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THEORETICAL AND EXPERIMENTAL STUDIES ON DISKS CENTRIFUGAL SEPARATORS

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Abstract

The paper presents an extension of classical thin shell theory to those with moderate thickness, having simplex order $h/R \leq 0.2 \ldots 0.33$. The application of the flexibility matrix method is studied to realise a computerised analysis of centrifugal disks separators for the influence of the central load F_a induced for assembling the bowl and for the critical area junctions. In the first time, the separator's bowl is divided in structural elements; each of the structural elements is at first considered separately and then the global generalised forces and bending moments are obtained from the displacement compatibility conditions at each element junction. Comparative experimental investigations based on recording surface strains at selected locations of the bowl performed with strain gauges reveal a reasonable agreement. The main features of the method as, (1) reasonable accuracy of the results, (2) reasonable positioning of the critical junctions, (3) low computational cost, give a real possibility in the design work.

Keywords: computerised analysis; centrifugal disks separator; flexibility matrix method; displacement computability; strain gauges.

1. Introduction

Disks centrifugal separator as separation machine of liquid suspension has arisen about 100 years ago; a lot of theoretical (analytical, numerical) and experimental methods concerning the mechanical analysis of them are available in the literature [1–8]. From the exact analytical method based on bending theory of axisymmetrical shells to the numerical techniques like FDM – finite difference method, DRM – dynamic relaxation method, FEM – finite element method or BEM – boundary element method, some common deficiencies as cumbersome theoretical approach, excessive idealisation of the system, the large system of algebraic equations that must be solved and which require skill and intuition, or to large computational time, affect the efficiency of the methods. In this paper an approximate analytical method developed as a numerical method is presented for the elastostatic analysis of one separator's bowl subjected to multiple loads as central axial load ($\mathbf{F}_a$), internal pressure ($\mathbf{p}_i$), and centrifugal force ($\mathbf{R}\cdot\omega^2$). The analysis presents the influence of the central load F_a induced for assembling the bowl for the critical area of centrifugal disks separators; these areas are the same junctions in which the increase of the stresses reveal the beginning of the cracking stage.

The proposed method of linear elastic solution is an extension of the well known flexibility method, strategies of this kind have been developed many years since and are known like the influence coefficients method. Not concerning centrifugal separator's analysis, the idea is not entirely new [9–11, 14–17], but they may be still valid. The techniques of this elastic analysis applied, generally follow the step approach: In the first step the real structure is divided into strips of the same rigidity that includes the main structural elements of the bowl. These are considered separately under conditions of external loads and internal forces acting along the wall-junctions. Corresponding to the released internal forces result conditions to analyse each element of the bowl subjected to inner pressure (p_i), centrifugal force ($r \cdot \omega^2$), central axial force (F_a) and internal forces and moments in the wall-junctions, ($Q, N_\varphi, N, M_\varphi, M$). The procedure of the force method allows to evaluate the coefficients of flexibility at the releases. In the second step, conditions of compatibility of displacement at the releases are formulated, resulting a system of simultaneous equations in compact matrix form, from which the magnitudes of the generalised internal forces and moments in junctions as boundary conditions for a nodal junctions are determined. In the third step continuous internal forces and moments are determined in each wall strip from the boundary conditions. In the fourth step the global stresses solution is obtained by applying the principle of superposition. Results presented in diagrams and tables are compared with experimental study; strain gauges were used to record surface strains at selected location of the bowl. Comparative results reveal that this method can be employed for a safe preliminary design of such structures and give a reasonably good assessment of positioning the critical junctions.

2. Theoretical Approach

The elastic analysis applied in this work involves the usual assumption of thin shell theory and the following main idealisation:

- shell is assumed to be homogeneous, isotropic and to obey Hooke's law;
- the transverse displacements are small compared to thickness;
- the Love–Kirchhoff's hypothesis of normal to the meridian is valid;

- present approaches neglected the transverse normal stress and restrict the investigation to a plane stress state;
- each structural element which presents geometrical simplex $h/R \leq 0.2$ will be considered thin shell elements, at the same time those who present geometric simplex $0.2 < h/R \leq 0.3$ will be considered moderate shell elements;
- every time, divisions ring having geometric simplex $h/R \leq 0.4$ will be considered moderate-thin, at the same time those who present geometric simplex $f/h \leq 0.2$ will be considered strongly stiffness elements;
- it is assumed the validity of principle of superposition at the same time as the linear governing equations between displacements and internal (external) forces.

