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DESIGN OF THE FRAME FOR THE UK TORQUE CALIBRATION MACHINE

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Abstract: At IMEKO - XV World Congress, the design of the first UK national torque calibration machine was described.

As part of the design of the machine frame, a structured finite element (FE) study was carried out to predict its behaviour. A series of models were created, with varying levels of idealisation. These were used to evaluate the linearity of response to load application, the deflection and stress states under differing load conditions, and dynamic response using both modal and post-dynamic analysis. The effect of using tubular columns in the upper structure in place of solid bars was also modelled.

The study provided confidence in the design of the machine frame and reduced the risk of post-manufacturing changes. It also contributed to the design of the software that controls the application of deadweight masses and the applied torque.

During commissioning of the machine, deflection and frequency measurements will be made to close the design loop and provide validity to the FE analysis.

Keywords: Static torque, FE analysis, 2 kN·m torque calibration machine

1 INTRODUCTION

The first UK national standard torque calibration machine will be of a lever beam/deadweight design with the transducer under calibration mounted vertically [1]. The target uncertainty budget of the machine is 1×10^{-5} over the majority of its operating torque. This dictates that any deflections of the machine frame are minimal. For the target uncertainty budget, it was specified that the allowable deflection of the machine's upper frame (the working envelope) should be less than the radial deflection of the central pivot air bearing of approximately 15 µm. In addition, deflection of the upper frame with an off-setting force of 500 N applied to the reaction drive gearbox should also not exceed this 15 µm value.

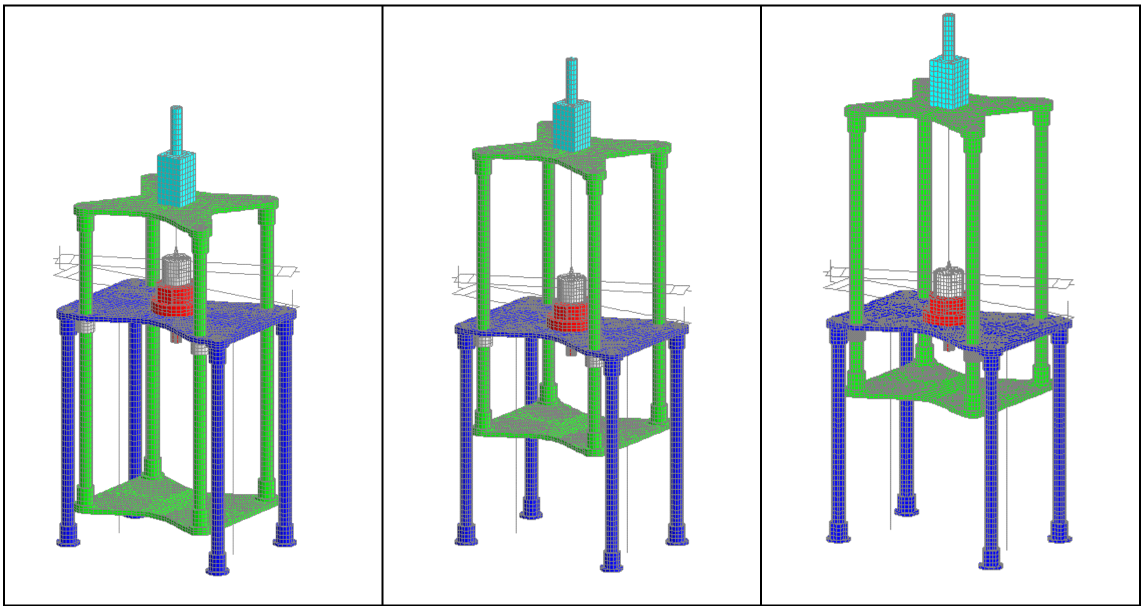
As part of the design of the machine frame, a structured finite element (FE) study was carried out to predict its behaviour. The study initially considered a frame construction using 100 mm diameter, solid stainless steel columns. A second study was then carried out to replace the upper frame with tubular columns with a 70 mm bore diameter. The benefit of tubular columns was in significant weight saving and therefore in the specification of the cross-head motor.

