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Analysis of pipe vibration in an ultrasonic powder transportation system[☆]

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ABSTRACT

The transportation of dry fine powders is an emerging technologic task, as in biotechnology, pharmaceutical and coatings industry the particle sizes of processed powders get smaller and smaller. Fine powders are primarily defined by the fact that adhesive and cohesive forces outweigh the weight forces, leading to mostly unwanted agglomeration (clumping) and adhesion to surfaces. Thereby it gets more difficult to use conventional conveyor systems (e.g. pneumatic or vibratory conveyors) for transport. A rather new method for transporting these fine powders is based on ultrasonic vibrations, which are used to reduce friction between powder and substrate. Within this contribution an experimental set-up consisting of a pipe, a solenoid actuator for axial vibration and an annular piezoelectric actuator for the high frequency radial vibration of the pipe is described. Since amplitudes of the radial pipe vibration should be as large as possible to get high effects of friction reduction, the pipe is excited to vibrate in resonance. To determine the optimum excitation frequency and actuator position the vibration modes and resonance frequencies of the pipe are calculated and measured. Results are in good accordance.

1. Introduction

One very effective set-up for transporting adhesive and cohesive powders consists of a pipe, which vibrates harmoniously in axial direction at low frequency combined with a pulsed radial high frequency vibration [1,2]. The high frequency vibration accelerates the particles perpendicular to the surface of the pipe, which in average leads to lower normal and thereby smaller friction force [3].

Fig. 1 shows the experimental setup of the ultrasonic transportation system. A Voice-Coil Actuator excites the pipe to harmonic vibrations at low frequency. Only when the pipe is moving backward the effective friction between powder and pipe is reduced by switching on a piezoelectric actuator, causing a high frequency radial pipe vibration. The powder is therefore accelerated relatively strong when the pipe is moving forward and slightly decelerated when the pipe is moving backward. Operation at an axial vibration frequency of 50 Hz results in a nearly continuous powder transport. The powder velocity is adjustable by altering the vibration amplitudes and frequencies of low and high frequency excitation as well as the switching time of the high frequency vibration. This makes the device versatile for comparable high volume and fine dosing using one setup.

A simple model of the system has already been published in [1], see Fig. 2. The model is based on the assumption that the powder behaves like a rigid body with the mass m_P. The effective friction coefficient $\bar{\mu}(t)$ switches between two values depending on whether the ultrasonic pipe vibration is switched on or off. The high adhesion forces between powder and pipe are considered by a constant normal force $F_{adhesion}$. Viscous damping is considered by a velocity-dependent force with a parameter $2\delta = d/m_P$. As for fine powders it is very hard to determine parameters like friction, adhesion and viscous damping, the parameters were estimated by fitting model results to measurements. Fig. 3 shows the comparison of the mean powder velocity from measurements and corresponding simulations.

However, the comparison of model and measurements often led to unexpected results. After removal and installation of the pipe, deviating results of the mean powder velocity were obtained. This observation suggested that the high frequency vibration had changed after re-installation of the pipe.

Also, as parameters were only estimated by fitting model results to measurements, the physical relationships between the pipe vibration and friction reduction are not considered in this model. Therefore, estimated friction parameters can only be used for

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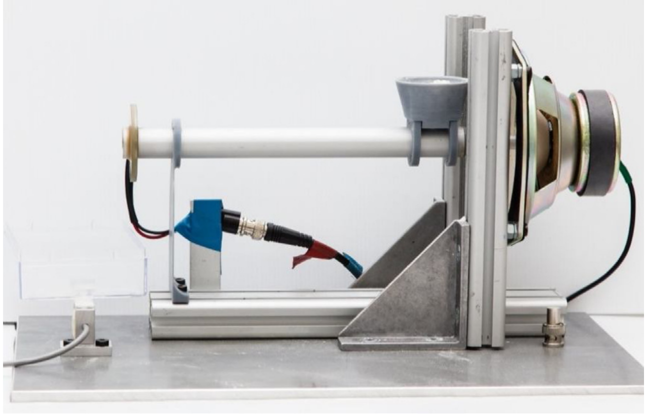

Fig. 1. Experimental setup of the ultrasonic powder transportation system.