It is well known that classical shell theory is suitable for shells called 'thin', having simplex order $h/R \leq 0.1$ (REISSNER, GALIMOV, VLASOV, KRAUSS, et al.), but also it is valid with reasonable accuracy of the results for shells called 'moderate', [12, 14, 19, 22] having simplex order $h/R \leq 0.2 \ldots 0.33$; the differences did not exceed 3–6% that is satisfactory in engineering practice. According to these assumptions and analytical methods available in literature [13–18, 25] we assume that the formulation of thin shell theory could be still applied to our bowl's separator, under axisymmetric loading conditions, without so important changes. The main structural element of the centrifugal disks separator is a typical example for the jointed shells and for small or moderate centrifugal disks separator is a typical joint of short shells. The governing differential equations, for a thin isotropic general shell element under axisymmetric geometric and loading conditions, reported in many other textbooks, e.g. [11, 12, 19, 22], are considered in the manner of FLUGE (1960) and KRAUS (1967) and are reported separately on every structural element of the bowl. Because this analysis is rather cumbersome [1, 4, 12, 23, 24] only the results will be summarised here. Thus the complete equilibrium equations for bending on every structural element of the bowl may be developed in explicit manner with linear terms, or in the following compact matrix form:

$$\{\mathbf{F}_{ik}\} = [\mathbf{a}_{ik}] \times \{\mathbf{C}_i\} \quad \text{for generalised and proper forces,} \qquad (1)$$

$$\{\Delta_{ik}\} = [b_{ik}] \times \{C_i\} \quad \text{for junctions displacements.} \qquad (2)$$

Our notations $\{\mathbf{F}_{ik}\}$ proper forces vector, $\{\Delta_{ik}\}$ partial displacements matrix, $[a_{ik}]$ partial stiffness matrix, $[b_{ik}]$ partial flexibility matrix, $\{C_i\}$ constant matrix elements. A possibility to obtain explicit solutions for the governing differential equations as, internal forces and moments, $Q, N_\varphi, N, M_\varphi, M$ or nodal edge's displacements δ, θ, are based on modified Bessel's functions generally presented in terms of Kelvin's or Thomson's functions. If the following geometrical restrictions are valid:

$$40^\circ \geq \varphi \geq 30^\circ \quad \text{respectively} \ , \ 60^\circ \geq \alpha \geq 30^\circ \qquad (3)$$

and the real part 'x' of Bessel's function are too large as

$$x = 2.828 \cdot \beta \cdot \mathrm{ctg}\,\alpha \geq 10 \quad \text{or} \quad 10 \cdot \mathrm{ctg}\,\varphi \leq \beta = \frac{1.285}{(r_2/h)^{0.5}}, \tag{4}$$

where

$$\beta = 1.285 \cdot \left(\frac{r_0}{h \cdot \cos\alpha}\right)^{0.5} \quad \text{and} \quad r_2 = \frac{r_0}{\sin\varphi}, \tag{5}$$

the explicit expressions for modified Bessel's functions of the second kind of order may be given in asymptotic expansion solutions, to Kelvin's functions. For many technical situations these restrictions and assumptions facilitate a reasonable and quick assessment of different actions with the minimum amount of space and computer time.

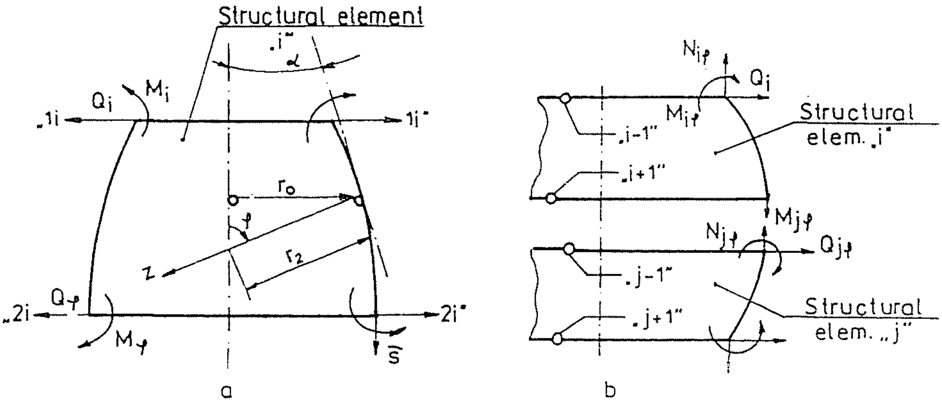

Fig. 1. Structural general element
a. Boundary stress, geometrical conditions and co-ordinate system for a general structural element
b. Theoretical junction between two adjacent structural elements

