20th International Conference on Structural Mechanics in Reactor Technology (SMiRT 20)
Espoo, Finland, August 9-14, 2009
SMiRT 20-Division 6, Paper 1658

Active Control of Vibrations in Piping Systems

Carsten Block[a], Jürgen Engelhardt[a], Fritz-Otto Henkel[a]

[a] *Woelfel Beratende Ingenieure, Hoechberg, Germany, e-mail: block@woelfel.de*

Keywords: Active vibration control, piping systems, active vibration absorber

1 ABSTRACT

Dynamic loads acting on piping systems often are not considered in the planning phase of plants. Therefore, in most cases no design at all or insufficient design against vibrations is performed.

However, vibrations in piping systems can lead to major problems. This paper presents a method to actively reduce vibrations in piping systems.

2 INTRODUCTION

Piping systems in plants can fulfil different tasks. Therefore, the dimensions and the used materials vary significantly. In addition, the loading highly depends on the intended use. Beside the common static load cases dead weight, pressure and temperature dynamic load cases may occur. Table 1 presents an overview of typical dynamic load cases for piping systems in plants.

In particular it is hard to predict steady state vibrations due to e. g. pressure surges or physical / chemical reactions occurring during operation. For that reason plants are not designed for such dynamic loads in most cases. But vibrations occurring in piping system after the initial start up of operation or after the backfitting of a plant can lead to serious problems. Beside safety-related problems, which can result in a downtime of a plant, the productivity or the quality of the product can be affected. Both cases lead to high economic loss.

Table 1: Typical dynamic load cases for piping systems by Schalk (1990)

Operating State	Reason / Examples	Characteristics
Normal Operation	Vibrations of directly connected machines - motors, pumps, compressors - mixers - centrifuges	deterministic steady state (harmonic/periodic) or transient normally force-time-curves
	Excitations from external events - machines in the vicinity - construction sites, traffic	deterministic steady state or transient Time dependent base excitation
	Pressure surges - switching of pumps - actuation of valving	deterministic transient force-time-curves
	Physical-chemical reactions	stochastic steady state, power spectrum
	Wind, vortex shedding	periodic, force-time-curves
	Motion of the sea for maritime constructions	deterministic/stochastic steady state force-time-curves, power spectrum
Upset operation / Incidents	Earthquake	stochastic/deterministic transient time-curves, response spectra, base excitation
	Explosion pressure waves	deterministic transient, pressure-time-curves, response spectra
	Pipe burst Machine faults	deterministic transient, force-time-curves
Tests	Loads from tests - dynamic excitation - snap back tests	harmonic/periodic/stochastic transient or steady state

If the resulting vibrations exceed permissible values and a smooth operation of a plant is not possible remedial measures have to be taken. VDI Guideline 3842 offers a good overview on passive methods to reduce vibrations in piping systems. Common measures for vibration reduction in piping systems are:

- reduction of excitation
- detuning
- damping
- passive tuned mass dampers

The efficiency of conventional passive methods for vibration reduction is limited. The application of passive viscoelastic dampers, for example, requires a fixed support which is not always possible to realise due to cramped confines. Tuned mass dampers demand a mass ratio of about 10 % to reach a good attenuation. Therefore, tuned mass dampers often are unfeasible due to high static load. In addition, tuned mass dampers are restricted to one natural frequency and get ineffective if the dynamic properties of the piping system change.

To overcome problems with conventional passive methods of vibration reduction, active vibration control systems can be used.

3 ACTIVE VIBRATION CONTROL

In active vibration control external energy is transformed e. g. in a mechanical force by means of actuators. This mechanical force is used to actively reduce vibrations in a controlled manner. (Figure 1).

