con KEF-4,5 (100), which has high elastic characteristics. The analysis of real products for resonance and the ability to modify them in order to remove from the range of forbidden eigenfrequencies remain an actual problem in the design of sensitive elements of the pendulum micromechanical accelerometers.
The main constructive node of the micromechanical accelerometer is a sensitive element which includes mass and elastic elements of the suspension, is attached to the support frame (base). Elastic elements of the suspension are located on the console or bridge scheme.
Under the bridge scheme, the suspension M moves strictly along the measuring axis. Multilayer bridge suspension M is characterized by low sensitivity to transverse actions, high rigidity and basic self-frequency. The dynamic analysis seeks to calculate of the resonant (own) frequencies and their corresponding forms of oscillation.
The module COSMOSWorks implements the classical finite elemental method, which has the following limitations: damping is not taken into account; the presence of friction is ignored; the external load which changeable is absent.
The consequence of the first limitation is the inability to obtain information on the state of the design at the moment of resonance. None of the parameters (displacement, deformation, stress) is not calculated. Also, the analysis of behavior during loading of loads is not available.
However, even with these restrictions, the program allows you to solve the most urgent task - to perform the analysis of real products on the resonance and to modify them in order to remove from the range of forbidden eigenfrequencies.
The finite element method was used to study influence of the geometric parameters of the elastic suspension and the mass of the sensitive element on the frequency of natural oscillations with correction for damping. In the modeling of suspensions of different shapes, a stress-strain state of the sensitive element was determined and an analysis of the elastic characteristics was conducted to select the optimal design.
Key words: pendulum micromechanical accelerometer, sensitive element, finite element method, frequency of natural oscillations, damping ability.

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OPTIMIZATION OF TURBINE TYPE FLOW RATE TRANSDUCER WITH HYDRODYNAMIC BALANCING OF SENSITIVE ELEMENT

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In the article the accuracy increasing necessity of flow rate transducer by optimizing their design parameters is proven. Optimization of turbine type flow rate transducer with sensitive element hydrodynamic balancing parameters is one of its operation principle mathematical simulation stages. Optimization criteria are formulated in the article allowing estimating the operation of the transducer in a dynamic mode. The design objectives choice is substantiated. Their boundary conditions have been analyzed and determined. The optimization results of the flow rate transducer design according to the developed algorithm are presented.

Keywords: *optimization, mathematical modeling, turbine type flow rate transducer, sensitive element, hydrodynamic balancing.*

Introduction

Fluid flow rate and volume measuring instrument accuracy is provided by the characteristics of their individual elements. Most modern secondary transducers of these instruments are universal and do not create additional resistance to the measured medium flow. Therefore, an accuracy increase of the measuring instruments provides the fluid flow transducers improvement.

In turn, increasing the accuracy of measuring transducers is possible by developing new measurement methods or by improving measuring instruments and systems based on existing methods. The improvement is realized by the results of numerous experimental studies or by the optimization of the measuring instrument elements directly at the design stage. The last is the most effective way to obtain high metrological performances of measuring instruments.

Hence the question arises about the systems creation for the automated design of measuring instruments with improved metrological performances. Such systems make it possible to study the measuring instruments operation in various operating conditions and to determine the design elements rational parameters that ensure high metrological performances [1].

Currently, more attention has been paid to the use of optimization methods for solving applied technical problems [2 – 7], but the objectives of improving turbine flow transducers (TFT) by optimizing their design parameters have not been adequately considered in the literature.

Mathematical modeling is widely used in modern practice of designing measuring instruments. It takes the leading place among other research methods.

Optimization of TFT with Sensitive Element (SE) hydrodynamic balancing parameters is one of its operation principles mathematical simulation tasks [1]. There are two optimization ways. An experimental path requires appropriate laboratory equipment and huge time and financial expenses. The theoretical way allows to significantly reducing the number of experimental studies, which are necessary only to refine the optimization results in this case.

Formulation of the problem

The TFT optimization consists in the search for its internal parameters rational values, providing a minimum or maximum of the objective function by a certain criterion.

To optimize the parameters of the flow rate transducer, the following must be done: to determine the optimization criteria and the objective function that confirms the measuring system operation quality; to identify the most influential objective function design parameters; to demonstrate the constraints in which the design parameters may change; to choose optimization method; to develop an optimization algorithm; to create an application software package; to check the TFT modeling results using CFD-technologies (Computational Fluid Dynamics). This allows us to specify the instrument features at measuring range certain points under conditions that are as close as possible to the real ones [8 – 11].

