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THE CALCULATION OF THE IMPELLER OF THE PUMP

Abstract. The article presents the results of theoretical studies of the parameters of the impeller of a centrifugal pump, set limits on the width of the blades, which will not have a negative impact on the effective performance of the pump and provides optimal capacity with a minimal power consumption, as well as the performance of the reactor accelerator.

Key words: parameters of the pump, the impeller, the acceleration process of petrification reaction, the reactor production of biodiesel, the number of blades.

Introduction. The mechanism of accelerating the process of obtaining biodiesel isconsidered, which represents the screw-shaped surface of the accelerator and Laval's snot, accelerates the methanolysis reaction (*triglycerides* in vegetable oil being reacted with methanol), which takes place in a reactor with a constant working volume and in the presence of alkali (as a catalyst) when heating the initial components. The main elements of the mechanismanacceleration reaction is the vane pump and byimproving its parameters, the optimum performance of the accelerator is achieved. On the basis of the above, the theoretical studies to justify the parameters of the pumpisconducted.As a parameter,that affecting for the number of blades, often acts a coefficient of specific speed, which depending on the flow rate, head and shaft speed.

This approach has the following disadvantage - the lack of consideration of the thickness of the blades when determining their quantity. If, when calculating high-pressure pumps with a large flow, the influence of the blade thickness is not significant enough, then for low-flow pumps, the effect of the blade thickness is so significant that the number of blades, which obtained when considering only the coefficient of specific speed, is incorrect. This leads to the need to find a universal formula for the optimal number of impeller blades for all ratios of thickness, flow and shaftspeed of the pump.

Suppose that all the essential design parameters of the pump as the rotational speed of the pump shaft, the impeller inlet and outlet diameters, the width of the blades at the inlet and outlet of the impeller, the angles of the blades at the inlet and at the blade output and the design of the outlet at the time of determining the optimal number of blades have already defined and unchanged.

Methods. It is also assumed that this calculation is being done at a fixed rate. The design scheme of the considered pump is a single-stage cantilever pump with a spiral outlet. To solve this problem, we will determine the optimization criterion - pump head, since this is one of the most important parameters of the pump [1, 2].

To reduce head pressure losses, the width of the blades at the entrance is selected in such a way that the speeds of the blades at the entrance to the wheel and at the entrance of the blades are equal. Therefore, the area of the corresponding sections will beequal.

$$\frac{\pi}{4}\left(D_0 - d_{CT}^2\right) = \pi\, D_1\, b_1; \quad hence\;\; b_1 = \frac{\left(D_0 - d_{CT}^2\right)}{\eta\, 4 D_1},$$

where μ ≈ 0,9 – the coefficient of constraint in the input section bythe impeller blades.

As there is no flow swirl at the impeller inlet, then

$$C_{1r} = C_1 = \frac{Q}{\pi D_1 b_1 \eta \eta_0}.$$

The peripheral speed at the impeller inlet and the blades inclination angle will be determined by the formulas:

$$u_1 = \frac{D_1 \omega_0}{2}; \qquad \beta_1 = arctg\,\frac{C_{1r}}{u_1}$$

Usually this angle of the blades ranges from 20 to 30°.

To reduce hydraulic losses in the impeller, the normal [3] component of the liquid flow rate in the impeller is taken constant, that is,

$$C_{2r} = C_{1r} = C_1.$$

Outlet blade angle β_2 ranges from 20 to 70°, in that regard we choose $\beta_2 = 45^0$. Then from the basic equation of the impeller, we can determine the necessary peripheral speed at impeller exit:

$$u_2 = \frac{1}{2}\left[C_{2r} ctg\beta_2 + \sqrt{(C_{2r} ctg\beta_2)^2 + \frac{4gH}{\eta_г}}\right]$$ considering $ctg45^0 = 1$,

$$u_2 = \frac{1}{2}\left[C_{2r} + \sqrt{C_{2r}^2 + \frac{4gH}{\eta_г}}\right]$$

The impeller outer diameter, the blades width on the outer diameter and the blades number are determined by the formulas:

$$D_2 = \frac{2u_2}{\omega_0}; \quad b_2 = \frac{b_1 D_1}{D_2}; \quad Z = 6{,}5\mu \frac{(D_2 + D_1)}{(D_2 - D_1)} \sin\left(\frac{\beta_1 + \beta_2}{2}\right).$$

Typically, the number of blades Z = 8.

