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PULSIM, a powerful tool for the control of vibrations in pipe systems

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1. Introduction

Pulsations or pressure waves are flow dynamic phenomena, which can cause various kinds of problems in installations in the process industry, the power industry and the oil and gas industry. For example pulsations can cause:

- Vibration and fatigue problems in piping and fluid machinery (reciprocating compressors and pumps, screw compressors, rootsblowers and centrifugal compressors and pumps).
- A decrease of the efficiency of running equipment (increase of power consumption or reduction of capacity).
- Damage to fluid machinery (e.g. compressor or pump valves or impellers).
- Cavitation problems.
- Errors in flow meter readings and process control systems.
- Increased noise levels.

To avoid these problems a simulation program, called PULSIM (PULsation SIMulator) has been developed within the TNO-TPD. Simulation of flow and piping dynamics in the design stage of a system can reduce the number of problems during operation.

2. History Pulsim

In 1965 the so-called PULSIM project has been started. An electrical analogue simulator (PULsation SIMulator) has been built within the department of Instrumentation of the TPD. Several Industrial partners: Shell, Unilever, AKZO, Thomassen and Gasunie financially contributed to the project.
With this analogue computer rather simple pipe systems of reciprocating compressors have been investigated during approximately 15 years.

Since 1980 a digital simulation program has been developed within the department of Flow and Structural Dynamics of the TNO-TPD (PULSIM). With this flexible program all kinds of dynamic flow phenomena can be investigated in very complex and long pipe systems. New modules for the modeling of fluid machinery with their pipe system have been added to PULSIM over the years, i.e. detailed compressor valve dynamics, liquid pumps, centrifugal compressors, burners, flow induced pulsations and start-up and blow-down conditions.

Moreover, the PULSIM program has been coupled to the finite element program ANSYS for the mechanical response analysis. In this way the calculated pulsation induced forces (amplitude and phase) are used directly for the calculation of vibrations and cyclic stress levels of the piping. The results of the analyses are continuously being checked in practice for instance during trouble shooting of existing systems.

Nowadays the Flow and Structural Dynamics Department carries out approximately 30 project per year for clients all over the world [7]. These clients are mainly manufacturers of compressors and pumps, engineering companies, oil and gas industries, chemical and petrochemical industries. The analyses are carried out according to the API Standard 618, fourth edition of June 1995, design approaches 1, 2 or 3 for reciprocating compressors [1] and according to the API Standard 674 of June 1995, design approaches 1 and 2 [8] for plunger pumps. For a full design 3 approach analysis two steps are taken:

1. The acoustical analysis with PULSIM
2. The mechanical analysis with ANSYS

The results of the analyses, preferably performed during the design stage of a system, are recommendations to reduce pulsations, vibrations and cyclic stress levels in the system to acceptable levels.

The PULSIM program, combined with ANSYS, has been proven to be a powerful tool for the design of safe systems and optimal processes [2]. Moreover, the PULSIM team acts as a partner during the complete course of the analysis and even when the system has been put into operation.

3. A short description of the PULSIM program

3.1 Basic equations

The following equations describe the flow through a pipe, assuming that the flow velocity is constant in the pipe cross section:

- The continuity equation. Eq(1)
- The momentum equation. Eq(2)
- The effect of gravity and friction is taken into account by the source term. Eq(3)
- The energy equation. Eq(4)

To complete the set of equations, a relation between pressure, density & temperature, the ideal gas law is used Eq(5).

The basic equations for the flow in pipe systems can be simplified considerably, if the following conditions are fulfilled:

- The fluctuations of pressure and density are small, compared to their mean values.
- The (total) flow velocity is small compared to the wave propagation speed (velocity of sound).
- There is no heat exchange with the pipe wall.
- The fluidum can be described as an ideal gas.
- The acoustic wavelength is much larger than the pipe diameter.

The simplified equations are called the acoustic approximation. For this approximation 2 equations remain:

- The mass balance. Eq(6)
- The momentum balance. Eq(7)

3.2 The analog simulation method

For the solution of the acoustic approximation of he flow equations, the space variable x has to be made discrete, This is equivalent to dividing a long pipe into small sections. In this way the two partial differential equations are converted into two ordinary first order differential Eq(8) and Eq(9). In these equations W represents the pressure loss due friction and proportional to $u|u|$. For the analogue solution this term is linearized, so that friction can be modelled by a linear resistance.

Equation Eq(8) can be considered as a mass balance for the pipe element, for which the electric analog is a capacitor. Equation Eq(9) represents the impulse balance for a pipe element, which can be represented electrically by an inductor. In this way a sequence of pipe elements can be represented by an inductor – capacitor network.

3.3 The digital simulation method

A method suitable for the solution of the one dimensional (hyperbolic) flow equations on a digital computer, is the method of characteristics. This method, which is very popular in engineering, leads to minimal computation time and thus is very efficient. In this paper the method of characteristics will not be given in detail. Instead, a qualitative description will be given of the numerical solver chosen by TNO-TPD. In the method of characteristics both time and space variable x are made discrete. The time step and the length of a pipe element are coupled by the equation Eq(10). This relation represents the so-called characteristic directions in the x-t plane in case of the acoustic approximation. A fixed grid of characteristics can be drawn in the x-t plane. In case the flow velocity cannot be neglected with respect to the wave propagation speed, the characteristic directions depend on the local flow velocity and the grid of characteristics is not fixed. It is obvious that in that case the solution will be far more complex. By introducing so called characteristic variables Φ Eq(11) and Ψ Eq(12), the differential equations are simplified to ordinary equations Eq(13), Eq(14) and Eq(15), which are coupled by the friction term W. Starting with a known initial condition, the values of the characteristic variables for the next time step can be calculated by solving equations Eq(13) and Eq(14) for each point on the x-axis. Apart from the calculation of the friction term W, the solution is reduced to shifting the characteristic variables along the characteristic directions, Φ along the positive and Ψ along the negative characteristic direction. Besides, the value of Ψ at the right-hand boundary and the value of Φ at the left-hand boundary have to be known for the next time step. These values are called boundary values.

Apart from the method of solving the partial differential equations the so called boundary conditions, which are terminating elements at both ends of a pipe section are equally important. The boundary conditions describe for instance components like a compressor cylinder, a Tee connection, a reducer, a volume, a closed valve (zero flow pulsations), a very large volume (zero pressure pulsations) etc. In the PULSIM program, a flexible data structure is available with which a flow simulation model of any pipe system can be defined. A complex pipe system consists of various interconnected pipe sections. The model data structure consists of tables with pipe sections, start and end nodes of each pipe section and the boundary condition at each node. The model size is basically unlimited. A simple example is given in chapter 4.2.1.

