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The Vibroacoustic Analysis of The Hydrocarbon Processing Plant Piping System Operating at Elevated Temperature

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In this paper it is presented the vibroacoustic analysis of the selected section of the hydrocarbon processing chemical plant piping system operating at elevated temperature and subjected to dynamic load exciting vibration of the structure. The pump suction and discharge piping system is a part of chemical plant for processing hydrocarbon mixture at 270 ^{o} C. Elevated temperature is one of static loads that influences the boundary conditions of the piping structure thus generating pump nozzle loadings leading to possible pump body deflection. Deflected shape of the pump body results in generation of flow fluctuation, visible and measurable as a pressure pulsation. This kind of fluctutation has been assumed further to be one of the dynamic loading on piping system structure. The dynamic analysis was performed to quantify the loading effect of pressure pulsation excited in the pump discharge nozzles on the structure of pipelines and the connected pump nozzles. The simulation was based on the numerical analysis of the excitation by acoustic waves propagation in subjected piping system. Measured on–site pressure pulsation at pumps nozzles has been identified and assumed to be the source of the acoustic waves. In the simulation elastic features of the piping structure as well as the fluid, and pressure loses in pipes, taken into account. Final result of the acoustic part of the simulation was spectral characteristics of the acoustic shock forces, defined further as harmonic loads for the dynamic structural analysis. To observe an influence of the acoustic excitation on the piping there was performed structural analysis of the piping system and the combined results of static and dynamic loading influence determined. This part of the analysis has been perfomed by means of FEM computer software Bentley AutoPIPE as well as some use of ANSYS FEM program. Important step in this simulation there was the theoretical modal analysis. This analysis allows to predict possible vibroacoustic resonance in the structural system under specific conditions of the coincidence between acoustic excitation and modals. The results of the combined static and dynamic loadings analysis contain the information on the node displacements, internal forces, resulting stresses in the pipe walls and loads on the pump nozzles and piping supports.

Keywords: vibroacoustic analysis, pressure pulsation, plant piping system.

1. Introduction

Pipings systems are basic components of the media transportation systems in the process industrial plants. As mechanical structures they are subjected to a variety of static and dynamic loads, which sources include the self-weight, the internal media pressure and the elevated temperature of the piping wall, also seismic, wind and other environmental loadings. An important role in process piping systems play loads derived from dynamic phenomena in the fluid media. Depending on many factors they may or may not influence significantly the operation of piping plants. In this group of loads it is important to predict these responsible for resonant vibration excitating and undertake efforts to protect the piping system against them. Depending on the structure complexity and its operating conditions, it may be difficult task.

The structural analysis procedures of industrial piping systems are formalized to some extent, strictly defining numerous requirements and technical parameters. An example of such formalization is the European Standard EN 13480-3 "Metallic Industrial Piping – Part 3: Design & calculation" [2]. This standard allows designers involved in the design and analysis, construction and operation of the installation capabilities of defining the list of loadings that are included in the formal stress analysis of the system. This leads to situations that piping systems are put into service after being only partially verified in the static stress analysis. As a rule, from which one may encounter occasional exceptions, is skipping the impact of any dynamic loads in the system structural analysis.

The dynamic loads threaten especially the piping systems structure due to the slenderness of the linear structural elements (pipes) equipped with the concentrated masses distributed on the system (valves, equipment etc.). These structures tend to vibrate in operating conditions at frequencies much below 500Hz, what is in accordance to the usually occuring in plants excitation frequencies. The excitating frequencies depend on the type of equipment and several other factors. For flow pulsation excitating sources e.g. reciprocating compressor they may start from 10Hz, while for rotating pumps it is much higher, as this is usually no less than blade passing frequency if no other source is located in the system. Further in this paper pressure pulsation is subjected to the field measurement, although it is worth to mention that it can be exchanged into flow pulsation, as these two parameters must change in parallel due to physical laws governing the fluid flow phenomena.

For the effective application of dynamic analysis it is necessary to perform the theoretical modal analysis first. This is necessary for application of the methodology adopted for the structure nodes displacement, internal forces and moments calculation, but also very usefull for estimation the possible correlation of structure modes and acoustic modes. Theoretical modal analysis of the structure allows to identify the tendency of mechanical oscilation and the solution of it is used in the quantitative dynamic response analysis under defined acoustic loading.

In the case of simulations described in this article, e.g. the flow / pressure pulsation

it is assumed the 100% filling the interior space of pipelines and there are other assumptions typical for one–dimensional acoustic analysis described in [1].
The phenomena is the subjected system may occur in wide variety of gas and liquid piping systems, not limiting the considerations to the hydrocarbon plants. Therefor conclusions may be applied to e.g. power piping systems as well as gas plants piping systems.

2. The hydrocarbon plant piping system subjected to analysis

The subject of the research simulation and analysis is the mixture of liquid hydrocarbon plant piping system. The core of this system are three centrifugal pumps, that operate either as a single or double pumping system, depending on current process requirement.

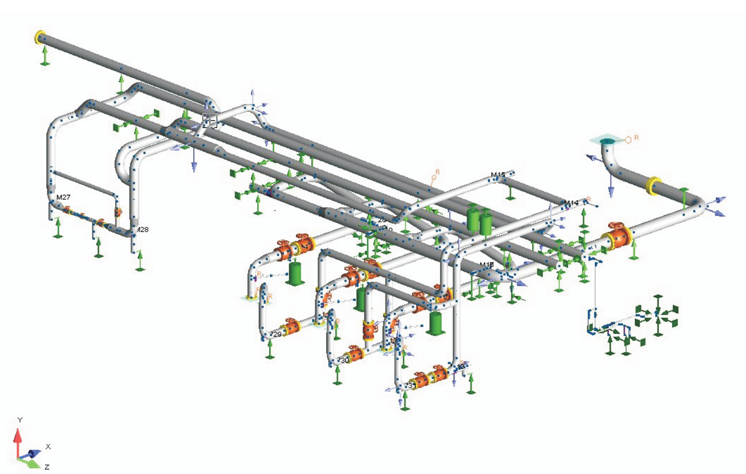


Figure 1 The overall view of the piping system under consideration. AutoPIPE model

During normal operation, two pumps are on, and the third is the hot spare. It is also possible the start-up scenario: a single pump is operating. In the Fig. 2 there is visible a layout of suction pipelines of process pumps, Fig. 3 shows discharge.
There was significant vibration of the piping system in the start-up phase, still present at the operating conditions, however only at certain rotating speed and certain operating configurations of the system, described in details in [5]. The vibration parameters were subject of the on-site research and registered. Sample of it is presented below, see Fig. 4.

In the system there have been installed three single-stage centrifugal pumps, equipped with six blades impeller of double suction layout. The synchronous frequency of rotation of these pumps engine is 1500 RPM. Flow system channels of pump consists of the centrifugal impeller, the inlet volute casing and the double outlet volute casing.