The parameters optimization purpose of TFT with SE hydrodynamic balancing is to obtain measuring instruments with higher metrological characteristics. The transducer parameters quantity that determines its metrological properties can be different. The quality of the instrument is affected not only by individual parameters, but also by their combination. Therefore, metrological and operational features improvement is possible under the condition of choosing rational relations between the values of various transducer parameters.

Optimization criteria definition

Optimization of the TFT is expedient to carry out on a complex criterion, which would allow evaluating its operation in a dynamic mode. The dynamic mode of the flow rate transducer is characterized by the values of relative and average integrated errors in measured flow rate wide range.

For turbine flow measuring instruments it is advisable to use functionals, which characterize accuracy as optimization criteria. The relative error in measuring the SE rotational speed is determined by expression

$$\Delta n = \frac{n_{id} - n_r}{n_{id}} 100 \; . \qquad (1)$$

Where, n_r is the SE real rotation frequency of TFT (considering resistance to rotation); n_{id} is the SE ideal rotation frequency of TFT SE (without resistance to rotation).

The component n_p is a functional of the measured medium flow physical properties and the inner surface geometric features of the flow rate transducer measuring chamber. This makes the determination of the relative error (1) very difficult. This value is determined by solving a differential equations system that describes the SE rotational and translational longitudinal motions [12 – 15]:

$$\begin{cases} 2\pi J \dfrac{dn(t)}{dt} = M_D - \sum M_R \\ m \dfrac{d^2 x}{dt^2} = \sum F_i \end{cases} . \qquad (2)$$

Where, $n(t)$ is the SE rotational speed; J is the SE moment of inertia; M_D is the driving torque; M_R is the SE rotation resistance moments; m is the SE mass; x is the SE longitudinal displacement; ΣF_i is the main vector of forces acting on the SE.

The system coefficients (2) are determined by the transducer design parameters and the measured flow physical properties. Hence, to obtain each specific value of Δn, it is necessary to calculate the equations system coefficients (2) and solve it. So, it is impossible to write down the objective function explicitly.

The flow rate transducer relative error can be determined in the form of a mean integral error

$$\Delta n_{CI} = \sum_{i=1}^{n} \Delta n_i P_i \; . \qquad (3)$$

Where, Δn_i is the relative error value at the i-th flow rate; P_i is the weight coefficient [16], which is the relative volume measured at the i-th flow rate (Q_i)

$$\sum_{i=1}^{n} P_i = 1{,}00 \; .$$

Choice of design parameters

The design parameters for the investigated transducer are the geometric characteristics of the measuring chamber design elements, which significantly affect its metrological performances and the work as a whole.

These transducer geometric parameters are: number of blades *z*; blades profile thickness *h*; radial clearance, which is the difference between the measuring chamber inner surface radius (r_K) and the SE hub radius (r_{BT}); blade angle β; the ratio of the SE hub radius and the blades outer surface radius (r_H); turbine axial length *s*; the ratio of the input fairing maximum radius (r_O) and the SE hub radius.

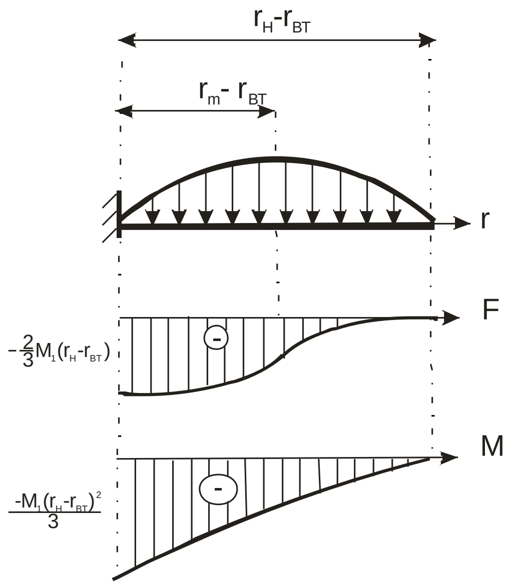
Boundary conditions

The transducer construction elements values vary in certain ranges.

This is due to the processes physics occurring in the transducer, on the one side, and to the features of the measuring chamber internal surface, on the other side.

Blades profile thickness h. The turbine blades should be as thin as possible, for technological and constructive reasons.

