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ANALYSING OF SEALING PROPERTIES ON A BOLTED JOINT WITH A STEEL GASKET BY USING FINITE ELEMENT METHOD

ANALYZOVANIE TESNIACÝCH VLASTNOSTÍ SKRUTKOVÉHO SPOJA S OCEĽOVÝM TESNENÍM POUŽITÍM METÓDY KONEČNÝCH PRVKOV

Abstract

This article deals with study of contact pressure/gap distribution on a bolted joint of a hydrostatic transmission. The study was performed via numerical analyses with support of ANSYS a finite element method software package. The main focus in the results evaluation is aimed on evaluation of the contact pressure/gap on the bolted joint after tightening individual bolts and after applying operational loads on the hydrostatic unit. Operational loads in the analyses were represented by applying of an operational pressure into the individual areas of the unit. The operational pressure was applied in two time steps of the Finite Element Method analysis to simulate normal and reverse mode of the hydrostatic transmission. Numerical results were verified through experimental measurements carried out by using FujiFilm tests.

Abstrakt

Tento článok sa zaoberá skúmaním rozloženia kontaktného tlaku resp. kontaktnej vôle skrutkového spoja hydrostatického prevodníka. Skúmanie týchto ukazovateľov bolo realizované numerickými analýzami s využitím konečno-prvkového softvéru ANSYS Workbench. Cieľom príspevku je vyhodnotenie kontaktného tlaku/vôle skrutkového spoja po uťahovaní jednotlivých skrutiek a po aplikácii prevádzkového zaťaženia hydrostatickej jednotky. Prevádzkové zaťaženie v analýzach predstavovalo aplikáciu prevádzkového tlaku do jednotlivých oblastí jednotky. Pracovný tlak bol aplikovaný v dvoch časových krokoch konečno- prvkovej analýzy, nakoľko bol simulovaný normálny a reverzný chod hydrostatického prevodníka. Numerické výsledky boli následne porovnané s experimentálnymi meraniami, ktoré boli vykonané pomocou FujiFilm testov.

Keywords

Hydrostatic transmission, Tightness, Contact pressure, Gasket sealing, Bolted joint, FEM, FujiFilm test, Experiments.

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1 INTRODUCTION

There is no doubt, that bolted joints are commonly used in many applications as the preferred fastening method due to their high efficiency, low price and ready availability. Although bolted joints have been replaced by welding in many applications, mainly due to the extra cost of fasteners and their labour intensive installation. However, the biggest advantage of the bolted joint is that it is not a permanent connection and can be disassembled easily should the need arise. This makes them irreplaceable in the engineering environment.

The bolted joint plays a significant role in the assembly of hydrostatic machines, especially on a hydrostatic unit. Hydrostatic machines are commonly used in wide range of applications in mobile machinery. Some advantages of hydrostatic machines are very impressive in several cases, but the constant demands of the market place and the customers ever changing requirements, force the manufacturers to provide better products, offering higher performance and efficiency, which means working at higher pressures. Hydrostatic machines use hydraulic fluid to operate and there is always a great emphasis placed on the impact that these operating fluids have on our environment should spillage or leaking occur. The two reasons above lead us to study the sealing properties of the bolted joint. More about hydrostatic transmissions see in [2], [5], [7], [12].

Nowadays, numerical simulations make the design process of products much easier and more cost effective because they offer several solutions, determined using several conditions in the simulation environment without having to manufacture and test all the options (more information about finite element method see in [1], [3], [4], [8], [9], [11]). Firstly this can lead to saving unnecessary expenditure & secondly can save one from wasting time. Some factors that determine the correct sealing properties of the bolted joints & seal being used are, their material, shape and size of the bolts as well as the pretension force applied. The stiffness of the flanges being clamped also play a major role in the seal. In order to study these factors we have used numerical tests within the ANSYS Workbench. The case of analyzing a bolted joint with a steel gasket on the aforementioned hydrostatic unit in conjunction with an axial piston pump. To prove the accuracy of the results obtained, experimental measurement were carried out via a Fujifilm test.

