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EXPERIMENTAL PROBLEMS OF NEW CONSTRUCTIONS OF PORTABLE SUBMERSIBLE PUMPS FOR MINING WITH ELECTRIC MOTOR COOLED BY WATER JACKET

EXPERIMENTÁLNÍ PROBLÉMY NOVÝCH KONSTRUKCÍ PŘENOSNÝCH PONORNÝCH ČERPADEL PRO DŮLNÍ TĚŽBU S ELEKTRICKÝM MOTOREM CHLAZENÝM VODNÍM PLÁŠTĚM

Abstract

The article presents a new construction of a submersible pumping engine where new original solutions in flowing system and control system where introduced It let obtain high efficiency and fulfill user's expectations. The article also contains experiments of the new pumping engine and their results. The results of numerical analysis of the movement of liquid in flowing channel is described and the analytical characteristics are compared with ones measured in laboratory. The work also presents the problems with constructing, research and certification of new submersible pumping engine which construction fulfils ATEX requirements for machines working in explosive conditions areas

Abstrakt

Článek představuje novou konstrukci ponorného čerpadla, kde byly představeny nové originální řešení v proudícím systému a řídicím systému. Toto řešení dovoluje dosáhnout vysoké účinnosti a plní uživatelské očekávání. Článek obsahuje experimenty nového čerpadla a jejich výsledky. Je popsán výsledek numerické analýzy pohybu kapaliny v průtokovém kanálu a porovnány analytické vlastnosti s měřením v laboratoři. Práce také představuje problémy s konstrukcí, výzkumem a certifikací nového ponorného čerpadla, který konstrukčně splňuje ATEX požadavky pro stroje pracující ve výbušných podmínkách.

Keywords

Construction, prototype, research, implementation

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1 INTRODUCTION

Submersible mine pumping engines known as submersible mine pumps have electric engine and electronic control system and they must fulfill restrictional ATEX requirements. The described pumping engine was constructed as a plant of the group of first category M2 flame-proof structure. Fig. 1 shows intersection of the pumping engine where a new construction of pulling out power lead [1] and innovative flowing system [2] [3] were used.

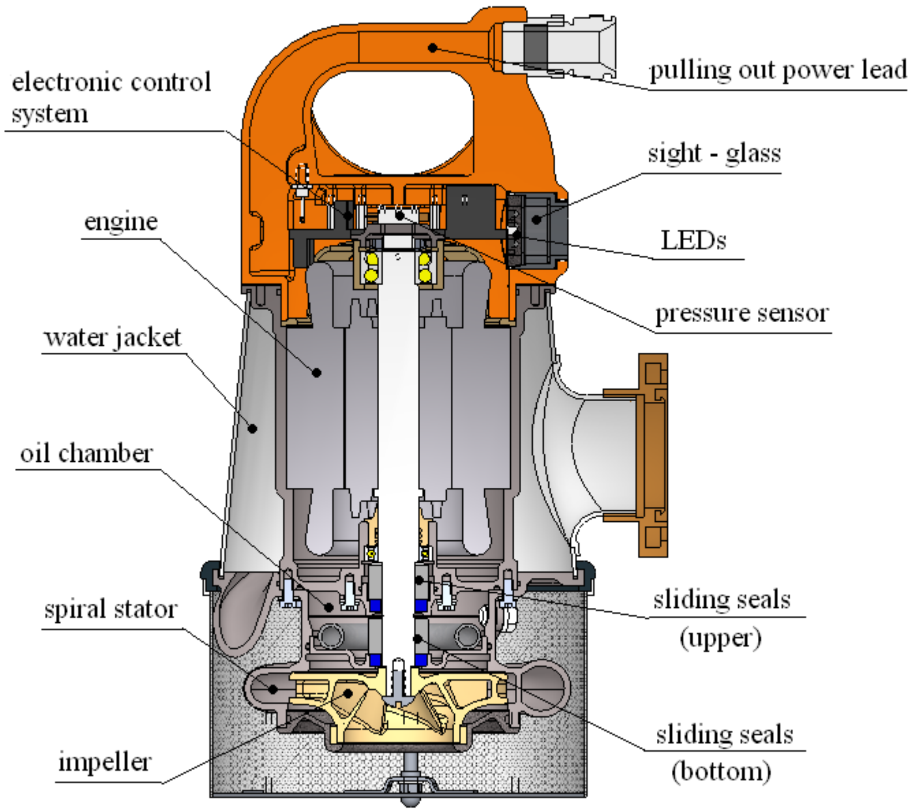

Fig. 1. Submersible mine pumping engine

2 CONSTRUCTION OF THE SUBMERSIBLE MINE PUMPING ENGINE

Using water jacket so cooling the engine with pumping water lets significantly increase its power over the rated power described while working in the area of 25°C. It forced to use stator as element of carrying away pumping liquid from the impeller to the water jacket. The pump of new construction has innovative flowing system [2], [3], thanks to which it gains higher efficiency in comparison with known constructions of the same type. Fig. 2 shows parts of flowing system

a) impeller

b) spiral stator

c) water jacket.

Fig. 3 shows the basic characteristic. The broken line shows efficiency of the pump. The full line shows efficiency of the pumping engine. Fig. 4 shows the test stand of the pump of new construction.

