A MULTI-DOF ACTIVE VIBRATION ISOLATION SETUP FOR A CORIOLIS MASS FLOW RATE METER

K. Staman[1], L. Van de Ridder[1], W.B.J. Hakvoort[1,2], D. M. Brouwer[1], J. van Dijk[1]
[1]Laboratory of Mechanical Automation and Mechatronics
University of Twente, Enschede, The Netherlands
[2]DEMCON Advanced Mechatronics, Enschede, The Netherlands

INTRODUCTION

A Coriolis mass flow meter (CMFM) for small mass flows, as presented by Mehendale [1,2], is limited from measuring more accurately due to vibration disturbances introduced through the device's frame as shown by Van de Ridder [3]. Considerable improvement in measurement performance is expected from the application of passive and active vibration isolation strategies. Using feedback control, an attenuation of 42 dB of translational disturbances is expected [4]. Where the field quantity definition of dB used, i.e. $value_{dB} = 20 \log_{10}(\text{ ratio}))$. The component of the CMFM that needs to be isolated from disturbances is the tube window (see figure 1). This fluid-conveying tube is actuated in oscillation around the actuation-axis. A fluid flow in this oscillatory rotating tube results in a Coriolis force induced motion around the Coriolis-axis in the order of sub nanometres for the lowest flow range. A mixed signal of the actuation- and Coriolis-deflection is measured at the sensing locations, where the phase difference of both sensor measurements is linearly dependent on the mass flow, as derived by Mehendale [1].

This work extends the mentioned research with the design, modelling and validation of an active vibration isolation system that serves as a proof of principle for multiple vibration isolation strategies. The designed system is experimentally validated and active vibration isolation by means of a feedback control scheme is tested. In this paper, the design of the system is discussed, followed by the properties obtained from a dynamic model. An attenuation of the main source of disturbance by 50 dB is expected theoretically. With implementation of the same feedback scheme, an attenuation of 48 dB is obtained experimentally. The work concludes with a discussion of the presented system and results.

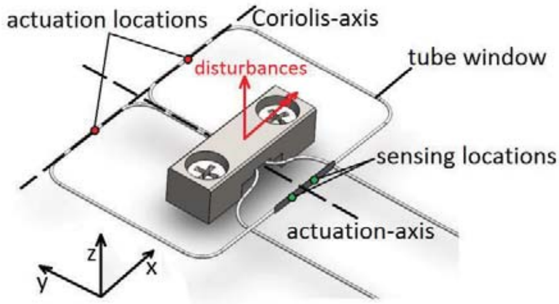

FIGURE 1. Solidworks® model of the fluid-conveying tube window and its socket.

CONCEPT DESIGN

Disturbances introduced through the device's frame can cause a measurement error. This occurs when the tube experiences motion similar to the Coriolis-deflection at the sensor locations. When this motion is around the measurement frequency of 170 Hz it will be indistinguishable from fluid flow and result in a measurement error. The two main contributors to this error are a translation perpendicular to the tube plane (z-direction) and a rotation around any axis parallel to the Coriolis-axis (x-axis) (see figure 1) as derived by Van de Ridder [5]. So the active vibration isolation system should isolate the tube window and its actuation and sensing components, together called the measurement stage, in both these directions from frame disturbances. Furthermore, the setup will be used to test if the tube window can be actuated by rotation of the tube's socket about the actuation-axis (y-axis),

Isolation from frame disturbances by means of feedback control is achieved in two steps. First, the measurement stage is suspended. The suspension provides passive vibration isolation above its resonance frequency (called suspension frequency in the following). The term suspension is used for a support with a designed stiffness significant to the involved dynamics. A suspension frequency of 30 Hz was determined by internal research as a lower bound for sufficient mechanical properties when the device is unpowered and the control is deactivated. Second, a controller is designed in conjunction with sensors and actuators to close a feedback loop, providing active vibration isolation. This loop has the advantage that the suspension frequency can be lowered virtually, without compromising the low frequency behaviour of the system. Acceleration feedback is used with proportional and integral (PI) control action to add virtual mass and skyhook damping, respectively, to the system [6]. This concept is illustrated for a single dimension in figure 2, where the spring with stiffness k forms the suspension. The measurement stage is represented by the mass, m. Its acceleration, a_1, is measured and used by a feedback controller, C, to produce an actuation force, F, that reduces the influence of the frame disturbance acceleration, a_0, on the measurement stage.