Basic equations:

Equation 1:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial x} = 0$$

Equation 2:

$$\rho \frac{\partial u}{\partial t} + \rho u \frac{\partial u}{\partial x} = \rho f - \frac{\partial p}{\partial x}$$

Equation 3:

$$f = g + \frac{2fa}{D} u|u|$$

Equation 4:

$$\frac{\partial h}{\partial t} + u \frac{\partial h}{\partial x} = \frac{\partial p}{\partial t} + \frac{u \partial p}{\partial t} - q$$

Equation 5:

$$p = \rho \cdot R \cdot T$$

Equation 6:

$$\frac{\partial p}{\partial t} + c^2 \frac{\partial \rho u}{\partial x} = 0$$

Equation 7:

$$\frac{\partial \rho u}{\partial t} + \frac{\partial p}{\partial x} - \frac{2\rho fa}{D} u|u| - \rho g = 0$$

Equation 8:

$$\frac{dp_i}{dt} = -c^2 \frac{(\rho u)_i - (\rho u)_{i-1}}{\Delta x}$$

Equation 9:

$$\frac{d(\rho u)_i}{dt} = \frac{p_i - p_{i+1}}{\Delta x} - W_i$$

Equation 10:

$$c = \pm \frac{\Delta x}{\Delta t}$$

Equation 11:

$$\Phi = p + \rho \cdot cu$$

Equation 12:

$$\Psi = p - \rho \cdot cu$$

Equation 13:

$$\Phi_i(t + \Delta t) = \Phi_{i-1}(t) - W_{i-1}$$

Equation 14:

$$\Psi_i(t + \Delta t) = \Psi_{i+1}(t) - W_{i+1}$$

Equation 15:

$$W_i = W(\Phi_i, \Psi_i)$$

Notation:

c	Wave propagation speed
D	Pipe diameter
f	Volume force
fa	Fanning friction factor
h	Specific enthalpy
p	Pressure
q	Heat flow
t	Time
T	Temperature
u	Flow velocity
x	Distance along pipe axes
Φ Ψ	Characteristic variables
ρ	Density
κ	Isentropic constant
W	Friction Term

3.4 Presentation of results.

To understand the acoustical behaviour of the system and to present the simulation results the PULSIM program offers various options:

1. The frequency response curves of the system.
2. Plots of the calculated peak to peak level of pressure, flow and force at a selected node or pipe section as a function of the velocity of sound or compressor speed.
3. Plots of the peak to peak pulsation level of a selected compressor harmonic at a certain condition as a function of the location in the pipe system.
4. Plots of the time function of pressure, flow or force during the revolution time of the compressor.
5. Tables with the calculated pulsation level at selected points for various operating conditions.

Examples of the plots and tables are given in chapter 4.

4. Steps in a pulsation analysis

4.1 Collecting data for the analysis ("Kick off" meeting)

4.1.1 Scope of the analysis.

The scope of the analysis has to be defined very well. In case more than one pulsation source is present in the piping, it has to be decided for which source or combination of sources the pulsations have to be calculated. To find the maximum possible worst-case pulsations and pulsation-induced forces, all possible combinations have to be analyzed. For each source the operating conditions and the type of capacity control play an important role for the resulting pulsations. For instance for a reciprocating compressor the following parameters are very important:

- Compressor speed (fixed or variable).
- Number of stages and cylinders per stage.
- Single or parallel running compressors.
- Type of capacity control (suction valve unloaders, reversed flow control, clearance pockets)
- Pressure ratio (fixed or variable pressures in the system).
- Gas composition. Fixed or variable.

In case many of the above mentioned parameters vary significantly in practice, a selection of the cases to be investigated will be made to limit the calculations. To guarantee a safe operation for all possible operating conditions much experience is required to make an optimal selection.

4.1.2 In search for good boundary conditions.

As described in 3.3 the boundary conditions play a very important role in the accuracy of the simulation results. Good boundary conditions to start or stop the simulation are:

- Large volumes. A volume can be considered to be large in case the volume is 5 to 10 times a well designed pulsation damper in the system to be investigated. For instance the suction K.O. drum should be 5 to 10 times the suction pulsation damper.
- A closed valve. For instance the normally closed relief or bypass valves. Also valves with sufficient pressure drop (more than 5% of the static pressure in the system) can be considered as being acoustically closed.
- A large diameter change in the piping. For instance in case the compressor piping is connected to a header with considerable larger diameter. A diameter ratio of 4 can be considered to be sufficient. However, the length of the piping with large diameter should be sufficiently long.
- A very (infinitely) long pipe system. In an infinitely long pipe system the pulsations, generated by the compressor will not be reflected and therefor reflections will not increase the pulsations near the compressor. A pipe system can be considered to be infinitely long in case the length is more than 10 times the wavelength of the lowest compressor harmonic. For instance for an air (velocity of sound 340 m/s) compressor with a compressor speed of 300-rpm (lowest harmonic 5 Hz) the wavelength is 68m. So the length should be more than approximately 680 meters before the length can be considered as infinitely long.

During the "Kick off" meeting Process Flow and Instrumentation Diagrams are very helpful to define the points to start and stop the simulation. Also detailed data of process equipment like heat exchangers or separators need to be collected. Moreover, good insight in the process conditions (pressures, temperatures, and densities) and pressure losses of equipment and valves should be gathered.

4.2 The check of the pulsation dampers

4.2.1 The original pulsation damper

Instead of the formula given in the API Standard 618, paragraph 3.9.2.2.2, for the design of pulsation dampers, PULSIM applies a different approach.
For the design of the damper, the cylinder internals with a simple compressor valve model and a suction and discharge damper are modelled. The suction and discharge piping is replaced by infinitely long lines. A sketch of such a model is given in figure 1. The structure of a part of the PULSIM model is shown in the pipe table of figure 2.

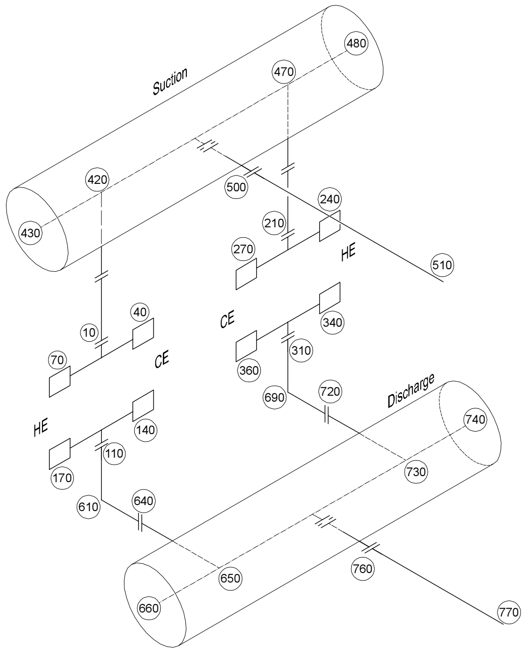


Figure 1.

In this model the PV diagram and the flow pulses are calculated based on the actual suction and discharge pressure and depend on the pulsations in the cylinder passage.

The calculated time function of the discharge flow pulse, generated by one cylinder side, is shown in figure 3. Moreover the frequency analysis of the pulse is given from the 1st to the 15th harmonic. It is obvious that the interaction between cylinder, cylinder passage and damper effects the shape of the flow pulses. The PV diagram shows the effect of the pulsations on the power consumption and capacity. In figure 4 the calculated PV diagram is given at a certain velocity of sound.

The generated flow pulses cause pressure pulsations in the system. In figures 5 and 6 the plots of the time function of the calculated pulsations near the discharge compressor valves and at the outlet of the discharge damper are given. In figure 7 the plot of the time function of the calculated pulsation induced force on the discharge damper is shown. In figures 8 through 10 the corresponding calculated peak to peak levels as a function of the velocity of sound are given.