Based on the foregoing calculationsof the pump impeller parameters and the equations of the pump shaft power [4].

$$N = \frac{\rho\, g\, H\;\, Q}{\eta} \tag{1}$$

As well as the pump capacity isassociated with the average speed at the exit of the impeller with the ratio:

$$Q_p = \pi\, D_2\, b_2\, C_{2r} \tag{2}$$

Taking into account the expression of thepump capacitythrough the parameters of the disk D_2 and $b_2$equation (2) to gets the form:

$$N = \frac{\rho\, g\, H\;\, \pi\, D_2\, b_2\, C_{2r}}{\eta} \tag{3}$$

Consideringη = 0, 96; andρ = 0,850kg/m^{2}for oil;and by the Euler equation, we determine the theoretical head of flow at the outlet of the centrifugal pump:

$H = \frac{u_2\, C_{2u}}{g}$, and in $\beta_2 = 45^0$ then $C_{2u} = C_{2r}$ and considering $u_2 = \frac{D_2 \omega_0}{2}$

Taking into account the above values, as well as the expressions $Q = C_{2r}^2 \omega_0$ and after some abbreviations, equation (3) transforms into:

$$N = \frac{4 D_2 b_2 Q}{\eta} \tag{4}$$

Equation (4) allows the construction of operating characteristics of the pump, taking into account the regulation of consumed N - the pump shaft power, kW; Q - volumetric pump flow, m^3/h; H - pump head, pump head, (m water column) m w.c.; and η- the pump efficiency, depending on blade width b_2 changes, based on the use of proportionality formulas.

In the majority of literary sources [5, 6], for constructing the operating characteristics of pumps in the frequency control of performance, an approach, based on the use of proportionality formulas has been proposed.

The formulas of proportionality, obtained from the provisions of the theory of similarity of dynamic machines, reflect the change in the operating parameters of the pump with a change in the rotor speed, impeller diameter, etc. So, if the pump characteristics are known at the nominal rotor speed, whenits changing, the operating parameters can be determined by the expressions

$$\frac{Q}{Q_p} = \frac{n}{n_н} \frac{\eta_{\text{rot}}}{\eta_{\text{rot,p}}}; \qquad \frac{H}{H_p} = \left(\frac{n}{n_p}\right)^2 \frac{\eta_{\text{h}}}{\eta_{\text{h,p}}}; \quad \frac{N}{N_p} = \left(\frac{n}{n_p}\right)^3 \frac{\eta_{\text{p}}}{\eta}; \tag{5}$$

where Q - volumetric pump flow, m^3/h; H - pump head, m.w.f.; N - pump shaft power, kW; n - the number of revolutions of the pump rotor, r/min; η_{rot}, η_{h}, η - volume, hydraulic and full efficiency of the pump, respectively; the index "n" indicates the value of the parameter in the nominal mode of the pump, i.e. at the nominal rotor speed.

For practical calculations, formulas (5) are applicable only conditionally, since the functions of changing η_{rot} and η_{h}depending on the rotor speed in most cases are absent. In this regard, it is recommended to use simplified expressions obtained under the assumption that the hydraulic and volumetric efficiency of the pump remain unchanged at any rotor speed:

$$\frac{Q}{Q_p} = \frac{n}{n_p}; \quad \frac{H}{H_p} = \left(\frac{n}{n_p}\right)^2; \quad \frac{N}{N_p} = \left(\frac{n}{n_p}\right)^3; \quad \eta = \eta_p \tag{6}$$

Taking into account the presented results of the analysis, it became necessary to develop a more universal mathematical model, which allows to predict with sufficient accuracy the character of pump performance at varying rotor speed even in the absence of a large amount of experimental data [7, 9].