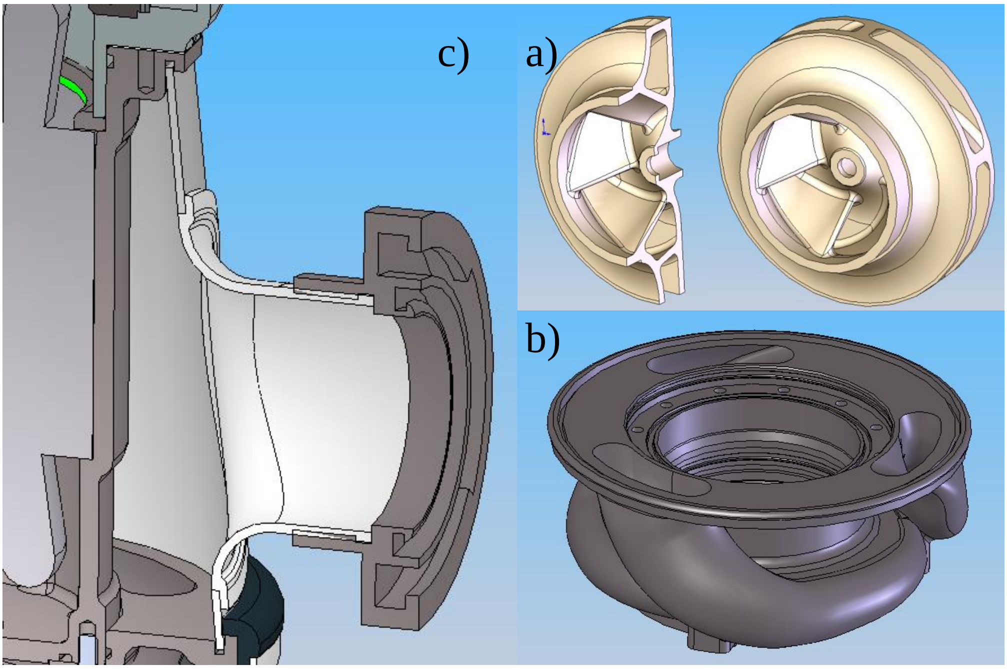

Fig. **2**. Parts of the flowing system in the submersible pumping engine.

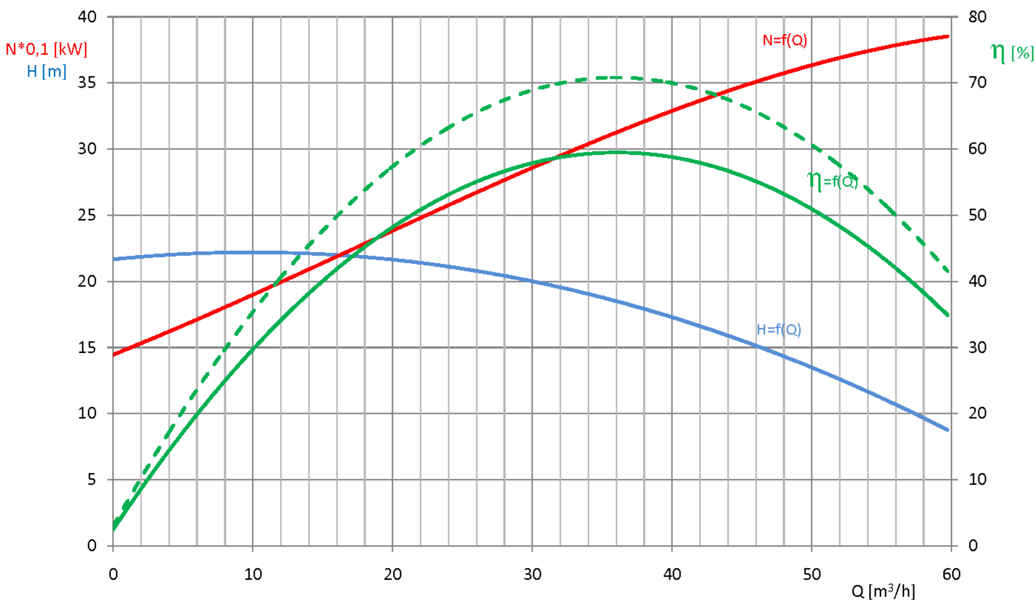

Fig. **3**. Characteristics of submersible mine pumping engine..

Fig. **4**. The test stand of the pump of new construction.

The submersible mine pumping engine is called mine face pump because of the place where it is used. As a pump draining mine face it has not immovable place of work. The pump is repeatedly installed in different places where it works in different conditions. Changing conditions of work cause that the pump is endandered by

a) wrong phase sequence on the side of electronic supply of the engine

b) dry running which is connected with the fact that the engine works with low power

c) overload of the engine because of pumping liquid with too high density which consequence is excessive current consumption

d) increase of the temperature of the engine which can also be caused by pumping too hot water

e) increase of the temperature of bearing that can be caused by excessive axial force or mistakes made during assembly

f) failure of bottom sliding seal letting the pumping liquid get to the oil chamber and causing danger of damaging upper seal

The user expects that the pumping engine will have protection system against mentioned dangerous situations. Innovative protection system against the damage of bottom sliding seal is used in the pump [4]. The pumping engine has electronic protecting system and control system against the danger.

3 ELECTRONIC CONTROL SYSTEM [6]

All the dangerous situations mentioned above are mosty connected with the engine driving the submersible pump. The consequences can be counteracted by turning off the engine. In the presented project a solid-state relay was used to turn on and turn off the engine. This SSR is controlled by microprocessor connected with sensor signaling mentioned danger or informing about operating conditions of the pumping engine. Fig. 5 shows the block diagram of the electronic control system.

Sensors transmitting signals that identify typical danger are used in the pump. Depending on the danger group of sensors placed in sight-glass display proper signal describing the condition of the pumping engine. Correctness of working of the electronic control system was checked with concurrent measurement of its parameters of work. There were tests where the control system was outside the pumping engine. The determinant of the correct construction of the electronic control system is the temperature of work of the plate (PCB). Fig. 6 shows tests stand of the control system.

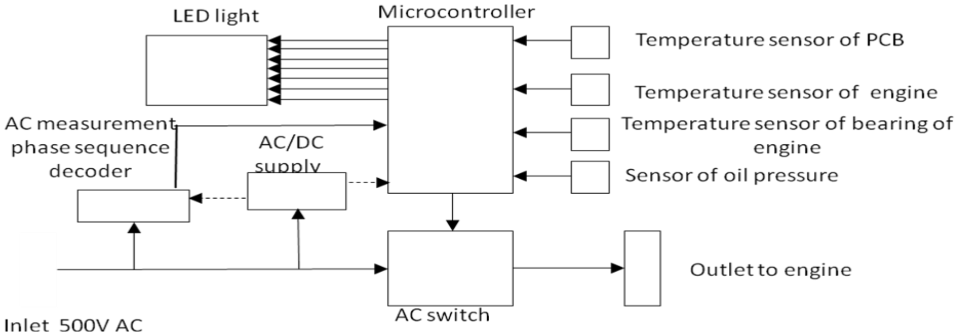

Fig. **5**. Block diagram of the electronic control system.

