LOW RIPPLE SOLUTION: VANE-IN-GROOVE PUMP WITH PRECOMPRESSION

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ABSTRACT

Paper presents new approach to pressure ripple and acoustic noise reduction in a vane pump with working chamber located in annular groove of a rotor face (Vane-In-Groove pump) by means of working fluid precompression.

Vane-In-Groove pumps with adaptive rotor comprise means for variation of the volume of working fluid transferred portions and pressure in them. This feature as well as lengthy enough transfer zone makes possible precompressing the transferred portions so that to balance working fluid pressure in them with the output pressure. Thus the origin of decompression impacts eliminates completely.

Two main methods for adjusting the degree of the working fluid precompression are described. The first method works with adjusting the amplitude of the transferred portions volume variation depending on operational pressure and is preferable for reduction of remaining kinematical ripple down to 0.2% and lower. The second method works with adjusting the total transfer angle and is preferable cost wise. One embodiment of the second method was tested and described in details.
The tests equipment, schematic diagram and procedure for output pressure ripple measurement are presented as well as the tests results for 28 cm^3 pump. The test results illustrate low levels of output pressure ripple at 500 – 2000 RPM and 10 – 30 MPa.

KEYWORDS: low pressure ripple, quiet operation, vane pump/motor, Vane-In-Groove, decompression flow, working fluid precompression

1. INTRODUCTION

The origin of pressure ripple in the Vane-In-Groove pumps comes from a common shortcoming of all existing positive displacement pumps that have high volumetric efficiency. The point is that when a closed transferred volume of a working fluid moves from the suction to the pumping cavity the pressure in it does not reach the outlet pressure by the moment of merging with the pumping cavity. At that moment, due to compressibility of a working fluid, a counter flow (*decompression flow*) of the working fluid from the pumping cavity to the transferred volume arises. The decompression flow equalizes the transferred fluid pressure and the pressure in the pumping cavity and gives rise to the delivery pulsations as well as the pressure ripple in an outlet line. This in it's turn lead to power losses, vibrations and noise and can cause destruction of outlet line elements and a pump itself.

2. VANE-IN-GROOVE PUMP WITH WORKING FLUID PRECOMPRESSION

Known methods of reducing a decompression impact can be divided in two types. The first one use a counter flow (actually leakage) created between a transferred volume and an outlet cavity before they merge. This allows fluid pressure in a transferred volume to approach the outlet pressure thus reducing the pressure drop that generates a decompression flow. There are many implementations of this method in all types of positive displacement pumps. This method can be called passive from the point of view that no actions are taken on a transferred volume while it travels between a suction and a pumping cavities. Two major disadvantages of this method are obvious. Firstly, some useful power dissipates in mentioned counter flow heating a working fluid and a pump. Secondly, the method can be only optimized for a certain combination of parameters: fluid compressibility and viscosity, rotation speed, delivery and outlet pressure.

Real solution of the decompression problem can be achieved by the active pulsation suppression system based on the working fluid precompression provided in transferred volumes while they are separated from both a suction and a pumping cavities.

This solution can be referred to the second type of known methods. Usually [8] precomrpession is provided in inter-vane cavities of radial-vane pumps due to special shape of a cam ring. However this implementation of the precomrpession method has the same disadvantage [7], namely it can be only optimized for a certain combination of working parameters. Another disadvantage is it's inapplicability for variable displacement pumps.

Vane-In-Groove pump with Adaptive Rotor (see [9]) offers easy implementation of the working fluid precompression based on the ability of the supporting part of the adaptive rotor to move relative to its working part. In particular, the supporting part (1) of the rotor can be tilted at a small angle g around the axis perpendicular to the axis of rotation of the working part (2) of the rotor, as is shown in Fig. 1. Since the rotor parts are hydraulically connected to each other via the force chambers of variable length (3) and each force chamber is a part of the corresponding transferred volume (4), when it's length is changed the corresponding transferred volume is changed as well. If the tilt axis of the supporting part of the rotor lies parallel to the line connecting the middle points of the forward and backward transfer areas, the volume of the force chambers will cyclically vary according to the sine law thus providing sine variation of the transferred volumes.

Detailed description of the active pulsation suppression system and it's mathematical model is given in [5]. The system provides several methods of regulation of the transferred volumes variation needed for adjusting the system to outlet pressure and delivery changes. Paper [5] describes one of these method, namely the amplitude regulation of the transferred volumes variation. The other methods as well as their embodiments in a Vane-In-Groove pump/motor are described in details in patent application [6].