The resulting equations are refined to fully account for the physical nature of the processes occurring in the pumps when the rotor speed changes, as well as to summarize many experimental data. Basic calculation expressions:

$$Q = Q_p \left(\frac{n}{n_p}\right)^{r+2} = \left(\frac{\dfrac{H}{AQ_p^2}}{\dfrac{H_p}{AQ_p^2}}\right)^{0,5}; \qquad H = H_p \left(\frac{n}{n_p}\right)^2; \tag{7}$$

$$N = \frac{\rho g H_p Q_p}{1000*3600\eta_p} \left(\frac{n}{n_p}\right)^3; \quad \eta = \eta_p \left(\frac{n}{n_p}\right)^{r+1} \left(\frac{\dfrac{H_p}{AQ_p^2}}{\dfrac{H_p}{AQ_p^2} - 1}\right)^{0,5}; \tag{8}$$

$$A = \frac{1}{1620000 g \pi^2} \left(\frac{1}{d_h} - \frac{1}{d_s}\right)^3; \tag{9}$$

where Q – volumetric pump flow, m^3/h; H – pump head, m.w.c; N – the pump shaft power, kW; η_p – full efficiency of the pump; n – the pump rotor speed, r/min; A – auxiliary complex; g – acceleration of gravity, m/s^2, ρ – the average density of water in the pump, kg/m^3; d_h and d_s – diameters respectively of the head and suction nozzles of the pump, m; r – the model identification parameter; the index "n" indicates the value of the parameter at the nominal rotor speed.

To determine the characteristics of the pump at different rotor speeds (Q_p, η_p), we perform calculations in tabular (in table 1) and in a functional, graphical form (figure 1), specifying the parameters of fluid flow and calculating efficiency indicators (η_p) of the pump at various numbersof revolutions, can be calculatedthe operational characteristics and build a graph.

Table 1 – Indicators *of variation* of the pump efficiency

Fluid flow Q m^3/h	η_p pump efficiency at various numbers of revolutions			
	n=1000 r/min	n=800 r/min	n=600 r/min	n=400 r/min
0	0	0	0	0
0,2	0,5	0,5	0,5	0,5
0,4	0,8	0,78	0,7	0,65
0,6	0,9	0,86	0,75	0,7
0,8	0,9	0,8	0,7	0,65
1	0,82	0,65	0,56	0,45

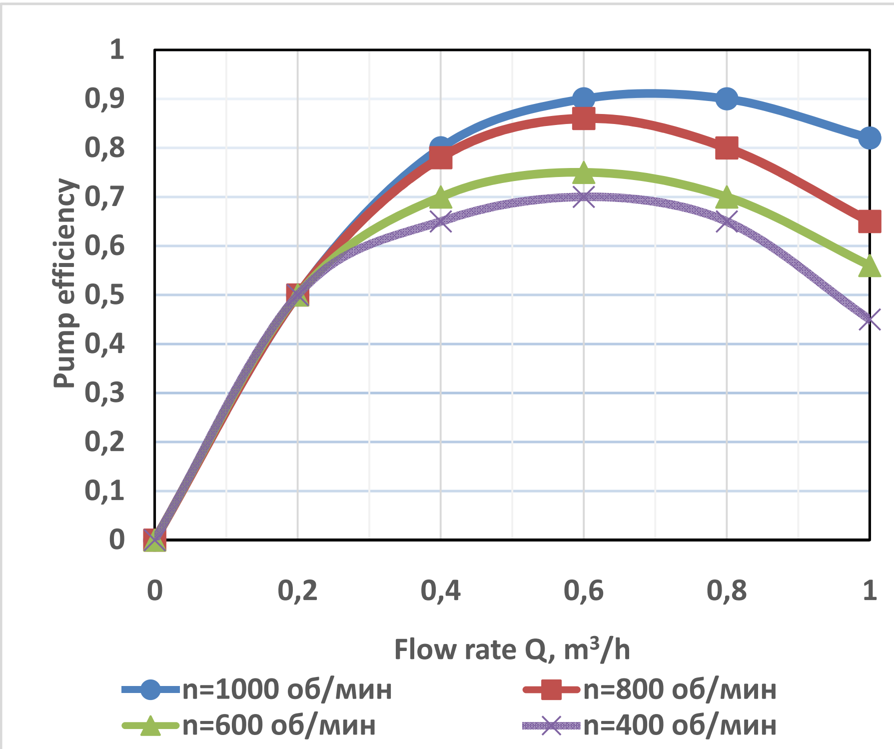

Figure 1 – The graph of changing the pump efficiency (η_p) depending on the increase in flow rate (Q_p) when the numbers of revolutions equals (n) (from 400 to 1000) r/min

Figure 1 presents the results of calculations using formulas (4, 7, 8) for the above example with a pump. In this case, the deviation of the calculated values (η_p) of the pump efficiency from various flow rates and numbers of revolutions was 5.8%.

