STRESS ANALYSIS OF STRAIN-GAUGE BALANCE FOR «LIFTING BODY» MODELS AERODYNAMIC TESTS

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Abstract

The considered type of strain-gauge balance was designed and manufactured for wind tunnel testing the models of descent vehicles type “lifting body” (the examples are the descent vehicles “Clipper” [1] and “Pre-X”).

Six-component combined high-load strain-gauge balance described in [2] is the prototype for the balance. The balance has a mono-piece structure and includes two elements:

- a longitudinal octahedral beam connected to a tail sting in the wind tunnel and supplied with strain-gauges placed on the beam facets for direct reading Y normal and Z side forces, Mx roll, My yaw and Mz pitch moments;
- a cylindrical piston placed coaxially on the beam cantilever and supplied with X- longitudinal force strain-gauge dynamometer incorporated in the piston walls.

3D-model of the balance body was designed using Solid Works codes. Finite-element analysis (FEA) of the balance body stressed state was fulfilled using PATRAN/NASTRAN codes. Calculations were provided for several cases of the balance loading, including loading by single force/moment component and simultaneously six force/moment components.

Calculation results on the stressed state parameters for several characteristic points of the balance (points of strain-gauges’ positions) were compared with measurement results.

Introduction

A peculiarity of aerodynamics of the “lifting body” space descent vehicles (such as “Clipper” [1] and “Pre-X”) is lift-to-drag ratio ~0,9-1,0 at the trim angle of attack ~20°-40° that provides required side manoeuvre (up to 500 km) at descent trajectory. The strain-gauge balances intended for testing such type flight vehicle (FV) models in wind tunnels have to provide, along with the above lift-to-drag ratio, high values of roll, yaw and pitch moments required for controlling the FV manoeuvre, to be rigid enough, compact and also do not introduce noticeable distortion in a flow around tested model.

Present requirements imposed on accuracy of FV aerodynamic tests are very strong and can be provided by strain-gauge balances with accuracy 0,05-0,1. Creation of a balance with such accuracy demands special approaches for each stage of the work: design, materials and component parts selection, springy body fabrication, measuring circuits assemblage and adjustment, balance calibration.

At the design stage FEA of the balance stressed state under action of the loads to be measured is of great importance. It permits to optimize a complicated spatial structure of the balance taking into consideration required sensitivity of the balance measuring elements with required rigidity and strength of the balance structure. The design codes Solid Works, CATIA, Pro Engineer/Pro Mechanica, PATRAN/NASTRAN, ANSYS [2-7] are used for 3-D model of the balance springy body construction and loading state simulation.

Verification of numerical technique is a goal of the presented work. A sample of “lifting body” strain-gauge balance was designed, manufactured and examined. Measured values on the balance stressed state characteristics were compared with computational data.

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**Section I**

**Strain-gauge structure, models and design situations**

The prototype of the balance under development is a six-component combined strain-gauge balance with tubular and rod springy elements [2]. The balance springy body consists of two parts: a longitudinal tetrahedral beam connected to a tail wind tunnel sting, and a hollow cone-cylinder shell placed coaxially on the beam cantilever. Strain-gauge transducers (SGT) are placed on the beam facets and wired in bridge measuring circuits for direct reading of Y and Z forces, My and Mz moments. The shell incorporates strain-gauge dynamometers of X force and Mx moment; they present as two rings – groups of springy parallelograms. The parallelograms of the first ring – with shear sensing elements and SGT on them – are used for reading X force, and the parallelograms of the other ring – with longitudinal springy bending elements and SGT on them – for reading Mx moment. The examined model is mounted on the shell surface, and all sensing elements of the balance are arranged inside the model.

The “lifting body” balance was developed taking into consideration relationship between dimensions and combination of the load components specific for “lifting body” FV models; it consists of springy body and cowling. The balance 3-D model shown in Fig.1 was designed using Solid Works.2007.

![Fig.1](image)

1 – longitudinal beam with sensing elements for Y, Z forces and Mx, My, Mz moments; 2 – piston with sensing elements for X-force; 3 – cowling; 4 – strain-gauge transducers.

On the contrary to the prototype [2], the springy body is one-piece, and longitudinal springy beam is octahedral (like the balance [8]) on which facets additional SGT of measuring circuits for reading Mx moment are mounted. The shell with X force dynamometer is manufactured as a piston placed coaxially on the beam cantilever; the piston surface is used for fixing the model to be tested.

