristics. For the calculation of the fluid-mechanical values, a system of equations comprising the mass-conservation equation, the conservation of momentum equation and the conservation of energy equation is applied. In order to obtain a solution, the hyperbolic partial differential equation system is transferred into a common differential equation system. The solution of the equations is then only possible along characteristic curves (eigenvalues of the matrices = characteristics) [4]. With a temporal and spatial discretization as well as the ensuing local linearization of the characteristics, the system of equations is numerically solved.

The equivalent model for the Rijke tube has a length of l = 1 m and a diameter of d = 0.06 m. Air is assumed as fluid. In this simulation, the heat exchanger is usually simulated at position *1*/4. A quasi-stationary heat transfer model is used for calculation based on heat transfer data according to Michejew and van Leyen [5]. The time dependent heat input into the gas caused by fluctuating flow velocity due to disturbances results from the dependence of the Nusselt number on the time and space dependent Reynolds number. The phase relation between a fluctuation of velocity and the heat input is predetermined for the one-dimensional simulation. Usually, the heat input lags the velocity fluctuation by Π/4.

2.3 Results of the simulation calculations

A systematic variation of potential influencing factors was carried out in the simulation calculations. Due to experiences gained from measurements in gas pressure regulation and metering stations, the applied flow velocity in the pipe, the heat transfer temperature and the position of the heat exchanger were varied at continuous heat input. In order to receive further knowledge on the development of thermoacoustic vibrations, the impulse response of the system to a heat-flow impulse and the influence of the phase between velocity and heat flow pulsation were also investigated. At first, an example will illustrate the vibration behaviour of the investigated model after the start of the heat exchanger.

2.3.1 Vibration after starting the heat exchanger

With the simulation of the Rijke tube in the way described above, it is possible to simulate the oscillation behaviour of a thermoacoustic vibration. A precondition for the development of such a vibration is the already existing fundamental oscillation that can be amplified. For the simulation, an oscillation was applied to the average velocity – independent of the function of the heat exchanger – which corresponds to human conversation at room volume in the amplitude $\hat{p}_0$ (52 dB).

As frequency, the first natural frequency of the tube of 171.6 Hz at 20°C (293 K) was chosen. The heat exchanger is set at position l/4 and put into operation at a certain turn-on instant and remains permanently activated.

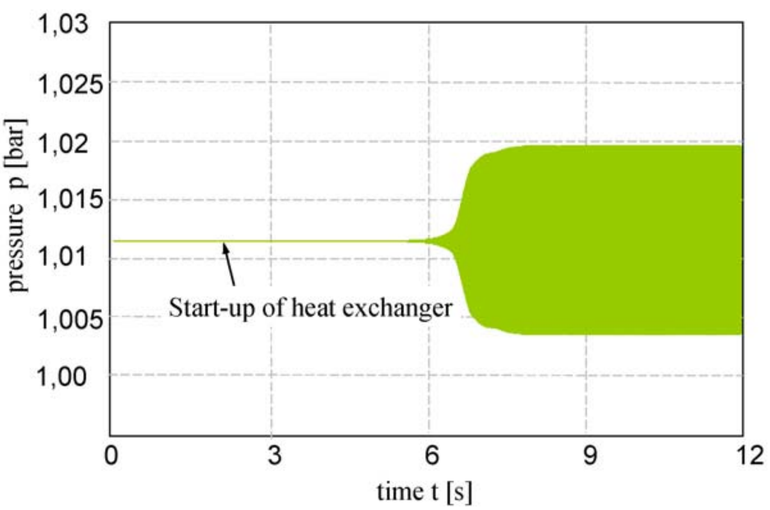

Fig. 6 Amplification of vibration after starting the heat exchanger (simulation)

A solid phase Π /4 is preset between the velocity fluctuation and the heat flow fluctuation so that the heat input correspondingly lags the velocity fluctuation. An initial exponential vibration growth is to be seen with a time lag to the turn-on instant. Afterwards, the vibration amplitude converges towards a fixed value where the pulsation excitation and the dissipation are at equilibrium. This behaviour was also observed in the investigations in gas pressure regulating and metering stations.