114001 *** PULSIM 1.11 *** 14-MAY-2001 11:28:32
Customer : Cetim SYSNR: 1
System : example MODNR: 0
Duty : Natural Gas SPEED: 756.0 rpm
Part : Cylinder with pulsation dampers VOS : 402.9 m/s
Condition: 100% load PMEAN: 1600.0 kPa

Listing of pipes

Pipe	From-To	bound.	L [m]	D [m]	v [m/s]	r [kg/m3]	f [-]	q [kg/hr]	P [kPa]
1	10- 420	RE-TE	0.470	0.070	403	13.2	0.004	-3165	1493.9
	410	KW	0.270						
	419	KW	0.190						
	420	TE	0.010						
2	420- 430	TE-Q0	0.350	0.330	403	13.2	0.004	0	1497.3
3	420- 440	TE-TE	1.750	0.330	403	13.2	0.004	-3164	1497.3
4	210- 470	RE-TE	0.470	0.070	403	13.2	0.004	-3057	1494.0
	460	KW	0.270						
	469	KW	0.190						
	470	TE	0.010						
5	470- 480	TE-Q0	0.350	0.330	403	13.2	0.004	0	1497.3
6	470- 440	TE-TE	1.750	0.330	403	13.2	0.004	-3057	1497.3
7	440- 510	TE-Z0	1.210	0.107	403	13.2	0.004	-6221	1522.6
	490	OR	0.010						
	500	KW	0.200						
	510	Z0	1.000						
8	10- 20	RE-TE	0.050	0.087	403	13.2	0.004	3165	1493.6
9	20- 40	TE-VO	0.250	0.110	403	13.2	0.004	1535	1493.6
	30	KW	0.225						
	40	VO	0.025						
10	40- 50	VO-CY	0.050	0.080	403	13.2	0.004	1535	1493.3
11	20- 70	TE-VO	0.250	0.110	403	13.2	0.004	1630	1493.6
	60	KW	0.225						
	70	VO	0.025						
12	70- 80	VO-CY	0.050	0.080	403	13.2	0.004	1630	1493.2
13	50- 140	CY-VO	0.050	0.080	465	28.7	0.004	1535	4885.1
14	140- 120	VO-TE	0.250	0.110	465	28.7	0.004	1535	4884.8
	130	KW	0.025						
	120	TE	0.225						
15	80- 170	CY-VO	0.050	0.080	465	28.7	0.004	1630	4885.3
16	170- 120	VO-TE	0.250	0.110	465	28.7	0.004	1630	4885.3
	160	KW	0.225						
	120	TE	0.025						
17	120- 110	TE-RE	0.050	0.087	465	28.7	0.004	3165	4884.7
18	210- 220	RE-TE	0.050	0.087	403	13.2	0.004	3057	1493.6
19	220- 240	TE-VO	0.250	0.110	403	13.2	0.004	1571	1493.6
	230	KW	0.225						
	240	VO	0.025						
20	240- 250	VO-CY	0.050	0.080	403	13.2	0.004	1571	1493.3
21	220- 270	TE-VO	0.250	0.110	403	13.2	0.004	1485	1493.6
	260	KW	0.225						
	270	VO	0.025						
22	270- 280	VO-CY	0.050	0.080	403	13.2	0.004	1485	1493.3
23	250- 340	CY-VO	0.050	0.080	465	28.7	0.004	1572	4887.9
24	340- 320	VO-TE	0.250	0.110	465	28.7	0.004	1572	4887.5
	330	KW	0.025						
	320	TE	0.225						
25	280- 370	CY-VO	0.050	0.080	465	28.7	0.004	1485	4887.8
26	370- 320	VO-TE	0.250	0.110	465	28.7	0.004	1485	4887.8
	360	KW	0.225						
	320	TE	0.025						
27	320- 310	TE-RE	0.050	0.087	465	28.7	0.004	3057	4887.5

Figure 2.

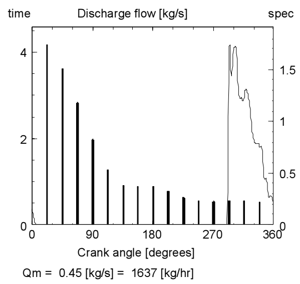


Figure 3.

In case special attention has to be paid on the performance of the compressor valves, the acoustical model is extended with a mechanical model (2nd order system) of the compressor valves. For each acoustical time step the response (lift) of the compressor valve is calculated. For this type of simulation many parameters are necessary. Research, together with compressor and compressor valve manufacturers, to optimise this type of simulation is still necessary. It is clear that a poorly designed compressor valve can have a strong effect on the damper analysis and can generate high frequency pulsations by itself.

In case the piping is replaced by an infinitely long line, there is no contribution of reflections of the pipe system to the calculated pulsation level and in most cases the minimum pulsation level is calculated in this way. Therefor, we take into account a certain safety margin for the allowable level of the calculated pulsation level at the line connection of the damper.

In case there is only one pulsation suppression device is attached to the pipe system, we design the damper such that, with an infinitely long line, the pulsations at the line connection are 80% of the allowable level. In case more pulsation suppression devices are attached to the pipe system, 70% of the allowable level is applied. This approach has been proven to be very practical. The check or design of the pulsation damper can be carried out in an early stage of the project as the piping lay out need not to be known yet. Only in very exceptional cases it appeared, during the investigation of the complete system, that installation of an additional volume was still necessary.

4.2.2 Recommendations

In case the calculated pulsations or pulsation induced forces exceed allowable levels, modifications are investigated to reduce these levels.

Simple acoustical design rules for improving pulsation dampers are:

Cylinder connection:

- As short as possible and maximum diameter.
- Equal length between cylinder connection and caps (minimum forces).

For dampers with more cylinders:

- Install baffle plate.
- Or extend cylinder connection in the damper to the axial middle of the damper.

In most systems, the installation of orifice plates in the pipe section between the cylinder and the damper is effective. The optimal location is as close as possible to the damper.

For the system as shown in figure 1 it was recommended to:

- Increase the diameter of the piping between the cylinder.
- Install baffles in both dampers.
- Install orifice plates in the pipe sections between the damper and the cylinder (close to the damper).

In figures 11 through 17 the simulation results are shown for the recommended damper and can be easily compared with the results of the original damper (figures 4 through 10). The results show clearly that the power consumption, the pulsation levels and the pulsation-induced force on the discharge damper have been reduced considerably.

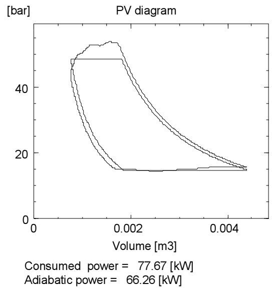
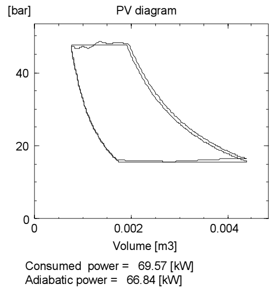


Figure 4.



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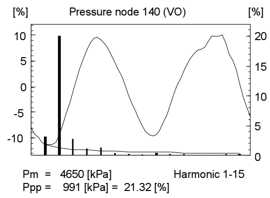

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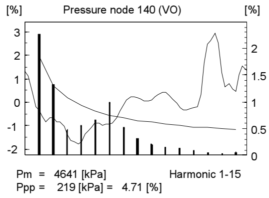

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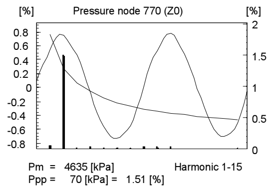

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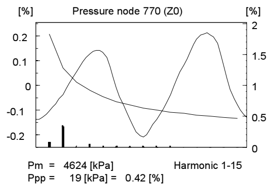

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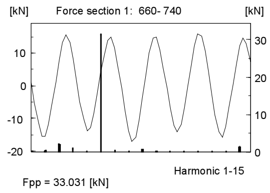
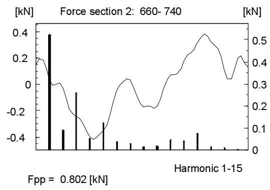


Figure 7.



