

# REDUCTION OF VIBROACOUSTIC LOADS IN AVIATION COMBINED PUMPS

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## Abstract

*Considerable number of previous research has shown that aviation combined pumps, which consist of a screw and centrifugal wheels and a gear stage, are the most loaded units of the gas turbine engines. Thus a fuel pump is the key component that limits the reliability and resource of the fuel system and, as a result, of the whole engine.*

*The main problem of this study was reducing so called vibroacoustics loading of combined pump by means of new pump entrance redesign measures aimed at reducing pump loading. This loading included both pressure pulsation and vibration loading. It had to be done without any changes of operating regimes of aviation fuel system. This paper describes produced pump enter redesigns and experimental results of their efficiency. In addition this paper describes the developed test bench and measurement techniques. The results illustrate how the proposed redesigns reduce the flow unsteadiness in combined pump at its different operating regimes. We can predict that the suggested measures will enhance the reliability and endurance of aviation fuel pumps.*

## 1 Introduction

The statistics of aircraft failures shows fuel pumps mounted at engines are key components limiting resource and reliability of the whole engine. Why is it so? Papers [1 - 5] showed the elements of an aircraft fuel pumping system and particular gear and screw centrifugal pumps are under a significant vibration load. It's well known that aircraft engine fuel system

contains a large number of functionally related hydro-mechanical components, each of which can be a source of vibration, pressure, flow and noise.

The aviation fuel pump system has as a rule a low pressure pump and a high pressure pump. Considered in this study fuel system has also two these pumps: a low pressure screw centrifugal pump which is combined with high pressure gear pump in one unit to get lighter and more reliable pump (Fig. 1).

Let's depict some causes of loading of pumps considered in present study. One of the most significant sources of increased loads for this type of pumps is free gas which can enter the pump [6]. There are two causes of free gas entry. The first is failure to operate of cut-off valves located in aircraft's tanks. The second is absolutely evident. Its well-known all fuels are able to dissolve a significant amount of gas. Therefore vortex disturbances occurring in the supply pipeline contribute to the bubble formation. This leads to breakdown of the dynamic equilibrium in the fuel-air system, accompanied by the liquid-vapor transition. As a result cavitation properties decrease significantly [7, 8]. Paper [9] presents an experimental dependence to estimate the cavitation performance change on account of the free air. This work shows that even a small amount of free gas in fluid significantly decreases pump cavitation performance.

Appeared gas bubbles collapse in regions with higher pressure. Bubble collapsing in its turn leads to erosion and as a result to highly intensive stresses on the pump's elements [10]. As shown from [1, 2], cavitation erosion is a major factor that leads to increased wear of the angular contact sleeve bearing of combined pumps. Due to its increased wear the touching occurs between the blades of impeller and

volute. This leads to pump failure. Paper [11] experimentally and theoretically studies the effects of bubble collapse within blades of an inducer.

Papers [10, 12] show backflow vortex presents almost at all pump operating regimes. They are additional source of flow instability at pump entrance and as a result increased loading of pump's elements.

Thus, the aviation fuel systems with combined pumps are very complex in which there is a large number of forced oscillation sources. In spite of a lot of works discussing working processes of different types of pumps there are scarcity of works studying free gas influence, cavitation influence and flow instability in combined pumps. As a result there is a lack of works describing measures helped for reducing combined pump loading.

## 2 Combined pump

The aviation combined pump (Fig. 1) considered in this study is a commercial type. This pump consists of two stages. The first is screw centrifugal stage with an open type impeller with 11 straight blades and double-lead screw. It has the single unvaned volute casing. The shape of the single volute casing is designed according to the theory of a constant average velocity for all sections of the volute. The second stage is gear stage. Gear wheels as well as centrifugal wheel also have 11 teeth. The working fluid is kerosene.

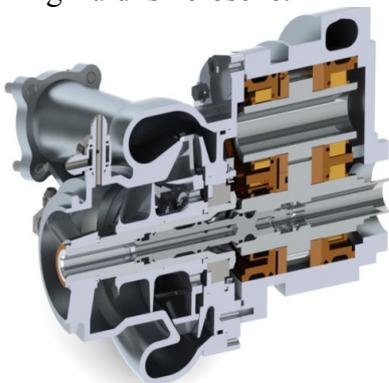


Fig. 1. Combined pump

## 3 Test bench

We developed a test bench for experimental research of proposed pump entrance redesigns efficiency. Its hydraulic schematic is shown in Fig. 2. Hydraulic and connections schemes of the developed test bench were as close as possible to the flight conditions on the aircraft.