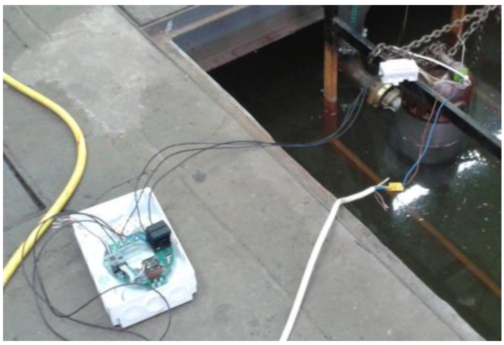

Fig. **6**. The test stand of the electronic control system.

The maximal measured temperature of work of the control system was 85°C. The control system worked correctly with the pump when it was outside it. In spite of it it was not possible to carry out tests when the control system was placed in the pumping engine. It was caused by fire in the area of engine block and cover during start-up. These events are being explained in cooperation with the producer of the electronic control system and the producer of engines. Fig. 7 shows the pump after the fire.

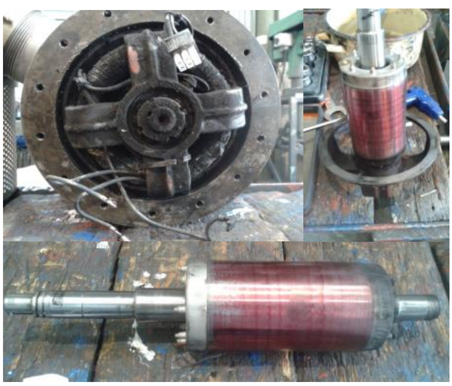

Fig. **7**. Parts of the electronic control system damaged by the fire.

4 STAND TESTS

The tests of the pumping engine were carried out on a lab test stand shown on fig. 4. The testes were carried out using pure water at the temperature of 19°C and rotational speed of 2860 rotations/minute. A series of investigation were made with the pumping engine using the impeller of maximal diameter of 135 mm and trimming blades. The trimmings were made for the impeller of diameters from 135 to 110 mm. The pump of the construction for series production gains maximal efficiency of ŋ=71%. Considering entrance loss to discharge branch which at the nominal efficiency ($36m^3/h$) is about 3%, the efficiency gained by the pump is consistent with requirements given in world literature [5]. This efficiency is also about 10% higher than the efficiency of the series of types accessible on the market now. The efficiency of the pump was calculated assuming the efficiency of the engine at about 84% which was proved by the measurement on the engine test house. Fig. 10 shows characteristics of the pump for the impellers of the diameter from 135 to 116,2 mm.

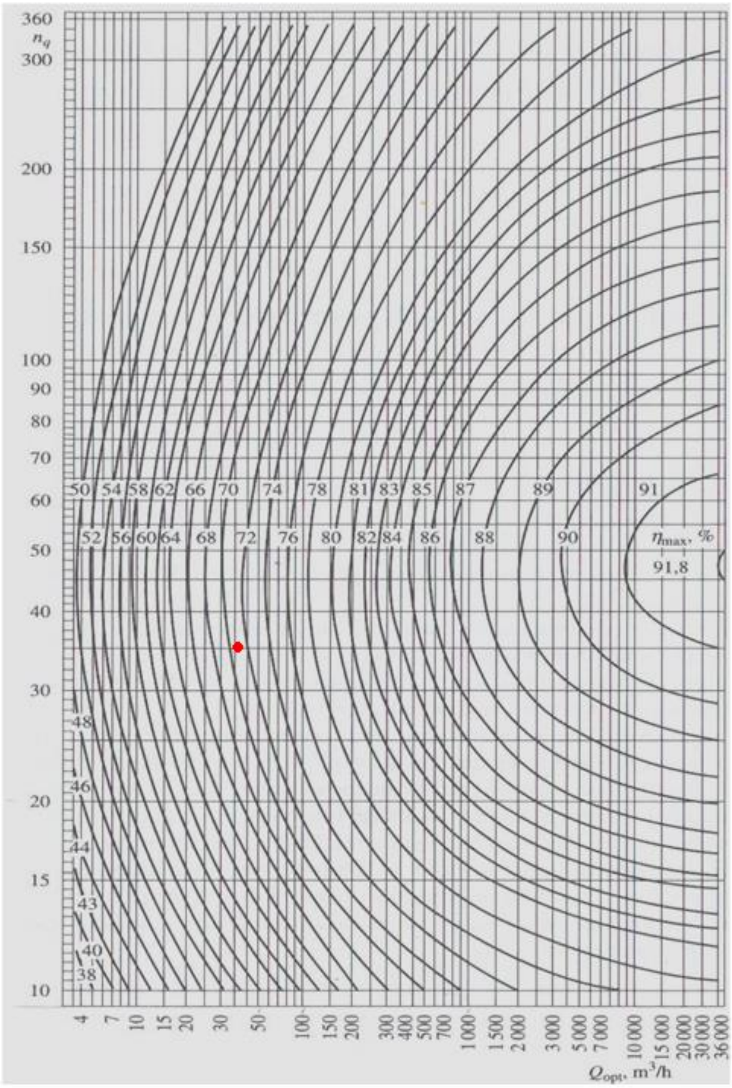

Fig. **8**. Characteristics of total efficiency in the function of capacity and shape number for single-stage pumps. The red point on the diagram shows the efficiency of the pump of new construction

