Pulsations Induced by Vibrations in Aircraft Engine Two-Stage Pump

S A Gafurov[1,2], V A Salmina[1] and H Handroos[2]

[1] Department of Automatic Systems of Power Plants, Samara National Research University, Moskovskoe Shosse 34, 443086, Samara, Russian Federation
[2] Lappeenranta University of Technology, Skinnarilankatu 34, 53850, Lappeenranta, Finland

E-mail: gafurov@ssau.ru

Abstract. This paper describes a phenomenon of induced pressure pulsations inside a two-stage aircraft engine pump. A considered pumps consists of a screw-centrifugal and gear stages. The paper describes the cause of two-stage pump elements loading. A number of hypothesis of pressure pulsations generation inside a pump were considered. The main focus in this consideration is made on phenomena that are not related to pump mode of operation. Provided analysis has shown that pump vibrations as well as pump elements self-oscillations are the main causes that lead to trailing vortices generation. Analysis was conducted by means FEM and CFD simulations as well by means of experimental investigations to obtain natural frequencies and flow structure inside a screw-centrifugal stage. To perform accurate simulations adequate boundary conditions were considered. Cavitation and turbulence phenomena have been also taken into account. Obtained results have shown generated trailing vortices lead to high-frequency loading of the impeller of screw-centrifugal stage and can be a cause of the bearing damage.

1.Introduction

A model of flow induced vibrations in the two-stage pump was developed in paper [1]. Essence lies in trailing vortex stalling with a certain frequency (figure 1). This trailing vortex forms in a radial gap between screw blades and pump case. At that, screw blades are loaded by high-frequency pressure oscillations spreading from a gear stage to screw-centrifugal stage inlet (figure 2). These oscillations can coincide and lead to vibration loading of the pump rotor. Experimental results obtained in paper [2] does not confirm this model as high-frequency oscillation from the gear stage were not registered at the screw-centrifugal stage entrance.

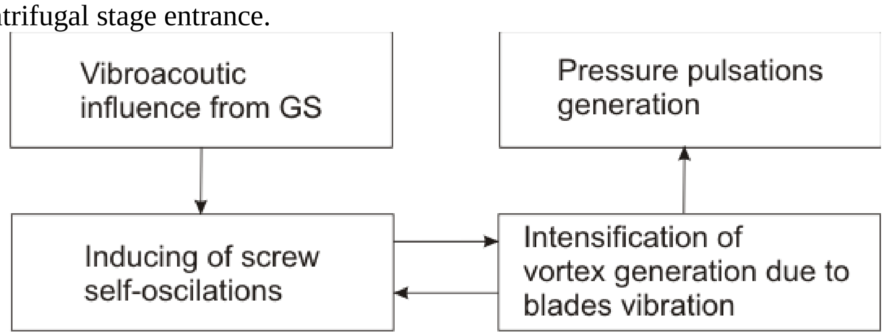


Figure 1. A model of flow induced vibrations in the two-stage pump

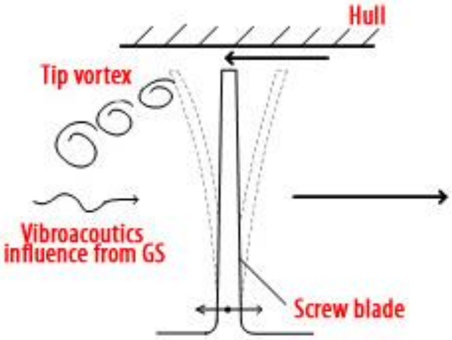


Figure 2. Scheme of pressure pulsations induced vibrations due to the trailing vortex stalling and gear stage influence

A number of factors such as boundary conditions, cavitation model, turbulence model, mesh quality and others influence on accuracy of numerical simulation. CFD approach was used in paper [3] for investigation of the sources of dynamic loading of the radial and thrust bearing inside pumps ND-25 and ND-32. Obtained results showed that the main cause was the so-called "acoustic-vortex resonance". Acoustic-vortex resonance is the phenomenon when natural frequencies of a screw, pressure pulsations caused by a gear pump and eddy frequency coincide with each other. Star-CD commercial software was used for CFD analysis. However, low quality of the computational mesh, using of the standard $k-\varepsilon$ turbulence model for near-walls flows did not allow to get accurate results and to make final conclusions about the real causes of bearing damages. It is worth to note that some construction elements were not taken into account. Additionally, there is a lack of verification of obtained results. At that, experimental results obtained in paper [3] did not confirm high-frequency pulsations at screw-centrifugal stage inlet. It discards the assumptions made by authors.

