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Highlights

- Pressure Vessel Steel P355 N
- Submerged Arc Welding Parameters
- Research on Partial Reduction
- Devices for Complex Mechanical

Discovering Thoughts, Inventing Future

VOLUME 23 ISSUE 4 VERSION 1.0

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Mechanical and Mechanics Engineering

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The Influence of Submerged arc Welding Parameters on Bending Strength of Pressure Vessel Steel P355 N

By Cioroagă Bogdan-Dorel, Emanoil Linul, Cioată Vasile George & Dascăl Amalia
Politehnica University Timişoara

Abstract- To be able to optimize the quality of the welded joints of products such as pressure vessels, it is necessary to know the demands to which they are subjected during operation and the mechanical characteristics that need to be considered in order to be followed to obtain the best possible quality from the point of view of mechanical resistance. This paper is focused on the specific bending strength of the welds executed on the circular contour of the pressure vessels. The parameters of the welding regime that are considered are: the voltage of the electric arc, the intensity of the electric arc and the welding speed, for which are determined evolution trends of the bending mechanical characteristics.

These trends are compared with each other, and the contribution of each parameter is observed, as well as which has the most pronounced effect on the bending resistance characteristics. The collection of data regarding the bending resistance characteristics is carried out with the help of the stress-deflection curves resulting from the bending testing of the specimens taken from welded joints under different welding regimes.

Keywords: *welding, parameter, stress, strength, bending.*

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Strictly as per the compliance and regulations of:

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Keywords: welding, parameter, stress, strength, bending.

I. Introduction

The submerged arc welding technology is mostly used in the fabrication of industrial pressure vessels. To ensure the durability and safety in operation of products such as pressure vessels that can work at room temperature or high temperatures, ensuring good resistance in maintaining the high pressures that develop inside the vessels it is necessary that the welded joints have good mechanical properties.

The mechanical properties of the welded joints are mainly influenced by the applied welding regime, having as the main parameters the welding voltage, the intensity of the welding current and the welding speed. The welded joints found for the fabrication of pressure vessels are executed in 2 distinct ways, continuous longitudinal welding or continuous welding on the circumference.

The welded joints executed by the longitudinal welding method mostly support stretching loads, due to the pressures developed inside the pressurized vessels, having an effect of uniform stretching of the walls of the vessels. In the case of the welded joints executed on the circumference of the pressure vessels, the stresses that take place are generated by bending loads, the situation being presented in figure 1.

The pressurized vessels are usually made of two main parts, the first being the central cover made of a laminated steel sheet, rolled and closed by longitudinal welding which can be seen in figure 1, detail X. The other component is the cap, which is present on both ends, it is also made of laminated steel sheet, obtaining its shape by embossing, its connection with the central cover is made by circular welding on the exterior round contour. [1]

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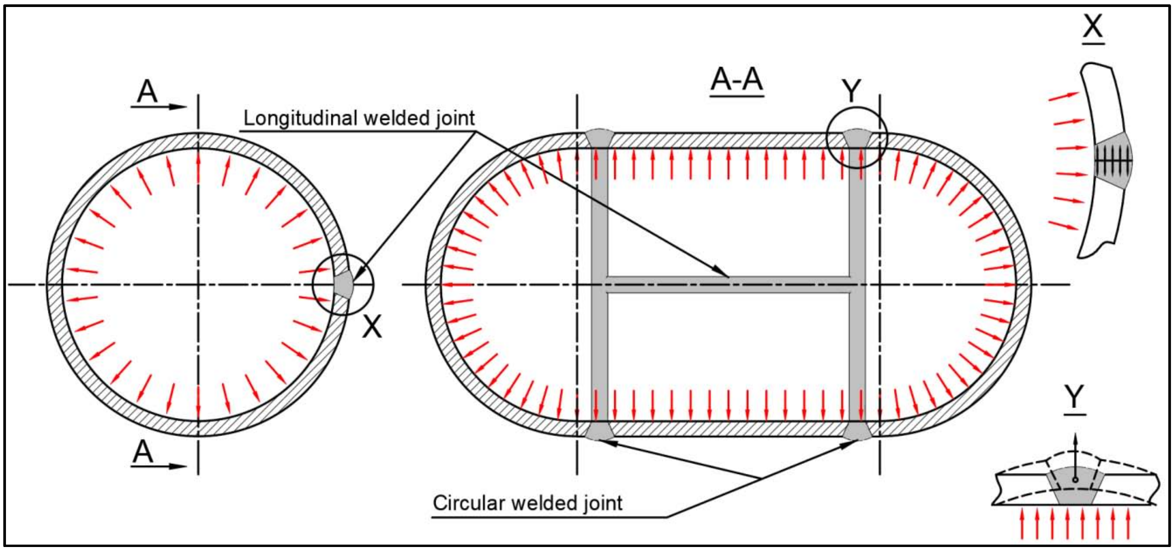

Figure 1: Pressure Vessel Joints and Stress Distribution [2]

In this paper the focus will be on determining the influence of the parameters of the welding regime on the bending strength of the welded seams using the technology of submerged arc welding on the P355 N pressure vessel steel from which the tank wagons for the transportation of petroleum products are made.

The effects of the parameters of the welding regime will be compared by means of the stress-deflection curves resulting from the testing of specimens for bending tests taken from welded seams using different variations of the 3 parameters under study. To carry out the experiments, a set of 4 specimens were taken for bending tests specific to each welding regime. The purpose of the experiments is to develop a statistic regarding the evolution of the bending strength characteristics of the welded seams depending on the welding regime parameters. The data obtained is processed for determining and compare trends in the evolution of changes in the bending resistance characteristics of the welded joints, data that can be used to optimize the welding process, improving aspects regarding productivity and quality.[3][4]

II. Experimental Material and Methods

To carry out the experiments, welded samples were made from 6 mm thick plates joined by butt welding made from P355 N steel, standard SR EN 10216–3:2003. From these samples, bending test specimens were taken, with dimensions of 250x25x6 with the welded joint arranged in the middle of the specimen. The welded samples were made in the industry on a semi-automatic submerged welding machine, the welds being executed respecting the following process characteristics.[5][6]

- *Welding procedure:* Welding under flux with wire electrode, codification 121, according to standard EN ISO4063. [7]
- *Joint type:*
  - o Butt welding with full penetration, codification BW, according to standard SR EN 287–1. [8]
  - o Welding with root support, codification mb, according to standard SR EN 287–1. [8]
  - o One-pass welding, codification sl, according to standard SR EN 287–1
- *Welding position:* Horizontal, codification PA, according to standard SR EN ISO 6947. [9]

The filler material used in the welding process is intended for submerged arc welding of a carbon steel, being a molybdenum-alloyed copper carbon steel, found in the form of a round section electrode of 3, 2 mm in diameter. The manufacturer of the electrode classifies it in the material catalogs as type OK Autrod 12.24. [10]

The material that constitutes the protective environment of the electric arc and the molten weld is a layer of granular flux. Classified by the is manufacturer as type OK flux 10.47, being deposited on the welding joint with an advance before the electrode and later being collected by suction after the passing of the electrode, to be recirculated through the deposition installation which creates a thick layer of approximately 25 mm. [10]

Figure 2 shows an image of the welding process, it can be seen that the electrode is plugged by the flux layer during welding process and the way the flux is deposited and collected during the welding process. The plate on which the weld is made contains 2 samples from which the specimens were taken for the bending test.

Figure 2: The Process of Submerged ARC Welding Samples in the Industry

From one welded sample, 4 bending testing specimens were obtained, each sample being welded with a different welding regime. The processing of the specimens was carried out by the mechanical process of cutting the sample sheet with the guillotine, in order to avoid the thermal effects that other mechanical cutting processes have.

The standard welding regime used in the industry is the reference base to which the other welding regimes subject to the study will be related, as well as their specific strength characteristics. The standard welding regime parameters are the welding voltage 33 V, welding current intensity 480 A and welding speed 60 cm/min.

The structure of the welding regimes under study is composed of 3 main parameters each having 3 stages of variation to determine a trend of the evolution of the strength characteristics depending on the value of the parameters involved. The classification of welding regimes subjected to the study is reproduced under the following structure:

1. Welding regimes that involve the variation of welding voltage:
   - 22V, 480A, 60 cm/min
   - 30V, 480A, 60 cm/min
   - 38V, 480A, 60 cm/min
2. Welding regimes that involve the variation of welding current intensity:
   - 33V, 300A, 60 cm/min
   - 33V, 500A, 60 cm/min
   - 33V, 700A, 60 cm/min
3. Welding regimes that involve the variation of welding speed:
   - 33V, 480A, 25 cm/min
   - 33V, 480A, 75 cm/min
   - 33V, 480A, 125 cm/min

The testing of the specimens took place on an A009 (TC100) universal testing machine of 100 kN capacity, using a device with two support rollers on which the specimen was placed and pressed onto the weld bead with a rounded prism. In figure 3, images from the testing are shown, where it can be seen how the bending prism acts on the sample.

