STRUCTURAL INTEGRITY DESIGN AND VERIFICATION FOR A SYSTEM HAVING A HEAT PUMP FUNCTION

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SUMMARY: This paper is the first one of a series of research work where will be identified an optimal topology, from a lifetime, cost and thermodynamic point of view, for a device with a principal role of a heat pump. The device itself has a construction similar to hydraulics cylinders. In this paper is presented the general design of the sytem, a work cycle and the first stage of designing, optimising and verifying the integrity of the piston of the first version of the device. The optimisation is realised in Ansys code and consists in a parametric optimisation. The objective function is the reducing of mass, the restrictions consisting in a minimum lifetime of 36600 cycles and a safety coeficient of stability equal or higher than four.

KEY WORDS : Ansys, parametric optimisation, F.E.A., heat pump

1. Introduction

The general subject of this paper is represented by the first stage of optimising the piston of a device with a general function of heat pump, but it is also presented the general design of the device and a work cycle. The purpose of this research is to obtain an optimal topology of the device piston from the mass point of view. This was performed realizing a paramteric optimisation in Ansys code, which has an objective function of reducing the mass, and two restrictions: a minimum lifetime of 36600 cycles and a safety factor of stability equal of higher than four.

2. The general design of the device

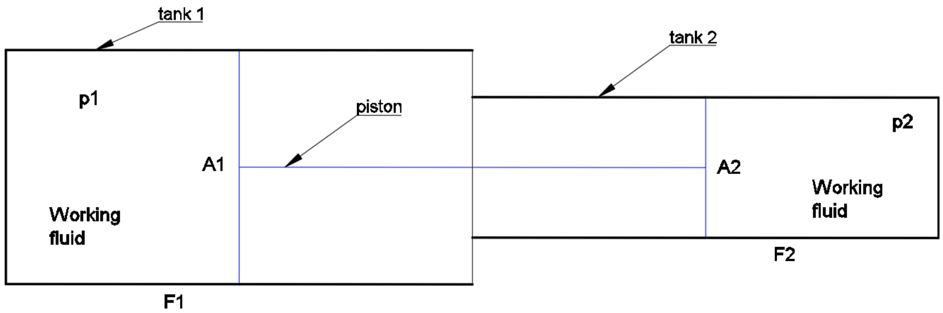

Fig. 1. Sketch of the general design (A – area, p – pressure fluid, F – force)

In Fig. 1. are represented the principal components of the device: two tanks with different diameters, each one of them having a lid who can slide in them. The two lids are rigidized one to another by an element which will be called core. This three part assembly will be called piston. The working fluid is a substance that will constantly have two aggregation states (the gaseous one and the liquid one). This substance is on the liquid curve. In this moment the device is designed and optimised to have carbon dioxide as an working fluid, but once will be idetified an optimal topology for the device, it will be possible to find a working fluid with superior thermodynamic propreties.

3. The work cycle

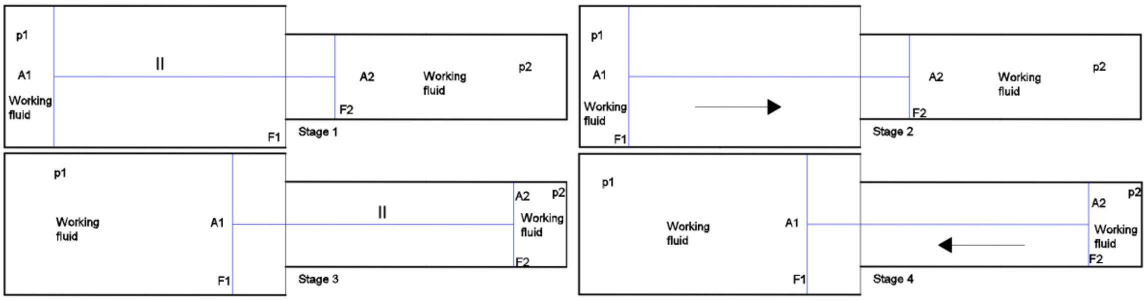

Fig. 2. Four figures collage representing the main stages of the work cycle

In Fig. 2. are represented the stages of the work cycle. Each assembly of tank+lid type is characterized by a pressure and an area, therefore a force. In the first stage, the piston is maintained in the initial position by an external action. In this stage the pressure from the two tanks are equal and defined by the ambiental temperature. The second stage starts once the piston is released, because of the equal pressures of the tanks but also the different areas of the lids a dynamic imbalance occurs witch is producing a movement of the piston in the sens of compression the working fluid from the second tank. This compression is generating the variation of enthalpy on the liquid curve of the carbon dioxide, witch is translated by a heat release. After a period, the piston will stop, this represents the third stage. In the forth stage the piston is returned, by an external action, to its initial place, with the purpose of starting the next cycle.