The designed vibration isolation system is shown in figure 3. It consists of the measurement stage components (with all but the tube window omitted for clarity) mounted on a measurement frame. This measurement frame is mounted on a three degrees of freedom suspension (see figure 4), constraining three dimensions, while the other three dimensions are controlled by three voice coil actuators and three acceleration sensors. A degree of freedom (DOF) is an independent absolute or relative motion within the system describing relevant system motion. The three DOF not constraint by the suspension are the z-direction translation and rotations around the x- and y-axes. These are the first three eigenmodes of the suspension. All suspension frequencies are designed close to 30 Hz to keep the suspension design symmetric. The three actuators and sensors are used to close the feedback loops in the z-translation and x-axis rotation to provide active vibration isolation. The third DOF (the y-axis rotation) is used for the aforementioned measurement actuation.

With this setup it is also possible to investigate if it is sufficient to suspend and actively isolate only one direction, resulting in a more efficient solution. By attaching one or both optional constraints at the sides of the setup (see figure 3), the measurement frame can be constraint in its rotation about either or both the x- and y-axis in the suspension plane, reducing the system's number of DOF. The optimum configuration being a research objective. It might be possible that one of the frame disturbances, a translation in y-direction, can cause significant unwanted motion in the tube window. Therefore an interchangeable four DOF suspension (see figure 4) is created to convert the setup into a three DOF active isolation setup with one passively suspended DOF.

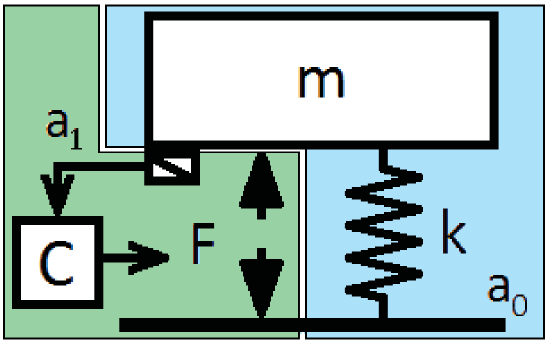

FIGURE 2. Dynamic model of suspended measurement stage (right block) with active feedback vibration isolation (left block).

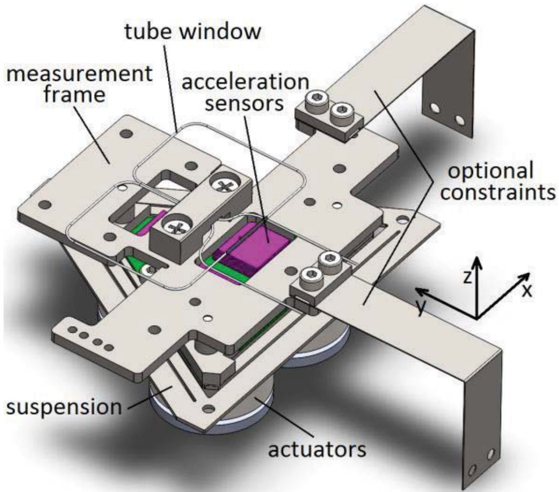

FIGURE 3. Solidworks® model of the active vibration isolation setup with optional constraints and simplified measurement stage.

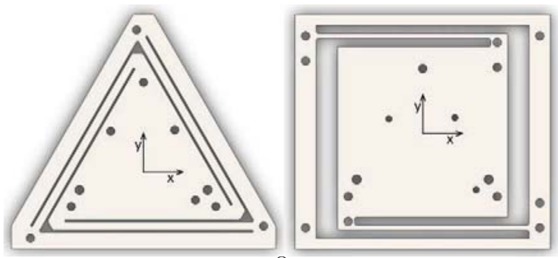
FIGURE 4. Solidworks® models of the 3 DOF suspension (left) and alternative 4 DOF suspension (right).