2 THE PROBLEM DESCRIPTION

First of all, it is necessary to note that analyzed bolted joint was on the hydrostatic unit of the axial piston pump of swash plate design (See Figure 1). When variable displacement is required for a machine, piston pumps are the most commonly used mechanisms. They offer certain amount of advantages but especially higher operating pressure. To avoid pressure losses it is needed to secure proper sealing properties of the pump joints.

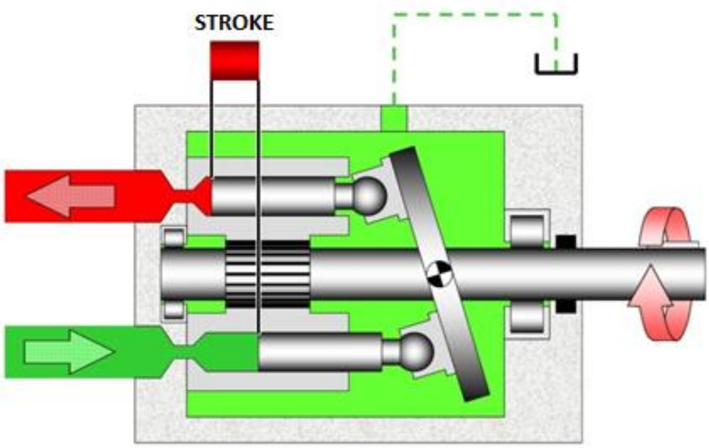

Fig. 1 Axial piston pump [10]

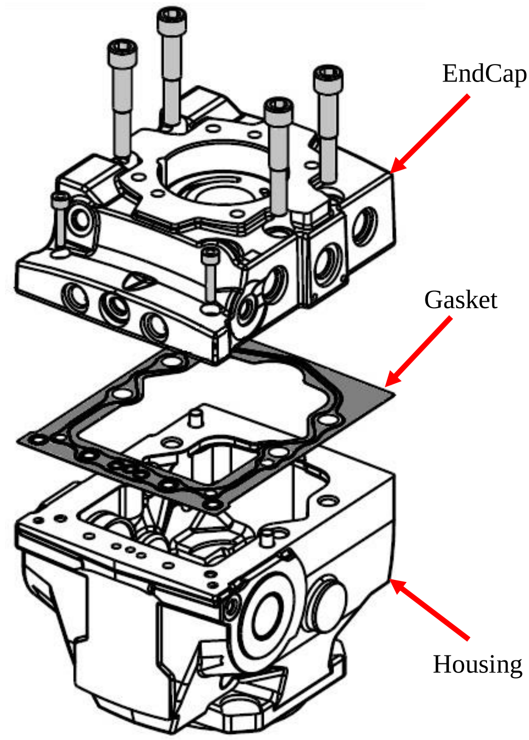

Fig. 2 Area of the interest (CAD software IDEAS)

In the following text, the area of the interest, interface between EndCap and Housing shown on the Figure 2, will be analyzed.

A couple of simplifications were applied on the analyzed model of the hydrostatic unit to reduce the size of the FEM model. It wasn't possible to apply a symmetry plan due to holes on the side of the housing, therefore entire model had to be analyzed. Bolt bodies were created without the threats, glued contact was used instead to simulate bolt screwed up into the flange (housing). EndCap and Housing were clamped together by using 6 bolts. During the simulation two kinds of loads were applied - bolt pretension and operational loads of the piston pump. Bolts were tightened and therefore steel gasket was compressed and "sealing lines" were created. After this, two most important information were obtained: distribution of the contact pressure and contact gap. This distribution of the contact pressure/gap was analyzed under operational loads afterwards.

