Theoretical Evaluations and Numerical Researches of Possibilities of Service Life Extension for Stacked Torsion Members of Helicopter Main and Tail Rotor Hubs

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Currently it has become a widely spread practice in the helicopter industry to use stacked rubber-metal bearings (LRMB), having a number of advantages as compared to conventional stacked torsion members, as hinges in helicopter main and tail rotor hubs. However, the high cost of LRMB leaves certain niche for conventional stacked torsion members as well (further on referred to as "torsion members").

It is known that hub hinges must meet high rigidity requirements as regards resistance to centrifugal forces and low rigidity requirements as regards torsional and flapping motion resistance. Conventional design torsion members do have all these properties but fatigue strength of plates sharply varies in terms of stack thickness. As for exposure to the aggregate of operating forces, the most loaded elements are external plates of the torsion member stack, their fatigue strength usually determining the torsion member total service life. At the same time central plates virtually have unlimited service life.

Previously, for torsion members of helicopters built by Mil Moscow Helicopter Plant, a torsion member design and a designing method were offered that made it possible to create torsion members with predetermined plate strength. Work (1) dealt with some basic features of this design. Extra computations and experiments made it possible to improve the capabilities of the proposed design even more. Thus, fatigue strength of plates can be adjusted by a set of simple engineering techniques. These include the use of different types of plates in the stacked torsion member: non-slotted plates (placed in the torsion member center), flat slotted plates, slotted plates with a structural curve which are placed in the stack external portion along with the use of plates manufactured from materials with different moduli of elasticity.

In addition, for concentrated stress areas (openings for bolts, slot roundings, etc.) special methods to reduce stress concentrations employing simple engineering techniques in those zones have been developed.

According to theoretical estimations, application of the proposed techniques makes it possible to extend the service life of stacked torsion members tens of times as compared to conventional stacked torsion members. This enables the proposed design to successfully compete with LRMBs both in terms of cost and performance.

Conventional designations

B – torsion member plate width

b – band width in the plate

E – modulus of elasticity of torsion plate material

F – total cross-section of all plates

F_1 – total cross-section of flat plates

F_2 – total cross-section of structurally curved plates

H – torsion member total height. $H=h_0+h_1+2\cdot h_2$

h_0 – total height of spacers between plates

h_1 – height of flat plate stack

h_2 – half of total height of curved plate stack

L – length of torsion member flat plate (distance between bolt centers)

L_1 – slot length

ΔL – difference between lengths of flat and curved torsion plates

N – centrifugal force

N_{10} – part of centrifugal force taken up by flat plates before straightening out of curved ones

m – total number of plates

n – Wöhler curve degree index for the material under consideration

X – a coordinate along the torsion member calculated from the axis of one of the bolts

Y – distance from the plate neutral surface to the torsion member symmetry plane

U – plate end face linear displacement due to the turn

Δ – structural curve deflection

δ – plate thickness

φ – cyclic pitch angle – torsion member twisting angle

$\sigma_{vSym.}$ – symmetrical equivalent cycle stress amplitude

σ_v – variable stress amplitude

σ_m – centrifugal force stress in a homogeneous stack

σ_{fl} – centrifugal force stress in flat plates

σ_{arc} – centrifugal force stress in curved plates

$[\sigma]$ – permissible static stresses

T_k – designed life of the product k-th version or k- th plate

Dimensions used in the work: force in kg, length in mm.

Torsion member plate load analysis. Building basic dependencies

Considered below is a torsion member exposed to two operating forces – centrifugal force and rotor blade tilted to a cyclic pitch angle (Figure 1).

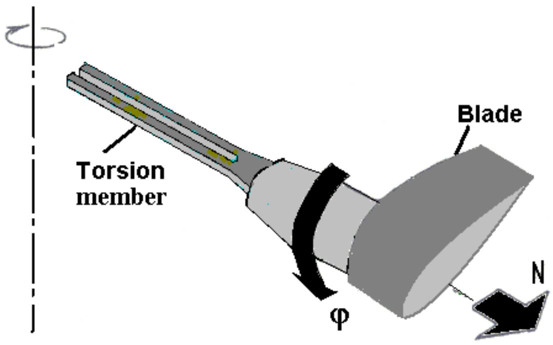

Figure 1

To build simplified design dependencies, let us assume that the centrifugal force is constant over time whereas the cyclic pitch angle is a variable load factor. Let us also assume that all torsion member plates are made from the same material.

The torsion member being one of the most critical helicopter assemblies is imposed with high service life and reliability requirements. Therefore in the course of the torsion members designing one of the natural tasks is to create a structural member with maximum service life determined by the combination of load resulting from the centrifugal force and twisting.