4. The device topologies that will be designed, optimised and verified to find the best option

In this series of research, to identify a suitable topology for the functional role of such a system and the loads to which is subjected, there will be designed, optimised and verified a variety of topologies.

A topology of system includes predominantly the design and the structural verification of the tank and the piston (including its lids).

Topology classification criteria:

1) By the type of the main degree of freedom:
   a. System with the main degree of freedom of the translation type;
   b. System with the main degree of freedom of the rotation type.
2) According to the perimeter shape of the piston ends (criterion valid only for systems with the main degree of freedom of translation type)
   a. System with circular perimeter of the piston ends;
   b. System with stepped shape of the piston ends.
3) By the number of stiffeners placed between the piston ends (criterion valid only for systems with the main degree of freedom of translation type)
   a. System with 16 stiffeners placed between the piston ends;
   b. System with 12 stiffeners placed between the piston ends;
   c. System with eight stiffeners placed between the piston ends;
   d. System with four stiffeners placed between the piston ends;
   e. System with one stiffener placed between the piston ends (the stiffener is continue on all the circumference of the end of the piston).
4) By the presence of a stiffener plate positioned on the middle of the piston (criterion valid only for systems with the main degree of freedom of translation type)
   a. System with a stiffener plate positioned on the middle of the piston;

b. System without a stiffener plate positioned on the middle of the piston.

In the followings, not all combinations that can result from the upper criterions will be studied. The study will start from one of the possible topologies, after this, using the initial configuration will be compared al the posibilities present at the forth criterion. With the optimal version, will be analised in the same manner all posibilities from the third criterion and the procedure will repeat until it will be identified an overall optimal topology.

5. First topology – general elements

The first topology, according to the criterias mentioned above, is a system with main degree of freedom of the translation type, with circular perimetral shape of the ends of the piston, with 16 stiffeners between the ends of the piston and without a stiffener plate positioned on the middle of the piston.

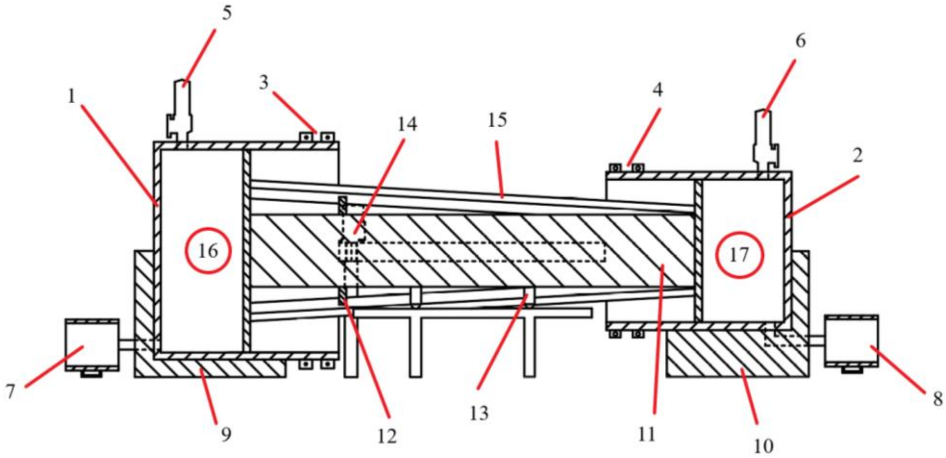

Fig. 3. Sketch of the heat pump with the identification of the main components

Legend: 1, 2 – principal tanks, 3, 4 – stiffening collars of the main tanks, 5, 6 – safety valves, 7, 8 – auxiliary tanks featured with pumps and sense vales, 9, 10 – grounding system, 11 – piston, 12 – stopping element, 13 – the system of ensuring the orientation of the device, represented by guideways (roller supports), 14 – gear electromotor with rack and pinion, 15 – stiffeners of the ends of the piston, 16, 17 – working fluid (represented by carbon dioxide)