MODELING

The entire setup has been modelled using the non-linear finite element flexible multibody software package SPACAR [7]. This model (see figure 5) is used to obtain equations of motion for suspension design and transfer functions for control synthesis.

MODEL RESULTS

The three DOF have a suspension frequency between 30 and 33 Hz. The alternative four DOF suspension has an additional DOF in the y-direction, which is left passively suspended with a frequency of 34 Hz. The optional constraints are omitted in the rest of this work and will be used for future research.

Of primary interest is the transfer function from six (three orthogonal translations and three orthogonal rotations) frame acceleration disturbances to the tube window Coriolis-displacement at the sensing locations. This is called the transmissibility and it has six inputs and one output, thus it is a multiple-input single-output (MISO) system. As mentioned, the tube window displacement is mainly influenced by two measurement stage directions. Without taking the tube dynamics into account, the transmissibility of interest is thus the multiple-input multiple-output (MIMO) transfer function from the same six inputs to two stage acceleration measurements. These are the acceleration signal of the top sensor, a1,I and the mean of the bottom sensors, a1,II, (see figure 5). These signals are a combination of the z-translation and x-axis rotation. From the new transmissibility only three frame accelerations appear to be significant due to symmetry, being a translational disturbance in z-direction, a translation in y-direction and a rotation around the x-axis. The other directions already have an influence below -100 dB. So only the three main transmissibility directions are considered when evaluating the performance of the setup.

For control synthesis the transfer function from actuator forces to measurement stage accelerations is required. This is called the plant and this transfer function has three inputs and three outputs. This transfer function has non-zero off-diagonal elements and thus coupling of every input to every output, where for control purpose it is desirable to have a diagonal matrix.

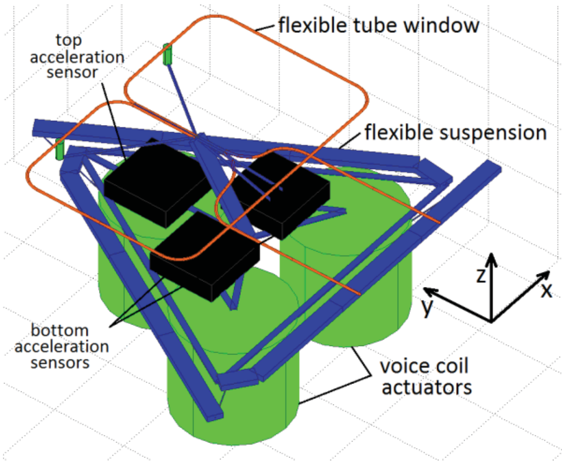

FIGURE 5. SPACAR model of the setup without optional constraints.

The plant transfer function can be decoupled by using the eigenvectors obtained from an eigendecomposition according to Owens [8]. This results in a predominantly diagonal transfer function matrix where influence of the off-diagonal elements is about 30 to 40 dB lower across the frequency region of interest. Three independent SISO plants remain for which three SISO controllers can be designed. The third decoupled plant direction is the transfer function from actuator input to sensor output that corresponds to the third DOF of the suspension, a rotation about the y-axis. In this direction no feedback loop will be closed as this DOF was only included to test tube window actuation. On the other two directions acceleration feedback will be applied. The applied SISO controllers consist of the aforementioned PI control together with three loop shaping filters. These add to the performance and stability of the feedback system. The additional filters used are a second order high-pass filter to limit actuator saturation, a second order low-pass filter to increase attenuation at the actuation frequency of 170 Hz and an zero at the crossover frequency of 300 Hz to increase stability. The parameters of the PI controller are calculated from performance requirements (lowering of the suspension frequency and adding damping) and the respective SISO plant transfer functions.

