REDUCING OF DYNAMIC DEFLECTION OF SHAFTS FOR MIXING DEVICES BY OPTIMIZING THE DISTANCE BETWEEN SUPPORTS

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Abstract: For improving of the design for vertical reactors with mixing devices, the establishing of optimal length between the supports of bearings is an important step in developing of operational parameters of these apparatus.
Usually, for the shafts having a high rigidity, the distance between the bearings is bigger than for the elastic shafts having the same length.
The clearance in bearings has an important influence on the optimal length of the distance between supports of bearings and on the absolute value of the deflection.
Generally, the deviation $l_{1,opt}$ to small values is fewer dangerous than the deviation to high values.
Keywords: own theoretical frequency, own real frequency, optimal length between bearings

1. INTRODUCTION

The vibrations which accompany the elements of the reactors with mixing devices contribute to reducing of quality of mixing process or to damaging of these components**.**

The amplitude of the vibrations may be influenced by the same external factors. So, m many cases, a bearing support or a lid on which the support is set having a low rigidity, may contribute to appearing of vibrations of mixing devices. This means that the whole device (impeller, shaft vessel tower, lid) must have a good rigidity**.**

The mixed materials (in which the shaft with its impeller rotates) influence both the level of own frequency and the amplitude of vibrations.

Under the action of the external stresses, the shafts of the mixing devices may have flexional. torsional or axial vibrations. Practically, the vibrations problem is simplified to verify the behavior under flexional vibrations, because they are dangerous for the shafts of mixing devices.

On the fundaments of experimental researches, it reached to the conclusions [6] that the own pulsating (or critical rotating speed) of "j" degree, has the form .

$$\omega_{cr,\,j} = (k_a\, k_\mu\, k_\rho\, k_b\, k_s\, k_g)\, \rho_j \tag{1}$$

p *j* - own pulsating of the mixing devices, which is considered in vacuum
$k_{a\,r}\, k_g$ - coefficients which introduce the effect of the axial force and of the gyroscopic moment
$k_{b\,r}\, k_s$ - coefficients which show tie influence of the bearings and of the sealing of the shaft
$k_\mu\, k_\rho$ - coefficients which introduce the influence of the viscosity and of the density of the mixed fluids

2. THE INFLUENCE OF THE LENGTH BETWEEN THE BEARING SUPPORTS OF THE SHAFTS TO ITS VIBRATION

The study of mixing devices supposes the optimization of its constructive and functional solutions and establishing of same optimization criterion which must be generally valid (without the type "of mixing device). In specialty publications it is recommended to use the following optimization criterion :

- mixing device type
- necessary power
- minimum sag (deflection
- critical rotational speed
- mass of the device
- price of the device

Theoretical calculus and the experiment prove that the problem of the minimum deflection and of the critical rotational speed (out of the elastic deformation of the shaft, the clearance inside the bearing supports), may be solved also through the optimization of length of the distance between the supports.

The noise and the vibrations which accompany the working of the elements of reactors with mixing devices may be reduced by the optimum placing of bearing supports for rotating shaft for minimum deflection under the bending force.

So, a method for quality improving and working safety consist in establishing the length of the distance between the bearing supports, using an optimization computation, considering a minimum displacement of the extremity of the shaft which is placed in cantilever (figure 1)

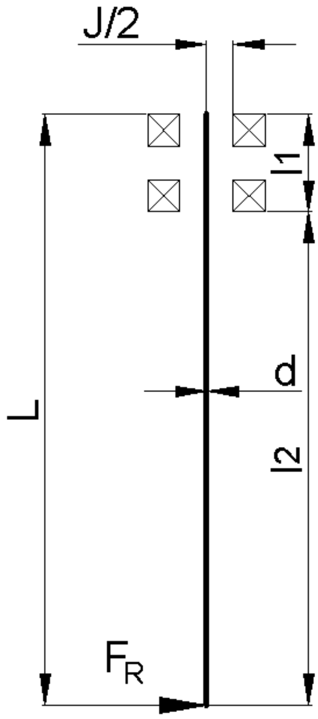

Figure 1 Computation scheme for the cantilever driving shaft of the mixing device

In specialty papers, [5] [8] [1], present computing formulas for optimal length of the distance between the bearing supports and for the ratio of the optimal length between bearings and the cantilever length.

The drawback of these formulas consists in the manner in which the optimum length of the distance between the bearing supports is not simultaneously dependent on the type of the beatings and on their hydrodynamical loading.

The author of this paper based on the theoretical studies and on experimental determinations, have obtained a formula for computation of (be length of the distance between bearing supports, depending on the type of the bearings and the force: F^*_R.

Figure 1 presents the loading scheme for vertical cantilever shaft with impeller placed at the extremity. The upper supports take-up the axial and the radial stresses, and down support only takes- up radial stresses.

The value of the deflection y, has the general form:

$$y_t = y_t + y_2 + y_3 \tag{2}$$

where: y_1 - the deflection caused by elasticity of the shaft
y_2 - the deflection caused by elasticity of the supports
y_3 - the deflection caused by clearance inside the bearing support

The value of deflection, which is caused by elasticity of the shaft y_1, depends on the value of radial component of resulting force: F_R* taking into account the radial hydrodynamic force: R_R, and the centrifugal force F_R caused by the remanent unbalancing (from defective assembling or manufacturing) – e, e_c is residual eccentricity of the centre of mass.

$$F^*_R = F^*_{hd,R} = + / - F^*_{cf} \tag{3}$$

The "+" sign is accorded to the case when centrifugal force contributes to the increasing of the deflection (caused by the radial hydrodynamical force), and the "-" sign is accorded in the case when this contributes to decreasing of the defection.