In the cases when the structure is exposed to the combination of constant and variable components, the stresses being equivalent as regards service life are determined using Oding formula. The aggregate operating stresses are recalculated into the symmetrical cycle using formula (1):

$$\sigma_{vSym} = \sqrt{\sigma_v^2 + \sigma_v \cdot \sigma_m}\ . \tag{1}$$

Formula (1) makes it possible to qualitatively compare various versions of a design solution based on a single criterion σ_{vSym}.

However, the strength of any product depends not only on the variable cycle stresses and endurance of the material. Here, also some requirements to statistic strength must be met which can be presented as follows:

$$|\sigma_v| + \sigma_m \leq [\sigma], \tag{2}$$

Thus, a rationally designed torsion member must meet the following requirement

$$\sigma_{vSym} \rightarrow \min, \tag{3}$$

at

$$|\sigma_v| + \sigma_m \leq [\sigma]$$

Let us consider, at a greater length, the stress on the torsion member plates exerted by each of the load components. We shall analyze regular zones only. Specific features of the plates stressed condition in the zone where they are connected to the bolt and stress concentration in other plate zones will be considered below.

Figure 2 shows three plates singled out from the torsion member cross section.

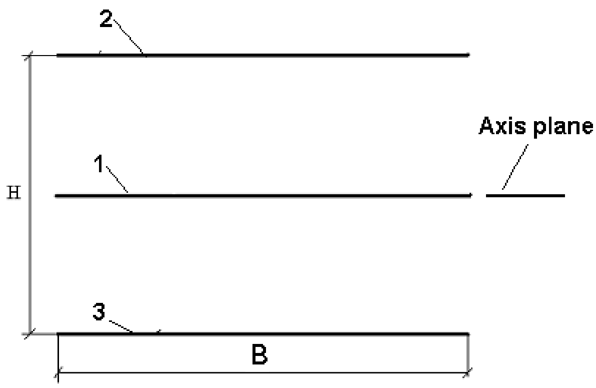

Figure 2

Plate 1 is located in the symmetry plane. Two others (2 and 3) are maximally distanced from this plane. All the plates have the same geometry and thickness. The centrifugal force (disregarding the bolt pliancy) is evenly distributed among all the plates. Stresses from the centrifugal force can be determined as follows:

$$\sigma_m = N/(B \cdot m \cdot \delta), \tag{4}$$

In the course of the torsion member twisting let us assume that one end face of the torsion is fixed while the other is turning. Figure 3 shows the position of turned end faces of three plates with the torsion member twisted through angle φ.

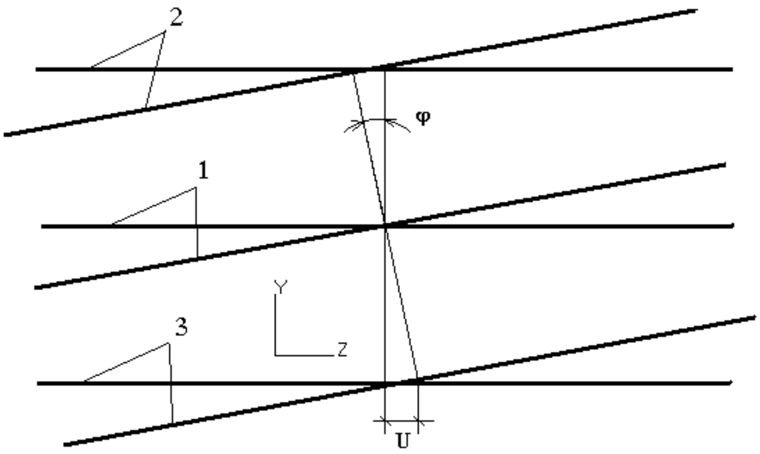

Figure 3

The plate 1 is just beginning to twist. In it, a flow of tangent stresses emerges that counterbalances the torque. Estimations show that these tangent stresses are negligibly small (given the real geometry of the plates) and their impact on service life can be ignored. The stresses of the symmetrical equivalent cycle in this plate are equal, in accordance with (1), to zero.

Peripheral plates 2 and 3, while twisting, act differently. The total displacement can be regarded as a sum of two components. The first component is a linear displacement of one of the plate end faces relative to the other one to value U, while the other component is twisting through angle φ. As mentioned above, the twisting proper does not affect the stressed condition significantly. Therefore let us consider the contribution made to the stressed condition by a displacement to value U along axis OZ.

Stresses from the displacement of the torsion member plate located at distance Y from the symmetry plane to value U can be determined from the following formula:

$$\sigma_V = \frac{3E\ Ub}{L^2}\ , \tag{5}$$

$$U \approx Y \cdot tg(\phi),$$

then

$$\sigma_\gamma = \frac{3 \cdot E \cdot Y \cdot B \cdot tg(\varphi)}{L^2}\ , \tag{6}$$

(5) and (6) were obtained from the calculation scheme of a beam fixed at both ends, one of which is displaced to distance U.