To clarify the characteristics of the pump at different rotor speeds (N_p, Q_p), we will perform calculations in tabular (in table 2) and in a functional, graphical form (figure 2), Specifying the parameters of fluid flow and calculating indicators (N_p) of the pump power consumption at various number of revolutions, can be calculated the operational characteristics and build a graph.

Table 2 – Indicators *of variation*of the power consumption

Fluid flow Q m^3/h	N_p(kW) power consumption of the pump at various numbers of revolutions			
	n=1000 r/min	n=800 r/min	n=600 r/min	n=400 r/min
0	0,7	0,56	0,4	0,3
0,2	0,85	0,72	0,6	0,56
0,4	0,98	0,82	0,7	0,65
0,6	1,1	0,96	0,8	0,75
0,8	1,2	1,1	0,92	0,86
1	1,4	1,2	1,1	0,98

Figure 2 presents the results of calculations using formulas (4,7,8) for the above example with a pump. In this case, the deviation of the calculated values (N_p) of the power consumption from various flow rates and numbers of revolutions was 3,8%.

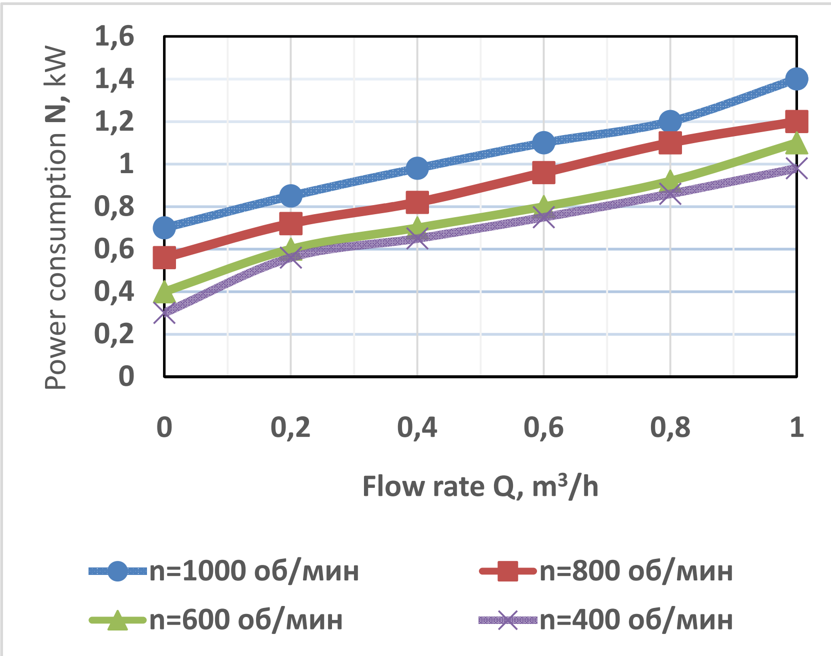

Figure 2 – The graph of changing of the power consumption (N_p) depending on the increase in flow rate (Q_p) when the numbers of revolutions equals (n) (from 400 to 1000) r/min

To clarify the characteristics of the pump at different rotor speeds (H_p, Q_p), we will perform calculations in tabular (in table 3) and in a functional, graphical form (figure 3), specifying the parameters of fluid flow and calculating indicators (H_p) fluid pressure generated by pump at various number of revolutions, can be calculated the operational characteristics and build a graph.

Figure 3 presents the results of calculations using formulas (4,7,8) for the above example with a pump. In this case, the deviation of the calculated values of the fluid pressure generated by pump at various number of revolutions was 4,8%.

