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CALCULATION OF THE TUBULAR ELEMENTS MADE OF PRESSED PROFILES

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This paper is dedicated to ensuring the strength of cargo equipment elements of transport aircraft. The strength of rollers of roller conveyors, which are made of standard pressed aluminum tubular profiles or composite tubular elements, is considered. The main disadvantage of these semi-finished products is the deviation of the diameter of these standard aluminum profiles, which leads to the emergence of eccentricity Δ between the axes of the outer and inner surfaces. The influence of eccentricity on the change in the values of normal and tangential stresses is considered. This analysis was carried out for standard diameters of tubular profiles at values Δ equal to half of the standard limit deviation of the outer diameter D. Calculations of normal and tangential stresses and their comparison with nominal stresses that occur in the absence of misalignment have been carried out. Calculations were made of the value c of the removal of the center of rigidity from the center of the circle of the outer border of the cross-section at different cross-section sizes of standard profiles with values of Δ equal to half of the standard limit deviation of the outer diameter D inclusively. The calculations showed an increase in tangential stresses τ in some cases by 64% and even by 213%. The obtained results indicate that the presence of Δ≠0 will have a negative effect on the resource of these elements. In order to eliminate the negative consequences, it is necessary to increase the requirements for the shape deviation of the tubular profiles in the input control.

Keywords: *cargo equipment of transport aircraft, roller conveyor, tubular profile, misalignment, influence of eccentricity, thin-walled rod, center of cross-sectional rigidity, normal and tangential stresses, shape deviation, resource.*

Introduction

The design of cargo aircrafts includes roller conveyor equipment designed to move cargo in the fuselage compartment. The working elements of the roller conveyor are rollers made of pressed aluminum tubular profiles or composite tubular elements. The limit deviations of the diameter for standard aluminum profiles are: ±0.5…±1.2 mm, and for composite ones they have a similar or greater value. Naturally, in the process of manufacturing profiles by pressing or by the pultrusion method (if a composite material profile is used), an eccentricity Δ occurs between the axes of the outer and inner surfaces of the profile (Fig. 1).

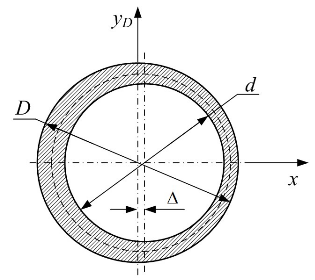

Fig. 1. Cross-section

Moreover, if some defects in the form of warping of aluminum profiles can be eliminated with the help of special technological procedures, it is not possible to correct the misalignment of the inner and outer surfaces of the profile. The consequences of the presence of Δ are considered below.

Changes in calculations in the presence of eccentricity Δ

It was assumed that distributed forces act on the tubular element of the roller conveyor in a vertical plane that passes through the axis, which is the rotation axis of this element. There are internal force factors in the cross-section – bending moment and transverse force acting in the specified plane.

The diagram of the transverse force loading of the tubular element is shown in Fig. 2 conditionally taking into account its rotation. In fact, only one transverse force Q_x (Q_x=Q_y), which is shown in Fig. 2 in order to consider the case of the roller loading when it turns on 90° (with Q_y=0), acts in the cross-section of the roller. Perfectly shaped roller (Δ=0, Q_x=0) is in a state of transverse bending (Fig. 2).

The specified eccentricity can negatively affect the tension state of the roller in a number of cases (with significant values Δ). In the future, the roller will be considered as a thin-walled rod [1]. Moreover, the wall thickness of the section of this rod is variable: for the upper half of the section it is $\delta(\alpha)=\frac{D-d}{2}-\Delta+\frac{2\Delta}{\pi}\alpha$. The variable thickness affects other geometric characteristics of the cross-section and causes certain corrections in the calculations of dangerous stresses.

