STRESS STAT IN THE RIVET JOINTS: THE APPLIED THEORY OF THE FATIGUE FRACTURE

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Abstract

In the present activity of fundamentals of applied theory of a fatigue fracture of rivet joints and its applications are stated in a generalized aspect. The basic meaning for the residual stresses definition has a solution of the elastic-plastic task about a plate with a cylindrical hole the surface of which is subject to the action of constant pressure. The model of active loading is carried out for homogeneous stresses on the infinity and also concentrated force in center of an insertion. Special experiment confirms the existence of a local curve fatigue for some class of rivet-joints.

1 General Introduction

In this paper there are some results of the European project AISHA. The project aim is the aircraft health monitoring, using ultrasonic Lamb's wave technology. The final link in the health assessment process is: using information about the extent of size to predict remaining strength and lifetime.
The safe use of complex engineering structures such as aircrafts can only be guaranteed when efficient means of damage assessment are in place. Whereas aircraft design is nowadays based on a damage tolerance approach and time based inspection cycles, it is envisaged that the large cost associated with this approach can be drastically reduced by switching to a condition based maintenance schedule. This does require continuous health monitoring capabilities using integrated sensing technology and autonomous damage assessment. This technology must be combined with the development of autonomous signal analysis routines and adequate models for remaining lifetime prediction. There are many articles about theory of predicting of widespread fatigue damage in joints [1-7] together with environmental influence [8,9].

Damage in most cases consists of multi-site cracking, e.g. from rivet holes. Fracture mechanics based analyses at the design stage commonly treat a single crack, such as in a wing skin extending over two bays. When an aircraft is aging, shorter cracks initiate and grow from stress concentration sites under fatigue. These cracks are distributed over a region and can cause failure by coalescence and linkage. For safe performance of the aging aircraft, an assessment must be made as to how much remaining strength and life can be reliably depended upon. Such assessment will allow determination of the inspection procedure as well as whether a part should be repaired or replaced.
The rivet joints widely applied in a modern airframe as a mean of connection of thin-wall units. On the one hand, it is connected with the lack of reliable alternate means of junction of structural units of aluminum alloys. On the other hand, during the last decades of the performance of strength and, especially, fatigue life of rivet joints have sharply increased owing to the application of new process engineering of assembly. There are the numerous means of manufacturing rivet joints of high resource. But in the fundamentals of all these modes there is a usage of favorable effects of influencing residual stresses obtained in the round zone of

fastening points. They appear as a result of those or others technological handling receptions of holes for the rivets. The most effective means of an increasing of a rivet joint fatigue life is the creation of large radial tightness between a rivet and joined parts during the assembly of joint. As a rule, large radial tightness is accompanied by an appearance of plastic strain around of a hole and a favorable field of residual stresses. The influence of radial tightness is induced basically by two mechanical effects. Firstly, the stress concentration factor in critical points of the surface of an rivet hole decreases, so, the amplitude of alternating stresses at this point from an operation of fissile external loading on junction also decreases. Secondly, the residual stresses at the same points are negative, so that, putting up with stresses of an exterior active load, they also call for decreasing the mean stress during the activity of junction. As result,

2 Fundamentals of the applied theory of a fatigue fracture of rivet joints and stress multilevel analyses

According to this theory all the aggregate of factors influential in fatigue life of a structure, is divided into two groups. In the first group there are fixed factors, which one should be identical to all junctions of the given class: mark of materials of joined units, a technology of manufacturing process of joint elements, their mechanical and heat treatment, process engineering of assembly, geometrical sizes of rivets and holes for them, type of rivets and the some other performances. In remaining joint of the given class can essentially differ. For example, in a Fig. 1 the examples of segments of compound structures are shown, the units which one are connected by rivet joints (joint of two sheets, joint of a sheet to an "abandoned" stringer, reinforcement of a sheet with an hole by a fish plate).