Like for the previous bending analyses, membrane solution, for every structural element of the bowl subjected to external loads as uniformly internal pressure ($\mathbf{p}_i$), and centrifugal force ($\mathbf{R} \cdot \omega^2$), can be expressed in compact matrix form:

$$\{\Delta_i^*\} = [bk_i^*] \cdot \{e_i^*\}, \tag{6}$$

where $\{\Delta_i^*\}$ additional displacements vector; $[bk_i^*]$ additional flexibility matrix assembling influence coefficients of external stress and $\{e_i^*\}$ partial additional vector of external forces for the 'i-Th' structural bowl's element.

According to the procedure of the force method, with the exact displacement functions solved in modified Bessel functions, dimensional relations (*Table 1*) and other solutions available in literature [12, 16, 17, 23, 24], the explicit functions of internal forces and moments, $Q, N_\varphi, N, M_\varphi, M$, and junctions displacements δ, θ, were determined. Based on previous relations the total boundary displacements and general stress resultants, for any structural element, can be expressed in a general compact matrix form as:

$$\{\Delta_{ik}\} + \{\Delta_i^*\} = [\mathbf{b}_{ik}] \times \{\mathbf{C}_i\} + [\mathbf{bk}_{ij}^*] \times \{\mathbf{e}_{ij}^*\}, \tag{7}$$

$$\{F_i\} = [ak_i] \times \{C_i\}, \tag{8}$$

$$\{\Delta_i\} = [b_i] \times \{e_i\}. \tag{9}$$

Table 1. Geometrical dimensions for structural elements of junction area

Element 's'	Shell	L [mm]	h [mm]	β	ξ	Type
3	Tr. conical	86	3	3.150	7.54187	Interm
4	Cylindrical	30	3	1.035	2.40002	Short
5	Tr. conical	12	3	1.075	1.02557	Short
6	Cylindrical	16	5	1.129	0.71580	Short
7	Plate Ring	$DR = 12$	$g = 8.500$	1.170	–	–
8	Tr. conical	48	8	1.880	2.44137	Short
9	Plate Ring	$DR = 25$	$g = 13$	3	–	–

From the displacements compatibility condition, called deformation continuity, in every wall junction '$i - j$', the following equation in matrix form can be obtained:

$$\{\Delta_i\} - \{\Delta_j\} = 0 \tag{10}$$

or:

$$\{\Delta_i\} - \{\Delta_j\} = [\mathbf{b}_{ij}] \cdot \{\mathbf{e}_{ij}\}, \tag{11}$$

where: $\{\Delta_i\}, \{\Delta_j\}$, displacements matrix (radial displacements and rotations) on outer border of the 'i-Th' element, $[\mathbf{b}_i]$ flexibility matrix of the 'i-Th' element (*Fig. 1*), assembling all dependence factors of internal forces and loading, acting the 'i-Th' element; $\{\mathbf{e}_i\}$, proper total internal forces and loading vector of the 'i-Th' element $[\mathbf{b}_{ij}]$ and $\{\mathbf{e}_{ij}\}$ have the same significance as in previous expression, but relating the total dependence to any junction '$i - j$' of the structure. Ensuring that *Eq.* (11) is satisfied at the same time in each junction of the bowl, a system of simultaneous equations in conventional compact matrix form may be specified:

$$\{\mathbf{D}\} = \{\mathbf{E}\} \cdot [\mathbf{A}] \tag{12}$$

where $\{\mathbf{D}\}$, the vector of structural nodal junction displacements; $\{\mathbf{E}\}$, the vector of equivalent structural forces, and $[\mathbf{A}]$, the equivalent flexibility matrix of the structure.

