

## CHARACTERIZATION AND QUALIFICATION OF A FORCE DYNAMIC CALIBRATION DEVICE

*R. R. Machado<sup>1</sup>, R. S. Oliveira<sup>1</sup>, L. C. C. Freitas<sup>1</sup>, C. A. Koch<sup>1</sup>, A. Knott<sup>2</sup> and R. S. Dias<sup>1</sup>*

<sup>1</sup>Instituto Nacional de Metrologia, Qualidade e Tecnologia - Inmetro, Duque de Caxias, Brasil,  
[rrmachado@inmetro.gov.br](mailto:rrmachado@inmetro.gov.br)

<sup>2</sup>National Physical Laboratory – NPL, Teddington, UK

### Abstract:

This paper describes the procedures and the tests that have been carried out to characterize and to qualify two specific designs for Dynamic Calibration Devices (DCDs) which are intended to be used in the calibration of dynamic force testing machines in accordance with ISO 4965.

The dynamic and static behaviour of these DCD structures were analyzed by Finite Element Method (FEM), followed by preliminary experimental characterization tests. Results of these measurement practices are shown and the proposed design is discussed in regard to their application to the standard's requirements.

**Keywords:** Dynamic force, Calibration, ISO 4965, DCD design.

### 1. INTRODUCTION

The requirement for dynamic force measurements is constantly increasing due to a number of factors which come mainly from industrial and research applications. Nowadays users are convinced that the force transducer used in dynamic force measurement should be calibrated not only statically but also dynamically [1, 2]. Consequently, the use of a system capable of carrying out force calibration dynamically became necessary.

The ISO/TC 164 committee is working on the revision of the standard ISO 4965, which specifies the procedure for calibrating fatigue testing machines dynamically. The latest draft documents, ISO/FDIS 4965-1 [3] and -2, require the calibration to be performed using a Dynamic Calibration Device (DCD) such as a reference test piece. However, there is a need for the specification of these devices and that need forms the focus of this research.

Following on from this research, the authors present a study on the suggested design for the DCD, to be applied within a range of compliance. Two DCDs were developed and constructed to perform these tests and be able to calibrate dynamic force testing machines. The paper presents the procedures and some preliminary tests developed for the characterization and qualification of this DCD design.

### 2. METHODOLOGY

A main design is proposed for the DCDs and detailed with description of its basic mechanical principles of operation. A sequence of experimental and theoretical simulation tests should be carried out for proof of concept.

Previous analyses by FEM were done at the first step for the whole assembly structure to predict its behaviour. Static deformation analyses evaluated values for the project compliances to be achieved. Also, the tension distribution should be evaluated to guarantee the integrity of the structure.

Modal analysis of the structure assembly was also carried out for fundamental resonant frequencies, which shall be compatible with the calibration frequencies to be used.

Based on this proposed design, two DCDs were constructed and subjected to practical static and dynamic tests within the sequence of steps to approach the most important parameters for their characterization as prototypes.

Therefore, the real deflection and compliance can be measured, comparing with FEM results and set as the nominal value for the DCD.

Force alignment during application through the device should be also checked in order to work as reference for adjustments to be done between parts and couplings of the assembly.

After static studies, the devices should be tested under dynamic conditions. Inertial contributions should be also measured and might be included as correction factor during calibration.

Finally, the DCDs shall be able to make the first tests as reference sensor for the dynamic calibration of a testing machine. Experimental results obtained during the calibration procedure should be plotted and discussion should be also presented in the final part of the paper.

### 3. DESIGN OF THE DCD

The draft standard ISO/FDIS 4965-1 does not point a direction for the principle of functionality of a DCD. The Standard only determines that the DCD shall be a strain-gauge replica test-piece, what means it will maintain the characteristics of mass and compliance of a chosen test

piece to be tested in the machine, or it must be a proving device with known compliance. The DCD shall be able to produce an electrical signal proportional to the force applied on the machine under some specific mechanical condition.

The Standard also calls for the need of a DCD Instrumentation definition, issue that is treated specifically in part-2 [4] of the standard and will not be on focus on this article.