2.3.2 System response to heat pulse input

Based on the Rayleigh criterion, two extreme cases for heat input limited in time were simulated. In one case, the singular heat pulse is added at the moment of highest pressure, in the other case at the moment of lowest pressure. According to the Rayleigh criterion, maximal amplification and dampening are expected.

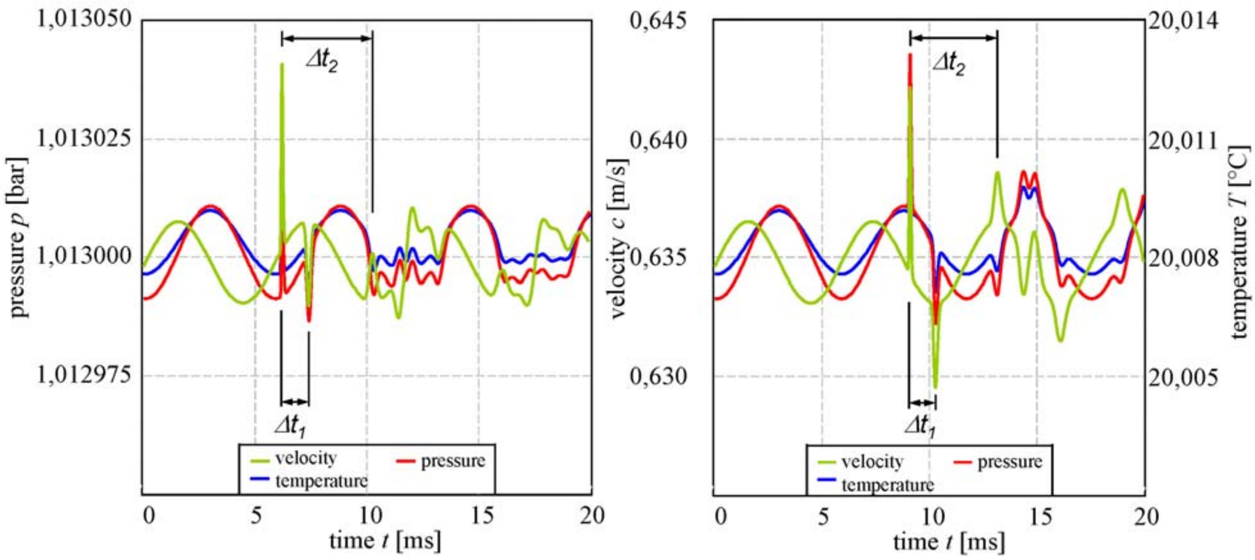

Fig. 7 System response to heat pulse input (simulation)

The simulation results in the right diagram show the heat pulse added at the moment of the highest pressure. Prior to that, the standing wave in the pipe can be seen, with a phase shift of PI/2 between pressure and velocity pulsation. The specific addition of a heat flow pulse initiates a disturbance which, from the point of heat input, spreads into both directions at speed of sound. After a time interval Δt_1, the reflected pulse which is locally nearer to the acoustically open pipe end is clearly visible. The pulse after Δt_2 is triggered by the pulse reflected by the other acoustically open end of the pipe. It is obvious that heat input at the moment of the highest pressure results in an amplification of the velocity amplitudes and thus in an amplification of the vibration. Pulse activation with the same absolute value can also be noticed when the heat flow pulse is activated at the moment of the lowest pressure. In this case the two reflected pulses result in a dampening of the fundamental vibration. Thus the thermoacoustic effect is due to the phase between heat flow fluctuation and pressure fluctuation as well as to the place of heat input. Moreover, a phase shift between velocity and pressure fluctuation of Π/2 supports the build-up of the oscillation, which is given in case of a standing wave in the system.

2.3.3 Influence of the phase shift between velocity and heat flow fluctuation

Following the examination of the pulse excitation, the influence of the phase between velocity and heat flow input is systematically varied from 0 to Π, with the heat flow fluctuation generally lagging the velocity fluctuation. A parabolic dependence between the phase and the resulting pulsation amplitude $\hat{p}$ (amplification) – related to the amplitude of the excitation $\hat{p}_0$ - is to be noticed, which shows a maximum at a phase difference close to Π/2 (see Fig. 8). In this case, the heat input is in phase with the pressure pulsation. In the range of 0 to Π/12 and 11/12 Π to Π, the influence of the damping parameter of the pipe prevails and vibration is damped with an amplitude ratio of less than one. The standardized illustration of the Rayleigh integral shows a curve comparable to the amplification so that it can be assumed that, with a higher value of the Rayleigh integral, the energy input into the acoustic wave is encouraged.

