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STIRRER BLADE OPERATIONAL LOADS IDENTIFICATION

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ABSTRACT: This paper presents a particular problem of safe-life prediction of the stirrer blade – a work of research team of the Czech Technical University (CTU). Only internal forces on input and output shaft of single blade were determined experimentally, in operation. The pressure field acting on blade surface was assessed as a linear combination of 6 unit load states of finite element (FEM) model of it. Coefficients of linear combination are to minimize difference between the internal forces determined experimentally and by FEM.

1 Introduction

This paper describes a particular problem of safe-life prediction of rubber mixture stirrer blade. The interest is focused at operational loads identification and numerical stress analysis of the stirrer blade. Complete task, is being solved by a team of researchers working at Department of Elasticity and Strength of materials of Faculty of Mechanical Engineering, CTU in Prague in collaboration with manufacturer of the stirrer blades.

The task was formulated as to evaluate safe-life increment in consequence of internal surface of blade cast improvement. It requires to determine strain or stress field, depending on time, during the typical stirring process. Operational loading spectra are the basic input of the task. Inertial forces of the blade itself may be neglected in comparison to the rubber mixture resistance. Assuming this, the motive torsional momenta in the input shaft is to be equilibrated by the momenta of resistance forces. Remaining components of resistance forces are caught by bearings (fig. 2). Rubber mixture rheological response is not simple. It may change during the stirring process depending on time of stirring, temperature and other parameters. That's why the idea to determine loads experimentally was preferred. As the stirring function of blades was found satisfactory, no external surface changes are required. This fact enabled to identify the forces on the original stirrer, directly in operation, without manufacturing the improved one. Unfortunately, no quantities, except internal forces on input and output shaft, could be experimentally determined under operational conditions. The forces measurements is described in [1]. Resistance forces are assumed in form of surface normal forces (pressures) on the stirrer blade external surface. They are constructed as a linear combination of "unit" surface loads (see fig. 2) the coefficients of which are determined to generate internal forces corresponding with measured ones.

point B, $M_{oy,C}$ and $M_{ox,C}$ in site C, torsional moment $M_{kz,B}$ measured in site B and axial (membrane) force $F_{z,C}$ in point C (see Fig. 2), all of them as a function of time during single stirring process were determined via strain-gauges measurements. This six quantities frames the base of identification.
Six "unit" load states S_1–S_6 (see Fig. 2) were outlined from simple idea about pressure distribution on stirrer blade surface

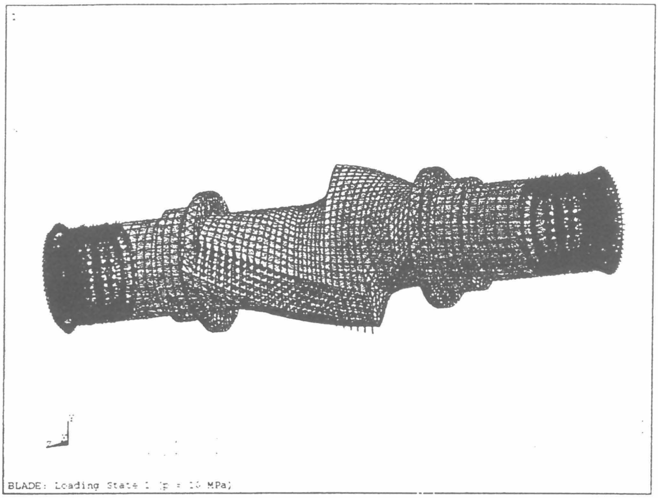

Fig. 2: Loading state S_1.

Integral internal forces are determined from nodal internal forces in cross-sections B and C computed by FE analysis for this load states. Resulting stirrer–blade load is assembled as a linear combination of load states S_1–S_6. Internal forces associated to single load state are superscripted by load state number. For instance

$$M^{3}_{oy,B}$$

signs bending moment acting about y-axis determined from internal forces from load state S_3 in nodes associated to cross-section B. Resulting bending moment $M_{oy,B}$ is

$$M_{oy,B} = \sum_{i=1}^{6} M^{i}_{oy,B}\, q_i$$

Let's denote vector of computed internal forces as $\vec{M}$

$$\vec{M} = \left\{ \begin{array}{cccccc} M_{oy,B} & M_{ox,B} & M_{oy,C} & M_{ox,C} & M_{kz,B} & F_{z,C} \end{array} \right\}^{T} \quad , \tag{1}$$

