Numerous experimental investigations of crankshaft strength have shown that none of the present design methods can be used to determine the forces acting in cross sections of the crankshaft with sufficient accuracy [1]. This includes the continuous method which considers actual shaft and bearing stiffness.

A new method has been developed at LenNIIkhimmash for more accurately calculating the strength of crankshafts of opposed compressors with cantilevered electric motor drivers. The calculation is carried out on a digital computer.

The special features of this method are: 1) inclusion of gaps in the crank bearings which means that the shaft does not have to be supported by all the bearings, 2) determination of design misalignment of the intermediate bearings, including the actual geometry of the bearing bushings and the crankshaft journals and the introduction of a relationship between the direction of reaction in the bearings and misalignment, and 3) more accurate determination of unidirectional magnetic attraction of the (electric motor) rotor to the stator during compressor operation.

Figure 1 is the calculation schematic for a five-bearing crankshaft of a four-row compressor. Projections $P_{i,x}$ and $P_{i,y}$ of active forces in representative sections 6-11 and projections $R_{i,x}$ and $R_{i,y}$ (here $i$ is the number of the cross section or bearing) of unknown reactions 1-5 of the bearings are shown in x-y coordinates. The coordinate origin is located on the axis connecting the centers of the outer (main) 1 and 5 bearings (supports) soles; the y axis is positive upwards and the x axis is positive to the right (looking at the compressor from the electric motor side).

The calculation is carried out with the assumption that the active forces acting in cross sections 7-11 are known. It is also assumed that the weight coefficients $V_{i,k,y}$ and $V_{i,k,x}$ are also known. These coefficients correspond to the magnitude of deflection in the i-th section when a unit force directed along the given axis is applied in the k-th section of the shaft supported only by the extreme bearings.

It is assumed in the calculation that each shaft support and each crankshaft journal represents a closed order curve within the surface perpendicular to the axis of the bearing soles (in further discussion this is called the bearing or journal contour) and is represented by the equation

$$Ax^2 + By^2 + Cx + Dy + 1 = 0.$$  (1)

Parameters A, B, C, and D take into account wear of the bearing bushings or crankshaft journals and are determined by measuring wear using probability methods [2].

Probable values of coefficients A, B, C, and D are determined from data of n measurements of wear by solving the equations on the following page.

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\[
A \sum_{i=1}^{n} x_i^2 + B \sum_{i=1}^{n} x_i y_i + C \sum_{i=1}^{n} x_i + D \sum_{i=1}^{n} y_i = 0
\]

(2)

Coordinates \( x_i \) and \( y_i \) are determined from the following functions:

for the bearing contour

\[
x_i = (h_i - h_{avg}) \cos \beta_i; \quad y_i = (h_i - h_{avg}) \sin \beta_i;
\]

for the journal contour

\[
x_i = (r_i - r_{avg}) \cos \beta_i; \quad y_i = (r_i - r_{avg}) \sin \beta_i;
\]

where \( h_i \) and \( r_i \) are the measured bushing thickness and modulus of the radius-vector of the journal cross section at angle \( \beta_i \) to the positive direction of axis \( x \) (at the shaft position corresponding to the first-row upper dead point) in m; \( h_{avg} \) and \( r_{avg} \) are the average (within the fabrication tolerances) bushing thickness and the radius-vector modulus of the shaft journal in m.

Simultaneous solution of Eq. (1) and the straight-line equation

\[
y_i = x_i \tan \beta_i.
\]

is used to find coordinates \( x_{i, s} \) and \( y_{i, s} \) of the intersection points of the bearing contour and the straight line drawn from the bearing contour center at angle \( \beta_i \) to the x axis.

Angle \( \alpha \) of crankshaft rotation, measured from the position corresponding to the upper dead point of the first compressor-piston row (Fig. 2), must be used in determining coordinates \( x_{i, s} \) and \( y_{i, s} \) of the intersection point of this line with the shaft journal contour in a stationary coordinate system. Coordinates \( \eta_{i, s} \) and \( \psi_{i, s} \) of the intersection of this straight line in the internal \( \eta-\psi \) coordinate system (this is rigidly connected to the shaft) can be determined by simultaneously solving Eq. (1) of the journal contour (replacing \( x \) by \( \eta \) and \( y \) by \( \psi \)) and \( \psi_{i, s} = \eta_{i, s} \tan(\beta_i + \alpha) \). The axes of coordinate system \( \eta-\psi \) match the axes of the stationary \( x-y \) coordinate system at \( \alpha = 0 \).

We next find

\[
x_{i, s} = \eta_{i, s} \cos \alpha + \psi_{i, s} \sin \alpha; \quad y_{i, s} = \psi_{i, s} \cos \alpha - \eta_{i, s} \sin \alpha.
\]

(3)

(4)

Two iterations are made in designing a shaft. Reactions of bearings at constant unidirectional magnetic attraction of the electric-motor rotor to the stator \( Q_m \) are determined in the first iteration (inner). This iteration is completed if the assumed conditions of shaft force on the bearings and the bearing reactions calculated are equal.

The second iteration (outer) establishes the equality of the assumed and calculated position of the rotor axis. It must be satisfied for more correct determination of forces \( Q_m \) since one more degree of static indeterminability is introduced into the calculation by \( Q_m \) as a function of shaft deformation at the point of rotor settling. It is obvious that the first iteration is an integral part of the second.