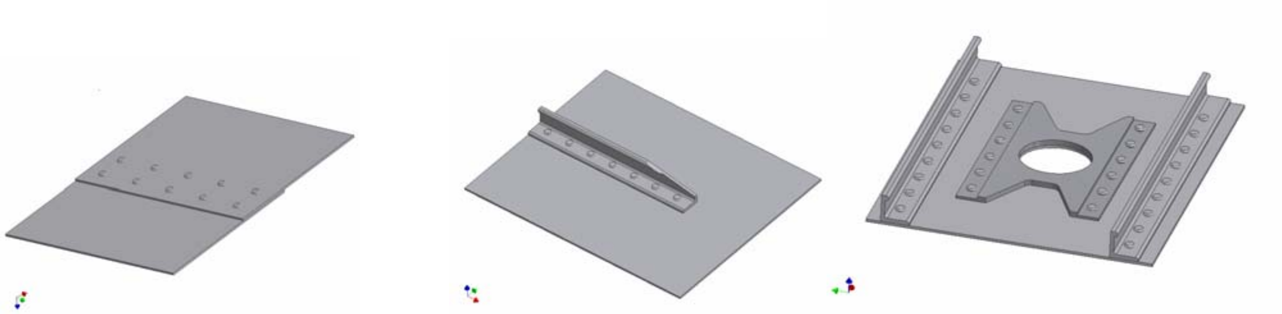

Fig. 1. The examples of the rivet joints application in the aircraft structures.

it calls an increasing of fatigue life of joint. At the same time it is obvious, that for security of reliability of a structure and preservation of its fatigue life the preservation of radial interference during all time of maintenance of a design is necessary. In the process of operational loading there can be peak loads (from extreme guest of air in a flight, at rough landings etc.), which one are capable to call local plastic strains for the most weighted fastening points. It can become a reason for decreasing of residual stresses and fatigue life of a junction.

Structurally they essentially differ each from other. However, if for any of joints to select a small zone of a sheet near of arbitrary selected rivet, the differences mainly mean different intensity of stress states. These differences compose the second group of factors, which one are called varied. For different versions of the given class joints the varied factors can vary in known limits, because their cumulative influence on fatigue life is mirrored in a stress state in a critical point of a structure. The stresses are a determinative of durability up to appearance of an endurance crack.

Thus, the group of varied factors includes the performances of the isolated fastening unit loading (Fig. 2): shear load on a rivet $\boldsymbol{P}$, the stresses in a sheet far from a fastening unit $\sigma_x, \sigma_y, \tau_{xy}$, angle α, a gap or tightness between a rivet and joined unit.

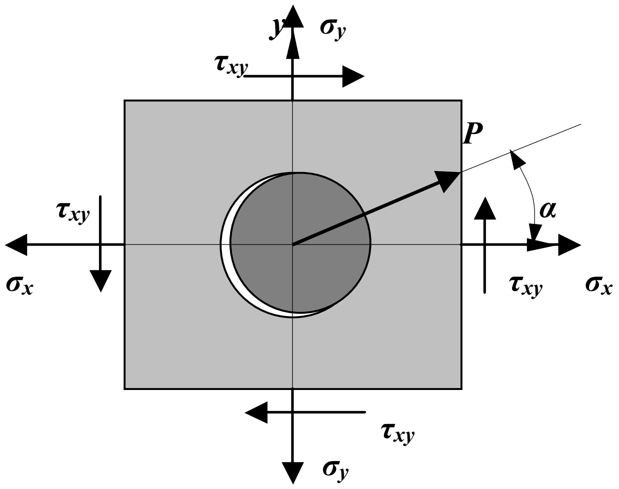

Fig.2. Scheme of an isolated fastening

The aggregate of these factors completely determines stress - strain state in a zone of a fastening unit and, in particular, the circumferential stresses in a critical point of a contour of an hole. Obviously, that the fatigue life up to an appearance of a crack in this point is determined by intensity of these stresses, which one are a function of the varied factors $\sigma_\theta=\sigma_\theta(\boldsymbol{P}, \sigma_x, \sigma_y, \tau_{xy}, \alpha, \varepsilon)$. Depending on a type of joint, standing in brackets the arguments can accept every possible combination. But that of them for which one the circumferential stresses are identical, there corresponds the same fatigue durability.

Thus, the main supposition of the theory is that for the class of joints with identical fixed factors there is a uniform local curve of a fatigue $\boldsymbol{N_0=f(\sigma_{\theta max}, \sigma_{\theta min})}$. Here $\sigma_{\theta max}$, $\sigma_{\theta min}$ are maximum and minimum circumferential stresses in a critical point of a contour of a rivet hole at a cyclic load, $\boldsymbol{N_0}$ is fatigue durability to an appearance of a crack in a zone of this most stressed point. The experimental data [10,11] confirm the existence of a local curve fatigue, besides with an especially higher accuracy, than the requirements of theory are more strictly fulfilled.