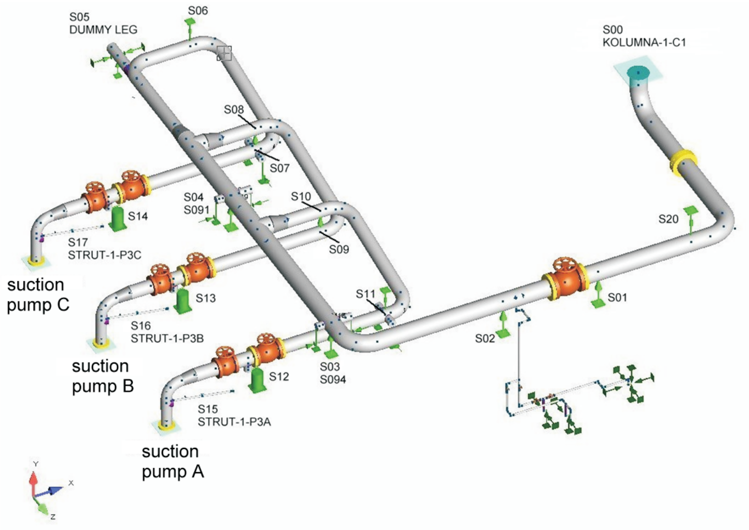


Figure 2 The view of suction pipes modeled in AutoPIPE

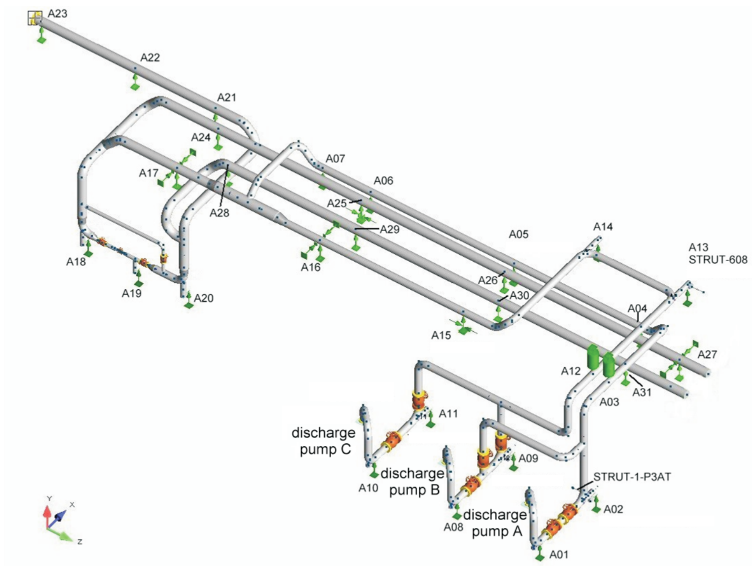


Figure 3 The view of discharge piping system model in AutoPIPE

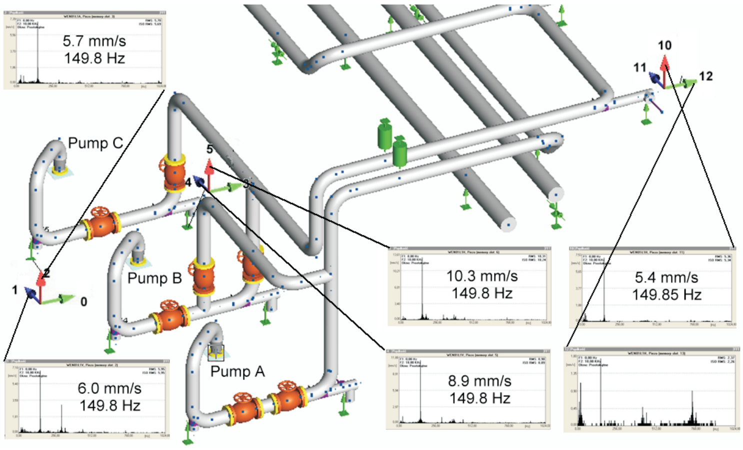


Figure 4 Sample on-site measurement recordings – pumps B & C operate at nominal speed

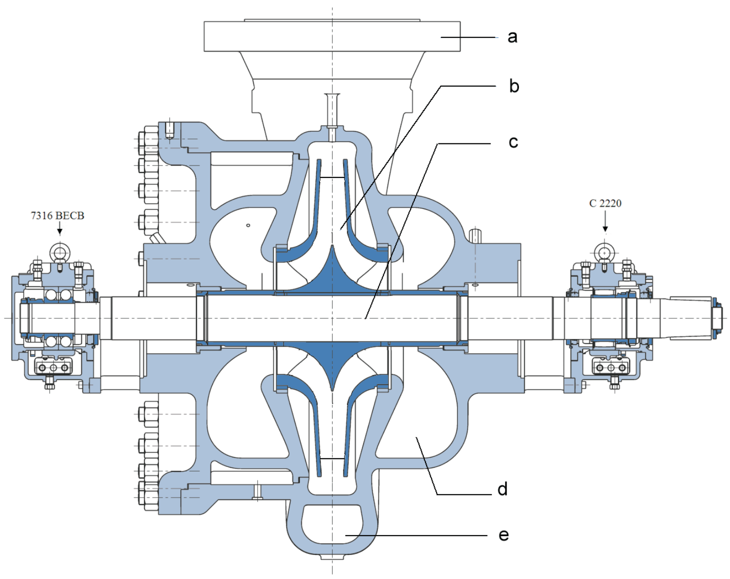


Figure 5 Meridional cross–section of the process pump

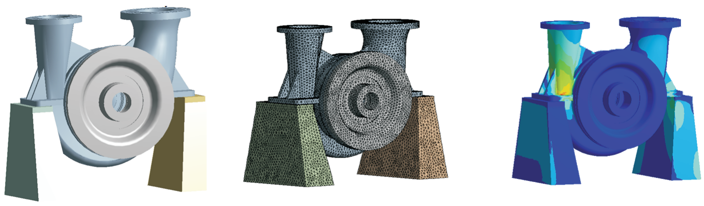


Figure 6 Finite Element Model of process pump

Table 1 The FEM calculated process pump nozzles stiffness

	Pump Nozzle stiffness						Thermal displacement (22°C - 272°C)		
	$Fx/\Delta x$	$Fz/\Delta z$	$Fy/\Delta y$	$Mx/\phi x$	$Mz/\phi z$	$My/\phi y$	Δy	Δz	Δx
	N/mm	N/mm	N/mm	Nm/1°	Nm/1°	Nm/1°	mm	mm	mm
Suction nozzle	98 887	299 279	1 892 149	1 541 669	289 442	1 544 873	3,86	- 1,52	0
Discharge nozzle	74 811	189 241	1 742 274	579 934	155 973	414 507	3,86	1.48	0

The rotating assembly of the pump has bearings:

1. on the opposite drive side, two angular, single–row ball bearings,
2. on the drive side, barrel bearing.

