EAN '97

35th International Conference on
Experimental Stress Analysis
June 4 - 6, 1997, Olomouc, Czech Republic

JUDGEMENT OF EFFECTIVE STRESS CONCENTRATION OF A LATHE CHUCK JAW FOR ITS UNLIMITED FATIGUE LIFE

Frémund J., Lašová V., Laš V., Plánička F., Václavík J.

The paper deals with a shape body influence on fatigue limit. An effective stress concentration of a lathe chuck jaw for its unlimited fatigue life is taken into account. Results obtained by numerical FEM ANSYS system and experimental resistance strain gage method are compared.

1. Introduction

The influence of body shape on fatige limit can be expressed by a fatigue notch factor or by FEM using a computational system. There was a problem to judge effective stress concentration of a lathe chuck jaw for its unlimited fatigue life. In literature from fatigue region there are for that body shape published fatigue notch factors only for a stepped flat bars with shoulder fillets subjected to tension or to bending, but in the plane of that bar. There was a task to solve that problem for mentioned above component.

2. Analysis of problem on a flat bar

The first step was to compare results of a numerical solution obtained by FEM ANSYS system and experimental analysis using electrical resistance strain gauge method for a simple model similar in profile to chuck jaw. Therefore a stepped flat bar with shoulder filets of thicknes 5 mm loaded by bending moment in the plane perpendicular to the plane of the body (Fig. 1) was taken into account. Material of the bar was a mild steel, Young's modulus $E = 2{,}1 \cdot 10^5$ MPa. Using linear regression analysis to fit least-squares line through scattered measured quontities and by extrapolation of strain component on the notch edge was determined bending stress $\sigma = 99$ MPa for loading force 50 N here. For the same shape of the body, material and loading conditions the magnitude of the stress determined numerically was 105,5 MPa. There is very good correspondence between both obtained magnitudes. Calculation and expreimental analysis of the real component were done in the next step.

3. Analysis of a real body

The chuck jaw was made from the steel 15241.7 ČSN 415241 with tensile ultimate strength $R_m = 1176$ MPa, yield point $R_e = 980$ MPa, Young's modulus $E = 2{,}1 \cdot 10^5$ MPa. It was loaded by a force $F = 720$ kN in its down and by $F = 200$ kN in its upper position respectively.

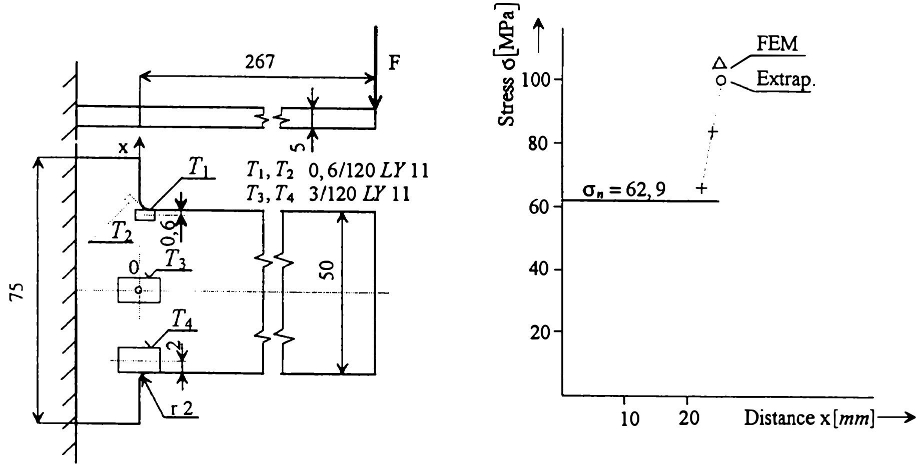

Fig. 1

3.1 Calculation using FEM

Results published here are for put in chuck jaw in its down position. A model for calculation was done by 6 and 8 nodes volume elements. The first calculation was realized at point of the surface where the jaw is supported by its guideway. Corresponding displacement component was restricted in boundary conditions, i.e. the jaw guideway was assumed to be absolutely rigid. Analysing obtained results was discovered extremely high magnitude of von Mises stress at one node in vicinity of recess. That stress magnitude was in all probability influenced by taken into account boundary condition when the jaw guidway was assumed absolutely rigid. Taking into account the fact, that the highest magnitudes of the von Mises stress are just along that rigid support it has been necessary to do new model of calculation to be nearer to reality. Therefore the boundary contitions along the support were asummed as a contact task.