During tests the balance is mounted downwards the wind tunnel flow, and the tested model is fixed on the piston at the angle respective to the trajectory angle of attack (~20°-40°), so the piston is wholly inside the model, and some sensing elements placed on the longitudinal beam in the model wake is shielded from the flow effect by cylindrical or rhomboid cowling.

FEA model of the balance springy body constructed with the help of MSC.PATRAN / NASTRAN.2005 is illustrated in Fig.2. The model is constructed of tetrahedrons with 10 nodes (4 vertexes and 6 nodes in the middle of the edges) with 3 degrees of freedom in each node. The model consists of 188294 elements and 281938 nodes. The problem dimension is more than 1,6 million degrees of freedom. Maximal edge length equals 2 mm, minimal length (in places of the grid crowding) – 0,05 mm.
SGT R1...R36 are placed on the springy body in five cross sections (Fig.3) and wired by the bridge measuring circuits (Fig.4).

Table 1 presents parameters of the balance loading cases examined in the paper.

<table>
<thead>
<tr>
<th>component</th>
<th>X</th>
<th>Y</th>
<th>Z</th>
<th>Mx</th>
<th>My</th>
<th>Mz</th>
</tr>
</thead>
<tbody>
<tr>
<td>single loading</td>
<td>1N</td>
<td>1N</td>
<td>1N</td>
<td>1N·m</td>
<td>1N·m</td>
<td>1N·m</td>
</tr>
<tr>
<td>simultaneous loading *</td>
<td>3000N</td>
<td>3000N</td>
<td>1000N</td>
<td>10N·m</td>
<td>60N·m</td>
<td>150N·m</td>
</tr>
</tbody>
</table>

* simultaneous loading is a nominal measuring range of the balance.

Design point of force application is located at 27 mm from the rear end of the piston (Fig.2). Computations were executed for the linear model of stressed state (\(\sigma = E \cdot \varepsilon\), here \(\sigma\) – stress, \(\varepsilon\) – relative strain, \(E\) – modulus of elasticity). Properties of the balance body material (ultimate stress \(\sigma_b = 1,5 \cdot 10^9\)Pa, \(E = 1,8 \cdot 10^{11}\)Pa, Poisson's ratio \(v = 0,3\)) correspond to steel type Armco 17-4PH [8].
order to restrain the level of loads the limiting value $\sigma$ was the following: for $SGT \leq 3 \cdot 10^8 \text{Pa}$ ($\varepsilon \sim 1.7 \cdot 10^{-3}$), and for other structural elements $\leq (0.6 - 0.7) \cdot 10^9 \text{Pa}$ ($\varepsilon \sim 5.3 \cdot 10^{-3}$).

![Fig.5](image)

The strain-gauge balance sample manufactured for verification tests is shown in Fig.5. The same way as [3], the stressed state parameters $\varepsilon$ or $\sigma$ of the balance springy body were measured at the characteristic points – the spots of $SGT$ location at dynamometers measuring circuits $X(R1,R2), Y(R17,R18), Z(R21,R22), Mx(R25,R26), My(R31,R32), \text{and } Mz(R35,R36)$.

**Computational results**

The results obtained with one-component loading are the following. With loading $X=3000\text{N}$ the balance deformation does not exceed $\sim 0.016 \text{ mm}$, and maximal local value $\sigma$ (at the surface of the dynamometer $X$ sensing element) equals $\sim 1.7 \cdot 10^8 \text{Pa}$. With loading $Y=3000\text{N}$ the balance face plane displacement is within $\sim 0.5 \text{ mm}$, and $\sigma$ value at the $Y$ dynamometer $SGT$ location equals $\sim 3 \cdot 10^8 \text{Pa}$; correspondingly, with loading $Z=1000\text{N}$ these values for $Z$ dynamometer are equal to $\sim 0.6\text{mm}$ and $\sim 2.5 \cdot 10^8 \text{Pa}$. With loading $Mx=10\text{N}\cdot\text{m}$ the piston rear end displacement does not exceed $\sim 0.04 \text{ mm}$, and $\sigma$ value at the $Mx$ dynamometer $SGT$ location does not exceed $2 \cdot 10^7 \text{Pa}$. With loading $My=60\text{N}\cdot\text{m}$ the balance face plane displacement $\sim 0.8 \text{ mm}$, and $\sigma$ value at the $My$ dynamometer $SGT$ location $\sim 2.7 \cdot 10^8 \text{Pa}$. Correspondingly, with loading $Mz=150\text{N}\cdot\text{m}$ these values are equal to $\sim 0.5 \text{ mm}$ and $\sim 6 \cdot 10^8 \text{Pa}$.