In the Fig. 5 there is shown the meridional cross-section of the pump with marked basic elements.
a – Discharge nozzle,
b – impeller,
c – shaft,
d – inlet volute casing,
e – double outlet volute casing.
Due to the fact that it was necessary to identify accurate physical data of the structural elements for the purposes of the dynamic analysis, there was necessary to build a model of the pump body and determine its stiffness, that is one of important boundary conditions for vibration analysis. It is usually not being under consideration neither for static, nor for dynamic industrial code-compliance analysis based on Code requirements. Later on it was confirmed that these values have significant impact on the modal analysis results.
In order to determine the stiffness of pump casing nozzles, there have been carried out the numerical computation of displacements and angles for both nozzles. The computations were performed for admissible loads given by the pump manufacturer. Computed values of stiffness are shown in the Tab. 1. These values are introduced in the coordinate system consistent with the standard ISO 13709 (API 610) [2].

3. The fluid–structure dynamic interaction and vibration excitation sources identification

One of possible reasons of piping structure vibration was assumed the piping–structure mechanical interaction. In this assumption the non-uniform static pressure layout and periodic changes along piplines results in dynamic forces acting on piping structure. The source of the pressure pulsation has been identified in the on–site measurement process.

Measurements were done by the use of fast changing pressure probes, resistant to elevated temperatures. These probes were assembled directly to the pipeline at the suction and discharge of the pump. Collecting the data was carried out continuously, and the figures show only a few seconds sample form the measurement process. Sampling frequency was 2 kHz. Measurement card had the anti–aliasing filter installed, set to 500 Hz.

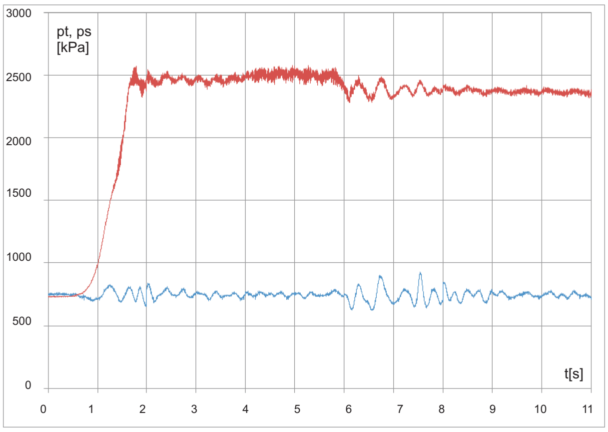


Figure 7 Suction pressure (blue line) and discharge pressure (red line) of process pump measured at pump start–up

The following Fig. 8 contains charts of the static pressure time history at suction and discharge nozzle as well as the spectral characteristics of the static pressure.

4. The combined acoustic – structural analysis

The combined analysis of the subjected piping system is compsed of several parts, described in [1]:

1. Modal analysis of the structural model of plant piping system;
2. Acoustic analysis i.e. pressure pulsation simulation in the piping system. The goal of this simulation is to define the fluid-structure dynamic loading acting on piping structure;

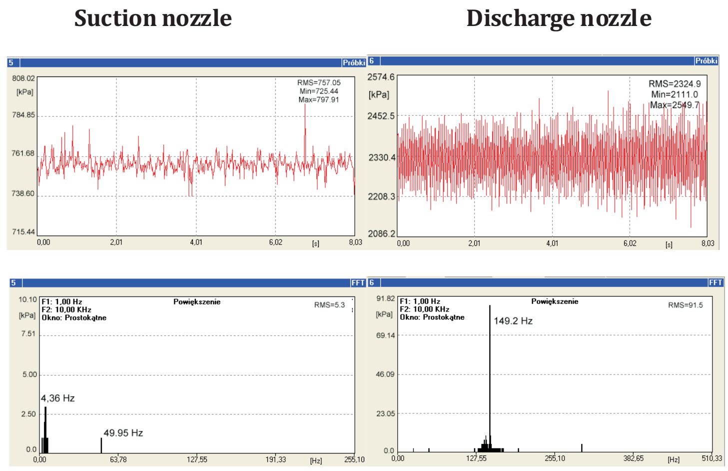


Figure 8 Static pressure at suction nozzle (left) and discharge nozzle (right) measured for pump A in operating conditions. Upper row: time history, lower: spectrum of the pressure signal

3. Structural analysis of the subjected piping system under dynamic loading defined in the acoustic analysis. In this analysis it is also possible to verify the combined static and dynamic effects on the piping structure.

5. Modal analysis of the piping structure of the plant

To complete the identification of vibration sources and confirm the possible fluid–structure interaction there were two analyses necessary, followed by correlation studies. The first one was the theoretical modal analysis.

Modal analysis is commonly applied in practice by the use of technique of dynamic characteristics examination technique for mechanical objects. The theoretical modal analysis applied for linear elastic model of the piping structure allows to predict free-vibration frequencies of the subjected structure and assigned shapes of combined harmonic motion. It also allows to predict the behavior of the structure as a result of any imbalance caused e.g. by external loading. This kind of analysis is used for the purpose of modifying the structure, structural health diagnosis, active vibration reduction and for the purposes of verification and validation of numerical models such as finite element or boundary elements method.

The natural vibration frequencies and shapes of the subjected system were determined, and the accuracy resulting from an adopted computational linear model confirmed in on-site tests and measurements. To increase accuracy of the numerical model there have been used the results of vibration measurements at supports.

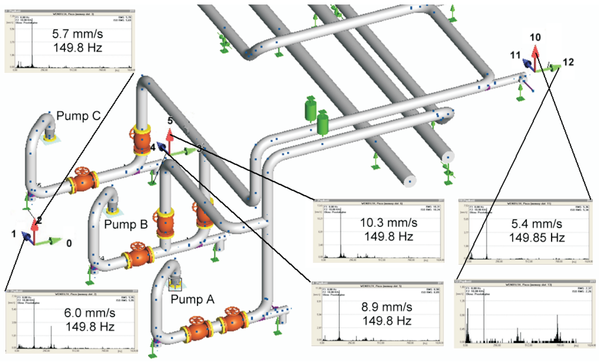


Figure 9 Measuring position of vibrations at discharge pipelines

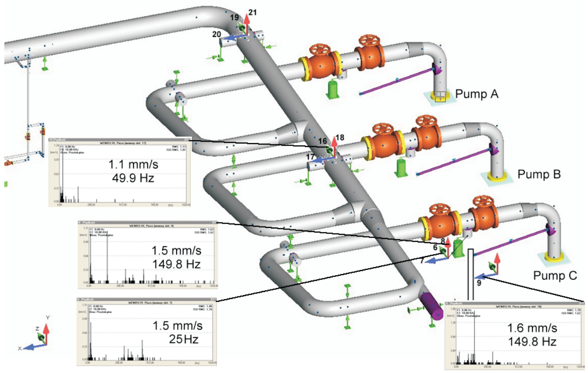


Figure 10 Measuring position of vibrations at suction pipelines

The subset of final results of the theoretical modal analysis of the model after modifications of the structural piping model are presented in Tab. 2. It was intentionally limited to the frequencies close to the excitation frequency 149.2 Hz shown in the bottom–right chart on the Figure 8, above.