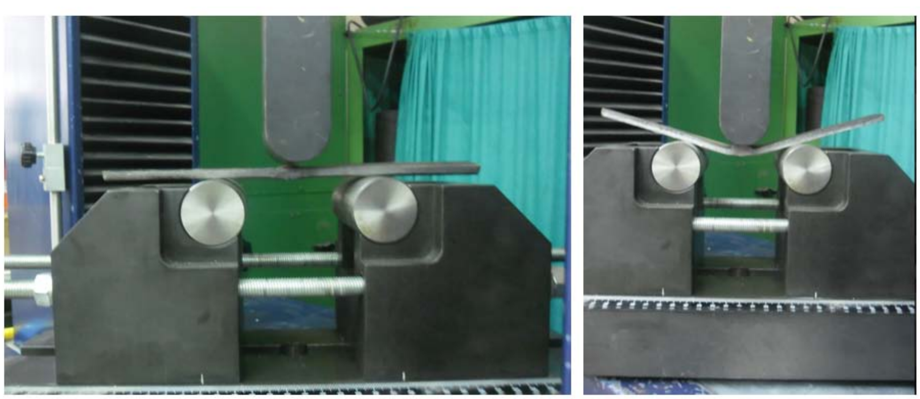

Figure 3: The Bending Test Process

The distance between the centre of the support rollers during testing was 120 mm, the rollers being 40 mm in diameter. The prism pressing on the sample had a radius of 20 mm in the pressing area. Figure 4 shows an image of how it was installed the test specimen in the test device. The testing of a specimen continues until the weld seam cracks or until the specimen is completely bent. The tests took place with a pressing speed of 50 MPa/s, the data acquired by the machine's data acquisition system are the displacement of the mobile beam in which the pressing prism is mounted, and the force applied throughout the experiment. [11]

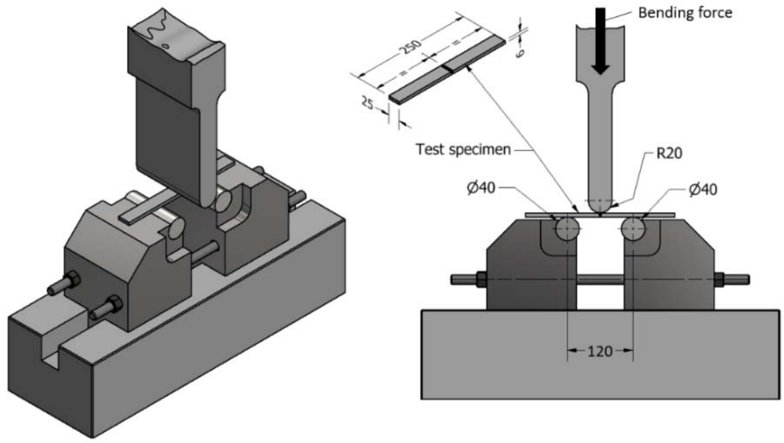

Figure 4: Installation of the Specimen in the Bending test Device

III. Data Analysis and Interpretation

As a result of the experiments, a set of stress-deflection curves were obtained, these being 4 for each welding regime under study, from which a representative curve was extracted for each group. Figure 5 represents a comparison between these stress-deflection curves specific to each welding regime that was involved in the study.

It can be seen the major difference between the sets of curves, a group that aims to study the influence of a specific parameter on the behaviour of welded joints, subjects to bending loads, is made up of 3 stress-deflection curves that are represented with the same colour, the line type of the curve represents the stage of variation for the respective parameter.

Curves with a more pronounced gauge reflect the presence of better mechanical properties, such as superior toughness and ductility compared to curves with a smaller coverage area on the graph that show a fragility of the welded joint.

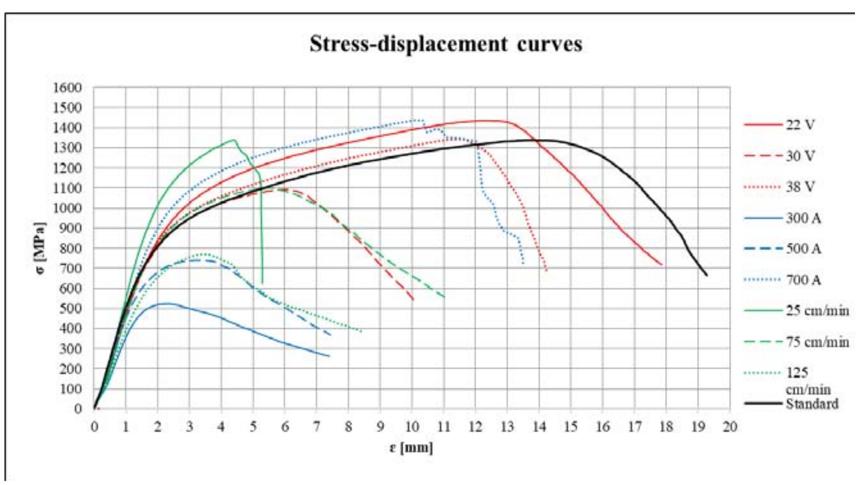

Figure 5: Stress-Strain Curves

Analysing the graph, it can be observed that the use of a welding voltage of 22 V presents a situation of stress-strain curves with superior mechanical properties, it can also be observed a trend of involution of the curve which is more and more pronounced once with the increase of the welding voltage from 22 V to 38. It can also be noted that the use of a welding regime with an amperage of 300 A presents the worst situation presented by the graph, having the lowest curve in terms of the maximum stresses developed in the welded joint.

A centralized situation of the resistance characteristics obtained after the bending tests is presented in tables 1, 2 and 3. The main characteristic followed for determining the bending resistance capacity of welded joints is the maximum bending force.

Using the maximum bending breaking force in formula (1) results in the value of the maximum bending stress developed during the bending test, this value largely depending on the geometry of the welded seam also.

$$\sigma_{max} = \frac{3 \cdot F \cdot L}{2 \cdot w \cdot h^2}$$

$$F - load\ at\ span\ center,$$

$$L - distance\ betwen\ supports,$$

$$w - width\ of\ specimen,$$

$$h^2 - thickness\ of\ specimen$$

Table 1: Strength Characteristics Collected from Stress-Strain Curves

Parameters	22 V	30 V	38V	U.M.
Maxim bending force	9,74	7,55	9,84	kN
Maxim displacement	29,21	10,04	14,27	mm
Maxim bending stress	1527,60	1092,17	1342,66	MPa
Displacement at maxim bending stress	19,23	6,07	11,47	mm
Bending energy	26330,60	5659,37	10305,71	KJ

Table 2: Strength Characteristics Collected from Stress-Strain Curves

Parameters	300 A	500 A	700 A	U.M.
Maxim bending force	3,47	5,60	10,85	kN
Maxim displacement	7,40	8,97	21,76	mm
Maxim bending stress	523,30	832,55	1597,01	MPa
Displacement at maxim bending stress	2,38	3,79	16,71	mm
Bending energy	1842,80	3782,13	27761,95	KJ

Table 3: Strength Characteristics Collected Ffom Stress-Strain Curves

Parameters	25 cm/min	75 cm/min	125 cm/min	U.M.
Maxim bending force	9,13	7,60	5,13	kN
Maxim displacement	5,29	11,10	8,44	mm
Maxim bending stress	1338,03	1098,98	769,41	MPa
Displacement at maxim bending stress	4,42	5,52	3,51	mm
Bending energy	3425,49	6144,46	2925,77	KJ

All the data found in tabular form were regrouped for an easier analysis in figure 6 in the form of a radar graph, highlighting the evolution of the strength characteristics depending on the welding regimes used. Analysing the radar graph, there are 2 welding regimes for which all the strength characteristics have higher values than those specific to the standard welding regime. The standard welding regime that are represented by the following values:

- Maxim bending force 9,35 kN
- Maxim displacement 19,28 %
- Maxim bending stress 1337,7 MPa.
- Displacement at maxim bending stress force 13.85 mm.
- Bending energy force increases from 20957,6 kJ.