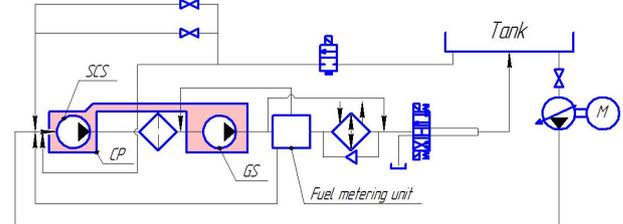


Fig 2: The hydraulic schematic of test bench

Eight strain sensors were mounted on angular contacting sleeve bearing housing to measure stresses (Fig. 3). Note that bearing housing is the flange which connects the CP's stages.

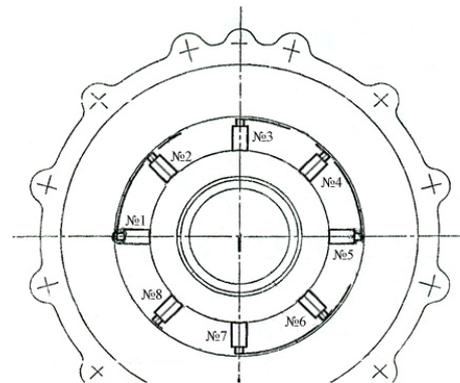


Fig. 3. Installation scheme of strain sensors

## 4 Operating modes

We carried out experiments on 24 different operating regimes of fuel system to estimate the proposed entrance redesigns efficiency. Regimes 1...6 correspond with the regimes of engine starting. Herewith rotational frequency of pump's rotor varied from 1400 to 5300 rev/min and the mass flow varied from 4000 to 10500 kg/h. Regimes 7...13 correspond with cruise rating of engine. Rotational speed and mass flow varied from 5200 to 6500 rev/min and from 11300 to 17500 kg/h accordingly. Regimes 14...17 correspond with engine ignition in-flight. Rotational speed and mass flow varied from 6600 to 7200 rev/min and from 18500 to 20000 kg/h accordingly.

Regimes 18...22 correspond to forcing regimes. Herewith rotational speed and mass flow varied from 7300 to 10000 rev/min and from 15400 to 20300 kg/h accordingly.

5 Pump entrance redesigns

The analysis of [2] and [3] allowed us to create the scheme of vibration and hydrodynamic interaction of combined pump stages (Fig. 4). Here  $C_{Ai}(t)$  is air concentration,  $P_i(t)$  – mixture pressure,  $Q_i(t)$  – volume flow rate,  $T$  – temperature,  $V_x, V_y, V_z$  – vibration acceleration,  $N$ - rotary speed.

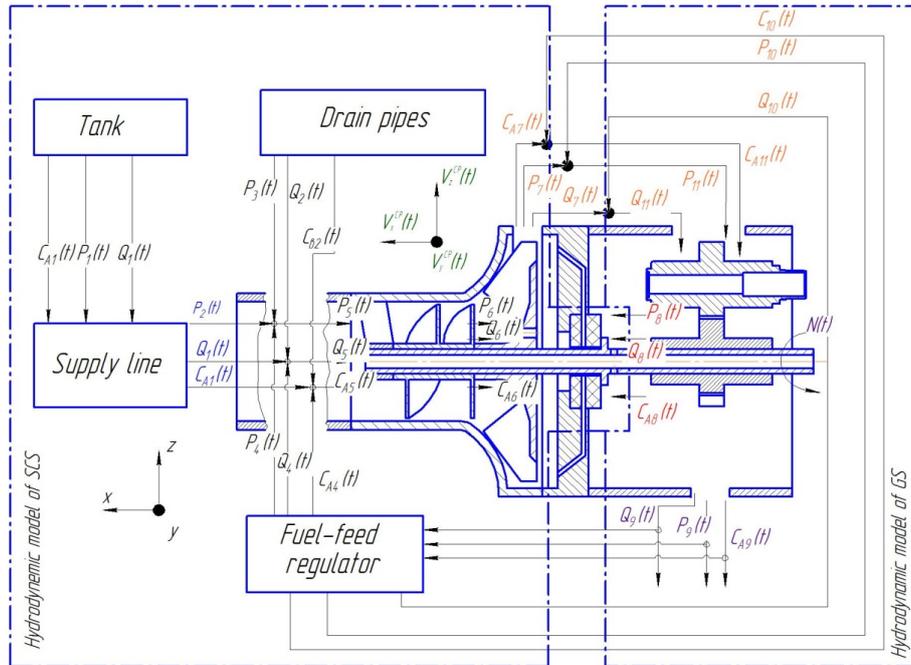


Fig 4: Scheme of vibration and hydrodynamic interaction of CP's stages

This scheme illustrates the main loading sources of pump's components. We proposed some pump entrance redesigns by means of this scheme:

- the first is transfer the drain pipelines to supply line (Fig. 5);

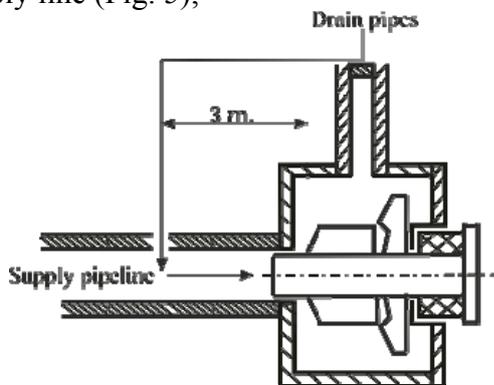


Fig. 5. Transfer the drain pipelines to supply line

- the second is screen installation at the screw inlet (Fig. 6) with gap forming between the back face of the screen and the vase case.

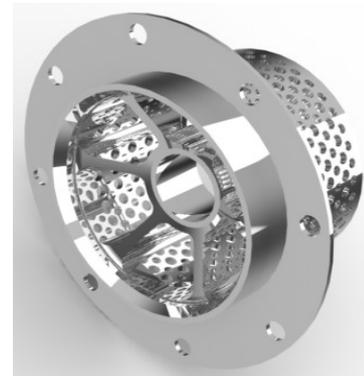


Fig. 6. Redesign with the screen

Screen is perforated cylindrical component with a flange at its inlet. It has 6 guide vanes instead of 3 guide vanes in the basic design. Additionally screen has two rows of holes with diameter 4 and 5 mm. Total amount of holes is 60. Their total resistance is equivalent to the formed gap total resistance.

The aim of this screen is divide main flow into series of smaller flows by means of its 6

segments. Thereby it stabilizes flow at pump entrance;

- the third is attaching the ring in addition to the screen for closing the gap (Fig. 7).



Fig. 7. Redesign with screen and ring

Closing the gap leads to flow drain pipes liquid only through the screen's holes;

- the fourth is spinner installing additionally to the screen and ring to prevent high frequency leakages from the gear stage and stabilizing flow structure (Fig. 8).



Fig. 8. Redesign with a screen, ring and spinner

The main purpose of proposed redesigns is considerable decreasing of whole vibroacoustics loading.

## 6 Results and discussion

We estimated the stress conditions of considered pump by means of vibration, pressure and stress measurements. We measured pressure in 6 points and vibration acceleration in 4 points to have full information about the system condition at any time (Fig. 9). We used hardware LMS SCADAS Mobile and software LMS Test. Xpress for data recording and analysis.

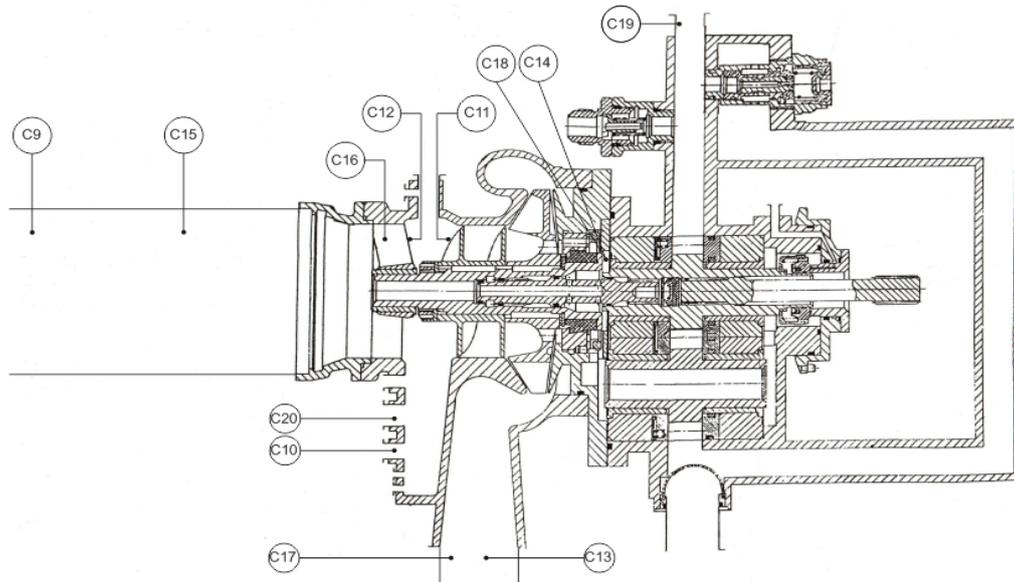


Fig. 9. Point for loading analysis

Pressure pulsation and vibration acceleration were estimated with help of RMS amplitude pressure and vibration acceleration fluctuation accordingly. RMS amplitude calculates by means of equation:

$$RMS = \sqrt{\frac{1}{n} \sum_{j=0}^{n-1} (y_j)^2}, \quad (1)$$

$n$  – number of time steps

$y_j$  – value of measured signal at defined value of time step.

### 6.1 Static pressure distribution

Pictures 10 and 11 show dependence measured static pressure at inlet and delivery pipelines (points C15 and C17 on Fig. 9) on system operating regime for different pump configuration. These pictures illustrate proposed redesigns lead to static pressure increasing at pump inlet (at lower regimes pressure buildup is 5-10%, at higher regimes – up to 96%). At the same time we got both negligible drop of static pressure at the pump outlet and drop delivery head (about 0.05 – 0.2 MPa) relative to basic pump design. It is largely due to pump entrance shadow caused enhanced number of guide vanes.

Picture 12 (point C14 on Fig. 9) shows static pressure change at the pot which situate between two pump stages. Leakages from gear stage pours into this pot. At basic design this pot connected with entrance of screw centrifugal stage by means of shaft internal diameter and by means of series of holes in bearing case it connected with pot situated after centrifugal wheel which in its turn connected with screw outlet by means of holes at centrifugal wheel disk.

At final pump design the inlet to shaft internal diameter on the part of pot was closed. Pressure in this pot circumstantial characterizes condition for bearing lubrication as it determines by pressure difference between this pot and outlet of centrifugal wheel. Maximum drop of pressure in 1.2 time gives the final pump design (screen, spinner and without gap) at regimes when mass flow exceeds.

### 6.2 Pressure pulsation

We produced and installed analyzer probe for estimate flow direction at pump entrance. Analyzer probe represents 2 transducers of static pressure. One of them installed downstream another one – upstream (points C11 and C12 on Fig. 9 accordingly). The pressure difference between these two transducers can show presence of back flow at pump entrance. Analyzer probe was installed near the pump case wall and in-between guide two vanes (Fig. 13 – for basic design).

At basic pump design we obtained that pressure downstream more often than not is higher than pressure upstream. Its confirm backflow at pump entrance in case its basic design. Fig. 14 shows measurements of pressure pulsations at the inlet of screw centrifugal in different redesigns.