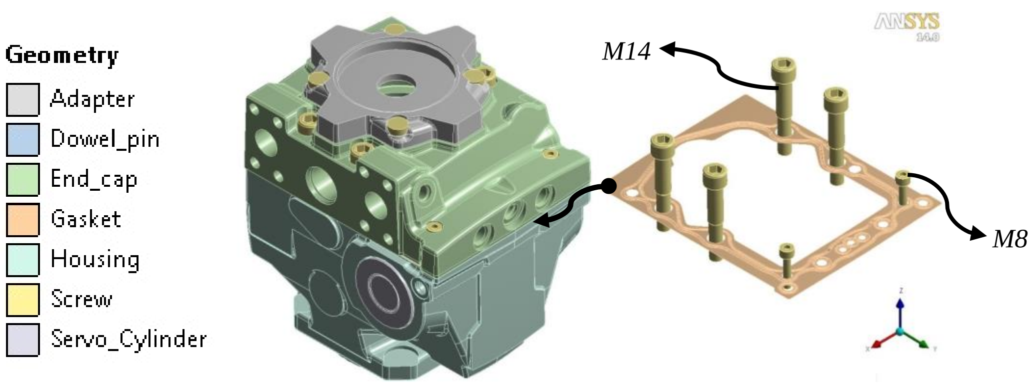

Fig. 3 CAD model of the Hydrostatic unit (ANSYS software)

3 SOLUTION APPROACH

Numerical tests were divided into two analyses. This was carried out to validate the numerical model with the experimental measurement by using a Fuji Film test. The FEA also studied the stiffness of the EndCap & Housing under operational loads of the hydrostatic pump. Secondly, an analysis including the steel gasket was done. The distribution of the contact pressure/ gap between the gasket and flanges was studied (End Cap and Housing).

3.1 Experimental validation

The Fuji Film pre-scale is a very thin paper which consists of a polyester layer that contains microcapsules. After applying force onto the film, the capsules rupture is caused across the contact area. Depending on the value of the pressure applied the colored image is created with variable color intensity. The most intense color indicates the areas experiencing the highest pressure concentrations.

In the experimental analysis Fuji Film (pressure range 10 - 50 [MPa]) was inserted between the EndCap and the Housing and six bolts were tightened in a prescribed order by applying specified bolt torque afterwards. Figure 4 shows very accurate results were obtained (contact pressure distribution).