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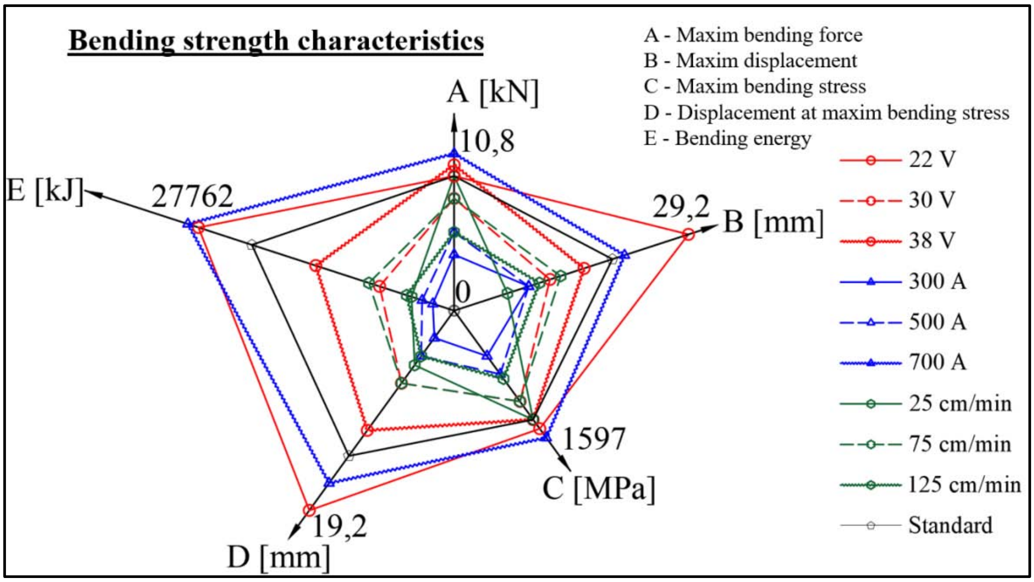

Figure 6: Evolution of Strength Characteristics Collected from Stress-Strain Curves

The welding regime for which the 22 V welding voltage is representative shows the following increases in resistance characteristics compared to the standard welding regime:

- Maxim bending force 9,74 kN resulting an increase of 0.39 kN, signifying a progression of 4,2%
- Maxim displacement 29,21% resulting an increase of 9.93 %, signifying a progression of 51,5%
- Maxim bending stress 1527,6 MPa resulting an increase of 189,9 MPa, signifying a progression of 14,2%
- Displacement at maxim bending stress force 19,23 mm resulting an increase of 5,38 mm, signifying a progression of 38,8%
- Bending energy 26330,6 kJ resulting an increase of 5372,9 kJ, signifying a progression of 25,6%

The second welding regime that has better bending strength characteristics than the standard one is the welding regime were the intensity of the electric arc was 700A, for which the values are:

- Maxim bending force 10,85 kN resulting an increase of 1,5 kN, signifying a progression of 16%
- Maxim displacement 21,76% resulting an increase of 2,48 %, signifying a progression of 12,8%
- Maxim bending stress 1597,01 MPa resulting an increase of 259,31 MPa, signifying a progression of 19,4%
- Displacement at maxim bending stress force 16,71 mm resulting an increase of 2,86 mm, signifying a progression of 20,6%
- Bending energy 27761,9 kJ resulting an increase of 6804,3 kJ, signifying a progression of 32,5%

IV. Conclusions

As a main conclusion it could be said that 2 welding regimes with better strenght characteristics than those of the standard welding regime were identified and this are:

- 33V, 700A, 60 cm/min
- 22V, 480A, 60 cm/min

The maximum bending force recorded is specific to the welding regime 33V, 700A, 60 cm/min with the value of 10,85 kN resulting an increase of 1,5 kN, signifying a progression of 16% comparing to the standard welding regime maximum bending force.

The maxim displacement is found within the welding regime o 22V, 480A, 60 cm/min with the value of 29, 21% resulting an increase of 9.93%, signifying a progression of 51, 5% comparing with the standard welding regime maxim displacement.

The maxim bending energy force is specific for the welding regime 33V, 700A, 60 cm/min and has a value of 27761,9 kJ resulting an increase of 6804,3 kJ, signifying a progression of 32,5% comparing to the standard welding regime bending energy.

The optimal welding regime being considered 33V, 700A, 60 cm/min.

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Preliminary Prototyping and Simulations to Explore Mechanical Properties of 3D Printed Materials for Supporting the Head

By Matthew Dickinson

Abstract- A novel exoskeleton design has been produced that assists the contraction of neck muscles. 3D printing has been employed to reduce costs of manufacturer. The two printing materials employed were Polylatic acid (PLA) and polyethylene terephthalate with carbon (PET-C), and the central spinal cord of the design being Nitrile rubber. The aim of this work was to study the use of 3D printed materials as the main skeletal structure to support the human head and neck. To identify if the 3D printable materials could be offered as an equivalent alternative to conventional more expensive materials. A simulation and proto type were created for this work. An exoskeleton was designed to assist with neck extension. A maximum load of lift force was calculated to be 20N, and this was incrementally reduced to study the effects on the material. A total number of 10 simulations were run to study the head in conditions with no muscular support through to 90% of operational support. When measured against the head, the material performed well within its operational value.

Keywords: 3D printing, exo-skeleton, PLA, PETC.

GJRE-A Classification: DDC Code: 519.2 LCC Code: QA273

PRELIMINARY PROTOTYPING AND SIMULATIONS TO EXPLORE MECHANICAL PROPERTIES OF 3D PRINTED MATERIALS FOR SUPPORTING THE HEAD

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Matthew Dickinson

Abstract- A novel exoskeleton design has been produced that assists the contraction of neck muscles. 3D printing has been employed to reduce costs of manufacturer. The two printing materials employed were Polylatic acid (PLA) and polyethylene terephthalate with carbon (PET-C), and the central spinal cord of the design being Nitrile rubber. The aim of this work was to study the use of 3D printed materials as the main skeletal structure to support the human head and neck. To identify if the 3D printable materials could be offered as an equivalent alternative to conventional more expensive materials. A simulation and proto type were created for this work. An exoskeleton was designed to assist with neck extension. A maximum load of lift force was calculated to be 20N, and this was incrementally reduced to study the effects on the material. A total number of 10 simulations were run to study the head in conditions with no muscular support through to 90% of operational support. When measured against the head, the material performed well within its operational value. The validation of this work showed a m difference which could have been related to the ability of the material to combine while being manufactured. This study presents work in the form of a novel exoskeleton that presents 3D printing as a possible alternative to conventional manufacturing.

Keywords: 3D printing, exo-skeleton, PLA, PETC.

I. Introduction

Over the past decade the development of wearable robotic structures for replacement limbs for rehabilitation's has received attention by many researchers and developers. [1–7] Exoskeleton systems have been traditionally designed for the manufacturing and military sectors, prioritising the ability to move a large mass as efficiently as possible. [8, 9] These systems primarily support an able person's lower body to assist in difficult tasks, such as carrying a load whilst navigating tough terrain, of which the human body alone would not be capable. One example of this type of development is the Berkeley Lower Extremity Exoskeleton (BLEEX) which was designed for applications where a wheeled vehicular transport is not practical. [10] In the past decade, with motors reducing in size and advancements in battery technology, there has been an increase in exoskeleton development for medical purposes. There are two main types of exoskeleton in this category; rehabilitative exoskeletons and assistive exoskeletons. Rehabilitative examples aim to assist with the recovery of total or partial motor abilities. These systems are designed to be reusable, but temporary solution for those patients who have suffered, or are suffering from, a curable illness or injury. Assistive exoskeletons are used for patients who have a permanent disability, such as an amputated limb, or a muscular degenerative disease. Both of these systems can greatly improve quality of life. Spinal cord injuries (SCI) are one of the most common reasons for paralysis. In 2010, Berkeley Bionics unveiled their latest exoskeleton development "eLEGS". This system allowed users who suffer with mobility disorders, or are paralysed, to stand and walk. This light weight (20.4 kg), lower limb exoskeleton structure allows wearers to walk upright with little physical exertion. To control the system, the user's intent is measured and this is then passed through a proportional–integral–derivative (PID) feedback-loop found on a micro controller. The output signal is then used to drive the motors in the system thus facilitating movement. [11] Dynamically driving a machine that will assist it the actuation of the human body is an extremely complicated problem. [12–14] Ensuring that muscles are helped to move, using a similar pattern to the human central pattern generator (CPG), [15] and ensuring that the system is lightweight and simple to use, prove to be some of the largest challenges in creating devices of this nature. A team at Columbia Engineering developed a Robotic Spine Exoskeleton (RoSE), designed to assist patients with spinal abnormalities. [3] Spinal deformities are usually treated by using a fixed brace around the torso and hips to correct the abnormality. [16] Studies showed that the device allows for the three dimensional forces of the human torso to be mapped via force and position sensors mounted to the actuators, allowing for control methodology known as impedance, which is a method of control which simply regulates the position or force of a system. The RoSE, consists of three rings placed on the pelvis, mid-thorax, and upper-thorax. These rings are connected by 6 pairs of motors to produce a total of 12 degrees of freedom. By using a PID controller, the system can apply corrective forces in any direction, focussing on specific areas of the spine whilst also having freedom of movement in ways which do not affect it. [3] Since the early 1980's when Charles Hull introduced the world first 3D printer, 3D printing has

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become a major interest in the research and development environment. [17] These machines have become widely available due the massive reduction in cost and increased availability of low cost technology. With the development of high performing source materials, researchers have begun use of 3D printed materials for orthotic and prosthetic 2 applications. [18–20] Zuniga presented a study that examined the use of Polylatic acid (PLA) as a material for manufacturing of low cost prosthetic limbs. The study successfully demonstrated the use of PLA for the manufacture of prosthesis could serve as an alternative to conventional high priced material, however, the conclusions did highlight a problem in that PLA has poor thermal performance and noted that fluctuations in heat can lead to components failure. [21] In an effort to address the low thermal performance of the material Bo-Hsin presented an improved PLA filament by modifying the –NCO (Cyanate) and the –OH (Hydroxide) ratio of the material, resulting in glass transition temperature (Tg) improvement from the standard 55oC to 64oC. [22] With these advancements further development into the use of 3D printed materials within the medical world have been untaken, more specifically single limb orthoses and upper limb prostheses. [23] By the use of 3D printing the manufacturing of components, the exoskeleton has the potential to offer an improved quality of life for millions of people around the world. The use of exoskeleton technology is in it's infancy with the problems of accessibility preside, at present the costs of these devices can be as high as $45,000 per system. [24] Offering these types of technology's to low income countries can prove to be extremely difficult. One possible way of lowering the price of these skeletons is to consider the costs for manufacturing and ease of maintenance, by incorporating the use of 3D printing technologies. Typically exoskeletons are made from rigid, structured materials such as aluminium. [5, 25–28] Recently, 3D printing has proven to be a tool that has the potential to impact the field of prostheses and orthoses. [20, 29] Similar to traditional methods, the use of 3D printing allows the designer to offer a tailored fit to the patient's limb or amputated area. 3D-printed objects can be made from Polylactic acid (PLA), which is a cheap and strong alternative to a traditional material, such as Aluminium. [30] This paper presents the use of 3D printed materials for exoskeleton structures. The work forms part of a larger exoskeleton project that we have named "E'ssist".