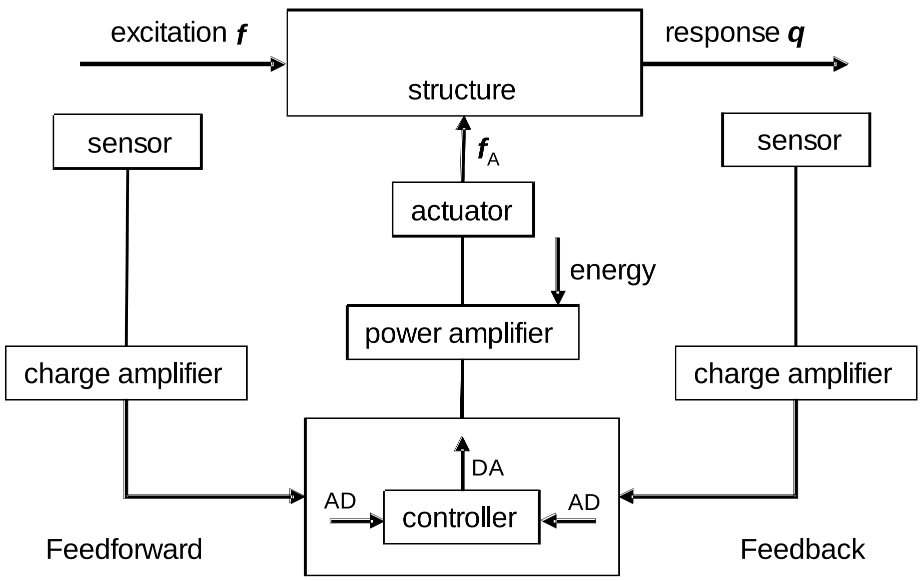

Figure 1: Components of a system for active vibration control

There are two main control strategies: feedforward and feedback control. In the case of feedback control, the response of a system due to excitation is measured with a suitable sensor and fed back into a controller to generate a proper control signal. This signal drives the actuator to attenuate the vibration. In feedforward control, the excitation and not the response is measured and fed forward into a controller to generate the control signal. Feedforward control is only applicable if a signal correlating to the excitation is available. For this case, the response of the system can theoretically be forced to be zero.

The choice of an adequate system for active vibration control depends on many different factors. These are for example the type of the dynamic excitation, which can differ in direction, amplitude and frequency range, the requirements on the degree of vibration reduction and whether the vibrations shall be reduced globally or in a limited region (for more details see Block (2008)).

4 ACTIVE VIBRATION ABSORBER AVA

In the present contribution the possibilities of active vibration reduction in piping systems using an **A**ctive **V**ibration **A**bsorber (AVA) are shown. For a verification of the vibration reduction potential the AVA shown in Figure 2 was designed, manufactured and tested (see Engelhardt et al. (2007))

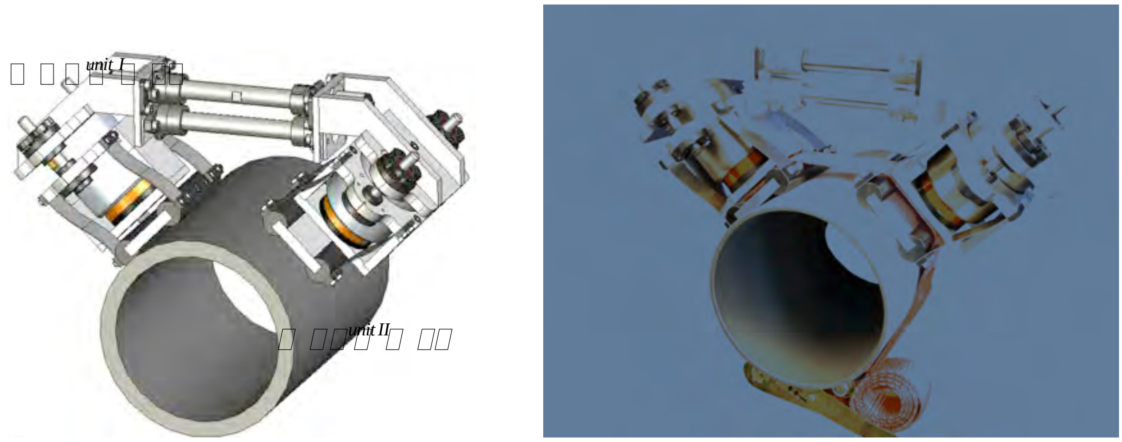

Figure 2: Active Vibration Absorber AVA (left: design, right: realization)