It should be noted that (6) is an approximated dependence disregarding the impact of N force: due to the relative displacement of end faces of one and the same plate, the centrifugal force creates additional torques which can be estimated rather simply. However, in all torsion member designs researched by the authors, stresses from such additional torques proved to be low. Therefore, in the present section, which deals mainly with the quality analysis, those torques and stresses induced by them are omitted.

Then, for any torsion member plate, the stresses of the symmetrical equivalent cycle can be estimated from the following formula:

$$\sigma_{vSym} = \sqrt{\left(\frac{3 \cdot E \cdot Y \cdot B \cdot tg(\phi)}{L^2}\right)^2 + \frac{3 \cdot E \cdot Y \cdot tg(\phi)}{L^2} \cdot \frac{N}{m \cdot \delta}}. \qquad (7)$$

From (7) it follows that the shortest service life is provided by the plates farthest from the symmetry plane, i.e. plates with the highest Y. In order to increase the service life it is necessary, in the first place, to reduce the stress of the symmetrical cycle exactly in those plates.

Let us consider the situation when N=0. The stresses of the symmetrical cycle will then be equal to torsional stresses

$$\sigma_{vSym} = \frac{3 \cdot E \cdot Y \cdot B \cdot tg(\varphi)}{L^2},$$

and will change linearly along coordinate Y. According to the Wöhler formula, service life is determined by power dependence with a certain index of n degree. The relation of service lives of two plates located on coordinates Y_1 and Y_2 can be presented as follows:

$$\frac{\mathrm{T}_1}{\mathrm{T}_2} = \left(\frac{Y_2}{Y_1}\right)^n.$$

From (7) it follows that the plate on symmetry plane (Y=0) virtually has unlimited service life. Plates being farthest from the symmetry plane have minimum service life. Service life of those plates is determined by the torsion member total service life.

The relation of the central plate service life to that of extreme ones is as follows:

$$\frac{T_1}{T_2} = \left(\frac{Y_2}{Y_1}\right)^n -> \infty \qquad (8)$$

Despite the fact that force N somewhat equalizes service lives across stack height, from (8) it follows that in order to increase the torsion member endurance as a whole it is necessary, in the first place, to increase the endurance of extreme plates.

Ways to increase the torsion member service life in regular zones (zones outside of concentrators)

Let us determine, based on (7), the torsion member parameters being modified. The geometrical parameters of L and H are invariable being determined by the hub dimensions. Also invariable is product m·δ since this value linearly relates to H. Thus, reduction of the first summand value in (7) is only possible by changing parameter B, i.e. plate width. The solution is well known: slotted plates (ref. Figure 4). The band width in the plate will be designated as b.

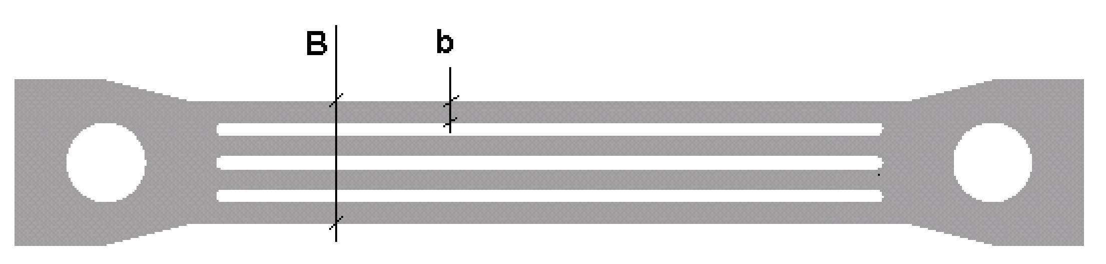

Figure 4

The borderline case of reducing parameter b is a wire torsion member similar to that installed on the Mi-26 main rotor. That torsion member consists of several thousands of wires.

There are no other possibilities to influence the first summand in (7) in a homogeneous plate stack. The second summand determines the contribution to the symmetrical cycle stress made by constant load. Its impact is not as great as that of the first summand. Nevertheless, let us consider what can be obtained by changing this summand.

Above, it was assumed that the plate stack is homogeneous and consists of similar plates. However, without excessively sophisticating the torsion member manufacture, it is possible, for instance, to use plates with different moduli of elasticity. In so doing, for exterior plates it is expedient to use a softer material, for instance, titanium, while for plates inside the stack, a harder material, for instance, steel. In this case, stresses in the peripheral plates will be reduced both by the centrifugal force and by twisting. So, the service life of the entire torsion member can be extended substantially.