II. Design

The initial design challenge for the exoskeleton was to develop an actuating system that assist in the extension of the neck. The system has been designed to be used in conjunction with a neck support wrap. Two Actuonix linear actuators where used to provide alternative to the typical muscle strength, each motor was attached to a connector which in turn provided the assistive moment to the head. The position of the connectors were deemed one of the most important elements of the prototype exoskeleton, as wrongly positioned actuation points could lead to further muscle damage and/or injury to the subject. The trigger points of the trapezius muscle were used as key areas for actuation. Work noted by Palastanga confirmed the muscle is the main point of head support for mobility. [31] The exoskeleton under development is predominantly aimed at children with muscular deficiencies, and therefore a user height range of 100 cm 3 to 180 cm was chosen and the system was designed accordingly. The strength range that influenced motor selection was based on a study by Hosking, where 19 children aged between 4 and 13 years of age were subject to a series of strength tests. [32] In this work, only the upper trapezius muscles were assessed through "Neck Flexor" examinations. Further work from Sandercock confirmed these examinations remain relevant. [33] A study completed by Villia explicitly examined skeletal muscle function in children, at the age of 11 years 33. The results from both studies agreed, and they are listed in Table 1.

Table 1: Neck Muscle Strength

Muscle Group	Min (height = 100 cm)	Max (height = 170 cm)
Trapezius	20 N	80 N

To verify the Trapezius results noted in these studies, the human head force (HF) was modelled as a Class 1 lever. The centre of gravity (CG) relative to the pivot point around the atlas vertebra acts at a displacement of roughly half of the force lever arm.

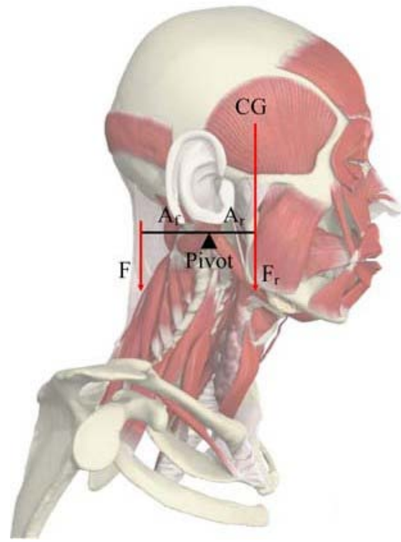

Figure 1: First Class Lever Human Head

Figure 1 shows the human head, for which a Class 1 lever is assumed to be an appropriate model, where A_r, A_f ,F_r and F are the resistance arm, the force arm or the lever, resistance force of the human head and force generated by the muscle to operate the lever, respectively. Palastanga [31] suggests that the force and the resistance arms are directly related to each other, thus the resistance arm of the lever is calculated as:

$$A_r = \frac{A_f}{2} \quad (1)$$

Where A_f and A_r is the distance from the atlas vertebra to the point of muscular interface and resistance arm, respectively. The mass of the head is assumed to be in the range 4 kg - 5 kg.

$$H_F = \frac{F_r A_r}{A_f} \quad (2)$$

Based on this, a pair of linear actuators were used to share the loading of H_F. The motors performance rating is noted in Table 2.

Table 2: Linear Actuator

Motor	Force (N)	Key point muscle group area
L12 Micro Linear Actuator	42N	Trapezius

Table 3: Material Masses

Compnents	Material	Mass (g)
Top Segment	PLA	54.1
Middle Segment	PLA	26.6
Bottom segment	PLA	68.6
Connectors x 4	PET-C	14
Actuators x 2		56
Total mass		317.3

The design of the structure is shown in Figure 3.

To enable smooth transition from the force delivered from the motor through to the body, a novel connection method was proposed Figure 2.

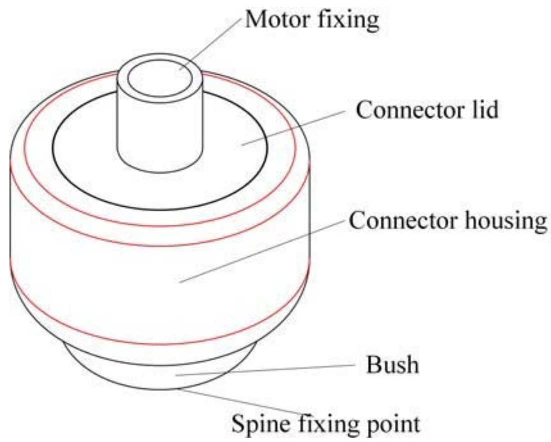

Figure 2: Connector Assembly

The connector ensured that the muscle did not receive direct force from the motor as this may cause damage to the subject (Figure 6). This also added a three degrees of freedom relationship between the spine segment and the linear actuator connecting point. Both connector lid and housing were made from a Polyethylene terephthalate with carbon mixture (PET-C). These components were 3D printed with 40 % infill of material with a triangular support structure. Both the bush and the central cords were Nitrile rubber (NBR).

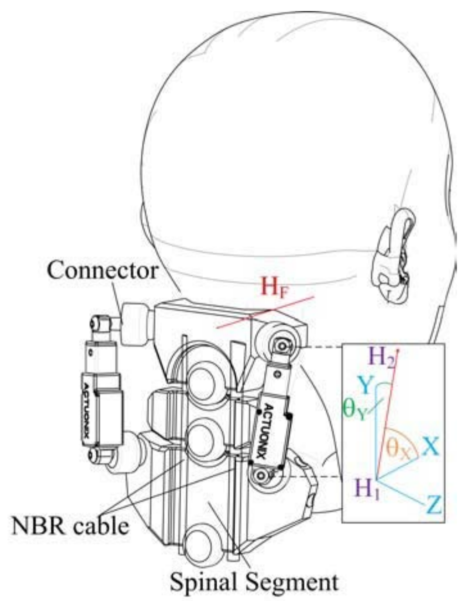

Figure 3: Neck Exo-Structure

By taking the precise locations for H_1 and H_2 which are the coordinate locations of where the actuators connect respectively, it was possible to calculate the vectors, representative of the actuation during operation, as follows:

$$F_x = 42N sin\theta_x \tag{3}$$

$$F_y = 42N sin\theta_y \tag{4}$$

Where F_x and F_y are the vector force for both x and y components respectively. The maximum output from the actuators used is 42 N and where ϑ_x and ϑ_y are the directional vector angles in respects to H_1 and H_2. To consider the variation in neck muscular assistance, the simulations were run with a variation of H_F. 2 N assumes that 90% of the muscles are actively supporting the mass of the head and at 20 N it was assumed that there was no muscular assistance, hence all support drive was provided by the exoskeleton. From the minimum value Increments of 10% in H_F were studied. As noted by Abbot, when simulating 3D printed materials it is good practice to verify the material used within the simulation. [34] To verify the material values used within the simulation the deformation and stress concentration were calculated using the deflection of a cantilever. This is represented as:

were calculated using the deflection of a cantilever. This is represented as:

$$y = \frac{PL^3}{3EI} \tag{5}$$

Where y, P, L, E and I are the maximum deflection, load, length, modulus of elasticity and moment of inertia, respectively.

$$\sigma_{bend} = \frac{Py}{I} \tag{6}$$

Both equations were applied in a simple bending test to verify the material data being used in the simulation. As the spinal structure is connected with NBR cords, hyper elastic phenomena will occur during this operation. One of the most popular models for the extension of elastic material is Storakers equation [35], more specifically for this work the Ogden model was used [36], which operates by the materials strain energy function:

$$\omega = \sum_{k=1}^{N} \frac{2\mu_k}{\alpha_k^2}\left[\lambda_1^{\alpha_k} + \lambda_2^{\alpha_k} + \lambda_3^{\alpha_k} - 3 + \frac{1}{\beta_k}\left(J_{el}^{-\alpha_k\beta_k - 1}\right)\right] \tag{7}$$