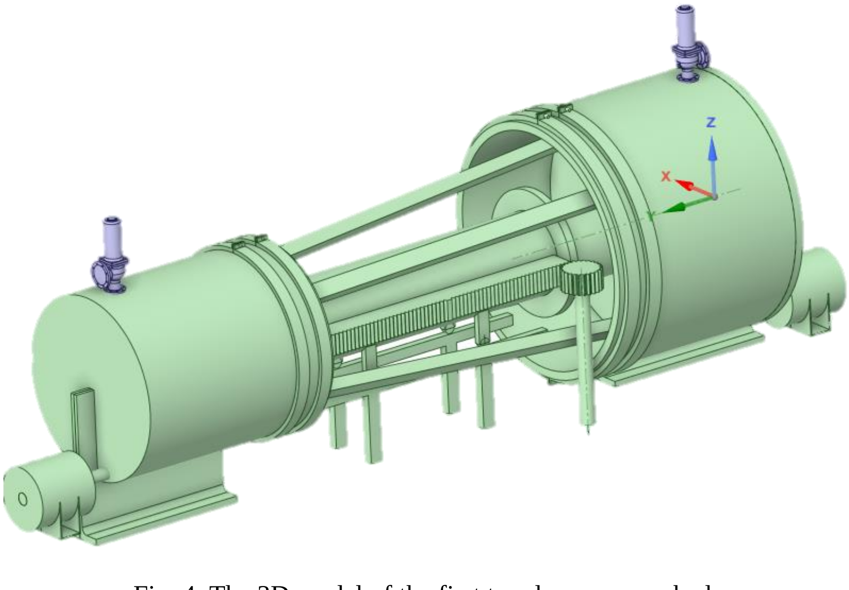

Fig. 4. The 3D model of the first topology approached

6. The optimisation and structural verification of the piston of the first topology

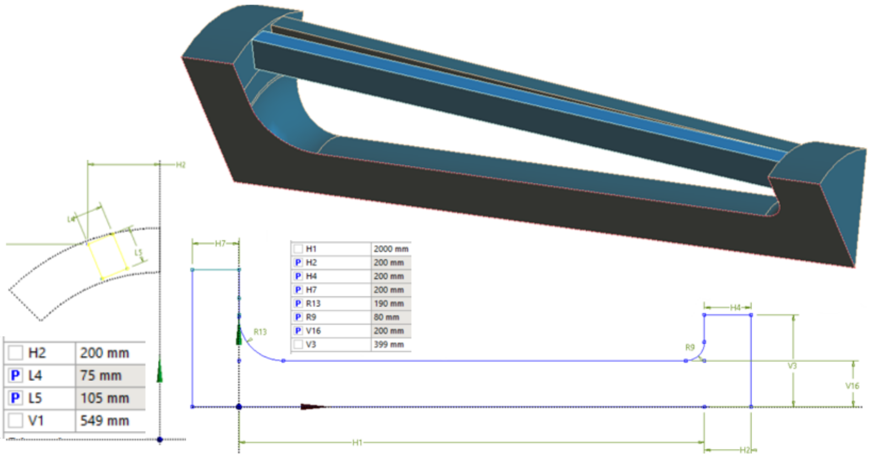

Fig. 5. Collage representing the geometry used to define the finite element model ()

The geometry was made in Design Modeler and not in Space Claim. The advantages of Design modeler are not only the fact that is very clear and easy to set the input parameters for the optimisation analysis (in Space Claim, some of the parameters can be dificult to set), but also, it is posible to create mathematical relations between them, which is what i did to get more control over the geometry. All the elements with a blue “P” in front of them are the input parameters for the optimisation analysis and i preset all the values that they can take during the analysis. The model is representing one eight of the whole piston.

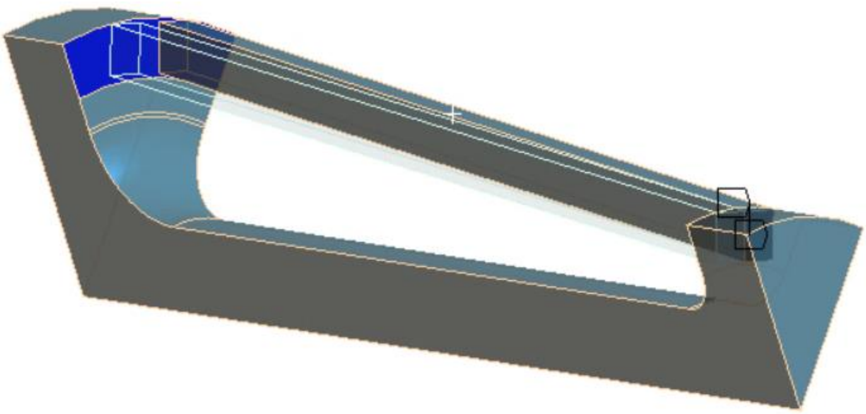

Fig. 6. One of the two defined contacts of “bonded” type, realised between the stiffeners and the edges of the piston