Table 2

Variant of pumps operation			Free vibration frequency
Pump A	Pump B	Pump C	
running	running	standby	144.98, 146.95, 148.02, 150.65, 153.25, 155.00
running	standby	running	145.14, 146.49, 150.73, 152.56, 154.95, 155.00
standby	running	running	145.24, 146.67, 147.95, 148.67, 150.80, 152.56, 154.96

For computations there have been adopted the following pipelines temperatures:

1. 272^{o}C – pipelines temperature, which hold a flowing agent,
2. 150^{o}C – pipelines temperature with a pump standby,
3. 5^{o}C – pipelines temperature which do not hold a flowing agent.

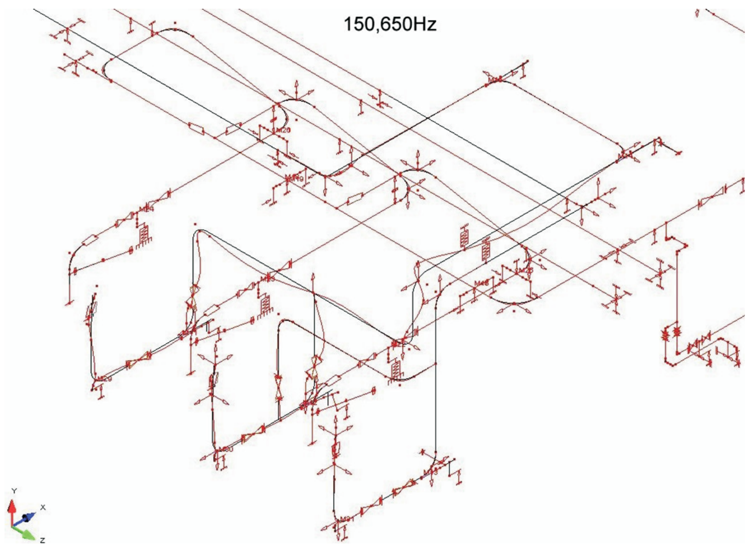


Figure 11 Free vibrations shape of discharge pipelines. Pumps A and B running and pump C standby

It is important to emphasize the fact that inclusion of the calculated stiffness of process pumps nozzles, increased the accuracy and allowed to precisely identify shapes and frequencies of natural vibrations of pipeline systems.

In most piping systems pressure pulsation does not cause noticeable and dangerous effects. It is due to relatively substantial stiffness and high attenuation of the actual structure. This may change substantially when forces from the acoustic phenomena stimulate construction vibration under resonance conditions. For this reason, modal analysis is an essential part of the study in terms of sound using their results to the analysis of the vibration risk.

6. Acoustic analysis goals and rules

The aim of the acoustic simulation and further structural analysis is to confirm or reject the thesis of the possible origin of the source of pulsation and resultant vibration of excited piping system. If this analysis allowed the qualitative and/or quantitative analysis of this kind of load it would be possible to identify and apply effective methods of pulsation prevention and damping, hence minimize the causes and results of vibration.

For the subjected piping system has been made the numerical simulation of acoustic wave propagation in the pipeline using a pressure pulsation source. This theoretical pulsation force has been defined according to result from the field measurements. The simulation takes into account the elastic parameters of the fluid and the walls of the pipe as well as flow frictional losses.

The result of the above simulation are spectral characteristics of acoustic shock forces (dynamic forces used later on in dynamic structural analysis) for specific, selected rotating speed of pumps. These forces act along the piping elements between consecutive bends axially, so under stationary operating conditions stimulate the vibration of the system. Actual systems attenuate vibrations, so no visible results are encountered. However there is number of piping systems where resonant conditions occur, vibration may rise to noticeable and even dangerous level. Having the spectral characteristics of these forces it is possible to perform the dynamic analysis of structures in AutoPIPE and thus obtain information on the nodal displacements, forces, moments and ultimately determine the stresses in the walls of the pipes. Obtaining these values allows the code assessment of actual object as well as to identify problems and restore the proper functioning of the system.

The qualitative and quantitative verification of the simulation should be done as a series of on–site measurements of vibration parameters in key point of the piping system. These parameters are then compared to their predicted values of vibration amplitude and frequency.

As a result of preliminary studies and on-site measurements, it was assumed that the source of pressure pulsation is the flow fluctuation caused by internal geometry of pump channels near the discharge nozzle.

There was taken into account in further research, that several physical parameters of the fluid may affect significantly simulation results. Thus the key parameters fluctuation sensitivity analysis has been also done. Further in these paper it is presented result of such analysis for fluid density and bulk modulus change. This is very important as they change significantly with temperature changes, thus affect acoustic simulation.

7. Acoustic analysis description

From the large number of models there were several selected as the most representative for the analysis:

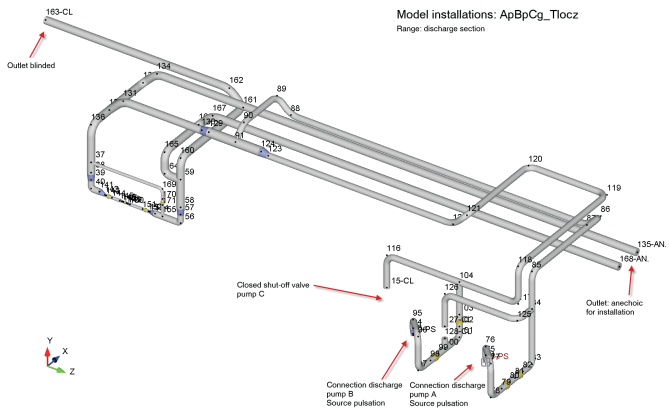


Figure 12 Model #1 labeled A_p_B_p_C_g_272_5: discharge piping system for defined for acoustic analysis in operating case with pumps A & B on and the pump C shut-off valve closed. On piping end 163 is the blind flange, ends 135 and 168 connected to anechoic, long piping system

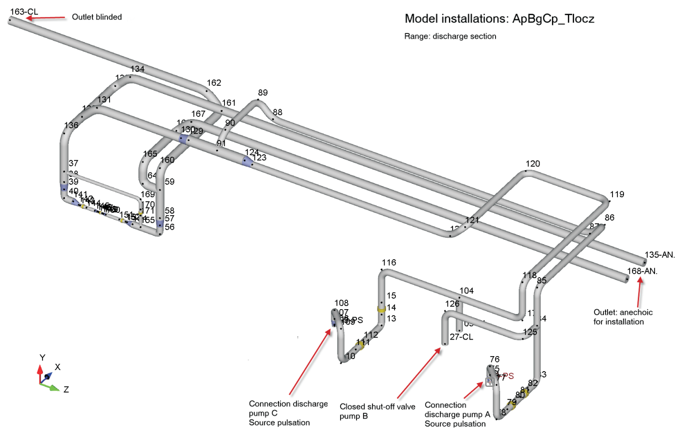


Figure 13 Model #2 labelled A_p_B_g_C_p_272_5: discharge piping system for defined for acoustic analysis in operating case with pumps A & C on and the pump B shut-off valve closed. On piping end 163 is the blind flange, ends 135 and 168 connected to anechoic, long piping system