Where λ_1, λ_2 and λ_3 are the distortional principal stretches of the material. ω is the strain energy fuction, J_{el} is the elastic ratio of the material, μ_k, α_k *and* β_k are functions of the material properties. When simulating the hyper elastic material in an FE package, these parameters need to be provided as an input, and the values used in this work are taken from. [37] As the notation of the bush on the connector is subject to rotational strain, we used the Mooney-Rivlin two parameter model, as this model great shear behaviour. [38] The Mooney-Rivlin model is commonly seen as an extension to the to the Neo-Hookean model, by providing a greater accuracy on the elastic strain coefficient I_2^* variable by incorporating the use of the Helmholtz free energy per unit reference.

$$\Psi\left(C_{10}, C_{01}, \kappa\right) = C_{10}\left(I_1^* - 3\right) + C_{01}\left(I_2^* - 3\right) + \frac{\kappa}{2}\left(J - 1\right)^2 \tag{8}$$

As noted in equation 8, the nearly incompressible Mooney-Rivlin method operates with three key properties of the material. C_{10} is the representation of elastic behaviour of the material, C01 this is the none linearity component of the material, and κ is the bulk modulus. Showing the connection between Strokers and Mooney in $I_n = \lambda^2 + \lambda^2 + \lambda^2$. To ensure our results remained accurate, the 3D printed components were modelled in Autodesk Inventor with an internal geometry representative of the infill structure.

The test equipment was produced on an Ultimaker S5, both the connector and spine segments were produced with a 0.4 mm extruder head and 0.1 mm layer thickness. The 6 mm NBR cord was purchased and cut to size. Each connector was attached to the spine using a NBR bush Figure 4. The assembly was controlled through an Arduinouno and coded to cycle to maximum extension and maximum contraction. All cycles were run at 0 – 3 seconds. At the connection point between the segment and the connector a BF350 strain gauge was placed.

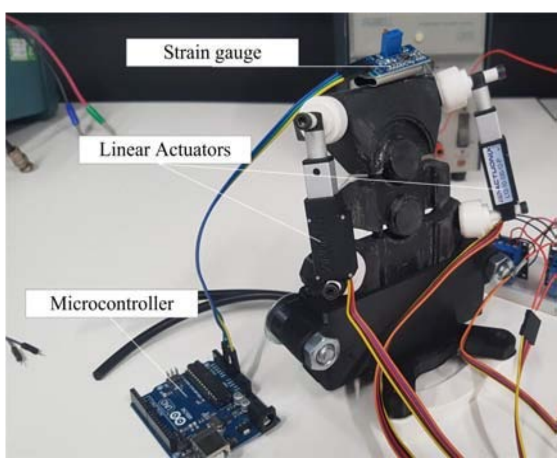

Figure 4: Neck Spine test Equipment

III. Results

Simulation of the neck assembly was performed with COMSOL Multiphysics and validated with experimental data using BF350 strain gauge and Arduino Uno on the linear actuator connector. Von Mises Stress are calculated because this is a way of testing to failure of the material.

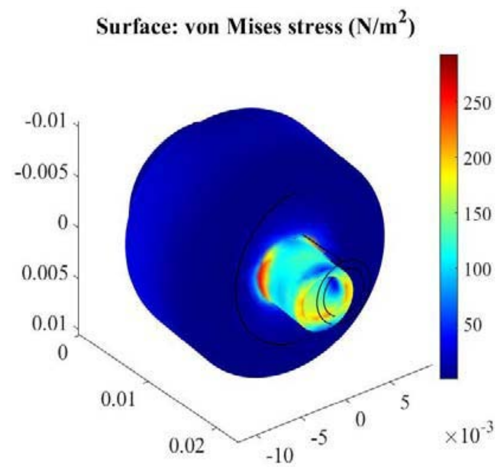

Figure 5: Connect Force Distribution

Figure 5 shows the connector after running the test at the maximum loading of 42 N. The resulting deformation and surface stress shows a maximum of 250 9 N/m^2 at the point of actuator connection and also shearing force around the Nitrile interface. As the linear actuators are assumed to be in maximum operation, these results have been applied to the spinal segments in the simulation of all head force variations.

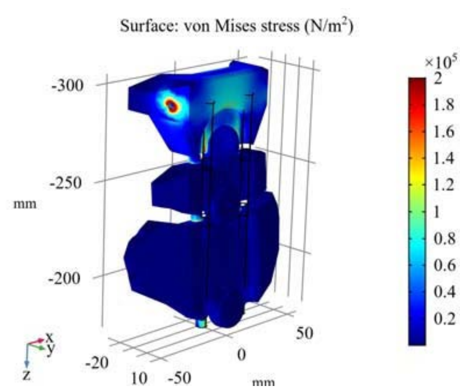

Figure 6: Neck Assembly with 42 N Linear Force

Figure 6 shows the neck in an isometric view aimed at the front and the rear of the assembly. The top segment that interfaces with the head and spine connection shows the most concentrated area of stress. This is located in particular around the NBR connections and the linear actuator interfaces. The maximum stress is noted to be be 2x10^5 N/m^2 which remains well within the Young's modulus of both PLA and the nitrile. Further testing was run to examine the neck assembly in variant conditions simulating situations where a subject would be capable to offering differing degrees of self-actuation of neck muscles.

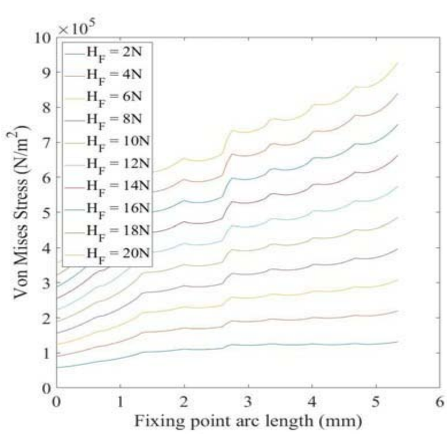

Figure 7: Neck Stress around Mounting Hole

Figure 7 shows the results from the simulation when varying the head force in the range of 2 – 20 N. When the H_F is tested at 20 N this assumes no assistance from muscular support, hence the exoskeleton is holding the full weight of the head. The results from the simulation show that the exo-skeletal structure receives below $9.5x10^5$ N/m^2 of stress.

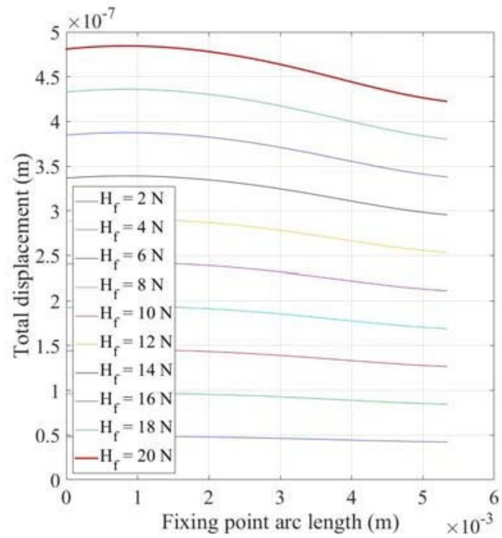

Figure 8: Top Joint Displacement

Figure 8 shows the displacement of the top joint after the range of H_F being applied. This showing that when maximum H_F is applied at 20 N the maximum deformation of the joint is 480 μm and demonstrating a linear relationship between H_F and displacement.

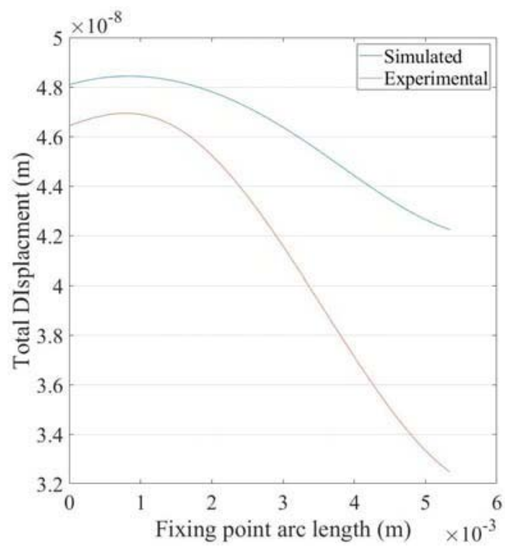

Figure 9: Simulated Vs Experimental Result

Figure 9 shows the results from the physical test equipment and also the simulation at head force of 2N. The simulation deformation begins at $4.8x10^{-8}$ m and ends above $4.2x10^{-8}$ m, the physical results show above $4.6x10^{-8}$ m and ends above $3.2x10^{-8}$ m, showing $1x10^{-8}$ m difference between both outputs.

