FATIGUE DESIGN PROCEDURE OF LONGITUDINAL AND TRANSVERSE FUSELAGE JOINTS

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Abstract

A new fatigue design procedure of longitudinal and transverse medium-range civil aircraft skin joints was verified while taking into account the geometrical non-linearity (bending stress) based on results of the experimental study of specimens that simulate different variants of fuselage skin joints.

Following the tests results a number of main joints failure types were determined and each of them is provided with a particular algorithm to calculate the reduced stresses given.

The state of stress (SS) calculation of all joints variants tested is given. The calculation vs. experiment shows that the developed procedures are capable to provide a precision that is enough to be used practically to predict the fuselage joints fatigue stregnth taking into account the bending stress.

1 Introduction

The state-of-the-art approaches to set and to extend the aircraft service life are based to a greater degree on knowledge of primary load-bearing airframe components fatigue and fatigue strength properties as well as on those approaches that facilitate assessing the structural service life characteristics. The substantial extension of current aircraft (transport aircraft included) design service life issues the challenge to provide a sure method to determine the fatigue strength of longitudinal and transverse fuselage skin joints that should be exempt from multisite fatigue cracks during life time.

The principle fuselage joints peculiarity is expressed by the factual high bending stresses level that may thrice and more surpass the nominal stresses. Besides, when the joint has sufficient bending stress the geometrical nonlinearity phenomenon appears i.e. the state of stress depends on the loading level. So, the SS and the fatigue strength were estimated taking into account the bending stresses.

In the study a procedure of longitudinal and transverse fuselage joints fatigue strength design was verified based on results of experimental researches of numerous joints specimens. The fatigue strength of specimens that simulate different variants of medium-range civil aircraft longitudinal and transverse fuselage skin joints is investigated.

2 Test specimens

The transverse fuselage joints were simulated by a triple (three-row) single-shear transverse joint of fishplate connecting onto two plates. The longitudinal joint were simulated by a single-shear transverse joint of two overlapping plates. Specimens of both joints are shown in Fig. 1 and 2.

Besides the study of relationship between the fuselage joints fatigue strength, the plate thickness and the fuselage skin regular area smooth thickness change technology, the influence of type and diameter of fasteners (rivets, bolt-rivets of 4 and 5mm diameter, countersink depth and so on) on joints service life characteristics was investigated.

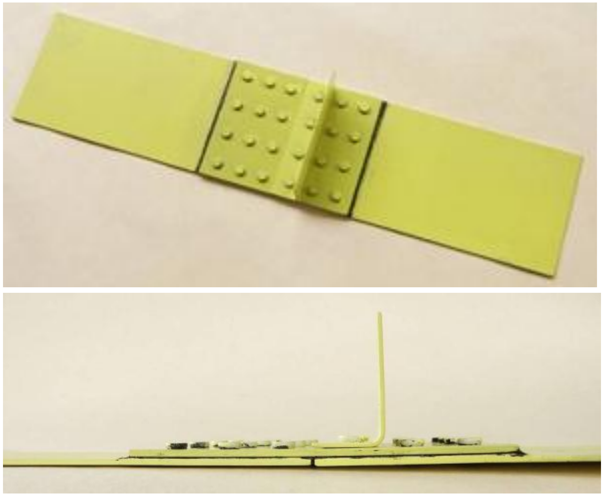

Fig. 1.Transverse joints specimens

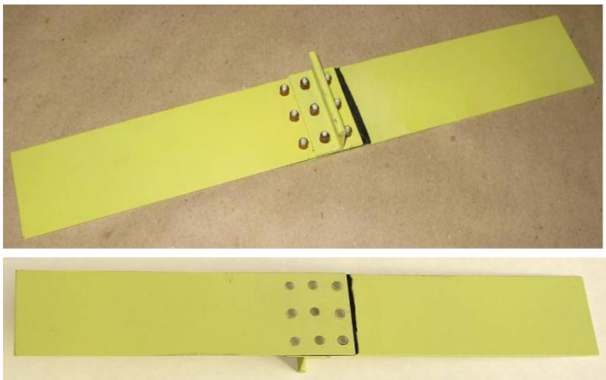

Fig. 2. Transverse joints spécimens

3 Stress of state calculation procedure

To calculate the SS a special algorithm was used. The algorithm treats the joined elements as bars that are working in bending and tension and the fasteners are treated as elastically based beams. The developed procedure enables enough rapidly and precisely the calculation of joints SS with taking into account geometrical nonlinearity of joints, that simplifies the parametric analysis.

