Tuned mass dampers help FORTiS™ encoders achieve class-leading vibration resistance

In 1907, the American mechanical engineer Frederick Winslow Taylor described machining vibrations as "the most obscure and delicate of all the problems facing the machinist". From our extensive machining experience gained over nearly five decades both here at Renishaw and through close collaboration with our global customer base, we appreciate his point. Despite the advances made over more than a century and development of the modern high-quality, high-speed CNC machine tools available today, vibration can and does occur. Typical applications that may cause it include heavy roughing cuts, intermittent cutting, thin-walled components, and the machining of especially hard and exotic materials. Correct tool selection and the optimisation of feeds and speeds are essential measures to help reduce or dampen vibration. To ensure optimum performance of the machine's motion control system and therefore its output, the need to balance cycle time with accuracy and quality of finish is ever present.

Paul Maxted – Director of Industrial Metrology applications – Renishaw plc.

Problem

Machine tools can produce significant vibration during operation and high levels of machine vibration can adversely affect the enclosed position encoders installed on these machines, leading to measurement inaccuracies. The quality of axis position measurement can directly impact aspects of process quality, such as feature accuracy and surface finish. Position measurement improvements that reduce the influence of vibration can significantly improve the quality of production. The main causes of vibration in machine tools are:

i. Tool chatter, which can occur during the cutting process under certain conditions. For example, when milling hard materials, deflection of either the workpiece or the tool can occur due to excessive cutting forces.

ii. Inhomogeneities in the material being machined and built-up edges on the cutting tools. Due to a sudden increment in machining difficulty, impulsive force is generated which causes vibration.

iii. Intermittent cutting, common in milling, causing impulsive forces to be generated and resulting in vibration.

iv. Disturbances due to unbalanced rotating masses, changes in transmission damping, such as bearing wear, or poor workpiece holding.

v. Worn or poorly maintained machine tools and poor selection of cutting tools, non-optimised spindle speeds and feed rates.

Solution

Renishaw's extensive experience of machine tools has enabled successful partnerships with major machine tool builders as well as end-users. The FORTiS™ enclosed encoder was conceived to address known issues with machine tool vibration and its effects on position measurement. Three features of the FORTiS encoder design combine to improve robustness against mechanical vibration and enhance the encoder's ability to stop high-amplitude disturbances from entering the position control loop:

1. Conventional enclosed optical encoders feature a sprung, wheeled carriage that supports the readhead body as it runs along the encoder scale, as shown in Figure 2a. At any given driving frequency, the amplitude and phase of vibration of the machine guideway (V_g) supporting the readhead will be different to that of the encoder's scale and enclosure (V_m), which are fixed on the machine mounting surface; this difference in the amplitude and phase response must be absorbed by the flexures and couplings in the wheeled carriage, as seen in Figure 1. FORTiS encoders employ a contactless design which effectively isolates the readhead body from its enclosure, as shown in Figure 2b.

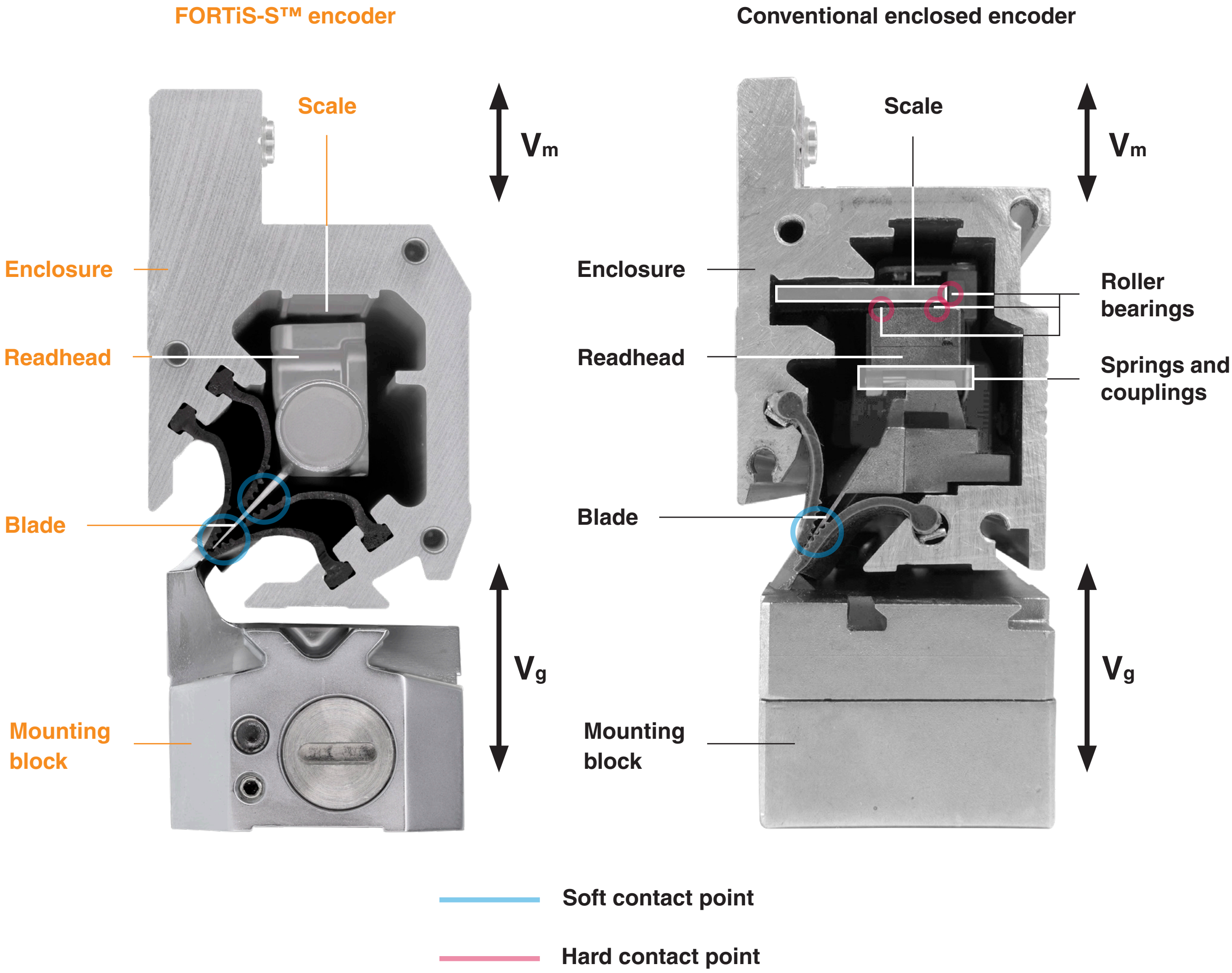


Figure 1: Vertical sections through a FORTiS-S encoder and a conventional enclosed encoder.

