VIBRATION ISOLATION BY COMPLIANT SENSOR MOUNTING APPLIED TO A CORIOLIS MASS-FLOW METER

L. (Bert) van de Ridder  
Mechanical Automation Laboratory  
Faculty of Engineering Technology  
University of Twente  
P.O. Box 217, 7500AE, Enschede  
The Netherlands  
Email: l.vanderidder@utwente.nl

Wouter B.J. Hakvoort  
Demcon Advanced Mechatronics  
Institutenweg 25, 7521 PH, Enschede  
The Netherlands  
Email: wouter.hakvoort@demcon.nl

Johannes van Dijk  
Mechanical Automation Laboratory  
Faculty of Engineering Technology  
University of Twente  
P.O. Box 217, 7500AE, Enschede  
The Netherlands  
Email: j.vandijk@utwente.nl

ABSTRACT

In this paper a vibration isolated design of the Coriolis Mass-Flow Meter (CMFM) is proposed, by introducing a compliant connection between the casing and the tube displacement sensors with the intention to obtain a relative displacement measurement of the fluid conveying tube, dependent on the tube actuation and mass-flow, but independent of casing excitations.

Analyses are focussed on changing the transfer function of support excitations to the relative displacement measurement. The influence of external vibrations on a compliant sensor element and the tube are made equal by tuning the resonance frequency and damping of the compliant sensor element and therefore the influence on the relative displacement measurement is minimised. Based on simulation results, a prototype is built and validated.

The validated design shows a 20dB reduction of the influence of external vibrations on the mass-flow measurement value of a CMFM, without affecting the sensitivity for mass-flow.

Keywords: Coriolis Mass-Flow Meter, Floor vibrations, Internal mode, Transfer function, Compliant mechanism.

INTRODUCTION

Vibration isolation is extremely important in high precision machines for surface treatments (e.g. lithography machines) or measurements (e.g. scanning electron microscopes) that should be accurate to nanometre level. Vibration isolation can be achieved with passive isolators. Passive isolation consists of several stages of mass-spring-damper systems between the floor and the casing of a machine [1], the parameters are adjusted to achieve high-frequency attenuation, which is appropriate for many applications. The better the vibration isolation system the better the decoupling of the internal measurement system from any environmental disturbances. In this paper a passive vibration isolated measurement system applied to a Coriolis Mass-Flow Meter (CMFM) is presented.

A CMFM [2] is an active device based on the Coriolis force principle for direct mass-flow measurements independently of fluid properties. The CMFM contains a fluid conveying tube, which is actuated to oscillate in resonance with a low amplitude. A fluid flow in the vibrating tube induces Coriolis forces, proportional to the mass-flow, which affect the mode-shape of the actuation mode. Measuring the tube displacements allows measuring the mass-flow. Besides an effect of the mass-flow on the mode-shape, support excitations can introduce motions that cannot be distinguished from the Coriolis force induced motion [3, 4], thus introducing a measurement error. The influence of floor vibrations on the measurement value can be estimated quantitatively as shown in [5].

A possible solution is to find a balancing mechanism for flow meters, which allows accurate measurements for various process conditions and changing fluid densities [2]. Because there is only attenuation needed in relative small range around the actuation frequency [5], passive vibration isolation solutions can be applied to accurate measure the displacements of the internal actuation modeshape.

In this paper a novel design of a CMFM is proposed, by introducing a compliant connection between the casing and the tube displacement sensors with the objective to obtain a relative displacement measurement of the tube, dependent on the tube ac-
tuation and mass-flow, though independent of casing excitations. The design is analysed to find out how the transfer function of the support excitation on the relative displacement measurement can be minimised. The influence of external vibrations on the compliant sensor element and the tube are made equal by tuning the resonance frequency and damping of the compliant sensor element and therefore the influence on the relative displacement measurement is minimised. Based on simulation results, a prototype is built and validated.