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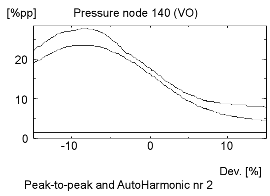
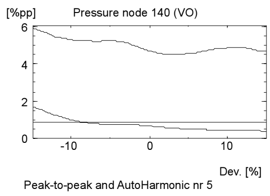


Figure 8.



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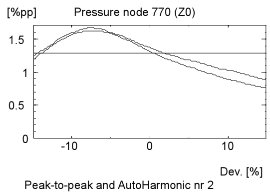

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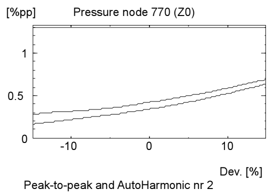

Figure 16.

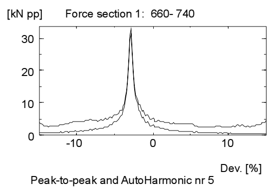

Figure 10.

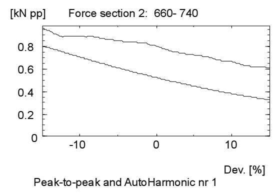

Figure 17.

4.3 Investigation of the pipe system:

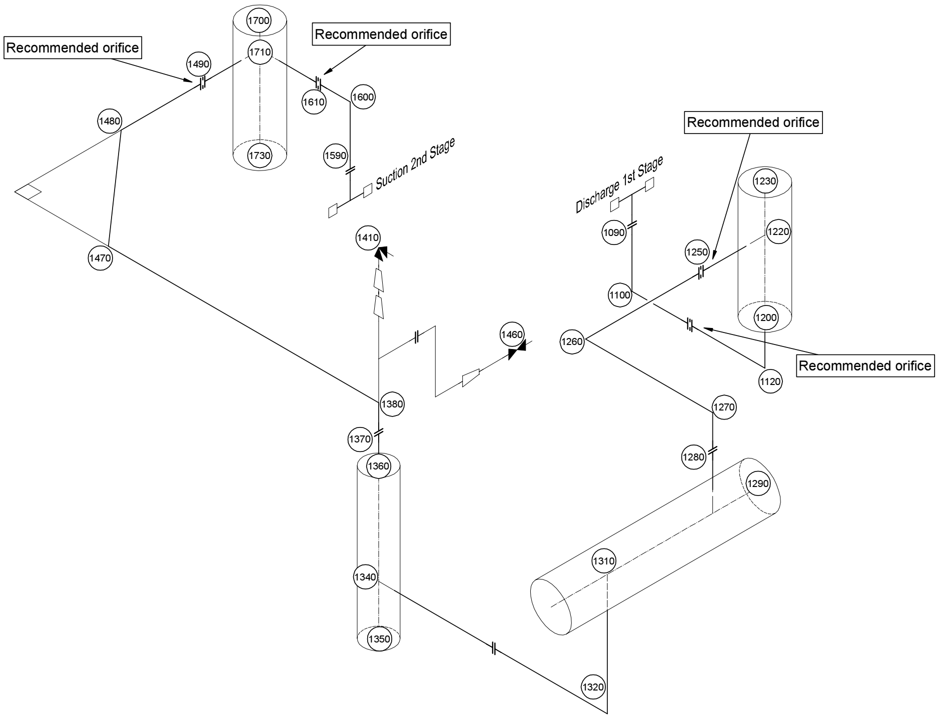

Figure 18.

After the check or the redesign of the damper has resulted in allowable pulsation levels and dynamic forces, the acoustical model of the pipe system is connected to the model of the damper. An example of a pipe system is shown in figure 18. For the analysis of the complete pipe system the results can be presented in the same way as for the check of the damper design. Some examples of the calculated levels (time function of the pulsation levels for one revolution) are shown in figures 19 and 20.

Another useful way of presenting the simulation results, to understand the acoustical behavior of the pipe system, is a plot as shown in figure 21. In this plot the calculated peak to peak pulsation level is given as a function of the location in the pipe system between both dampers. With this type of plot acoustical standing waves can be analyzed.

Because it is not useful to present a large number of plots of the calculated pulsations throughout the pipe systems, normally the simulation results of pipe systems are presented in tables. A number of presentation points in the pipe system are selected and the calculated pulsations and forces at these points are given for a range of the velocity of sound or for the range of the compressor speed.

In the tables the actual peak to peak pulsation levels can be given (see figure 22) or for each presentation point a comparison can be made with the allowable level. In case the calculated pulsation level exceeds the allowable level, the ratio of the calculated and allowable level is printed.

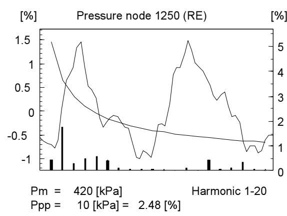

Figure 19.

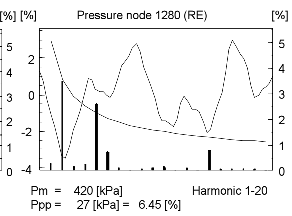

Figure 20.

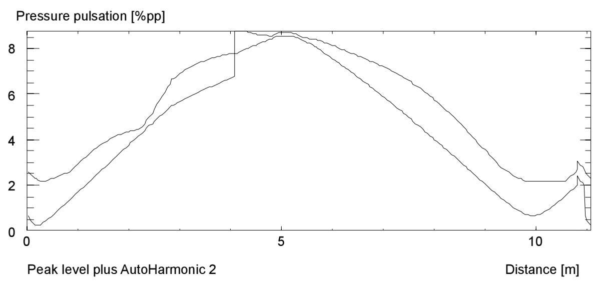

Figure 21.

For calculated pulsations below the allowable level the following symbols are used (see figure 23):

*	Calculated pulsation level between 0.9 and 1.1 times the allowable level
+	Calculated pulsation level between 0.5 and 0.9 times the allowable level
-	Calculated pulsation level between 0.1 and 0.5 times the allowable level
.	Calculated pulsation level below 0.1 times the allowable level.

The figure behind the slash sign indicates the dominant harmonic of the calculated pulsation level. In this way, the tables give a clear overview at which location and which velocity of sound (or compressor speed) the pulsations exceed the allowable level.

```
 114001 *** PULSIM 1.11  STANDARD      ***  15-MAY-2001 11:52:48
Customer : Cetim                                                SYSNR:       29
System   : example                                              MODNR:        0
Duty     : Design                                               SPEED:   585.0 rpm
Part     : Interstage I                                         VOS  :   638.0 m/s
Condition: 100% load                                            PMEAN:   429.6 kPa


Pressure pulsations [%pp], based on system mean pressure
Variation of velocity of sound
---------------------------------------------------
VOS [m/s]    558    590    622    654    686    718
DEV   [%] -12.5   -7.5   -2.5    2.5    7.5   12.5
---------------------------------------------------
User defined allowable level of 14.00 [%pp]
1020        13/4   15/5   15/5   12/5   12/6   15/6
1060        13/4   15/5   15/5   12/5   12/6   15/6
1520        12/6  8.5/6  7.4/7  9.8/7  7.5/7  8.1/8
1560        12/6  8.4/6  7.5/7  9.9/7  7.4/7  8.3/8
According to API 618 June 1995, for compressor valves 6.3 [%pp]
1090        13/4   14/5   14/5   11/5   11/6   14/6

1590        10/6  7.5/6  6.7/7  8.9/7  6.8/7  7.4/8
According to API 618 June 1995, for piping
1250       3.2/15 3.3/13 2.9/5  2.5/2  2.7/15 2.4/6
1260       4.2/13 3.4/13 2.5/15 3.0/15 3.3/15 2.6/17
1280       4.8/4  2.8/13 4.3/5  6.5/5  6.9/2  6.4/2
1330       2.5/2  3.0/2  4.6/2  7.5/2  9.7/2  9.7/2

1370       2.8/2  3.1/2  4.6/2  7.6/2   10/2   10/2
1410       4.2/13 5.1/13 5.7/13 7.7/2   10/2   10/2
1460       8.3/4  7.3/13 8.8/13  13/2   16/2   14/2
1470       5.0/4  3.4/13 4.9/5  6.3/5  6.7/2  6.7/2
1490       2.0/2  2.3/12 2.9/13 3.0/2  3.3/2  3.3/2
---------------------------------------------------
```