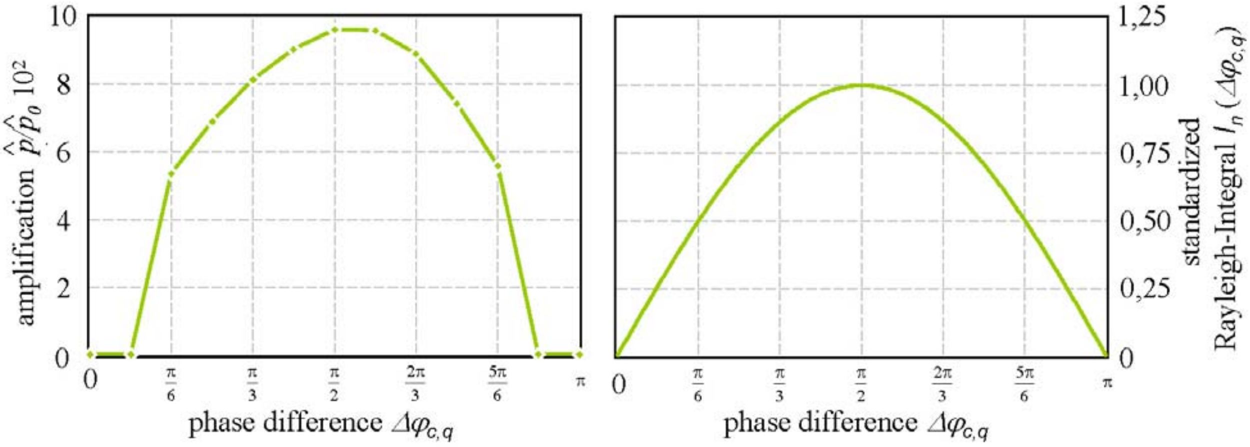

Fig. 8 Influence of phase shift between velocity and heat flow fluctuation

Thus, the simulation of the Rijke tube confirms the theoretical connection between the influence of the phase difference between velocity and heat flow pulsation on the amplification and the Rayleigh integral as measure for the energy input.

2.3.4 Influence of the average velocity

The heat transfer from the heat source to the air depends on the Nusselt number. This, in turn, depends on the Reynolds number calculated at the heat source in dependence on time. In case of a fluctuating flow velocity, also fluctuating Nusselt numbers and thus fluctuating heat flows are to be expected. Fig. 9 shows the dependence of the vibration amplification on the time-averaged Reynolds number. This is directly proportional to the applied mean velocity in the Rijke tube.

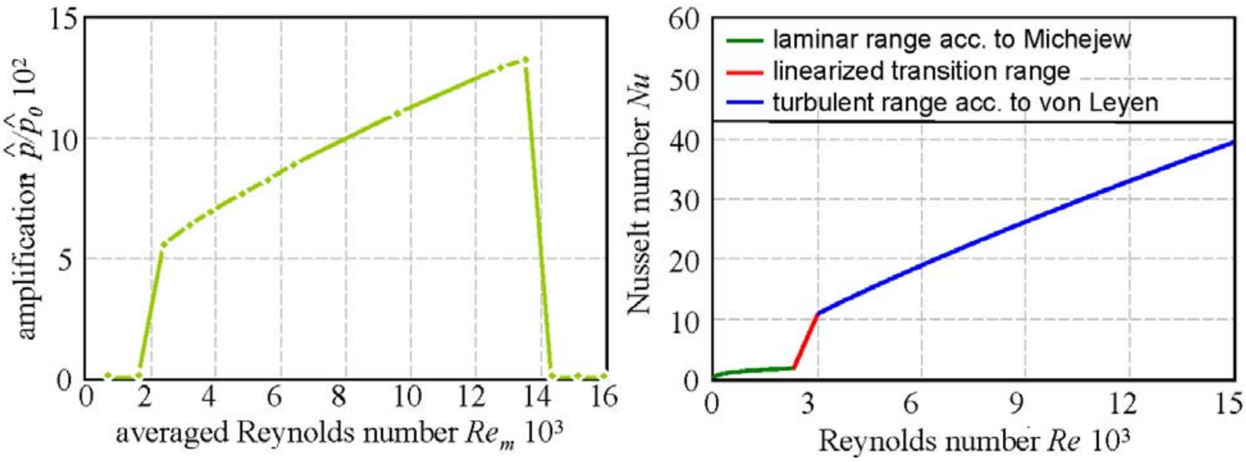

Fig. 9 Influence of average velocity on amplification of vibration (simulation)