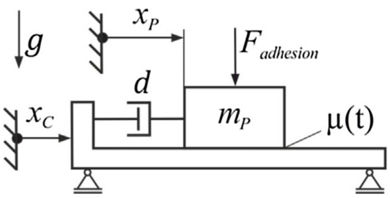

Fig. 2. Rigid body model of the powder transport system.

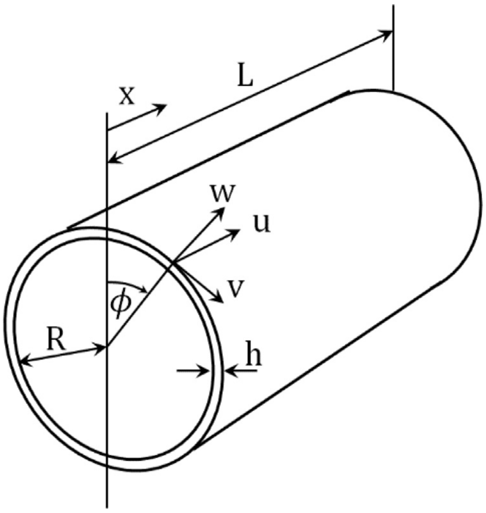

Fig. 4. Pipe coordinate systems.

unchanged excitation of the high-frequency vibration. A physical coefficient of friction can not be used since the coefficient of friction depends primarily on the radial vibration, which – at this high-frequency vibration – is not constant over the pipe length. Therefore, a locally resolved calculation is needed to analyze the friction reduction in detail.

Within this contribution the pipe vibration is analyzed using the finite element method and measurements.

2. Pipe vibration analysis

In the following, the pipe vibration of an aluminum pipe with an outer diameter $D_o = 20\,\text{mm}$, inner diameter $D_i = 18\,\text{mm}$ and length $L = 235\,\text{mm}$ is analyzed. The vibration is excited by a ring-shaped piezoelectric ceramic (PIC181) with dimensions $D_o = 50\,\text{mm}$, $D_i = 20\,\text{mm}$, $L = 5$ mm, which is bonded to the aluminum pipe using a two-components epoxy adhesive (Pattex Stabilit Express).

Fig. 4 shows the pipe coordinates used within the analysis. The variables u, v, w are displacements in axial, torsional and radial direction of each point of the pipe shell, whereas x and ϕ describe location coordinates. To get highest amplitudes, the pipe vibration is excited at a resonance frequency of about 35 kHz by applying a harmonic voltage on the piezoelectric actuator, which is adhered to the pipe at $x \approx 0$.

The pipe vibration was measured with different kinds of Laser-Doppler-Vibrometers. For measuring the radial vibration amplitude $\hat{w}$ along the longitudinal axis x of the pipe an out-of-plane Scanning Vibrometer was used whereas the longitudinal vibration $\hat{u}$ along the longitudinal axis x was measured with an in-plane Vibrometer. These measurements were done for different angles ϕ on the pipe surface.

Measurements of the radial pipe vibration are shown in Fig. 5. The vibration amplitudes differ for different angles ϕ. It is therefore not radial-symmetric. The maximum vibration amplitude was measured at $\phi = 20^\circ$. Also there is a phase difference of about 180° between the radial amplitude $\hat{w}(x)$ at $\phi = 0^\circ$ and $\phi = 90^\circ$, which suggests an elliptical deformation of the pipe profile.