Because the structure of the bowl must to be in static determination throughout the axle of symmetry, together with equilibrium conditions for internal forces and moments, expressed in every nodal point of junction '**NP**' as

$$\sum(Q_i, Q_j)_{ij} = 0 \text{ and } \sum(M_i, M_j)_{ij} = 0, \tag{13}$$

the solution of the system of simultaneous equations (12), in conventional compact matrix form gives the values of the junctions' end forces $(\mathbf{M}_i, \mathbf{Q}_i)$. Based on boundary prescribed forces and moments at the boundary junctions' end $(\mathbf{M}_i, \mathbf{Q}_i)$, both meridional stress σ_{1c}, and hoop stress σ_{2c}, for bending on every structural element of the bowl may be calculated at any required point of the structural element of the bowl. According to the principle of superposition for a plane stresses state, the general meridional stress $\sigma_{1,3}$, and general hoop stress σ_2, can be expressed as:

$$\begin{aligned} \sigma_{1,3} &= \sigma_{1,3c} + \sigma_{1,3m}, \\ \sigma_2 &= \sigma_{2c} + \sigma_{2m} . \end{aligned} \tag{14}$$

Membrane stresses, $\sigma_{1m}, \sigma_{2m}, (\sigma_{3m})$ for any structural element of the bowl are resulting from the external loads as $(p, r \cdot \omega^2, Fa)$; concrete expressions of these membrane stresses are well known and given elsewhere, e.g. [8, 12, 20, 23, 24], thus they are excluded from this work. Based on von Misses' yield criteria,

$$\left(\sigma_1^2 - \sigma_1 \cdot \sigma_2 + \sigma_2^2\right)^{0.5} = \sigma_e \tag{15}$$

resultant stresses, σ_e, can be calculated at any required point of the element.

3. Numerical Applications

Based on the approaches outlined in the preceding paragraphs, a computer program, for a small centrifugal disks separator having nominal diameter $D_n = 0.160$ m, with maximum flow capacity $Q_{\max} = 400 \ldots 1000$ l/h, written in Turbo Pascal and implemented on an IBM compatible PC, has been developed [23, 24]. This allows an elastostatic analysis of the bowl under specified axisymmetric loads as: (1) central load induced on mounting 'Fa', (2) uniform pressure or pressure varying between adjacent nodal lines 'p_1', and (3) centrifugal load $(r \cdot \omega^2)$. First the structure of the bowl is subdivided into a number of various elements of constant thickness (*Fig. 2b*), so any junction adjacent elements become junctions related to a circumferential nodal line at the midshell of the wall. Then *Eqs.* (12)–(13) are solved together loading conditions to evaluate the structural response fields; some dimensional details of the test structure are given in *Table 1*. A number of experimental studies [1, 3, 4] reveal the beginning of the cracking stage more