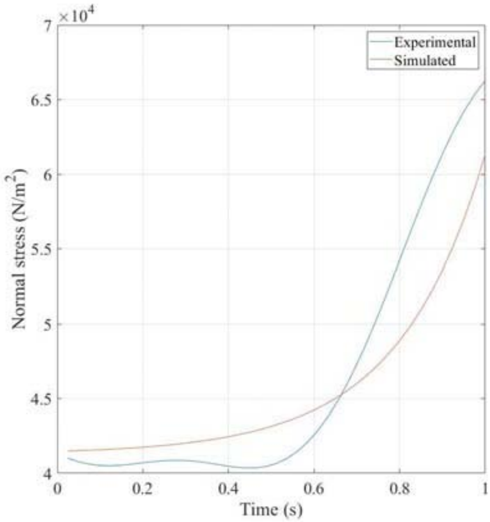

Figure 10: Caption

Figure 10 shows the results from the physical and simulated test result of H_f of 2N. The simulated stress begins 4.2x104 N/m2 of force and ends above 6x104 N/m^2. The experimental results beneath 4.2x104 N/m2 and end above 6.5x104 N/m^2, showing almost a 0.5 N/m^2 difference between the two results.

IV. Discussion

In all simulations and experiments, the linear motors where actuated to their maximum displacement of 10 mm resulting in an arc length of the upper edge of the top segment of 5 mm, A resistance force representing the assistance or lack of assistance from a subject was varied between 2-20 N, and the resulting surface stress was simulated, firstly within the connector with an actuator force directly applied and secondly as part of a complete assembly of the neck prototype. As the connector was produced using PET (Carbon) the 250 N/m2 stress exhibited are well below the yield stress of 2.7 GPa published for this material. [39] The stress predicted within the Nitrile rubber bush are also well within the tolerable limits published for this material. The results from the assembly simulation demonstrate the largest stress concentration are found in the upper segment where the mass of the subject's head would be supported. The concentrated forces in this segment result in stresses as high as 2 x 105 N/^{2} but once again, these lie well within the published yield stresses of PLA. Head force variation applied at 2N increments of resistance force, which assumes minimal assistance to the patient, the result demonstrate a linear relationship between deflection and Head Force 480μm - 48μm, however a linear relationship between max stress and HF is not observed directly only through calculation of 13 rate of change of H_F with respect to stress is a linearity observed. To validate the results from the neck exoskeleton, the system was manufactured and run in the same conditions. The linear actuators where moved by the full 10 mm and a resistance force was placed on to the neck of 2 N. When compared against the simulation results the initial starting displacement shows almost over 0.166 x 10^{-8} m difference between the simulated material and the manufactured material. The elastication of the material also shows a greater displacement than what is seen in the simulation, noting a difference of 0.976 x 10^{-8} m. The normal stress between the simulated and the manufactured material showed a maximum of 10% difference. As the material used within the simulation has been selected based on values presented within the COMSOL software, the variation of results could be an indicator that the PLA used has a slight greater performance as the one presented within the software. The experimental results show good corroboration with those from COMSOL simulation, error is deemed acceptable. The double nitrile lines which continue up the spine hold similar stress concentration to what is seen within the top section of the spine however, the patent will be unable to perform all ranges of motion, as head tilt is unachievable.

V. Conclusion

This study examines the use of 3D printed materials for use in wearable robotic structures. The human neck is a complex series of connected segments which work together to create a system that allows for 6 degrees of freedom (DoF). To ease the process of design this study has focused on the pitch element of human head motion (looking up and down). This study has demonstrated that 3D printed structures are suitable for use with exoskeleton components and the materials chosen here for this prototype design are able meet the physical demands expected during operation. Additionally, these results have also demonstrated this prototype design is over engineered while at the same time providing a suitable method of refining design to reduce materials used and maintain crucial structural integrity.

VI. Further Work

To further this work testing of the raw material to produce a detailed mechanical data base is recommended. With information of this nature the error difference between modelled and tested would become less. As the material is show to be able to support the body for assistive movement it is suggested that this work be continued to the lower spine section to examine the use of the structure on the torso area. Further to the design it is suggested that the duel nitrile lines be revised to a design which contains a single feed as full mobility of the neck and head could not be achieved in this solution.

Author Contributions

All authors contributed equally in the preparation of this article.

Declaration of Conflicting Interests

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Entropic and Statistical Analysis of Existed Refrigeration Plant for Retail

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The paper presents results of analysis existed refrigeration plants used for retail applications.

The ESMA was applied to investigate some operating refrigeration plants equipped with monitoring system and define elements for further optimization. The initial date for analysis were date taking from monitoring system (pressures and temperatures in working points). The description of calculation method was given.

The degree of thermodynamic efficiency was calculated.

This result could not be achieved using a traditional method of comparison refrigeration systems by using a coefficient of performance (COP) or seasonal efficiency as an only efficiency criterion.

The prospect of the ESMA application to analyse operation and optimization of existing refrigerating systems as well as to design new refrigerating systems consists in that it allows refusing expensive equipment for improvement refrigeration plant efficiency.

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GJRE- A Classification: *LCC: TJ163.5*

ENTROPICANDSTATISTICALANALYSISOFEXISTEDREFRIGERATIONPLANTFORRETAIL

Strictly as per the compliance and regulations of:

Entropic and Statistical Analysis of Existed Refrigeration Plant for Retail

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I. Introduction

Power consumption of refrigeration plants makes 48% to 60 % from total power consumption of retail store.

Because of ozone depletion and global warming risks, we ought to use new ecological friendly refrigerants, which require improving refrigeration components, processes and cycle architecture to increase energy efficiency.

We need to use methods that could help to improve energy efficiency based on scientific methods.

There are some methods that used for thermodynamic analysis – energy method, exergy method and entropic and statistical method.

Energy method of analysis based on first law of thermodynamic and does not allow getting information about energy losses by system components. Exergy and entropic and statistical method based both on first and second laws of thermodynamic, but usage of entropic and statistical methods for system analysis used for cold generation is preferable.

In this paper introduced usage of entropic and statistical method of analysis for refrigeration plants operation.

II. Entropic and Statistical Analysis of Refrigeration Plants

a) Entropic and Statistical Method of Analysis

Real processes in refrigeration plants are irreversible and nonequilibrium.

The reason of irreversibility is finite difference of mass streams potentials (temperature difference, pressure difference etc.) due to processes nonequilibrium.

The measure of irreversibility is entropy production. According to second law of thermodynamics:

$$dS \geq \frac{dQ}{T} \qquad \text{Eq. (1)}$$

The entropy has property of additivity e.g. total entropy production of the system equal to the sum of entropy change of each sub-system.:

$$\Delta S = \sum_i \Delta S_i \qquad \text{Eq. (2)}$$

where $\sum_i S\Delta_i$ - entropy production in sub-systems, *i* – sub-systems quantity.

According to Gyui-Stodola's theorem, for it is necessary to perform work for entropy production compensation. This work will be transferred to environment as heat. The total work spent for refrigeration plant operation will equal:

$$L_{tot} = L_{min} + \Delta L \qquad \text{Eq. (3)}$$

L_{min} – minimum work for ideal refrigeration cycle with total reversible processes.

Distribution of energy expenses for entropy production compensation for one stage refrigeration cycle is shown on Figure 1.

Ideal Carnot cycle (c1-c2-c3-c4) is totally reversible, it means that work, which is spent for cold generation, will be minimum and is expressed by the area c1-c2-c3-c4. Real cycle is expressed by the area 1-2-3-4-5-6. Work, which is necessary for entropy production compensation in compressor, is expressed

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by area d-e-e3-e2. Area e3-2-3-4-e1 or equal to it in size e-f-e4-e3 expresses work, which is necessary for entropy production compensation in condenser. Work, which is necessary for entropy production compensation in evaporator, is expressed by area c-d-e2-c2, in throttling devices – by area a-b-c3-e1.