Figure 22.

```
 114001 *** PULSIM 1.11  STANDARD      ***  15-MAY-2001 11:52:48
Customer : Cetim                                                SYSNR:       29
System   : example                                              MODNR:        0
Duty     : Design                                               SPEED:   585.0 rpm
Part     : Interstage I                                         VOS  :   638.0 m/s
Condition: 100% load                                            PMEAN:   429.6 kPa


Legend    | Symbol    Range     | Symbol    Range     | Symbol    Range     |
          |   .    0.00 .. 0.10 |   -    0.10 .. 0.50 |   +    0.50 .. 0.90 |
          |   *    0.90 .. 1.10 | VAC          vacuum | CAV     cavitation  |

Pressure pulsations [comparison], based on system mean pressure
Variation of velocity of sound
---------------------------------------------------
VOS [m/s]    558    590    622    654    686    718
DEV   [%] -12.5   -7.5   -2.5    2.5    7.5   12.5
---------------------------------------------------
User defined allowable level of 14.00 [%pp]
1020         *      *      *      +      +      *
1060         *      *      *      +      +      *
1520         +      +      +      +      +      +
1560         +      +      +      +      +      +
According to API 618 June 1995, for compressor valves 6.3 [%pp]
1090       2.0/4  2.2/5  2.3/5  1.8/5  1.8/6  2.3/6

1590       1.7/6  1.2/6    *    1.4/7    *    1.2/8
According to API 618 June 1995, for piping
1250         +      +      +      -      +      +
1260       2.0/13 1.6/13 1.7/15 1.8/15 1.4/15 1.2/17
1280       1.8/4  1.1/13   +    1.1/5  1.6/2  1.6/2
1330         +      +    1.2/2  2.0/2  2.8/2  2.8/2

1370         +      +    1.2/2  2.0/2  2.8/2  2.8/2
1410         *    1.4/13   *      *    1.4/2  1.4/2
1460         *    1.2/13   *    1.4/2  1.9/2  1.9/2
1470       1.5/4  1.5/13 1.2/5  1.1/5  1.4/2  1.4/2
1490         -      -      *      +      +      +
---------------------------------------------------
```

Figure 23.

In case more than one compressor is acting on a pipe system PULSIM applies the following criterion [1]:

- The pulsations caused by each compressor separately should be within the allowable level.
- The pulsations caused by each compressor separately are added at each point of the pipe system and the sum of the pulsations of n compressors running parallel should be within $\sqrt{n}$ times the allowable level for each compressor separately.

In case the pulsations exceed the allowable level, modifications are investigated to achieve acceptable levels.

Pulsations in the piping can be reduced in the following ways:

- To install orifice plates or other damping devices [3] which dampen the pulsations during acoustical resonance conditions.
- To shift acoustical resonance conditions out of the operation range. This can be achieved by changing pipe length or partly changing the diameter of the piping.
- To install more volume by increasing the pipe diameter.
- To install an acoustical filter. This can be achieved by installing a secondary volume near the existing damper. With these two volumes and the pipe section in between an acoustical filter can be created with a low cut-off frequency. The filter should be designed such that the cut-off frequency is below one or two time compressor speed (for single or double acting cylinders).

Which modification is preferred depends on which parameter is most critical in the system or which modification is less expensive on the short or long term. For instance orifice plates are not expensive and can be installed easily in most cases, but the additional pressure loss can reduce the capacity and gives energy costs during the lifetime of the compressor. So these matters need to be discussed in detail with all parties (compressor manufacturer, engineering company and end user).
In tables 24 and 25 the calculated pulsation levels (actual levels and comparison) are given in case in the system of figure 18 optimal orifice plates are installed.
From the plots with the remaining pulsation induced forces as a function of the velocity of sound (see figures 26 and 27), the acoustical resonance conditions (the peaks) can be selected and used as worst-case conditions for the mechanical response analysis (maximum vibrations and cyclic stress levels).

```
 114001 *** PULSIM 1.11  STANDARD      ***  15-MAY-2001 14:46:56
Customer : Cetim                                                SYSNR:      29
System   : example                                              MODNR:       2
Duty     : Design                                               SPEED:   585.0 rpm
Part     : Interstage I                                         VOS  :   621.9 m/s
Condition: 100% load                                            PMEAN:   431.7 kPa
Orifices at cyl. connections dis. bore=123mm, suc. bore=111mm
Orifice at line connection dis. bore=90mm, suc. bore=87mm (dP=0.5%)

Pressure pulsations [%pp], based on system mean pressure
Variation of velocity of sound
---------------------------------------------------
VOS [m/s]   544    575    606    637    669    700
DEV   [%] -12.5   -7.5   -2.5    2.5    7.5   12.5
---------------------------------------------------
User defined allowable level of 14.00 [%pp]
1020      9.4/4  8.7/4  7.9/4  7.3/4  6.9/6  6.8/6
1060      9.2/4  8.5/4  7.9/4  7.3/5  6.8/6  6.8/6
1520      6.5/27 5.1/6  5.1/29 4.7/7  4.1/7  3.9/7
1560      6.0/27 4.8/6  4.7/29 4.1/7  4.0/7  3.9/8
According to API 618 June 1995, for compressor valves 6.3 [%pp]
1090      8.7/4  8.0/4  7.4/4  6.5/5  6.5/6  6.4/6

1590      4.2/6  4.3/6  4.0/7  4.0/7  3.6/7  3.4/7
According to API 618 June 1995, for piping
1250      2.5/15 2.3/13 2.0/17 2.0/17 2.0/15 1.9/19
1260      1.9/13 2.0/13 1.5/13 1.7/15 1.7/15 1.3/17
1280      1.7/13 1.6/13 1.4/13 1.8/15 2.0/15 2.1/2
1330      1.6/2  1.7/2  1.8/2  2.1/2  2.2/2  2.3/2

1370      1.8/11 1.7/2  1.8/2  2.2/2  2.1/2  2.2/2
1410      2.9/11 2.1/12 2.0/12 3.0/13 2.2/2  2.3/2
1460      6.2/11 4.0/12 3.6/12 5.2/13 6.9/4  7.6/4
1470      1.8/11 1.3/12 1.2/12 1.7/13 1.3/2  1.4/2
1490      2.6/11 1.9/12 1.9/12 2.4/13 1.7/2  1.7/2
---------------------------------------------------
```