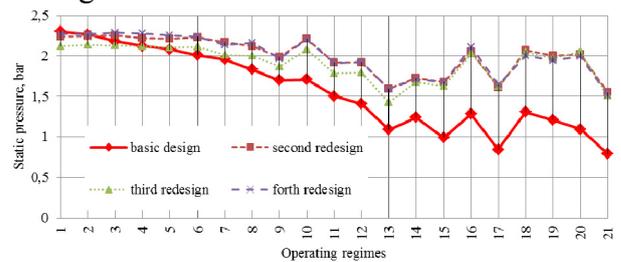


Fig 10. Static pressure change at inlet manifold

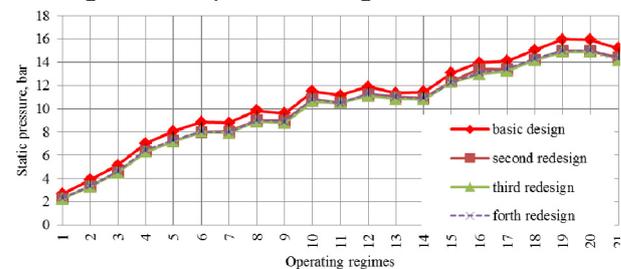


Fig 11. Static pressure change at screw centrifugal pump outlet

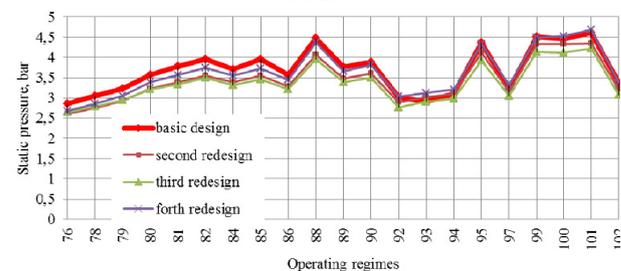


Fig 12. Static pressure change at the pot situated between pump's stages

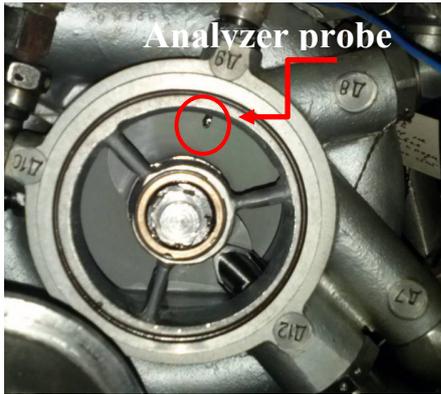


Fig 13. Photograph of analyzer probe installing

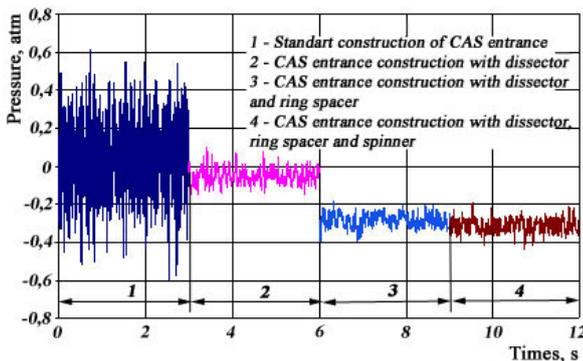


Fig 14: Pressure pulsation at pump entrance for different pump designs

The figure shows that the design modification results in reduction of low-frequency oscillation of backflow vortex before screw. As a result it leads to flow stabilizing at pump entrance. We archive decreasing of pressure ripple amplitude measured downstream from 0.09 MPa to almost 0 MPa and upstream from 0.045 MPa to 0.12 MPa. Therefore the final design removes backflow at all investigated regimes.

We obtained that transfer the drain pipelines to supply line do not prevent backflow.

Therefore the most efficient design is the design with screen, spinner and without gap. Herewith mounting of drain pipes is the same like in basic design.

Picture 15 shows pressure pulsation change in inlet manifold of screw centrifugal stage (point C9). It illustrates pump redesign leading to

decreasing of pulsation amplitude up to 45% relative basic pump design

Picture 16 shows change of pressure oscillation at the pot between pump stages (point C18). As we showed earlier pressure oscillations occurred in the pump entrance, in screw and centrifugal wheels, in pot behind centrifugal wheel caused additional pump loading. Our results show pressure oscillations are almost constant and reach 0.01...0.06 MPa when static pressure is 0.26...0.35 MPa at lower pump regimes. At higher regimes they reach 0.07...0.11 MPa and 0.36...0.45 MPa accordingly. Herewith pressure oscillations in pot under consideration mostly depend on mass flow rate.

Redesign of pump entrance also leads to negligible changes of pressure oscillations at outlet of screw centrifugal stage (Fig. 17. Point C13). These oscillations mostly depend on operating regimes.

Pump entrance redesigns also lead to increase of pressure fluctuation at drain pipelines up to 0.1...0.15 MPa at all fuel system operating regimes (Fig 18).

We also obtained that proposed redesigns also effect on pressure oscillations at outlet of gear stage (Fig. 19). At lower operating regimes this influence is not so significant as at higher regimes when we get pressure pulsation drop in 6 times.

### 6.3 Stress changes

Fig. 20 illustrates change of average stress which we measured at bearing case by means of 8 transducers. It shows that pump entrance redesign leads to significant reduction of average stress at bearing surface. We obtained that maximum axial force acting on bearing in basic design is 120 kilogram-force.

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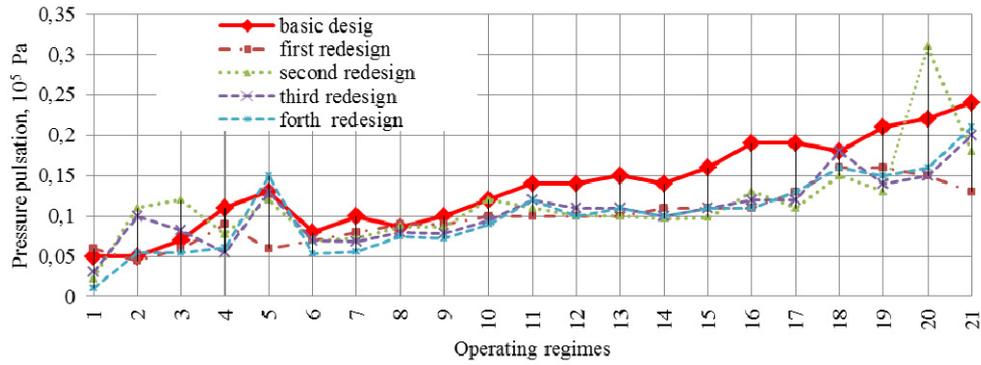