The AVA's function is based on the principle that an accelerated inertial mass generates a reaction force in the supporting structure:

$$F = -m \cdot \ddot{x} \qquad (1)$$

The reaction mass is connected to the structure by means of a spring. An actuator is located parallel to the spring. By accelerating the reaction mass, using the actuator in combination with an appropriate control algorithm, a resultant force for vibration reduction is obtained.

A straightforward control strategy consists in the Direct Velocity Feedback, by means of which a broadband vibration reduction can be achieved.

In order to achieve an observability and controllability of all vibration modes perpendicular to the pipe axis, the AVA is designed as a 2 DOF system. The two identical units of the AVA are oriented perpendicular to each other and are connected by means of a rigid connection. Via this connection the same reaction mass is obtained for both effective directions. Consequently, lower actuator strokes at the same total weight are needed. The major part of the reaction mass is provided by the actuator magnets (Figure 3e). For actuation two electrodynamic voice coils are used. These customised actuators provide a constant force of 110 N (330 N peak force) and a stroke of 20 mm. The actuator's magnets and coils are guided relative to each other by an adjustable linear bearing that is free of clearance (Figure 3d). The coil is connected to the fixed part of the linear bearing by means of a frame (Figure 3c). This frame is connected via leaf springs (Figure 3f) to the interface (Figure 3g) that is attached to the piping. The interfaces are clamped to the pipe with a tension belt (not shown).

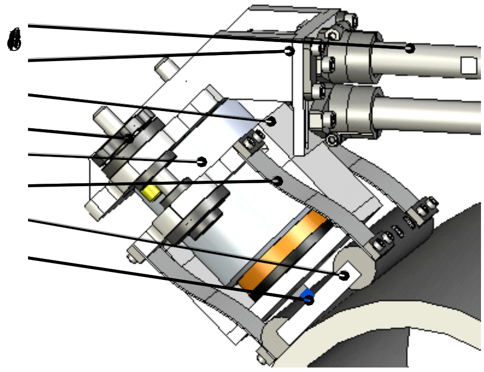

Figure 3: Components of the AVA

The leaf springs fulfil various tasks:

- provide compliance of each AVA-unit perpendicular to the actuators effective directions
- guide the AVA-units in the actuators effective directions
- transmit the reaction forces to the piping
- carry the static load of the reaction mass

By varying the thickness of the leaf springs, the AVA's natural frequencies can be tuned. By means of the setting mechanism (Figure 3a) the centre positions of the actuators are adjusted. The setting mechanism is also used for adapting the AVA to different pipe diameters. The two accelerometers needed for the controller are mounted on the pipe interface (Figure 3h).

Figure 4 shows the mode of operation. The static load is accepted carried by the leaf springs of both AVA units. For vertical motion both actuators are driven by an identical signal, for horizontal motion by an identical signal with opposite sign. The preconditioned sensor signals are separated into vertical and horizontal components and supplied to the decentralised controller.