Figure 24.

```
 114001 *** PULSIM 1.11  STANDARD      ***  15-MAY-2001 14:46:56
Customer : Cetim                                                SYSNR:      29
System   : example                                              MODNR:       2
Duty     : Design                                               SPEED:   585.0 rpm
Part     : Interstage I                                         VOS  :   621.9 m/s
Condition: 100% load                                            PMEAN:   431.7 kPa
Orifices at cyl. connections dis. bore=123mm, suc. bore=111mm
Orifice at line connection dis. bore=90mm, suc. bore=87mm (dP=0.5%)

Legend    | Symbol    Range    | Symbol    Range    | Symbol    Range    |
          |   .    0.00 .. 0.10 |   -    0.10 .. 0.50 |   +    0.50 .. 0.90 |
          |   *    0.90 .. 1.10 | VAC          vacuum | CAV     cavitation |

Pressure pulsations [comparison], based on system mean pressure
Variation of velocity of sound
---------------------------------------------------
VOS [m/s]   544    575    606    637    669    700
DEV   [%] -12.5   -7.5   -2.5    2.5    7.5   12.5
---------------------------------------------------
User defined allowable level of 14.00 [%pp]
1020        +      +      +      +      -      -
1060        +      +      +      +      -      -
1520        -      -      -      -      -      -
1560        -      -      -      -      -      -
According to API 618 June 1995, for compressor valves 6.3 [%pp]
1090      1.4/4  1.3/4  1.2/4    *      *      *

1590        +      +      +      +      +      +
According to API 618 June 1995, for piping
1250        +      -      +      +      -      -
1260        *      *      -      +      +      -
1280        +      +      -      -      -      -
1330        -      -      -      +      +      +

1370        -      -      -      +      +      +
1410        +      -      -      +      -      -
1460        +      -      -      +      *    1.2/4
1470        +      -      -      +      -      -
1490        *      -      -      *      -      -
---------------------------------------------------
```

Figure 25.

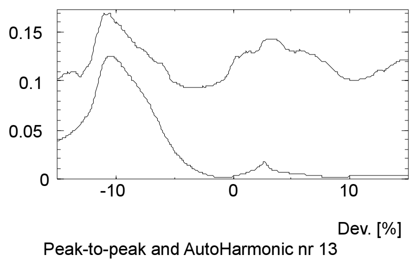

Figure 26.

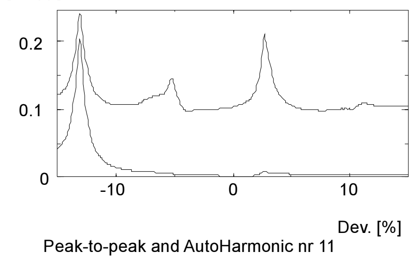

Figure 27.

5. Flow pulsations

Besides the pressure pulsations, also the flow pulsations in each pipe section are a result of the simulation. The level of the flow pulsations play an important role at the following devices:

- Non return valves. In case of strong acoustical resonances in a pipe system it is possible that flow reversal occurs for a part of the revolution time of the compressor. This will cause "hammering" of non return valves.
- Flow measuring equipment.

Many types of flow measuring devices are sensitive for flow pulsations and the flow pulsations will cause a measuring error. For measuring orifice plates, the (always-positive) measuring error can be calculated. For other types of measuring devices, for example turbine meters or vortex flow meters, sometimes data of the measuring error caused by pulsations is available at the manufacturer [4].

6. The mechanical response analysis

6.1 Introduction

One of the objectives of the pulsation analysis is to reduce the pulsation induced vibration forces to a minimum. However, unallowable vibrations and cyclic stresses can occur in case a mechanical natural frequency is close to or coincides with a frequency component of the pulsation induced vibration forces, even in case the pulsation levels itself are below the allowable level.

The objective of the mechanical response analysis is therefore to check the design of the pipe system, including the pipe supports and the construction on which the supports are mounted, to make sure that the vibration levels and the cyclic stresses are lower than the allowable levels.

The mechanical response analysis is carried out according to the API Standard 618, design approach 3, which also includes an analysis of the compressor manifold system. However, the analysis of the compressor manifold system is only included upon request of the customer [5].

For the mechanical analysis the general-purpose finite element program ANSYS is used. The pipe system is divided into basic parts (called finite elements) such as straight pipe sections, elbows, T-joints, flanges, reducers, constructions on which pipe supports are mounted, etc. The pipe supports up to the supporting beam are mostly regarded as infinitely stiff elements.

The elements which are used in the calculations are so called Timoshenko beam elements (shear deflection is included). Special attention is paid to the flexibility of the nozzle/shell intersections. The flexibility of these intersections is calculated with the finite element program FE-PIPE and the results are used in the mechanical response analysis with the ANSYS program. In case the compressor manifold is taken into account so called substructures from the shell element models are made of the crankcase, crosshead guide, distance piece and cylinder to reduce computer time. An example of such a model is shown in figure 28.

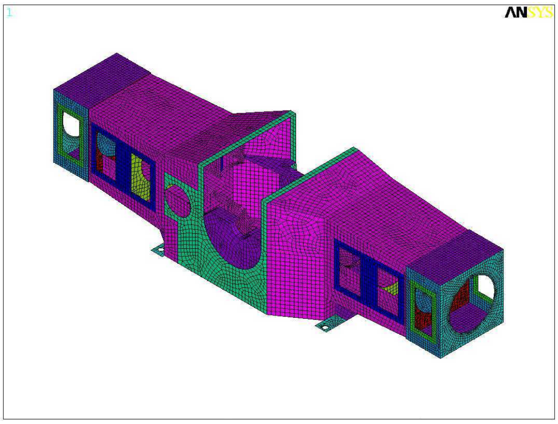

Figure 28.

6.2 Step 1: Calculation of the mechanical natural frequencies and mode shapes of the pipe system

For these calculations, the pipe system is usually split up into parts, which have no, or only small interaction. When the compressor manifold is not taken into account in the analysis, the cylinders are taken as rigid points.

The model consists of the pipe system including the supporting constructions on which the pipe supports are mounted and models of the equipment (dampers, coolers, and separators with their supports). An example of a plot of such a system is given in figure 29. This systems consists of all the piping and equipment on the skid and the supporting beams on the skid of a 4 stage compressor. In figure 30 the plot of the model is shown with all the locations where the piping is restrained and in which direction. In the plot of figure 31 the node numbers of the model are shown of which the calculated results can be given.

At first, the lower mechanical natural frequencies of these models are calculated to get a feeling of the flexibility of the system and the mode shapes. For this system the lower mechanical natural frequencies are shown in figure 32.

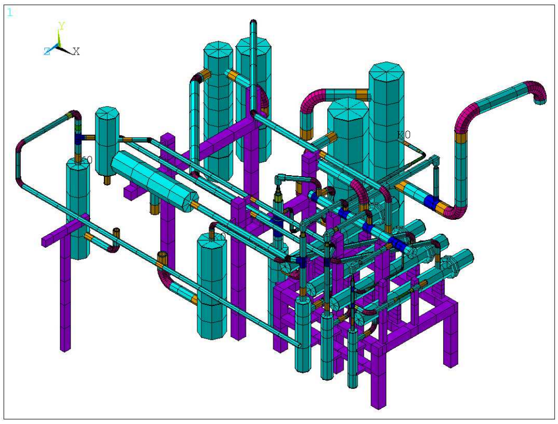

Figure 29.

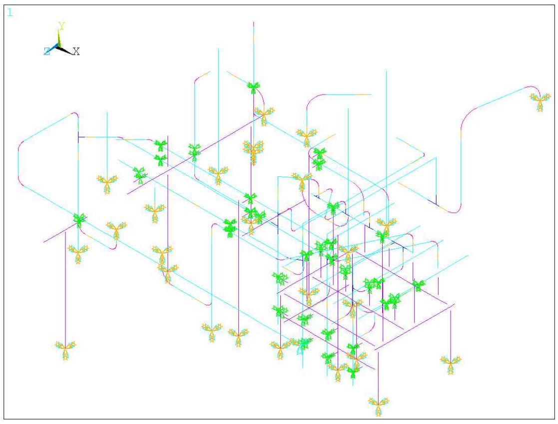

Figure 30.