Fig 15. Pressure pulsation at point C9

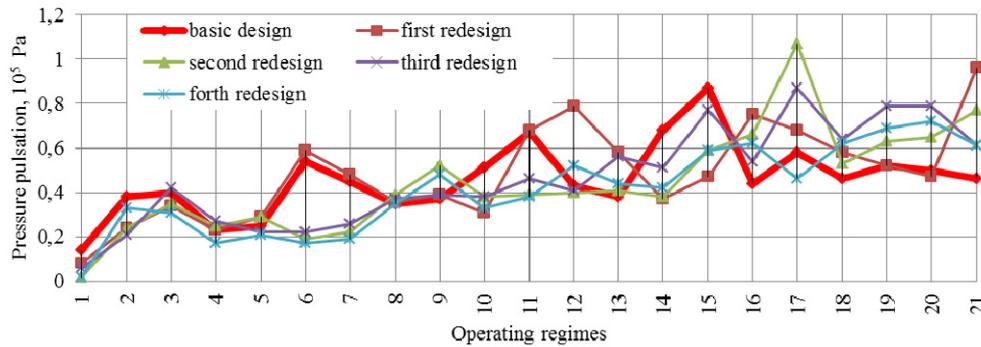


Fig 16. Pressure pulsation at point C18

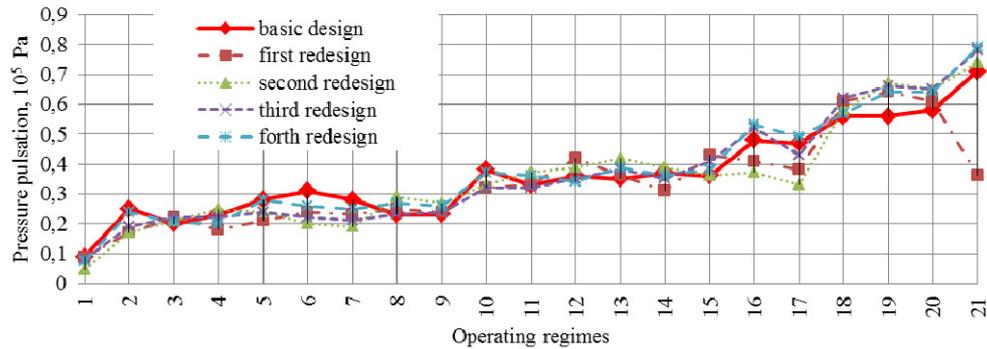


Fig 17. Pressure pulsation at point C13

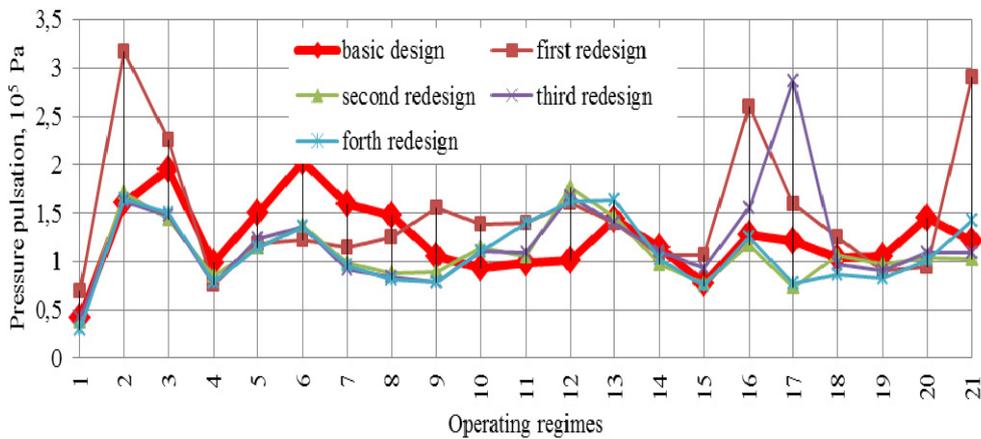


Fig 18. Pressure pulsation at point C10

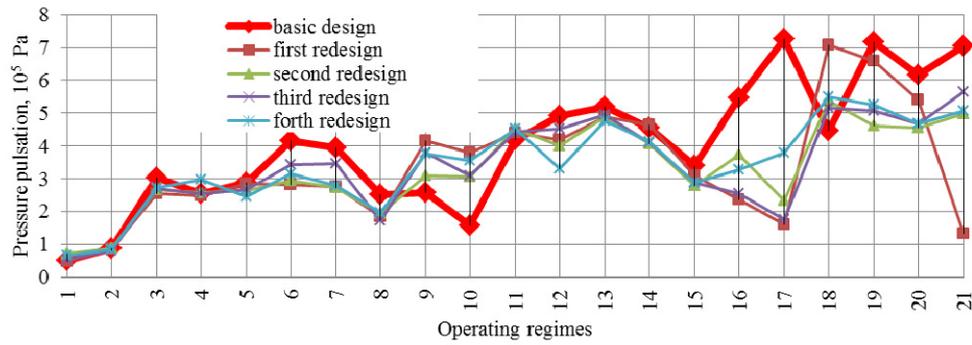


Fig. 19. Pressure pulsation at point C19

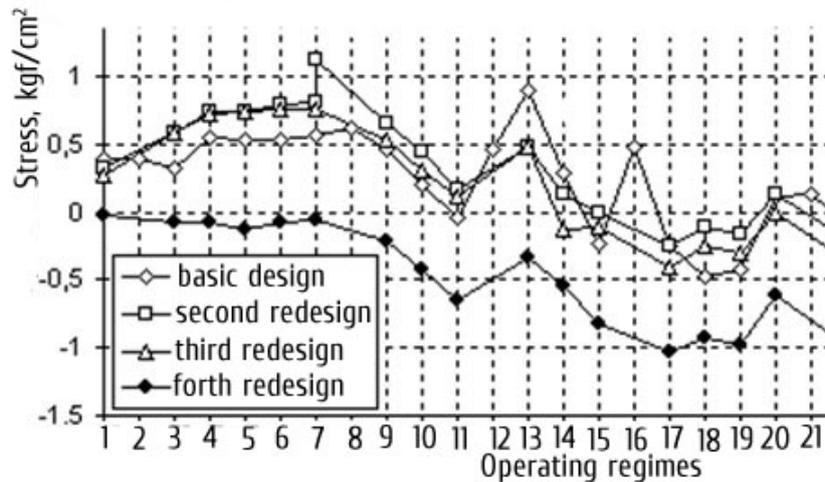


Fig. 20. Average stress at bearing surface

#### 6.4 Vibration acceleration changes

Figures 4.24 – 4.27 show change of vibration acceleration in investigated points. We measured them on pump flange in three directions (on XOY, YOZ and XOZ planes – see Fig. 5). We also measured vibration acceleration on flange that connected two pump stages in one direction – on XOY plane. Pump entrance redesigns reduce vibration