An appropriate model of the pipe vibration was built using finite-element-simulation in Ansys. The pipe was meshed using a number of each 50 3D volume elements along the pipe coordinate x as well as the pipe coordinate ϕ. 2 elements were set on the pipe thickness h. Material parameters for the aluminum pipe were taken from literature (modulus of elasticity $E = 70GPa$, Poisonís ratio $\nu = 0.34$, density $\rho = 2700\,\text{kg/m}^3$). Using a harmonious analysis with excitation of the piezoelectric actuator at a frequency of 35 kHz the pipe vibration was analyzed.

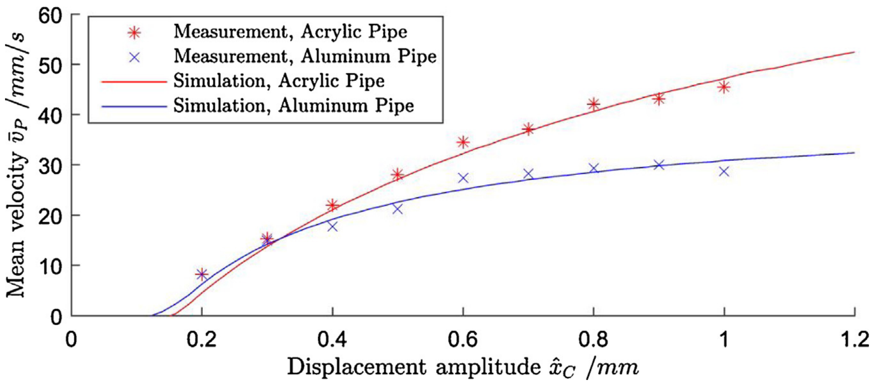

Fig. 3. Comparison of mean powder velocity from simulation and measurements. Measurements were done at an electrical power of $P_{Piezo} = 0.5\,\text{W}$ at the piezoelectric actuator. Simulation parameters were $\bar{\mu}_{US}/\bar{\mu}_0 = 0.63$, $\delta = 8\,\text{kg/s}$ for the acrylic pipe and $\bar{\mu}_{US}/\bar{\mu}_0 = 0.4$, $\delta = 30\,\text{kg/s}$ for the aluminum pipe.

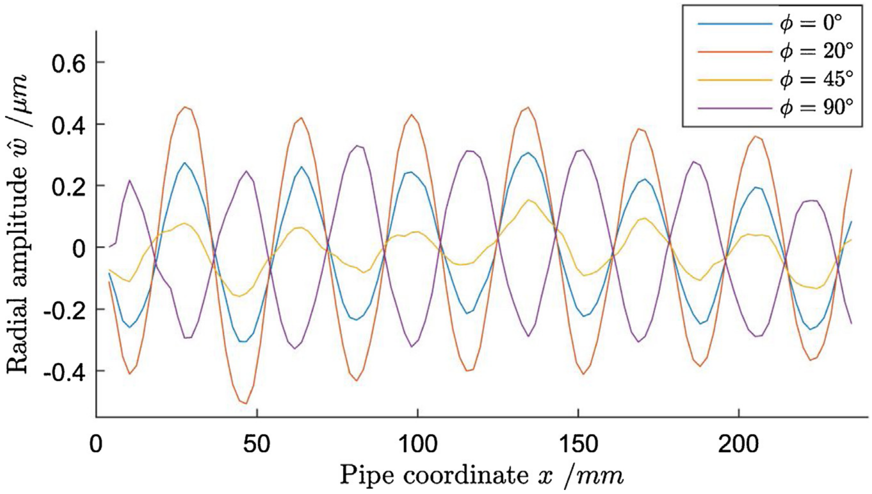

Fig. 5. Radial amplitude $\hat{\boldsymbol{w}}$ over the pipe coordinate $\boldsymbol{x}$ at different angles ϕ..

The result of the radial vibration from the finite-elements simulation is shown in Fig. 6. The vibration mode shape agrees with the measurements as the number of radial vibration peaks is the same. Also, it is verified, that the pipe profile is deformed elliptically. However, the amplitudes from the finite element simulation differed from measured amplitudes.