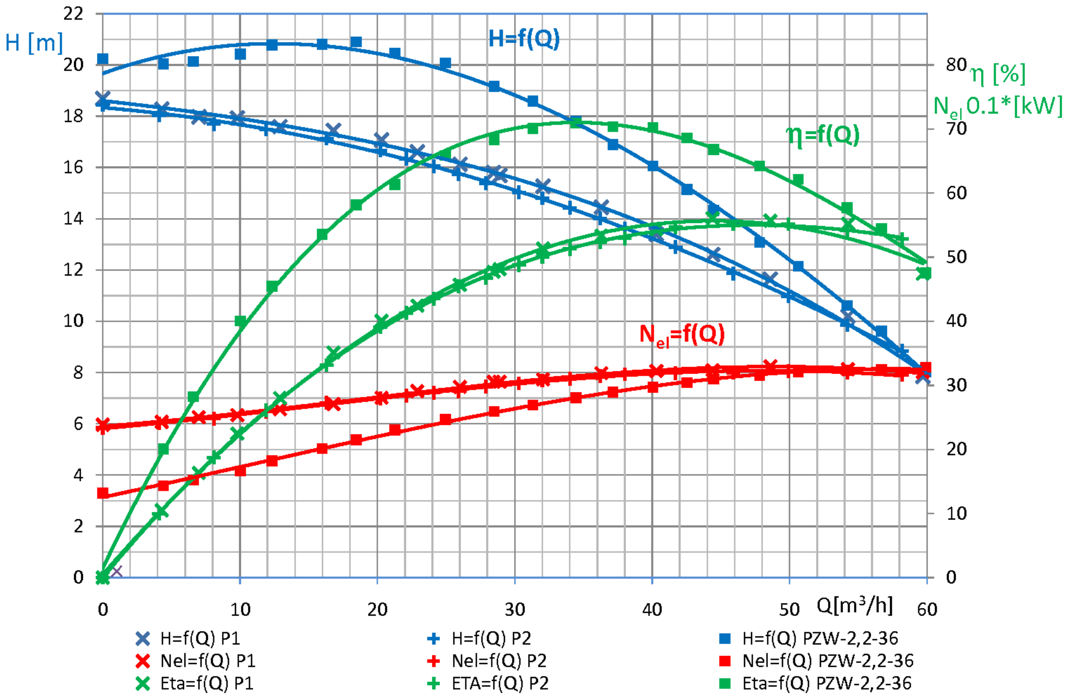

Fig. **9**. Characteristics of the pump of new construction PZW-2,2-36 (d2=133) and two examples of pumps accessible on the market.

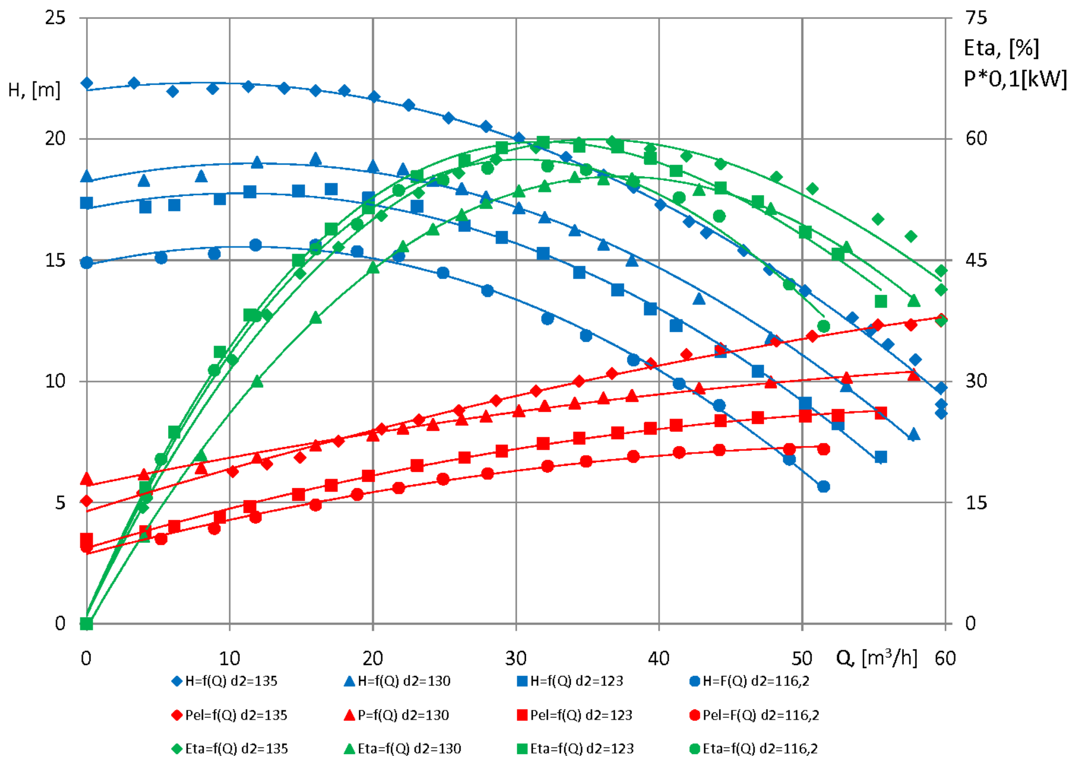

Fig. **10**.Characteristics of the pumping engine of new construction with trimming impeller

5 OIL CHAMBER

In the submersible pumping engines the electric engines are separated from the pumping liquid by the oil chamber with pair of sliding seals. The first seal (bottom seal) separates pumping liquid from the oil. The second seal (upper seal) separates the oil chamber from the area of the engine. Usually both the seals are of the same construction however because of different medium they work on, seal saces (materials of the rings making the pairs) are different. Because the first seal has contact with the liquid it will be damaged as first.

In case of bottom seal break-down the liquid flows through it to the oil chamber what leads to increase of the pressure to the value of pumping and the mixture of oil and water will make emulsion. The pump can still work because the upper seal does not let the moisture get to the engine. This way seals put in series increase the reliability of the seal system which stops functioning after the failure of both seals [7].

The pressure of the oil chamber will increase because of failure of the first seal, turing normal work because of the leak through seal saces or because of heating oil. The temperature of the oil will increase because of loss of friction in sliding seals, increase of the temperature of the shaft taking the heat from the engine, increase of the temperature of the pumping liquid.