The one type transversal joints SS computation results are shown in Fig. 3&4. The plot of the joint total loading versus the relative forces that act upon the most loaded joints rows is shown in Fig. 3 and the plot of the joint total loading versus the bending stresses in the same rows is shown in Fig. 4. Based on SS computation results it is to conclude that:

– the joint SS has a nonlinear dependencies on the loading value, in so doing, as far as the load grows the bending stresses influence on joints rows force distribution decreases as well as the local bending stresses value;
– in dependence on load magnitude the bending stresses can exceed the regular stresses near joints by more than 3 times. At load magnitude that is typical for aircraft fuselage the bending stresses may exceed the regular stresses range up to ÷ 1.8 times;
– while estimating the fatigue strength and its equivalents for joints under bending load it is crucial to allow for dependence of stress-state on load magnitude (Fig. 3 and 4). As a rule, the low load levels compared to real life situations are not included into experimental tests load programs for the sake of tests time saving. Taking into account the SS nonlinearity it may lead to fatigue strength safety factor overestimation.

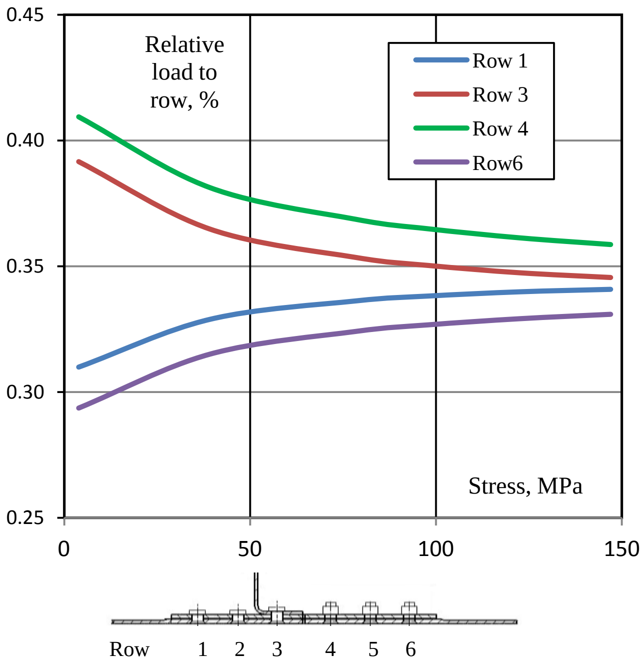

Fig. 3. The transversal joints row relative load vs. the external loading magnitude

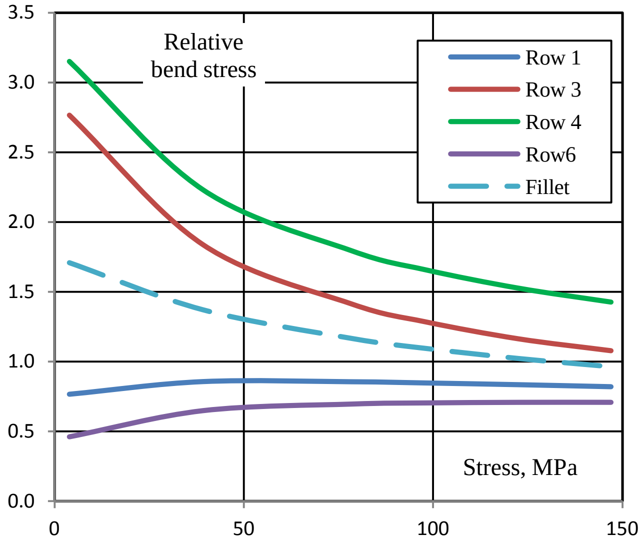

Fig. 4. The transversal joints row relative bending stresses vs. the external loading magnitude

4 Joints fatigue strength computation procedure

The tests results show three main types of failure: the loaded hole failure, the edge of head of rivets failure and the fillet (smooth thickness change joint zone) failure.

To compute the reduced stress value for each type of failure the following relationships are used.

4.1 Loaded hole failure

The structural elements joints fatigue strength computations performed bases on procedure which reduces a combined SS near loaded hole to that of near unloaded hole. The σ^* reduced stresses are predicted by the formula:

$$\sigma^* = k_1 \left(\begin{array}{l} (1-\alpha)\sigma_n + K_{br}\alpha \cdot abs(\sigma_n)\dfrac{B}{d} + \\ K_{bend}\sigma_{bend} \end{array} \right), \quad (1)$$

where σ_n – stresses acting in front of a given concentrator;

k_1 – coefficient that takes into account the difference between concentration factor of a given element and a specimen used to obtain stress-cycle diagram;

α – the ratio of a force transferred by a fastener (bearing load) to the force acting on a given element;

B/d – the ratio of a given element width B to the fastener hole diameter d;

K_{br} – bearing intensity factor that takes into account the difference between the bearing load and the by-pass load on concentrator;

σ_{bend} – bending stress that act in the zone of a given concentrator;

K_{bend} – intended for computing the joint area bending intensity factor that takes into account the difference between damaging bending load and by-pass load on concentrator. In this paper the ratio is equal to 0.5.