Another way to reduce the second component in (7) is creation, for some portion of the plates, of a preliminarily designed curve with certain bending deflection Δ, as shown in Figure 5.

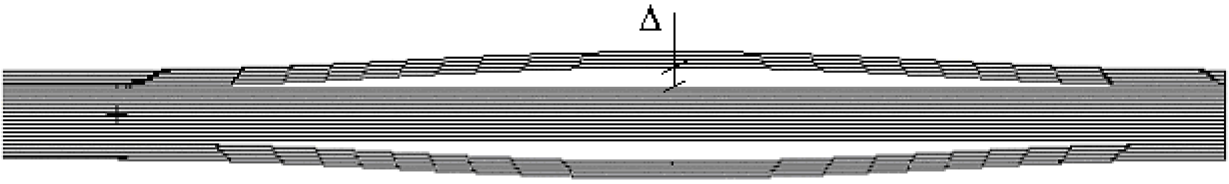

Figure 5

In this case, during the initial stage of the centrifugal force build-up, only flat plates get loaded. At some elongation of the flat plates, the structurally curved plates straighten out and they also begin taking up the centrifugal force. Obviously, stresses in them from the centrifugal forces will be lower than in flat plates. Accordingly, the service life of extreme plates will increase while that of internal (flat) ones will decrease.

Thus, the use of any of these two ways will increase the torsion member endurance. The analysis shows that a heterogeneous stack of torsion member plates may be more preferable than a homogeneous one. From the above it follows that the plates (or plate bands) in the torsion member center may have maximum width. In particularly, these can be unslotted plates. Further, at a distance from the symmetry plane, slotted plates can be located, while at the periphery slotted plates and those with a structural curve (and/or from a softer material) may be used. So the torsion member plate stack becomes heterogeneous.

Methods of designing a new type of torsion members

For designing a torsion member being heterogeneous in its composition, a special method has been developed previously, which includes two steps.

The first step is the determination of the torsion member key parameters. These include plate types, number of plates of each type, plate material and structural curve deflection of external plates.

The second step is the adjustment of plate geometrical parameters to reduce the working stress in concentration areas.

Step 1

To implement the first step we shall supplement the design dependences with due regard of availability of structurally curved plates.

Obviously,

$$N_{10} = \frac{\Delta L E F_1}{L}. \tag{9}$$

For the curved plates to take up some part of the centrifugal force, the following requirements shall be met:

$$\Delta L \le \frac{N\ L}{E F_1}. \tag{10}$$

Stresses in flat plates from the centrifugal force can be determined as follows:

$$\sigma_{fl} = \frac{\Delta L E}{L} + \frac{NL - \Delta L E\ F_1}{L(F_1 + F_2)}. \tag{11}$$

In this case stress in the curved plates will be as follows:

$$\sigma_{arc} = \frac{NL - \Delta L E F_1}{L(F_1 + F_2)}. \tag{12}$$

The dependences shown in this section are built with the assumption of a known difference in the length between flat and curved plates. However, it is inconvenient to set a length difference because in this case it becomes necessary to determine curve bending deflections based on the given length difference. (The curve bending deflection is such that the curve is resilient).

Figure 6 shows the elastic line of a structurally curved plate in unloaded condition.

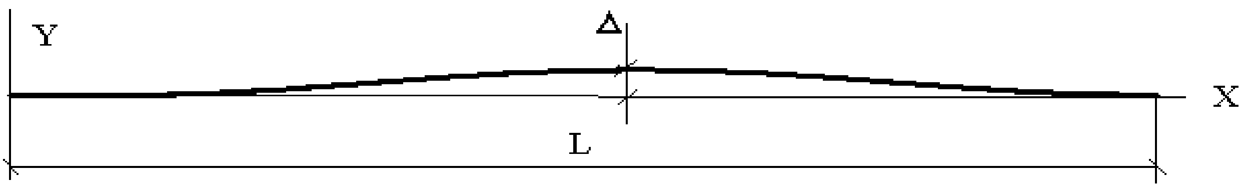

Figure 6

On either ends of the plate the twist angles are equal to zero. The twist angle in the plate center is also equal to zero. Any way of shaping a structural curve creates a similar shape of the plate elastic line. For computations and comparison of versions, only the first element from this row is used. That is the shape of plates with a structural curve in unloaded condition is as follows:

$$Y = \Delta/2 \cdot (1 - \cos(2 \cdot \pi \cdot X/L)), \tag{13}$$

Computation program for Step 1 implementation

Based on the above mentioned dependences and adjustments made, which are not given herein, a computation program has been developed, which makes it possible to carry out design and check computations of a torsion member.