The minimum value of the blade profile thickness is determined from the blade strength design under the action of forces and moments, which are created by the maximum flow rate [17]. In this case, the blade is viewed as a cantilever beam fixed to the turbine hub. The construction results of the bending moment diagram indicate that the dangerous cross-section is at the blades location on the hub (Fig. 1).



$$0 = r_H\ ,$$

$$r_H \le r \le r_{BT}\ ,$$

$$F(r) = -2M_1\left(\frac{(r_H - r)^2}{(r_H - r_{BT})^2} - \frac{2}{3}\frac{(r_H - r)^3}{(r_H - r_{BT})^3}\right),$$

$$F(0) = 0\ , \qquad F(r_{BT}) = -\frac{2}{3}M_1(r_H - r_{BT}), \tag{4}$$

$$M(r) = -\frac{M_1(r_H - r_{BT})^2}{3}\left(2\frac{(r_H - r)^3}{(r_H - r_{BT})^3} - \frac{(r_H - r)^4}{(r_H - r_{BT})^4}\right),$$

$$M(0) = 0\ , \qquad M(r_{BT}) = -\frac{M_1(r_H - r_{BT})^2}{3}\ .$$

Fig. 1. Distribution diagrams of shear force and bending moment for the sensitive element blade. Where, r_m is the radius appropriate to the maximum speed value; M_1 is the driving torque moment acting on one blade

An expression, determining the minimum allowable turbine blade thickness, can be obtained from the system of conditions and equations (4)

$$h_{\min} = \sqrt{\frac{3k^2F^2 + \sqrt{9k^4F^4 + 144l_Л^2M_{\max}^2[\sigma]^2}}{2l_Л^2[\sigma]^2}}\ . \tag{5}$$

Where, *k* is a coefficient that depends on the cross-sectional shape; *F* is the force applied to the blade; $l_Л$ is the blade profile chord length; $M_{\max}$ is the maximum value of the moment; $[\sigma]$ is the allowable bending stress [17].

The maximum allowable turbine blade thickness is determined from the conditions of constructive and technological expediency. That is, the blade thickness and channel width between blades must be equal, on the one side. Blades should be located evenly on the SE hub surface on the other side

$$h_{\max} = \frac{\pi r_{BT}}{z}\ . \tag{6}$$

Radial clearance (r_K-r_H). The turbine outer diameter determines the radial clearance between the blades and the measuring chamber inner surface. When choosing its value, it is necessary to take into account:

– the flow overflow through the blades caused by the difference in pressure on the opposite blade sides, lead to a decrease in the driving torque;

– the clearance should be greater than the boundary layer thickness on the body.

Since the radial clearance magnitude is affected by the r_H value, the allowable values of the radial clearance value are determined by the limiting values of the ratio r_H/r_K.

As a rule, the clearance size is selected [14] from the range

$$(r_K - r_H) = (0{,}0192 \div 0{,}0392)r_K\ . \tag{7}$$

Hub-tip ratio (r_{BT}/r_H). The SE hub diameter has no fundamental importance in the investigation of the two-dimensional flow model. The actual flow spatial picture is complicated by the unevenness of the centrifugal forces field, the inconstancy of the attack angles along the blade height, and so on. To reduce the secondary flows harmful effect associated with it the difference between r_H and r_{BT} should not be very large.

However, a decrease in the blades height causes a number of constructive nature complications, so it is recommended that r_{BT} be selected in the range [14]

$$r_{BT} = (0{,}48 \div 0{,}78) r_K . \qquad (8)$$

Number of blades z. The turbine blades form a so-called lattice of profiles. An important geometric characteristic of such lattice is its step

$$t = \frac{2\pi r_{CP}}{z} .$$

Where, r_{CP} is average turbine radius.

The limiting allowable values of turbine blades number are determined from the conditions of the location uniformity along the cross section of the flow turn and prevention of the boundary layer detachments on the blades surface. In this case, the lattice step should not exceed the blade height. Since the blade height is

$$(r_H - r_{BT}) = (0{,}48 \div 0{,}2) r_K ,$$

the number of blades is determined from expression

$$\frac{2\pi r_H}{z} = (0{,}48 \div 0{,}2) r_K . \qquad (9)$$

Solving (9) with relatively to z, we obtain $z_{\min} = 12$, $z_{\max} = 30$.

In practice, it is advisable to choose, as boundary conditions $z_{\min} = 4$ та $z_{\max} = 30$ [14].