The fundamental meaning for adequacy to model of a fatigue fracture has exactitude of definition of a stress state in a zone of a rivet. Its precise definition is rather difficulty and in essence probably only in the case if the task about a stress state is considered as the full task about a contact interaction of units of a concrete joint. In this case a calculation of fatigue durability of a rivet joint is very unwieldy or generally impracticable without usage of the powerful computing tools of a solution of the composite contact tasks. The approximated solution of a task about the distribution of forces in the joint and stresses in its units far from the fastening points is other alternative version of a solution. After that the plane problem for the isolated fastening unit is decided. Its calculation scheme is shown in a Fig. 2. After that it is possible to define the position of the most stressed point on a contour of a hole and a value of the circumferential stresses in this point and the fatigue life up to an appearance of a crack. With this purpose the local curve of a fatigue for this class of joints will be used. It is possible to obtain its using the experimental data and defining a correlation between a fatigue life and a circumferential stress in a critical point of the hole. Thus the same model of the isolated fastening unit should be used.

It is visible that local stress state near isolated fastener must be carried out by multilevel analysis of stresses.

Usually it consists:

1) Approximated definition of the stresses and contact efforts of interaction between the elements of an units. The calculated model for simplification of the analysis represents the monolithic structure formed from the same geometrical plants as the initial units. As a result of FEA the stresses in the first level model are determined. The analysis allows estimating the stresses in all elements and shearing efforts on planes of contact of elements.

$$\bar{\sigma}_i = \frac{\sigma_i}{p} = \varphi_i\left(\frac{D}{\delta}, \frac{L}{\delta}, \frac{l}{t}, \frac{t}{\delta}, \ldots\right),$$

$$\bar{\tau}_i = \frac{\tau_i}{p} = \phi_i\left(\frac{D}{\delta}, \frac{L}{\delta}, \frac{l}{t}, \frac{t}{\delta}, \ldots\right),$$

$$\bar{T}_i = \frac{T_i}{p\delta} = \frac{t}{\delta}\phi_i\left(\frac{D}{\delta}, \frac{L}{\delta}, \frac{l}{t}, \frac{t}{\delta}, \ldots\right)$$

2) Updated definition of stresses of each element of the units. Calculated models of elements represent parts of initial model with the loaded holes under rivets.
3) The analysis of localized stresses of each of elements near the most loaded fastening point. In this analysis the universal model of a contact problem about interaction of a flat body with the inserted cylinder is used. If the rivet-joint has highly resource caused by a high tightness formed at unit mounting the model should take into account presence of a field of residual elastic stresses. By results of the analysis the most probable place of appearance of a fatigue crack and its shape at the initial stage is determined.

$$\bar{\sigma}_\theta = \frac{\sigma_\theta}{\sigma} = f_i\left(\frac{\sigma_{bear}}{\sigma}, \frac{\sigma_2}{\sigma}, \theta, \frac{d}{\delta}, \ldots\right)$$

3 Model of the isolated fastening unit

Thus, the model of the isolated fastening unit is the basis of the practical calculations of fatigue life of rivet joints. It represents a plate of unlimited sizes with a circle hole. The circle elastic insertion is inserted into this hole with a clearance or tightness. At mounting the units of model are deformed, as a rule, elastic-plastically. In the total in joint after mounting there can be residual stresses. The basic meaning for their definition has a solution of the elastic-plastic task about a plate with a cylindrical hole the surface one is subject to action of constant pressure. Explicitly solution of this task for a material with degree of hardening is given in [14…18].
The analysis of distribution of circumferential stress in a zone of a hole is carried out. In a Fig.3 the outcomes of numerical calculations are shown at three values of a degree of hardening and at the pressure $p=\sigma_0$. It is visible, that the