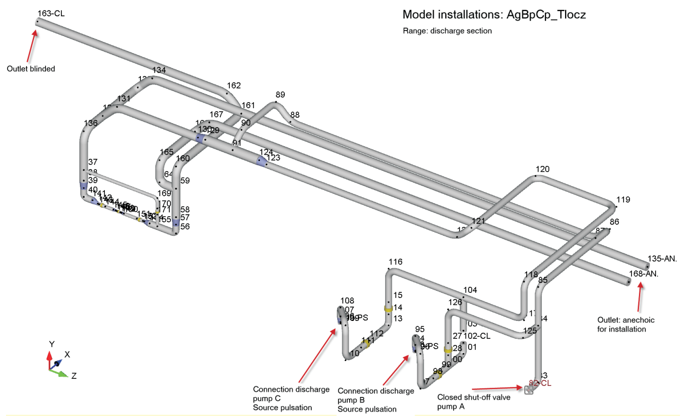


Figure 14 Model #3 labeled A_g_B_p_C_p_272_5: discharge piping system for defined for acoustic analysis in operating case with pumps B & C on and the pump A shut-off valve closed. On piping end 163 is the blind flange, ends 135 and 168 connected to anechoic, long piping system

The next paragraph of this paper discusses the progress of execution of simulation and analysis results for Model #1. Other models brought the results allowing for a detailed analysis of the subjected piping system.

8. Model #1 acoustic analysis

Physical data of the fluid are critical parameters of the acoustic analysis. They significantly affect the quality of the acoustic wave propagation analysis results.

1. density = 609 kg/m3
2. dynamic viscosity = 0.1495 cP
3. fluid bulk modulus = 1400 MPa
4. Speed of sound in fluid = 1516.2 m/s
5. Speed of sound in fluid in elastic pipe = 1360 m/s

8.1. *Location and characteristics of pulsation sources*

The spectral characteristics of the sources of pressure pulsations were determined based on on-site measurements made. In the case of Model #1 source pulsations are located in the pump discharge pipe connections A and B. The amplitude–frequency characteristics of pulse sources defined for acoustic analysis are presented below:

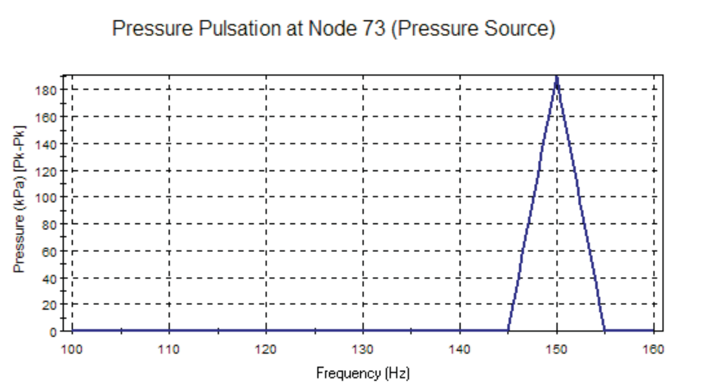


Figure 15 Spectral characteristics source of pressure pulsation on pump nozzle A

8.2. *Acoustic analysis results*

As a result of the numerical simulation of pressure waves propagation, the following values of harmonic acoustic shock forces were found. They are assumed to be the dynamic load for further structural analysis.

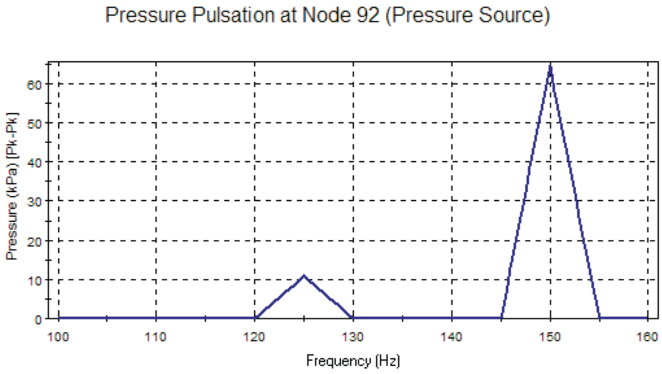


Figure 16 Spectral characteristics source of pressure pulsation on pump nozzle B

Freq. (Hz)	Point	F (N)	Phase
148.0	F00	1139.0	179.4
154.0	F00	962.7	-9.5
148.5	F02	3871.0	178.1
154.0	F02	1772.0	-11.6
150.0	F03 F	12400.0	-179.9
153.5	F04 F	8707.0	175.0
150.0	F05 F	16744.0	0.8
122.5	F10 F	162.0	0.0
153.0	F10 F	8747.0	177.2
122.5	F11	1861.0	-17.9
150.0	F11	5403.0	1.2
122.5	F12 F	3383.0	-17.6
150.0	F12 F	11119.0	-178.0
149.5	G00	847.0	-138.1
150.0	G02	2422.0	-157.9
125.0	G03 F	615.0	-179.0
150.0	G03 F	4237.0	-179.8
146.5	G04 F	1913.0	-8.3
125.0	G05 F	1099.0	0.7
149.0	G05 F	5147.0	-54.2
125.0	G10 F	641.0	-176.0
149.0	G10 F	4191.0	146.7
150.0	H11 F	2273.0	11.8
149.0	H13 F	5149.0	-65.8
122.5	H14 F	1183.0	166.3
154.0	H14 F	624.6	158.4
122.5	H16 F	1346.0	-15.4
149.0	H16 F	7100.0	120.5
122.5	I01 F	1790.0	162.1
150.0	I01 F	5119.0	1.1

8.3. *Structural analysis of the piping system under combined: static and dynamic loading*

The acoustic analysis results have been applied in AutoPIPE structural model as a loading case H1. For dynamic analysis needs there was also applied modal analysis up to 1.5 of the largest frequency of the dynamic load, i.e. 240 Hz.
The detailed study of the modal analysis results showed that there are some modal shapes/frequencies that coincident with the pressure pulsation excitation. It means that there is the risk of resonant vibration excitation - the example of this is presented below, see Fig. 17, where for pump A exists the modal at f = 154.9986 Hz and its shape contains axial movement of pipe segment between consecutive bends.