2. Conventional enclosed optical encoders use a glass scale of relatively high mass, suspended from one side of the enclosure. To avoid unwanted oscillations in the enclosure due to the vibration of the cantilevered scale, the FORTiS encoder uses a light steel scale that is fixed along its full length to the inside of the enclosure body.

3. The third vibration-reducing design feature of the FORTiS encoder is the use of tuned mass damping. A tuned mass damper (TMD) is a mechanical device, mounted to a specific location in a structure, to strongly damp resonant vibration. Two TMDs are used to counteract vibration around the vertical and horizontal axes.

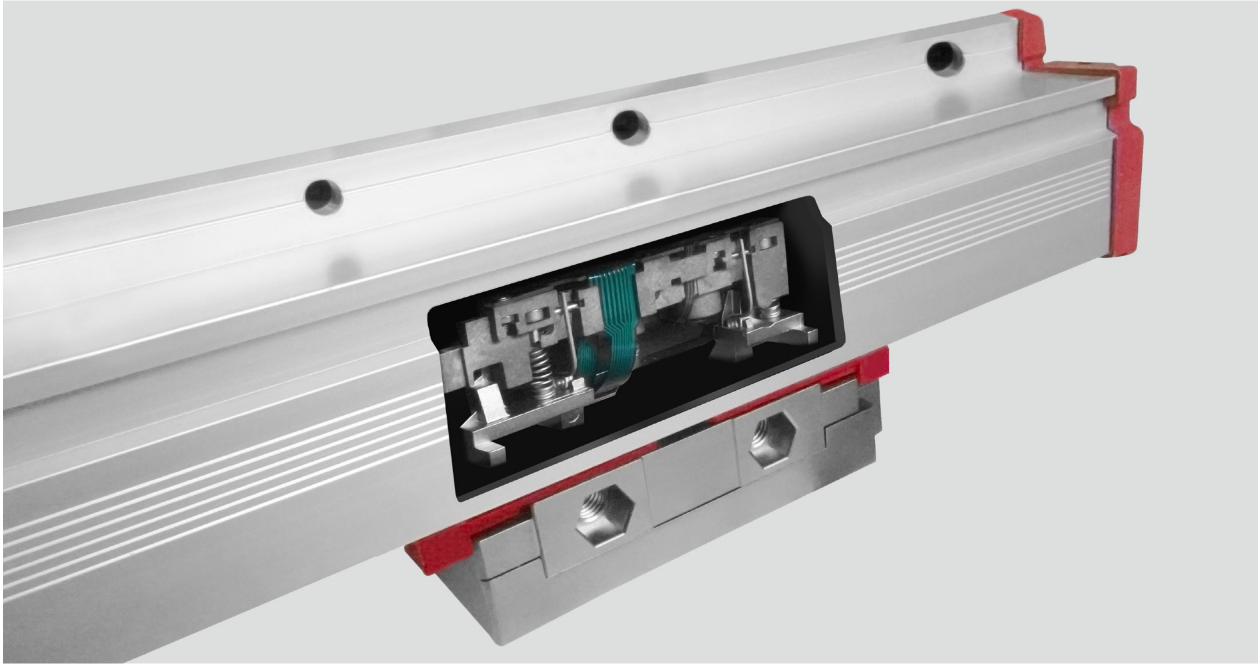


Figure 2a: Conventional enclosed encoder with cutaway showing the readhead body and supporting wheeled carriage inside its enclosure. Note that the readhead mechanism is completely exposed to contaminants that enter the enclosure.

Figure 2b: FORTiS-S enclosed encoder with cutaway showing non-contact, sealed readhead body inside its enclosure.

An introduction to tuned mass damping

Tuned mass dampers are used in a wide range of engineering applications where it is important to damp mechanical vibration in an object with a well-defined resonant frequency. Perhaps the best known example of TMD use is in the construction of super-tall skyscrapers such as the famous Taipei 101, where a large, tuned mass damper is used to reduce vibrations in the building due to high winds or earthquakes, as shown in Figure 3. In other examples, TMDs are fitted to power transmission lines, aircraft wings, car crankshafts, bridges and the FORTiS encoder.

Tuned mass damping is employed on the FORTiS encoder readhead in both the vertical (Z-axis) and horizontal (Y-axis) directions. The basic design of the FORTiS TMD comprises two O-rings applied to spigots at the ends of a damper mass, which is installed inside a reamed pocket to control the compression of the O-rings, as shown in Figure 4.

Extensive development work on the TMDs has enabled the FORTiS encoder to achieve a 5.3 times reduction in the peak acceleration at the end of the readhead (the optics carrier) furthest from the mounting points.