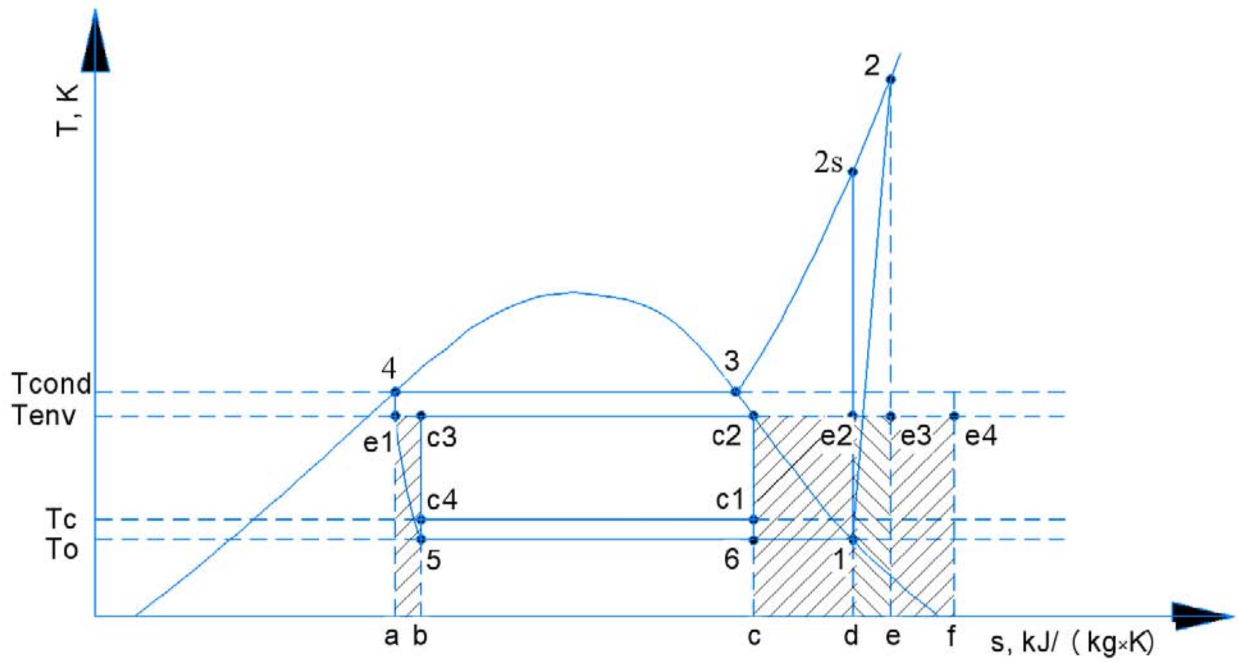

Figure 1: Distribution of Energy Expenses for Entropy Production Compensation by Elements of Single Stage Refrigeration Cycle

Total real work for single stage refrigeration cycle is expressed by the sum of areas.

$$L_{tot} \propto (c1\text{-}c2\text{-}c3\text{-}c4) + (d\text{-}e\text{-}e3\text{-}e2) + (e\text{-}f\text{-}e4\text{-}e3) + \\ + (c\text{-}d\text{-}e2\text{-}c2) + (a\text{-}b\text{-}c3\text{-}e1) = (1\text{-}2\text{-}3\text{-}4\text{-}5\text{-}6) \propto (h_2 - h_1) \quad \text{Eq. (4)}$$

Analysis of the single stage refrigeration cycle analysis will be carried out in the following sequence.

Specific mass cooling capacity at evaporation temperature.

$$q_o = h_1 - h_4 = h_1 - h_5 \quad \text{Eq. (5)}$$

Minimum specific work which is necessary for cold generation.

$$l_{min} = q_o \times \frac{T_{env} - T_c}{T_c} \quad \text{Eq. (6)}$$

Adiabatic compression work, calculated by using h-lgp diagram:

$$l_s = h_{2s} - h_1 \quad \text{Eq. (7)}$$

Actual specific compression work.

$$l_{comp} = q_{cond} - q_o = h_2 - h_4 - (h_1 - h_4) = \frac{l_s}{\eta_s} \quad \text{Eq. (8)}$$

Degree of thermodynamic efficiency.

$$\eta_{therm} = \frac{l_{min}}{l_{comp}} \quad \text{Eq. (9)}$$

COP at adiabatic compression.

$$\varepsilon_s = \frac{q_o}{l_s} \quad \text{Eq. (10)}$$

Actual value of COP.

$$\varepsilon_{act} = \frac{q_o}{l_{comp}} \quad \text{Eq. (11)}$$

Part of compression work, which is necessary for entropy production compensation in condenser, is the sum of parts of compression works for entropy production compensation in gas cooling process Δl_{sh} and condensation process Δl_{cd}:

$$\Delta l_{cond} = \Delta l_{sh} + \Delta l_{cd} \quad \text{Eq. (12)}$$

where

$$\Delta l_{sh} = (h_{2s} - h_3) - T_{env} \times (s_{2s} - s_3) \quad \text{Eq. (13)}$$

$$\Delta l_{cd} = T_{env} \times (h_3 - h_4) \times \left(\frac{1}{T_{env}} - \frac{1}{T_{cond}}\right) \quad \text{Eq. (14)}$$

Part of compression work, which is necessary for entropy production compensation in throttling process.

$$\Delta l_{thr} = T_{env} \times (s_5 - s_4) \quad \text{Eq. (15)}$$

Part of compression work, which is necessary for compensation of entropy production in heat transferring processes from cooling object to refrigerant (evaporation).

$$\Delta l_{e.evap} = (h_1 - h_5) \times T_{env} \times \frac{T_c - T_o}{T_o \times T_c} \quad \text{Eq. (16)}$$

Part of compression work, which is necessary for compensation of entropy production in heat transferring processes from cooling object to refrigerant (total).

$$\Delta l_{evap} = \Delta l_{e.evap} \quad \text{Eq. (17)}$$

Specific adiabatic compression work is a sum of parts of compression work for compensation of entropy production in all processes of refrigeration cycle.

$$l_{s.calc} = l_{min} + \Delta l_{cond} + \Delta l_{thr} + \Delta l_{evap} \quad \text{Eq. (18)}$$

Energetic losses in compressor.

$$\Delta l_{comp} = l_{comp} - l_{s.calc} \quad \text{Eq. (19)}$$

Rated compression work.

$$l_{comp.calc} = l_{s.calc} + \Delta l_{comp} \quad \text{Eq. (20)}$$

By using analysis results, the chart of distribution losses on system elements can be constructed.

b) Refrigeration Plant Analysis in Case of Condensation Pressure Algorithm Change

One of the goals, which is very important to refrigeration equipment end user, is to evaluate results of implementation of new technical solution (usage of more efficiency compressors, heat exchangers, new control algorithms etc).

The goal of refrigeration plant optimization (it is located in Moscow, Russia) was energy consumption reduction. This plant work with R404A as refrigerant and used for 10 medium temperature consumers. Design date is shown in Table 1.

Table 1: Design Date of Refrigeration Plant

Cycle	Single Stage
Refrigerant	R404A
Evaporation temperature, °C	-10
Condensation temperature, °C	+45
Suction gas superheat, K	15
Liquid subcooling, K	0
Condenser type	Air cooled
Cooling capacity, kW	33,8
Quantity of compressors, pcs.	2
Compressor type	scroll
Model	ZB76KCE-TFD - 1 pcs. ZBD76KCE-TFD – 1 pcs.
Consumers quantity, pcs.	10
Thermal processing	Storage
Product type	Egg (1 consumer), milk products (6 consumers), sausages (2 consumers), fish (1 consumer)

There were two measurement periods – 4.5 days with fixed condensing set point (condensation pressure was keeping on the same value) and 4.5 days with floating condensing set point (condensation pressure was changing depending on ambient temperature).

Initial date (average values for considered period) is given in Table 2, changing of operating parameters is shown on Figs. 2 and 3, where System 1 is refrigeration plant before algorithm change, System 2 is refrigeration plant after algorithm change.

Table 2: Initial date for Analysis

Temperatures, ºC	System 1	System 2
Evaporation	-9.0	-9.07
Condensation	+34.79	+19.33
Ambient	+8.51	+8.18
Air in cooling volume	+1.95	+2.1
Adiabatic efficiency was taken from selection software of compressor manufactures		
Adiabatic efficiency	0.7117	0.6661

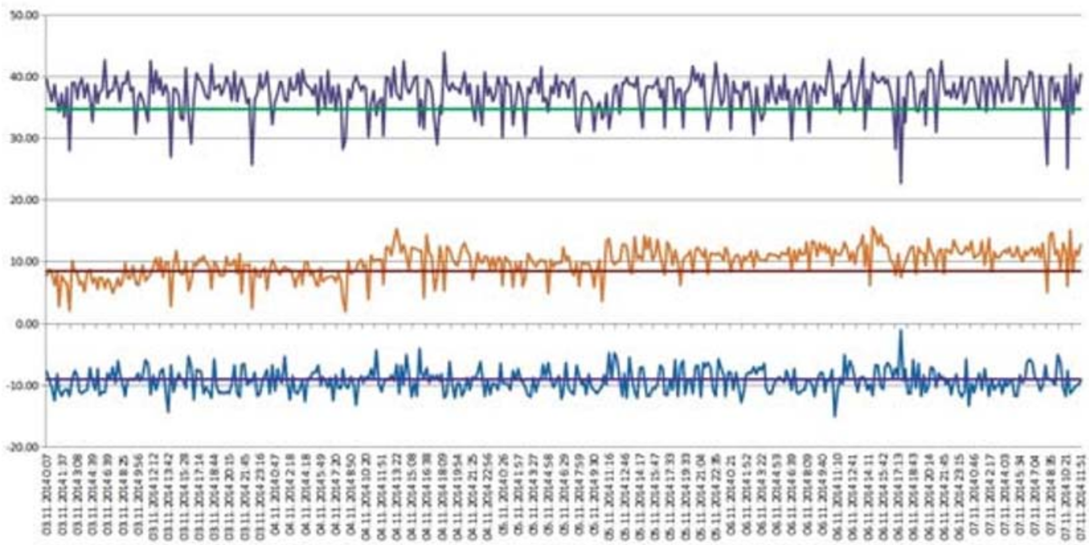

Figure 3: Schedules of Change of Condensation Temperature (Top Schedule), Ambient Temperature (Middle Schedule), Evaporation Temperature (Lower Schedule) and Their Average Values (Horizontal Lines) For System 1

