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ANALYSIS OF RESULTS FROM BALANCE INVESTIGATIONS OF CENTRIFUGAL PUMPS 6E20 AND 6E32 WITH AIR-WATER TWO-PHASE FLOW PERFORMANCE

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ABSTRACT:
In this paper are given the results from balance investigations of centrifugal pumps 6E20 and 6E32 with air-water two-phase flow performance. On this basis a comparative analysis is made of the behavior of the two pumps in these performance conditions.
KEYWORDS:
centrifugal pumps, two-phase flow, void fraction, specific speed

1. INTRODUCTION

The most used pumps for transport of air-water two phase flow performance are centrifugal pumps. The research in this area is trying to predict how they will react in these conditions of work and also how to fix/settle the reasons for their indices going down when the volumetric of gas in the two-phase flow performance is going higher .An important factor influencing the pump's work with a two-phase fluid of air and water is the geometry of their impeller. The research of many authors in this area are concentrated on pumps of same type In the literature we do not find any information about the influence of the specific frequency of rotation - on the indices of the pump when it works in these conditions.

In this paper are offered some experimental results from the balance of the energy one-stage centrifugal pumps $6E20$ with $n_S = 85\,\min^{-1}$ and $6E32$ with $n_S = 60\,\min^{-1}$ in work with air-water two-phase flow performance. Using the experimental results, a comparative analysis of the performance of these two machines at their indicated conditions of work has been done.

2. MAIN BODY

The experimental research in pump's working with air-water two phase flow performance are coordinated with the method and shown in $[1]$.The results of these investigations are shown in a dimensionless aspect in the mode of dimensionless coefficients .The water's flow-rate through the pump is presented by the coefficient of the flow-rate φ which is defined by the formula:

$$\phi = \frac{Q_L}{\pi \cdot D_2 \cdot b_2 \cdot u_2} \tag{1}$$

where Q_L is the water's flow-rate, D_2, b_2 and u_2 are the diameters of the web-wheel, the width of the channels and the transmission speed of its exit.

The pressure of the pumps is presented by its coefficient Ψ, which is found with the formula:

$$\Psi_{tp} = \frac{g \cdot H_{tp}}{u_2^2} \tag{2}$$

In the same way the hydraulic loses in the impeller Δh_{imp} and in the volute Δh_{volute} are made dimensionless too.

In figures 1-4 are shown pressure characteristics, coefficient of efficiency and hydraulic loses in the impeller and also the hydraulic loses in the volute of centrifugal pump 6E20, with different values of air at the entrance-α_1, defined in [1, 3, 4].From a first look we can see that when α_1 is going up the hydraulic loses in the impeller are going up too-very intensively and those in the volute stay almost the same. We have similar characteristics for pump 6E32.They are shown in a way which in a way which makes it possible to compare the two pumps work with air-water two phase performance.

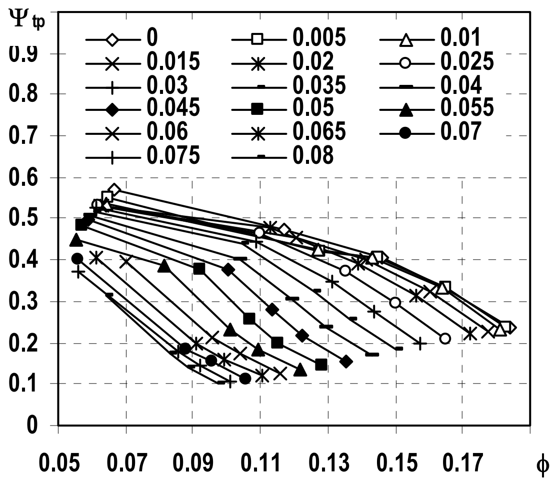

FIGURE 1. Pressure characteristics of a pump 6E20 with different value of air-content

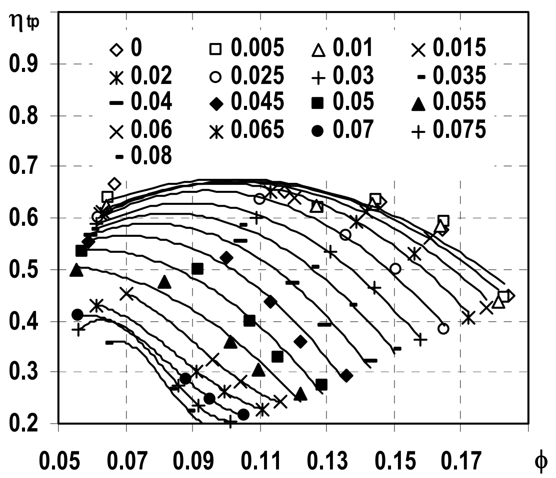

FIGURE 2. Coefficient of efficiency of a pump 6E32 with different value of air-content