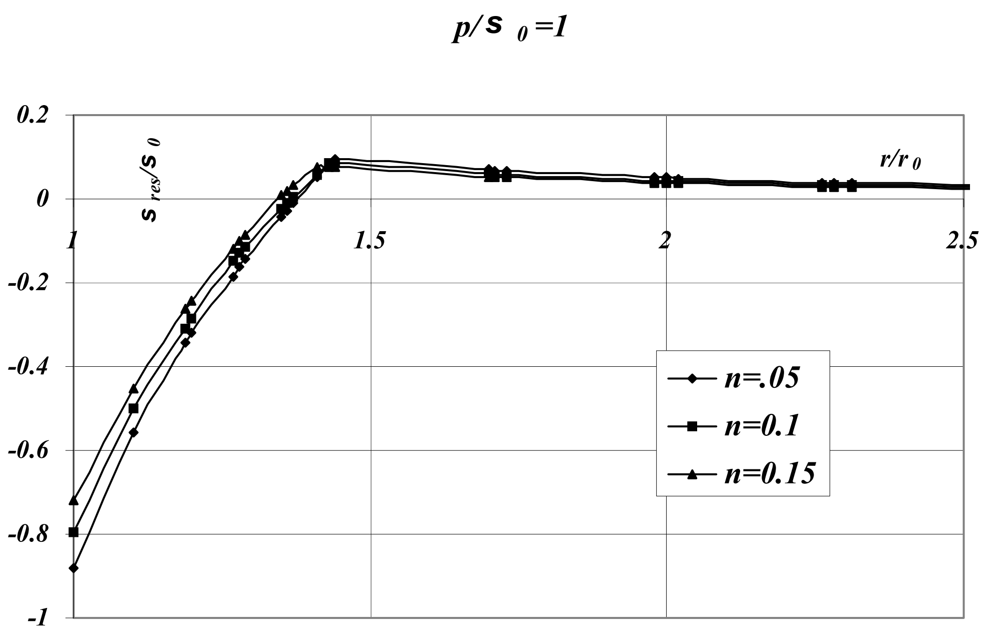

Figure 3. The outcomes of numerical calculations of residual stresses $\sigma_{\theta res}$ are shown at three values of a degree of hardening and at the pressure $p=\sigma_0$.

stress in points of a surface of hole is negative and bigger for material with bigger degree of hardening. In a radial direction this stress increases and has maximum rating on boundary of ring plastic zone. In an elastic range of stress monotonically decreases at an increasing of distance from a hole.

As a rule the processes of assembly of joints for deriving high resource suppose the technological receptions, at which one the cylindrical surface of a hole for an including is exposed to the uniformly distributed pressure. In outcome at sufficient intensity of pressure around of a hole the zone of plastic deformation appears. After full or partial unloading in a ring zone around of

a hole there is a field of residual stresses. Thus, in points of a surface of a hole these stresses are more often compressive.

The active loading of model is carried out for homogeneous stresses on the infinity and also concentrated force in center of an insertion.

The solution of this elastic task about a plate with a circle hole the elastic smooth circle insertion (generally from other material) is inserted into which one can be used as the acceptable calculated scheme for the analysis of a stress state in a zone of a fastening point. The insertion is loaded with a central concentrated force and the plate has a homogeneous state of stress in an indefinitely remote point. The insertion can be inserted into a hole with a gap or tightness. In the latter case at an installation of a rivet in a hole there is a contact pressure p on a contact surface.

In paper [10] it is shown that the solution of this task is reduced to an integer-differential equation for definition of contact stresses. This equation is well known in theory of finite-span wing and in the contact tasks of a theory of elasticity. There are effective approximate methods of a numerical solution of an equation. The feature of the given task consists in higher rate of indeterminacy of the arc of contact sizes in comparison by classic tasks about interior contact of cylindrical domains.

Let in an hole the diameter **d** in an elastic isotropic plate posed elastic, smooth circle insertion a diameter $d + 2\varepsilon$, and ε is small positive constant $(\varepsilon << d)$.

equilibrium remains On a contour of contact of a plate and insertion there is a pressure which one at $\varepsilon > 0$ is determined by the following expression

$$p_0 = -\frac{4 \cdot \mu \cdot \mu_0}{2\mu_0 + \mu(\chi_0 - 1)}\bar{\varepsilon}\,, \tag{1}$$

where $\bar{\varepsilon} = \dfrac{\varepsilon}{R}$ – relative interference, $R = \dfrac{d}{2}$,

$\mu_0 = \dfrac{E_0}{2(1+\upsilon_0)}$, $\mu = \dfrac{E}{2(1+\upsilon)}$, $\chi_0 = \dfrac{3-\upsilon_0}{1+\upsilon_0}$,

E, υ and E_0, υ_0 - Young's modulus and Poisson's constants of materials of a plate and insertion (here and hereinafter parameters concerning to the insertion are marked by a index "0").