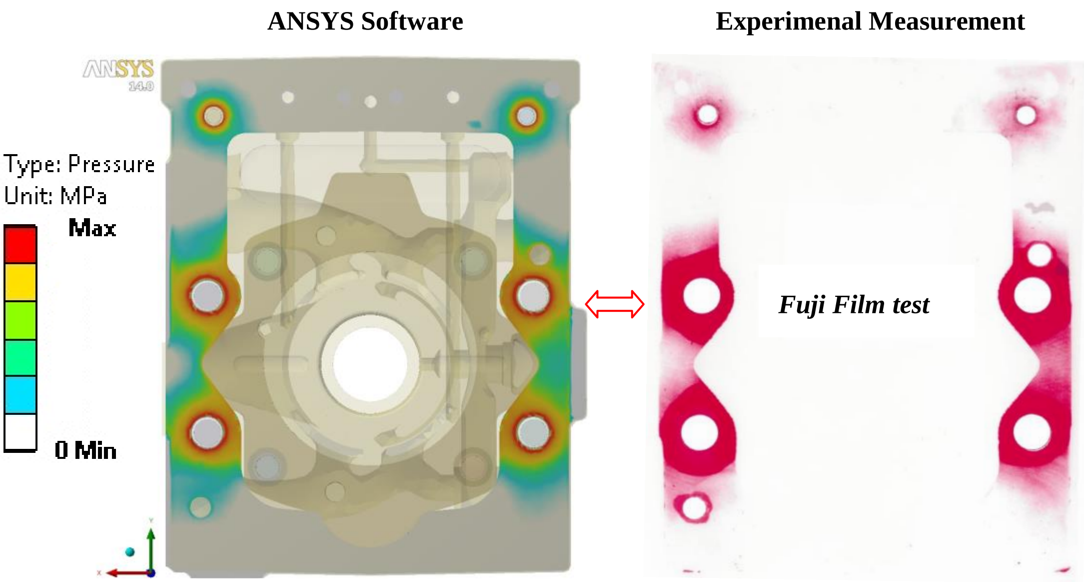

Fig. 4 Experimental validation of the FEA model

3.2 Stiffness study

The main focus of the results evaluation, of the analysis without a gasket was aimed at observation of the contact gap between the Housing and the EndCap. As it was previously mentioned the biggest influence of this parameter is the stiffness of the Housing and EndCap itself. This physical phenomena was studied under operational loads of the hydrostatic unit, where the pressure was applied into two steps. This was done because the analyzed hydrostatic unit can work in normal and reverse mode. Firstly, pressure was applied to the Core A (high pressure) and inside the Housing (case pressure) besides the others such as Servo, Gallery pressure etc. (see Fig. 5) and secondly, pressure was applied to the Core B (high pressure) and inside the Housing also besides other applied pressures (see Fig. 6).

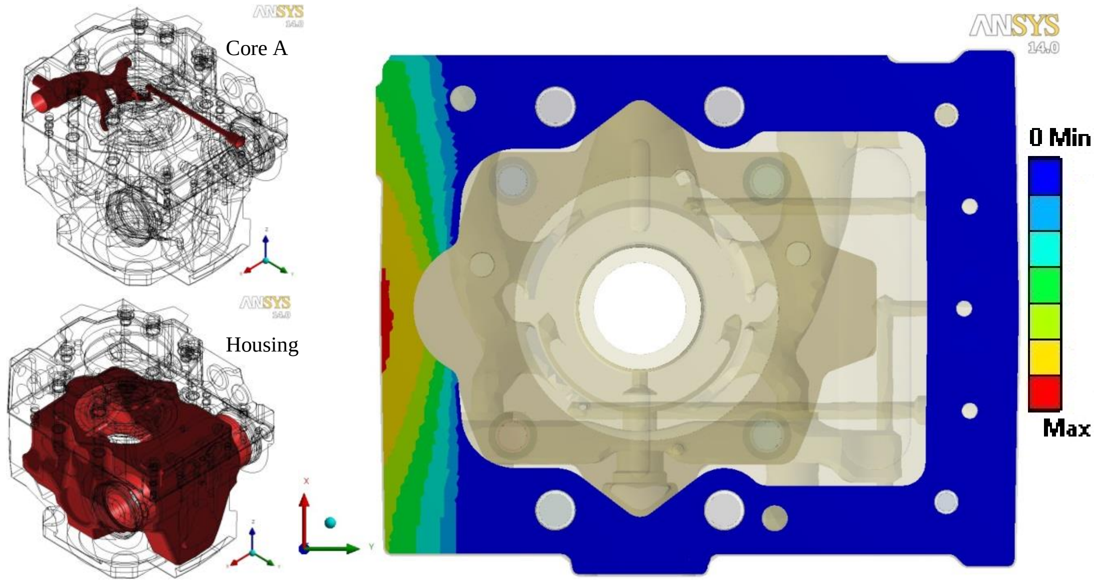

Fig. 5 Contact gap [mm] - High pressure Core A (ANSYS Software)