The initial data for the program are the torsion member dimensions, loads, material specifications (moduli of elasticity, maximum permissible static stresses, fatigue limit, basis which was used to obtain this fatigue limit) and level of stress concentration in the plates (the stress concentration is determined based on the preliminary FEM computation for one plate subjected to elongation). Standard flight parameters are also known.

For choosing the torsion member most rational parameters a target function is generated, i.e. designed service life of the entire torsion member determined as a minimum value of the designed service life of each plate. The plate service life is computed based on available material specification data and computed constant and variable components for the stressed condition.

The computation designing is carried out in such a way as to ensure the possibility of using the torsion member based on its condition. For this purpose the target function is supplemented, based on the penalty function method, with extra elements which ensure the minimum designed service life for structurally curved plates. Then, due to inherent strain energy, the bent plates will be straightening out, thus visually indicating the beginning of their breakdown. Figure 7 shows the nature of the torsion member deformation in unloaded condition in case of breakdown of one band in the structurally curved plate (obtained through FEM computation).

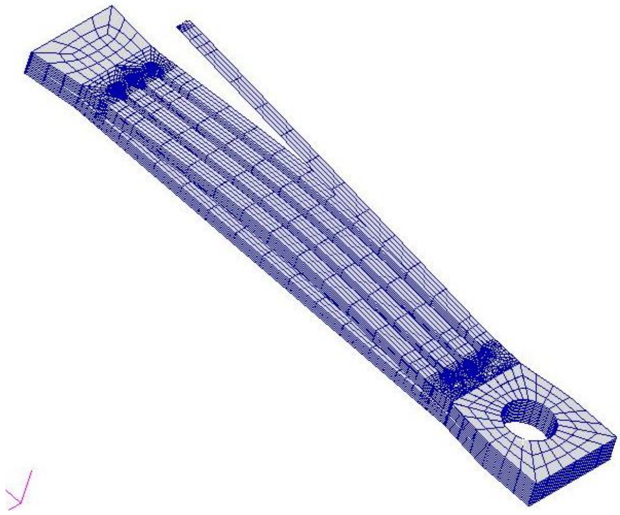

Figure 7

In addition, the target function is also adjusted by static stress limits using, among other things, penalty functions.

The computation block provides for a possibility to design a torsion member having both similar and variable bending deflection of the structural curve for all curved plates. If a variable curve bending deflection is used, then it changes discretely, from plate to plate, from zero to a certain value determined by computation. As a result of the design computation, the number of plates of each type is determined, as well as structurally curved bending deflection and the product in-flight endurance corresponding to the most rational design.

When carrying out the check computation, the user sets all of the torsion member parameters and obtains its service life value in "flights" as an output.

Step 2

During this step plate geometrical parameters are adjusted in order to reduce the working stress level. It mainly concerns the zones with stress concentration: in the area of bolts and slot roundings. The computation is made with regard to possible contacts both between plates and between plates and connecting bolts in the equivalent operating (specification of operational fatigue) and critical (check of torsion member static strength sufficiency) modes.

At this stage, modern FEM computing systems of high class are used making it possible to study the torsion member design with all its structural features and with due regard to all non-linear factors. Figure 8 shows a fragment of the typical finite element model of a torsion member.

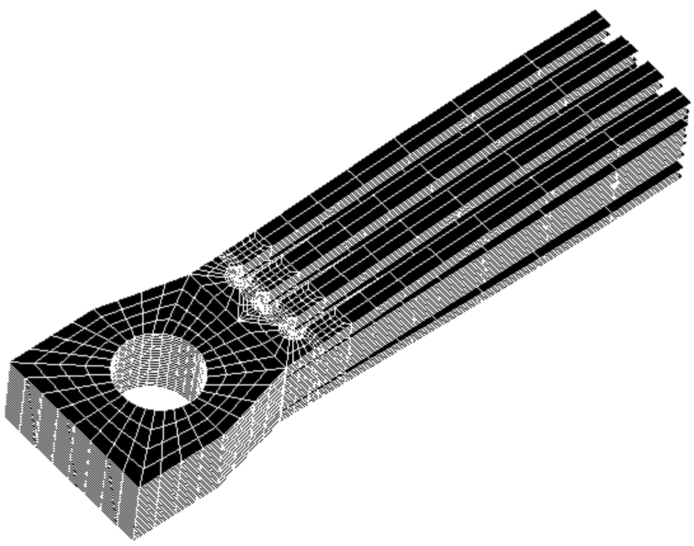

Figure 8

Stress concentration reduction

The above results are valid for regular zones free from stress concentrators. However, the edge of plate slots and bolt openings are serious concentrators that can limit the torsion member service life. Figure 9 shows one of possible versions of the torsion member plate (half to the symmetry plane).