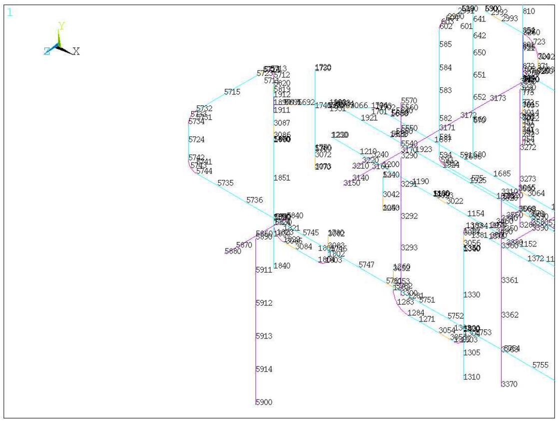

Figure 31.

```
 114001 *** PULSIM 1.11                 ***  22-MAR-2001 13:58:03
Customer : Cetim                                          SYSNR:      20
System   : example                                        MODNR:       0
Duty     : Natural Gas                                    SPEED:   585.0 rpm
Part     : Skid piping                                    VOS  :   548.9 m/s
Condition: 100% load                                     PMEAN:    199.0 kPa

List of the lower 25 natural frequencies
----------------
       Frequency
Number      [Hz]
----------------
     1      6.16                     16      27.28
     2      8.52                     17      27.93
     3     13.64                     18      28.46
     4     15.95                     19      29.06
     5     17.42                     20      29.70

     6     17.92                     21      30.35
     7     18.45                     22      31.56
     8     18.65                     23      32.40
     9     21.01                     24      32.50
    10     22.22                     25      34.01

    11     22.49
    12     23.78
    13     24.30
    14     25.06
    15     27.19
```

Figure 32.

6.3 *Step 2: Calculation of the mechanical response of the system*

The pulsation forces are calculated with the PULSIM program for the pipe system with the modifications, which have been accepted by the client. The mechanical response calculations are carried out for all important acoustical resonance conditions and the calculated amplitude and phase of the pulsation forces for each acoustical resonance condition are transferred automatically from the PULSIM program to the ANSYS program. For the calculation of the natural frequency an inaccuracy of ± 20% is taken into account. Therefore, natural frequencies, which deviate less than ± 20% from one of the harmonic components of the pulsation forces, will be excited by these components in the response calculation. In figure 33 a plot of a vibration mode of a natural frequency which is excited by the pulsation-induced forces is shown. For this vibration mode the calculated vibration level exceeds the allowable level. The maximum calculated vibration level (120mm/s peak to peak) is indicated in the plot.

Instead of presenting a lot of plots as shown in figure 33 we also present the calculated vibration and cyclic stress levels in tables. These tables have the same layout and same symbols as the tables used for the presentation of the results of the acoustic analysis (See 4.3 and figures 34 and 35). When the calculated vibration levels and cyclic stresses exceed the allowable levels, modifications are investigated to decrease the levels to allowable levels. This can be achieved in many cases by shifting the natural frequencies far enough from the excitation frequency. Shifting the natural frequency can be achieved by installing extra pipe supports or by increasing the stiffness of the structures on which the supports are mounted.

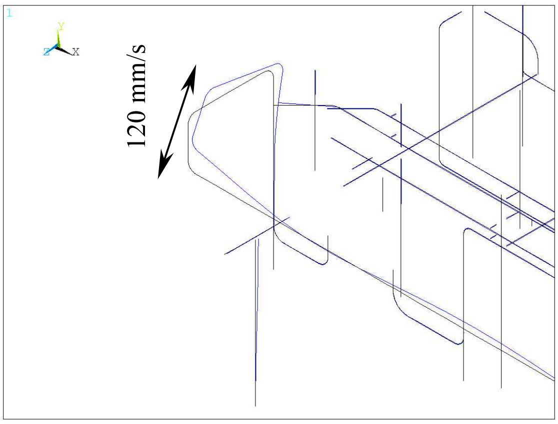

Figure 33.

```
 114001 *** PULSIM 1.11                   ***  22-MAR-2001 22:27:58
Customer : Cetim                                    SYSNR:      20
System   : example                                 MODNR:      0
Duty     : Natural Gas                                      SPEED:   585.0 rpm
Part     : Skid piping                                      VOS  :   548.9 m/s
Condition: 100% load                                        PMEAN:   199.0 kPa


Legend   | Symbol    Range    | Symbol    Range    | Symbol    Range    |
         |   .    0.00 .. 0.10 |   -    0.10 .. 0.50 |   +    0.50 .. 0.90 |
         |   *    0.90 .. 1.10 |

Vibrations [comparison]
Variation of velocity of sound
-------------------------------------------------------------------------
VOS [m/s]   467    489    505    525    541    549    562    583    609    631
DEV   [%] -15.0  -10.9   -8.0   -4.4   -1.5    0.0    2.3    6.2   10.9   15.0
-------------------------------------------------------------------------
Nodes above 1.2 x constant allowable level of 80.00 [mm/s pp]
 5715      -      +    1.2/2  1.6/2  1.7/2  1.6/2  1.3/2    +      -      -
 5724      -      *    1.6/2  2.2/2  2.2/2  2.1/2  1.7/2  1.2/2    +      -
 5732      -      *    1.6/2  2.3/2  2.3/2  2.2/2  1.8/2  1.2/2    +      -

 5733      -      *    1.7/2  2.3/2  2.4/2  2.3/2  1.9/2  1.3/2    +      -
 5734      -      *    1.7/2  2.3/2  2.4/2  2.3/2  1.9/2  1.2/2    +      -
 5735      -      +      *    1.3/2  1.4/2  1.3/2    *      +      -      -

 5742      -      *    1.5/2  2.1/2  2.1/2  2.0/2  1.7/2    *      +      -
 5743      -      *    1.5/2  2.0/2  2.1/2  1.9/2  1.6/2    *      +      -
 5744      -      +    1.4/2  1.9/2  1.9/2  1.8/2  1.5/2    *      -      -
-------------------------------------------------------------------------

Stresses [comparison]
Variation of velocity of sound
-------------------------------------------------------------------------
VOS [m/s]     467    489    505    525    541    549    562    583    609    631
DEV   [%]   -15.0  -10.9   -8.0   -4.4   -1.5    0.0    2.3    6.2   10.9   15.0
-------------------------------------------------------------------------
Nodes above 0.26 x constant allowable level of 60.00 [N/mm2 pp]
 2480         -      -      .      .      .      .      .      .      -      .

 5400         .      -      -      -      +      -      -      -      -      .
-------------------------------------------------------------------------
```