2. Two-stage pump modelling

In this work, another cause of high-frequency loading of the two-stage pump elements was accepted. Not only screw blades were considered but also blades of the impeller were taken into account. The schematic design of the pump shows that leakages from the gear stage are poured out to the chamber between stages (figure 3). This champed is also connected with the screw-centrifugal stage flow channel by means of the orifices in the bearing case (figure 4a) as well as by means of discharge holes in the impeller (figure 4b). Such a leakages propagation path results in high-frequency oscillations of blades of the screw and centrifugal wheel.

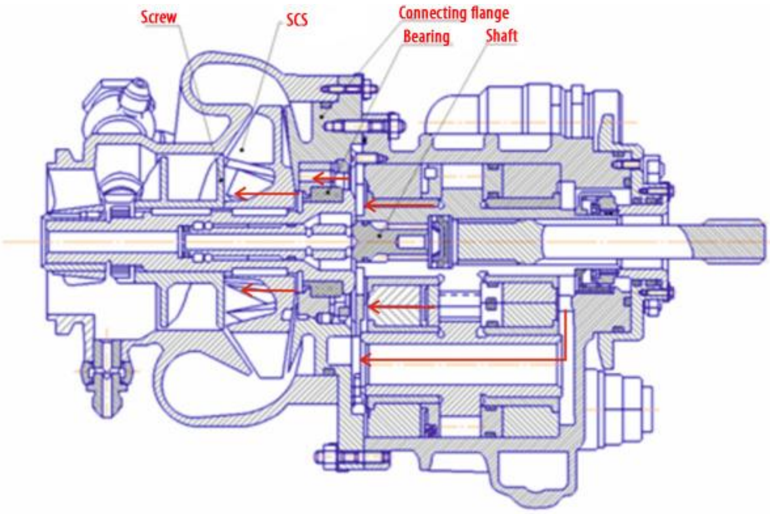
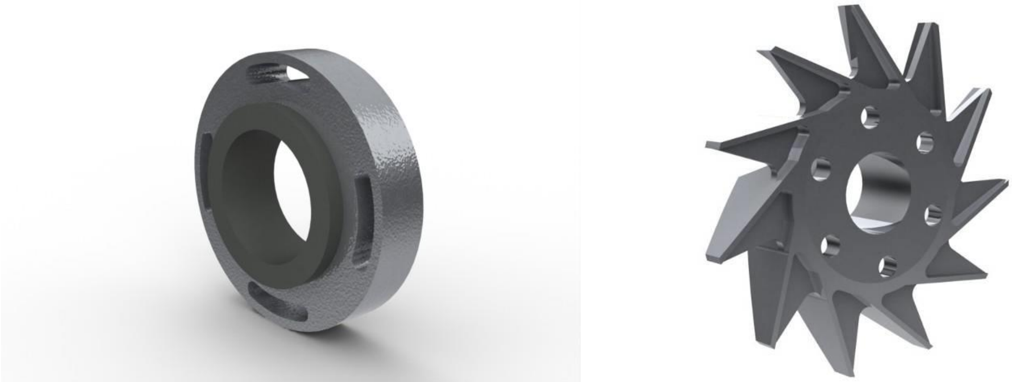


Figure 3. Leakages propagation path from the gear stage to the screw-centrifugal stage



(a) (b)

Figure 4. Construction elements on the screw-centrifugal stage. a) bearing; b) centrifugal

Results obtained in papers [4-6] show that the most important vibration sources of screw and centrifugal wheel blades are trailing vortices. Fluid flow passing the radial clearance between pump case and blades, interacts with the main flow and radial flow from root sections to peripheral. This interaction occurs at blades suction surfaces and results in rolling up of this flow into a vortex. Its radius and circulation depend on pump operating regime, its geometrical parameters and location of vortex appearance. Trailing vortex moves in the circumferential plane of the channel in a direction from the periphery of blades suction sides to their pressure sides.

Near-wall stream, occurring due to the radial flowing, is an unsteady [6]. Disturbances evolution leads to flow separation. These disturbances interact with vibroacoutic influence from the gear stage. This interaction results in oscillations of screw and impeller blades. Oscillations are significantly increased due to the coincidence of their frequencies and natural frequencies of the screw and impeller. Increasing of blades vibrations leads to intensification of pressure pulsations, caused by the hydrodynamic reasons [7,8], and to increasing of vibration loading of the end face of the bearing.

The block diagram of the interaction of vortex disturbances in the screw-centrifugal stage and screw and impeller blades, oscillated due to the vibroacoutic impact from the gear stage, is presented in figure 5.