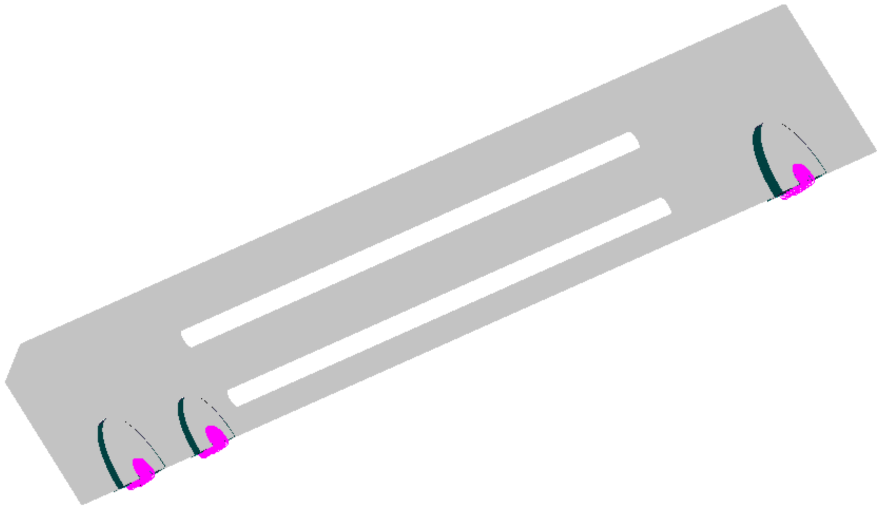

Figure 9

There are two bolts on the left, and one on the right. The slots end in radius roundings.

One of the methods for changing the stressed condition in slots with rounded ends is to change the rounding shape. Adopted as such a shape is an elliptical opening with a larger axis directed along the slot. The FEM computation (all FEM computations were carried out using MSC.Software software packages) studied the impact of ellipsis axes relation on stress concentration during elongation of the plate. Figure 10 gives a representative picture of stress distribution.

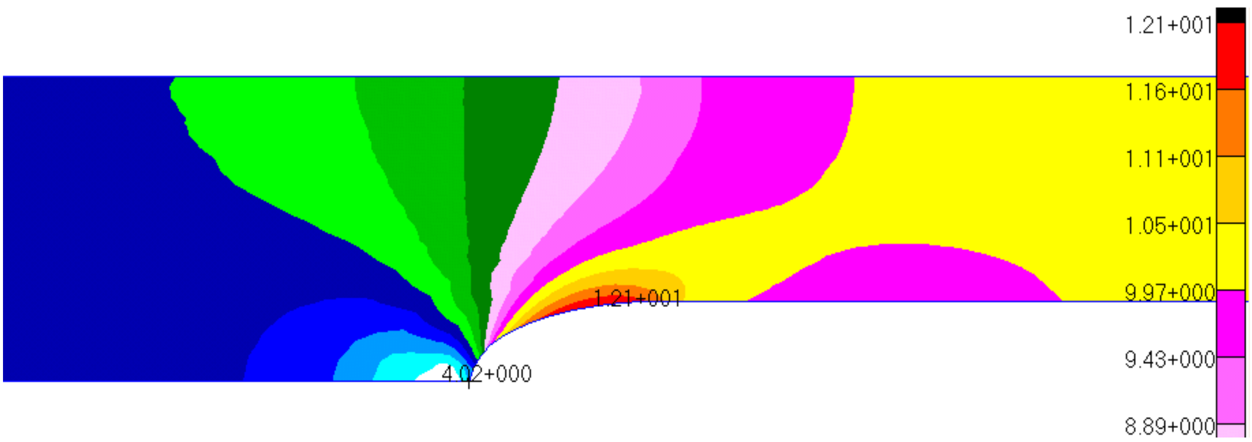

Figure 10

The maximum of stresses is obtained in the area of abrupt change of the contour edge curvature. The change in relation of the length of ellipsis major axis (b) to the length of minor axis (a) makes it possible to substantially reduce stress concentration K in the rounding area (Figure 11).