Present paper discloses simple and cost wise preferable method – phase (or total angle) regulation of the transferred volumes variation. The essence of this method is regulating the total angle f_{total} i.e. the angle of the rotor rotation between the angle of detaching the transferred volume from the inlet cavity f_{detach} to the angle of it's merging the outlet cavity $f_{merg.}0$. Thus the total angle is the angle range within which the transferred volume is separated from both the inlet and outlet cavities. Total amplitude of the transferred volumes variation at this method of regulation is chosen corresponding to the maximum pressure drop between the inlet and outlet ports and maximum pump displacement. The corresponding variation of the transferred volumes and pressure of the working fluid in it are shown in Fig. 2 (curves 1a, 1b). All graphs are traced against the angular travel of the transferred volume within the range from f_{detach} to $f_{merg.}0$.

Method of the phase regulation can be implemented as follows (Fig. 1): the total angle is changed by connecting the transferred volume (4) with the outlet cavity (5) ahead of time, at the moment when the pressure in the transferred volume becomes equal to the outlet pressure. This type of implementation is called the regulation of the angle of merger. As a result of the ahead of time connection further change of pressure of the working fluid in the transferred volume stops (curves 2b, 3b in Fig. 2). This ahead of time connection of the transferred volume with the outlet cavity can be done by shifting the vane (6) that is currently separating the mentioned transferred volume from the outlet cavity or by connecting the transferred volume with the outlet cavity through a normally closed bypass duct (7). In the latter case starting from the moment of unlocking bypass duct a part of fluid is displaced out of the transferred volume via the bypass duct to the outlet cavity (curves 2c,d, 3c,d in Fig. 2) (for a hydromotor – out of the outlet cavity to the transferred volume). Bypass duct can be unlocked, for example, by back-pressure valve (8) that is opened when the sign of the pressure difference between the ends of the bypass duct is changed. The shape of vanes used in Vane-In-Groove pumps is chosen so that to provide hydrostatic pressing of a vane to forward transfer limiter (9) where the pressure of a working fluid ahead of a vane is higher than behind it. This allows to use a vane itself as a back-pressure valve and no external bypass duct is needed.

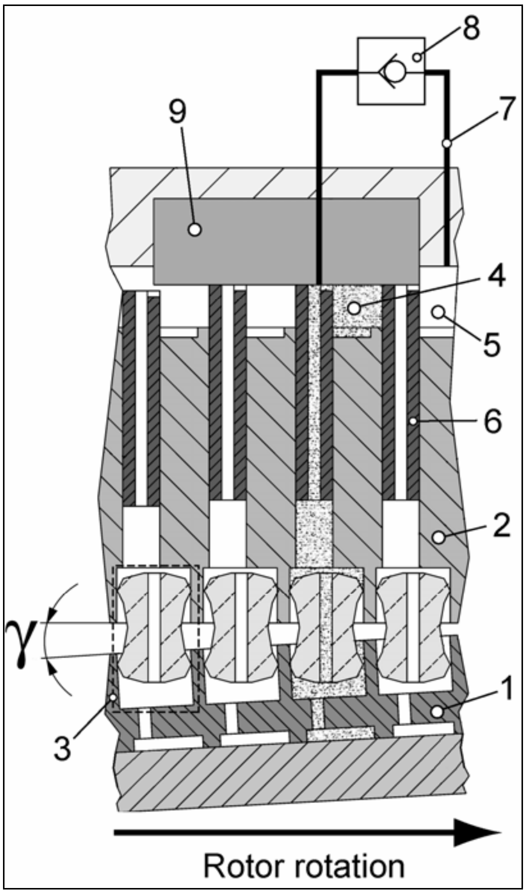

Figure 1: Schematic representation of the phase regulation method implemented in a form of regulating the angle of merger $f_{merg.}$