The arragement of calculation model consists of the lathe chuck jaw and it guideway screwed on to a face plate as shown in Fig. 2. Contact stresses between the jaw and its guideway were taken into account. There are several possibilities to solve contact stresses using ANSYS system. The most corresponding to reality by our opinions is so called „General Contact“, which is expressed by 5 nodes contact elements of the jaw on its guideway planes. The constant rigidity, friction factor and some other constants (for example tolerance of allowable penetration of components and so on) can be used for normal and tangential directions respectively. Two-way contact was assumed.

Detail of a calculation model of a zone surrounding recess is shown in Fig. 3. The maximum stress was calculated here. The highest value of the y stress component on the recess edge $\sigma_y = 425$ MPa.

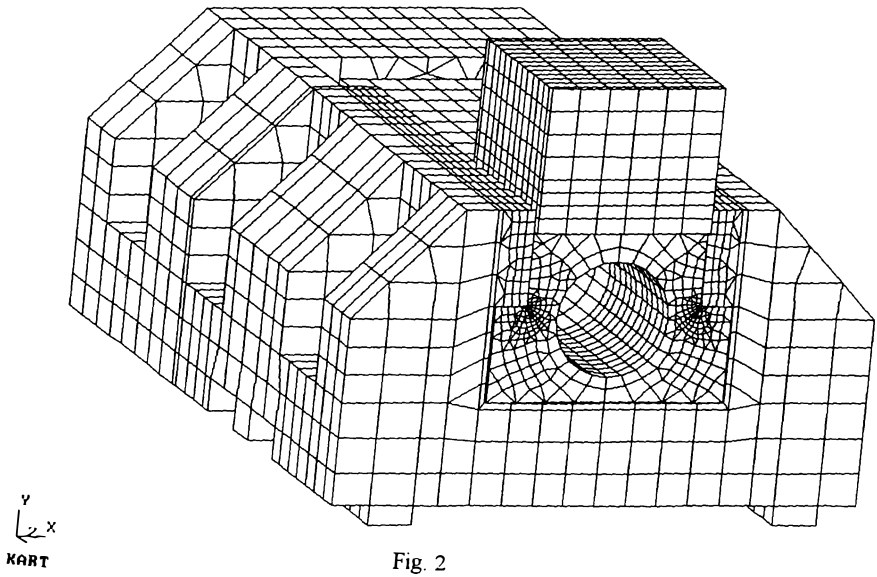

Fig. 2

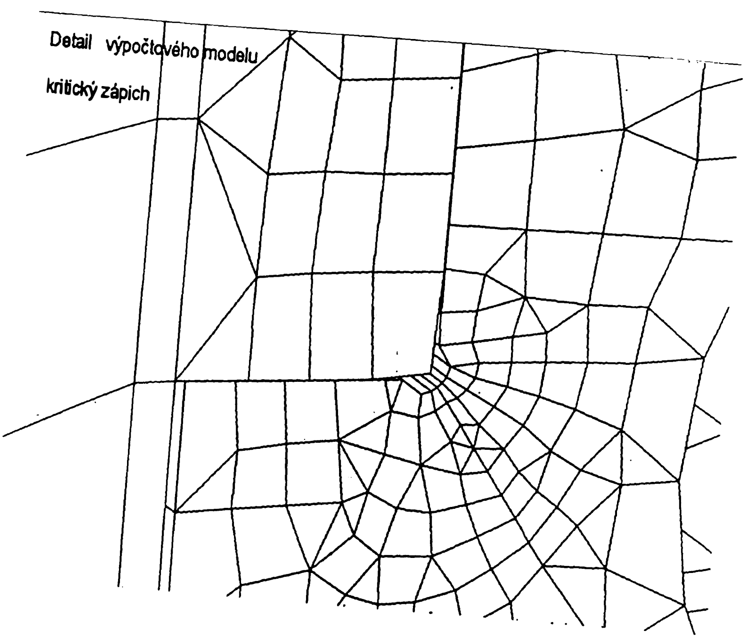

Fig. 3

3.2 Investigation on a Real Clamping Jaw

To verify the standard calculation, the FEM analysis and the measurement on the test bar mentioned above, the strain gauge investigation has been performed on the real clamping jaw of the lathe.