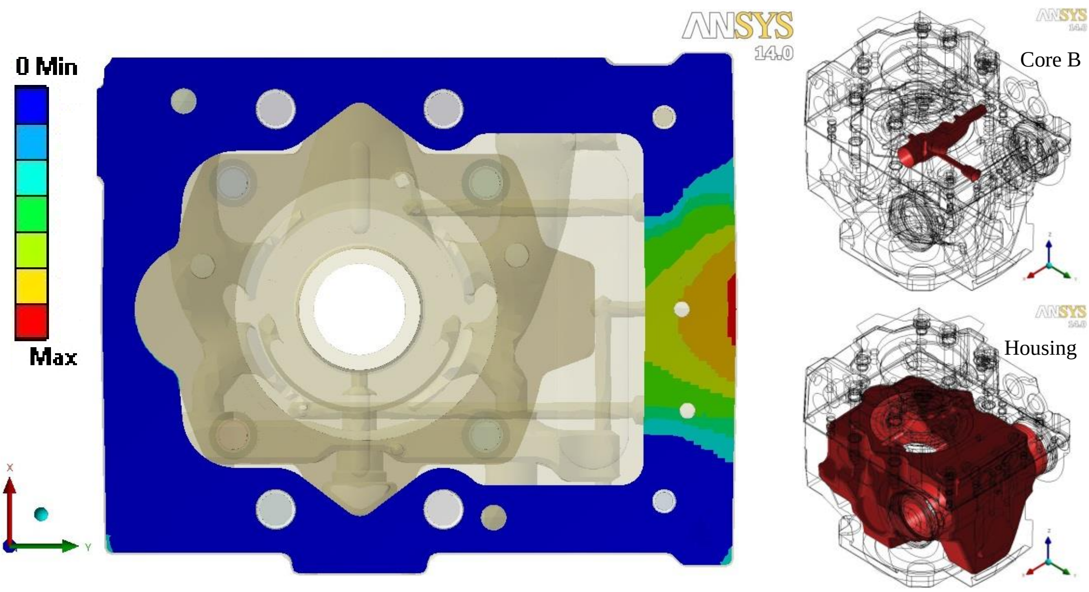

Fig. 6 Contact gap [mm] - High pressure Core B (ANSYS Software)

From the previous figures it's obvious that high pressure causes the gap between clamped flanges. This indicate low stiffness one of the clamped parts. After studying deformation and displacements on the analyzed model, the conclusion was that the EndCap has lower stiffness then the Housing, what is shown on the Fig. 7.

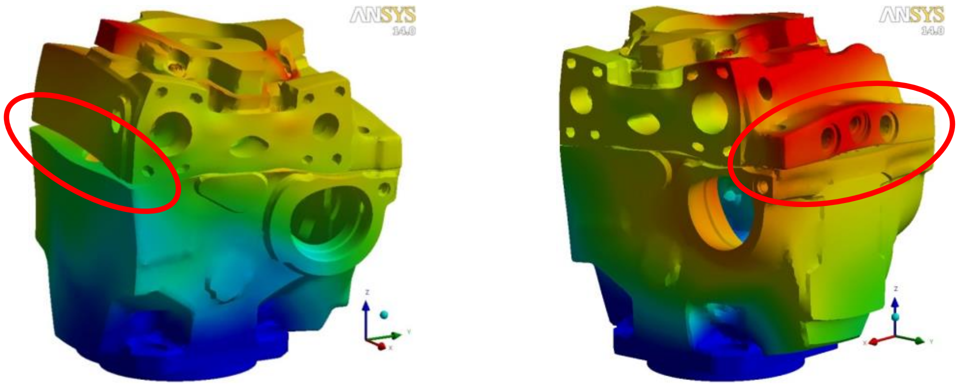

Fig. 7 Deformed shape (scaled 810x, ANSYS Software)

3.1 Contact pressure and gap study in FEA with steel gasket

In this numerical analysis, the same FEM model was used but steel gasket was added into the assembly and a few additional commands were defined. Gasket made from steel was represented by the shell elements with corresponding shell thickness. The most important task was creating the mapped mesh on gasket beads to obtain the most accurate results. Material properties were represented by using bilinear plastic material model with isotropic hardening.