4.2 Fillet failure

To compute the fatigue strength of fillets the stress-cycle diagram for specimen with unloaded hole was used. The reduced stresses are computed by the following formula:

$$\sigma^* = \sigma_n \frac{K_{eff}}{K_{eff}^{spec}}, \quad (2)$$

where K_{eff} – effective stress concentrator factor for a given concentrator;

$K_{эф.}^{обр.}$ – effective stress concentrator factor for a specimen with unloaded hole.

Based on the theory of similarity [1] the effective stress concentrator factor K_{eff} is possible to be computed by the following formula:

$$K_{eff} = \frac{2\alpha_\sigma}{1 + \left(\dfrac{88.3\bar{G}}{L} \right)^{\nu}} + \frac{1}{\beta} - 1,$$

where α_σ – theoretical stress concentrator factor a concentrator under consideration;

L – the highest stresses zone length;

$\bar{G}$ – relative stress gradient;

β – surface quality factor. According to test data it lies in the range from 0.2 to 1;

ν – obtained from test data constant, that is equal to 0.2.

4.3 Rivet-heads failure

The rivets and rivet-bolts heads edges failure represents one of the typical failures of joints under consideration. To compute the fatigue strength within these zones the stress-cycle diagrams of specimens with unloaded holes and the following formula for the reduced stress were used:

$$\sigma^* = \frac{\sigma_n + K_{bend}^{head}\sigma_{bend}}{K_{spec}}, \qquad (3)$$

where K_{bend}^{head} –factor that takes into account the stress concentration in the point fastener installation zone. This factor is possible to be represented and used as the following formula:

$$K_{bend}^{head} = 0.75\frac{B}{D}\gamma;$$

D – fastener head diameter;

γ – constant that will be obtained afterwards based on experimental data analysis result.

5 Experimental results data analysis

The relationship between the effective stress concentration factors and the total external load magnitude on joint is shown in Fig. 5 for three failure types that are considered in Section 4. The effective stress concentrator factor is equal to ratio of reduced stresses **σ*** to the nominal stresses in zone of the joint. As Fig. 5 shows the fasteners hole zone is characterized by the minimum fatigue strength in all practical stress range.

The preliminary calculation showed that regardless of the existing nonlinearity of reduced stresses, within the loading magnitudes range (80 ÷150 MPa), that is approved for experimental research the rated joint curve is deviated less than by 0.5% vs. the straight line in double logarithmic axes. That's why the S-N diagram of joint was approximated by the linear relationship within the double logarithmic axes in subsequent calculations.

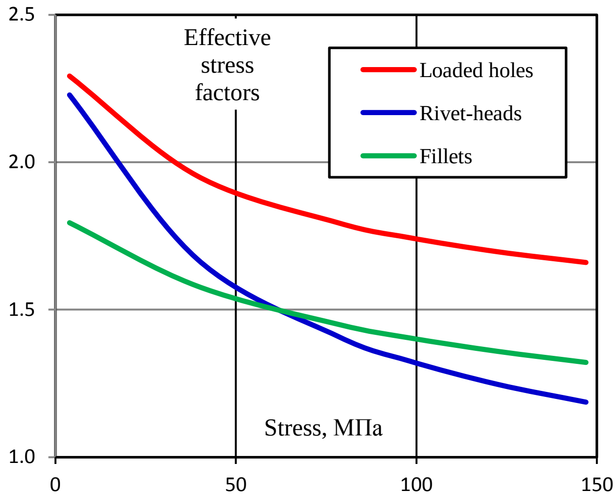

Fig. 5. The effective stress concentration factor vs. the external loading magnitude