The blades installation angle β on the average radius can vary from π/6 to π/3 [14].

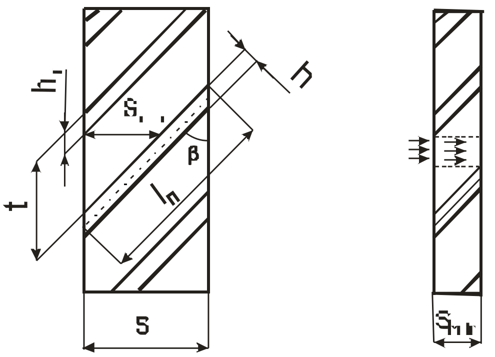
Turbine axial length s. The *s* value can also vary in a certain range only. The turbine axial length minimum value should be such that the flow completely fills the channel between blades, that is, all flow elements act force on the blade. If the *s* value is made smaller, then the part of measured flow will pass through the SE without causing its rotation (Fig. 2). This leads to an increase in the flow rate measuring error.

Based on this, the minimum allowable value of the turbine axial length is given by

$$s_{\min} = \frac{(2\pi r_{CP} \sin\beta - hz)\cos\beta}{z \sin^2\beta} . \qquad (10)$$

The *s* maximum allowable value is determined on the assumption that the blade length should not be greater than the one complete turn of the cylindrical helix length (Fig. 3)

$$s_{\max} = 2\pi r_{CP} tg\beta . \qquad (11)$$



a) b)

Fig. 2. To determining the value $s_{\min}$

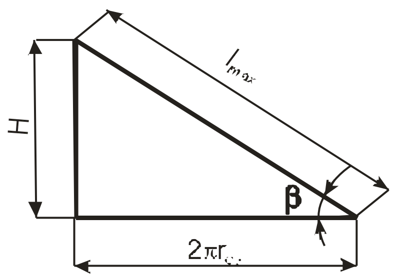


Fig. 3. To determining $s_{\max}$

On the other side, the helical line parametric equations are written as follows

$$\begin{cases} x = a\cos t \\ y = a\sin t \\ z = amt \end{cases} . \qquad (12)$$

Where, *a* is the radius of the cylinder on which the helix is wound; *t* is the central angle, $m = tg\beta$.

Define the helix turn length. In accordance with [18], if the function is given parametrically then its line length is determined by the expression

$$s = \int_{t_1}^{t_2} \sqrt{[x'(t)]^2 + [y'(t)]^2 + [z'(t)]^2} . \qquad (13)$$

In accordance with (12), we obtain

$$dx = -a\sin t dt , \quad dy = a\cos t dt , \quad dz = amdt . \qquad (14)$$

Hence, taking into account (13) and (14), for the maximum allowable blade length, we can write

$$l_{Л\max} = a\int_0^{2\pi} \sqrt{1+m^2}\, dt .$$

Because the $s_{\max} = l_{Л\max} \sin\beta$,

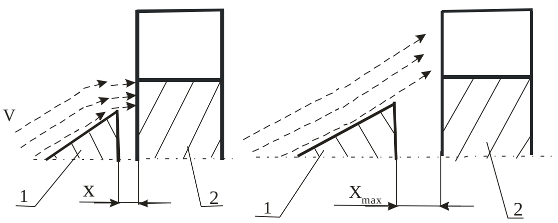
then

$$s_{\max} = 2\pi r_{CP}\sqrt{1+tg^2\beta}\sin\beta\,. \qquad (15)$$

The inlet fairing maximum radius r_O and the SE hub radius r_{BT} ratio. The inlet fairing radius value is determined from the conditions for the regulating force existence [13]:

$$r_O = r_{BT} - x_{\max}\sin\alpha\,.$$

Where, $x_{\max}$ is the maximum possible distance between the SE and the inlet fairing (is given for constructive expediencies); α is the angle between the height and the inlet fairing generator (Fig. 4).



a) *b)*

Fig. 4. To determining the r_O: 1 – inlet fairing; 2 – SE; x – distance between SE and fairing; V – flow velocity

Taking into account expression (8), the r_O variation limits are determined by the values

$$r_{O\min} = 0{,}48 r_K - x_{\max}\sin\alpha\,,$$

$$r_{O\max} = 0{,}78 r_K - x_{\max}\sin\alpha\,.$$

Additional constraints are a feature of the transducer with SE hydrodynamic balancing parameters optimization. These constraints are imposed by SE axial balancing conditions and determine the measurable flow rates range [13, 19]. The complete SE hydrodynamic balancing is provided by mutual compensation of all forces applied to it in a wide range of the transducer operating modes.