Figure 34.

```
 114001 *** PULSIM 1.11                   ***  22-MAR-2001 22:27:58
Customer : Cetim                                    SYSNR:      20
System   : example                                 MODNR:      0
Duty     : Natural Gas                                      SPEED:   585.0 rpm
Part     : Skid piping                                      VOS  :   548.9 m/s
Condition: 100% load                                        PMEAN:   199.0 kPa


Legend   | Symbol    Range    | Symbol    Range    | Symbol    Range    |
         |   .    0.00 .. 0.10 |   -    0.10 .. 0.50 |   +    0.50 .. 0.90 |
         |   *    0.90 .. 1.10 |

Vibrations [mm/s pp]
Variation of velocity of sound
-------------------------------------------------------------------------
VOS [m/s]   467    489    505    525    541    549    562    583    609    631
DEV   [%] -15.0  -10.9   -8.0   -4.4   -1.5    0.0    2.3    6.2   10.9   15.0
-------------------------------------------------------------------------
Nodes above 1.2 x constant allowable level of 80.00 [mm/s pp]
 5715     20/2   62/2   96/2  131/2  137/2  129/2  107/2   70/2   35/2   13/2
 5724     27/2   81/2  126/2  172/2  179/2  169/2  140/2   92/2   46/2   17/2
 5732     28/2   85/2  132/2  180/2  188/2  177/2  146/2   96/2   49/2   18/2

 5733     29/2   88/2  137/2  187/2  195/2  184/2  152/2  100/2   50/2   18/2
 5734     29/2   87/2  135/2  185/2  193/2  182/2  150/2   99/2   50/2   18/2
 5735     17/2   50/2   78/2  106/2  111/2  104/2   86/2   57/2   29/2   10/2

 5742     26/2   77/2  120/2  164/2  171/2  161/2  133/2   88/2   44/2   16/2
 5743     25/2   75/2  116/2  159/2  166/2  156/2  129/2   85/2   43/2   15/2
 5744     23/2   70/2  109/2  149/2  155/2  146/2  121/2   79/2   40/2   14/2
-------------------------------------------------------------------------

Stresses [N/mm2 pp]
Variation of velocity of sound
-------------------------------------------------------------------------
VOS [m/s]     467    489    505    525    541    549    562    583    609    631
DEV   [%]   -15.0  -10.9   -8.0   -4.4   -1.5    0.0    2.3    6.2   10.9   15.0
-------------------------------------------------------------------------
Nodes above 0.26 x constant allowable level of 60.00 [N/mm2 pp]
 2480        17/3  6.3/3  4.1/3  3.6/2  3.5/3  3.6/3  3.8/3  4.0/4   12/4  4.9/4

 5400       4.6/2   14/2   22/2   30/2   31/2   29/2   24/2   16/2  8.0/2  2.8/2
-------------------------------------------------------------------------
```

Figure 35.

6.4 *Allowable levels*

6.4.1 Allowable cyclic stress

The calculated cyclic stresses include the stress intensification factors for T-joints, flanges, elbows and reducers and are taken according the piping code ANSI B31.3. When the peak value of the cyclic stresses exceeds the endurance limit of the material, fatigue problems will occur.

Unless the client explicitly specifies another criteria, an allowable cyclic stress according the API Standard 618 of 179 N/mm2 pp will be used for each excited natural frequency. This value has been based on the endurance limit and is only valid for carbon steel pipe with an operating temperature below 371 C. All other stresses must be within the applicable code limits.

For welds an extra stress concentration factor of 3 is used. However, because not all the field weld locations are known during the mechanical response analysis, the weld stress concentration factors are not included in the stress calculation. Therefore, the allowable level of 179 N/mm2 will be divided by the stress concentration factor of the weld. An allowable cyclic stress level of 60 N/mm2 will therefore be used in the tables.

6.4.2 Allowable vibration levels

To make sure that equipment such as temperature transmitters, pressure gauges, valves, flanges etc. will not fail due to too high vibrations the vibration levels are also calculated.

During the analysis the vibration levels are calculated during coincidence of acoustical and mechanical resonance conditions (worst case conditions). Therefore, we apply an allowable level of 80 mm/s pp during the analysis. In practice, during measurements, we apply an allowable level of 42 mm/s pp (15 mm/s RMS).

6.5 How vibration levels and cyclic stresses can be reduced

When the vibration levels and/or the cyclic stresses exceed the allowable level there are several possible modifications to reduce the vibration levels and/or cyclic stresses. Some possible modifications are explained below.

Shifting resonances

An effective way to decrease the vibration levels and/or cyclic stresses is to shift the mechanical natural frequencies in such a way that they cannot be excited by the pulsation forces.

This can be achieved by installing additional supports, relocating of existing supports or by stiffening the structures on which the supports are mounted. It is also possible to shift the natural frequencies by changing the diameter or the layout of the pipe system. However, this will often have a considerable effect on the pulsation levels and pulsation induced forces so that the pulsation analysis would have to be carried out again for the modified layout.

Changing pipe properties

When the cyclic stresses only exceed the allowable level at stress risers in the pipe system such as T-joints, flanges, reducers, etc. it is possible to increase the local stiffness at such points. The increase in stiffness can be achieved by increasing the pipe diameter and/or wall thickness. When the stiffness is only increased locally this will have in most cases a minor influence on the pulsations and pulsation forces.

When extra supports are installed, or when the stiffness of the structures on which the supports are mounted is increased, the expansion stresses will be probably increased. It is important that the expansion stresses are always within the allowable levels according to the applicable piping code.

So the static and dynamic aspects compete in the design of a pipe system. This problem can be solved by searching acceptable compromises. At locations in the pipe system where the static and dynamic designs compete, e.g. a so-called spring hold down support can be used. When the friction force between the spring hold down support and the structure on which the support is mounted is equal to the peak value of the pulsation force than the spring hold down support restrains the dynamic motion.

The expansion stresses are mostly very large in comparison with the friction force and therefore the spring hold down support is free for the thermal motion.

7. Other dynamic flow phenomena investigated with PULSIM

Besides the pulsation analysis according the API Standard 618, design approaches 1, 2 and 3 and according appendix M of this Standard, points M1 through M 11, the PULSIM program has been extended for the investigation of the following kind of dynamic flow phenomena:

- PULLIQ. This is a special version of the PULSIM program for the investigation of pipe systems of reciprocating pumps. This analysis is carried out according the API Standard 674. In this analysis the pipe elevation (static pressure) and the static pressure loss in the piping and equipment play an important role to investigate if cavitation can occur.
- PULTRAN. This is a special version of the PULSIM program for the investigation of high pressure amplitude transients. These high amplitude transients (shock waves) can for instance occur in case an emergency valve is opened within a very short time. During such a transient the pressure variation is high and the density cannot longer be considered constant and in PULTRAN the density variation is taken into account.
- The investigation of so called FIPS (Flow induced pulsations) [6]. These pulsations are generated by unsteady separation of the flow at sharp edges, for instance at valves, orifice plates, tee connections and sharp bends. Strong pulsations can occur when the unsteady flow separation is coupled with acoustical resonances in the pipe system. In case the source (frequency and amplitude) can be

predicted with sufficient accuracy and is implemented in the PULSIM model, the pulsations thoughout the system can be calculated.

9. Further improvements

To improve the PULSIM package, continuous research and development is necessary. If possible the R&D is carried out together with our clients. At this moment, the following developments are ongoing within the flow and structural dynamics department of TNO.

- The interaction of compressor valve dynamics and pipe system. An improved valve model in PULSIM can lead to:
  - better compressor performance predictions, both in power consumption and capacity,
  - better prediction high frequency pressure fluctuations caused by valve instabilities
  - a more detailed pulsation analysis which can help the manufacturers in the design of pulsation dampers and the selection of compressor valves.
- The development of accurate source models for centrifugal compressors and pumps. Increasingly, pressure pulsations are observed in process installations with centrifugal compressors and pumps caused by an interaction with the dynamics of the pipe system. A dynamic source model can lead to accurate predictions of the stability of these systems.
- Source models of flow-induced pulsations. These can be caused by unsteady flow separation for example at T-joints, bends and valves. In contrast to external pulsation sources such as reciprocating compressors the internal sources are influenced by the flow itself. The geometry and the gas velocity according to a specific Strouhal number determine the pulsation frequency.
- New models for pulsation damping. Apart from the standard orifice plates, used for the reduction of low frequency pulsations, new types of dampers are now available, such as perforated plates. Experimental validation has been carried out to evaluate the performance of various types of damperplates for a wide frequency range [3].

10. References

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