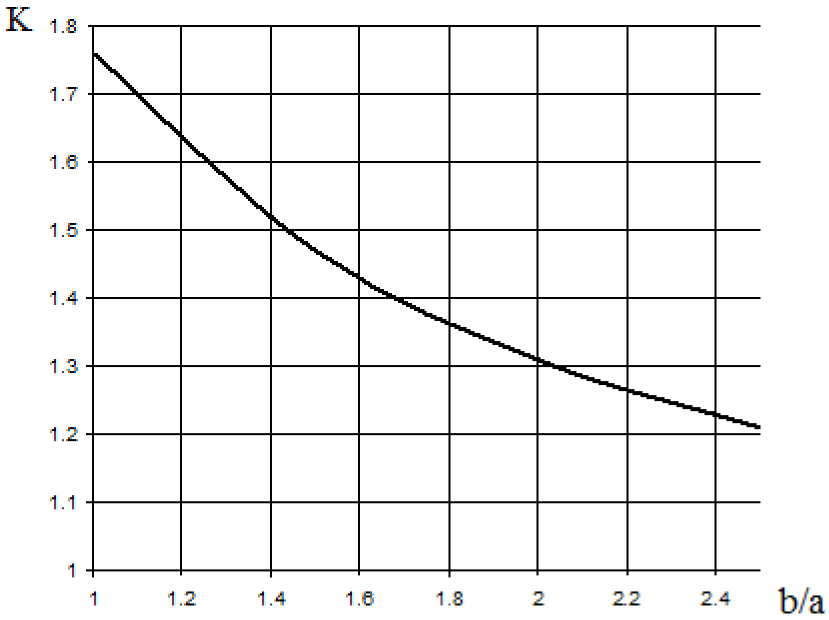

Figure 11

Figure 11 shows a 1.5-time reduction of stress concentration which ensures more than 5-time benefit in terms of service life for steel plates.

This method of stress concentration reduction is also considered for the bolt-to-plate connection. The load is transferred from the bolt to the bush and from the latter to the plate. Figure 12 shows a portion of the plate with an elliptic bush. Specifically, the opening in the plate and the bush external contour are supposed to be elliptic, while the bolt and the bush opening are supposed to be round. The FEM calculation considers a version when the ellipsis axes ratio is equal to 1:1.3.

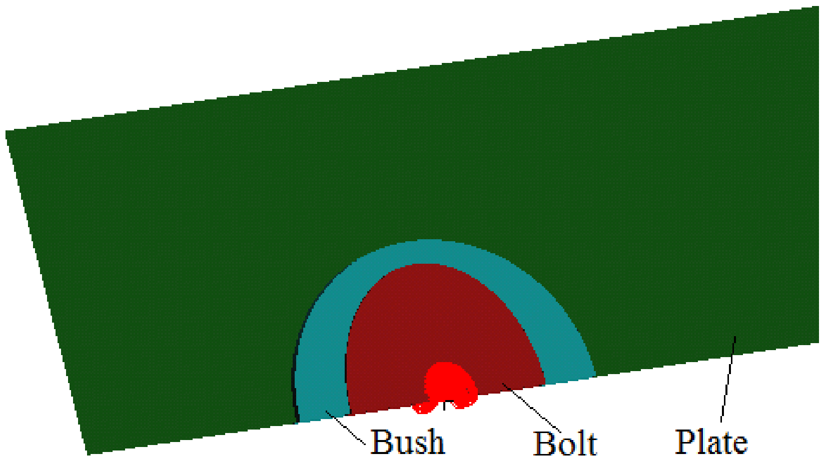

Figure 12

Calculations showed that the stress in the plate dropped by nearly 30 % as compared to the round opening. Accordingly, the service life in the concentrator area demonstrated a nearly 4-time increase. Possibly, higher results can be obtained at a different ratio of ellipsis axes lengths.

Considering modern technological capabilities (NC machines, laser-aided material cutting, etc.) it should be admitted that the approach under study is quite realistic and can dramatically increase the service life of stacked torsion members in the areas of slot roundings and bush openings.

Stress redistribution in bolt opening areas

In case of in-line arrangement of bolts (ref. Figure 9, left side) it is possible to control stress distribution between the bolts (and stresses in bush openings of plates) by rational distribution of gaps between bolts and bushes. Figure 13 shows dependences of maximum stresses in plate openings on the difference of gaps between bolts.