If diameter of insertion is less than diameter of an hole $(\varepsilon < 0)$ that $\bar{\varepsilon}$ is the relative gap size and p_0 in the formula (1) corresponds to the stresses of interaction indispensable for a realization of a contact between a plate and an insertion in this case.

After the insertion installation in a hole during the loading the concentrated force P is affixed in centre of insertion (Figure 2).

At the general case of loading on a part L_1 of boundary between a sheet and an insertion there is a contact interaction and along other part of boundary L_2 the clearance is uncovered. The boundary conditions look like

$$\left.\begin{array}{ll} T = T_0 = 0 & v_\rho^{(0)} = v_\rho - \varepsilon \\ N_0 - iT_0 = N - iT & \end{array}\right\}, \quad on \quad L_1$$

$$N = T = 0, \qquad on \quad L_2$$

where N_0 and N are normal stresses , and T_0 and T are shearing stresses on a surface of contact, $v_\rho^{(0)}, v_\rho$ are the radial displacement on contact surface of a plate and an insertion. In result the integer-differential equation for definition of contact pressure is

$$kN(\sigma_0)+\frac{\sigma_0}{\pi i}\int_{\gamma_1}\frac{N'(\sigma)d\sigma}{\sigma-\sigma_0}-\frac{p}{2\pi i}\int_{\gamma_1}\frac{N(\sigma)d\sigma}{\sigma}=$$

$$-2p\Gamma-\frac{p_q}{\pi}\left(\sigma_0+\sigma_0^{-1}\right)-$$

$$-3p\left(\Gamma'\sigma_0^{\,2}+\overline{\Gamma}'\sigma_0^{-2}\right)+\bar{\varepsilon}k_0,$$

$$\text{where } k=\frac{\mu_0(\chi-1)-\mu(\chi_0-1)}{\mu_0(\chi+1)+\mu(\chi_0+1)}, \qquad (2)$$

$$p=\frac{\mu_0(\chi+1)}{\mu_0(\chi+1)+(\chi_0+1)},$$

$$q=\frac{\mu_0\chi+\mu}{\mu_0(\chi+1)+\mu(\chi_0+1)},$$

$$k_0=\frac{4\mu\mu_0}{\mu_0(\chi+1)+\mu(\chi_0+1)},$$

γ_1 is arc of a circle of single radius γ appropriate to an arc of contact L_1 in a plane ξ.

For a resolvability (2) should be fulfilled the condition of an insertion equilibrium which one in the given task means that a principal vector of forces operating on actuation is equal by zero.

$$-i\int_{\gamma_1}N(\sigma)d\sigma=P\quad, \qquad (3)$$

To simplify a solution of the task, the following rational transformation of a variable is introduced

$$\sigma=-e^{i\varphi}\frac{t-i\beta}{t+i\beta}, \qquad (4)$$

where $\beta=\frac{\sin\theta^{o}}{1-\cos\theta^{o}}$, $2\theta^{o}$ is central angle of an arc of contact of a plate and an including , a φ is angle between a positive direction of an axis **x** and direction of vector of a point dividing middle of a contact arc in halves. With the help (4) circles of single radius γ of a plane of a variable ξ are mapped on a real numerical axis, and a contact arc γ_1 , is transferred in a segment [-1,1].

The equation (2) is conversed to a following view

$$k\frac{\beta}{x^2+\beta^2}\bar{\sigma}_\rho(x)-\frac{1}{2\pi}\int_{-1}^{1}\frac{\bar{\sigma}_\rho'(t)dt}{t-x}=f(x), \qquad (5)$$

where it is accepted $\bar{\sigma}_\rho(t)=\bar{\sigma}_\rho(\sigma(t))$,

$$f(x)=\frac{\beta^2 p}{\pi(x^2+\beta^2)}\int_{-1}^{1}\frac{\bar{\sigma}_\rho(t)dt}{t^2+\beta^2}-$$