The general model consists of the sub-models:

- Hydrodynamic model of the screw-centrifugal stage. This model allows to estimate the structure of the flow inside the pump.
- Finite-elements models of the screw and impeller to estimate their natural frequencies.
- Model of leakages from the gear stage. They are represented as sources of oscillations. Their parameters can be obtained theoretically or experimentally.

ANSYS CFX	ANSYS Mechanical		Experimental data
3D geometric model of SCS	**3D geometric model of screw**	**3D geometric model of centrifugal wheel**	**Leakages from GS**
CFD model of SCS	**FEM of screw**	**FEM of centrifugal wheel**	
MODEL OF INTERACTION BETWEEN SCS IMPELLERS AND TIP VORTEX TAKING INTO ACCOUNT GS			

Figure 5. The block diagram of the interaction of vortex disturbances in the screw-centrifugal stage and oscillated screw and impeller blades

2.1 Boundary conditions, pressure and flow simulation

Boundary conditions for the general model are:

- The eddy frequency and its parameters such as its structure, pressure and velocity distribution inside the eddy.
- Amplitudes of pressure pulsations obtained by the numerical simulation of the screw-centrifugal stage.
- Natural frequencies of the screw and impeller.
- Vibroacoutic performances of the gear stage.

Developed numerical model of the screw-centrifugal stage allows to estimate the separation of the flow inside a pump which results in forming of eddy zone. An accuracy of these turbulence flow can be estimated by yplus parameter distribution along the solid walls. This distribution is shown in figure 6.

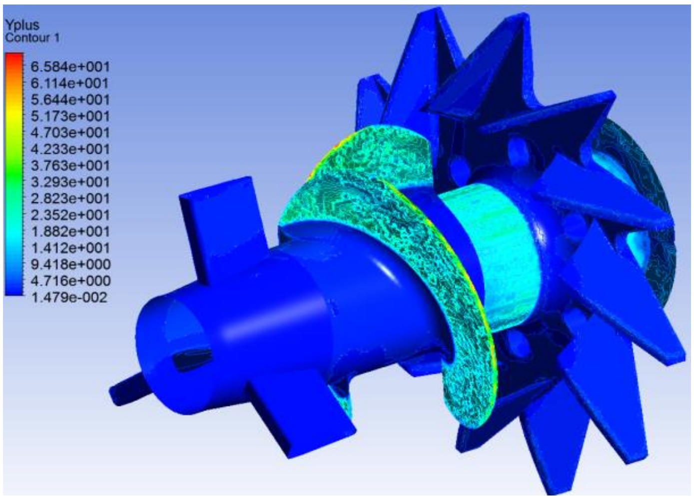
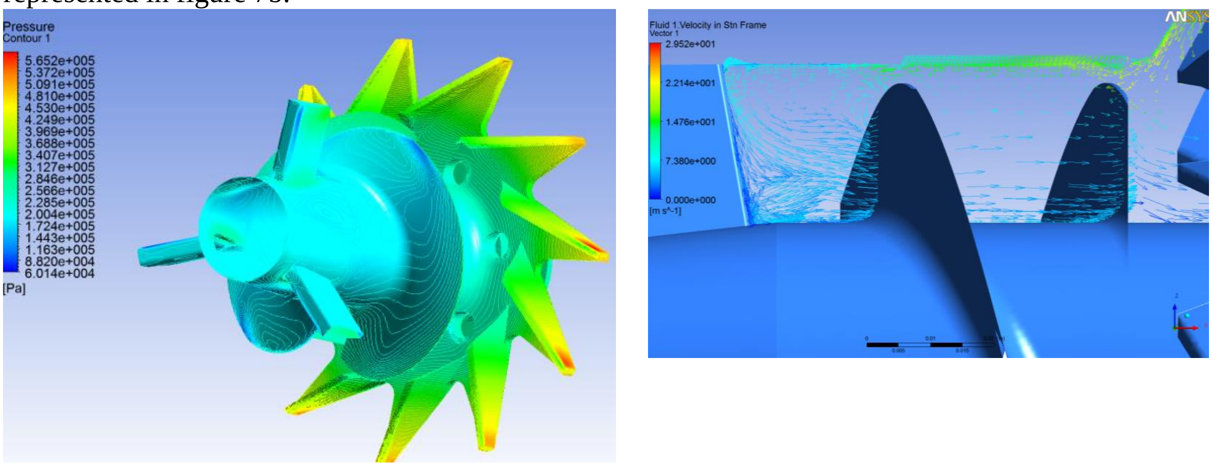


Figure 6. The distribution of yplus parameter along the rotor walls of the screw-centrifugal stage

Figure 7a represents the pressure distribution along the solid walls of the screw-centrifugal stage rotor. There is a separated zone on the suction side of the screw. The flow structure inside the screw is represented in figure 7b.