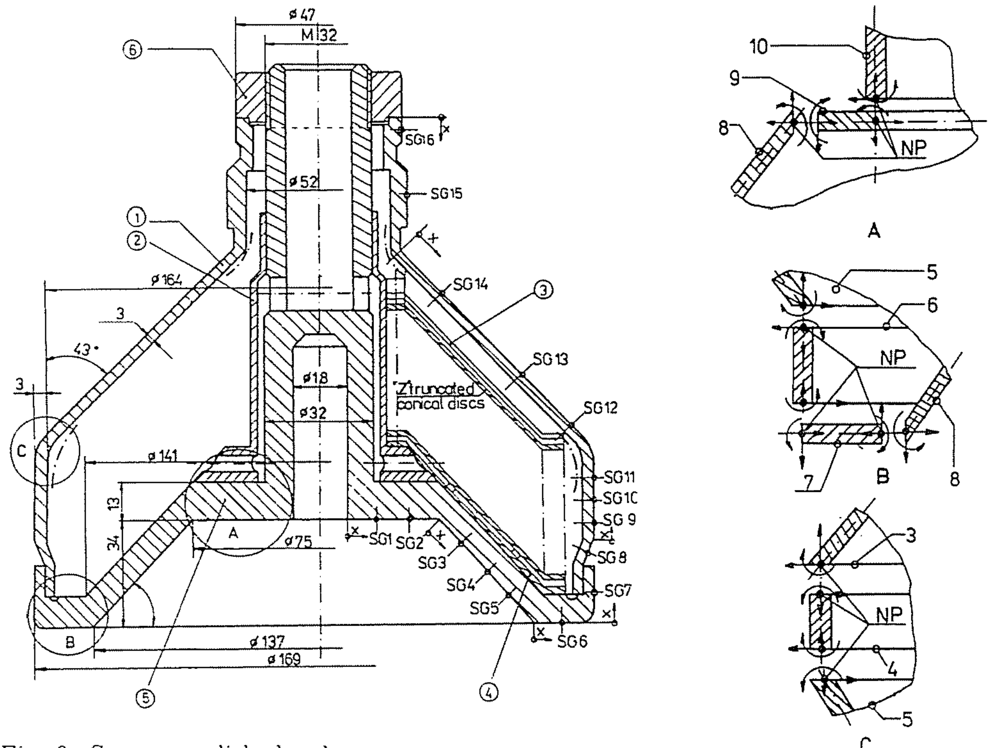

Fig. 2. Separator disks bowl

a. Cross-sectional half view in elevation of the bowl and real displacements for strain gauges: (1) – outside part of the bowl; (2) – guiding disks mark; (3) , (4) – separator disks, $Z = 40$ pieces; (5) – inner bottom of the bowl; (6) – central nut for assembling the bowl;

b. Discretising junction area and replacement of the bowl by a discrete system of structural elements;

frequent in junctions corresponding to '*A*', '*B*' and '*C*' area, (*Fig. 2ab*); by this reason in the study to follow we assign a special importance to these areas.

Others employing conditions of the test structure are central load induced on mounting, as $\mathbf{F_{1a}} = 1480$ N, $\mathbf{F_{2a}} = 5120$ N and angular velocity ω, as $\omega_1 = 136\ \mathrm{s}^{-1}$ and $\omega_2 = 450\ \mathrm{s}^{-1}$.

Because in this study the separator's bowl is one empty without any suspension, internal pressure $\mathbf{p_1} = 0$. Typical results of different bowl's actions and the influence of the central load F_a induced for assembling the bowl in the critical area of centrifugal disks separators are presented in form of tables and graphs for stresses. Tightening load, 'F_a' induce on mounting by central nut (6) for assemble the bowl, tensions or compresses

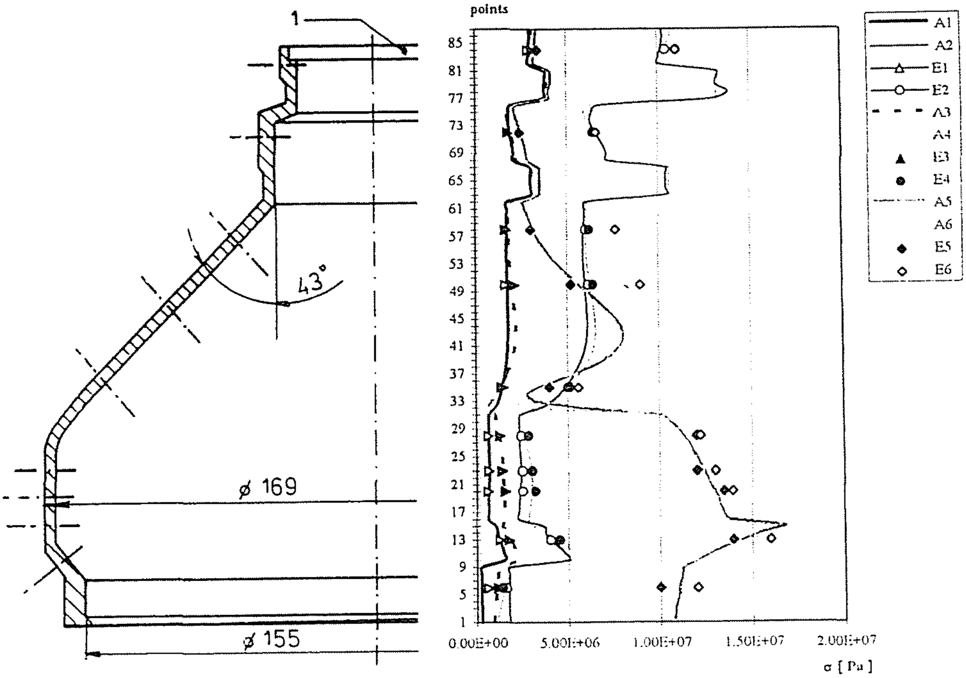

Fig. 3. Variation of equivalent stress along the outer surface of the outside part of the bowl, structural element (1)
$A1, E1$ – theoretical and experimental stresses at mounting with prestressed F_{a1}; $A2, E2$ – theoretical and experimental stresses at mounting with prestressed F_{a2}; $A3, E3$ – theoretical and experimental stresses at central load F_{a1} and angular velocity ω_1; $A4, E4$ – theoretical and experimental stresses at central load F_{a2} and angular velocity ω_1; $A5, E5$ – theoretical and experimental stresses at central load F_{a1} and angular velocity ω_2; $A6, E6$ – theoretical and experimental stresses at central load F_{a2} and angular velocity ω_2