The authors propose the use of a dynamic force transducer as the sensor element for the measurements, once it has well-known behaviour in terms of stability, metrological characteristics and nominal values. However, these types of force transducers are very stiff to be used as the single component between grips of the machine to be calibrated, and do not respond accordingly to dynamic cyclic force application. For that reason, the DCD shall have some mechanical parts attached to the transducer in order to reduce the stiffness of the overall device.

When using method B of the ISO/FDIS 4965-1, the main functional characteristic of a DCD is its compliance response under axial applied forces. Therefore, the design of these devices should be based on the range of stiffness/compliance proposed to be covered by the calibration procedure. Dyson et al [5] presented some designs using studs and beams that could be attached and removed from the reference test piece, obtaining different values of stiffness for the assembly with the same instrumented sensor element.

The principle adopted by the authors is based on the Dyson design and it focus on the application of beams with different dimensions and material (Figure 1).

In order to identify an actual range of compliances and forces to be used in these first devices, a survey shall be run and these values shall be set for each DCD design.

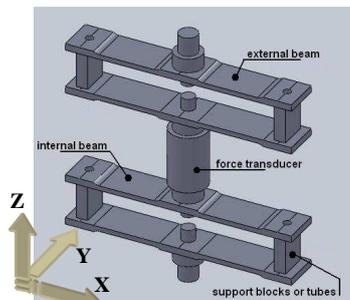


Figure 1. General design of a DCD using beams and tubes attached to a force transducer.

The FEA (Software SolidWorks) is applied to predict the static displacement and tension values for the project stiffness (compliance) to be achieved, while keeping the integrity of the structure. Two DCDs were constructed (Figure 2) trying to cover a range of compliance from  $7 \cdot 10^{-4}$  mm/N (DCD#1 - 148 x 40 x 4.3 mm) up to  $3 \cdot 10^{-3}$  mm/N (DCD#2 - 240 x 40 x 4.3 mm). Some preliminary experiments following the practical tests for static and dynamic parameters analyses were done for DCD#1, with lower compliance (higher stiffness).

The force range was chosen to be around 1500 N, compression only, with the force transducer used being an HBM U10M type, with 5 kN nominal force. That gives a

0.340830 mV/V Net reading corresponding to 1550 N. Instrumentation equipment HBM Quantum and Catman Software.



Figure 2. DCD#1 mounted in a the test machine and beams for DCD#2.

#### 4. COMPLIANCE VERIFICATION TEST

In a simpler approach, the analysis is done in one isolated pair of beams, which shall support the same load as the whole structure. Meanwhile, the deformation of the transducers and the connection tubes is very much lower than the total deformation of the whole DCD assembly. So, it was considered that the total deformation of the whole DCD is two times the central deformation of the isolated beam. Figure 2 shows the results of displacement simulation for one DCD pair of beams under a force value of 1550 N, highest value applied according to the Standard.

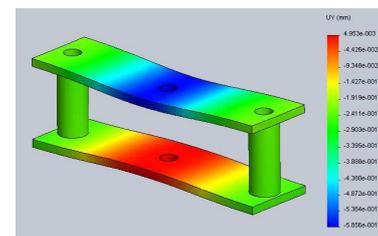


Figure 3. Axial displacement due to deformation of parts.

Therefore, the total displacement predicted for this DCD is 1.2 mm, what returns a linear compliance of  $7.56 \cdot 10^{-4}$  mm/N. Then, the DCD was tested statically in an electric fatigue test machine, but in both levels of force, at the top and at the bottom of the range. The compliance of the DCD was measured using the values of displacement of the actuator and also the force indication from the force transducer of the testing machine. The compliances calculated for both levels were different, such as  $7.5 \cdot 10^{-4}$  mm/N and  $9.8 \cdot 10^{-4}$  mm/N, for 1500 N and 500 N respectively.

This nonlinear behavior demands for a deeper study not only at the DCD but if the machine used has a linear relationship between applied force and displacement of the actuator, such as stated by the Standard. One real value of compliance should be assigned as the nominal reference of the DCD.

At this point it is necessary to highlight that, although the magnitude of compliance for each DCD is important, the most important point is that the final system covers the range proposed by the requirements. Therefore, some deviation can appear between the real measured compliance

and the designed compliance, such as we can check in the section about compliance test.

A stress analysis was also carried out and results are plotted in figure 4.