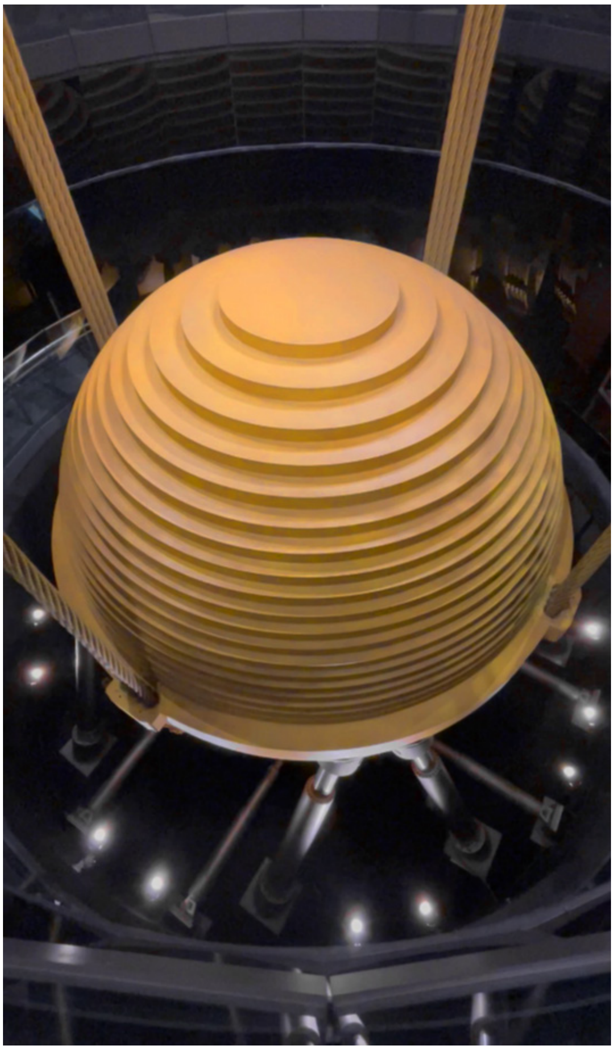


Figure 3: TMD system on the Taipei 101 skyscraper.

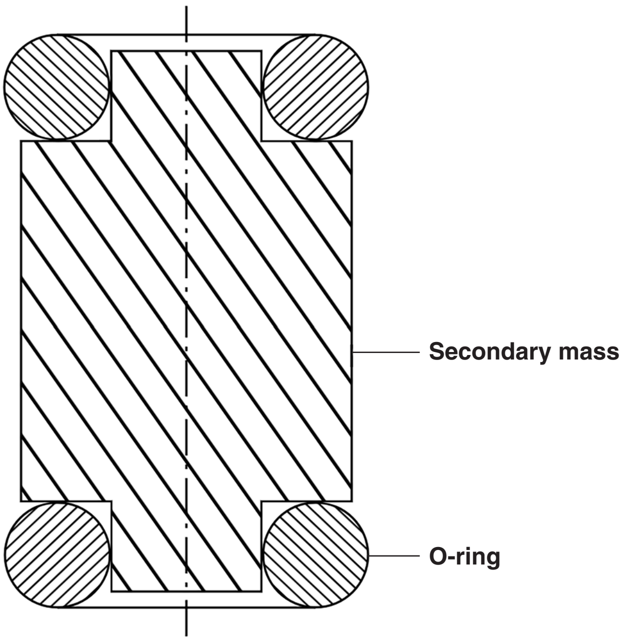


Figure 4: Vertical section through the TMD, designed for the FORTiS encoder, with the O-rings visible at each end.

Tuned mass damper theory

This section describes the basic theory of a tuned mass damper for a one degree of freedom (DoF) system. A problem frequently encountered by mechanical and civil engineers is resonance when a system produces high amplitude oscillations in response to an input excitation.

Resonant systems can be understood in terms of a driven, simple harmonic oscillator such as a mass (m) on a spring of stiffness (k), as shown in Figure 5. In this case, the familiar equation of simple harmonic motion applies, where x is the linear displacement from static equilibrium.

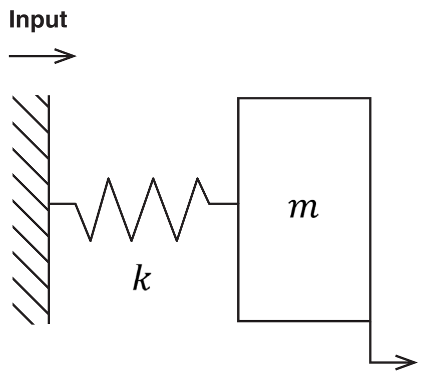


Figure 5: Mass – spring system with 1 DoF.

Equation 1: $m\ddot{x} + kx = input$

Equation 1 implies that a mass on a spring will have a natural frequency of sinusoidal response as shown in Equation 2:

Equation 2: $frequency\ (Hz) = \frac{1}{2\pi}\sqrt{k/m}$

If the frequency of a system input, either a force or displacement, is close to the resonant frequency of Equation 2 then the result will be a large resonant response with potentially destructive consequences, as shown in Figure 6.

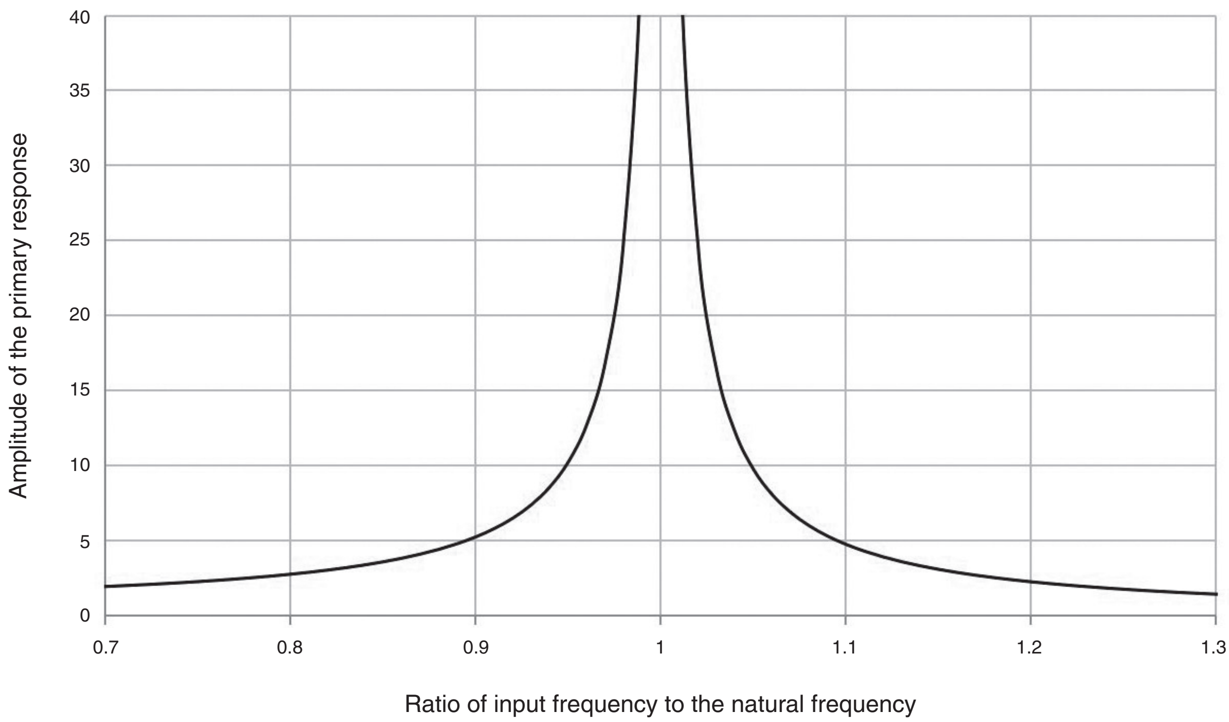


Figure 6: Resonant response of a mass on a spring.