In the first following section the performance criterion is given for the level of vibration isolation. Secondly, the transmissibility of floor vibrations to an internal deformation is derived, based on a mass-spring model for a reference instrument and for the new design. In the third section a dimensional design is presented, based on the concept presented in section two. The design is validated experimentally in the fourth section. Results are discussed in section five and the conclusions of this paper are presented in the sixth section.

**PERFORMANCE CRITERIA**

Before we can look to vibration isolation a performance criterion is needed. In this section a definition is given, taking into account how external vibrations affect the mass-flow measurement of a Coriolis Mass-Flow Meter (CMFM).

A functional model of a CMFM is given in Fig. 1. Of this instrument a flexible multibody model [5] is made to derive the influence of external vibrations on the measurement value, using [7][8]. A representation of the model is given in Fig. 2. The model consists of a rigid casing and a flexible tube-window, conveying the fluid flow, which is actuated by two Lorentz actuators act\(_1\) and act\(_2\). The displacement of the tube-window is measured by two optical displacements sensors s\(_1\) and s\(_2\) [9]. On the casing an input vector a\(_0\), consisting of three translation and three rotational floor movements, is imposed.

A Lorentz actuator is used to oscillate the tube-window around the \(\theta_{\text{twist}}\)-axis in resonance by the frequency \(\omega_{\text{act}}\). The actuation displacement is defined as a displacement due to a rotation around \(\theta_{\text{twist}}\)-axis on a certain distance. In the model the measured actuation displacement is the difference between the two sensor signals, located on equal distance of the rotation axis:

\[
y_{\text{act}} = \frac{1}{2}(s_1 - s_2)
\]  

(1)

Due to a rotating reference frame, the tube, and a moving mass, the fluid, there is a Coriolis force. This force is acting on the tube-window and is proportional to the actuation velocity and the mass-flow \(\Phi\) through the instrument:

\[
F_{\text{cor}} \propto y_{\text{act}} \times \dot{\Phi}
\]  

(2)

this force \(F_{\text{cor}}\) results in a rotation of the tube-window around the \(\theta_{\text{swing}}\)-axis (see Fig. 2). A rotation around this axis results in a displacement on the location of the sensors. This measured displacement is defined as a Coriolis displacement:

\[
y_{\text{cor}} = \frac{1}{2}(s_1 + s_2)
\]  

(3)

The Coriolis displacement, due to a fluid flow, is a harmonic with the same frequency \(\omega_{\text{act}}\) as the actuation displacement, though 90° out of phase. This is due to the velocity dependency of the force, expressed by Eq. 2. A phase difference between the two harmonic sensor signals \(s_1\) and \(s_2\), which are shifted 180 deg nominally, can be approximated by:

\[
\Delta \phi \approx 2 \left| \frac{s_1 + s_2}{s_1 - s_2} \right| = 2 \left| \frac{y_{\text{cor}}}{y_{\text{act}}} \right|
\]  

(4)
The actuation displacement $y_{act}$ is controlled with an internal actuator to be constant and the Coriolis displacement $y_{cor}$ is due to a mass-flow through the instrument, resulting in a phase difference $\Delta \phi$ which is proportional to the mass-flow $\Phi$. Therefore the phase difference is the measurement value of a CMFM. A Coriolis displacement $y_{cor}$ not related to a mass-flow through the instrument, but as a result of casing excitations can result in a measurement error.

Thus the mass-flow measurement value is obtained by a phase difference between two harmonics, which are the tube displacements measured by two sensors. The frequency of the harmonic is known, the actuation frequency $\omega_{act}$, whereby the width is dependent on the required response time of the measurement value due to a flow rate change.

In this paper the performance criterion is defined as: minimisation of the transmissibility $T_{y_{cor},a_0}$ of external vibrations $a_0$ to an internal deformation $y_{cor}$ around the actuation frequency $\omega_{act}$ without affecting the transmissibility $T_{y_{cor},F_{cor}}$ of a Coriolis force $F_{cor}$ to the internal deformation $y_{cor}$. To increase the vibration isolation the transmissibility $T_{y_{cor},a_0}$ needs to be minimised.