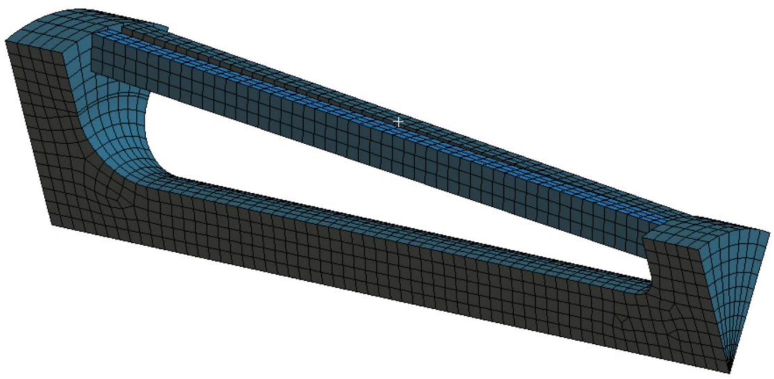

Fig. 7. The controlled (in local and global sizing, mesh method and element order) mesh of the model

The global elelemt sizing is also parametrised, the reasoning will be explained later in this paper.

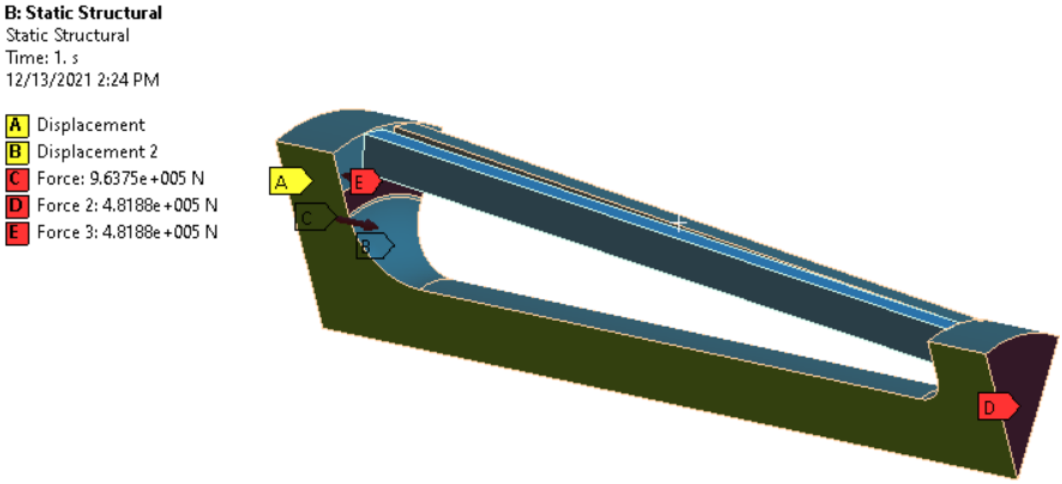

Fig. 8. The boundary conditions (including the loads that appear on the structure)