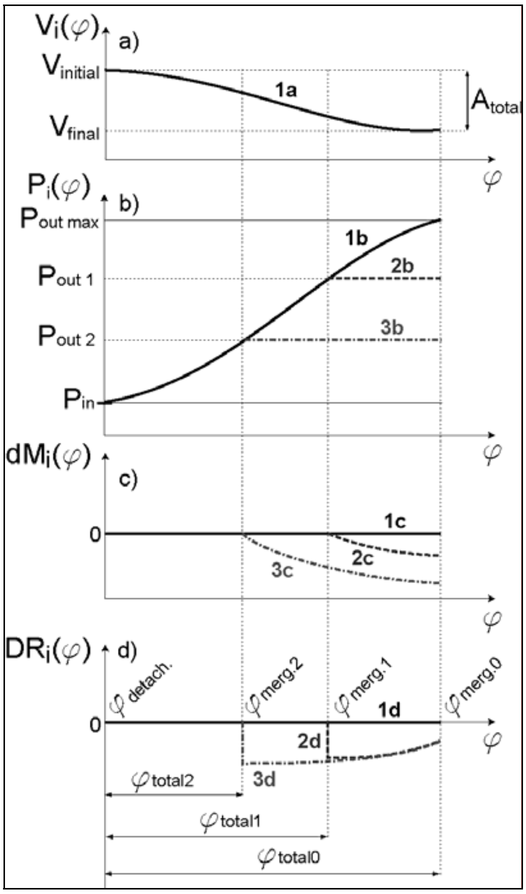

Figure 2: Variation of the transferred volume (a), change of mass (c) and pressure of the working fluid in it (b) and fluid flow rate between the transferred volume and outlet cavity (d)

Obvious advantage of the phase regulation method is it’s simplicity. The major disadvantage of this method is secondary flow pulsations appearing at the outlet pressure lower than maximal operational pressure. Since the total amplitude of the transferred volumes variation is adjusted for the maximal outlet pressure the fluid pressure in the transferred volume reaches the outlet pressure at the moment before their merger. Since that moment the transferred volume is no longer separated from the outlet cavity and some part of the fluid is displaced from the transferred volume to the outlet cavity generating additional delivery. This process repeats for each transferred volume producing secondary flow pulsations. The good thing is that these flow pulsations don’t have high frequency components because sharp decompression impact has been completely eliminated.

To verify the active pulsation suppression system in real-life prototypes with phase regulation of the transferred volumes variation (namely regulation of the angle of merger by vane shifting) have been designed assembled and tested on our internal testing stand. All they have shown functionality and good performance. One of the Vane-In-Groove pump prototypes with fixed displacement of 28 cm^3 was put to the comparative test for outlet pressure ripple in two modifications: with the active pulsation suppression system adjusted for 20 MPa and without it. Required adjustment is done by choosing corresponding tilt angle (equal to 0.9 degree) of the rotation axis of the supporting part of the rotor.

3. TEST INSTALLATION AND PROCEDURE

Figure 3 presents schematic of the internal testing stand. The test circuit is closed type without backup. Industrial oil similar to HLP 46 was used as a working fluid. The water cooling system (11) working on counter-current flow principle kept up the fluid temperature within the limits of 30±5 °C. Adjustable orifice (9) was used as a load. Maximal drive power provided by the electromotor was 30 KW. Outlet pressure and pressure ripple oscillogram, oil temperature and pump housing temperature were recorded.

The purpose of this test was to compare the outlet pressure ripple amplitudes produced by “regular” Vane-In-Groove pump and the same pump provided with the active pulsation suppression system adjusted for 20 MPa. This will give an idea on how the fluid born component of the pump noise changes when the active pulsation suppression system is used.

The test matrix was defined as follows:
- operating pressure: 10, 20 and 30 Mpa, i.e. the pressure lower, equal and higher than 20 MPa that the active pulsation suppression system was adjusted for.
- rotation speed: 1500 revolutions per minute

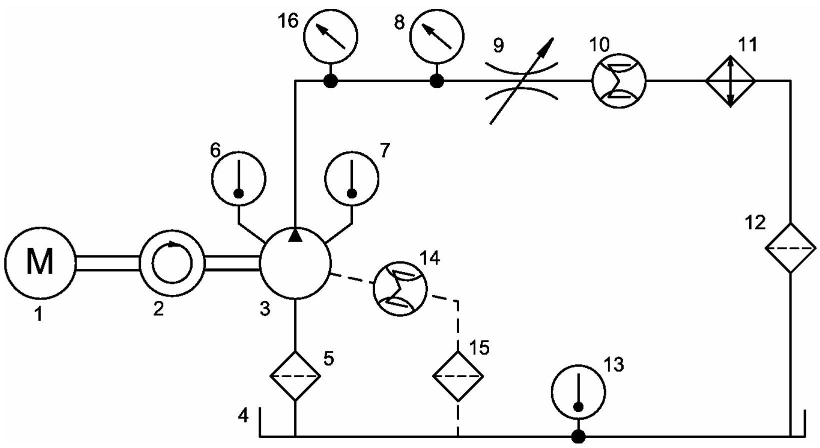

Figure 3: Scheme of the internal testing stand (auxiliary and safety equipment is not shown) 1– electromotor, 2 – tachometer, 3 – tested pump, 4 – tank, 5, 12, 15 – filters, 6, 7, 13 – thermometers, 8 – dynamic pressure sensor, 9 – adjustable load throttle, 10, 14 – flow meter, 11 – counter-current flow cooler, 16 – pressure gauge