For laminar flows in the range of up to a Reynolds number of about 2400, dampening prevails in the tube. At the transition from laminar to turbulent flow, the amplification curve rises steeply. From a time-averaged Reynolds number of 13600, the amplification curve falls abruptly. The rise of the amplification at the transition from laminar to turbulent flow is explained by the large gradient of the Nusselt number due to the heat transfer model. With slight changes of the current Reynolds number, the heat transfer to the gas changes considerably and results in high heat flow pulsation. In the subsequent range of further increase, the absolute values of the energy amount transferred to the gas increase, because the Nusselt numbers continue to rise in dependence on the averaged Reynolds number. From a time-averaged Reynolds number of about 13600, the friction loss increasing by the square of the averaged flow velocity prevents the excitation of the gas column oscillation in the simulation model of the Rijke tube. Consequently, above an averaged Reynolds number, the damping which increases with the average flow velocity abruptly prevents the excitation and thus the development of the single tone. In this case, the value of the Reynolds number depends on the existing attenuators (e.g. wall friction, inflow and outflow losses or changes in cross section due to fittings). This behaviour was also observed in the examinations on the gas pressure regulation and metering station, which suddenly, above a certain volume flow, did not show any vibrations.

2.3.5 Influence of heat exchanger temperature

According to Fig. 10, the influence of the heat exchanger temperature results in a parabolic increase of the vibration amplification with increasing temperature. Below a temperature of 275° C, a slight damping of the vibration occurs. The decrease is caused by the pressure losses which are due to the friction effects in the pipe. Below a certain temperature of the heat exchanger and/or heat flow input, the acoustic energy transferred into the gas is lower than the losses. Calculations show that, assuming lower loss coefficients, the amplification of the wave occurs even at lower heat exchanger temperatures.

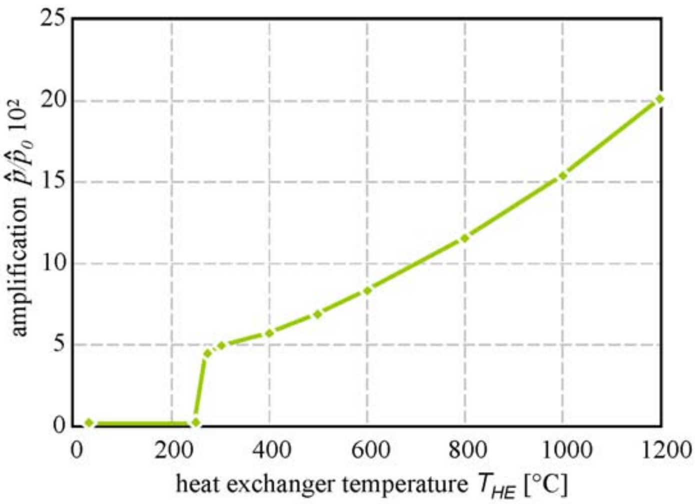

Fig. 10 Influence of heat exchanger temperature on amplification of vibration (simulation)

2.3.6 Position of the heat exchanger

The position of the heat exchanger has a decisive influence on the amplification or damping in the Rijke tube. Fig. 11 demonstrates that amplification occurs in the left part of the Rijke tube at an average flow from left to right. In the right part of the tube, damping takes place. In the simulated case, amplifications up to a factor of 960 occur depending on the position of the heat exchanger. Compared to the Rayleigh integral, the maximum of the amplification is slightly shifted to the centre of the tube. The reason for this is a shift of the acoustic natural frequency of the pipe due to the heat input and the related local increase in the speed of sound.