A common strategy, adequate in many cases, is to add mechanical damping and shift the natural frequency of the system away from the excitation frequency.

However, this approach is not always feasible. A steel framed skyscraper will sway at its natural frequency and there is nothing nearby to serve as an anchor for reinforcement or damping. In such cases, when access is a challenge, a potential solution is a tuned mass damper.

The mass of the FORTiS encoder readhead is supported by the blade which acts as a spring due to being thin in order to facilitate sealing integrity. External machining vibrations may induce undesirable resonance, unless controlled by employing TMDs within the readhead.

Practical design of TMDs requires careful development work but the basic concept can be visualised by starting with the undamped mass – spring system and the problem of resonance at its natural frequency.

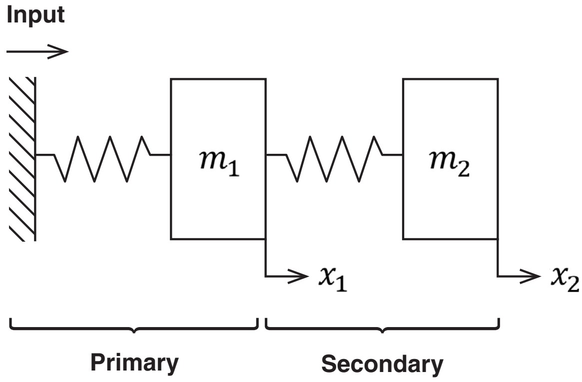


Figure 7: Mass – spring system with 2 DoF.

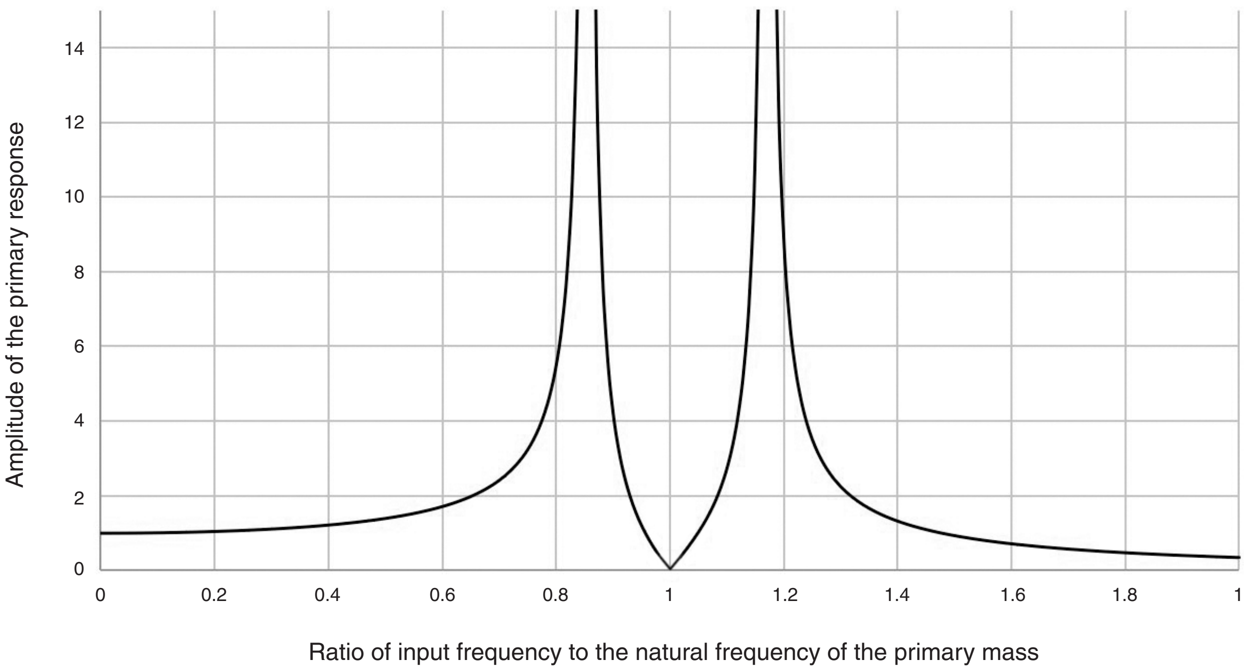


Figure 8: Resonant response of an undamped 2 DoF mass – spring system.

Suppose that a relatively small secondary mass (m_2) is coupled to the primary (original) mass (m_1) by a spring that gives the secondary mass the same natural frequency. The overall system, shown in Figure 7, is now said to have 'two degrees of freedom' which causes the original resonance peak to split into two. At the first (lower) natural frequency, both masses move in phase and in the same direction, while at the 2nd natural frequency they move in opposite directions.

Furthermore, the primary mass has zero amplitude when driven at the original natural frequency, while the secondary mass oscillates with finite amplitude. Thus, the resonance of the primary mass is suppressed at the expense of two new unlimited resonances at different frequencies, as seen in Figure 8.

Left unchecked, these resonances could be destructive but the key advantage of the TMD is that they can be controlled locally by a damper across the spring supporting the secondary mass, as shown in Figure 9.

Summarising, the TMD has three fundamental design parameters which can be adjusted: the ratio of the secondary mass to the primary mass (m_2/m_1), the natural frequency ratio of the secondary and primary masses (tuning frequency), and the damping factor of the damper.