4. TEST RESULTS

The outlet pressure ripple amplitude was measured by dynamic pressure sensor, at stationary working points of the test matrix and the results are presented as oscillograms in Fig. 6, Fig. 7, Fig. 8, Fig. 9. High frequency oscillations (about 1450 Hz) are the result of wave reflections in the outlet line between the outlet cavity and the load. For comparison Figure 10 is given that presents theoretically calculated outlet pressure ripple for “ideal” model (with no leakages) of the tested pump at 40 MPa (solid curve).

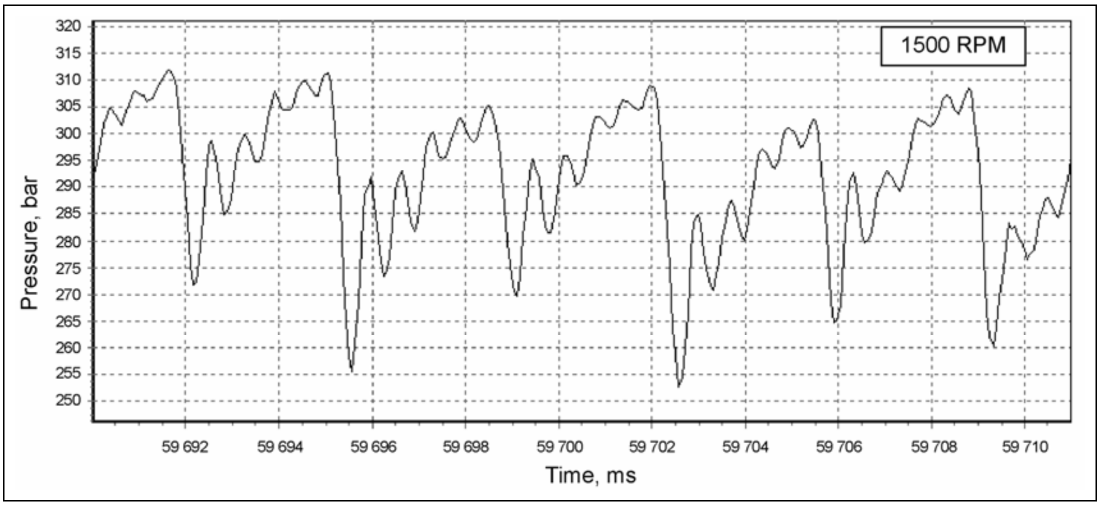

Figure 6: Amplitude of the outlet pressure ripple at 30 MPa (300 bar) and 1500 RPM against time for the pump **without** the active pulsation suppression system

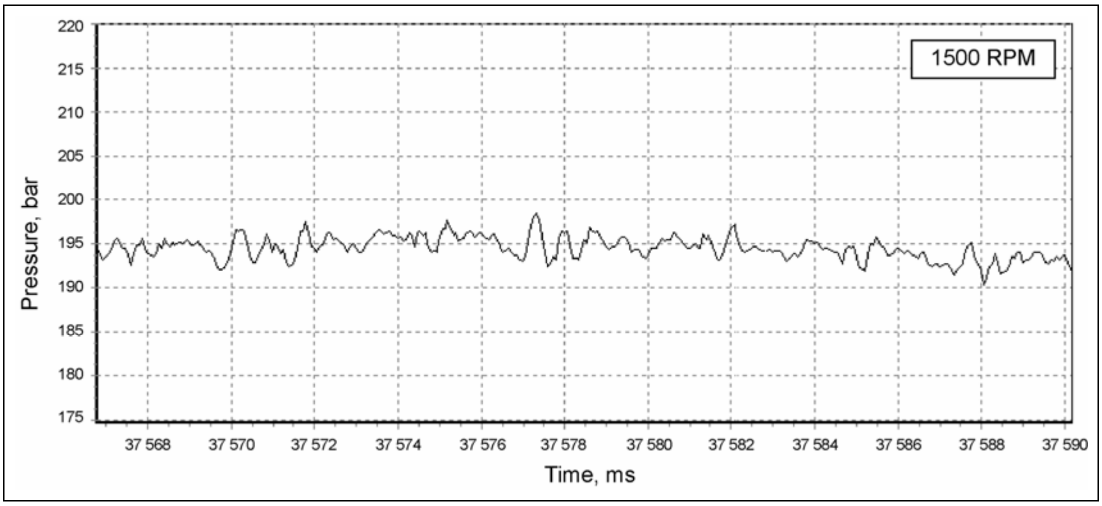

Figure 7: Amplitude of the outlet pressure ripple at 20 MPa (200 bar) and 1500 RPM against time for the pump **with** the active pulsation suppression system adjusted for 20 MPa (200 bar).