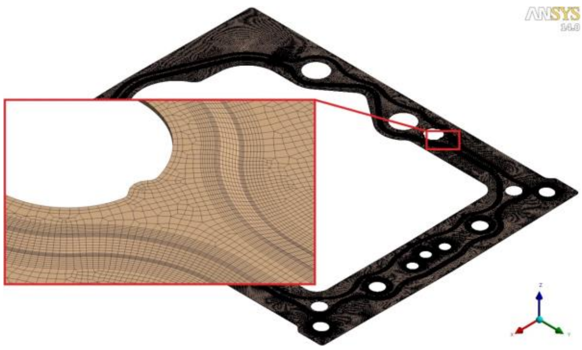

Fig. 8 Meshed gasket - Bilinear plastic material model (ANSYS Software)

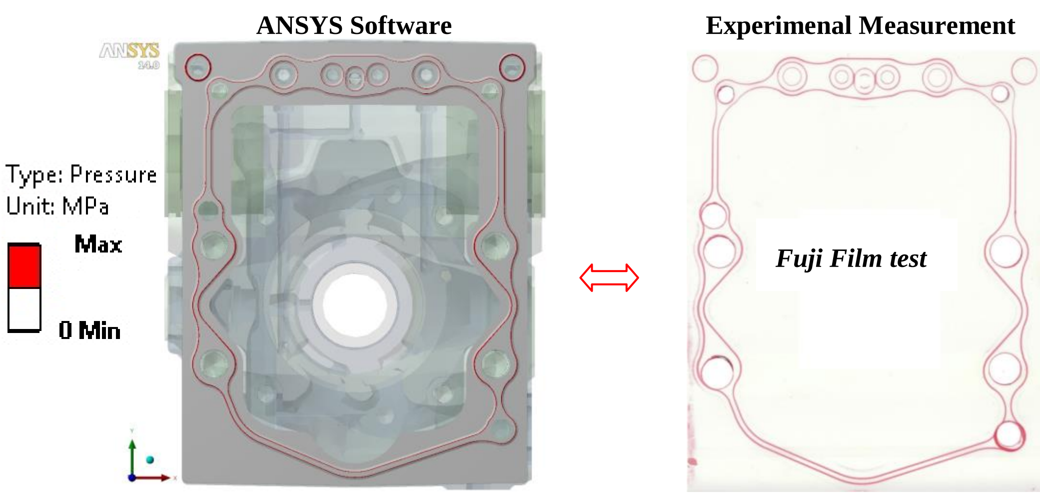

Fig. 9 Experimental validation of the gasket material model

In the Figure 9 is shown verification of the material model which represented gasket behavior. Comparison was done after tightening four bolts M14 and two bots M8 without applying operational loads. Fuji Film pre-scale was used with pressure range 130 - 300 [MPa]. In order to compare the results, the range of color bar in numerical results had to be redefined to the same range. After this comparison it was proved that used material model of the gasket represented it's behavior correctly.

The next step was evaluating of the contact pressure and gap distribution between gasket and two flanges. This was done in three steps - after tightening all six bolts and after applying high pressure into the Core A and to the Core B afterwards. This is shown in the following pictures.