vector of linear combination coefficients as

$$\vec{q} = \left\{ \begin{array}{cccccc} q_1 & q_2 & q_3 & q_4 & q_5 & q_6 \end{array} \right\}^{T} \tag{2}$$

and matrix of forces from "unit" loads in form

2 Modelling

Programm **ANSYS** was used on server *"jumbo.cis.vutbr.cz"* to do the analysis. Stirrer blade body is modelled in 3D continuum using isoparametric twenty-node elements. Brick elements (mapped) meshing possibility was preferred during geometric model creation at the same time with flexibility (to enable design changes in future). This demands lead to the using of macros structured as profile generator (2D) and the final body generator (3D). Input and output shafts are created interactively. FE mesh of final stirrer blade body is shown in Fig. 2. It consists of **19000** elements and **57000** nodes. Linear isotropic material satisfying Hook's law is quantified by Young's elastic modulus E = 210000 MPa and Poisson's ratio μ = 0.3. Motive torsional moment is modelled via tangential direction fixing of nodes associated with left shaft face. This produces nodal reactions resulting just into the motive torsional moment equilibrating (together with remaining reactions) resistance forces on the blade. All geometric boundary conditions are shown in Fig. 2.

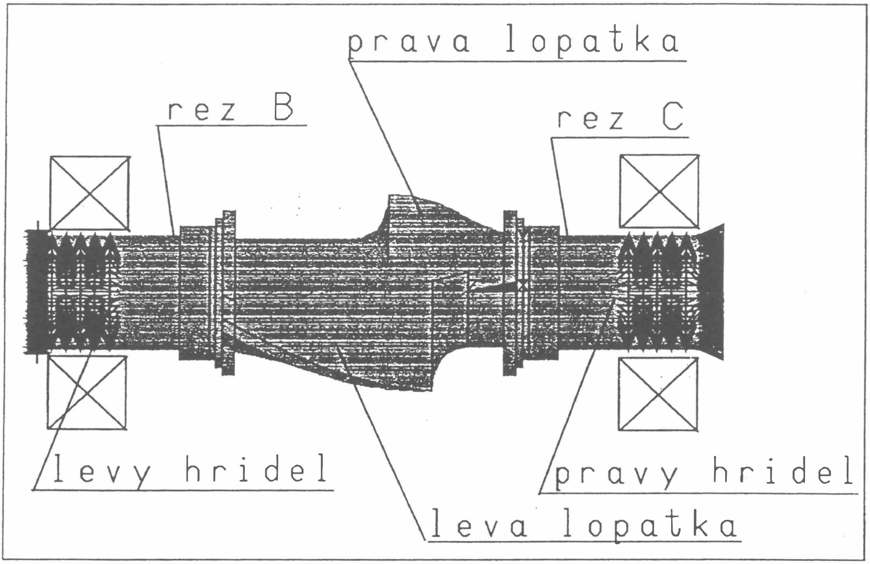

Fig. 1: Stirrer-blade model, kinematic conditions and loads.

3 Identification

The idea of identification of resistance forces is based on presumption of linearity. This allows us to express resulting fields of displacements and stresses as a linear combination of such fields obtained for single "unit" loads (procedure known as *principle of superposition*). Differences between computed and measured internal forces are to be minimized by coefficients of the linear combination.

Only several sites on shafts of the stirrer blade were suitable for strain-gauge installation. That's why integral internal forces (membrane and shear forces, bending and torsional moments) were used to compare computation and experiment. This eliminates local influences and decreases demands to the FE mesh density under strain-gauges. Bending moments $M_{oy,B}$ and $M_{ox,B}$ in

$$\underline{\mathrm{A}} = \left\{ \begin{array}{cccccc} M^1_{oy,B} & M^2_{oy,B} & M^3_{oy,B} & M^4_{oy,B} & M^5_{oy,B} & M^6_{oy,B} \\ M^1_{ox,B} & M^2_{ox,B} & M^3_{ox,B} & M^4_{ox,B} & M^5_{ox,B} & M^6_{ox,B} \\ M^1_{oy,C} & M^2_{oy,C} & M^3_{oy,C} & M^4_{oy,C} & M^5_{oy,C} & M^6_{oy,C} \\ M^1_{ox,C} & M^2_{ox,C} & M^3_{ox,C} & M^4_{ox,C} & M^5_{ox,C} & M^6_{ox,C} \\ M^1_{kz,B} & M^2_{kz,B} & M^3_{kz,B} & M^4_{kz,B} & M^5_{kz,B} & M^6_{kz,B} \\ F^1_{z,C} & F^2_{z,C} & F^3_{z,C} & F^4_{z,C} & F^5_{z,C} & F^6_{z,C} \end{array} \right\} . \tag{3}$$