the structural elements of the bowl (*Fig. 2*). Therefore equivalent stresses increase varied in any structural element but gradually with the increasing of the central load (*Figs 3–4*). Though the stresses are small in nominal value in this stage (*Figs 3–6*), for running centrifugal separator they reveal evidently increasing with angular velocity ω. The point that weighs, for the inner bottom of the bowl (5), is that the discontinuity and the intensity of stresses state decease simultaneous with the increase of central load and rotational speed. Necessity for high prestressed central load leads to strong stress in the vicinity of junctions 'B' and 'C' on the assembling stage; their general results consist in decreasing of stresses simultaneous with the increasing of

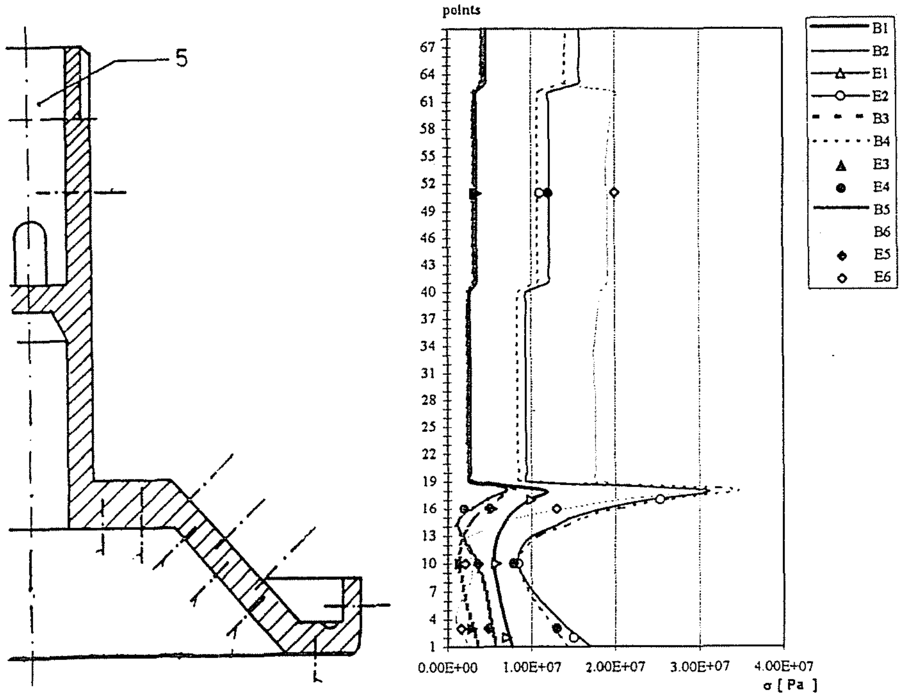

Fig. 4. Variation of equivalent stress along the inner surface of the inner bottom of the bowl, structural element (5)
$B1, E1$ – theoretical and experimental stresses at mounting with prestressed F_{a1}; $B2, E2$ – theoretical and experimental stresses at mounting with prestressed F_{a2}; $B3, E3$ – theoretical and experimental stresses at central load F_{a1} and angular velocity ω_1; $B3, E3$ – theoretical and experimental stresses at central load F_{a2} and angular velocity ω_1; $B5, E5$ – theoretical and experimental stresses at central load F_{a1} and angular velocity ω_2; $B6, E6$ – theoretical and experimental stresses at central load F_{a2} and angular velocity ω_2

rotational speed. For the outside part of the bowl (1) is establish a highly increasing (*Fig. 3*) of the prestress state of the bowl due by the increasing of the central load, 'F_a' induced on mounting, simultaneously with increasing of rotational speed. The faster increase of the stress state, with prestressed central load $\mathbf{F}_a$ and rotational speed $\mathbf{n}$ occurs in junctions corresponding to '*A*', '*B*' area (*Figs 5–6*). The same trend, concerning stress state increasing, is established in '*C*' area, but the rise is not so fast; maximum values of the stresses, approaching a constant value of about 80 MPa occur in junction '*C*', [23, 24] it is much less than admissible value of stresses about 150 MPa. The influence of the tightening load, 'F_a' strongly decreases around the value angular velocity $\omega = 890$ s^{-1}, which represents the nominal value for industrial condition of this type of centrifugal separator, when rotational speed is about $n = 8500$ rot/min. According to the previous analyses we can specify, for this small type of discs centrifugal separator with central nut for assembling, the major influence of the central load F_a induced on mounting, for the stress state and implicit for the field of industrial speeds.