A series of models and levels of idealisation were created. These were used to evaluate the linearity of response to load application, the deflection and stress states under different load conditions, and the dynamic response using modal and post-dynamic (forced dynamic) analysis. This enabled each stage of the design to be investigated efficiently.

The analysis followed the critical design stages of the machine, concentrating initially on the mass suspension system and end fittings before considering the lever and reaction beams. The beams were modelled both individually and with the air bearing as a complete sub-assembly [2]. Finally, models of the entire machine were generated at three cross-head heights simulating three dissimilar torque transducers of varying capacities, torsional rigidities, and length (Figure 1). This analysis made extensive use of lumped and simplified parameters to investigate deflection and dynamic response.

This scaled detailed approach meant that deformation and vibration characteristics of the various components and sub-assemblies could be traced through the complete machine and the effect of component interaction assessed. It also ensured that the differing levels of idealisation employed are self-referenced, improving the robustness of the modelling approach.

The predictive modelling techniques will be subsequently validated experimentally to provide a detailed picture of the machine behaviour. Such analysis not only supplies confidence in the machine structure, thereby reducing the need for machine modification and cost over-runs but also assists in the design of the operating software which controls the application of deadweight masses and the applied torques. This practical validation of the machines behaviour will close the design loop and provide validity to the FE analysis approach.



Lower cross-head position	Middle cross-head position	Upper cross-head position

Figure 1. Overall machine models in three operating positions

2 THE ANALYSIS

Cosmos/m is a general purpose finite element code. Initial analysis centred on evaluation of a mesh density to provide accurate results within the time-scale. To this end a ¼ model was used with a mesh of 8 node brick density, with the plate sections meshed with 2 elements through the thickness. This was found to provide a converged solution of acceptable time-scale.

In order to allow the consideration of tubular columns in the upper frame section, the model was generated with the centre section of the columns defined so that these sections could be easily removed to create tubes.

Lever and reaction beam elements were used to represent the sub-assembly and these were attributed appropriate properties and values. The central pivot air bearing was a simplification of the model described in [2].

2.1 Material Model and Material Properties

The stainless steel sections were modelled as a linear, isotropic material with a Young's modulus of 190 $GN{\cdot}m^{-2}$, a Poisson's ratio of 0,29 and a density of 7,8 x 10^3 $kg{\cdot}m^{-3}$ .Where complex components were modelled, these were given an appropriate outer profile with the density scaled to give the correct mass. Scaling was applied to the following components:

Reaction drive gearbox	:	mass and modulus scaled
Deadweight suspension system	:	mass and modulus scaled
Central pivot air bearing	:	mass only scaled

The composite sections were modelled as a linear orthotropic material with a longitudinal modulus of 172 $GN{\cdot}m^{-2}$, and circumferential and radial modulus of 22 $GN{\cdot}m^{-2}$, a Poisson's ratio of 0,276 and a density of 1,56 x 10^3 $kg{\cdot}m^{-3}$, as used in [2].

2.2 Torque transducer stiffness

Three dissimilar torque transducers were selected such that the machine frame was evaluated both at the extremes of cross-head travel and in the central position; the conditions were as listed below:

Upper position, 1 kN·m torque transducer of stiffness	68,4 x 10^3 N·m·rad^{-1}.
Middle position, 2 kN·m torque transducer of stiffness	296 x 10^3 N·m·rad^{-1}.
Lower position, 2 N·m torque transducer of stiffness	202 N·m·rad^{-1}.

2.3 Structural Constraint Definition

For simplicity of the various models the base of the columns and ballscrews (used to drive the cross-head) were assumed fully fixed.

2.4 Structural Loads

The structural loading of the machine used in the model considered: (i) frame deflection resultant from the self-weight of the lever beam (10 kg), and reaction beam and castored pulleys (60 kg), and (ii) deflection resultant from the application of a maximum deadweight force (1 000 N), either applied singly or at both ends of the lever beam. In addition, a loading of 500 N to the reaction drive gearbox in the x, y and z directions was also analysed. Gravity was taken as 9,81 m·s^{-2}.