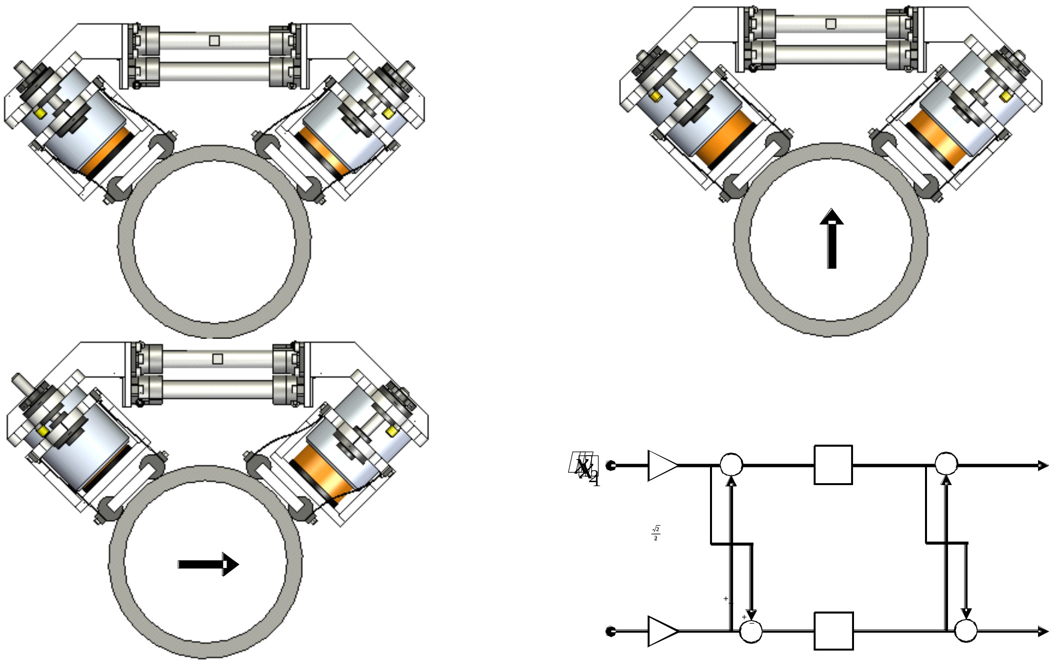

Figure 4: Mode of operation of the AVA with a decentralized controller

The efficiency of the AVA is shown in the next section on the basis of a mock-up built up within the scope of the European research project „SAFE PIPES“.

5 EXPERIMENTAL STUDIES

The mock-up is shown in Figure 5. The outer diameter of the steel pipe is 220 mm with a thickness of 17.5 mm. The pipe is welded on a quadratic base plate with a thickness of 56 mm. The base plate itself is connected to the floor with four bolts. The lower part of the pipe (in the area of the highest stresses) is strengthened to avoid damages in the case of high vibration amplitudes. A hanger acting as a vertical support is placed in the horizontal part of the pipe.