There are two known ways of detecting the failure of the bottom seal:

a) measurement of the oil resistance in the oil chamber. This way of protection shows some difficulties with interpreting the results and the necessity of supplying electric voltage to the oil chamber. It cannot be used in the area with danger of explosion.

b) measurement of pressure in oil chamber. The pressure sensor gives signals to turn off the pumping engine after exceeding the set limiting pressure.

   To determine the limiting pressure the measurement of the pressure was carried out behind the impeller at d_2 diameter and behind the impeller in front of stable ring of the bottom seal. The results of the measurements are presented as a diagram (Fig.12) The diagram shows the limiting pressure (straight line) that the sensor was set on and the maximal pressure that can appear in the oil chamber during normal work of the pump.

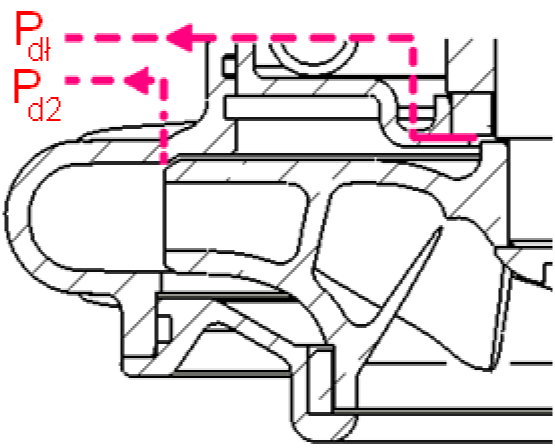

Fig. 11. Points of measuring the pressure on the full diameter of the impeller and in front of the bottom seal.

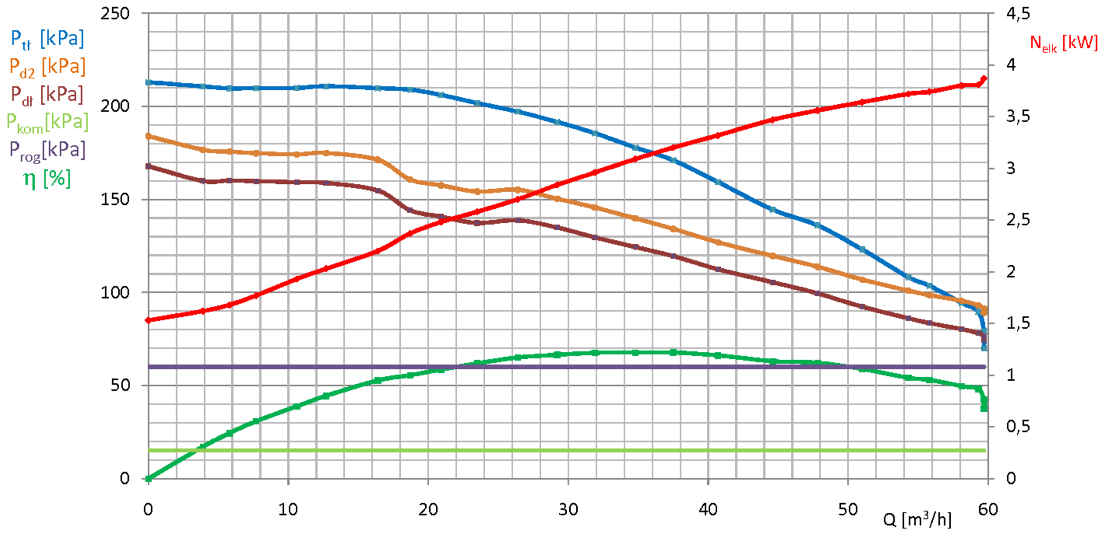

Fig. 12. Basic characteristics of pump $P_{tł}$(Q), N_{el}(Q) i Eta(Q) and course of the height of pressure

P_{d2}(Q)- on the full diameter, $P_{dł}$(Q)- in front of bottom sliding seal, P_{kom}(Q)- the level of the pressure in the oil chamber increased because of exploitation of the pump, P_{prog}(Q)- accepted increase of the pressure in the oil chamber.

Analyzing the results of pressure measurements behind the impeller it was said that set to counting axial force coefficient K of pressure distribution behind the impeller is much higher than the set during the measurements so the axial force is higher than the counted value. Repeated analysis of choice of bearing showed decrease of life of used thrust bearing but verified life is in the limits of the life for impeller pumps.

The constructional problems with using the pressure sensor in the oil chamber are

a) the choice of place where the sensor will be set and the way of leading the pressure to the sensor

b) the way of delivering the signal from the sensor to the control system

The sensor should be placed in aflame-proof space because outlet of the wiring from that area is connected with using parts described and required by ATEX regulations. They also must be of big dimensions. It is assumed that the sensor will be placed on the bottom of the engine block under the stator. The pressure from the oil chamber to the sensor will be led by a capillary tube maintaining fire clearance. It is very difficult to decide how to lead the signal from the pressure sensor to the control system from under inserted stator. Leading the signal using a duct was connected with the risk of damaging the duct during inserting the stator in spite of used grooves for ducts in the shaft of the engine. When the sensor was set under the stator after inserting, checking if the ducts are not damaged was quite difficult and during production it would not be realized. This is why the pressure sensor was transferred to the cover under the stator and the pressure signal was led in a capillary groove where the ducts were placed (Fig. 14). Probability of damaging the capillary is much smaller than damaging the ducts and checking patency of the capillary after inserting the stator is not a problem.

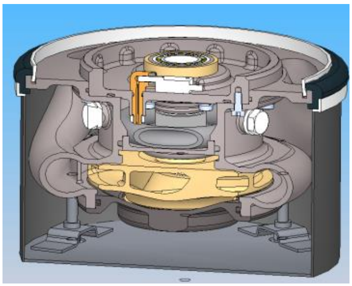

Fig. **13**. The way and place of building the pressure sensor at the bottom of the engine body under the stator.