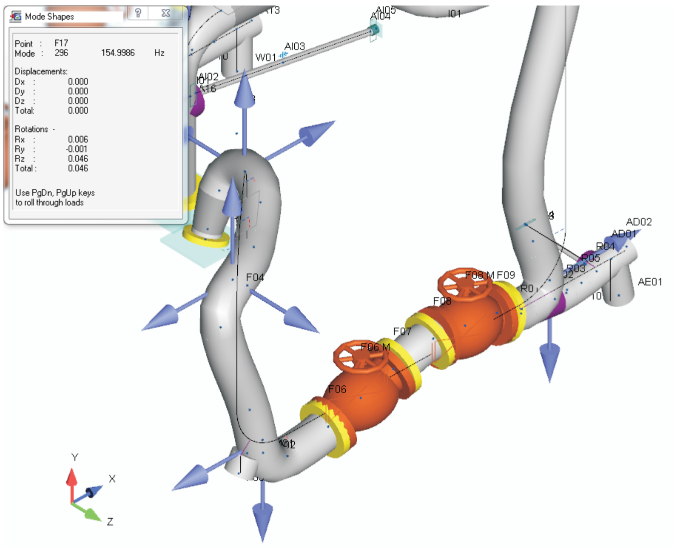


Figure 17 Modal analysis results: mode shapes $f = 154.9986$ Hz

Therefore it can be assumed that the pressure pulsations, excited in the internal channels of the pump is propagated in the form of acoustic waves in pipes. In certain circumstances of piping geometry this lead to formation of harmonic axial forces in the straight sections between consecutive bends.

8.4. *Loading cases and their combinations*

In order to determine the structural response the combined load calculations were performed. In this process there were determined nodes displacements, internal forces, moments, including the loads on nozzles and supports. One of load combinations presents all the values for pressure pulsation excitated loading – it is labeled **Harmonicz.1**

Combination	Print	Auto aktual.	Method combin.	Type combination	Version combination
	☑	☑			
Gravit.{1}	☑	☑	Sum	Default	OWN WEIGHT (INSTALLATION + LIQUID)
User 1{1}	☑	☑	Sum	Default	ONLY REACTIONS HYDRODYNAMIC
GT1P1{1}	☑	☑	Sum	Default	STATIC WITHOUT HYDRO. AND PULSATION
Harmonicz.1	☑	☑	Sum	Default	ONLY PULSATION
GT1P1U1	☑	☐	Sum	User-Defined	STATIC
GT1P1U1H1	☑	☐	Abs. S	User-Defined	STATIC + PULSATION (DYNAMIC)

In–depth results analysis of resultant stress in pipe wall as well as pump nozzle load, shows a significant effect of the dynamic load of pressure excited pulsations on the piping system structure.

Seg	Point	Combination	Category	StressCalc	StressAccept	Indic.	On_press	On_bend	StressElong	StressShear	Ma (Long)	Mb (Sporadic)	Mc (extend)	ForceLong.	SIF(i)	Nr. equ.
				MPa	MPa		MPa	MPa			Nm	Nm	Nm			
F	F03 F-	GR + MaksP{1	Long-term	19.11	129.32	0.15	18.00	1.14	0.00	0.00	868.95	0.00	0.00	0.00	2.63	2-1
F		Dltr + U1{1}	Sporadic	19.25	129.32	0.15	18.00	1.29	0.00	0.00	868.95	110.01	0.00	0.00	2.63	3-1
F		Dltr + U1{1} +	Sporadic	20.70	129.32	0.16	18.00	2.74	0.00	0.00	868.95	1210.51	0.00	0.00	2.63	3-1
F		ONLY H1	Sporadic	1.47	129.32	0.01	0.00	1.47	0.00	0.00	0.00	1114.00	0.00	0.00	2.63	3-1
F	F03 F+	GR + MaksP{1	Long-term	18.54	129.32	0.14	18.00	0.58	0.00	0.00	868.95	0.00	0.00	0.00	1.00	2-1
F		Dltr + U1{1}	Sporadic	18.62	129.32	0.14	18.00	0.65	0.00	0.00	868.95	110.01	0.00	0.00	1.00	3-1
F		Dltr + U1{1} +	Sporadic	19.35	129.32	0.15	18.00	1.39	0.00	0.00	868.95	1210.46	0.00	0.00	1.00	3-1
F		ONLY H1	Sporadic	0.74	129.32	0.01	0.00	0.74	0.00	0.00	0.00	1113.92	0.00	0.00	1.00	3-1
F	F04 N-	GR + MaksP{1	Long-term	18.79	129.32	0.15	18.00	0.83	0.00	0.00	1242.99	0.00	0.00	0.00	1.00	2-1
F		Dltr + U1{1}	Sporadic	18.88	129.32	0.15	18.00	0.91	0.00	0.00	1242.99	124.58	0.00	0.00	1.00	3-1
F		Dltr + U1{1} +	Sporadic	19.82	129.32	0.15	18.00	1.86	0.00	0.00	1242.99	1540.42	0.00	0.00	1.00	3-1
F		ONLY H1	Sporadic	0.95	129.32	0.01	0.00	0.95	0.00	0.00	0.00	1417.27	0.00	0.00	1.00	3-1
F	F04 N+	GR + MaksP{1	Long-term	19.60	129.32	0.15	18.00	1.64	0.00	0.00	1242.99	0.00	0.00	0.00	2.63	2-1
F		Dltr + U1{1}	Sporadic	19.77	129.32	0.15	18.00	1.80	0.00	0.00	1242.99	124.58	0.00	0.00	2.63	3-1
F		Dltr + U1{1} +	Sporadic	21.63	129.32	0.17	18.00	3.67	0.00	0.00	1242.99	1540.42	0.00	0.00	2.63	3-1
F		ONLY H1	Sporadic	1.87	129.32	0.01	0.00	1.87	0.00	0.00	0.00	1417.27	0.00	0.00	2.63	3-1

Figure 18 Comparison of stress values at selected nodes of the model. Stress due to load from pressure pulsations marked red

Seg	Point	Combination	DN	GlobalFX	GlobalFY	GlobalFZ	GlobalRFZ	GlobalMX	GlobalMY	GlobalMZ	GlobalRMZ
			mm	N	N	N	N	Nm	Nm	Nm	Nm
F	pmpAd F00	GT1P1{1}	0.00	-12495.01	-4222.40	1102.89	13235.19	13058.82	-9754.95	-1717.64	16390.31
F		Harmonicz.1	0.00	1743.79	5186.16	4691.73	7207.59	318.84	235.88	276.95	483.74
F		GT1P1U1	0.00	-12500.04	-2597.17	1215.60	12824.74	13090.98	-9694.96	-1751.23	16383.92
F		GT1P1U1H1	0.00	12500.04	5847.62	1215.60	13853.64	13090.98	9814.95	1751.23	16455.21
G	pmpBd G00	GT1P1{1}	0.00	-5417.26	-7859.54	-7834.87	12349.26	1094.15	-7757.89	-489.39	7849.93
G		Harmonicz.1	0.00	1222.68	3202.47	4055.72	5310.33	154.19	186.97	201.95	315.46
G		GT1P1U1	0.00	-5405.41	-6222.72	-7543.92	11173.70	1222.25	-7634.25	-517.22	7748.76
G		GT1P1U1H1	0.00	5429.10	9496.37	8125.81	13626.63	1222.25	7881.52	517.22	7992.48
H	pmpCd H00	GT1P1{1}	0.00	-2484.96	-7469.12	-6862.37	10442.93	-1909.96	305.84	555.20	2012.40
H		Harmonicz.1	0.00	762.79	308.78	178.34	842.02	33.12	31.02	145.29	152.21
H		GT1P1U1	0.00	-2487.42	-7427.14	-6467.67	10157.77	-1680.57	483.68	569.81	1839.28
H		GT1P1U1H1	0.00	2487.42	7511.10	7257.07	10736.34	2139.36	483.68	569.81	2266.16