acceleration by 30...40% relative to basic design on XOY plane at 76...96 regimes. At higher regimes (97-102) we vice versa obtained increasing of vibration acceleration by 20...30%. This result

can be explained by nonoptimality of flow around centrifugal pump.

We archived decreasing of vibration acceleration by 5...35% relative to basic design on YOZ plane in cases with pump entrance redesigns.

Our results also show that all pump entrance compositions provide reducing of vibration acceleration on XOZ plane in average by 6%

Analysis of pump vibration state shows that increasing of operating regimes leads to rising of vibration load.

In such a manner the best composition is the composition with screen, spinner and without gap.

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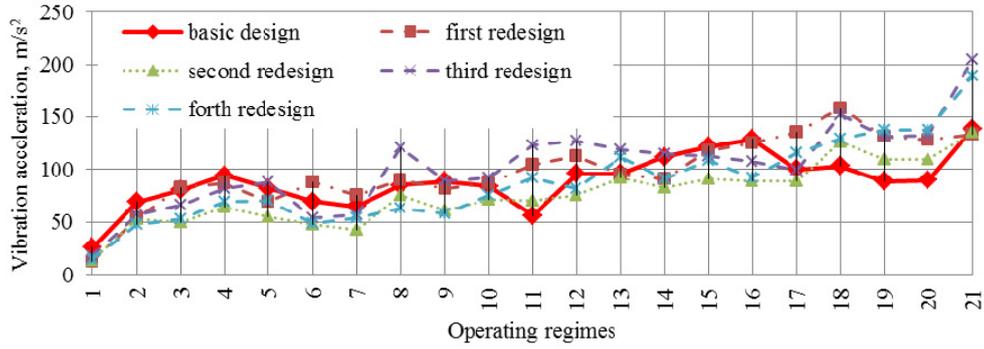