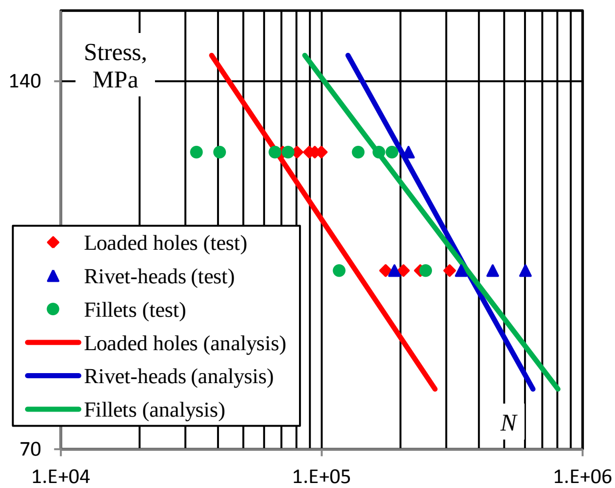

Fig. 6. Results of calculations and tests of transversal joint fatigue strength

Based on analysis results of various joints types the calculated values for loaded holes using (1) (bending is accounted) are at the bottom limit of experiment data. As a pattern (Fig. 6) the computation results for three types of joint failures are given (relationship (1), (2) and (3)) as well as the corresponding test results for a one type of transversal joints. The exception was only for fillet areas of some joints types for which the effect of fillet production technology on fatigue strength was investigated. As it is known the grooves' presence in fillets causes the significant increase of efficient concentration factor the magnitude which of may surpass 5, that is much higher vs. the corresponding concentration factors for other joint areas.

Thus, practically all the experimental points are higher than the computed stress-cycle diagram (for loaded holes), although when taking into account that the computation used a stress-cycle diagram for averaged fatigue strength values, the computed diagram must be approximated to averaged values. In this case (for the type of joint Fig. 6) the large number of damages was initiated by rivet-heads failures or by combined mechanism: simultaneously both by the rivet-heads and the holes. This fact shows the relationship between the fatigue strength and the interference that is not taken into account in the computed model used.

The fit of rivet or rivet-bolt body into the hole under interference leads to fatigue strength growth that is getting higher when the interference is getting higher too. At the same time when the loading magnitude grows the interference influence is getting lower. That's why one of the following test scenarios was performed:

- Relatively high interference specimens have no loaded holes failures. The next critical failure type is realized i.e. the rivet- heads failure or the fillet failure.
- Almost all types of failure are realized for specimens with less interference magnitude as the fatigue strengths in all crucial zones became nearly equal.
- For the specimens with minimal interference magnitude all the failures are in loaded holes.

The tests results and the computed stress-cycle diagram for the latter scenario specimen are given in Fig. 7. The visually high angulation of stress-cycle diagram for the computed curve as compared to the experimental data seems to be associated with the fact that under the loading magnitude growth the influence of interference upon the fatigue strength decreases.

Fig. 8 shows the computed and experimental values of fatigue strength for all the specimens of transversal and longitudinal joints under consideration and that are fractured near the fasteners holes. The disposition of points (Fig. 8) demonstrates a sufficient precision of the computation procedure used. As it is mentioned above practically all the experimental values for other failure types have higher magnitudes than for the loaded holes. Thus, the analysis based on holes may be used to compute the fuselage joints fatigue strength.

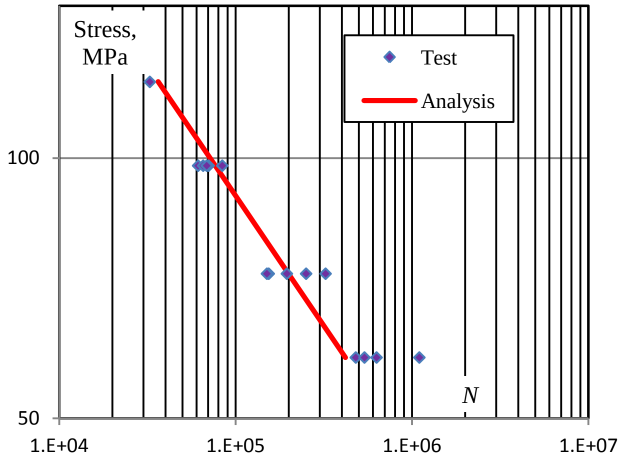

Fig. 7. Results of computing and testing the fatigue strength of longitudinal joint with minimal interference

Fig. 9 shows the computed and experimental values of fatigue strength for all the specimens of longitudinal joints under consideration and that are fractured at rivets heads of end row. The computation vs. the experiment showed that the use of relationship (3) at $\gamma = 1$ results into satisfactory fatigue strength evaluation that lies a little bit lower that the averaged values obtained experimentally.