Computational results for complex simultaneous loading of the balance $X=3000\text{N}, Y=3000\text{N}, Z=1000\text{N}, Mx=10\text{N}\cdot\text{m}, My=60\text{N}\cdot\text{m}, Mz=150\text{N}\cdot\text{m}$ show that the balance face plane deflection does not exceed $1.4 \text{ mm}$, and maximal $\sigma$ value in the structure elements (local concentration near holes of $X$ dynamometer sensing element) does not exceed allowable values $\sim 0.7 \cdot 10^9 \text{Pa}$.

Let’ note some features of stress fields at the points of $SGT$ location. The stress field for $X$ dynamometer sensing element with $X=1\text{N}$ at the points of $SGT$ $R1...R16$ location is shown in Fig.6. It is seen here that the compression–tension stress field along $SGT$ principal axes is uniform enough, and with $X=3000\text{N}$ the value of $\sigma$ at $SGT$ is $5.4 \cdot 10^7 \text{Pa}$.

![Fig. 6](image)  

**Fig. 5**

**Fig. 6**

**Fig. 7**
Stress field for the Y dynamometer sensing element with \( Y = 1 \text{N} \) at the SGT R17 location is shown in Fig.7. It is seen that the stress field along SGT principal axis varies on the length 3mm (over 15%) in proportion to the distance from the point of force application, and with \( Y=3000\text{N} \) the average value \( \sigma \approx 6\cdot10^7\text{Pa} \). Analogous situation takes place at SGT R18...R20 for Y force action and SGT R21...R24 for Z force action.

Typical stress distribution for My/Mz dynamometer SGT is illustrated in Fig.8 (SGT R35 for Mz=1N⋅m). Stress field nonuniformity (within 2-3%) is related to local peculiarities of construction (relation between dimensions of springy element and SGT, closeness to location of change in springy element section and shape, etc.)

Typical stress distributions for SGT of Y, Z, My, Mz dynamometers under effect of bending moment in the planes coinciding with SGT planes are illustrated in Fig.9. In this case one part of each SGT symmetrically to the neutral axis of bending is exposed to compression and another – to tension. If SGT axes of symmetry do not coincide with the neutral axis then integral value \( \sigma \) of each SGT is nonzero, that is one of sources of the corresponding component influence.

Stress fields at SGT R1 of X dynamometer under effect of Z force and My moment in SGT plane are shown in Fig.10 and 11; they demonstrate significant nonuniformity of the fields: difference of \( \sigma \) value at the length ~ 1mm lies from twofold (under Z force action) to tenfold (under My moment action).

Integrally, analyzing peculiarities of stress fields at SGT location note the following:
- under effect of the load components to be measured by corresponding SGT, stress field nonuniformity on SGT length does not exceed ~15-20%;
- under effect of the load components that are not to be measured by given SGT, stress values on SGT length/width may differ significantly (in times) and have opposite signs.

Due to observed nonuniformity it is necessary to choose carefully the positions of SGT location and SGT dimensions and to provide accurate and symmetric mounting of SGT that influence greatly on compensation level of the components’ interference in the measuring circuits and residue of this influence.
Comparison of calculation and measurement data

Table 2 presents the calculation and measurement data on the relative strain $\varepsilon$ at the points of SGT location for the balance sample sensing elements of force and moment components at one-component loading.