Firstly, the measurement was realized on the robust cast iron plate, because of the parallel realized holographic interferometry measurement. The clamping jaw was positioned in the originally guid body, the chucking force was realized with the clamping screw of the guid body against to the cross bar, mechanically. The force intensity was measured with the cylindrical strain gauge dynamometer with the help of the another cross bar. The chucking force has reached the maximum of 80 kN in this arrangement. The static strain gauge bridge DMD 20 with the 10-channel selector UMK 10 fy HBM was used for this investigation.

The strain gauges LY 11 0.6/120 fy HBM were cemented in the horizontal (ε_x) and vertical (ε_y) directions to the circular recess. The centres of its grid were positioned 1 mm from its edge (Fig.4). At the same time the strain e_y was measured with the use of the LY 11 3/120 strain gauges in the middle of the recess and the hole of 80 mm. The reference strain gauge LY 6/120 was positioned on the opposite side of the clamping jaw head.

Secondly, the measurement was realized directly on the lathe. Here, the guid body was clamped on the lathe faceplate, standardly.

The pneumatic screw runner was used, to chuck the workpiece of about 70 t weight, it was realized with the force of 370 kN with the position of the measured clamp jaw in the middle of the turned faceplate. Turning the faceplate, the force altered between the values of 300 kN and 370 kN in the top an bottom position, respectively. The difference of such position extreme strain values was found about 30%.

The measurement results with the pulled out clamping jaw are presented summary in Fig.5. Here, the beginning of the curves corresponds to the off machine measurement while the end to the on machine one. Presented curves differ from those ones, which were measured with the pulled in clamping jaw. In that case, the strain unevenness of the recesses to one another has grown up to 100%, independent on the chucking force position. This was caused by the different guide clearance of both jaw sides, probably.

The relation between the strain and the jaw position is presented in Fig. 3. Here, the curves are going close to one another when pulling out the jaw. The last values are in coincidence with the measurement on the lathe (Fig.5) with the pulled out jaw, while the measurement with the pulled in one could not be realised on the machine. Some strain value differences of both recesses were also presupposed due to the clearance elimination. They will be smaller than that 100% strain difference mentioned above and will have its maximum on the recess wall probably..

The stress concentration factor α is shown in Fig.5, too. It was calculated from the relation of values of both vertically positioned strain gauges with the pulled out clamping jaw. It is equal 2.0, approximately.

The maximum strain ε_y has reached the value of 820 μm/m with the chucking force of 390 kN, which represented the strain component σ_y = 170 MPa for the uniaxial state of stress on the recess edge. Using linear interpolation, the maximum stress was estimated to σ_y = 320 MPa for the calculated chucking force of 720 kN.

Because of impossibility to measure on the retracted clamping jaw under extreme load the measurement was realized for the load of 80 kN. The mean value of strains determined for recesses on both sides was taken into account. Strain $\varepsilon_y = 1810$ μm/m was determined for load $F = 720$ kN using a linear extrapolation. Corresponding stress component in the y direction was $\sigma_y = 380$ MPa.

It was found out after evaluation of the whole strain field using holographic interferometry, that the maximum values of the strain component ε_y were located on the recess edge and were by 20% greater than the values on the strain gauges place. The presented strain gauges data of the recess edge were by this factor recalculated.