Hill and Kime [4] show an approximate formula for computation of the radial component part of the hydrodynamical force:

$$F^*_{hd,R} = \frac{19 N_a k}{n d_a} \tag{4}$$

where: N_a - necessary power for mixing process
n - rotational speed
d_a - overall dimension of the impeller k - coefficient dependent on working conditions

For establishing of this kind of force it must take into account that it is an important difference between a stationary working conditions and non- stationary working conditions (when its value and direction arc continuously changed).

Also, for establishing the value of the radial component part of the hydrodynamical force, it is possible to use the dimensional analysis. So, Bernoulli's equation to:

$$F \sim p n^2 d^4 \tag{5}$$

This makes possible the determination of the components:

$$F_A = C_A pn^2 d^4$$
$$F_R = C_R pn^2 d^4 \quad (6)$$
$$F_T = C_T pn^2 d^4$$

where : p – density
n - rotational speed
d - overall dimension of impeller

$$F_{cf} = (m_a + m_b)(e_c + e)\ \omega^2 \quad (7)$$

where: m_a - the mass of impeller
m_b - the concentrated mass of the shaft, considered to be placed in the centre of mass of the impeller These masses depend on the kind of mixed suspension and on the number of impellers.
e_c - residual eccentricity of the centre of mass
e - the remanent unbalancing (caused by inadequate setting and manufacturing)

It is observing feat in accordance with the formulas: (3), (4), (5), (6), (7), the value of the F_R* doesn't depend on the length of the distance between the bearing supports, so it will be considered constant in deriving operation by $I_{1,apt}$

The deflection of the shaft y_1, which is caused by the stress of a radial force F_R*, is [3]:

$$Y_1 = \frac{F^*_R L}{3\ EI} (L+L_1) \quad (8)$$

where: L – the length of the shaft
1_1- the distance between bearing supports
E - modules of elasticity
I- moment of inertia

Usually, if "j" is the clearance of bearing (the eccentricity of external ring function by the internal ring), the deflection of the extremity of the shaft, under a force loading has the form [5]:

$$Y_2 = K_1 d^{1-2p} (F^*_R)^{\ p} + 0{,}6j \quad (9)$$

where ;
d - internal diameter of the ring (diameter of the shaft)
p - 2/3 in the case of the single -row roller bearing
K_1 - constant which depends on the type and on the serial of bearing.

Following of clearance "j" of bearings, it may be possible to define the adequate deflection, y_3-, will be [1]:

$$Y_3 = \frac{J(L - 1_1)}{1_1} \quad (10)$$

The values of the clearance "j^r is standardized in the catalogues. From the point of view of the maximum stress is considered the most unfavorable case when there is a deficient fitting of the bearing and the clearance remain the same after the fitting.

Considering the overlapping of the effects, according to the formulas (2), (8), (9), (10), it results:

$$Y_3 = \frac{F^*_R L}{3EI}(L - 1_1)^2 + K_1 d^{1-2p} (F^*_R)^p + 0{,}6j + j\frac{L - 1_1}{1_1} \tag{11}$$

For obtaining the optimum length of the distance between bearing supports, 1, it is given the condition for minimum: $\frac{dy_t}{dl_1} = 0$

$$\frac{dy_t}{dl_1} = 2L\, F^*_R 1_1 + 2\, F^*_R 1_1^3 - 3jEI = 0 \tag{12}$$

Aided by graphical solving, it was obtained the value from the table I:

Table 1

	d[mm]	Fr[N]	L[mm]	$I_{1,opt}$[mm]
1	50	30	1700	417.33856254
2	60	49	1860	477.55358770
3	65	58	1950	501.10337601
4	70	67	2070	540.74208987
5	80	82	2450	612.89913023
6	90	110	2840	674.05260433
7	95	122	3000	692.48433771
8	100	134	3220	711.57492394
9	110	160	3600	755.21223548
10	125	218	3890	790.21582622
11	130	250	4050	825.74890632

It is observing, from (12) that the $l_{1,opt} = f\,(d, F_R^*, L)$. So, the optimum length depends; on the radial component of the hydrodynamical force, on the total length of the shaft and on its diameter and on the type of bearing.

3. CONCLUSIONS

Specification that doesn't exist a study of vibrational behavior of the shafts for mixing devices has some deficiencies because the computation of the shafts of these apparatus lakes partly into account the real stresses. It is necessary to make the specification that it doesn't exit a unitary methodology for computation of this kind of shafts.

Praising the influence of the bearings on theoretical own frequency, authors make an optimization computation of the length of the distance between the supports of the bearings.

The obtained formula and the corresponding diagrams can be used by the mechanical engineers *on* design work, in the field of mixing devices.
There is an important different behavior under vibration using bearing with balls and with rollers. So, the distance between bearing supports is different for the shafts with the same length, having the same loading, but having different bearings.

Although, he used the similar formulas and similar computation methodology, Afanastev [1] has obtained a different result, more simple, because he considered that the cantilever length is constant, in point of 1_1. The authors of this paper considered this way is wrong and they expressed the cantilever length fraction by I_1 : L_2 = L- I_1 (figure 1). So, the single constant length, in derivation operation, in formula (12), is the total length L. It was obtained a computation formula for $l_{l,opt}$, more complex but more exactly. Also, the final computation formula, for $l_{l,opt}$, contains the value of clearance "j"; il is pointing out the dependence of l_1 by the type of bearing. This dependence doesn't appear in computation formula for $l_{l,opt}$, which was obtained by Afanastev [1]

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