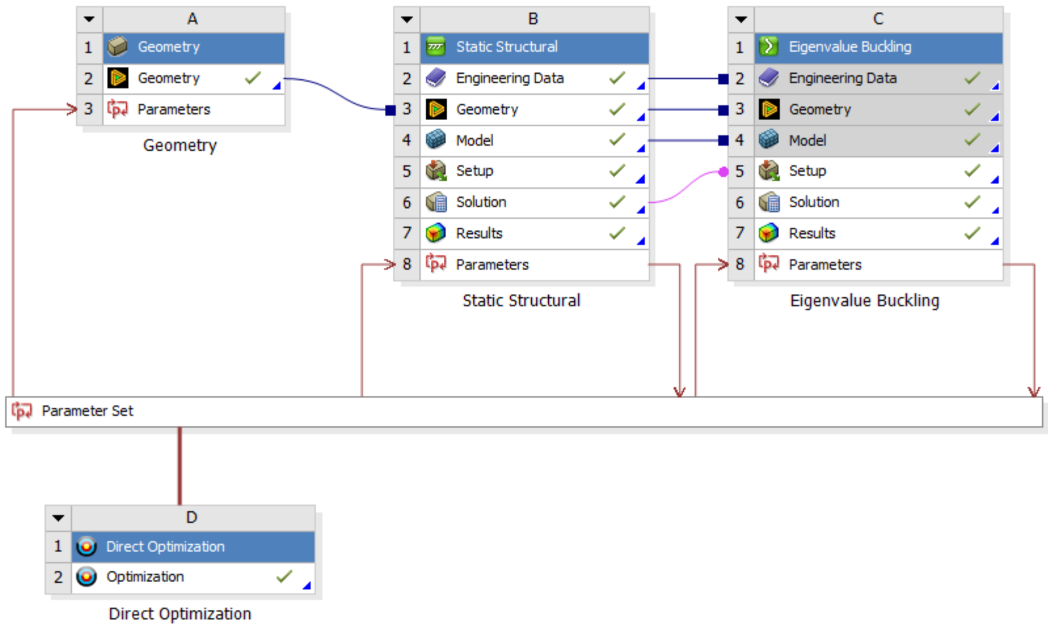

Fig. 9. The optimisation project, including a static model with fatigue calculus and a stability calculus module

The objective function of the optimisation was mass reduction, and it has two restrictions: a minimum lifetime of 36600 cycles and a safety factor of stability equal of higher than four.

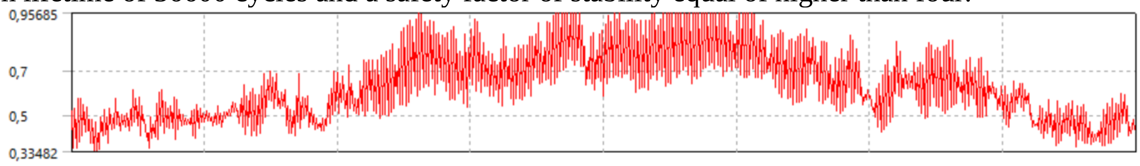

Fig. 10. The graphic representation of the fatigue cycle

The fatigue cycle was defined following the next described procedure. there were obtained daily temperature data with minimum and maximum values for a period of one year from a meteorologic station [3]. There were also imported point from the liquid curve of the carbon dioxide [4]. Because I did not have a continous function to describe the liquid curve I made a program using “C++” language

(Program 1). This program created a one degree polynomial curve between any two consecutive points who are on the liquid curve, after this the program exported as a file with the almost exact pressure that the tank will support at the every temperature received from the meteorological station. The resulted cycle can be saw in the previous figure. It is true that is not possible to know in this stage of the research the aproximative working cycle, but this is not extremely relevant because in this phase only a rought aproximation is needed so, the topologies in the scope of choosing the optimal one, were compared, version which will be further subjected to more realistic working cycles.

For the study of the lifetime it was used the Soderberg mean stress correction theory, which is the most restrictive from Ansys code and its returning a zero lifetime for any zone that have stress over the yield stress.

The optimisation method used was M.O.G.A. and it converged after computing 370 design points. The optimisation had eight input parameters with a total number of permutations of 0.1 bilions which was reduced by the first two inequations that can be seen in the following figure, inequations that also ensure that all the design point will have a geometry that can be generated. The last two inequations that can be saw also in the following figure represent a control in the global element sizing mesh. This is the reason for which it is considered a parameter. Controlling this global element mesh sizing one could obtain design points with simillar accuracy of the resulted stress and also similar accuracy of the loads multiplying coeficient needed to result in an unstable structure. All of the anterior mentioned advantages in the smallest computing time.

P20 <= (200[mm]-P22)+80[mm]	P20	<=	(200[mm]-P22)+80[mm]
P21 <= (200[mm]-P22)+190[mm]	P21	<=	(200[mm]-P22)+190[mm]
1	(P20*P20*(1-3.14/4)*2*3.14*(P20+P22)/2.37+P21*P21*(1-3.14/4)*2*3.14*(P21+P22)/2.37+564.2[mm]*564.2[mm]*3.14*P19+399[mm]*399[mm]*3.14*P17+2000[mm]*3.14*P22*P22)/(8*9000/4)	>=	P33*P33*P33
2	(P20*P20*(1-3.14/4)*2*3.14*(P20+P22)/2.37+P21*P21*(1-3.14/4)*2*3.14*(P21+P22)/2.37+564.2[mm]*564.2[mm]*3.14*P19+399[mm]*399[mm]*3.14*P17+2000[mm]*3.14*P22*P22)/(8*15000/4)	<=	P33*P33*P33

Fig. 11. The constraining inequations used in the optimisation

7. The results of the first phase of the optimisation of the piston of the first designed topology

The results of the Ansys optimisation are presented tabular, following this paragraph, being observed a mass reduction of 70%, percentage not relevant since I intentionally choosed a robust initial model. Other remark consists in the fact that the resulting section of the stffeners of the ends of the piston are similar to the ones obtained analitycally. A final note can be that it is clear that the model can be further optimized. The code in parentheses are for easy understand the mentioned dimmension in Fig. 5.