Since the values of vibration amplitudes could not be computed correctly in the simulation, the results of both the measurements and the simulation are normalized for better comparison. When comparing the normalized amplitudes from simulations and measurements, which are shown in Figs. 7–9, turns out that the simulation in general agrees with the measurements.

Fig. 7 shows the measured axial vibration amplitude $\hat{u}_{norm}$ over the pipe coordinate x from finite element analysis (FEA) and measurements. In FEA axial and radial vibration show a significant coupling as a waveform with same wavelength as the radial vibration occurs in the axial vibration. This can be attributed to the transverse strain.

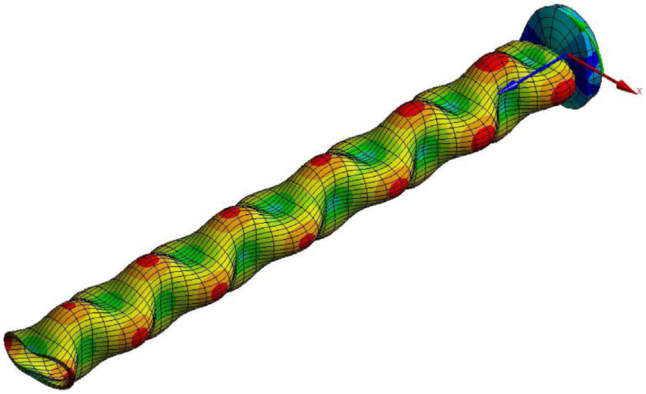

Fig. 6. 3D-view of radial vibration from the finite-element simulation at a frequency of 35 kHz.

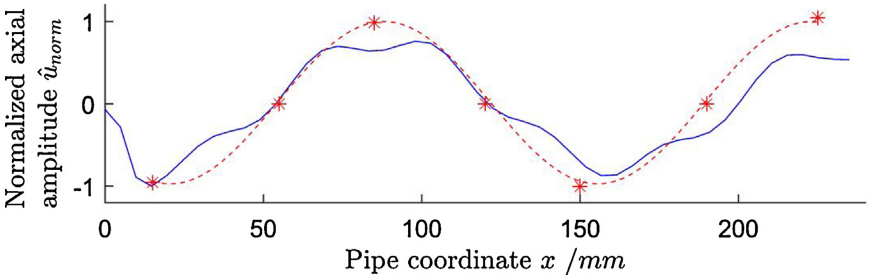

Fig. 7. Normalized axial amplitude of finite-element analysis (FEA) and measurement.

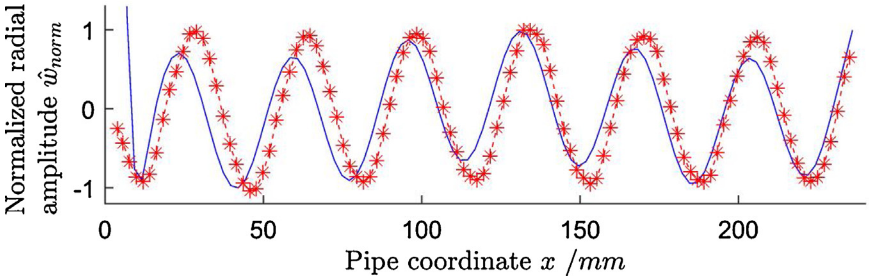

Fig. 8. Normalized radial amplitude $\hat{\boldsymbol{w}}_{\boldsymbol{norm}}$ of finite-element analysis (FEA) and measurement at $\phi = 20^{\circ}$.