$$-\frac{\beta}{x^2+\beta^2}\left[\frac{1}{2}p(1+\gamma)-\bar{p}_0(k-p)\right]+$$

$$\frac{4q\beta\bar{\sigma}_{br}}{\pi}\left[\frac{x^2-\beta^2}{(x^2+\beta^2)^2}\cos\varphi+\right.,$$

$$\left.+\frac{2\beta x}{(x^2+\beta^2)^2}\sin\varphi\right]+f_0(x)$$

$$f_0(x)=3p(1-\gamma)\beta\left[(x^4-6x^2\beta^2+\beta^4)\cos(\varphi-\alpha)+\right.$$
$$\left.+4x\beta(x^2-\beta^2)\sin 2(\varphi-\alpha)\right]/(x^2+\beta^2)^3$$

$$\bar{\sigma}_\rho=\frac{N(x)}{\sigma_1},\qquad \bar{p}_0=\frac{P_0}{\sigma_1},$$

$$\bar{\sigma}_{br}=\frac{P}{d\delta\sigma_1},\qquad \gamma=\frac{\sigma_2}{\sigma_1}$$

The equilibrium condition (2) is disintegrated on two

$$\beta\cos\varphi\int_{-1}^{1}\frac{(t^2-\beta^2)\bar{\sigma}_\rho(t)dt}{(t^2+\beta^2)^2}+$$
$$+2\beta^2\sin\varphi\int_{-1}^{1}\frac{t\bar{\sigma}_\rho(t)dt}{(t^2+\beta^2)^2}=\bar{\sigma}_{br} \qquad (6')$$

and

$$tg\varphi=\frac{2\beta\int_{-1}^{1}\frac{t\sigma_\rho(t)dt}{(t^2+\beta^2)^2}}{\int_{-1}^{1}\frac{t^2-\beta^2}{(t^2+\beta^2)^2}\sigma_\rho(t)dt} \qquad (6'')$$

After definition of contact pressure and sizes of an arc of contact the circumferential stresses in points of a contour of an hole under the formula can be defined also

$$\bar{\sigma}_\theta = \frac{\sigma_\theta}{\sigma} = \bar{\sigma}_\rho + \frac{\beta}{\pi}\left[(3-\nu)I_1 \cos\theta - Q\right] + \\ +1+\gamma-2(1-\gamma)\cos 2(\theta-\varphi), \qquad (7)$$

where $I_1 = \int_{-1}^{1} \frac{t^2-\beta^2}{(t^2+\beta^2)^2}\sigma_\rho(t)dt$,

$$Q = \frac{1}{\pi}\int_{-1}^{1}\frac{t\sigma_\rho(t)dt}{(t^2+\beta^2)^2}$$

Not stopping on details of the solution, it is possible to give some results of evaluations and on the basis of their analysis to define some main regularities of a stress state around of an insertion.

Two different versions of deformation are in general possible. The first assumes such combination of parameters of a problem at which the contact on all surface of a hole is saved. In this case contact stresses can be obtained by a degree approximation of a required function in an equation (2). For special loading variants the exact solution can be obtained. In particular, if on infinity takes place a single-axis stretching ***(σ_1=σ, γ=0)*** and its direction coincides about a direction of the force loading including stresses in points of a surface of a contact can be expressed by the following formulas

$$\bar{\sigma}_\rho = \frac{\sigma_\rho}{\sigma} = \frac{1}{2}\frac{\mu_0(1+\kappa)}{2\mu_0+\mu(\kappa_0-1)} + \bar{p}_0 + \frac{2}{\pi}\bar{\sigma}_{br}Cos\theta + \\ +\frac{3\mu_0(1+\kappa)}{\mu_0(3\kappa+1)+\mu(\kappa_0+3)}Cos2\theta \qquad (8)$$

where $\bar{\sigma}_{br} = \frac{\sigma_{br}}{\sigma}$ is relative bearing stresses.