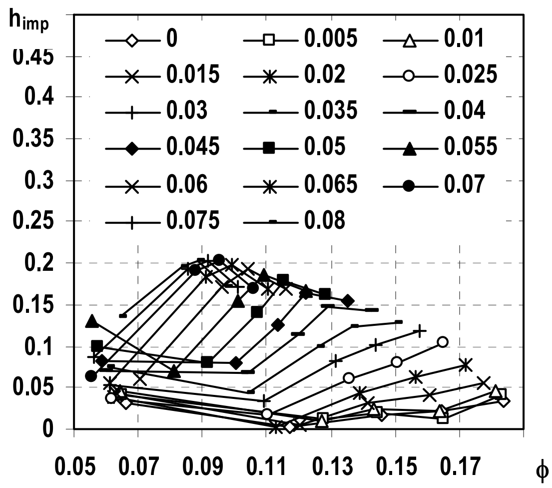

FIGURE 3. Hydraulic loses in the impeller of a pump 6E32 with different value of air-content

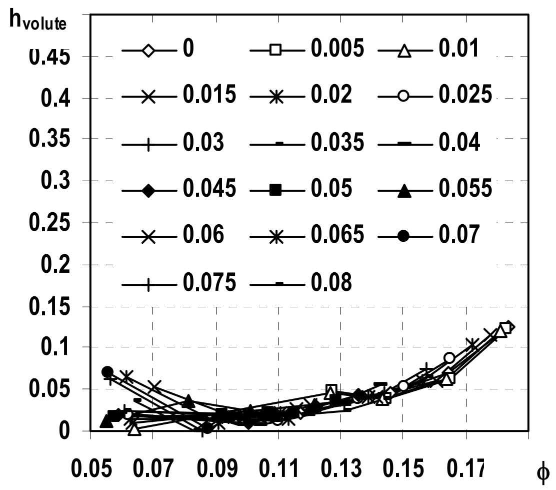

FIGURE 4. Hydraulic loses in the volute of a pump 6E32 with different value of air-content

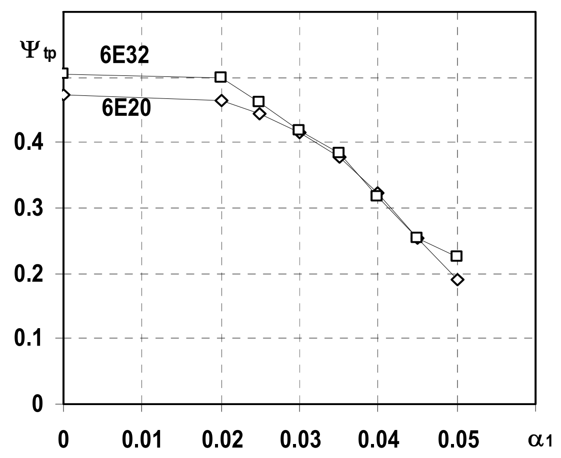

FIGURE 5. The pressure dependings of pumps 6E20 and 6E32 by the air-content when the flow-rate is nominal

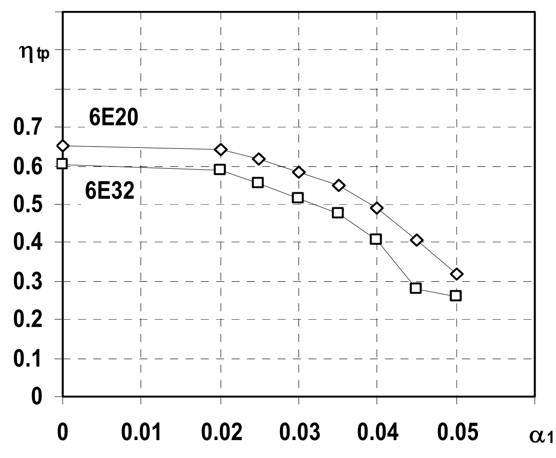

FIGURE 6. The coefficient of efficiency dependings of pumps 6E20 and 6E32 by the air-content when the flow-rate is nominal

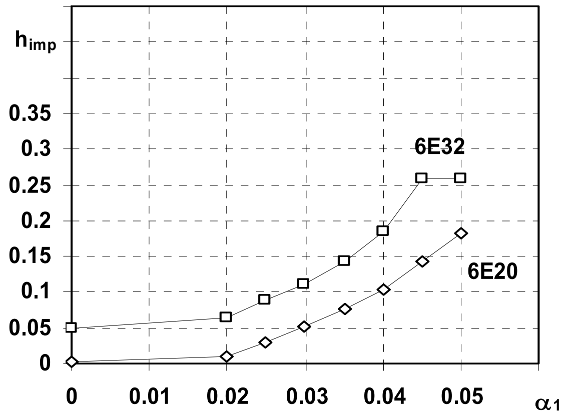

FIGURE 7. Hydraulic loses dependings in impeller of pumps 6E20 and 6E32 by the air-content when the flow-rate is nominal