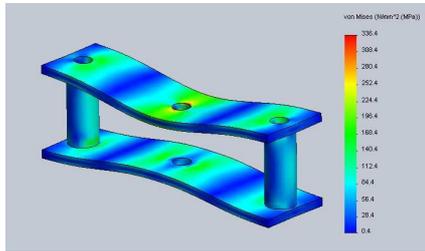


Figure 4. Stress analysis of the DCD.

Von Misses is applied and all the structure is under the considered yield stress limit for the SAE 1020 steel. Maximum values are found in the edges, what is minimized by machining of the parts.

FEA simulation was carried out for DCD#2 and compliance was calculated as  $3.3 \cdot 10^{-3}$  and stress analysis within the limits for the material.

## 5. FORCE ALIGNMENT VERIFICATION

The force alignment is important to be measured in order to look for the behavior of the DCD as maintainer of the axial force applied to it. Beams and tubes, if not well assembled, can generate parasitic components such as forces and moments.

The verification of these potential misalignments makes it necessary to adjust the position of the beams, the connection tubes and the torque applied to the bolts, as these are the elements which can contribute to this misalignment. The test was carried out through a GTM Multi-component transducer (5 kN nominal range) attached to the base of the DCDs. Forces were applied in 5x 100N steps and generated by dead-weights supported directly to the top of the sensors (Z axis). Directions are identified in figure 1.

Two main configurations were used; the initial checks are done with load applied to the assembly Force transducer + Multi components transducer, without beams (configuration#1). In the sequence, the beams are attached to the transducers and the same load steps are repeated for both DCDs (configuration#2).

Table 1 shows the results of the difference between force values measured in configurations #1 and #2 for DCD#1. Table 2 shows the results for DCD#2.

As it was expected, DCD#2 had larger components due to the lower stiffness of the beams assembly. However, relatively to the force applied, values are majority under 1% and instability has its cause in the instability of the structure. If values are higher than 1%, some analysis must be done order to check the tightened connections and even geometrical asymmetries.

Once it is impossible to completely remove misalignments, the actual intention is to minimize its effects and the degree of linear relationship between the axial applied force and the component forces is one of the parameters to be listed to check if there is any click in the

structure or even a loosening of the torque applied to the elements.

Although the authors selected screws and nuts to be the fixation components, another configuration can be the connections of the tubes, under a more robust design, by interference adjust. This can bring more stability to the assembly.

Table1. Differences in force between config#1 and config#2 (DCD#1). Values are in N.

| Force (Z axis) | Fx  | Fy   | Fz   | Mx  | My   | Mz   |
|----------------|-----|------|------|-----|------|------|
| 100            | 0.0 | -1.0 | 0.0  | 0.5 | -1.3 | 0.0  |
| 200            | 0.2 | -1.1 | -1.0 | 1.0 | -1.1 | -0.1 |
| 300            | 0.8 | -2.4 | 1.0  | 1.6 | -2.6 | -0.1 |
| 400            | 0.9 | -3.3 | 0.0  | 2.7 | -3.0 | -0.2 |
| 500            | 0.5 | -4.3 | 3.0  | 3.7 | -5.2 | -0.3 |

Table2. Differences in force between config#1 and config#2 (DCD#2). Values are in N.

| Force (Z axis) | Fx   | Fy  | Fz   | Mx   | My   | Mz  |
|----------------|------|-----|------|------|------|-----|
| 100            | -1.8 | 0.7 | -1.8 | -1.1 | -0.1 | 0.0 |
| 200            | -2.4 | 1.8 | -2.0 | -1.8 | 0.8  | 0.0 |
| 300            | -2.7 | 3.0 | -1.0 | -3.5 | 1.1  | 0.0 |
| 400            | -2.2 | 4.2 | -3.0 | -4.6 | 2.1  | 0.0 |
| 500            | -1.5 | 4.2 | 0.0  | -8.2 | 2.9  | 0.0 |

## 6. DYNAMIC TESTS

Modal analysis of the structure assembly is also carried out with FEA for fundamental resonant frequencies. For that, a simulation of the whole assembly was performed, including the application of different materials for the transducer and for the connection tube elements. DCD#1 simulation was evaluated and the first axial (Z direction) mode was found at 205 Hz, same result achieved by practical analyses of the structure (204 Hz).