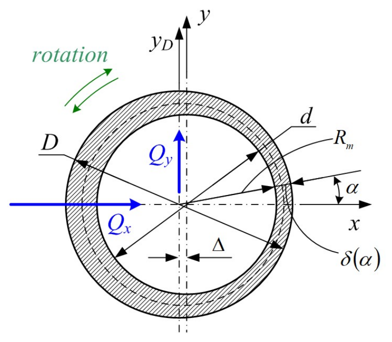

Fig. 2. Load in the cross-section

In the presence of Δ, the most dangerous from the point of view of normal stresses is the load of the roller in the position in which only the force Q_x acts. For the normal stress σ, which is calculated according to the basic formula of resistance of materials [2], the increase is caused by a decrease in the axial moment of inertia I_y and by an increase of the distance from the main central axis, perpendicular to the axis of symmetry, to the extreme fiber of the section in the place where the wall has the smallest thickness. Moreover, both of these parameter changes lead to an increase in stress σ in absolute value. It should be noted that the basic formula of material resistance is used not only when calculating normal stresses in beams and thin-walled rods [1].

Determination of tangential stresses τ at Δ≠0 is more difficult. Eccentricity during loading [3], as always, leads to a change in the character of the action of external forces. It is due to the fact that in the presence of a transverse force acting in the cross-section of a thin-walled rod, transverse bending and torsion occur at the same time, if the line of action of the force does not cross a special point of the cross-section [4, 5].

This special point in the cross-section of a thin-walled rod of an open profile is called the bending center, and in the cross-sections of a closed profile – the center of rigidity [4, 5] (or the center of torsion [1]). In an annular cross-section (Δ=0) the center of rigidity of the cross-section is located in the center of the circles and when the roller is loaded with a transverse force, only transverse bending occurs, and torsion does not occur [4, 5].

At Δ≠0 the geometric characteristics of the section will change and the center of rigidity will shift along the axis x (Fig. 2) in the direction of the greatest wall thickness. In this case, not only transverse bending, but also torsion will act in the cross-section of the tubular element. At the same time, the torque is $T=Q_y\cdot c$, where c – from the axis of the tubular element to the center of rigidity. In the considered thin-walled section, tangential stresses are constant throughout the thickness of the section wall [1] and are determined by the summation of two components: transverse bending and torsion [4, 6].

It should be borne in mind that at Δ≠0 the contour line is a circle of radius R_m with variable cross-section wall thickness $\delta(\alpha)=\frac{D-d}{2}-\Delta+\frac{2\Delta}{\pi}\alpha$. Function δ(α) is a linear dependence. Therefore, the calculation of this cross-section is not reduced to the application of simple compact formulas that were used for the case Δ=0. When determining internal tangential forces along the contour of the section of a thin-walled rod instead of tangential stresses (constant along the wall thickness at each point of the contour line, which has the shape of a circle of radius R_m) it is customary to use flows of tangential forces $q=\tau\cdot\delta$.

Flows q, caused by transverse bending (shear), are determined in this case by the formula

$$q(\alpha)=q_0+q_p(\alpha),$$

where q_0 is the constant part of the flow at the leftmost point of the cross-section (Fig. 2), from which the contour of the cross-section is bypassed (angle α); $q_p(\alpha)$ is the variable part of the flow.

Flow $q_p(\alpha)$ is found by a known dependence [4, 6]:

$$q_p(\alpha)=-\frac{Q_y}{I_x}S_x(\alpha),$$

where $S_x(\alpha)$ is the current static moment of the cross-sectional area.

Moreover $S_x(\alpha)=\int_0^\alpha y(\alpha)\delta(\alpha)R_m d\alpha$, where $y(\alpha)=R_m\sin\alpha$ is the distance from the point (α) on the contour line to the axis x.

From the condition of static equivalence of moments: the moment from the flow of tangential forces $q_0 + q_p(\alpha)$ must be equal to the moment from Q_y in relation to any point; taking this point as the center of the contour circle, we get

$$q_0 = -\frac{1}{\Omega}\oint q_p(\alpha) R_m^2 d\alpha ,$$

where $\Omega = 2\pi R_m^2$ is the doubled contour area.

Flows q, caused by torsion, are determined by Bredt's formula $q_t = \frac{T}{\Omega}$. To define T, c must be found – from the axis of the tubular element to the center of rigidity. The search for the position of the center of rigidity of the section is usually carried out by the fictitious moment method [5–9]. When using this method, T is found from physical considerations, from which $T = \theta \cdot GI$. Here θ is the linear angle of twisting of the cross-section, and I is the torsional constant, G is the modulus of rigidity.