Note: x direction - horizontal direction along the plane of the lever beam.
y direction - vertical direction of the machine frame.
z direction - horizontal direction normal to the plane of the lever beam.

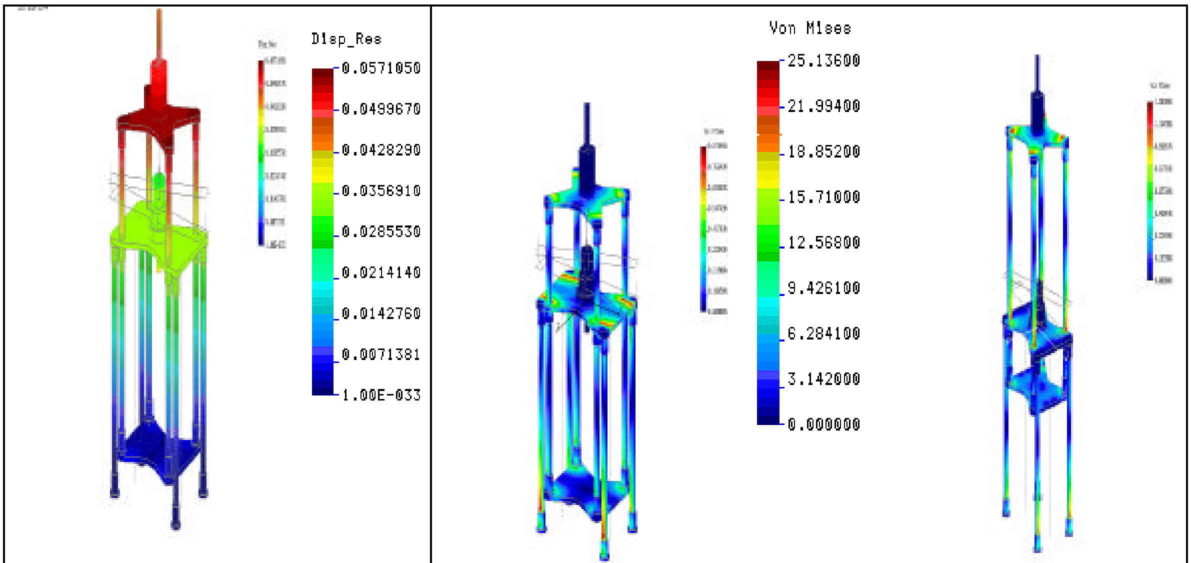
2.5 Dynamic Excitation

The analysis of the post-dynamic response model used a harmonic base excitation of 1 mm·s^{-2} and 1 % of critical damping between 0 Hz and 30 Hz, in the three orthogonal directions.

3 RESULTS

3.1 Frame Deflection and structural loads

Calculated deadweight forces were shown to have insignificant effect on the deflection of the solid column frame, similar results were recorded for the tubular upper frame. Figure 2 shows typical deflection and stress states for the machine resultant from a 500 N force applied to the reaction drive gearbox.



Displacement from head load	Stresses from head load - cross-head lower position	Stresses from head load – cross-head upper postion

Figure 2. Typical deflection and stress states of the machine caused by a 500 N reaction drive head load