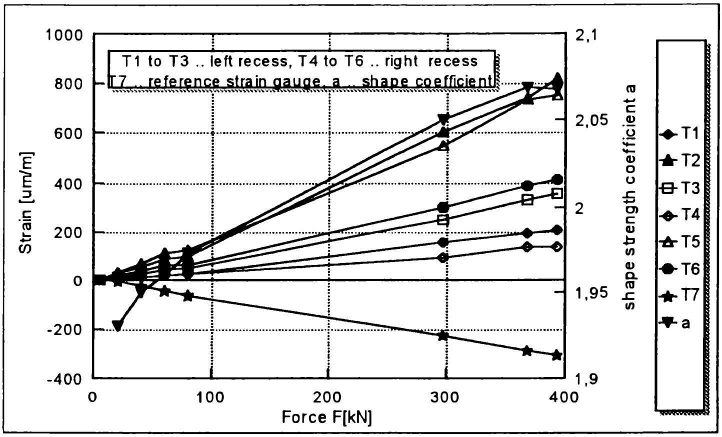

Fig.4 Measured strain versus chucking force

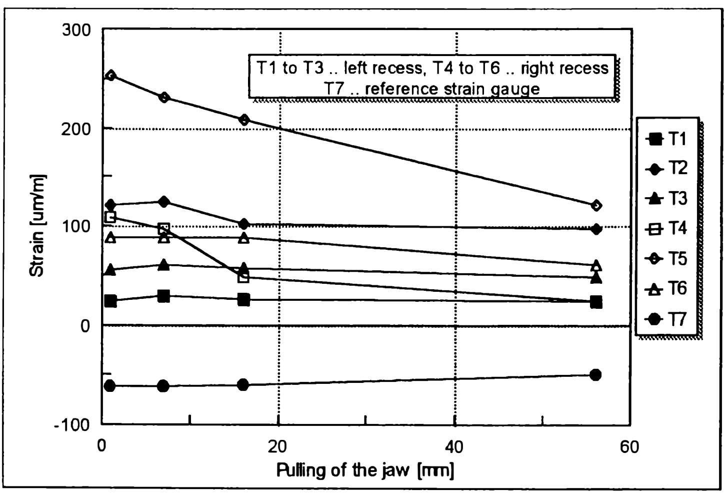

Fig.5 Measured strain versus clamping jaw position

4. Conclusion

The article deals with maximum stress determination in the region of high stress concentration due to shape change of the lathe chuck jaw. Comparison of stresses obtained using electrical resistance strain gage method and numerical FEM method for the stepped flat bar with shoulder filets loaded by bending moment in the plane perpendicular to the plane of the body is done. Using a convenient finite elements magnitude by both methods obtained values are nearly the same (difference is about 9%).

Comparison of stress magnitudes obtained by FEM numerical calculation and experimentally using electrical resistance strain gage method for real lathe chuck jaw under statical load and real conditions of chucking of a workpiece in a lathe was done. Numerical calculation was realized assuming contact problem. Maximum value of the stress component $\sigma_{y'}$ (tangential to the recess) was of 452 MPa.

Experimentally with a linear extrapolation on recess edge obtained magnitude of corresponding stress component was $\sigma_{y'} = 380$ MPa. Because of elastic stress concentration factor $\alpha \geq \beta$ the fatigue notch factor, there is reasonable to determine the elastic stress factor. The elastic stress concentration factor or numerical calculation using FEM including that stress concentration factor can be afterwards used for calculations under fatigue loading. The magnitude of elastic stress concentration α was about 2 for all mentioned above cases.

The stress magnitudes determination with sufficient accuracy is important for unlimited fatigue life calculation. Obtained results have proved that FEM numerical calculation can be used for stress concentration determination in complicated casses, if convenient method and element size are used.

References

[1] Laš, V. - Lašová, V. - Plánička, F.: Some Aspects for Machine Components Observations from Point of View of Unlimited Fatigue Life, 12th Conference with International Participation „Computational Mechanics '96", Pernink 1996, pp. 135-142 (in Czech).

[2] Růžička, M. a kol.: Dynamic Strength and Service life, ČVUT Praha 1992 (in Czech).

[3] Růžička, M.: The Destination of the Strain Response Using the Strain - Gauge Analysis, 33rd Conference of Experimental Stress Analysis, Třešť 1995, pp. 191-195 (in Czech).

Jan Frémund, Škoda Výzkum, s.r.o., Tylova 57, 316 00 Plzeň, tel.: 019/5044760, fax.: 019/533358

Ing. Václava Lašová, Škoda Machine Tool, P.O.Box 80, 303 80 Plzeň, tel.: 019/7735926, fax: 019/7735274

Doc. Ing. Vladislav Laš, CSc., Západočeská univerzita, Katedra mechaniky, Univerzitní 22, 306 14 Plzeň, tel.: 019/2171125, fax: 019/271744, e-mail: las@kme.zcu.cz

Prof. Ing. František Plánička, CSc., Západočeská univerzita, Katedra mechaniky, Univerzitní 22, 306 14 Plzeň, tel.: 019/2171131, fax: 019/271744, e-mail: planicka@kme.zcu.cz

Ing. Jaroslav Václavík, Škoda Výzkum, s.r.o., Tylova 57, 316 00 Plzeň, tel.: 019/7044861, fax.: 019/533358