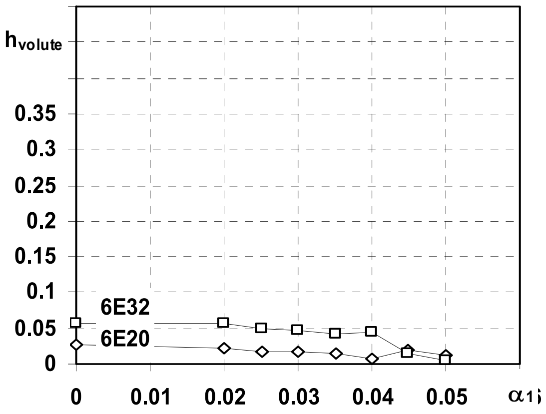

FIGURE 8. Hydraulic loses dependings in volute of pumps 6E20 and 6E32 by the air-content when the flow-rate is nominal

In evaluating the influence of the insoluble air on the dials of the pumps the following relationships were found: $\Psi_{tp} = f(\alpha_1)$, $h_{imp} = f(\alpha_1)$, $h_{volute} = f(\alpha_1)$ and $\eta = f(\alpha_1)$, with a constant water's pressure - ϕ_n. They are shown by the figures 5 – 8. With the help of the last analysis it is possible to compare the dials of the two pumps in work processes with different mixtures of air and water. In figures 3 and 4 we can see that a main part of the hydraulic loses in the pump, when air content is $\alpha_1 > 2\%$, is coming from the hydraulic lose in the impeller. In figure 7 are shown the results of $h_{imp} = f(\alpha_1)$ for the two investigated pumps. We can see that the pump with lower specific frequency of rotation - 6E32 is working with a higher hydraulic lose. That means also a smaller coefficient of efficiency for the pump (Fig.6). Hydraulic loses in the volutes of the two pumps are almost the same low value and they don't change their values when the air-content is going higher.

The increase of pressure loses in the impeller probably is coming from the air accumulating near the incoming edge of the blades, which is shown in paper 3-4. This accumulation is made worse by air movement across the blade - Kosyna's investigations [2]. In this reference [2] are shown some results for the distribution of the pressure on the front and back part of the blade for different values of air-content. As a result of this research in [2] – when the value of air-content in the two-phase fluid is 5%, we see that the blade is working with 60% of its surface.

The worse conduct of the pump with lower n_S is presented in [6]. In this paper the authors are making a comparison between the work of centrifugal and diagonal pumps in working with a mixture of diesel - CO_2. They say that the better dials of the pump with higher n_S, are coming from the better conditions at the entrance of the impeller (its bigger diameter and width of the channels between the blades). The critical value of air-content α_{cr} for the two pumps is around 5%. For values which are over than α_{cr} the pump stops its normal way of working. In figure 6 we see that the coefficient of efficiency is very much going down when it reaches to the critical value of air content. In the results shown in [4] it is clear that for this value of α_1, it starts connecting air/gas bubbles and accumulating of air near the entrance of impeller. When $\alpha_1 = \alpha_{cr}$ it is starting a process of cavitation which makes the pump's work efficiency go down.

Some depth analysis of the influence of some main factors (such as air-content, air's solubility and specific frequency of rotation) over the dials of centrifugal pumps in working with air-water mixture, could be done after an experimental investigation/research of a pump with $n_S = 120\ \text{min}^{-1}$ and use of a physico-mathematic model to predict the conduct of centrifugal pumps in work with two phase mixtures.

3. CONCLUSION

Based on the results from the balance research of one-stage centrifugal pumps 6E20 with $n_s = 85\ \text{min}^{-1}$ and 6E32 with $n_s = 60\ \text{min}^{-1}$ in a work with a two phase mixture of air and water, we can reach these conclusions:

The pump with the higher specific frequency of rotation $n_s = 85\ \min^{-1}$, 6E20 has better dials (higher coefficient of efficiency and lower values of hydraulic loses in the impeller) in work with air-water mixture, which results from the better conditions at the entrance to the impeller (its bigger diameter and its wider channels between the blades). A similar result is gotten from the author of [6].

The critical value of air content α_{cr}, when the work of both pumps is going down is around 5%. Similar values are found from the authors of references [3, 4].

A main part of the hydraulic loses in a centrifugal pump in working with a two phase mixture of air and water is coming from loses in the impeller when the loses in the volute are staying almost constant. Some of the authors [references 2,3,4] received the same results for the similar reasons.

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