CONCEPTUAL DESIGN

In this section a solution is proposed to minimise the transmissibility $T_{y_{cor},a_0}$. First it is derived for the reference system and secondly for the new design.

Model results indicate one dominant direction for the mass flow measurement, a translation in the direction out of the tube-window plane. Therefore the two effects are modelled in a simple and elegant manner by a simple mass-spring model in Fig. 3. The tube parameters are modelled with the modal mass $m_2$, damping $d_2$ and stiffness $k_2$ and there are two inputs, a casing excitation $a_0$ and a Coriolis force $F_{cor}$ and one output $y_{cor}$, which is a relative measurement between the mass $m_2$ and the casing. The modal model reduction of the system is presented in [5].

According to this model (Fig. 3) of the reference system, the relative displacement is equal to:

$$y_{cor} = \frac{1}{m_2 \omega^2 + d_2 + k_2} F_{cor} + \frac{1}{m_2 \omega^2 + d_2 + k_2} a_0$$

$$= T_{y_{cor},F_{cor}} F_{cor} + T_{y_{cor},a_0} a_0$$

(5)

This expression shows that the displacement is indeed dependent on the Coriolis Force and casing excitations $a_0$. Where the influence is dependent on the magnitude of the disturbance, the frequency and the physical parameters of the system. Those physical parameters cannot be changed without changing the sensitivity for a mass-flow.

To increase vibration isolation, without changing the mass-flow sensitivity, it is proposed to use a compliantly mounted sensor. The model is shown in Fig. 4. The relative displacement measurement is no longer between the casing $m_0$ and the internal mass $m_2$, but between an extra mass $m_1$ and again $m_2$. Where there is a certain stiffness $k_1$ and damping $d_1$ between the new mass and the casing. In this model the relative displacement is equal to:

$$y_{cor}' = \frac{1}{m_2 \omega^2 + d_2 + k_2} F_{cor} + \frac{1}{m_2 \omega^2 + d_2 + k_2} a_0$$

$$= T_{y_{cor},F_{cor}} F_{cor} + T_{y_{cor},a_0} a_0$$

(6)

Where the sensitivity for flow is equal, but the transmissibility $T_{y_{cor},a_0}'$ is dependent on the newly introduced parameters. By choosing the mass, damping and stiffness of the compliantly mounted sensor the influence of the casing excitations $a_0$ can be minimised.

An optimal result can be achieved when the following conditions are met:

$$d_1 = \frac{m_1}{m_2} d_2, \quad k_1 = \frac{m_1}{m_2} k_2$$

(7)

This is achieved when the resonance frequency $\omega = \sqrt{\frac{k}{m}}$ and damping ratio $\zeta = \frac{d}{2 \sqrt{km}}$ are equal for the internal mode of the tube-window and the compliant mounted sensor.

FIGURE 7. TRANSFER FUNCTION $a_0$ TO $y_{cor}$ - MODEL CONCEPTUAL DESIGN

The new transmissibility is compared to the transmissibility of the reference system, in Fig. 7 for the parameters given in Table 1. The transmissibility is zero when the parameters are exactly tuned. This is hardly achievable in reality, thus the transmissibility is given for a 20% increase in the damping and stiffness. The result shows that the influence is not zero anymore, though in the region of interest there is still attenuation achievable. This is an region of 5 Hz around the actuation frequency.
\[ \omega_{\text{act}} = 170 \, \text{Hz}, \] as explained in the performance criteria section. Thus the conceptual design indeed results in vibration isolation.

**DIMENSIONAL DESIGN**

Based on the conceptional compliant sensor mounting design an dimensional design is made to prove the principle.

The additional flexibility (Fig. 4) is applied to the CMFM model (Fig. 2) resulting in a model with a compliantly connection between the sensors and the casing, which is depicted in Fig. 5. The compliant mounted sensor (CMS) is presented in Fig. 6. A functional model of a CMFM (Fig. 1) is extended with a straight guidance mechanism. The five compliant wire springs provide an exactly constraint configuration with only one remaining degree of freedom, which is out of plane of the tube-window. This results in an extra degree of freedom between the casing and the printed circuit board (PCB) with the optical sensors.