Fig. 21. Vibration acceleration at screw centrifugal pump inlet. XOY plane. Direction is parallel to pump's axis

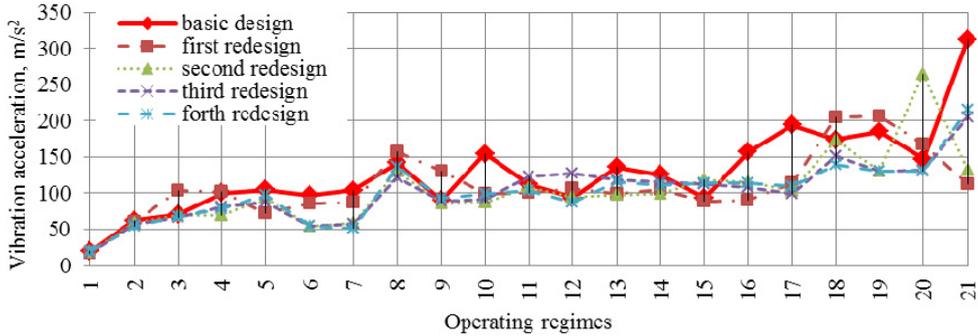


Fig. 22. Vibration acceleration at screw centrifugal pump inlet. XOY plane. Direction is perpendicular to pump's axis

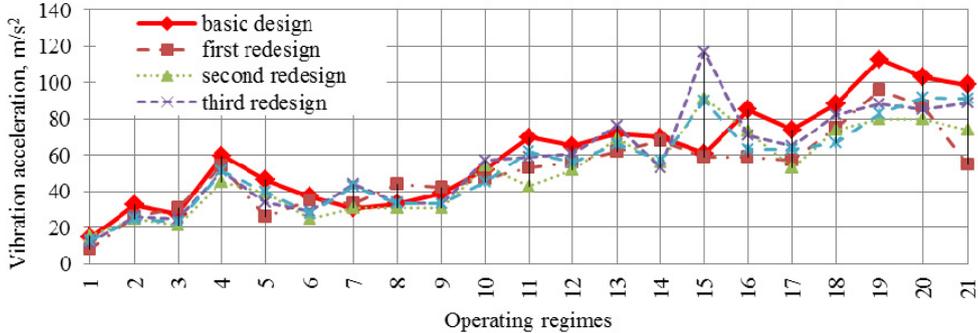


Fig. 23. Vibration acceleration at screw centrifugal pump inlet. ZOY plane. Direction is perpendicular to pump's axis

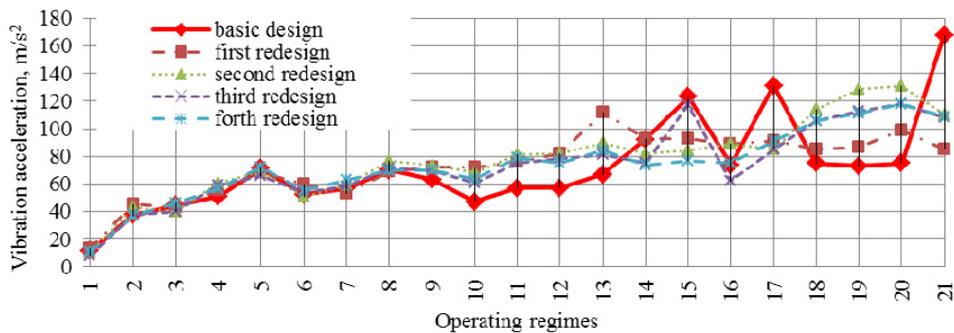


Fig. 24. Vibration acceleration at the flange connected two stages. XOY plane. Direction is parallel to pump's axis

## 7 Conclusions

1. The design modifications result in reduction of low-frequency oscillation of backflow vortex before screw. As a result they lead to flow stabilizing at pump entrance. Transfer the drain pipelines to supply line do not prevent backflow.
2. Proposed redesigns of initial section of screw centrifugal stage effect on pressure oscillations at outlet of gear stage.
3. Pump entrance redesigns lead to significant reduction of average stress at bearing surface.
4. Pump entrance redesigns reduce vibration acceleration up to 40%.
5. The best composition is the composition with screen, spinner and without gap.
6. The most efficient redesign is the design with screen, spinner and without gap.

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