Table 3 – Indicators *of variation* of the fluid pressure

Fluid flow Q m^3/h	H_p (mm.w.c.) fluid pressureat various numbers of revolutions			
	n=1000 r/min	n=800 r/min	n=600 r/min	n=400 r/min
0	120	110	98	86
0,2	117	106	92	80
0,4	112	98	86	78
0,6	105	94	82	72
0,8	92	82	72	65
1	78	65	56	48

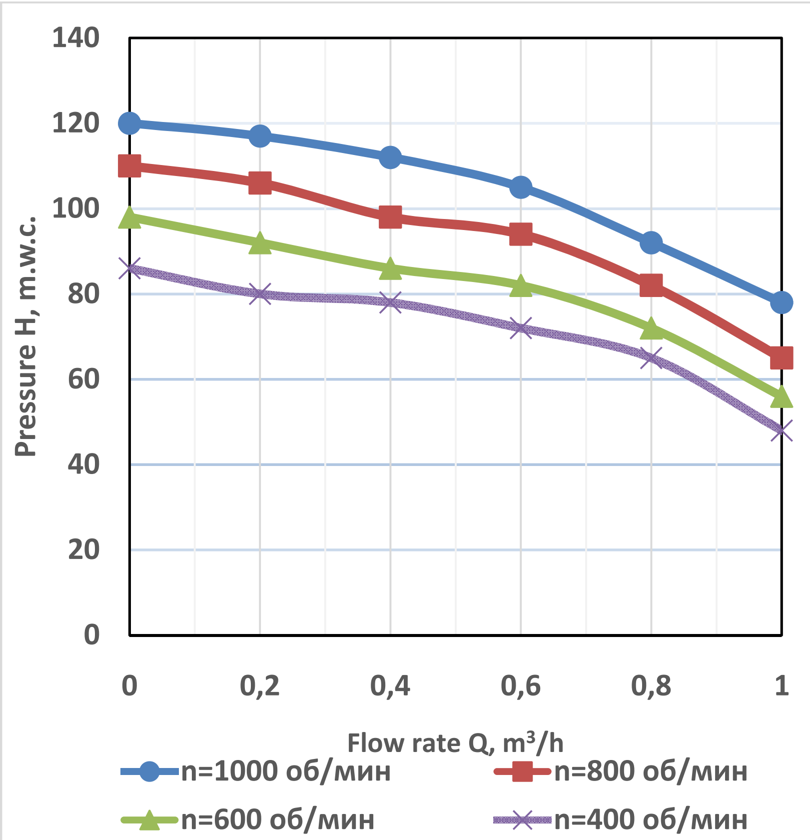

Figure 3 – The graph of changing of the fluid pressure (H_p) generated by pump depending on the increase in flow rate (Q_p) when the numbers of revolutions equals (n) (from 400 to 1000) r/min

We will calculate the liquid flow rate Q, m^3/h (capacity) and power consumption N, kW, depending on changes in the blade width of the pump disk, the results are shown (in table 4) and in a functional, graphical form (figure 4). Given parameters of the blade width (b_2), we calculate the pump indicators - (Q) capacity and (N) power consumption of the pump for different widths of the blades of the pump disk and construct a graph of f (b2) = (Q, N).

As a result of the calculations, a *dependency graph* of the capacity (Q) and pump power consumption (N) from changes of blades width of the pump disk was drawn which is shown in figure 4.

Table 4 – Indicators of variation of the fluid flow and pump power consumption

Bladewidthof the pump disk b_2	Fluid flow Q m^3/h	N_p(kW) power consumption of the pump
0,01	0,2	0,8
0,015	0,4	0,68
0,02	0,6	0,62
0,025	0,8	0,72
0,03	1	1,2
0,035	1,2	1,8