The multiplicity of transducer design parameters, the measured medium physical and hydrodynamic properties, and additional constraints caused by the SE balancing, indicate the multidimensionality of this problem.

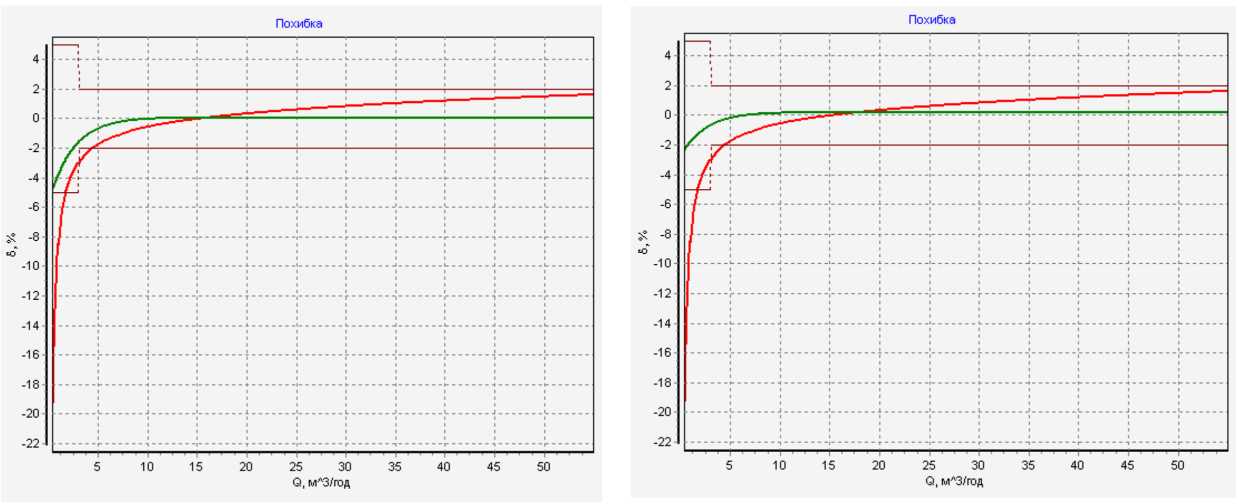
The methods used in multivariable optimization can be divided into three broad classes in accordance with the type of tasks and information used in their realization [2 – 7]:

– the direct search methods, based on the objective function values calculation only;
– the gradient methods in which the exact meanings of the objective function first derivatives are used;
– the second-order methods, in which, equally with the first derivatives, the second-order objective function derivatives are also used.

The analysis of multivariable methods optimization and components of the considered problem testifies that it is difficult and practically impossible to write down the objective function explicitly. In addition to the restrictions imposed on individual design parameters, there are also constraints imposed on the search area.

Consequently, the design parameters optimization of the transducer measuring chamber with SE hydrodynamic balancing by classical gradient methods and second-order methods is impossible, because of the objective function gradient obtaining complexity.

The TFT with SE hydrodynamic balancing optimization is carried out according to the direct search strategy based on the developed algorithm. This allowed us to take into account the changes limits in design parameters and restrictions on the search area (Fig. 5).



a) b)

Fig. 5. The TFT optimization results: a – minimum relative error optimization; b – minimum mean integral relative error optimization

As an optimization result by the criterion of the minimum mean integral relative error, the design parameters values are obtained, at which it does not exceed 0.24 %, and by the criterion of a minimum relative error of 0.25 %, which indicates their reliability. The obtained design elements values became the basis of the transducer experimental model.

Conclusions

The problem of optimization the turbine flow rate transducer design is solved using a direct search strategy based on a complex criterion, which is characterized by minimum values of the relative and average integral errors. In this case, the initial and boundary values of the design parameters are determined from the sensitive element balancing conditions in a wide measurements range.

Further work will be aimed at creating favorable conditions for the medium flow in the flow transducer measuring chamber and its bench research.