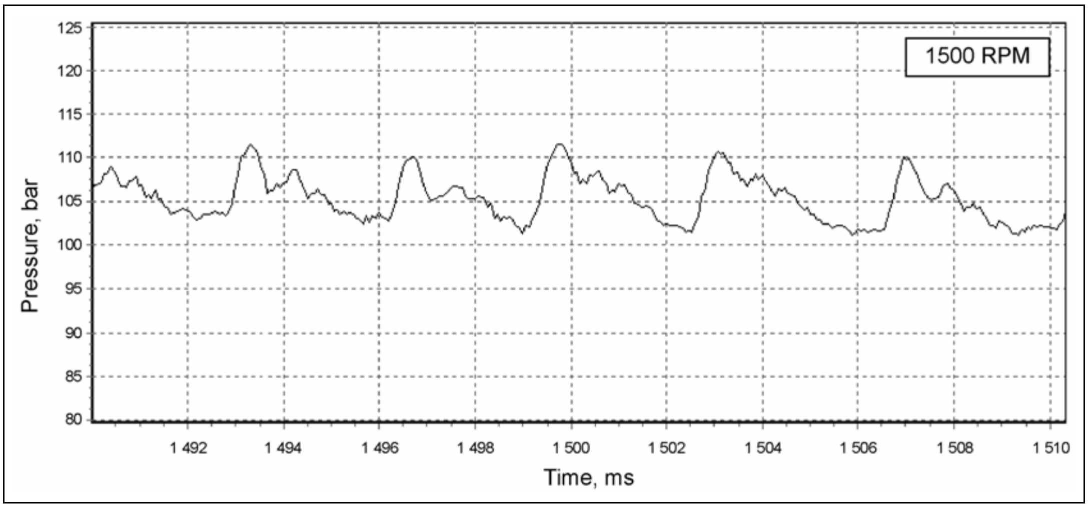

Figure 8: Amplitude of the outlet pressure ripple at 10 MPa (100 bar) and 1500 RPM against time for the pump **with** the active pulsation suppression system adjusted for 20 MPa (200 bar).

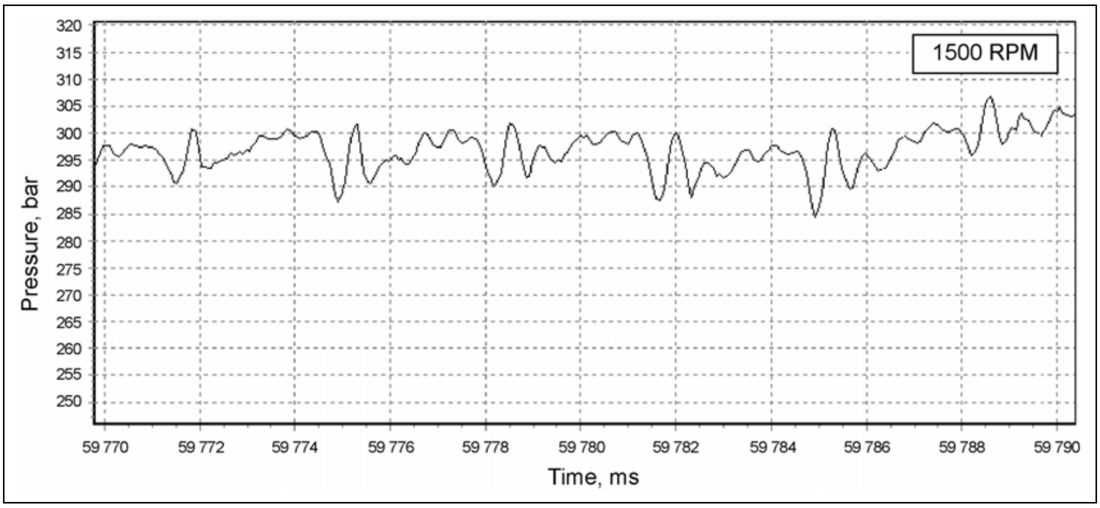

Figure 9: Amplitude of the outlet pressure ripple at 30 MPa (300 bar) and 1500 RPM against time for the pump **with** the active pulsation suppression system adjusted for 20 MPa (200 bar).