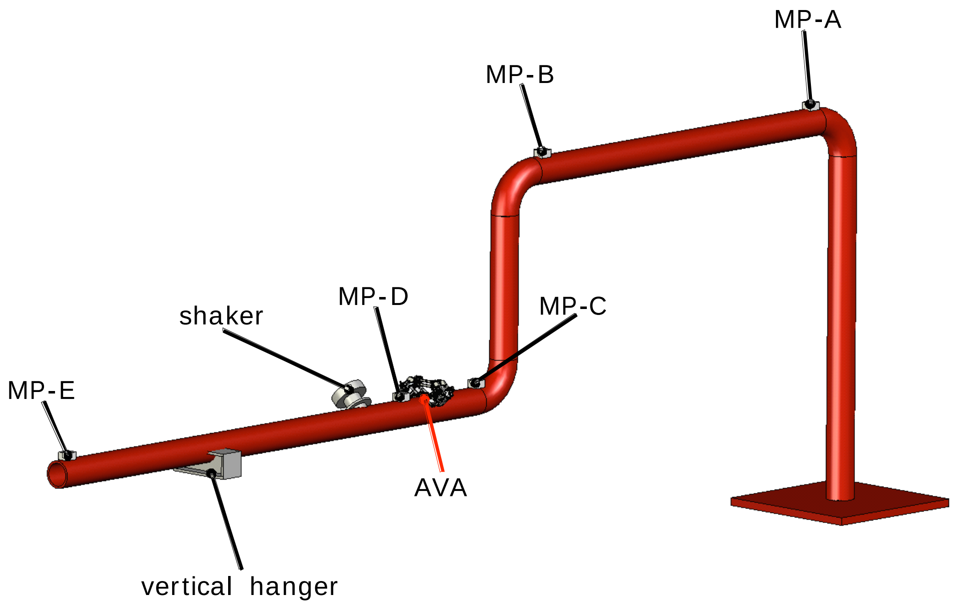

Figure 5: Mock-up with the positions of the accelerometers, the shaker and the AVA

The mock-up was excited with a shaker in the frequency range from 1 to 20 Hz and with snap back tests. The shaker is attached to the pipe at an angle of 45° in order to generate vibrations in the vertical as well as in the horizontal direction.For the snap back tests the pipe was deflected by 4.5 to 7 mm with a force of 5 to 10 kN. When the required force is reached the pipe is suddenly released and the pipe freely oscillates primarily in the vertical direction. For all tests accelerations in all three translational directions were measured for the positions MP-A to MP-E. The first two measured natural frequencies of the piping system in the vertical and horizontal direction are in the range of 5 to 10 Hz (see Table 2).

Table 2: Natural frequencies and masses

		natural frequency [Hz]			
	mass [kg]	vertical		horizontal	
mock-up	1300	5.3	8.2	6.1	10.3
AVA	16.25	3.2	-	3.2	-
	reaction mass: 13 kg				

The AVA is mounted next to the measurement point MP-D. Figure 6 shows the frequency response functions measured at the mock-up with and without the AVA for the three measurement points MP-A, MP-B und MP-E when excited with the shaker.