After the static studies, the devices should be tested under dynamic conditions. The evaluation of the sensitivity response of the DCD under the frequency range of interest and also the necessary characteristics of the acquisition system must be specified to guarantee confidence measurements. ISO 4965 part 2 refers specifically to the verification of the instrumentation components but it will not be in total focus of this study.

Trying to cover basic dynamic characteristics of the DCD, the transducer was tested under dynamic conditions and using the same instrumentation to be applied during its application in calibration of fatigue test machines. Also, the interpolation curve obtained from the static force calibration of the transducer was used to give force values to the readings.

For that, transducer of DCD#1 was tested under different acceleration values, applied through a hydraulic shaker, and acceleration measured by an accelerometer attached to the top of the assembly. A pair of beams was used under the transducer in order to play as an isolator from the shaker main table (Figure 5).

The values of the reference force were evaluated by the multiplication of the total mass attached to the transducer (2x 10 kg pieces + rod) by the acceleration measured.

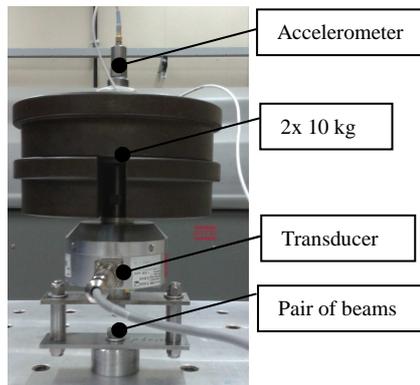


Figure 5. Assembly mounted on the shaker.

Table 3 shows the results for some force levels applied to the transducer. These values were obtained with the variation of frequency and amplitude of the shaker.

Table 3. Dynamic force comparison

| Freq. | Transducer | Reference | Deviation |
|-------|------------|-----------|-----------|
| Hz    | N          | N         | %         |
| 2     | 81.16      | 81.77     | -0.75     |
| 3     | 150        | 149.11    | 0.60      |
| 4     | 200        | 198.95    | 0.53      |
| 5     | 310        | 309.43    | 0.18      |
| 7     | 400        | 396.82    | 0.80      |
| 10    | 560        | 539.73    | 3.76      |
| 15    | 439.4      | 415.30    | 5.80      |

Values of force from the transducer were obtained through the FFT analysis. Results are in a good behavior once this can be considered not an assembly with the best characteristics and adjustments for a dynamic calibration of transducers. Results are useful to validate, even if in a simple approach, the instrumentation applied.

## 7. CALIBRATION TESTS

With dynamic and static behaviors well defined for a DCD design, it is time to make first tests for the dynamic calibration of a testing machine. At this point, the characteristics of the acquisition system and processing software are described and also integrate the whole reference kit.

One test was done in a fatigue test machine in the peak to peak range of 1000 N varying from 500 N up to 1500 N. The proceeding was according to the Standard and some highlights can be shown.

A frequency sweep was done from 1 Hz up to 15 Hz. The Standard determines that the values of peak and valley must be acquired from the machine force readings ( $F_i$ ) and also from the DCD electrical values ( $i_{DCD}$ ). This constitutes a slow routine of programming and a complex algorithm for linking different software programs.

Just in order to compare methodologies, Table 4 shows a comparison between the methods applied to verify each

peak and valley to evaluate their mean values and an FFT analysis applied to the same data. Fifty cycles were carried out for each frequency step with a 150 Hz acquisition rate and no filter applied.

Table 4. Dynamic force comparison between methods

| Freq. | FFT Amp. | Peaks and Valleys | Deviation |
|-------|----------|-------------------|-----------|
| Hz    | mV/V     | mV/V              | %         |
| 1     | 0.1098   | 0.109616          | 0.17      |
| 2     | 0.10944  | 0.109489          | -0.045    |
| 3     | 0.10958  | 0.109294          | 0.26      |
| 4     | 0.10924  | 0.108743          | 0.46      |
| 5     | 0.10403  | 0.105259          | -1.17     |
| 6     | 0.10815  | 0.107377          | 0.72      |
| 7     | 0.10994  | 0.106332          | 3.39      |
| 8     | 0.10641  | 0.105396          | 0.96      |
| 9     | 0.10521  | 0.104317          | 0.86      |
| 10    | 0.10466  | 0.104631          | 0.027     |
| 11    | 0.10612  | 0.101574          | 4.48      |
| 12    | 0.075373 | 0.10045           | -24.96    |
| 13    | 0.09631  | 0.098558          | -2.28     |
| 14    | 0.093628 | 0.095394          | -1.85     |
| 15    | 0.09558  | 0.094655          | 0.98      |

The results show there is not a tendency in the results of deviation but in some frequencies, due to the combination with the acquisition rate, some high deviation can be found, for example the 12 Hz step.