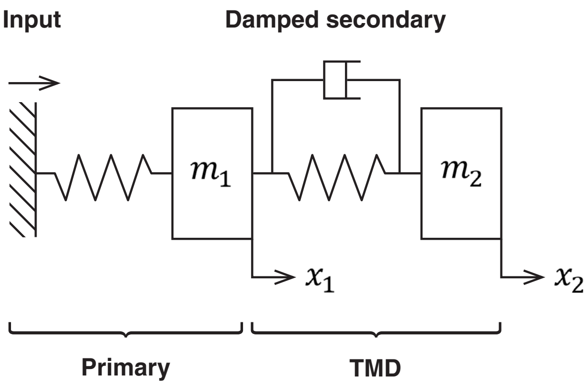


Figure 9: Tuned mass damper system with 2 DoF.

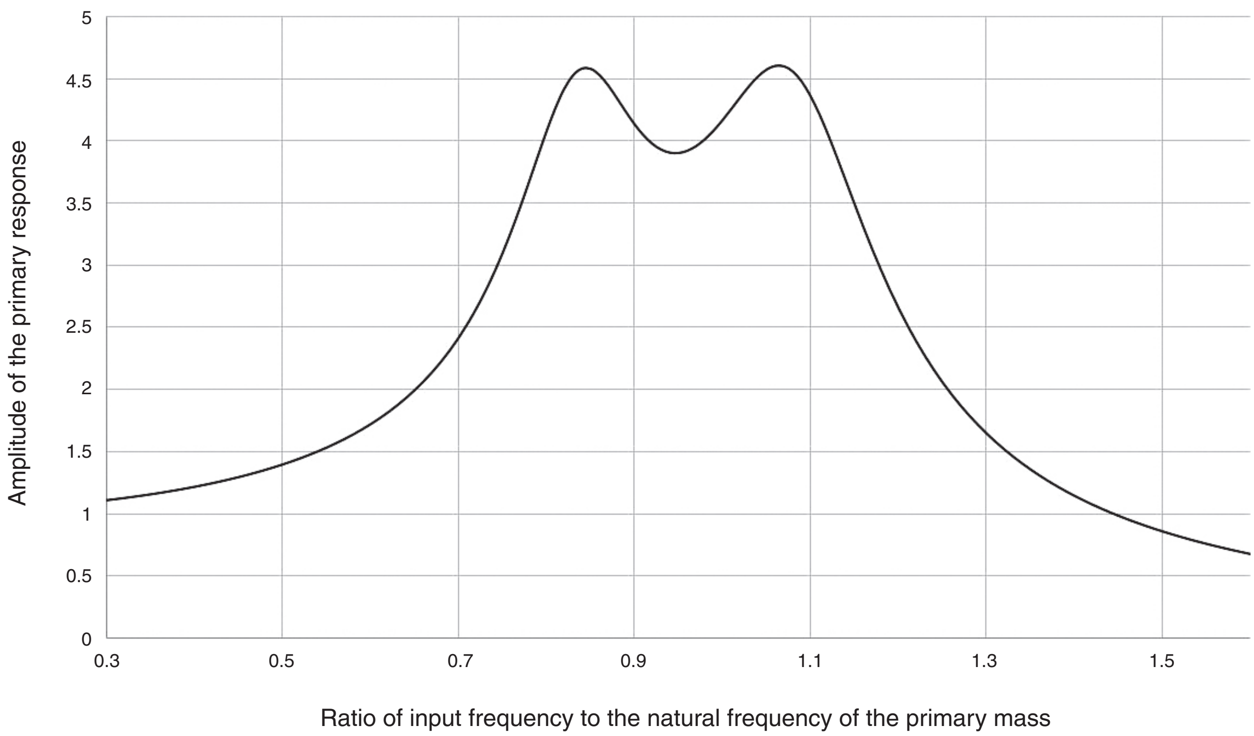


Figure 10: Shows the result, assuming a 10% mass ratio, after optimum tuning of the secondary to 91% of the original resonant frequency and with well chosen damping. The primary response is now always below 4.6 times the static, in contrast with the unlimited responses seen in Figures 6 and 8.

There are practical limits on the secondary mass, but good results are possible with a secondary mass of only 10% of the primary. The optimum tuning frequency turns out to be lower than the primary resonant frequency by an amount depending only on the mass ratio. Finally, the secondary damping factor is chosen to minimise the two peak responses, and control the amplitude response at all other frequencies.

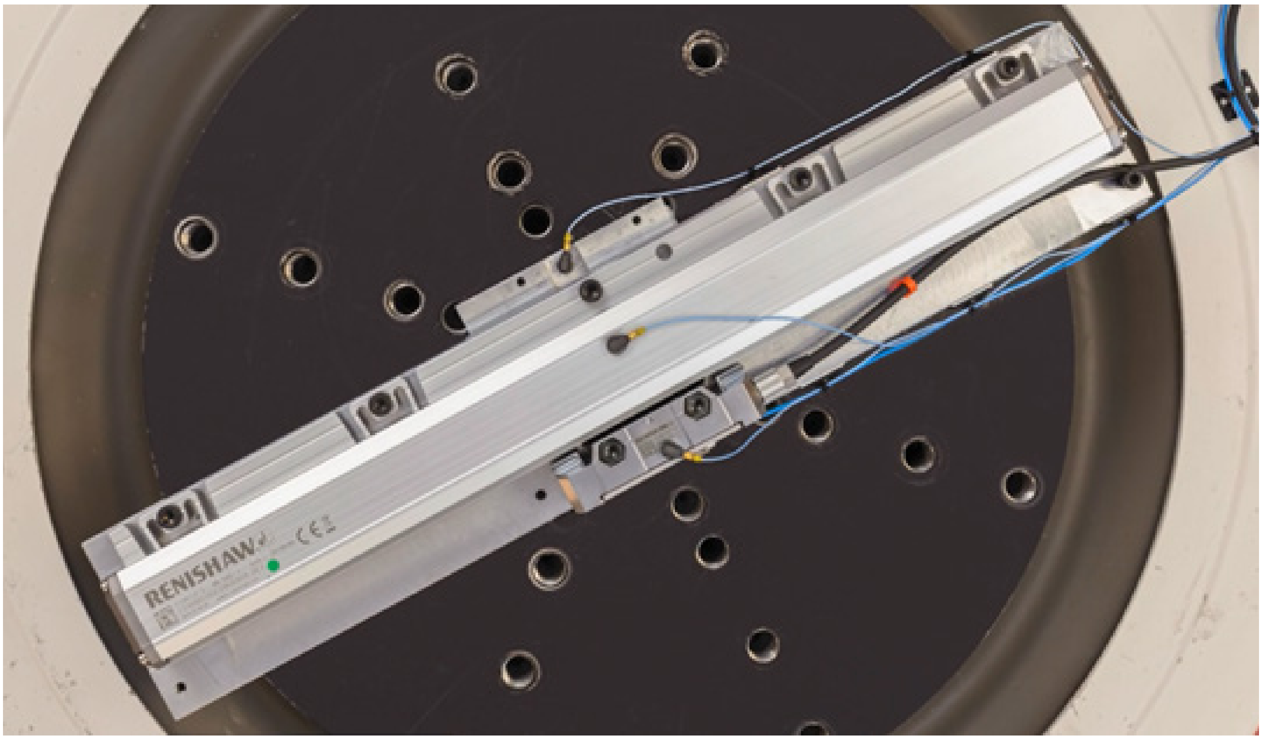
One of the first big challenges that we faced in the design of the TMDs was to predict the properties of the various candidate O-ring materials under dynamic conditions. I needed this data for simulation testing to select the optimum rubber hardness for spreading the frequency response and making the system less susceptible to dimensional variations in the various components.