a b

Figure 7. a) Pressure distribution along the solid walls of the screw-centrifugal stage rotor N=4325 rpm, G_{SCS}=8682 kg/h, P_{in}=1.9 atm, T=45 ^{0}C; b) Flow structure on the operating regime N=4325 rpm, G_{SCS}=8682 kg/h, P_{in}=1.9 atm, T=45 ^{0}C

2.2. Separation frequency determining. FEM-analysis

Numerical simulation allows to estimate the parameters of the eddies and to determine its separation frequency by mean of Strouhal number:

$$f = \frac{\upsilon \cdot Sh}{l_{cavity}} = \frac{Sh\sqrt{2g \cdot \Delta H}}{l_{cavity}} \qquad (1)$$

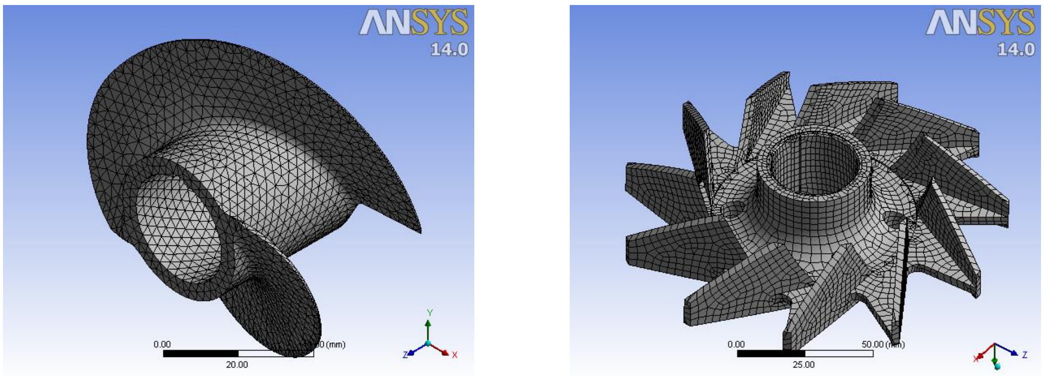
where ΔH is pressure difference on the peripheral section of the blade; l_{cavity} is the maximum length of the cavity before the stall break.

Conducted modal FEM-analysis has shown that natural frequencies of the screw and impeller are in the range of 3700-5000 Hz (Table 1).

Table 1. Natural frequencies of the screw and impeller

Natural mode shape	Natural frequency of the screw, Hz	Natural frequency of the impeller, Hz.
f_1	3310.6	3769.6
f_2	3812.7	3846.6
f_3	4373.3	3901
f_4	4887.8	3964.9
f_5	5420.9	4155.6

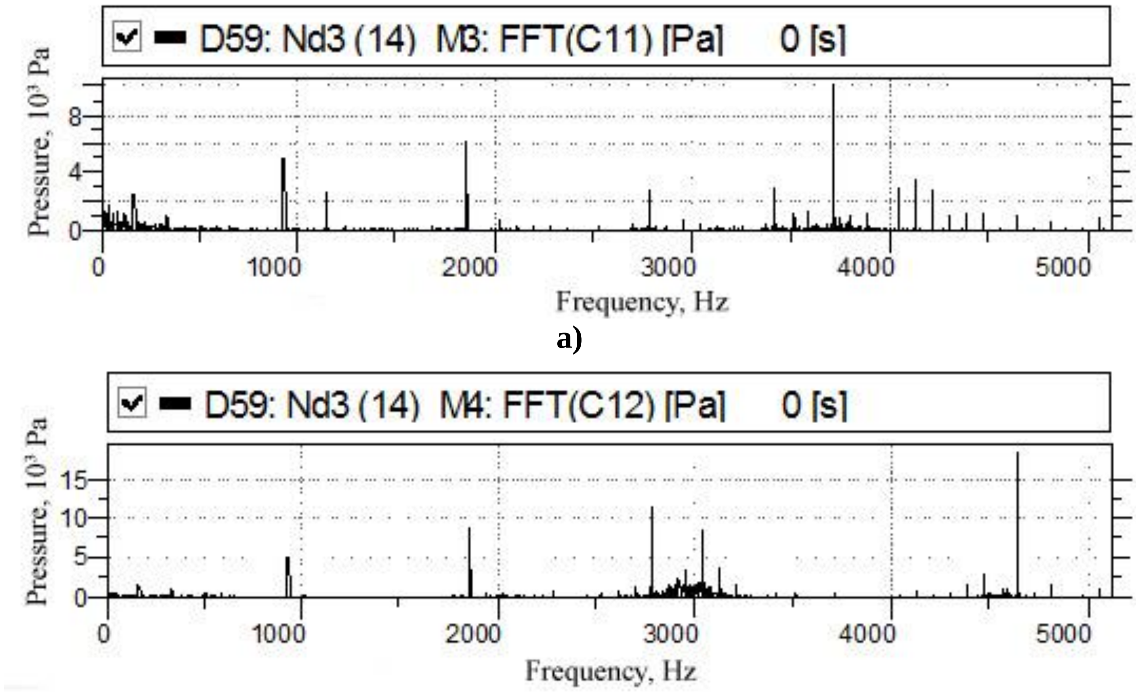
Modal analysis was performed by ANSYS commercial code. Finite-elements meshes for the screw and impeller are shown in figure 8. Size of the meshes does not exceed 3 mm.