Figure 19 Comparison of the forces and moments at the points of connection piping to the pump discharge nozzle. The loading combination Harmonicz.1 contains the load from the pressure pulsations only

8.5. *Sensitivity analysis: influence of the physical fluid characteristics changes on the acoustic analysis results*

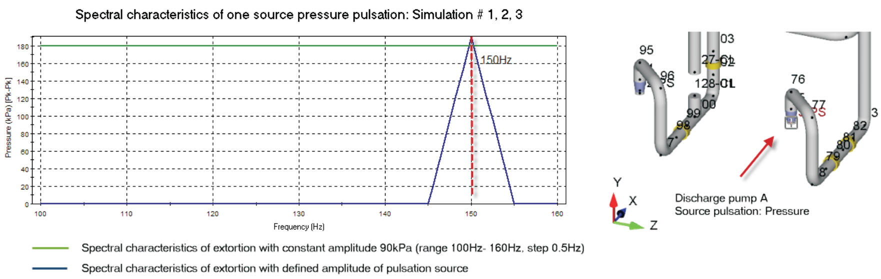
In previous chapters there was presented analytical confirmation of the thesis that the pressure pulsation propagated along the pipins system in certain conditions is one of significant loads on the plant structure. The accuracy of the simulations of acoustic effect and its quantitative influence to piping structure, significantly depends on the assumed physical properties of the fluid, ie. density and bulk modulus. In most of detailed calculations reported herein, the following values of these parameters were assumed:

1. density ρ = 609 kg/m3
2. bulk modulus B = 1400 MPa

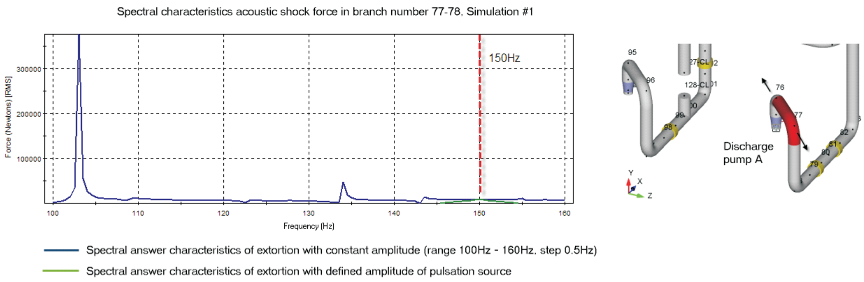
However, in the set of data accessible from various sources, parameters of exactly the same media may vary. Density may change between 627 kg/m^3 end 873 kg/m^3, depending on various factors, including temperature. This influences simulation results significantly.
Similarly, the value of the bulk modulus may vary from that used in the simulation calculations, based on the data provided. Literature gives different values of this parameter for petroleum products:
Fluid Bulk Modulus (MPa) Density ρ (kg/m^3)
Crude oil 1500 835
Oil 1100 – 1600 855 - 963
Considering the above, additional numerical simulations were carried out for different values of density and bulk modulus. In each simulation however all other data remained unchanged.

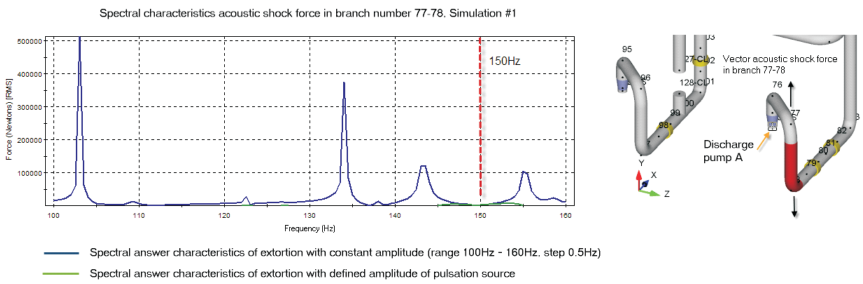
8.6. *Model #1; simulation case #1 (basic fluid data characteristics; all simulation cases):*

1. _density = **609 kg/m^3**
2. dynamic viscosity = 0.1495 cP
3. fluid bulk modulus = **1400 MPa**
4. speed of sound in fluid = **1516.2 m/s**
5. speed of sound in fluid in elastic pipe = **1360 m/s**



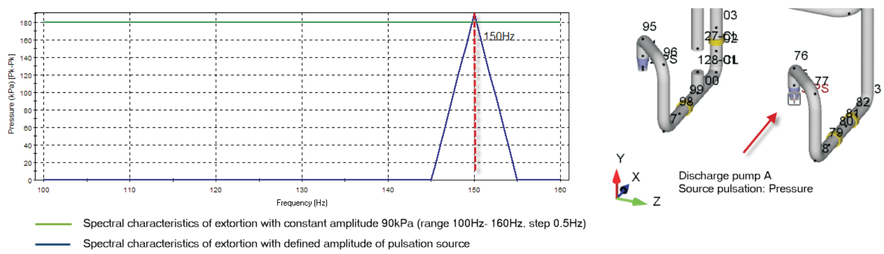
Sample results of simulation case #1:



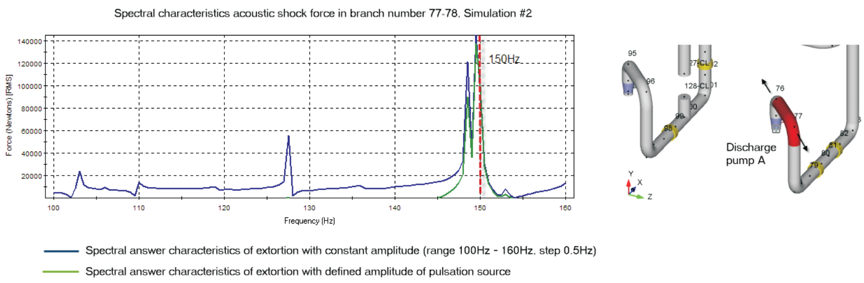


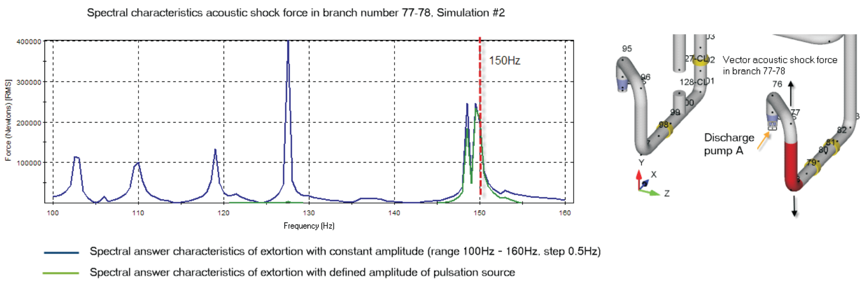
8.7. *Model #1; simulation case #2:*

1. _density = **873 kg/m³**
2. dynamic viscosity = 0.1495 cP
3. fluid bulk modulus = **1100 MPa**
4. speed of sound in fluid = **1122.2 m/s**
5. speed of sound in fluid in elastic pipe = **1042 m/s**