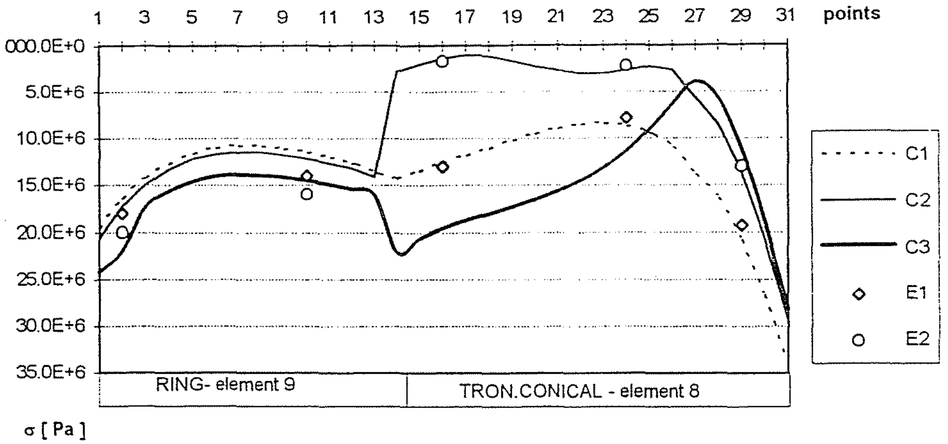

Fig. 5. Variation of equivalent stresses around the junction '**A**' of the bowl on the inner surface for central load $\mathbf{F}_{a2}$.
$C1, E1$ – theoretical and experimental values for angular velocity $\omega_1 = 136$ s^{-1}; $C2, E2$ – theoretical and experimental values for angular velocity $\omega_2 = 450$ s^{-1}; (c) Modelling for angular velocity $\omega_3 = 890$ s^{-1}

4. Experimental Study

One relative small bowl, having nominal inner diameter $D_n = 160$ mm, is used for the experiments in the previous conditions. It is a constitutive part of a centrifugal separator type TSL-400, with nominal flow capacity $Q_{\max} = 400$ l/h with $Z = 40$ separating disks at a maximum rotation speed, $n = 8500$ r/min. Experimental study based on recording surface strains at selected locations of the bowl is performed with strain gauges constantan wire type TER.5H. 120–INCERC Bucharest, having nominal resistance $R = 126\ \Omega$, and specific constant $K = 1.9 \pm 1.5\%$. Recording of specific strains has been done by strain bridge with 6 channels, type $N - 2302$ IEMI Bucharest, frequency band 5000 Hz, precision range $0.3 - 5\%$. Transfer of electric signal from strain gauges to recorder bridge was obtained by a collector with 4-mercur rings, for one point of location. Detailed dimensions of the test

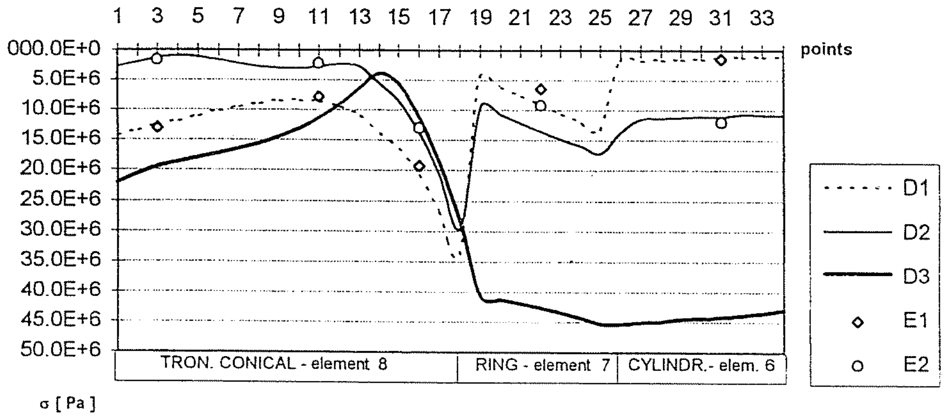

Fig. 6. Variation of equivalent stresses around the junction '**B**' of the bowl on the inner surface for central load $\mathbf{F}_{a2}$.
$D1, E1$ – theoretical and experimental values for angular velocity $\omega_1 = 136\ s^{-1}$; $D2, E2$ – theoretical and experimental values for angular velocity $\omega_2 = 450\ s^{-1}$; (c) Modelling for angular velocity $\omega_3 = 890\ s^{-1}$