The three main transmissibility directions are shown in figure 6 for the reference, passive and active systems. The reference system is an unsuspended measurement stage, in which a y-translation has no influence on z-translation or x-axis rotation of the measurement stage. It can be seen that the attenuation at the measurement

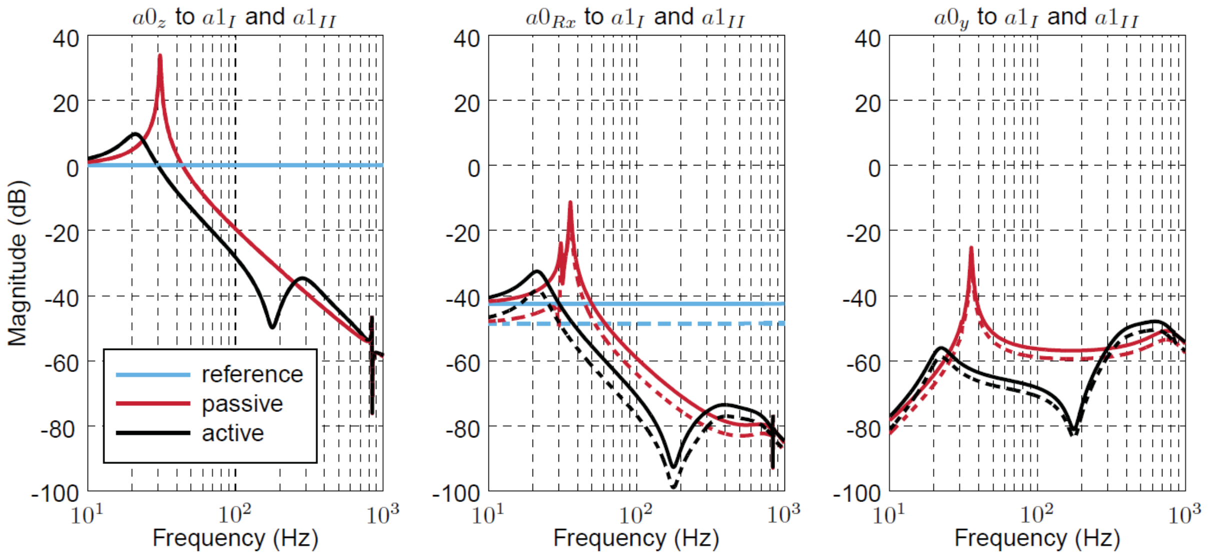

FIGURE 6. Bode magnitude plot of the main transmissibility directions (z-translation $a0_z$, -axis rotation $a0_{Rx}$ and y-translation, $a0_y$) to two measurement stage accelerations $a1_I$ (solid) and $a1_{II}$ (dashed).

frequency of 170 Hz for two of the three main directions is 50 dB with respect to the reference system. The third transmissibility shows that the vibration isolation system introduces the influence of a translational disturbance in y-direction at -80 dB.

EXPERIMENTAL RESULTS

The experimental setup consists of the suspended measurement stage, resembling the modelled setup of figure 3 when the optional constraints are omitted and the measurement stage components are added. This is mounted on a six DOF shaker platform used to supply external vibrations. An identification of the dynamics of this setup is performed over the frequency region between 5 and 2000 Hz by a modal analysis based on the method outlined by Tjepkema [9]. This results in three estimated decoupled SISO plants (shown in figure 7), similar to the plant obtained from the model and decoupling strategy. The main resonance frequencies in these transfer functions correspond to the three suspension frequencies, realized in the region between 25 and 34 Hz. An additional real pole is identified in each transfer function, originating from the (unmodelled) actuator induction. An internal mechanical mode is estimated at 282 Hz in one of the decoupled directions and in another directions an internal mode at 845 Hz is identified. Finally phase lag resulting from accelerometer signal filtering and digitization can be observed.

Based on the identified SISO plants the PI parameters are adjusted. The actuator induction poles are compensated by adding an extra zero and an additional notch filter is added at 845 Hz to stabilize the identified high frequency dynamics of the system.