(a) (b)

Figure 8. a) Finite-element meshes of the screw; b) impeller

Natural frequencies of the screw and impeller were also obtained experimentally by means of knocking method. Wheels were hung up on a thread and excited by means of a hummer with a wooden tip. The changing of the sound pressure was being registered on 30 cm distance. Theoretical and experimental spectrums of pressure pulsations are shown in Figure 9. There are coincidences of the eddy frequencies separated from the wheels and their natural frequencies on several operating regimes. It leads to increasing of vibration of the screw and impeller and, as a result, to intensification of the trailer cavitation.



b)

Figure 9. Spectrums of pressure pulsations measured on the pump operating regime - N=4325 rpm, G_{SCS} =8682 kg/h, P_{in} = 1,9 kgf/cm^2, T=45 ^{0}C;
a) outlet of the screw-centrifugal stage;
b) in the chamber between the pump stages

Additionally, operating characteristics of the pumps ND-25 and ND-32 were investigated in paper [9]. This paper shows that the main contribution in total level of vibration makes the high-frequency oscillations above 3 kHz. This paper did not determine the loading source. These investigations confirm the hypothesis assumed here.

3. Conclusion

FEM analysis of natural frequencies of the screw and impeller of the two-stage pump was provided. CFD analysis of flow structure inside the screw-centrifugal stage also was developed. During CFD analysis the main focus was made on pressure distribution and investigation of vortices flow inside the screw-centrifugal stage. FEM-analysis has shown that natural frequencies of the screw and impeller lie inside a range of 3700-5000 Hz. Obtained results were compared with experimental. The comparison has shown a good accuracy of the numerical results. Natural frequencies of screw and impeller and frequencies of trailing vortices were turned out to coincide on pump operating regime. Such coincidence leads to increasing of screw and impeller blades vibration, pressure pulsations, and, as a result, to intensification of the trailer cavitation.

4. Acknowledgement

This work was supported by the Ministry of Education and Science of the Russian Federation in the framework of the implementation of the Program ‘‘Research and development on priority directions of scientific-technological complex of Russia for 2014– 2020 and Grant # 01v-R200-201.

References

[1] Gasparov M K, Pressure pulsations at stream and separate flows, *Samara State Aerospace University*, Samara, (2009) pp. 77

[2] Korotkov F, Ocenka vlijanija jekspluatacionnyh faktorov na rabotosposobnost' agregata ND-25 [Estimation of exploitation factors influence on combined pump ND-25 durability], *OJSC "Scintific and production enterprice "Temp" named after F.Korotkov,* Moscow, (1990)
[3] Wang G H, Unsteady Dynamics of Cloud Cavitating Flows around a Hydrofoil, *Proceedings of the 7th International Symposium on Cavitation CAV2009*, (2009)
[4] Kinelev, V G, Fizicheskaja model' kavitirujushhego shneko-centrobezhnogo nasosa, rabotajushhego v shirokom diapazone rezhimov po rashodu Kiev [Physical model of cavitating screw-centrifugal pump operating in a wide range of mass flow], *Kavitatsionnyye avtokolebaniya v nasosnykh sistemakh*, Naukova Dumka, (1976), pp. 100-107
[5] Karelin Y, Cavitation phenomena in axial and centrifugal pumps, *Moscow:Mashinostroenie,* (1975)
[6] Shapiro S, Structure of real-scale flow in cenrifugal and axial flow, *Moscow: MGIU,* (2004)
[7] Antonov A K, Pressure pulsations at stream and separate flows, *Moscow: Mashinostroyeniye*, (1990)
[8] Rimskiy-Korsakov A B, Phisical principles of noise formation in blowing machines, *Moscow: Nauka*, (1988)
[9] OJSC, Samara construction bureau of engines design, Technical report, *Samara*, (1998) pp.25