Circumferential normal stresses on the surface of the hole in this case are determined under the following formula

$$\bar{\sigma}_\theta = \frac{\sigma_\theta}{\sigma} = \frac{1}{2}\frac{2\mu(\kappa_0-1)-\mu_0(\kappa-3)}{2\mu_0+\mu(\kappa_0-1)} - \bar{p}_0 + \\ +\frac{2}{\pi}\frac{(\kappa-1)}{(\kappa+1)}\bar{\sigma}_{br}Cos\theta - \frac{2\mu(\kappa_0+3)+\mu_0(3\kappa-1)}{\mu_0(3\kappa+1)+\mu(\kappa_0+3)}Cos2\theta$$

Shearing stresses are determined under the following
formula

$$\bar{\tau} = \frac{\sigma_\theta-\sigma_\rho}{\sigma} = -\frac{1}{2}\frac{\mu(\kappa_0-1)-\mu_0(\kappa-1)}{2\mu_0+\mu(\kappa_0-1)} - \bar{p}_0 + \\ +\frac{2}{\pi}\frac{\kappa}{(\kappa+1)}\bar{\sigma}_{br}Cos\theta - \frac{\mu(\kappa_0+3)+\mu_0(3\kappa+1)}{\mu_0(3\kappa+1)+\mu(\kappa_0+3)}Cos2\theta \qquad (9)$$

If the combination of exterior loads is those that the contact on all surface of a tangency of a plate and insertion is not saved the solution of an equation (5) is executed approximately. One of the most effective methods of the solution is Multopp's method explicitly advanced with

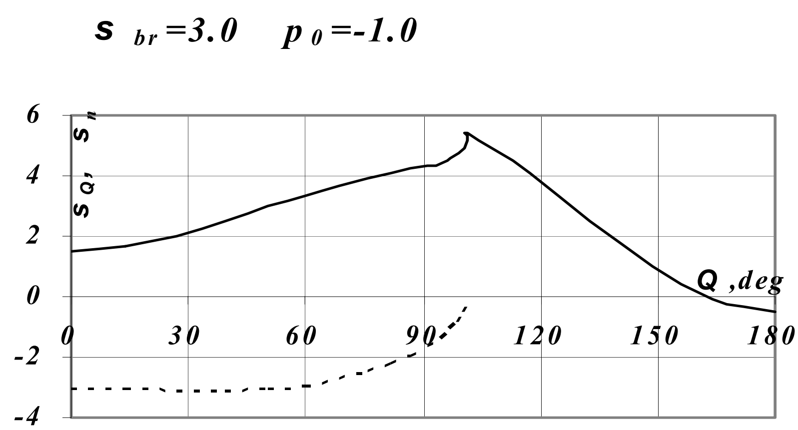

Figure 4. The distributions of circumferential (the solid line) and contact (the broken line) stresses on a surface of a hole at p_0 = *-1.0.*

reference to contact problems of a two-dimensional elasticity in [20]. The integer-differential equation (5) transforms to a system of linear algebraic equations for definition of contact pressure in properly selected points of a contact arc. The distinctive feature of this problem is that the integration limits (a size and a position of a contact arc) is unknown. Therefore the task is decided by an allocation procedure with refinement of the contact arc limits and checking of the insertion equilibrium conditions (6) at each step. On Fig.4 the distributions of circumferential and contact stresses on a surface of a hole are shown at $\boldsymbol{p_0}$ = -1.0. The similar distribution at presence of tightness $\boldsymbol{(p_0 = -2)}$ is shown on Fig.5.

Here values of pressure of tightness and stresses are expressed in relation to the first main stress on infinity $\boldsymbol{\sigma_1}$. Calculation is executed for a case of single-axis loading on infinity. The direction of tension coincides with a direction of a point force in center of insertion. On boundary of a contact arc the circumferential stresses have peak and its value descries with a tightness pressure increase. At the greater pressure of tightness this peak is saved but the maximal circumferential stresses in some interior point may be equal or exceed value of this stresses on the end of a contact arc.
value of this stresses on the end of a contact arc.