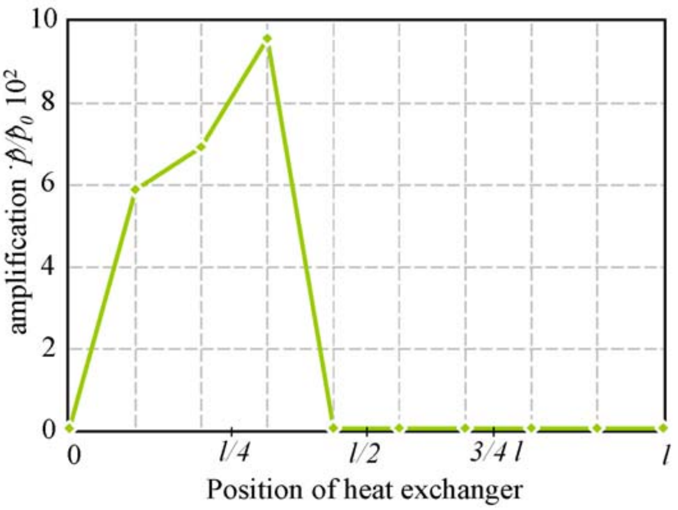

Fig. 11 Influence of the position of heat exchanger on amplification of vibration (simulation)

3. Problem-solving approaches to prevent thermoacoustically induced vibrations in gas pressure regulation and metering stations

Considering the presented measuring results, the results of the simulation of the Rijke tube can also be transferred to thermoacoustically induced vibrations problems in gas pressure regulation and metering stations. Moreover, it is suitable to transfer the calculation method to the simulation of entire gas pressure regulation and metering stations. In particular, when new stations with changing operating conditions are designed vibrations and vibration-induced disturbances of flow meters in the station can be precalculated and avoided in the further planning process. Based on the simulation, the amount of energy applied, the position of the heat exchanger, and the damping effects turned out to be decisive parameters for the avoidance of thermoacoustically induced vibrations.

3.1 Energy input

In particular in stations where the heat exchanger temperature is not lowered and/or adjusted at low volume flows, the development of thermoacoustic vibrations is encouraged. In stations where control is not provided, a possible switch-off of the heat exchanger should be considered at low flows.

3.2 Position of the heat exchanger

For a given station, the quantitative course of the Rayleigh integral along the pipe can be calculated in dependence on the geometrical dimensions and the first natural vibration form. Fig. 12 shows the range of the control path of the station presented in the first section in a simplified horizontal arrangement as well as the Rayleigh integral calculated for the station. It turns out that the heat exchanger in the operating line is in the range of the maximum of the Rayleigh integral and thus encourages the development of the thermoacoustically induced vibration.

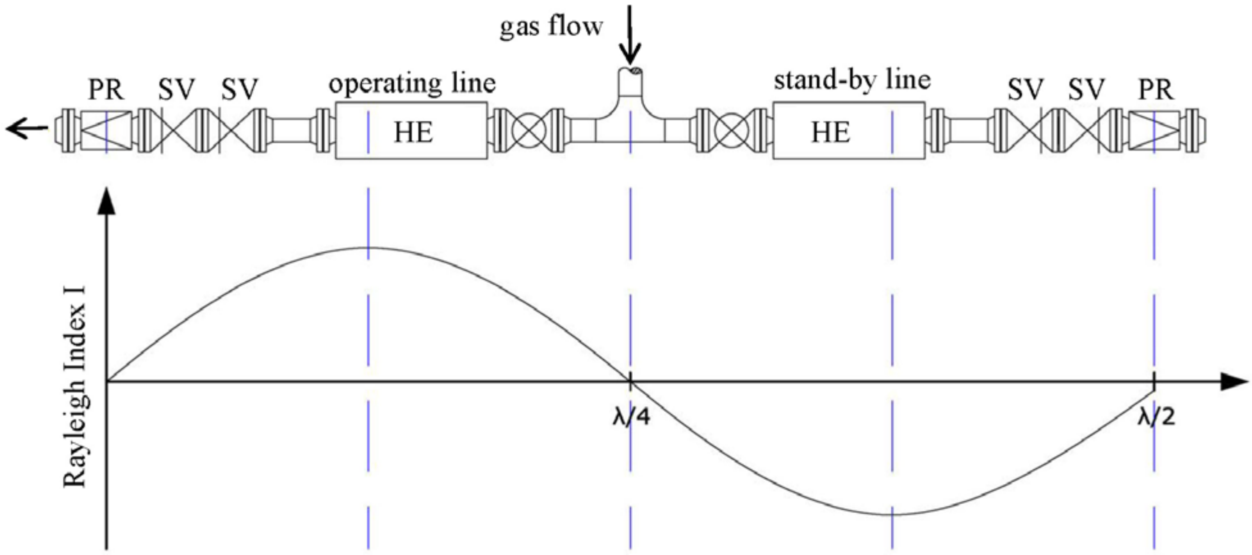

Fig. 12 Rayleigh Index calculated for the studied natural gas pressure regulation and metering station (right side closed)