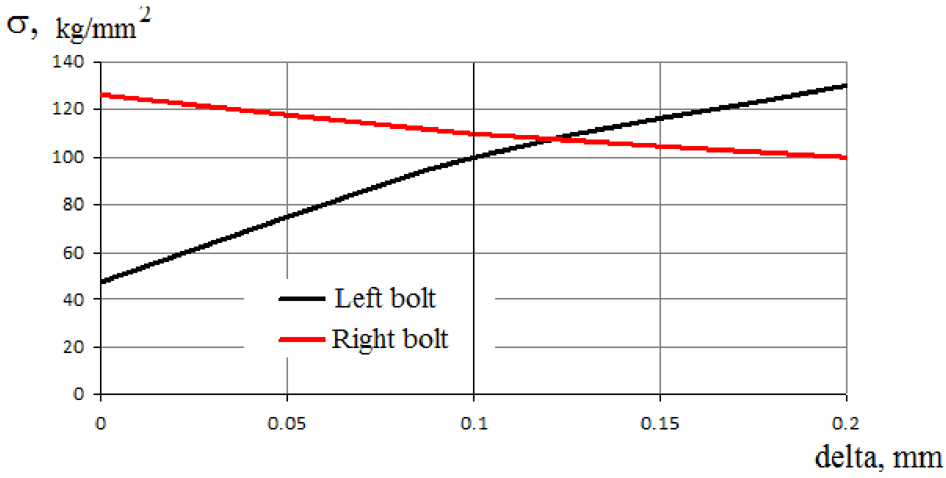

Figure 13

What is meant by "delta" gap difference is the following. On the abscissa axis a difference between bolt and bush gaps on the left and right bolts is plotted. That is the maximum value on the abscissa axis being equal to 0.2 mm means that the gap between the bolt and bush for the left bolt is larger by 0.2 mm than that for the right one. As Figure 13 shows, by changing the gap difference we can substantially redistribute stresses in the plate openings, thus controlling the service life.

Example of theoretical extension of service life for torsion member being in service

Considered as an example is a torsion member used in the main rotor hub of a helicopter. The torsion member basic parameters are shown in Table 1.

Table 1

B	b	H	h_0	δ	L	L_1^*	m
40	7	34	9.52	0.34	305	240	29

* average value of slot lengths. The torsion member has 3 slots, two 245-mm long and one 232-mm long.

All plates in the torsion member are similar. The assumed material is steel with the fatigue limit of 35 kg/mm^2 based on $1 \cdot 10^7$ cycles. The centrifugal force creates nominal stresses in the plates equal to 30 kg/mm^2. The rotor cyclic pitch angle is ±8 degrees. Designed service life T of this torsion member is assumed to be 1.

A design estimation of this torsion member has been carried out, assuming that it contains only two types of plates: flat ones with slots as in the original torsion member and curved ones with slots. Table 2 shows the results of design computations with two types of curve bending deflections. The first type is constant bending deflection for all curved plates. The second type is variable bending deflection that changes from zero to the maximal value at regular intervals. Table 2 shows the computation results.

Table 2

Δ, mm	Number of curved plates	T, flights
Const = 0	0	**1**
Const = 8	14	**6**
Variable, $\Delta_{max} = 8$	14	**12**

Thus, the use of structurally curved plates alone provides a 6-time extension of the designed service life.

Using the same torsion member with flat plates as a basis, we can create on it two bending deflections (Figure 14). All plates in the stack have slots $\Delta_{max} = 8$, $\Delta_1 = 4$.

There are 6 plates with bending deflection Δ_1 and 10 plates with bending deflection Δ_{max} –10.

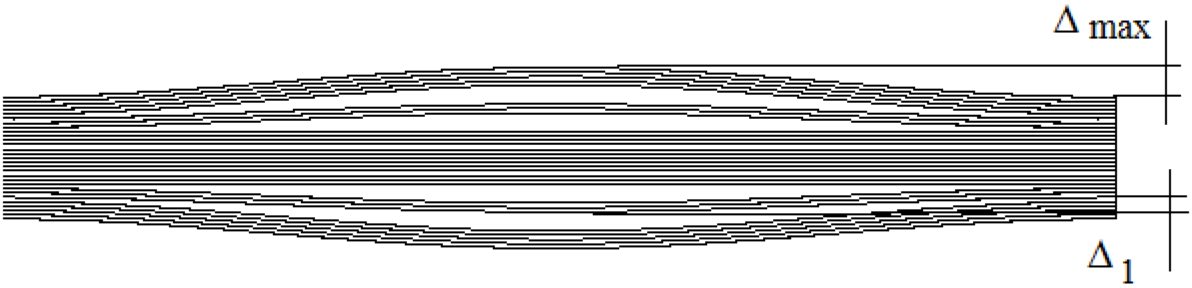

Figure 14

In this case T = 12.

Thus, more than a 10-time extension of the designed service life has been obtained.

Conclusions

The performed researches made it possible to develop a method for designing a stacked torsion member with structurally curved bending deflection with the aim of obtaining maximum designed service life. The algorithm and the computation program based on analytical dependences provide for designing stacked torsion members with long designed service lives. For concentrated stress areas some designing techniques have been developed, which extend the service life of those areas several times.

References

1. Yu.S. Aleksandrin, B.M. Pribytkov, V.P. Timokhin. "Analysis of static stressed-deformed state of stacked hinges and own shapes and frequencies of Mi-38 helicopter tail rotor fitted with stacked hinges." RosVO (Russian Helicopter Association) publications, 2010.