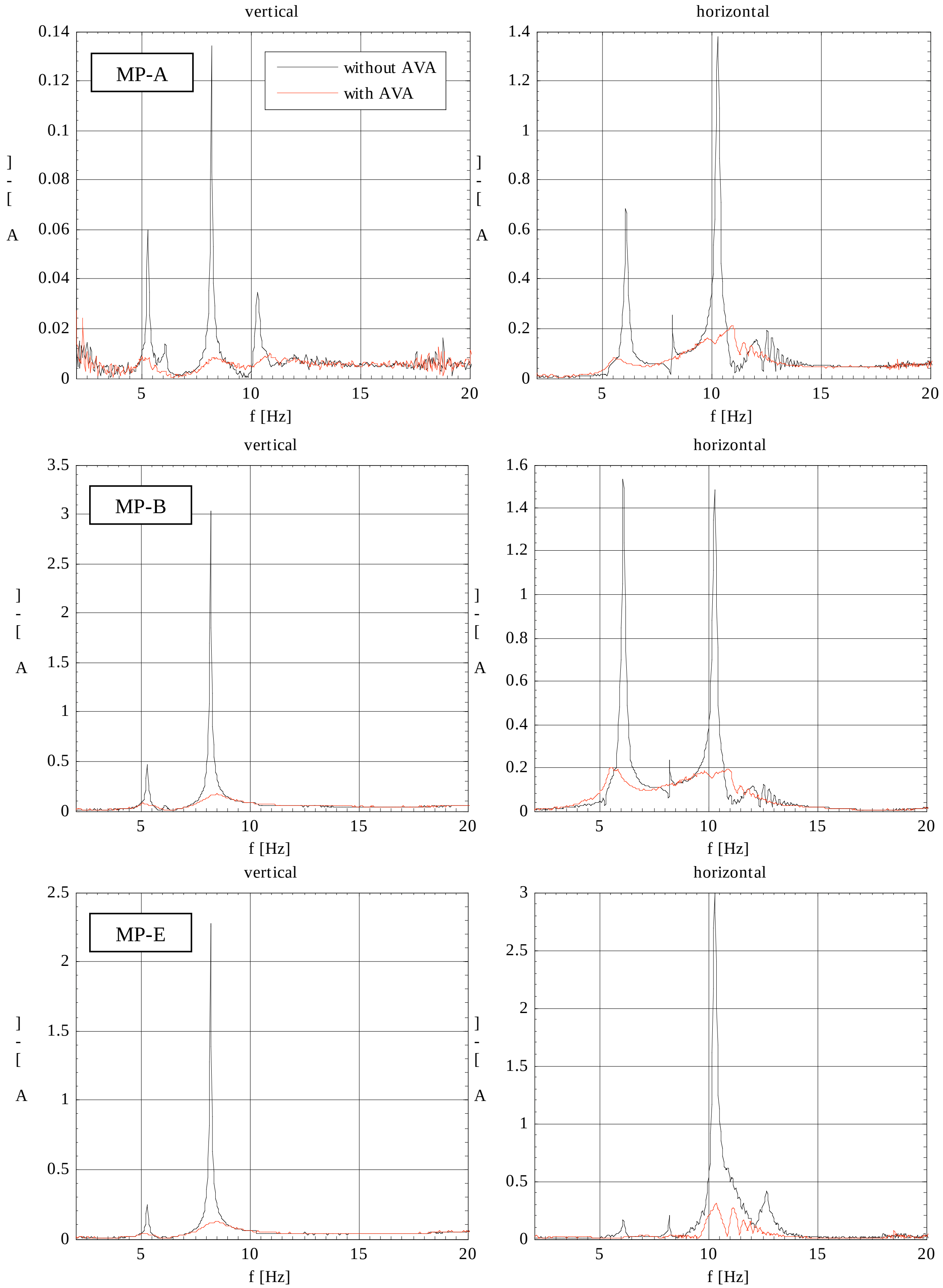

Figure 6: Frequency response function with / without AVA for an excitation with the shaker

The amplitudes are significantly reduced in the vertical as well as in the horizontal direction at all measurement points. A reduction factor of up to 30 is achieved in the range of the natural frequencies.

Figure 7 shows the behaviour of the piping system in the time and frequency domain for the vertical direction when it is excited by a snap back test.