Another interesting observation is the tendency of lost of load during increment of frequency. This lost of load is the main object of study of the Standard. So, the proceeding is to calculate the correction factor (C) as the rate between the variation  $\Delta F_{DCD}$  and  $\Delta F_i$ . Table 5 depicts two values of frequencies from the range, being 2 Hz at the beginning and 14 Hz close the end, and demonstrates how the factor C reacts in the machine.

Table 5. Correction factor

| Hz | $\Delta F_{DCD}$ (N) | $\Delta F_i$ (N) | C      |
|----|----------------------|------------------|--------|
| 2  | 995.2102             | 998.3609         | 0.9968 |
| 14 | 865.7361             | 968.0787         | 0.8942 |

According to the Standard, the correction factor within 1%, case of 2 Hz, it does not need to be corrected, but values above 10%, case of 14 Hz, the calibration is considered failed and the machine should not be used above this range.

## 8. DISCUSSION

This paper described, preliminarily, some results obtained from static and dynamic test forces carried out on a DCD prototype which are intended to be used in the calibration of dynamic force testing machines in accordance with ISO 4965. It was performed both numerical and experimental tests.

Modal analysis was applied by FEM and has shown the static and dynamic behaviour of the DCD structures. The FEA was also applied to predict the static displacement and

tension values for the project stiffness to be achieved, while keeping the integrity of the structure.

Experimentally the DCDs were tested under static forces condition in order to check their values of compliance that should be the one used as the nominal reference of the DCD. Dynamic experimental tests should be carried out to evaluate the compliance behavior of the DCD under the frequency range of interest.

Also the proceeding of the standard was used and deserves some attention in what refers to the use of the DCDs and the instrumentation applied. Below some points of future studies are highlighted:

- The DCD compliance and stiffness must be evaluated theoretically in a first approach, but due to deviations from the practical assembly, fixation components, material used in the beams and even the linearity of the system, values can differ a little must be checked experimentally.
- ISO 4965 part 2 must be exploited but not considering only the instrumentation components but also the transducer, or strain-gauge piece itself. Maybe not a complete calibration but some kind of dynamic analysis must be done with test piece.
- Acquisition rate and filters settings are interesting points of study and might be source of errors during the use of the equipment.
- The method used to calculate peaks and valleys must be analyzed through comparisons and indicates a very limiting factor due to time it can take to be evaluated in a complete sweep of frequencies.
- Although not shown in the study presented, the method B involves the analysis of two DCDs to be combined and returns a value of compliance envelop.
- In the same way, due to differences in mass and compliance, the second DCD might change the main resonant frequency of the whole system and some frequency steps calculated for the first DCD can now be within an invalid sub-range.

## 9. REFERENCES

- [1] Link, A., Glöckner, B., Schlegel, C., Kumme, R., Elster, C., & Bundesanstalt, P.-T. (2009). "System Identification of Force Transducers for Dynamic Measurements  $f(t)k$ ". IMEKO 2009, 205-207.
- [2] R. S. Oliveira, R. R. Machado, A. Knott, I. V. Caminha, C. R. Roesler "Dynamic Calibration of a Fatigue Testing Machine on Orthopaedic Implants in Accordance with the ISO 4965-1", 21st Conference on Measurement of Force, Mass and Torque, Pattaya, Thailand, 2010.
- [3] ISO/FDIS 4965-1:2011 Metallic materials - Fatigue testing - Uniaxial dynamic force calibration - Part 1: Testing systems.
- [4] ISO/FDIS 4965-2:2011 Metallic materials - Fatigue testing - Uniaxial dynamic force calibration - Part 2: Dynamic calibration device (DCD) Instrumentation.
- [5] B. F. Dyson, M. S. Loveday, M. G. Gee, Dynamic Calibration of Machines, CH. 4, 1992.