Then

$$\vec{M} = \underline{\mathrm{A}}\ \vec{q}$$

Matrix $\underline{\mathrm{A}}$ components values are determined from stress–fields computed by FE for single load states S_1–S_6. Coefficients of linear combination representing instantaneous loads in time t of single stirring process are determined from condition that vector $\vec{M}$ must equal to the vector $\vec{M}^E(t)$ of corresponding internal forces obtained from measurement in instant t. Vector $\vec{q}(t)$ of coefficients of linear combination may be obtained as solution of linear equations

$$\underline{\mathrm{A}}\ \vec{q}(t) = \vec{M}^E(t) \tag{4}$$

4 Solution

FE–analysis of load states was done as fully linear, using iterative "PCG" (preconditioned gradient) solver with precision 1×10^{-6}. Internal forces were determined using the theorem of dividing body on two separated parts by cross–section. By application of preceding procedure on all six load states we achieved matrix operator $\underline{\mathrm{A}}$ in form

$$\underline{\mathrm{A}} = 1.0 \times 10^7 \cdot \left\{ \begin{array}{rrrrrr} -1.761 & -3.677 & 1.263 & 2.260 & 0.001 & -0.000 \\ -0.181 & -0.041 & 0.194 & 0.678 & 0.000 & -0.000 \\ -3.337 & -5.536 & 0.561 & 1.841 & 0.001 & -0.000 \\ 0.452 & -1.330 & 0.794 & 0.174 & 0.000 & -0.000 \\ -8.788 & -5.187 & -5.113 & -2.958 & 0.006 & 0.003 \\ 0.018 & 0.010 & -0.017 & -0.010 & 0.000 & 0.000 \end{array} \right\} \tag{5}$$

5 Stirrer–blade stress under extremal loads

Coefficients of load states linear combination were evaluated in one of critical phases of stirring described in [1]. Vector of measured internal forces was evaluated as[1]

$$\vec{M}^E = \left\{ \begin{array}{cccccc} 57300000 & 0 & -53100000 & 0 & 60000000 & -636000 \end{array} \right\}^T \tag{6}$$

Linear equations set 4 solution was obtained as

$$\vec{q} = \left\{ \begin{array}{cccccc} 1.255 & 1.905 & 4.237 & 3.213 & 4627.5 & 7727.9 \end{array} \right\} \tag{7}$$

Displacement, strain and stress–fields were computed for previously determined combination of load states S_1–S_6. Largest displacement magnitude 0.416 mm acknowledges a'posteriori small displacement presumption. Von Mises'es equivalent stresses do not exceed value of 200 MPa in macroscopic volumes. Material is in plastic state in several sites on internal surface of right blade and in fillet area between right and left blades. Influence of this non–linearities in micro–volumes on global stress state may be neglected, however. It may be said, generally, that resulting

[1]forces are measured in [N] and moments in [N mm].

displacements and stresses under extremal load acknowledge the linear computational scheme. There are peak stresses between 250–300 MPa on internal surface of right blade (generally more stressed one) distributed near from leading edge. Left blade (significantly less stressed) has peak stresses about 150 MPa. Peak stresses locations correspond with peak points of bending moment in y–axis.

6 Conclusion

Stirrer blade loads identification is a particular solution included in complex task – safe life analysis of stirrer. From this point of view the output of presented solution is to be used as one of inputs for following analyses. Following conclusions may be outlined from results of presented work:

- FE mesh created compromises precision requirements and available computer and program capacity. Technique of submodelling may be used to increase the precision of local stresses.
- Designed set of loading states seems to be representative enough. Matrix operator $\underline{\mathbf{A}}$ (see equations 3 and 5) may be used for superposition coefficients evaluation in various phases of rubber–mixture stirring process. Time dependencies of stresses in critical localities of stirrer blade – the basic input of safe–life evaluation – will be determined this way in future.
- Superposition of critical phase of stirring process is described in section 5. Equivalent stresses in this phase may be considered as the worst stress in operation.

References

[1] J. Michalec and M. Růžička. Experimentální analýza namáhání hřídele hnětiče 250 L. VHČ, Strojní fakulta ČVUT, Praha, 1996.

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