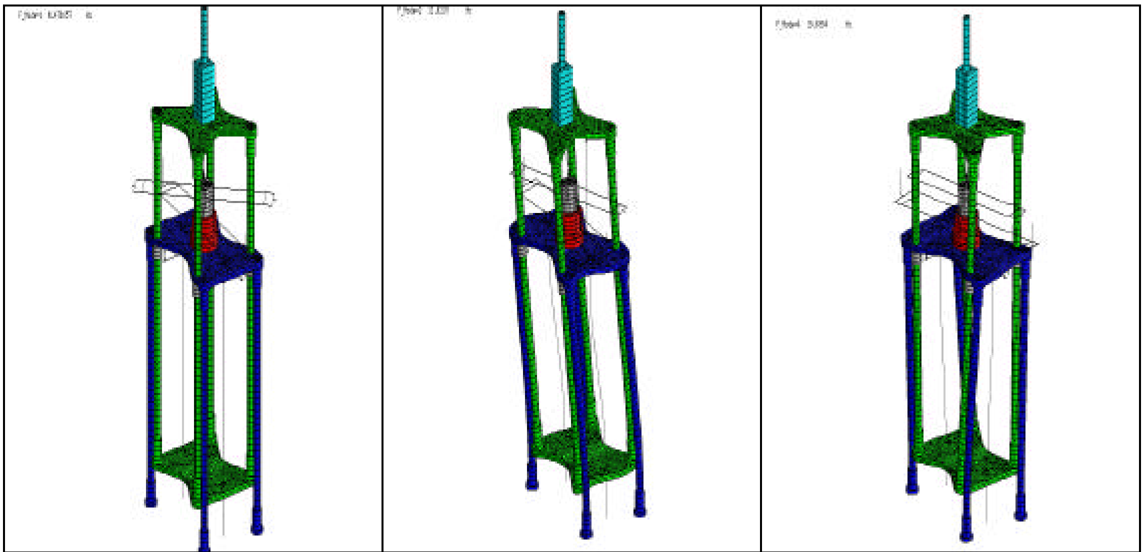
Frame deflections relative to the base of the columns, for both the solid and tubular upper frames, are shown in Table 1. These results show a marginal increase in frame deflection for the tubular upper columns. For both the solid and tubular upper frame construction the deflection increased as the cross-head is traversed to its upper location. The overall deflection of the machine frame serves only to make a simplified comparison between the solid and tubular upper frames. Of more importance is the deflection of the upper frame of the torque machine, i.e. the working envelope. Detailed analysis of the calculated overall machine frame deflections enabled the deflection of the upper frame to be determined. These calculations revealed that for this worst case condition that deflection of the upper machine frame was within the target specification of 15 µm, i.e., less than the radial deflection of the central pivot air bearing. Furthermore, this FE analysis calculated that under this worst case condition the level of stress within the frame was typically below 25 $N{\cdot}mm^{-2}$, and considered to be insignificant. It was therefore concluded that the frame provided a stable structure for construction of the 2 kN·m torque calibration machine.

Table 1. Frame deflection due to cross-head loading, relative to machine base.

	Solid upper columns	**Tubular upper columns**
Upper Position		
x direction force – resultant deflection	0,140 mm	0,160 mm
y direction force – resultant deflection	0,008mm	0,009 mm
z direction force – resultant deflection	0,160 mm	0,180 mm
Middle Position		
x direction force – resultant deflection	0,110 mm	0,120 mm
y direction force – resultant deflection	0,008 mm	0,008 mm
z direction force – resultant deflection	0,120 mm	0,130 mm
Lower Position		
x direction force – resultant deflection	0,057 mm	0,063 mm
y direction force – resultant deflection	0,006 mm	0,007 mm
z direction force – resultant deflection	0,068 mm	0,075 mm

3.2 Modal Analysis

Modal analysis of the machine frame and the component sub-assemblies was carried out for fundamental resonances up to mode 40. This analysis considered the three cross-head heights.



Mode 1 – 6,6 Hz	Mode 2 – 9,6 Hz	Mode 4 – 16 Hz

Figure 3. Typical mode shapes of machine.

Figure 3 shows typical examples of the modal analysis. A mode 3 plot is not included in this figure as it showed only a marginal difference in frequency response compared to a mode 2. The analysis confirmed that the frequency vibration range of the machine was above any expected excitation frequencies that may be generated during its operation. The results therefore indicate that the performance of the machine would not be compromised by any extraneous vibration noise. Marginal improvements in frame vibration frequency were calculated for the tubular upper frame.

3.3 Post-dynamic Response (Forced Dynamic Response)

An example of post dynamic response of the machine is shown in Figure 4: a calculated harmonic plot of 'x' direction displacement up to 30 Hz.