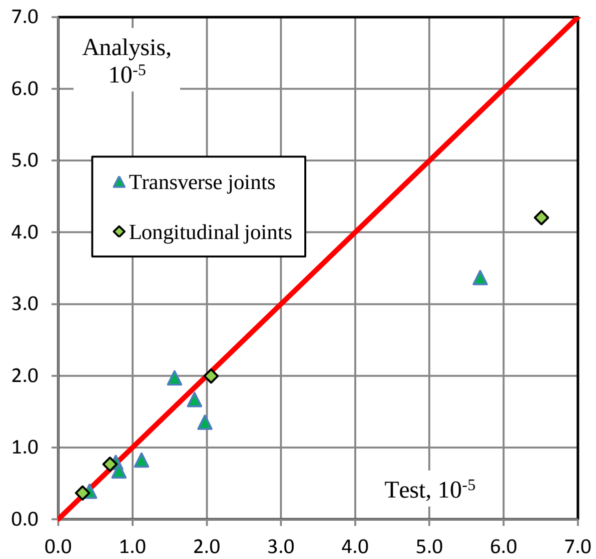

Fig. 8. Applicability analysis of loaded holes computation procedure

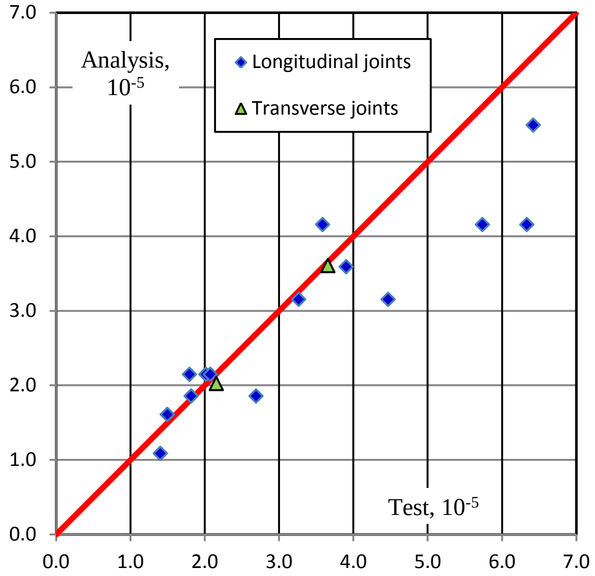

Fig. 9. Applicability analysis of rivets-heads computation

Fig. 10 shows the computed and experimental values of fatigue strength for all the specimens of transversal and longitudinal joints that are fractured near the fillets areas. The computation based on the relationship (2) results into the satisfactory fatigue strength evaluation if the β = 0.4 coefficient is used for the rough surface quality specimens in the zone under consideration. If no apparent defects are observed in fillet area, for example, for all the specimens of transversal joints the coefficient will be as follows: β = 1.

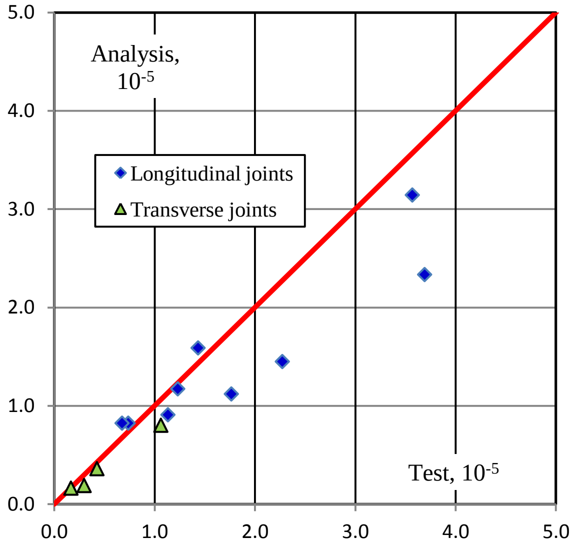

Fig.10. Applicability analysis of fillet area calculation procedure

6 Conclusion

Based on analysis results of several types of specimens of medium-range passenger aircraft transversal and longitudinal joints the calculation procedure precision was verified for three principle types of failure: the fasteners holes failure, the fasteners heads edge failure and the fillet area failure.

The analysis's findings are as follows:

– The computation of fasteners holes fatigue strength (the interference is not accounted) gives a conservative evaluation that lies at bottom limit of experimental data for all types of failure;

– The computation of fasteners heads fatigue strength using the relationship (3) at γ = 1 gives a satisfactory evaluation that lies somewhat lower than the averaged values;

– The computation of fillet area fatigue strength using the relationship (3) gives a satisfactory evaluation of fatigue strength except the cases when the rough defects take place in fillet areas. In these cases the β coefficient must be 0.4.

As a whole it is shown that the suggested calculation procedures allow computing precisely enough for practical application the fatigue strength of transversal and longitudinal fuselage joints.

References

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