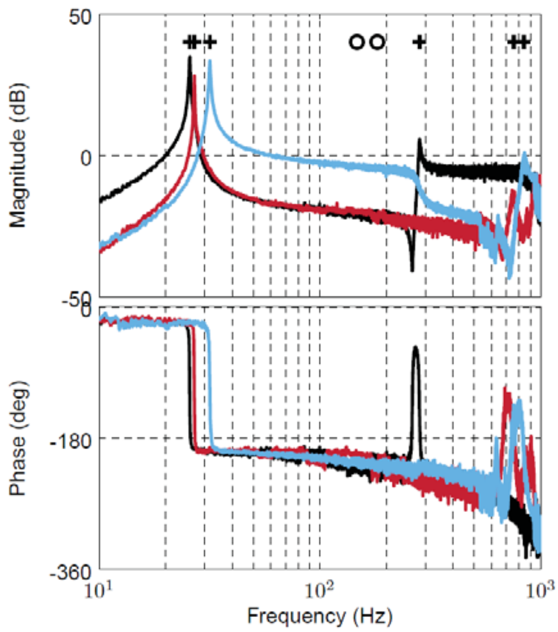

FIGURE 7. Identified plant (from actuator forces to three stage accelerations) with estimated complex pole pairs, +, and real poles, o.

As an initial validation only a disturbance in z-direction (the main disturbance direction as seen from figure 6) is introduced to the system. Both feedback loops are closed and performance is measured by evaluation of the power spectral density (PSD) plot of the measured stage acceleration signals (see figure 8). This PSD plot should resemble the modelled main transmissibility direction in the frequency region between 5 and 500 Hz as the input disturbance frequency content is uniform in this region.

The reference data is obtained by using the voice coil actuators to present a disturbance with an amplitude of 1×10^{-4} $(m/s^2)^2/Hz$ to the measurement stage. In the PSD of this signal the suspension frequency is seen and the identified internal mode is present at the expected frequency of 285 Hz. The passive and active accelerations show the expected behaviour within the frequency region of interest and the attenuation of frame vibrations in z-direction for the passive and active systems is 35 and 48 dB in the relevant frequency region around the actuation frequency, respectively.

When the influence on mass flow measurement is evaluated, a time domain signal, as shown in figure 9 is obtained. No real mass flow is provided to the device, so the measurement is a flow error due to the vibration disturbance, scaled to arbitrary units. The attenuation of the flow error obtained for the active system is lower than the expected 48 dB, due to an unexpected high measurement sensor noise. Larger disturbance amplitudes are provided to improve the signal to noise ratio, as seen from table 1, but the attenuation of 48 dB can still not be validated because the highest amplitude of 4×10^{4} $(m/s^2)^2/Hz$ is the upper limit of the used test setup.

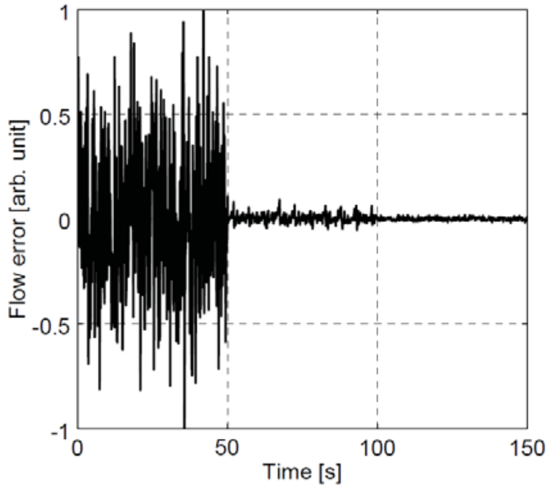

FIGURE 9. Time domain flow error with z-translation disturbance. Reference (0-50 s), passive (50-100 s), active (100-150 s).