Table 1. Results from the analysis computed on the initial and on the optimised model

Parameter name	The value for the initial model	The value for the optimised model
Tank two - diameter lid thickness (H4)	200 mm	80 mm
Tank one - diameter lid thickness (H7)	200 mm	80 mm
Tank one – stiffening radius (R13)	190 mm	190 mm
Tank two – stiffening radius (R9)	80 mm	160 mm
Core radius (V16)	200 mm	80 mm
Stiffeners of the lids section width (L4)	75 mm	45 mm
Stiffeners of the lids section lenght (L5)	105 mm	55 mm
Minimum lifetime	2.75e6 cycles	2.75e6 cycles
Equivalent von Mises stress	24,73 MPa	98,54 MPa
Multipliyng loads coeficient	57,25	5,84
One eight of the resulting structure mass	823,15 kg	253,28 kg

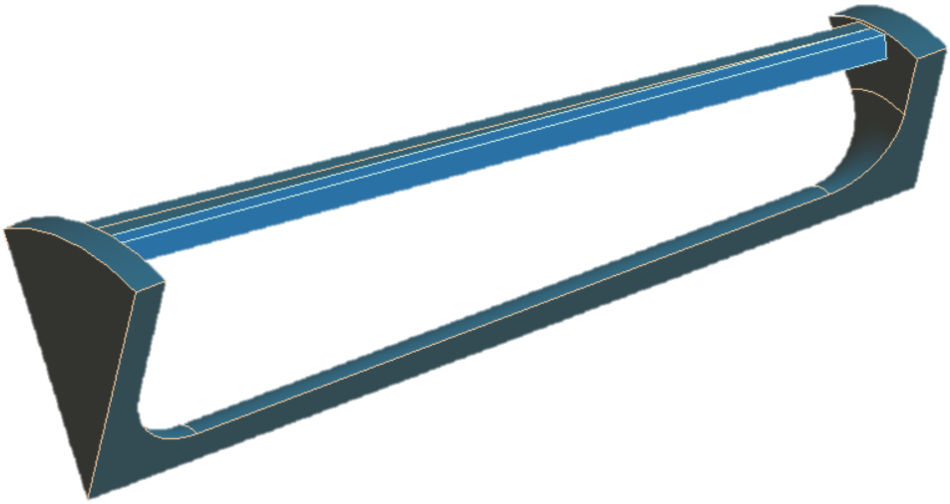

Fig. 12. The geometry of the resulted optimised model

8. Conclusions

In this paper, the first one of a series, the author presented the general design of the device that was conceived and created to be used as a heat pump that uses the ambiental temperature and energy to fulfill this fucntional purpose. It was also presented the work cycle and how this device is functioning. There were also reviewed the topologies that will be compared in time in the scope to identify the optimal one of them.

The design and the first part of the first technology structural piston verification and optimisation was also presented. The optimisation was realised using the parametric optimisation module in Ansys. The optimisation analysis had an objective function of reducing the mass, and two restrictions: a minimum lifetime of 36600 cycles and a safety factor of stability equal of higher than four. The mass reduction between the initial model and the final one was 70%, percentage which is not very relevant fot the moment.

On short term, the next steps will be in the direction of a further optimisation, a manual one this time, of this piston. As it has been mentioned before, from the results table it is easy to observe that the model can be further optimised. In this direction the author already started making some steps. He defined a new program (Program 2) using “C++” language. This program has as the input data all the design point calculated by the optimisation analysis made in Ansys. The program itself identifies all design points who are different by one single input parameter and then is defening a scalar that is representing the ratio between the variation of the mass of the system and the variation of the multipliyng loads coeficient to make the structure unstable, after this for all the ratios generatated by the same input parameter is defined the arithmetic average, which is a measurment of the sensibility. Using all the data generated by this program i will manually optimise three of the best candidates (with different construction) of the optimisation made in Ansys plus one chosen by me. This will represent the second phase of optimising the piston.

On long term the author will apply the same procedure on different tanks and, after this step, on other topologies with the scope of identification of the optimal verison for these devices.

9. Bibliography

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