The transmissibility of the CMFM model for external vibrations \( a_0 \) to the Coriolis displacement is depicted in Fig. 8 for the different configurations. A perfect match of the damping ratio and the resonance frequency no longer results in perfect vibration isolation because of the higher order dynamics of the tube-window. In the reduced modal model (Fig. 4 and 5) the higher dynamics are omitted, but in Fig. 8 the model presented in Fig. 5 is used. Fortunately, mistuning of those parameters can be used in our advantage to minimise the transmissibility in the region of interest by the introduction of an anti-resonance. Therefore the actually build concept is mistuned on purpose. The damping and resonance frequency of the different configurations are given in Table 1.

**EXPERIMENTAL EVALUATION**

The presented design is validated in two different approaches. First the transmissibility of external vibrations to the internal deformation, the Coriolis displacement, is validated and secondly a series of different external broadband excitations are applied and the phase difference is measured, which is proportional to the mass-flow.

The transfer function of the model is validated with an experiment, whereby the new design (Fig. 6) is mounted on a 6DOF Stewart platform (Fig. 10), acting as a shaker to apply a certain level of external vibrations in the dominant y-direction. The result for a CMFM with and without a flexibly mounted sensor

### Table 1. Damping and Resonance Frequencies of the Different Models

<table>
<thead>
<tr>
<th>Configuratie</th>
<th>( \zeta_1 [-] )</th>
<th>( \omega_1 , [\text{Hz}] )</th>
<th>( \zeta_2 [-] )</th>
<th>( \omega_2 , [\text{Hz}] )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference</td>
<td>-</td>
<td>-</td>
<td>3.33e-4</td>
<td>92.2</td>
</tr>
<tr>
<td>CMS</td>
<td>3.33e-4</td>
<td>92.2</td>
<td>3.33e-4</td>
<td>92.2</td>
</tr>
<tr>
<td>CMS (mis-tuned)</td>
<td>3.66e-4</td>
<td>101.0</td>
<td>3.33e-4</td>
<td>92.2</td>
</tr>
<tr>
<td>Realisation</td>
<td>t.b.d.</td>
<td>101.8</td>
<td>t.b.d.</td>
<td>92.3</td>
</tr>
</tbody>
</table>
is depicted in Fig. 9. Compared to a reference system without flexibly mounted sensor, a 20dB reduction of the influence of external vibrations on the Coriolis measurement value in the region of interest is achieved. Note that a peak is visible at 184.6 Hz. For this level of disturbance a second harmonic of the relatively undamped Coriolis mode ($\omega_2 = 92.3\,\text{Hz}$) is visible, due to the non-linear behaviour of the displacement sensors.

At low frequencies, outside the region of interest, the transmissibility is worsened with respect to the reference configuration. This is due to a mismatch in the frequency and damping of the flexibly suspended sensors. Although this is outside the region of interest it might deteriorate the performance, but this effect can be reduced by filtering both at the suspension frequencies and the higher internal modes of the Coriolis tube as expected from the tube model and presented in Fig. 8.

To test the direct influence on the mass-flow measurement, a certain level of external vibrations is applied on the casing of the reference instrument and the instrument with the compliant mounted sensor. The different levels are depicted in Fig. 11 which are compared with Vibration-Curve characteristics. Besides a flat broadband spectrum, the suspension modes of the shaker platform (Fig. 10) are visible. The corresponding measured RMS phase differences are given in Table 2 whereby a 10Hz bandpass filter and notch filters on the frequencies $\omega_1$, $\omega_2$ and $2 \cdot \omega_2$ are used.

With almost no external disturbance the error is equal for the reference system and newly introduced compliant mounted sensor system. This is the noise level of the CMFM, independent of external vibrations. For larger disturbances the influence is reduced by a factor 10, however decreasing for even larger disturbances due to non-linear effects.

The new design is also tested with certain flow levels, results indicate that the flow measurement is unaffected by the design change as expected.