For the cross-section under consideration, the torsional rigidity values and the angular twist angle are calculated according to the following formulas

$$GI = \frac{\Omega^2}{\oint \frac{R_m d\alpha}{G\delta(\alpha)}} \text{ and } \theta = \frac{1}{\Omega}\oint \frac{q(\alpha) R_m d\alpha}{G\delta(\alpha)} .$$

And when calculating θ the integral must be taken numerically.

The product of the linear twisting angle by the torsional rigidity will give the value T. By dividing the torque T on the magnitude of the transverse force Q_y, it is possible to find the distance c to the center of rigidity of the section. In addition, by magnitude T the value of the flow of tangential forces during torsion can be found using Bredt's formula (q_t). The magnitude of the total flow of tangential forces q_0+q_t will allow to determine the tangential stresses at various points along the contour and evaluate their changes compared to the stresses in an ideal section (Δ=0).

The influence of misalignment on stress magnitude

The problem of changes in the values of normal and tangential stresses due to misalignment, which was studied for standard diameters of tubular profiles at the values Δ equal to half of the standard deviation of the outer diameter D.

For normal stresses, the size of their increase depended on the wall thickness $\delta = \frac{D-d}{2}$ and medium diameter $D_m = 2R_m = \frac{D+d}{2}$ ratio.

For rollers D=40 mm at eccentricity Δ=0.5 mm and δ=2 mm calculations showed [2] an increase in normal stresses σ by 14.2%, and at δ=3.5 mm increase in stresses σ became 6,3%.

For rollers D=60 mm at eccentricity Δ=0.6 mm and δ=2 mm calculations showed an increase in normal stresses σ by 18.8%. At δ=4 mm increase in stresses σ became 7.2%, and at δ=5 mm – 5.3%.

For rollers D=82 mm at eccentricity Δ=0.9 mm and δ=2 mm calculations showed an increase in normal stresses σ by 34.5%. At δ=4 mm increase in stresses σ became 12.5%, and at δ=6 mm – 7.1%.

For rollers D=115 mm at eccentricity Δ=1.2 mm and δ=4 mm calculations showed an increase in normal stresses σ by 18.8%. At δ=6.4 mm increase in stresses σ became 10.7%, and at δ=8 mm – 7.2%.

However, for rollers of short length, tangential stresses are decisive in their work [2]. With an ideal shape (a hollow cylinder at Δ=0) these stresses are caused by transverse shear [4]. In the presence of misalignment (Δ≠0) torsional stresses are added to shear stresses. The component of torsional stresses q_0 is caused by the moment from the transverse force, since the line of its action does not pass through the center of rigidity [5].

When determining the increase in tangential stresses, the flows of tangential forces $q(\alpha) = q_0 + q_p(\alpha)$ are obtained in a closed contour for two options of the cross-section (Δ=0 and Δ≠0) [5]. Flow q_0 is the constant part of the flow of tangential forces. Composite q_p(α) depends on the contour coordinate α, the start of the detour along which is chosen at the point of the contour located at the intersection with the x-axis on the right (Fig. 2). Coordinate α increases counterclockwise.

The torque from the flow of tangential forces can be balanced only by frictional forces on the outer surface of the tubular element of the roller conveyor. In this case, a transverse shift and bending in the direction of the *x*-axis will occur. These problems were not considered in this paper, although the appearance of additional frictional forces will contribute to the reduction of the service life of these elements.

The location of the center of rigidity at $\Delta \neq 0$ is unknown.

Calculations of the value *c* have been carried out, *c* is the distance of the center of rigidity from the center of the circle of the outer border of the cross-section at different cross-section sizes of standard profiles at the values Δ, which are no more than half of the standard deviation of the outer diameter *D*.

Fig. 3 shows the calculation results of parameter c/R_m, which positions the position of the center of rigidity in the cross-section.