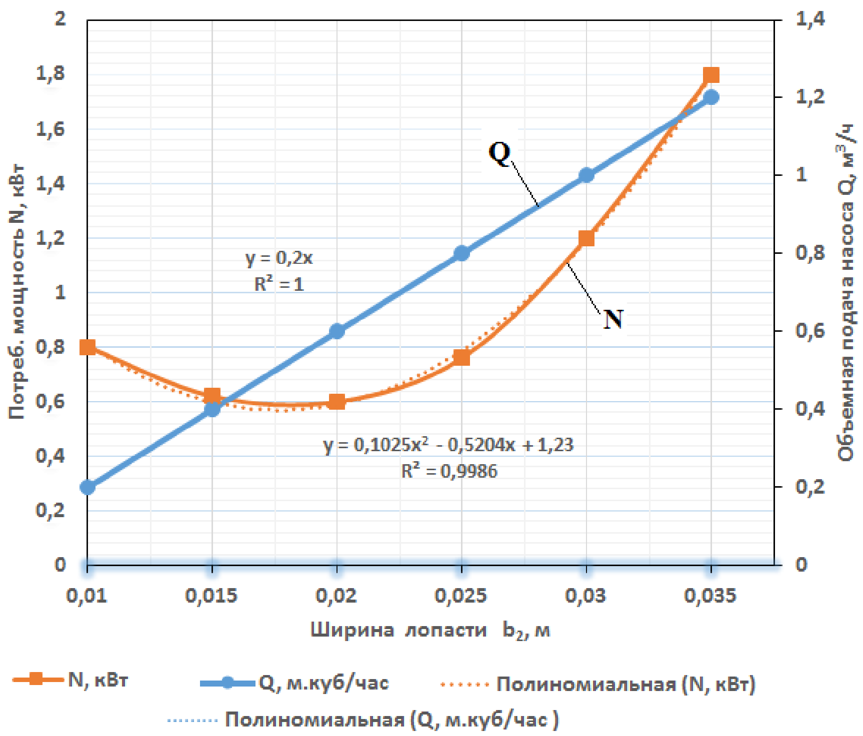

Figure 4 – A dependency graph of the capacity (Q) and pump power consumption (N) from changes of blades width of the pump disk

After approximation of the results of analytical studies, an empirical dependence of the power consumption on the change in the width of the blades of the pump disk is obtained.

$$\mathbf{N = 0{,}10\ b^2 - 0{,}52\ b + 1{,}23} \text{ herewith } \mathbf{(R^2 = 0{,}9986)} \quad (10)$$

where 0,10 – initial power consumption of the pump at minimum rotation, kW.

and pump capacity by the equation:

$$\mathbf{Q = 0{,}2\ b_3} \text{herewith } \mathbf{(R^2 = 1)} \quad (11)$$

where 0,2 - initial pump capacity at minimum rotation, m^3/h.

It can be seen from the graph that the previously accepted width of the blades is in the range of 0.015 m to 0.025 m, which is almost two times smaller than the width of the blades of existing pumps, which is equal to 0.052 m. At the same time, the maximum value of capacity Q = 0.8 m^3/h, and with further changes awidth of the blades towards an increase, the pump power consumption and itscapacity increases abruptly.

Results and discussion. As a result of the calculations, the proposed limit of the width of the pump blades was set from 0.015 m to 0.025 m, with an average rotation, the maximum capacity value was reached Q = 0.8 m^3/h, which ensures optimum pump capacity with minimum power consumption. In the results of analytical studies, an empirical dependence of the power consumption on the change in the width of the blades of the pump disk is obtained.

In view of the above, it can be concluded that the proposed limit of the width of the pump blades would not affect negatively for the effective performance indicators of the pump, while ensuring optimal capacity of the pump with minimal power consumption and efficiency of the reactor in general.

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РАСЧЁТ РАБОЧЕГО КОЛЕСА НАСОСА

Аннотация. В статьи приведены результаты теоретических исследований параметров рабочего колеса центробежного насоса, установлены пределы ширины лопастей, которые не окажет отрицательного влияния на эффективные показатели работы насоса и обеспечивается оптимальные производительности с минимальным потреблением мощности, а также работоспособности ускорителя реактора

Ключевые слова: параметры насоса, рабочее колесо, процесс ускорение реакцию петрификации, реактор производства биодизеля, число лопастей.

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СОРҒЫНЫҢ ЖҰМЫС ДӨҢГЕЛЕГІН ЕСЕПТЕУ

Аннотация. Мақалада орталықтан тепкіш сорғының жұмыс дөңгелегі параметрлерінің теориялық зерттеулерінің нәтижелері келтірілген, сорғы жұмысының тиімді көрсеткіштеріне теріс әсер етпейтін қалақтар енінің шектері орнатылған және қуатты барынша аз тұтынумен оңтайлы өнімділік, сондай-ақ реактор үдеткішінің жұмысқа қабілеттілігі қамтамасыз етіледі.

Түйін сөздер: сорғы параметрлері, жұмыс дөңгелегі, петрификация реакциясын жеделдету процесі, биодизель өндірісінің реакторы, қалақтар саны.

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