**FIGURE 9. TRANSFER FUNCTION $a_0$ TO $y_{cor}$ - EXPERIMENTAL FOR A BROADBAND DISTURBANCE OF $10^{-4} (m/s^2)^2/\text{Hz}$**

**FIGURE 10. SHAKER SETUP - THE CMFM (FIG. 1) IS MOUNTED ON A 6-DOF STEWART PLATFORM. VOICE COIL ACTUATORS ARE USED TO APPLY FORCES ON THE LOW FREQUENT (25HZ) SUSPENDED PLATFORM, WITH ACCELEROMETERS TO MEASURE ITS VIBRATIONS**

**FIGURE 11. APPLIED DISTURBANCE LEVELS COMPARED TO VIBRATION CURVES**

**DISCUSSION**

Perfect vibration isolation can be obtained when considering only a single mode related to the Coriolis motion, however when there are more internal modes (higher order dynamics) in the compliant sensor element or in the tube the maximum achievable reduction will reduce.

A small misalignment in the two resonance frequencies is not always resulting in an attenuation, it can be the opposite, a magnification of the external vibrations at frequencies around the suspension frequency of the sensor (Fig 8 and Eq. 6). This can result in additional uncertainties in the measurement value. The highly undamped resonance frequencies can still contribute to a measurement error because the attenuation value of band-pass filter is not low enough, thereto extra filters can be used for extra attenuation in this frequency range. Therefore the resonance frequencies of tube and compliant sensor element need to be known.

The influence can be reduced more by accurate tuning of the mass, damping and stiffness of the sensor suspension. It is pos-

<table>
<thead>
<tr>
<th>Disturbance</th>
<th>Reference</th>
<th>CMS</th>
<th>Att.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[rad]</td>
<td>[rad]</td>
<td>[dB]</td>
</tr>
<tr>
<td>1</td>
<td>1.000</td>
<td>1.022</td>
<td>+0.2</td>
</tr>
<tr>
<td>2</td>
<td>1.014</td>
<td>1.034</td>
<td>+0.2</td>
</tr>
<tr>
<td>3</td>
<td>1.603</td>
<td>1.008</td>
<td>-3.67</td>
</tr>
<tr>
<td>4</td>
<td>4.585</td>
<td>1.020</td>
<td>-13.1</td>
</tr>
<tr>
<td>5</td>
<td>15.41</td>
<td>1.394</td>
<td>-20.9</td>
</tr>
<tr>
<td>6</td>
<td>46.46</td>
<td>7.754</td>
<td>-15.6</td>
</tr>
</tbody>
</table>

possible to look to actively and passively tuned mass damper configurations to increase the performance. Active solutions should be considered to correct for damping and resonance frequency change due to temperature fluctuations and change in fluid density. Earlier work [5] showed that there is one direction is dominant for the influence of external vibrations on the measurement value. Therefore only one direction is presented in this work, however the influence in the other directions is not negligible and have to be looked into.

CONCLUSIONS
A vibration isolation solution using a compliant sensor mounting for measuring internal modeshapes is presented. The solution is applied to a Coriolis Mass-Flow Meter, which is able to obtain the mass-flow by measuring the internal tube deformations.

The introduction of a tuned compliant member between the casing and the displacement sensors, results in a displacement measurement of the tube, which is less affected by external vibrations. For the presented test case this results in a 20 dB attenuation of the influence of external vibrations on the mass-flow measurement value of a CMFM. This attenuation was expected in theory and verified experimentally.

For an optimal result the damping ratio zeta and the resonance frequency of the introduced compliant member need to match with the corresponding physical properties of the, to be measured, internal mode. This results in an attenuation in the frequency region higher and lower than the resonance frequency, though the influence around the resonance frequency is equal or even larger. Therefore this solution is only applicable for applications where signals around the resonance frequency are not of interest. When only information in a certain small frequency range is important, the damping and resonance frequency can also be tuned such that there is an anti-resonance, causing a large attenuation in a small region.

The presented performance is limited to a specific case, however, the concept can be used in every CMFM and even in other kinds of systems where specific internal deformations need to be measured independently of external vibrations.

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