A second big challenge was defining the properties of the chosen material to enable finite element analysis. Finally, we needed to optimise and tune the system which we validated with test data. The final design produces the best modal shape over the resonant frequency range of the readhead; the remaining harmonic responses of the tuned mass damper are also useful at different vibration levels.

Krys Jurczyks – FORTiS Senior Mechanical Design Engineer

Vibration testing: Sine vibration test

FORTiS encoders were tested to measure the deviation in absolute position reading, compared to the starting value, when exposed to swept sine vibration over a frequency range from 50 Hz to 2000 Hz, with the test repeated at vibration amplitudes of 1 g, 3 g, 5 g, 10 g, 15 g, 20 g, and 25 g. In other testing, not covered by this white paper, the test was repeated to cover the range of 30 g to 75 g. Furthermore, additional testing was carried out with high levels of random vibration.



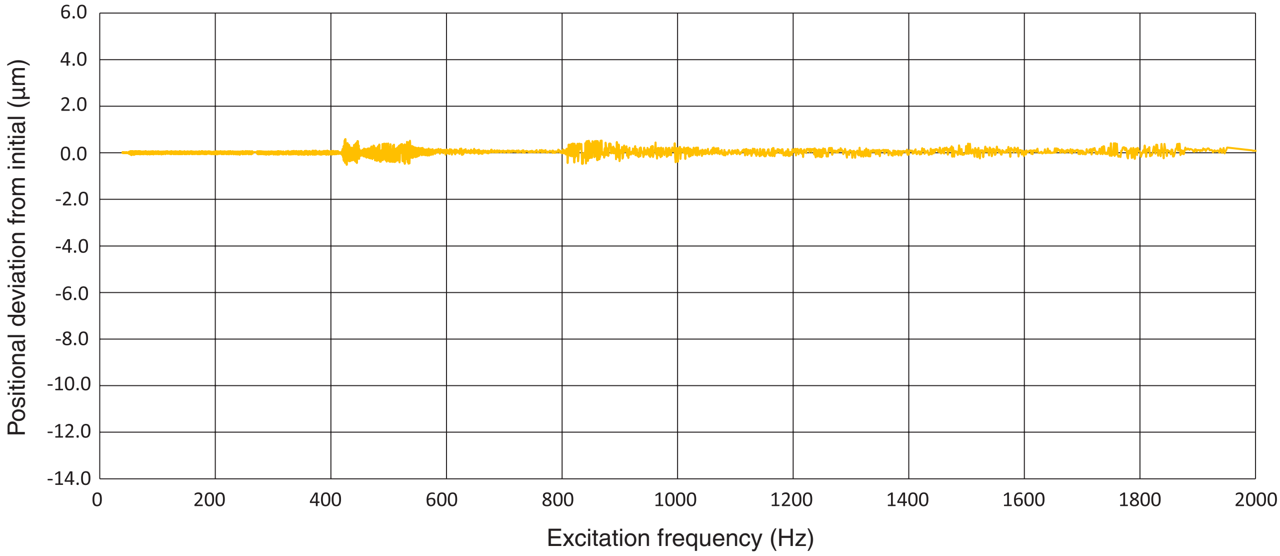
The tests were carried out on FORTiS-S encoders and FORTiS-N™ encoders directly mounted to a substrate, and FORTiS-N encoder fitted to a mounting spar. The FORTiS-N encoder uses the same technology as the FORTiS-S encoder but features a narrow cross-section for space-constrained applications.

For comparison, tests were carried out on conventional enclosed encoders, made by a competitor, of equivalent fit, form and function.

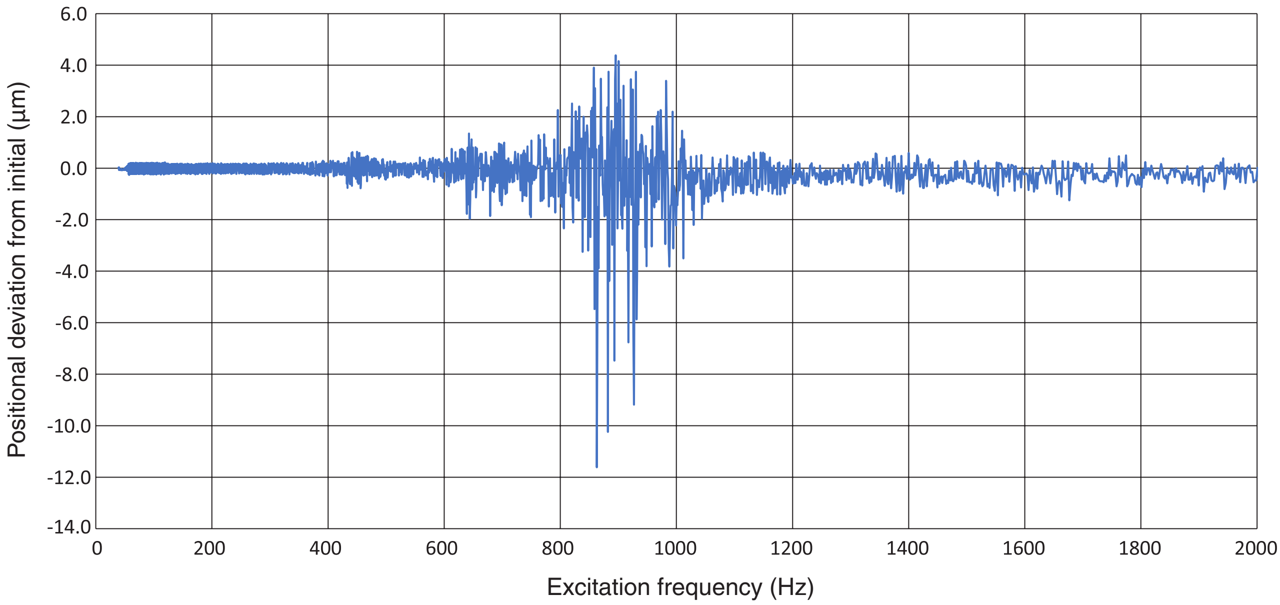
FORTiS-N encoder versus competitor (spar mounted)