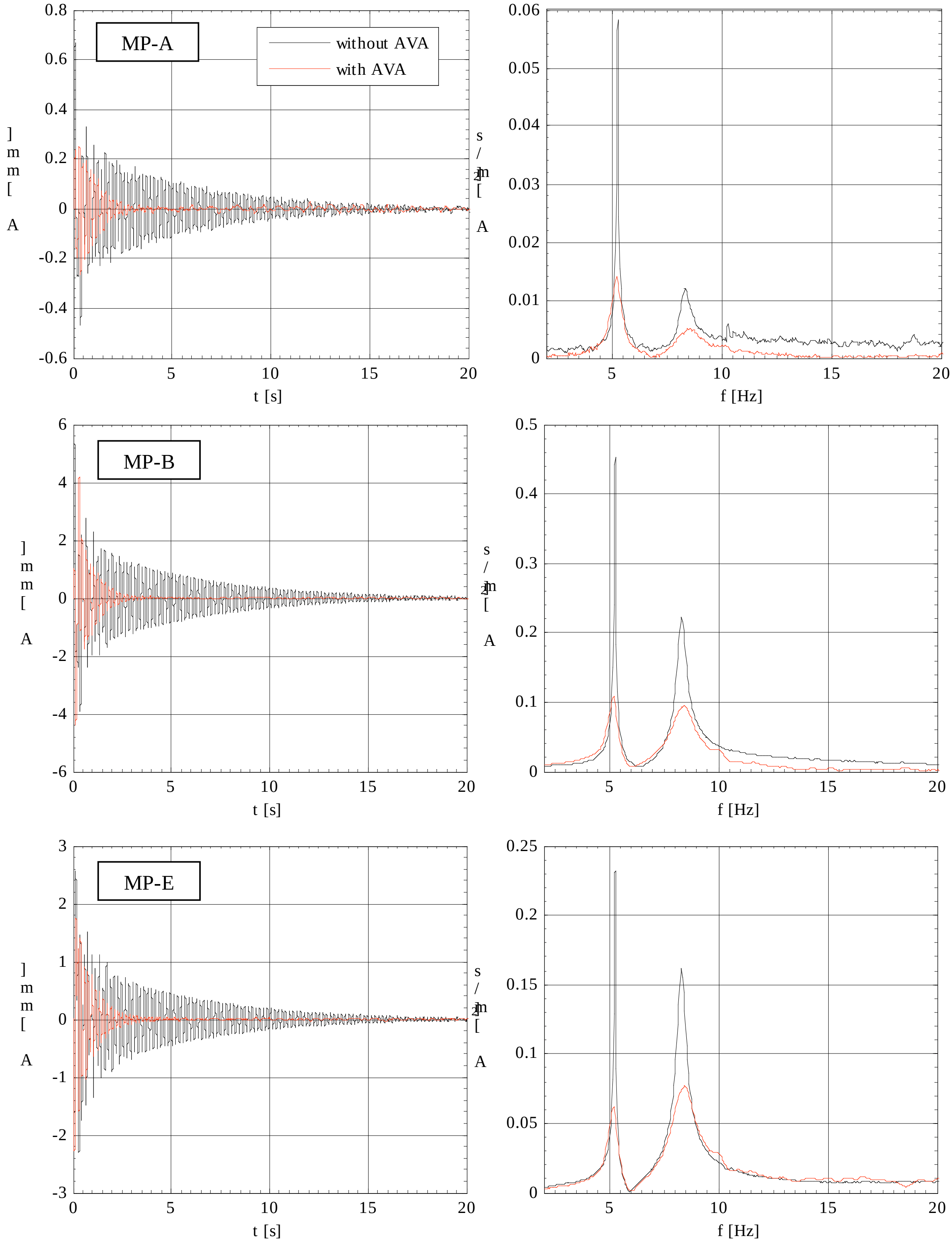

Figure 7: Behaviour of the piping system with / without AVA for snap back tests

For these tests as well a significant reduction of the vibrations can be achieved when using the AVA. In particular the long decay time of the vibrations without the active system is reduced.

6 CONCLUSION

The present contribution shows that active vibration control is suitable to efficiently reduce vibrations in piping systems.

A main advantage of the active vibration absorber AVA over conventional passive methods consists in the ability to reduce vibrations over a wide frequency range including several natural frequencies. Passive tuned mass dampers, for example, are tuned to a certain natural frequency. Therefore, they can only reduce vibrations of single modes.

Another advantage is that the control performance is not sensitive even to distinct parameter changes of the structure. In addition, the AVA does not need a fixed support as is required for viscoelastic dampers. This allows for an unproblematic implementation even after a plant started operation.

Currently the AVA is further developed within the scope of a European research project in order to implement the AVA in a real plant.

7 REFERENCES

Block, C.: Active control of human induced vibrations of long span civil engineering structures. Dissertation TU Darmstadt, Fortschritt-Berichte VDI, Reihe 11, Nr. 336, VDI Verlag, Düsseldorf , 2008

Engelhardt, J. et al.: Active Vibration Absorbers for Industrial Piping Systems. Adaptronic Congress 2007, 23-24 May, Goettingen Germany, 2007

M. Schalk, F.-O. Henkel: Rohrleitungsberechnungen für dynamische Lastfälle, 3rd international, Heft 9, 1990, Reprint in Handbuch Rohrleitungstechnik, 5. Auflage, Vulkan-Verlag

Safety Assessment and Lifetime Management of Industrial Piping Systems, SAFE PIPES, Sixth Framework Programme, Deliverable D07, NMP-CT-2005-013898, 2008

VDI Guideline 3842: Vibrations in piping systems, 2004