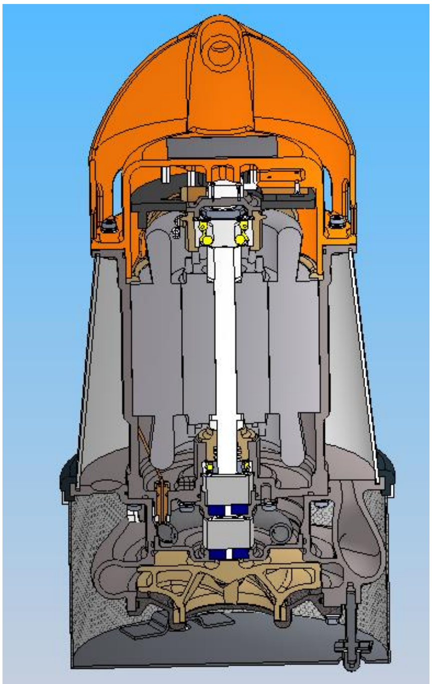

Fig. **14**. The place of building the pressure sensor over the stator

5 NUMERICAL RESEARCH

The results of measurements lead to validation of counting model. Basing on the results of counting the analysis of flow can be carried out in the hydraulic system of the pump (analysis of pressure distribution, velocity, stream line of the liquid). The analysis lets estimate the qualitative and quantitative parameters of flowing system. The basic equations describing the flow of working medium in modeling system are:

- equation of continuity of flow (conservation of mass)

$$\frac{\rho}{\partial t} + \nabla(\rho\varpi) = 0$$

- **-** Navier– Stokes equation (conservation of momentum)

$$\rho\frac{D\varpi_x}{Dt} = g_x\rho - \frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2\varpi_x}{\partial x^2} + \frac{\partial^2\varpi_y}{\partial y^2} + \frac{\partial^2\varpi_z}{\partial z^2}\right)$$

$$\rho \frac{D\varpi_y}{Dt} = g_y \rho - \frac{\partial p}{\partial y} + \mu \left(\frac{\partial^2 \varpi_x}{\partial x^2} + \frac{\partial^2 \varpi_y}{\partial y^2} + \frac{\partial^2 \varpi_z}{\partial z^2} \right)$$

$$\rho \frac{D\varpi_z}{Dt} = g_z \rho - \frac{\partial p}{\partial z} + \mu \left(\frac{\partial^2 \varpi_x}{\partial x^2} + \frac{\partial^2 \varpi_y}{\partial y^2} + \frac{\partial^2 \varpi_z}{\partial z^2} \right)$$

- \- equation of conservation energy

$$\rho \frac{Di}{Dt} = \nabla(k \nabla T) + \frac{Dp}{Dt} + \mu \Phi + q_v$$

where:

t – time $[\mathrm{s}]$
ρ – density $\left[\frac{\mathrm{kg}}{\mathrm{m}^3}\right]$
w – velocity
μ – viscosity
i – entalphy
k – overall heat -transfer coefficient
T – temperature
p – pressure
q – internal heat source

To solve systems of differential equations discretization of counting domain was carried out. Volume, limited by geometry of flowing system was divided into smaller parts (volumes). Fig. 15 shows types of pieces used in the process of discretization of the model.

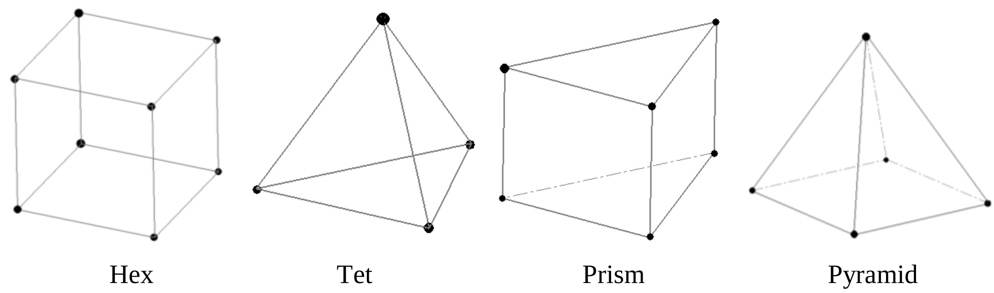

Fig. 15. Types of parts used in the process of discretization of the model

In the task the net of counting domain composing of different types of elements (model of discretization) was created. The basic elements were tetrahedral elements but to increase the accuracy of the analysis in the boundary areas, boundary layers were created and they were of prismatic elements.

Because most of the flows are of turbulent type, which basic feature is turbulence and fast changes of basic parameters such as velocity, pressure or density at each point, it was necessary to elaborate models of turbulence which qualities depend on empirical coefficients. The most common models of turbulence for counting are k-ε, k-ω, SST. In this analysis model of turbulence k-ε was used. To make the process of modeling faster a simplified geometry of flowing system was used (Fig. 16). On the planes of division of geometry into parts the condition of rotational periodicity was set. To join mobile and stationary elements (impeller and stator) stage interface was used. For the mobile elements (impeller) the value of set rotational speed was similar to nominal rotational speed of the impeller. The static pressure of value Pin=0,1 kPa was set on the inlet. On the outlet the value of mass flow was set. To gain the characteristics of capacities of the flowing system analysis of three values of flow were carried out:

$Q=27m^3/h$
$Q=36m^3/h$
$Q=45m^3/h$

To estimate local parameters of liquid flow in hydraulic channels numerical calculations were carried out for three capacities. Fig. 16 shows the elements of the flowing system.