The following graphs show the positional deviation for a FORTiS-N encoder versus a competitor's conventional enclosed encoder. In both tests, the encoders were mounted on a spar. The amplitude of the sinusoidal vibration test profile is 15 g in the Z-axis direction (into the encoder scale).

Positional deviation for the FORTiS-N encoder (on spar) with vibration amplitude of 15 g in the Z-axis direction over a frequency range of 50 - 2000 Hz

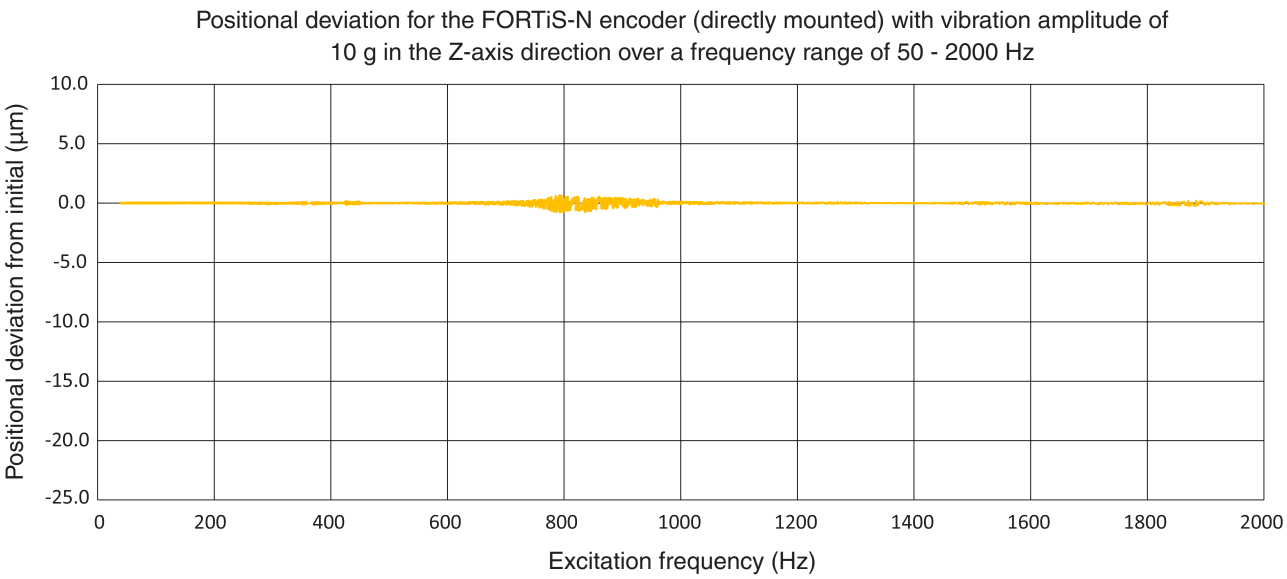


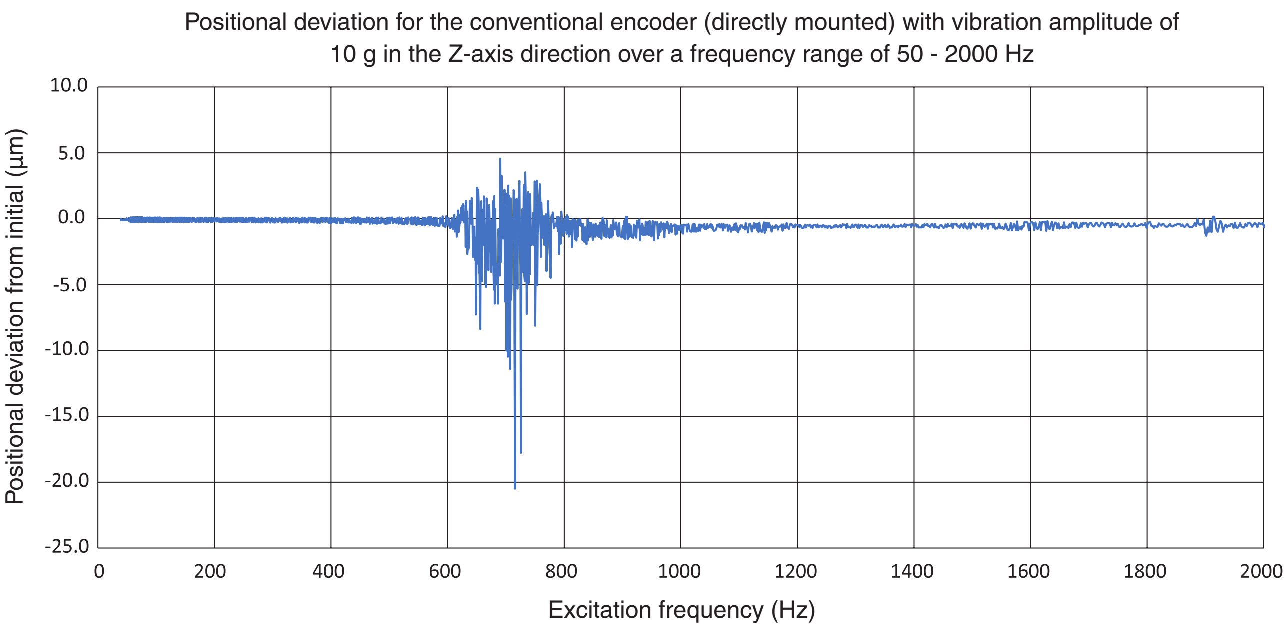
Positional deviation for the conventional encoder (on spar) with vibration amplitude of 15 g in the Z-axis direction over a frequency range of 50 - 2000 Hz