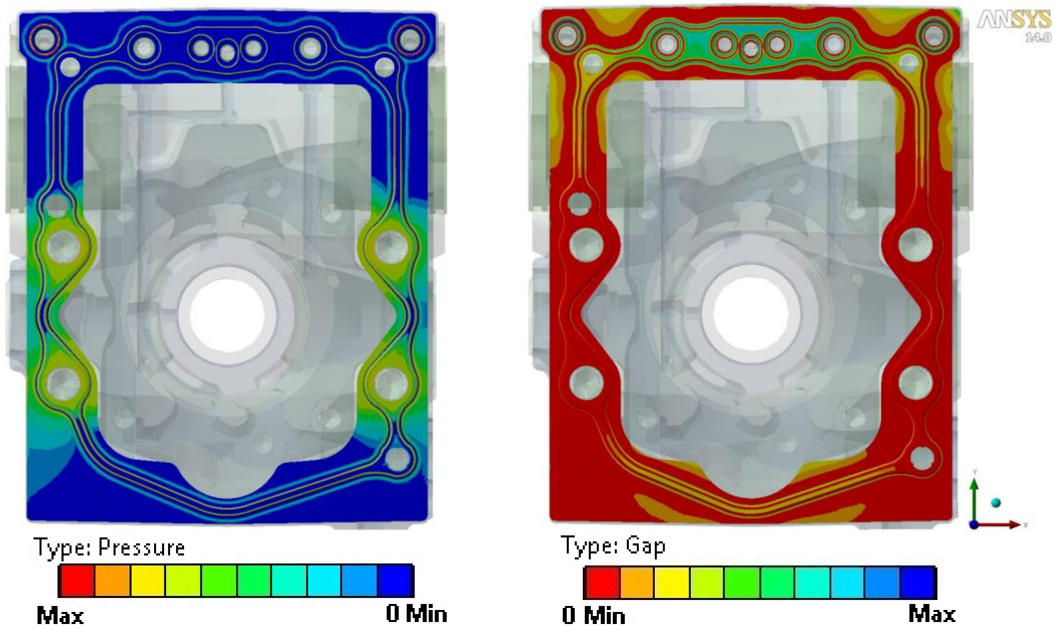

Fig. 10 All six bolts tightened

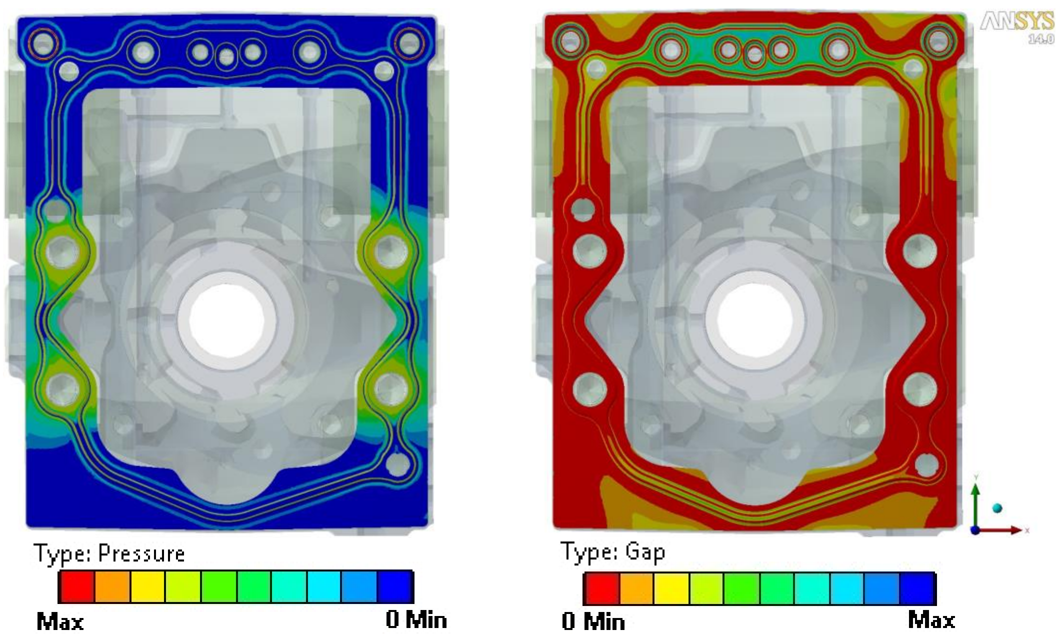

Fig. 11 High pressure Core A

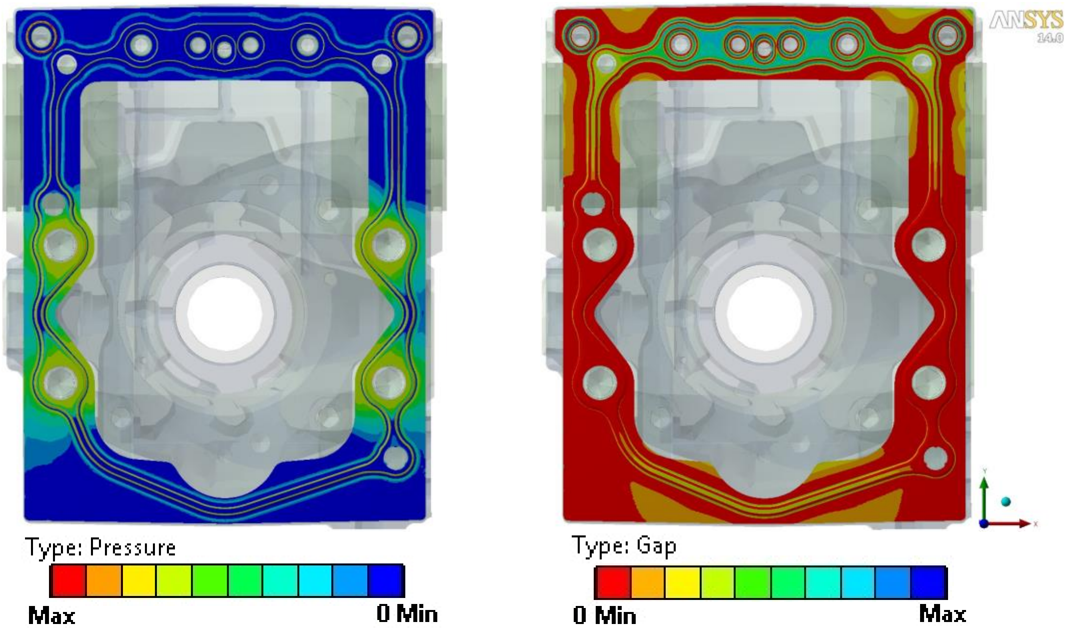

Fig. 12 High pressure Core B

It is necessary to note that from the previous figures (distribution of the contact pressure) it is obvious to see the continuous "sealing lines" which secure the beads of the gasket. Two kinds of sealing lines were created, the first one represents high contact pressure (thin yellow color) and the second one represents low contact pressure (light blue color). Zero pressure is represented by dark blue. Proper distribution of the pressure is around the M14 bolts. Deeper study of the light blue sealing line makes it possible to see the change in this colors intensity, which is caused by deformation of the End cap. This is proved by studying the contact gap in the bolted joint.

A problematic area is noted between the M8 bolts. Two factors cause this gap, low stiffness of the EndCap and high stiffness of the gasket beads in this area. From the study of the flange stiffness it was found that the EndCap has low stiffness in the marked areas (see Fig. 5 & 6) but an important note is that the high pressure in Core A (Fig. 5) does not cause insufficient sealing properties of the bolted joint in the marked area.

4 CONCLUSIONS

Based on the results obtained from the numerical analyses after one cycle of loading, the area of the bolted joint between the M8 bolts is observed as being potentially problematic. The problem arises from the following. The analyzed hydrostatic unit works in normal and in reverse mode, that means the gasket is under cyclic loading which leads to its cyclic plastic deformation. In that case a steel gasket can lose it's required sealing properties and therefore hydraulic fluid which is under pressure can enter into the space between the gasket and the flanges where the contact pressure is equal to zero. Further compressing and opening of the flange's in this area would cause the hydraulic fluid to leak out of the joint. The described problem was proved on a real hydrostatic pump by identifying the visible area of the fluid leakage.

In conclusion, after obtaining the above results, it is strongly recommended to increase the stiffness of the EndCap between the M8 bolts, and also change the position of the gasket beads in the problematic area. It is necessary to also mention that to make the EndCap stiffer, can be a difficult task due to its hydraulic connections. Once these changes have been made, further numerical tests should be carried out to see the improvement.

In the future, there is a huge possibility to apply probabilistic approach, for example see [6].

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