specimens and the test rig are given in *Table 2*, dial gauges arrangement is illustrated in *Fig. 2*; experimental work is performed employing the electrical resistance bridge method. It should be pointed out that recording of strains at maximum rotation speed $n = 8500$ r/min was not possible is safety conditions by our collector configuration.

Experimental values of the strains are processed assuming Hooke's law for a plane stresses state, corresponding to given measurements of principal strains $\varepsilon_1, \varepsilon_2$, or ε_3 as follows

$$\begin{aligned} \sigma_1 &= \frac{E}{1-\mu^2} \cdot (\varepsilon_1 + \mu\varepsilon_2), \\ \sigma_2 &= \frac{E}{1-\mu^2} \cdot (\varepsilon_2 + \mu \cdot \varepsilon_1). \end{aligned} \tag{16}$$

According to von Mises criteria and [21, 26, 27] an algorithm for the computation of the experimental stresses state was developed. The resultant equivalent stresses at selected locations presented together theoretical plots prove a reasonable agreement between the homologous values of theoretical and experimental stresses, the maximum casual difference between these was less than 10...15%.

Table 2. Some experiments stresses and position of strain gauges

Structural element 's'	Strain gauge	Experimental stresses for $\omega_1 = 136$ [s^{-1}] σ_e [MPa]		Experimental stresses for $\omega_2 = 450$ [s^{-1}] σ_e [MPa]		Position of strain gauges
		Fa$_1$	**Fa$_2$**	**Fa$_1$**	**Fa$_2$**	x [mm]
Tr. conical-3	SG12	1.5	5	4	5.6	12
	SG13	2.1	6.4	5.2	9	31
	SG14	1.7	6.2	3	7.7	75
Cylindrical-4	SG9	1.6	3.2	13.5	14	5
	SG10	1.4	3	12	13	15
	SG11	1.3	2.8	12	12.2	25
Tr. conical-5	SG8	1.8	4.5	14	16	7
Plate ring-9	SG1	–	18	–	20	5
	SG2	–	14	–	16	18
Tr. conical-8	SG3	2.9	13	4.85	1.65	5
	SG4	1.45	7.85	3.7	2.2	25
	SG5	5.3	19.3	5.13	13	45
Plate ring-7	SG6	–	13	–	1.65	8.5
Cylindrical-6	SG7	1.17	1.4	10	12	10

5. Conclusion

These analytical approaches and concrete employing of the flexibility matrix method have reasonable results as shown by comparison with experimental data. The method is applicable for structures of bowls subjected to various axisymmetric loading conditions and for different boundary conditions and jointed structural elements. In general the absolute maximum values for the stress resultants obtained from the present analysis are seen to be localised in the same junctions with those of beginning the cracking stage. The experimental study reveals a reasonable agreement between theoretical and experimental values. Reasonable positioning of the critical junctions and some major influences of the central load F_a induced on mounting for the stress state, give a real possibility in the design work; as easy, hasty and safe preliminary analysis of any structures of centrifugal disks separators is accessible.

Appendix

Other notations

h	–	customary thickness;
f	–	normal deflection for plate ring element;
R	–	customary radius;
$R_{med}, R_{max}, R_{min}$	–	average, maximum and minimum radius;
$B = R_{max}/R_{min}$	–	geometrical simplex;
$DR = R_{max} - R_{min}$;		
ξ	–	functional co-ordinate regarding internal proper forces and stress with explicit particular formulas, like $\xi = 1.285/(R \cdot h)^{0.5}$ for cylindrical element and $\xi = 2.57 \cdot (R \cdot \cos\alpha/h \cdot \sin^2\alpha)^{0.5}$ for truncated conical element
φ	–	opening angle of the section's element;
α	–	normal semiangle of element, or semiangle of truncated conical element;
L	–	physical length of structural element;
n	–	rotation speed, equal to $30 \times \pi \times \omega$
r_1, r_2	–	the first and the second principal radius of curvature;
ϑ	–	rotation of the meridional tangent
δ	–	radial displacement

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