FORTiS-N encoder versus competitor (direct mounted)

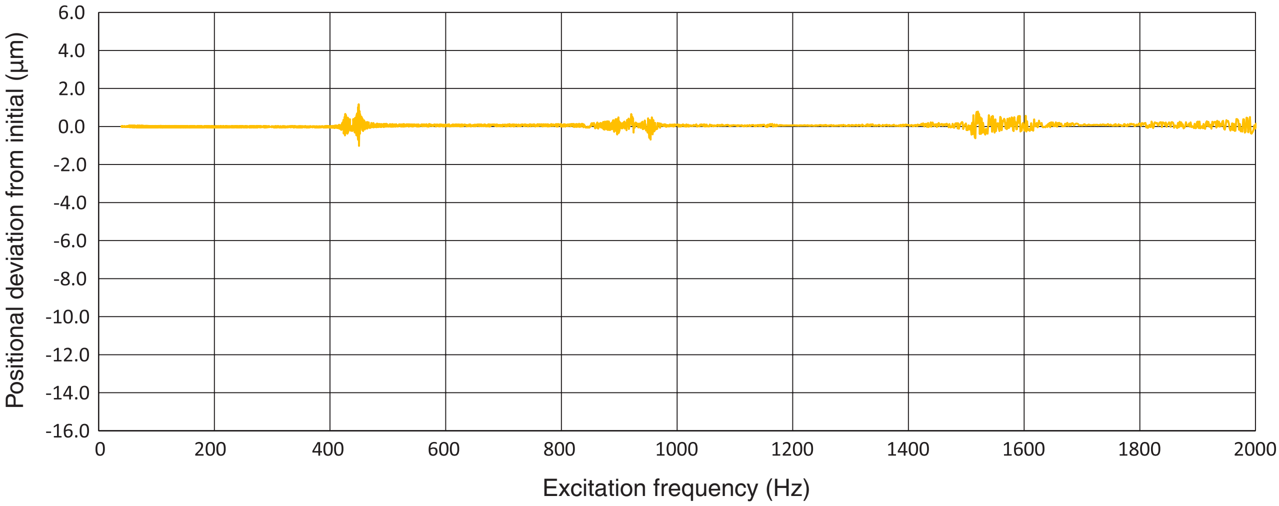
A second set of graphs show the positional deviation for a FORTiS-N encoder versus a competitor's conventional encoder, but with both encoders directly mounted to the substrate (test fixture). In this test, the amplitude of vibration is 10 g which is applied in the Z-axis direction.



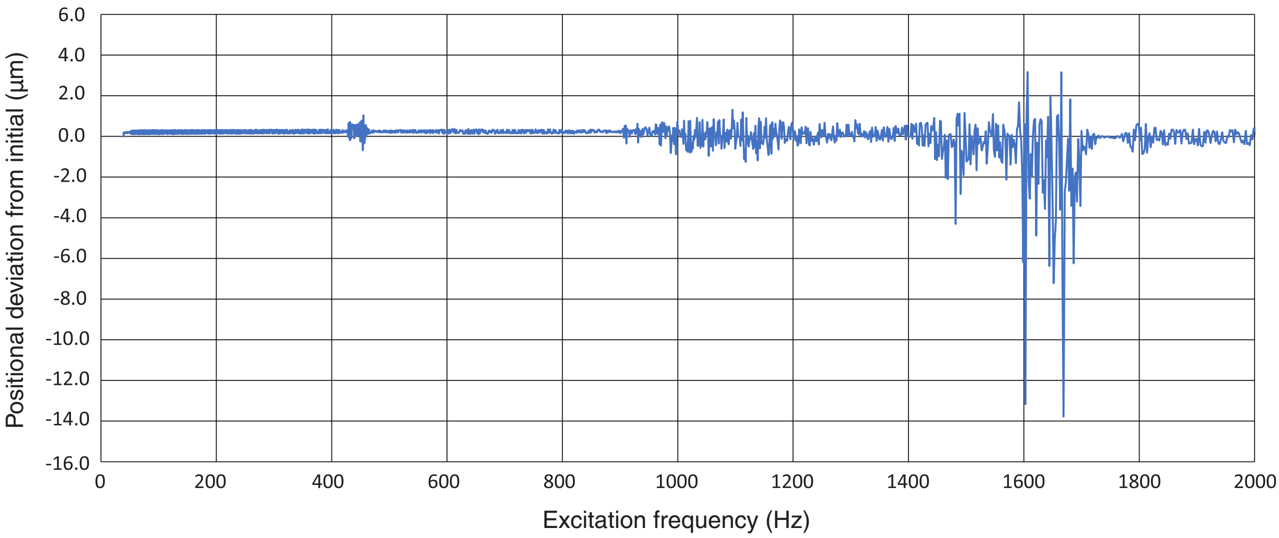


A third set of graphs show the positional deviation for FORTiS-S encoder versus a competitor's conventional encoder. On these standard-size encoders, they are both directly mounted to the substrate (test fixture). In this test, the amplitude of vibration is 25 g which is applied in the Z-axis direction.

Positional deviation for the FORTiS-S encoder (directly mounted) with vibration amplitude of 25 g in the Z-axis direction over a frequency range of 50 - 2000 Hz



Positional deviation for the conventional encoder (directly mounted) with vibration amplitude of 25 g in the Z-axis direction over a frequency range of 50 - 2000 Hz



Conclusion

In this paper, it has been shown that FORTiS encoders provide good positional stability and reliable operation, whilst being exposed to a wide range of vibration amplitudes, across the full frequency spectrum from 50 Hz to 2000 Hz. FORTiS encoders have also been shown to provide long life operation in high-vibration environments.

Comparison with conventional enclosed encoders that do not feature tuned mass damping demonstrates the superior vibration resistance of the FORTiS encoder. Therefore, FORTiS encoders offer superior positional stability for improved process control in machine tool applications.

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