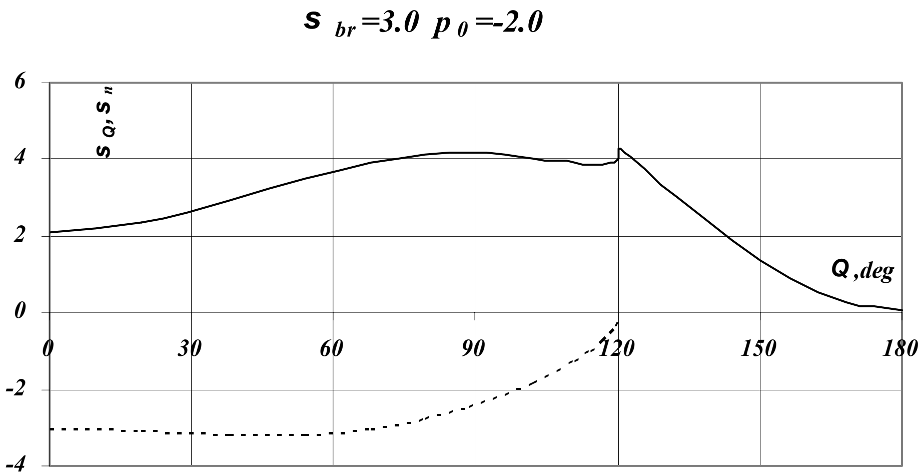

Fig.5. The distribution of circumferential (the solid line) and contact (the broken line) stresses on a surface of a hole at $\boldsymbol{p_0}$ = *-2.0.*

4 Comparison with experiment

The experimental investigation of remarkable theory was carried out on the samples those modelled the different kinds of rivet-joints. All samples are consisted one class of rivet-joints with the same fixed properties:

- materials is Al-Cu alloy D16T (similar to 2024-T4);
- the sheet thickness for all samples was 1.5-2 mm;
- the diameter of rivets 4-5mm;
- the rivet type in all cases was a flat countersunk head;
- the same technology of riveting;
- cyclical load ratio $\boldsymbol{r=0.5}$ and frequency 12.5Hz .

The following kinds of samples were tested (in brackets the appropriated symbol is showed):

- a stringer with free tip(middle picture on Figure 1) at the values of ‘rivet step/rivet diameter’ relation 5, 3,2;

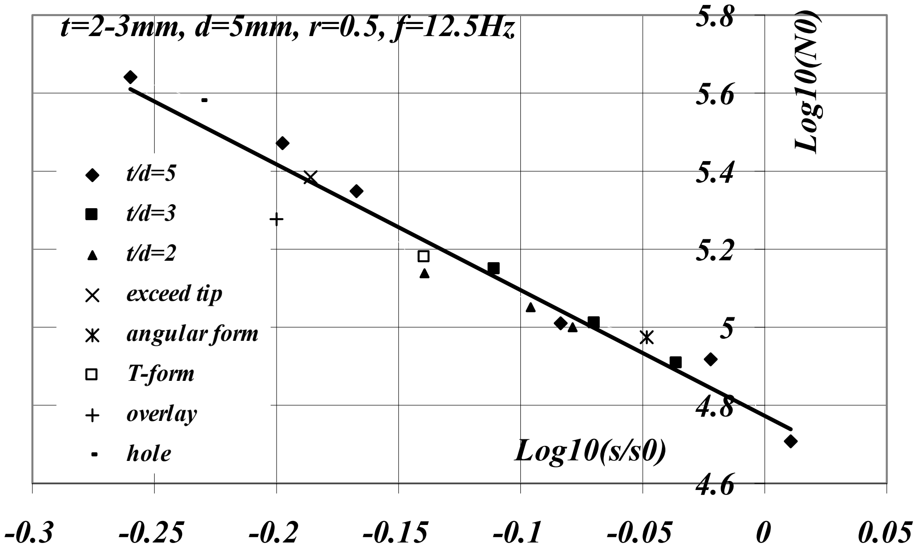

Fig.6. Local curve of fatigue of a rivet-joint class

- the same stringer, but with exceeded form of tip;

a stringer with free tip and angular form and T-form of cross-section;

- a sheet with overly;
- a sheet with round hole and overly.

The result of experiments is showed on Fig.6. Here N_0 is the average number of cycles to the appearance of initial fatigue crack (about 0.5 mm), but $\sigma_\theta = \sigma_{\theta max} - \sigma_{\theta min}$ is done as relation to yield stress. It is shown the trend line of all experimental data in logarithmical co-ordinate system is line. It means the function between σ_θ and N_0 can write in following view

$$\left(\frac{\sigma_\vartheta}{\sigma_0}\right)^m N_0 = C\,,$$

where $C=5.95 \cdot 10^4$, $m=3.226$.

The experimental data confirm the existence of a local curve fatigue, besides with an especially higher accuracy, than the requirements of theory are more strictly fulfilled.

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