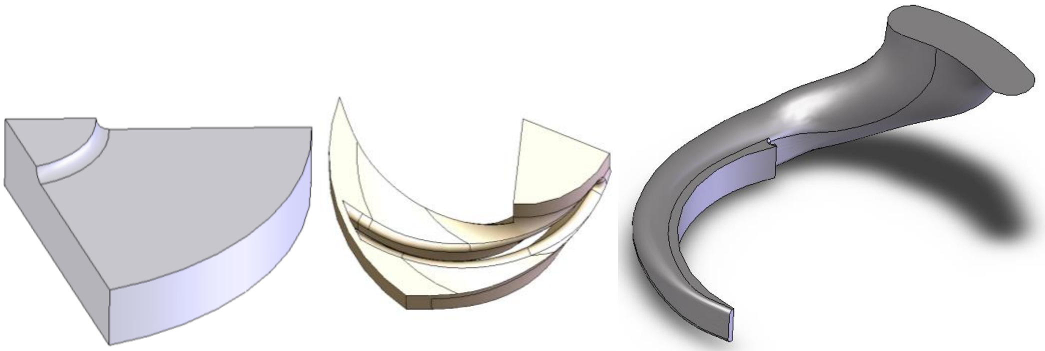

Fig. 16. Models of elements of the flowing system

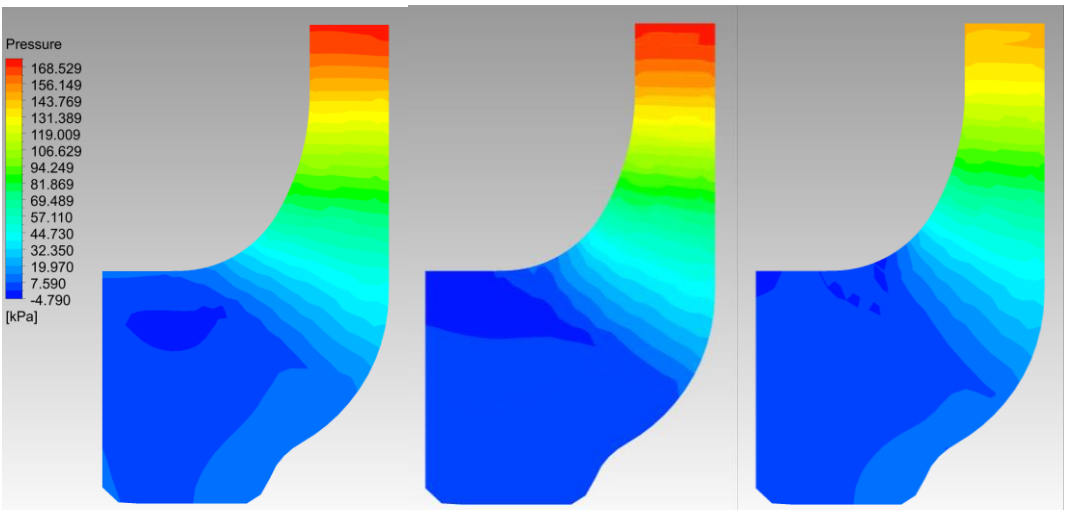

Fig. 17. Pressure field in the impeller in meridional planes $Q=27m^3/h$, $Q=36m^3/h$, $Q=45m^3/h$

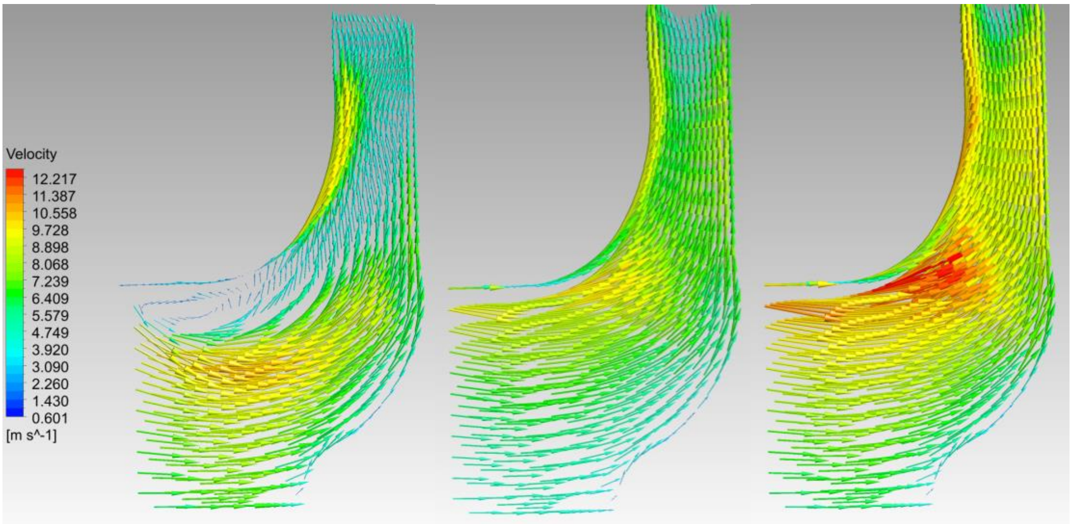

Fig. 18. Velocity vectors in the impeller in meridional plane for capacities of values $Q=27m^3/h$, $Q=36m^3/h$, $Q=45m^3/h$

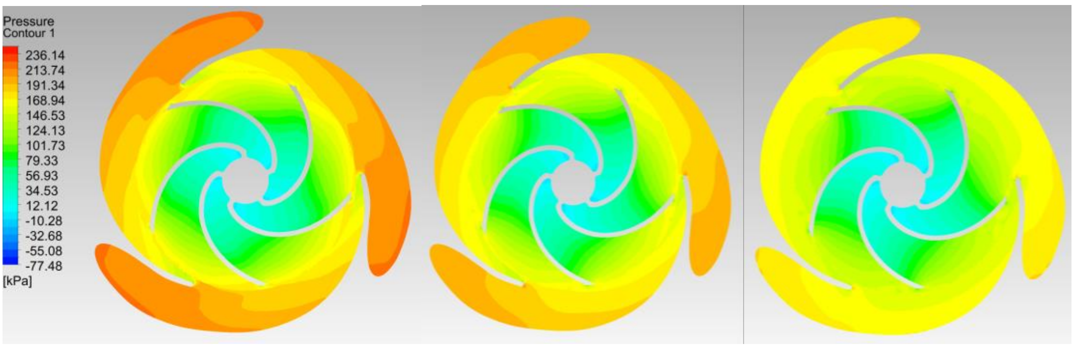

Fig. 19. Pressure fields in cross-section in the impeller and spiral casing.