This correlation can be taken into account when a new station is planned and vibration excitation can be avoided e.g. by positioning the heat exchanger more closely to the T-piece or into the feed line. Moreover, regarding existing stations there is the possibility to equip a potential fitting at the inlet of the stand-by line with a motor drive or to close it, in case the line is not used. The consequence is that the acoustic length of the pipe is reduced, which results in a shift of the maximum of the Rayleigh integral towards the pressure regulating valve. Thus, the amplification of the thermoacoustic vibrations is reduced and/or the excitation is completely avoided.

3.3 Damping

The simulation results have shown that from a certain damping in the system (dissipation), excitation of an existing disturbance does no longer occur. Consequently, thermoacoustically induced vibration can be avoided or reduced by increased damping in the frequency range of the first natural vibration of the pipe. Here it is suitable to install apertures or specially designed pulsation damping plates [6] in the area of high velocity fluctuations near the T-piece or at the inlet of the heat exchangers. Alternatively, also vertically arranged tube bundle heat exchangers with deflections can be provided, which generally cause a higher pressure loss compared to horizontally arranged heat exchangers and thus contribute to the desired minimum damping. Another possibility to avoid thermoacoustically induced vibration is avoiding operation of the station below certain volume flows (Reynolds numbers). The minimal permissible volume flows (Reynolds numbers) however depend on the given damping inside the respective pipe section and cannot be universally named.

4. Summary

A vibration problem and the impacts on the measuring behaviour of the flow meters contained is presented using a surveyed gas pressure regulation and metering station as an example. The root cause turns out to be a thermoacoustically induced vibration in the area of two adjacent regulating lines with heat exchangers.

The Rijke tube is a model experiment which enables the simulation of thermoacoustic vibrations. A numerical simulation of the Rijke tube by means of the method of characteristics was carried out and the decisive influencing factors as well as the physical mechanism of action for the development of thermoacoustic vibration were identified. The Rayleigh integral as a criterion for the excitation of thermoacoustic oscillations is confirmed.

Finally, approaches to the avoidance and/or reduction of thermoacoustically induced vibrations are given which can be transferred from the simulation results to gas pressure regulation and metering stations. The decisive influencing factors are the position of the heat exchanger, the amount of energy applied and the damping inside the gas pressure regulation and metering station.

For future investigations, it is planned to avoid the development of thermoacoustic vibrations e.g. by influencing the phase relation between velocity and heat flow fluctuations. Moreover, the thermoacoustic effect could be systematically used to dampen existing pulsations, e.g. in the field of positive displacement machines by applying electrically regulated heat flow. For this purpose, apart from theoretical considerations also experimental investigations are planned.

Acknowledgment

We would like to thank RWE Rheinland Westfalen Netz AG, Essen, FRG for the financial support granted to carry out the simulation with the aim of understanding the physical mechanisms of action of thermoacoustic vibrations and to show possible measures for the avoidance of such vibrations.

Abbreviations and formula symbols

c	velocity of gas
d	diameter of Rijke-tube
FS	filter separator
HE	heat exchanger
I	Rayleigh Index
K	Kelvin
l	length of Rijke-tube
Nu	Nusselt number
p	pressure
$\hat{p}_0$	pressure amplitude of excitation
$\hat{p}$	pressure pulsation amplitude after amplification
P	pressure pulsation measurement point
PR	pressure regulator
q	heat input
Re	Reynolds number
Re_m	averaged Reynolds number
S	vibration measurement point
SV	safety valves / shut-off valve
t	time
T	time period, temperature
TFM	turbine flow meter
TG	gas temperature measurement point
TW	water temperature measurement point
VFM	vortex flow meter
φ	phase

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