Sample results of simulation case #2:





8.8. *Model #1, simulation case #3:*

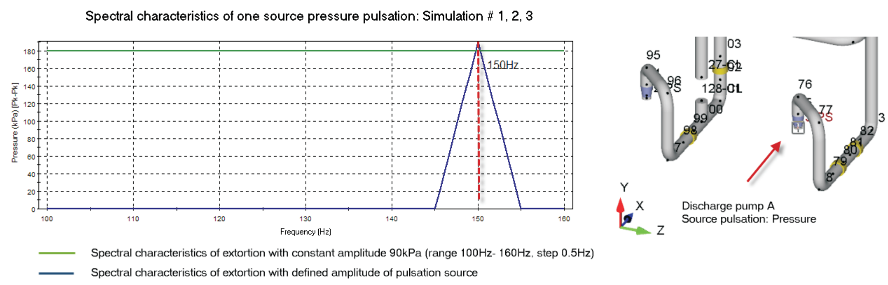
1. _density = **609 kg/m$^{\mathbf{3}}$**

2. dynamic viscosity = 0.1495 cP

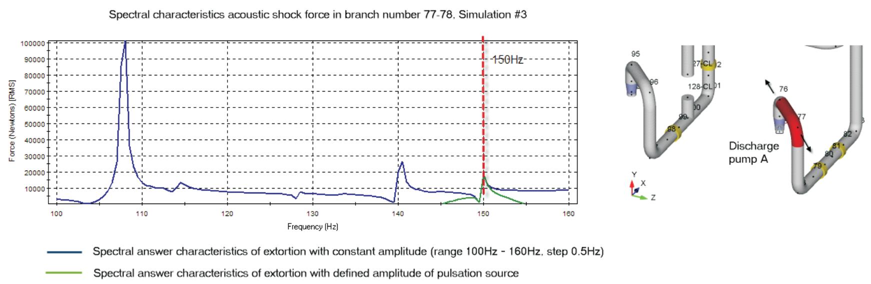
3. fluid bulk modulus = **1500 MPa**

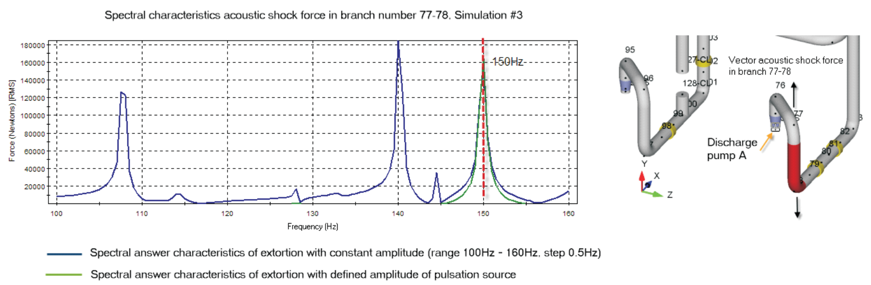
4. speed of sound in fluid = **1569.0 m/s**

5. speed of sound in fluid in elastic pipe = **1422 m/s**



Sample results of simulation case #3:

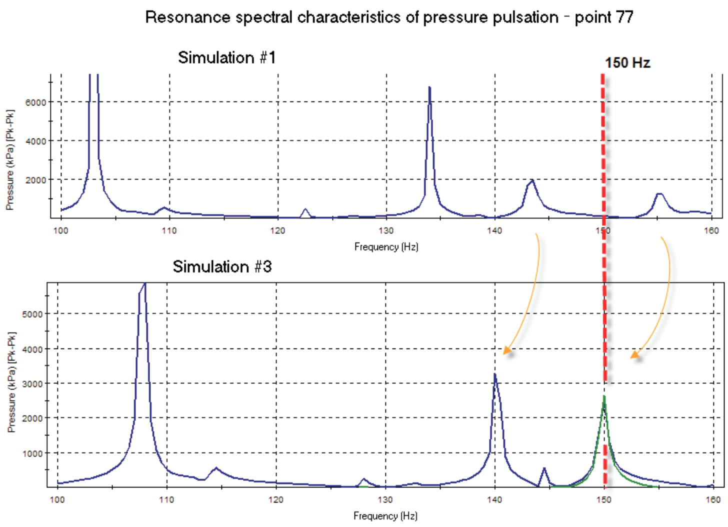




8.9. *The comparative analysis of results for cases #1, #2 & #3*

Comparing the simulation results it is clear that it is not possible to clearly confirm or exclude the risk arising from the pressure pulsations in the plant piping system under consideration without precise knowledge of the fluid data.

Comparing Simulation cases #1 and #3, where fluid density has the same value, while the bulk modulus values vary significantly, should be noticed that values of the acoustic shock forces at a frequency of about 150 Hz also vary. The reason is that bulk modulus value has significant impact on sound speed in fluid, what impacts resonance characteristics of the specific piping system. In case #3 the frequency 150Hz is one of resonant frequencies, while for #1 it is not – on the bulk modulus change the model of system has been retuned as far as acoustic characteristic is concerned.



As a result should be noticed significant difference of loading on the pump nozzle as well as actual stress in pipe walls, what is an observable factor that has been encountered in this pump piping system.

9. Summary and conclusions

1. The results of dynamic analyses of the subjected piping system show the significanty influence of the measured pressure pulsation on the loading of piping structure, including pump nozzles.

2. Calculated values of pressure pulsation and resulting forces applied as the dynamic load, determined in this sample model at the nominal operating conditions should be considered as approximate. Actual values may differ, especially if the design is 'susceptible dynamically' on the specified load type, ie. there are similar forms of load (axial forces) and modal shapes of construction. The problem in this specific case, as well as overall complex dynamic analysis of this type the accuracy of the physical elements, e.g. data of the fluid, where the pressure pulsation is propagated. A particularly important parameters in this case are bulk modulus of the liquid and its density.

3. Notwithstanding the results of detailed structural analysis, it is recommended to design the structure so the effects of acoustic resonances in pipelines are

properly separated from the modal shapes of piping plant construction. It is recommended detuning the piping system system so no fundamental acoustic frequency is below the pump speed rotational frequency neither close to the blade passing frequency of the rotor in centrifugal equipment as well as their harmonics.

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