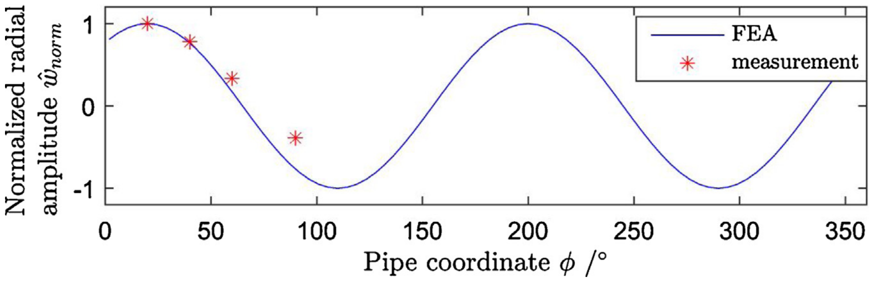

Fig. 9. Normalized radial amplitude $\hat{w}_{norm}$ of finite-element analysis (FEA) and measurement at $x = L$.

Measurements of the radial amplitude $\hat{w}_{norm}$ are shown in Figs. 8 and 9. The wavelength of $\hat{w}_{norm}(x)$ is much smaller than for the axial vibration. The waveforms of measurement and simulation show a good accordance although the coupling between axial and radial vibration can not be observed in the measurements. When looking at the radial amplitude $\hat{w}_{norm}(\phi)$ of the FEA, a nearly sinusoidal curve can be seen. The few measurements show a rather good accordance to that.

3. Conclusion and outlook

The results of the previous calculations and measurements can either be used for improving the construction of the powder transportation system and for improving the model of the powder transport. The pipe vibration is not constant over the angle ϕ and has paths along the longitudinal axis with optima of radial vibration amplitudes. Hence, when constructing such a dynamic system, it should be ensured by adjusting the angle ϕ that the maximum vibration amplitude is at the contact between powder and pipe surface.

Since the vibration amplitude is not uniform over the pipe coordinate x, the powder motion should be calculated locally for a large number of contact points. As the friction reduction is predominantly caused by the radial vibration of the pipe [1,2], it should be calculated depending on the actual position x_P in the pipe as well as the radial vibration amplitude $\hat{w}$ in this position. Hence, in the simulation the friction coefficient $\bar{\mu}\left(x_P, \hat{w}\right)$ should be determined using a look-up table.

Acknowledgements

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Biographies

Paul Dunst studied mechanical engineering at the University of Paderborn. During his studies he specialized in mechatronics, especially in sensor and actuator technology. He is currently research assistant in the group Piezoelectric and ultrasonic Systems and works on the development of a powder manipulation system.

Tobias Hemsel studied mechanical engineering at the University of Paderborn. After receiving his master degree in 1996, he was a research assistant in the field of piezoelectric ultrasonic motors at the Heinz Nixdorf Institute of the University of Paderborn. With the support of Prof. Dr.-Ing. Jörg Wallaschek he did his PhD with highest honour in 2001. Since then Dr. Hemsel holds a permanent position of an assistant professor within the chair of Mechatronics and Dynamics. His key aspects of activity in research and education are found in the area of sensors and actuators in mechatronic systems with a special focus on piezoelectric systems and ultrasound technology. During his scientific career Dr. Hemsel received twice the research award of the University of Paderborn, he published more than 100 papers. In 2009 he was awarded a 3 month scholarship as visiting professor at Tokyo University. Dr. Hemsel provides lectures on sensors and actuators, measurement technique, smart materials, and piezoelectric systems since about 15 years for undergraduate, master course and PhD students.

Walter Sextro studied mechanical engineering at the University of Hanover and Imperial College in London. After graduating, he designed and optimiseddrill strings for Baker Hughes Inteq research in Celle, Germany and Houston,Texas. He received his PhD from the University of Hanover in 1997. Subse-quently, he qualified as a professor in the field of mechanics and published hishabilitation thesis. In 2004 he was appointed as a professor at the institute of mechanics and gear trains at the Technical University of Graz, Austria. Since 2009 he leads the chair for mechatronics and dynamics at the University of Paderborn.