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ОПТИМІЗАЦІЯ ТУРБІННОГО ПЕРЕТВОРЮВАЧА ВИТРАТИ З ГІДРОДИНАМІЧНИМ ВРІВНОВАЖУВАННЯМ ЧУТЛИВОГО ЕЛЕМЕНТУ

Важливою задачею математичного моделювання роботи турбінних перетворювачів витрати рідини із гідродинамічним врівноважуванням чутливого елементу є їх оптимізація із вибором раціональних значень пара-

метрів конструкції. Причому можливі два шляхи такої оптимізації: експериментальний, що потребує відповідного лабораторного обладнання та великих часових і фінансових затрат; теоретичний, який дозволяє значно зменшити кількість експериментальних напівнатурних досліджень, необхідних, у такому випадку, тільки для уточнення результатів оптимізації. Оптимізація турбінного перетворювача витрати полягає у пошуку значень його внутрішніх параметрів, що забезпечують екстремум цільової функції за означеним критерієм. У статті сформульовано критерії оптимізації, що дозволяють оцінити роботу перетворювача у динамічному режимі, обґрунтовано вибір проектних параметрів, проаналізовано та визначено їх граничні значення за умов рівноваги чутливого елементу у широкому діапазоні зміни витрати.
Множина параметрів конструкції перетворювача витрати, фізичних і гідродинамічних властивостей вимірюваного середовища, та додаткові обмеження, викликані врівноважуванням чутливого елементу, вказують на багатовимірність задачі оптимізації. Аналіз методів багатовимірної оптимізації та складових наведеної задачі свідчить про те, що через складність отримання градієнту цільової функції, оптимізація параметрів конструкції вимірювальної камери турбінного перетворювача витрати із гідродинамічним врівноважуванням чутливого елементу класичними градієнтними методами та методами другого порядку неможлива. Тому доцільним у даному випадку шляхом залишається стратегія прямого пошуку, яка дозволяє ефективно врахувати межі зміни проектних параметрів та обмеження на область пошуку.
Виходячи з цього, поставлену задачу оптимізації конструкції турбінного перетворювача витрати рідини розв'язано із використанням стратегії прямого пошуку за комплексним критерієм, що характеризується мінімальними значеннями відносної та середньоінтегральної похибок. При цьому початкові та граничні умови проектних параметрів визначено із умов рівноваги чутливого елементу у широкому діапазоні вимірювань.
Ключові слова: оптимізація, математичне моделювання, турбінний перетворювач витрати, чутливий елемент, гідродинамічне врівноважування.

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ОПТИМИЗАЦИЯ ТУРБИННОГО ПРЕОБРАЗОВАТЕЛЯ РАСХОДА С ГИДРОДИНАМИЧЕСКИМ УРАВНОВЕШИВАНИЕМ ЧУВСТВИТЕЛЬНОГО ЭЛЕМЕНТА

Важной задачей математического моделирования работы турбинных преобразователей расхода жидкостей с гидродинамическим уравновешиванием чувствительного элемента является их оптимизация, предполагающая выбор рациональных значений конструктивных параметров, обеспечивающих экстремум целевой функции по определенному критерию. В статье сформулированы критерии оптимизации, позволяющие оценить работу преобразователя в динамическом режиме, обоснован выбор проектных параметров, проанализированы и определены их граничные значения при условии равновесия чувствительного элемента в широком диапазоне изменения расхода.
Множество параметров конструкции преобразователя расхода, физических и гидродинамических свойств измеряемой среды и дополнительные ограничения, вызванные уравновешиванием чувствительного элемента, указывают на многопараметричность задачи оптимизации. Анализ методов многопараметрической оптимизации и составляющих рассматриваемой задачи свидетельствует о том, что из-за сложности получения градиента целевой функции, оптимизация параметров конструкции измерительной камеры турбинного преобразователя расхода с гидродинамическим уравновешиванием чувствительного элемента классическими градиентными методами и методами второго порядка невозможна. Поэтому целесообразным в данном случае является применение стратегии прямого поиска, позволяющей эффективно учесть пределы изменения проектных параметров и ограничения на область поиска.
Исходя из этого, поставленная задача оптимизации конструкции турбинного преобразователя расхода жидкости решена с использованием стратегии прямого поиска по комплексному критерию, характеризующемуся значениями относительной и средне интегральной погрешностей. При этом начальные и граничные условия проектных параметров определены из условий равновесия чувствительного элемента в широком диапазоне измерений.
Ключевые слова: оптимизация, математическое моделирование, турбинный преобразователь расхода, чувствительный элемент, гидродинамическое уравновешивание.

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