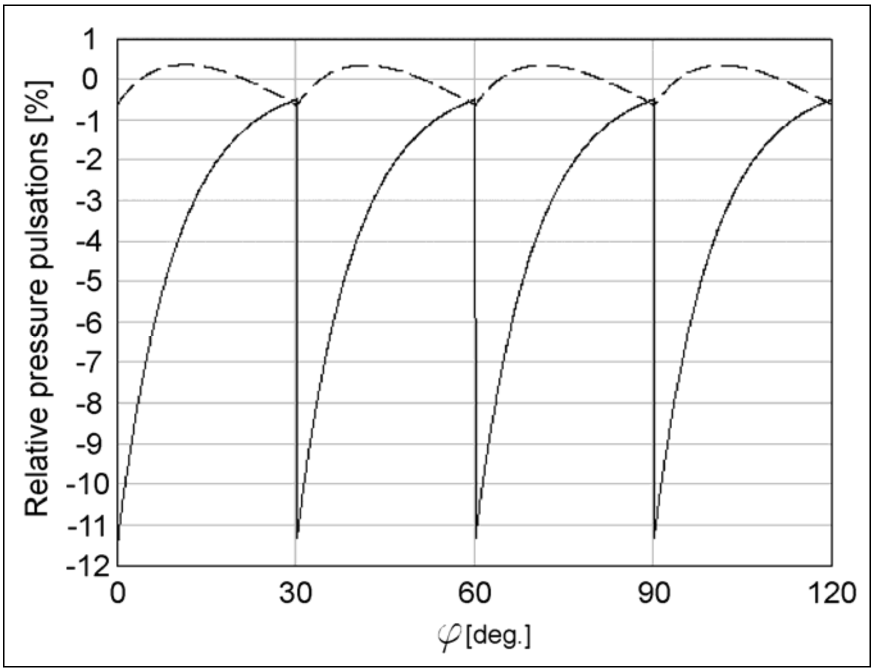

Figure 10: Outlet pressure behavior (average level and ripple) against the angle of the rotor rotation for "ideal" pump at 40 MPa (400 bar):
without (solid line) and with the active pulsation suppression system (dash line)

Figures 7 show significant reduction of the outlet pressure ripple amplitude especially at the vane's frequency when using the active pulsation suppression system. So, at 1500 RPM the pressure ripple amplitude drops down from 17% to 1,2% of the average outlet pressure. This corresponds to the theoretical calculations (Fig. 10) if high frequency ringing of the measuring tract is smoothed in Fig. 6 – Fig. 9.

Figure 8 shows reduction of the outlet pressure ripple amplitude in case of excess precompression (outlet pressure is lower than calculated for complete elimination of decompression). Since the fluid pressure in the transferred volume reaches the outlet pressure before it merges the outlet cavity the vanes in the forward transfer area unlock corresponding the transferred volumes ahead of time producing positive outlet pressure peaks (rather than negative). These peaks amplitude is defined by the pressure drop required to unlock the vanes. Some pump complication namely mentioned back-pressure valve installed in a bypass duct can reduce the amplitude of these positive peaks.

Figure 9 shows reduction of the outlet pressure ripple amplitude in case of insufficient precompression (outlet pressure is higher than calculated for complete elimination of decompression). Since the fluid pressure in the transferred volume never reaches the outlet pressure some decompression flow occurs when current transferred volume merges the outlet cavity. Residual decompression flow generates negative outlet pressure peaks. These peaks amplitude is defined by residual pressure drop at the moment of merging the transferred volumes with the outlet cavity. Even in this case the pressure ripple amplitude drops down from 17% (without the active pulsation suppression system) to 4% (with the active pulsation suppression system) of the average outlet pressure.

CONCLUSION

Data presented in 4 show significant reduction of the outlet pressure ripple and fluid born component of acoustic noise when using the active pulsation suppression system based on the precompression method. This system transforms sharp decompression pressure pulsations to relatively small secondary cinematic nonuniformity of displacement which can be readily surmounted in a pressure line and useful load by volumetric capacity. Additional positive effect of this system implementation is recycling some part of energy otherwise being wasted on decompression.

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