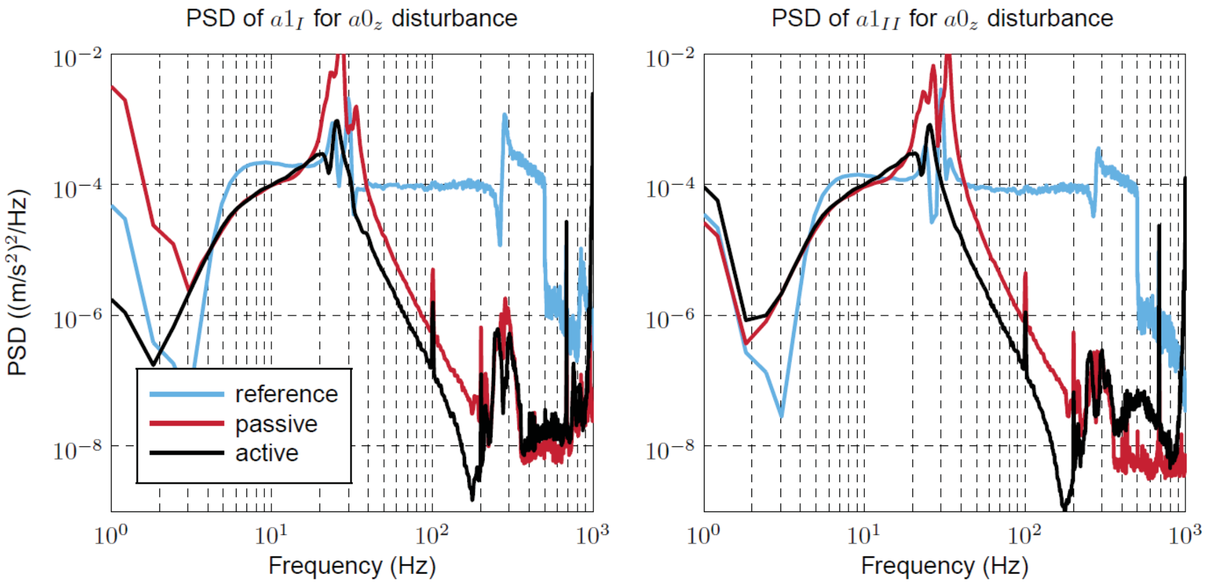

FIGURE 8. PSD of the two measurement stage accelerations for a translational disturbance (z-direction).

TABLE 1. Experimental RMS flow error in arbitrary units and the attenuation relative to the reference.

Disturbance	Reference		Passive suspension		Active feedback	
$(m/s^2)^2/Hz$	units	dB	units	dB	units	dB
1e-5	0.0551	0	0.0067	-18.53	0.0079	-17.44
1e-4	0.1741	0	0.0085	-26.28	0.0068	-28.44
4e-4	0.3515	0	0.0211	-24.41	0.0070	-34.03

DISCUSSION

To further identify the performance of this feedback system the influence of more disturbance directions (ultimately all six directions) will be tested. Also the optimal configuration of additionally constraint, passively suspended and actively isolated directions can be determined. The identification shows an internal mode at 282 Hz currently not seen in the dynamic model. Its origin should be identified, possibly modelled and its influence on control stability and performance should be further investigated. Furthermore robustification of the control for higher order dynamics of the setup should be investigated, for stability issues are encountered at higher specifications. The mentioned phase lag introduced by the digitization of the acceleration sensor signals has a negative influence on the system's stability and performance. It should be quantified and accounted for when determining obtainable specifications. Finally, in order to validate the attenuation of the influence of vibrations on the mass flow measurement a lower noise of the optical sensors used to measure tube displacement and thus mass flow is desired. The validation is currently limited to this noise level for low amplitude disturbances.

CONCLUSION

A valid conceptual design for the suspension and active vibration isolation of the Coriolis mass flow measurement stage in z-direction translation and x-axis rotation is created and three main disturbance directions are identified, being a translation in z- and y-direction and a rotation around the x-axis.

With a dynamic model of the presented setup design it has been shown that the theoretical attenuation of frame disturbances in the main directions is 50 dB.

Experimental validation of the active vibration isolation with feedback control shows an attenuation of about 48 dB when a disturbance is provided in the largest of the main disturbance directions, the z-direction translation.

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