For rollers *D*=40 mm at eccentricity Δ=0.5 mm and δ=2 mm calculations showed an increase in tangential stresses τ by 27.8%, and at δ=3.5 mm increase in stresses τ became 39.3%.

For rollers *D*=60 mm at eccentricity Δ=0.6 mm and δ=2 mm calculations showed an increase in tangential stresses τ by 35.5%, and at δ=4 mm increase in stresses τ became 51.5%, and at δ=5 mm – 63.7%.

For rollers *D*=82 mm at eccentricity Δ=0.9 mm and δ=2 mm calculations showed an increase in tangential stresses τ by 67.3%, and at δ=4 mm increase in stresses τ became 61.7%, and at δ=6 mm – 95.9%.

For rollers *D*=115 mm at eccentricity Δ=1.2 mm and δ=4 mm calculations showed an increase in tangential stresses τ by 64.0%, and at δ=6 mm increase in stresses τ became 213%, and at δ=8 mm – 240%.

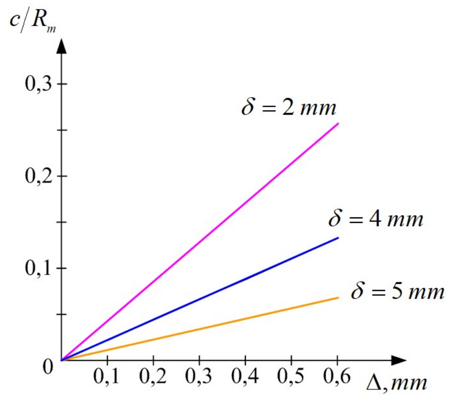

Fig. 3. The position of the center of rigidity in the cross-section D=60 mm

It should be pointed out that in the calculations for the given options, the reduction in the moment of inertia of the cross-section in comparison with the annular section did not exceed 3.2% in most cases (the exception is the option *D*=82 mm at eccentricity Δ=0.9 mm and δ=2 mm – 7.22%).

Conclusions

The obtained results indicate that the presence of $\Delta \neq 0$ will reduce the resource of these elements.

The presence of torsional deformation with torque balancing by frictional forces can lead to compressed torsion due to frictional forces in the direction of the axis of the tubular element, and this will have an even more negative effect on the stress state of this element.

To eliminate these shortcomings, the requirements for shape deviation should be increased in the input control of tubular profiles.

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Розрахунок трубчастих елементів з пресованих профілів

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Робота присвячена висвітленню питання забезпечення міцності елементів вантажного обладнання транспортних літаків. Досліджено міцність роликів рольгангів, які виготовляються із стандартних пресованих алюмінієвих трубчастих профілів або композитних трубчастих елементів. Акцентовано на тому, що головним недоліком цих напівфабрикатів є відхилення діаметру цих стандартних алюмінієвих профілів, яке призводить до виникнення ексцентриситету Δ між осями зовнішньої та внутрішньої поверхонь. Розглянуто вплив ексцентриситету на зміну величин нормальних і дотичних напружень. Аналіз проводився для стандартних діаметрів трубчастих профілів при величинах Δ, рівних половині стандартного граничного відхилення зовнішнього діаметра D. Виконано розрахунки нормальних і дотичних напружень та їх порівняння з номінальними напруженнями, що виникають за відсутності неспіввісності. Проведено розрахунки величини с − видалення центру жорсткості від центру окружності зовнішньої межі перерізу при різних розмірах перерізу стандартних профілів при величинах Δ, рівних не більш половини стандартного граничного відхилення зовнішнього діаметра D. Розрахунки показали зростання дотичних напружень τ у деяких випадках на 64% і навіть на 213%. Отримані результати свідчать, що наявність Δ≠0 негативно позначиться на ресурсі цих елементів. Для усунення негативних наслідків слід у вхідному контролі підвищити вимоги з відхилення форми до трубчастих профілів.

***Ключові слова**: вантажне обладнання транспортних літаків, ролик рольгангу, трубчастий профіль, неспіввісність, вплив ексцентриситету, тонкостінний стрижень, центр жорсткості поперечного перерізу, нормальні та дотичні напруження, відхилення форми, ресурс.*

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