<table>
<thead>
<tr>
<th>$\varepsilon$</th>
<th>$X(R1,R2)$</th>
<th>$Y(R17,R18)$</th>
<th>$Z(R21,R22)$</th>
<th>$Mx(R25,R26)$</th>
<th>$My(R31,R32)$</th>
<th>$Mz(R35,R36)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>calculation</td>
<td>$0,107\times10^{-6}$</td>
<td>$0,196\times10^{-9}$</td>
<td>$1,75\times10^{-8}$</td>
<td>$0,77\times10^{-8}$</td>
<td>$2,03\times10^{-8}$</td>
<td>$2,04\times10^{-8}$</td>
</tr>
<tr>
<td>measurement</td>
<td>$0,104\times10^{-6}$</td>
<td>$0,2\times10^{-9}$</td>
<td>$1,7\times10^{-8}$</td>
<td>$0,6\times10^{-8}$</td>
<td>$2,4\times10^{-8}$</td>
<td>$1,62\times10^{-8}$</td>
</tr>
<tr>
<td>$Y = 1N$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>calculation</td>
<td>$0,159\times10^{-6}$</td>
<td>$0,106\times10^{-10}$</td>
<td>$0,2\times10^{-8}$</td>
<td>$9,8\times10^{-9}$</td>
<td>$0,762\times10^{-9}$</td>
<td>$4,08\times10^{-9}$</td>
</tr>
<tr>
<td>measurement</td>
<td>$0,151\times10^{-6}$</td>
<td>$0,114\times10^{-10}$</td>
<td>$0,3\times10^{-8}$</td>
<td>$12,0\times10^{-9}$</td>
<td>$0,7\times10^{-9}$</td>
<td>$4,0\times10^{-9}$</td>
</tr>
<tr>
<td>$Z = 1N$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>calculation</td>
<td>$6,897\times10^{-9}$</td>
<td>$0,104\times10^{-10}$</td>
<td>$0,21\times10^{-6}$</td>
<td>$0,27\times10^{-8}$</td>
<td>$0,822\times10^{-8}$</td>
<td>$0,15\times10^{-10}$</td>
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<tr>
<td>measurement</td>
<td>$9,75\times10^{-9}$</td>
<td>$0,0216\times10^{-9}$</td>
<td>$0,32\times10^{-8}$</td>
<td>$0,943\times10^{-8}$</td>
<td>$0$</td>
<td></td>
</tr>
<tr>
<td>$Mx = 1Nm$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>calculation</td>
<td>$5,267\times10^{-6}$</td>
<td>$0,182\times10^{-8}$</td>
<td>$0,191\times10^{-8}$</td>
<td>$4,8\times10^{-6}$</td>
<td>$0,33\times10^{-8}$</td>
<td>$1,1\times10^{-9}$</td>
</tr>
<tr>
<td>measurement</td>
<td>$4,038\times10^{-6}$</td>
<td>$0,2\times10^{-8}$</td>
<td>$0,221\times10^{-8}$</td>
<td>$5,28\times10^{-6}$</td>
<td>$0,203\times10^{-8}$</td>
<td>$1,92\times10^{-7}$</td>
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<tr>
<td>$My = 1Nm$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>calculation</td>
<td>$0,101\times10^{-7}$</td>
<td>$0,145\times10^{-8}$</td>
<td>$1,096\times10^{-5}$</td>
<td>$0,33\times10^{-7}$</td>
<td>$11,3\times10^{-9}$</td>
<td>$1,2\times10^{-9}$</td>
</tr>
<tr>
<td>measurement</td>
<td>$0,111\times10^{-7}$</td>
<td>$0$</td>
<td>$1,226\times10^{-5}$</td>
<td>$0,365\times10^{-5}$</td>
<td>$10,75\times10^{-8}$</td>
<td>$3,53\times10^{-8}$</td>
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<tr>
<td>$Mz = 1Nm$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>calculation</td>
<td>$5,314\times10^{-6}$</td>
<td>$5,462\times10^{-9}$</td>
<td>$0,115\times10^{-9}$</td>
<td>$0,161\times10^{-8}$</td>
<td>$0,919\times10^{-8}$</td>
<td>$5,61\times10^{-8}$</td>
</tr>
<tr>
<td>measurement</td>
<td>$5,71\times10^{-6}$</td>
<td>$6,99\times10^{-6}$</td>
<td>$0,196\times10^{-8}$</td>
<td>$0,147\times10^{-8}$</td>
<td>$0,38\times10^{-7}$</td>
<td>$5,73\times10^{-8}$</td>
</tr>
</tbody>
</table>

The data in Table 2 printed in bold at the table diagonal demonstrate a good agreement of calculation and measurement results on $\varepsilon$ that characterize sensitivity of the balance measuring circuits to corresponding load components to be measured. And discrepancy between the calculation and measurement data does not exceed ~10%.

Discrepancies of $\varepsilon$ values that characterize mutual interference of the components up to ~30% (in the most cases) can be explained by inaccurate arrangement of SGT with noticeable stress field nonuniformity under action of the load components that are not to be measured by corresponding dynamometer, and also by calculation errors due to, in particular, partition of the balance springy body into elements.

Conclusion

Comparison of computational data on stress state characteristics of the “lifting body” balance springy body with the results measured on the balance sample demonstrates, in the whole, a good agreement. Therefore it is reliable to use the numerical simulation during the given type strain-gauge balance design at the stages of the balance structure formation and optimization taking into consideration various design requirements.

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References