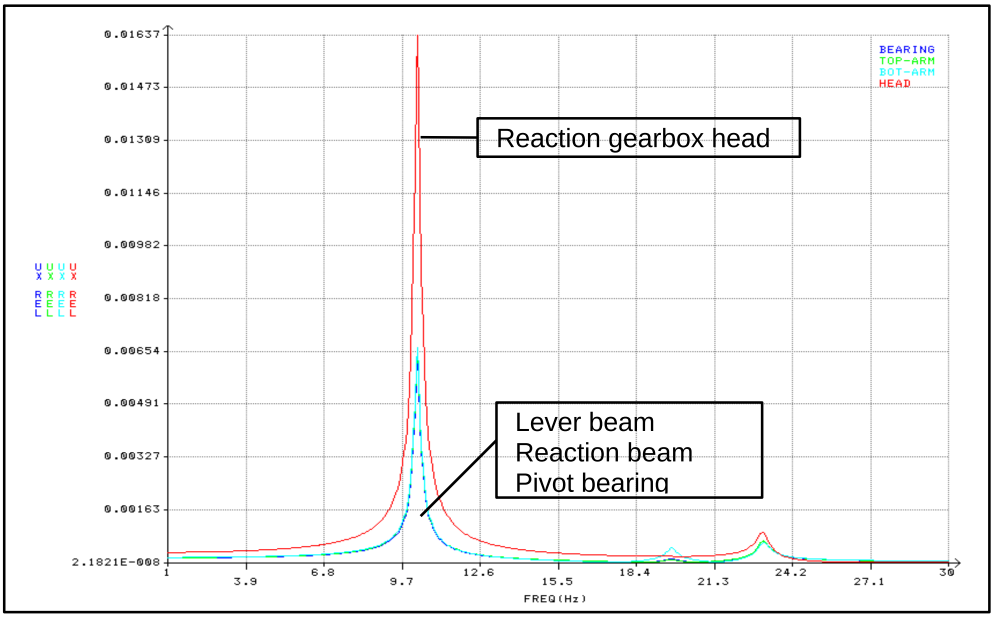


Figure 4. x direction response to base excitation

Figure 4 plots the relative displacement response in mm, i.e. the amplitude of vibration (y axis) against frequency for the x direction base excitation (x axis) at critical locations in the machine. These results will be used to validate the FE model against experimentation to demonstrate the importance of modelling during the design phase of the torque machine. This relationship between FE modelling and experimentation will provide additional confidence in design technique for future design improvements and development of this 2 kN·m torque standard calibration machine.

4 CONCLUSIONS

The FE analysis of the frame of the 2 kN·m torque calibration machine has shown that deflections due to deadweight loading have an insignificant effect. However, under an assumed worst operating condition - the application of an off-set loading of 500 N caused by misalignment of the reaction drive gearbox head - some deflection of the machine frame is predicted. Further, the analysis predicts a marginal increase in overall frame deflection for the tubular upper columns, this increasing as the cross-head is traversed to its upper location. Of greater importance is the deflection within the upper frame structure, this being the working envelope. Nevertheless, for the assumed worst case condition, the deflection of the upper frame was less than the target figure, i.e. the specified radial deflection of the central pivot air bearing of 15 µm. In addition, under this condition the calculated level of stress within the frame was typically below 25 N·mm^{-2}, and was considered to be insignificant.

The results of the vibration analysis showed that the frequency response of the machine frame was above any expected excitation frequencies that may be generated during operation of the torque calibration machine, and that marginal improvements in the frame vibration frequencies would be achieved by using a tubular upper frame.

From this FE analysis it was concluded that the frame provided a stable structure for construction of the 2 kN·m national torque standard calibration machine.

Practical experimentation is planned to provide a relationship between practice and theory and to close the design loop thereby demonstrating the usefulness of FE analysis as a design tool. This relationship will provide confidence in the design techniques used, necessary for the future design of torque standard machines and the development of this 2 kN·m torque standard calibration machine. A development of the present torque machine is the ability to carry out continuous calibration [3], i.e. "on-the-fly" calibration. This will provide a unique torque standard facility as there will be direct traceability for the dynamic calibration of a torque transducer to its static calibration.

ACKNOWLEDGEMENT

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