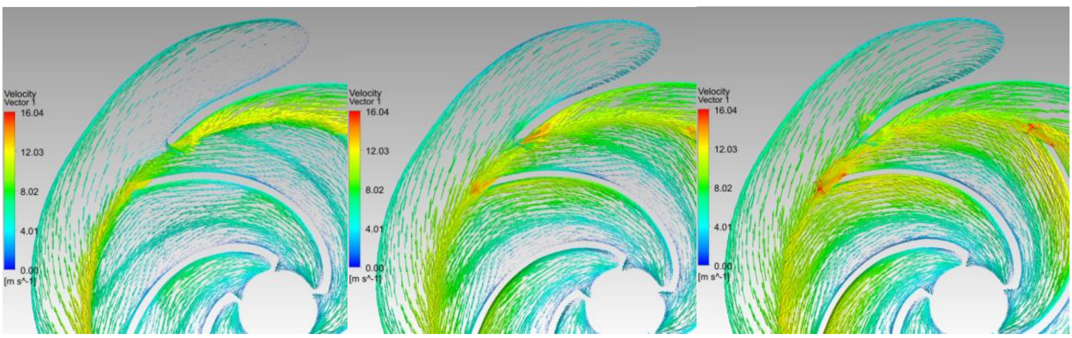

Fig. 20. Velocity vectors in cross-section in the impeller and spiral casing

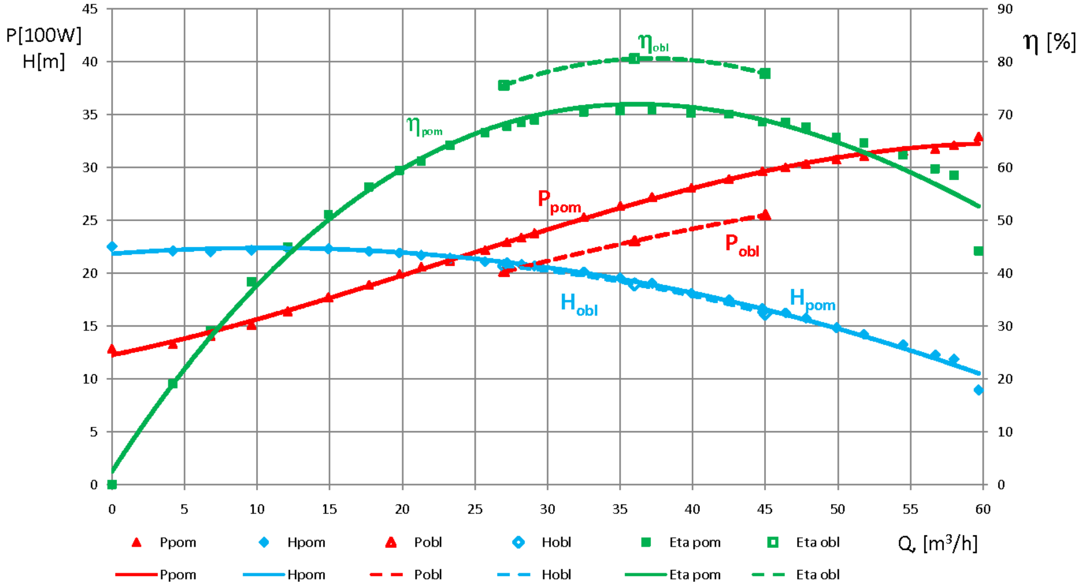

Fig. 21. The comparison of calculated characteristics of pump H_{obl}(Q), P_{obl} (Q) i Eta_{obl} (Q) with characteristics H_{pom} (Q), P_{pom} (Q), Eta_{pom} (Q) counted in a laboratory

From the comparison it is seen that characteritics of flow cover the counted working points H_{obl}(Q). The characteristics of power Pobl(Q) and efficiency Etaobl(Q) do not take mechanical loss in the bearing and seal and loss of leak in the neck of the impeller into account. The quality assessment basing on data of one of the producers of mechanical seal shows that loss of friction of the seal immersed on one and on both sides in oil and pumping water and mechanical loss of bearings will have the value .equalizing measured and counted power balance.

CERTIFICATION TESTS

Because the construction assumption was using the pumping engine in the explosive areas the engine must be set to certification tests. The tests are carried out in OBAC in Gliwice. So far these tests have been carried out:

Tab. 1 Certification tests which have been carried out

Lp.	Tested parameter	Standard
1	Testing of strength to striking	PN-EN 60079-0:2009 pkt. 26.4.2
2	Testing of flameproof enclosure– determination of the pressure of explosion	PN-EN 60079-1:2009 pkt. 15.1.2
3	Testing of flameproof enclosure – overpressure test	PN-EN 60079-1:2009 pkt. 15.1.3
4	Testing of flameproof enclosure–test of avoiding the transfer of internal explosion	PN-EN 60079-1:2009 pkt. 15.2
5	Testing of flameproof enclosure– checking of dimensions of flameproof joints	PN-EN 60079-1:2009

So far all the tests have been carried out successfully. The other tests will take place after solving the problem with internal fires of the engine.

Fig. 22. shows the parts of pumping engine prepared to test flameproof enclosure

Testing of flameproof enclosure– determination of the pressure of explosion
Mixture of 9,5 % CH_4 (methane) with air P_{max}= 2,952 [bar]

Testing of flameproof enclosure – overpressure test
During standstill of engine P_{max}= 6,06 [bar]
During motion of engine P_{max}= 6,72 [bar]

Testing of flameproof enclosure–test of avoiding the transfer of internal explosion
Mixture of 50% H_2 (hydrogen) with air P_{max} = 1,944 [bar]

Determined according to testing basic pressure for casing of pump
P_{od} = 3,204 [bar]

CONCLUSIONS

Implementation of new construction can face unexpected problems and it often causes changes in constructional documentation. Progress in technology of producing parts of machines, especially in founding and machining, lets increase the quality and realization of more difficult constructions. Constructional assumptions of the described pumps need a lot of knowledge and involvement. Only common actios during solving problems let gain satisfactory results. Implementation of the prototype has given three patents and one patent application. Optimalization needs further research and we also need to work on elaborating analytical algorithms of new flowing system which elements are slow-speed impeller of new construction, triple spiral stator and water jacket with confusor inlet.

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