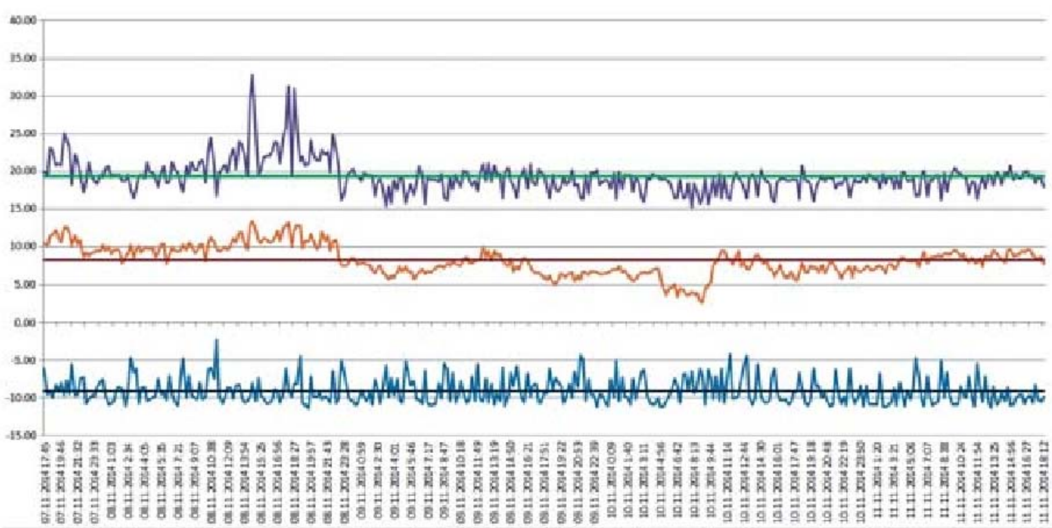

Figure 4: Schedules of Change of Condensation Temperature (Top Schedule), Ambient Temperature (Middle Schedule), Evaporation Temperature (Lower Schedule) and their Average Values (Horizontal Lines) for System 1

Because there was not possibility to measure all necessary values by using regular devices, some values were taken according to statistical information:

- Total superheat on suction line 12 K;
- Calculation of energetic losses in evaporator was done by using average value of cooling air temperature in all cooling room.

Analysis results are given in Table 3 and on Fig. 5.

Table 3: Analysis Results

-	System 1	System 2
q_o, kJ/kg	119.71	144.52
l_{min}, kJ/kg	2.88	3.19
$l_{s.calc}$, kJ/kg	27.96	19.32
l_{comp}, kJ/kg	39.24	29.00
η_{therm}	0.0795	0.1184
ε_{act}	3.05	4.98
Δl_{cond}, %	34.45	23.98
Δl_{thr}, %	16.47	10.29
Δl_{evap}, %	12.91	21.32
Δl_{comp}, %	28.83	33.4

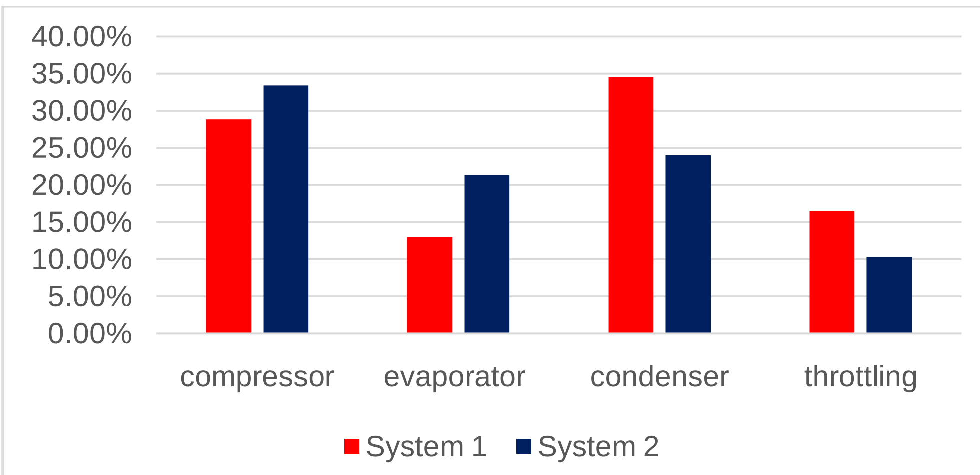

Figure 5: Part of Compression Work (Energetic Losses) which is Necessary for Compensation of Entropy Production in System Elements for Refrigeration Plant before and after Optimization, % from Compression Work

c) *Efficiency Analysis at Refrigerant Plant Operation with Different Working Cycles*

One more and, perhaps, more relevant task is the analysis of various systems with different working cycles.

Comparison of two systems is given below. System 3 works by using single stage refrigeration cycle, located in Moscow. System 4 works by using refrigeration cycle with economizer, located in Volgsky, Russia. Design date is shown in Table 4.

Table 4: Design Date of Refrigeration Plants

	System 3	System 4
Refrigerant	R404A	R404A
Refrigeration cycle	Single stage	With economizer
Evaporation temperature, °C	-35	
Condensation temperature, °C	+40	
Suction superheat, K	15	
Liquid subcooling, K	0	
Condenser type	Air cooled	
Cooling capacity, кВт	90,2	92,0
Compressor type	reciprocating	Scroll with economizer
Quantity of compressor, pcs.	3	8
Model	D8DJ-600X with inverter drive – 1 pcs., D8DJ-600X – 2 pcs.	ZF40KVE-TWD – 8 pcs.
Quantity of consumers, pcs.	39	26
Thermal processing	Frozen food storage	

As in the previous case, operation parameters were taken from monitoring system for specified period. Initial average date for analysis is given in Table 5.

Table 5: Initial date for Analysis

	System 3	System 4
Temperatures, ºC		
evaporation	-33.46	-32.58
condensation	+36.75	+31.35
ambient	+10.58	+15.98
air in cooling volume	-15.14	-16.25
Average superheat in evaporators, K	13.67	9.73
Adiabatic efficiency	0.673	0.632

Analysis results are given in Table 6 and on Fig. 6.

Table 6: Analysis Results

	System 3	System 4
q_o, kJ/kg	101.89	167.31
l_{min}, kJ/kg	10.73	22.17
$l_{s.calc}$, kJ/kg	50.67	61.91
l_{comp}, kJ/kg	75.31	97.97
η_{therm}	0.149	0.214
ε_{act}	1.35	1.71
Δl_{cond}, %	19.59	14.64
Δl_{thr}, %	22.58	9.41
Δl_{evap}, %	10.88	13.2
Δl_{comp}, %	32.7	36.78
Δl_{eco}, %	-	3.34

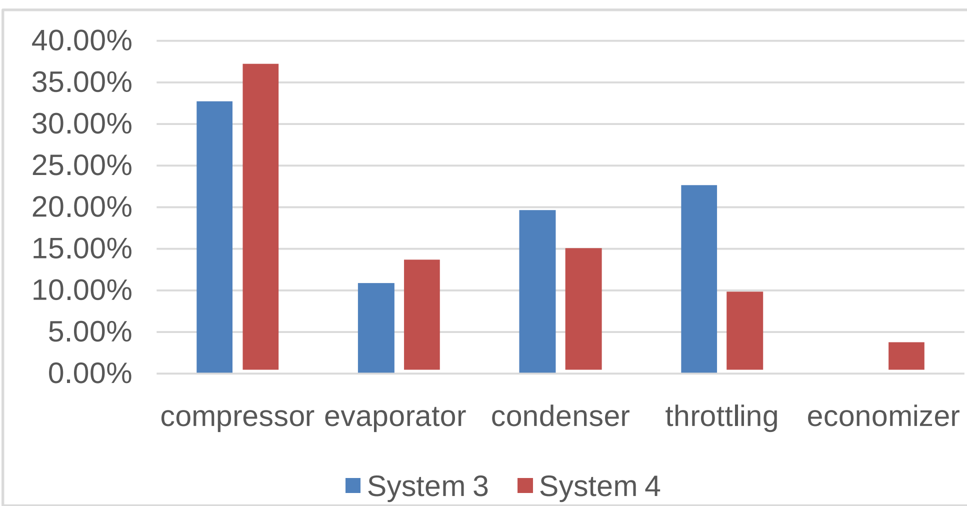

Figure 6: Part of Compression Work (Energetic Losses) which is Necessary for Compensation of Entropy Production in System Elements for Refrigeration Plant before and after Optimization, % from Compression Work.

III. Conclusions

Application of entropic and statistical method of the analysis during cooling plant operation allows increasing energy efficiency because we can get information about losses in different refrigeration plant components and take measures to increase their operation efficiency. It is possible to use regularly installed sensors.

The most appropriate value for comparison of energy efficiency different cooling plants is the degree of thermodynamic efficiency.

Changing control algorithm of condensation pressure from maintenance of the fixed setting to floating setting allow to increase the degree of thermodynamic efficiency on 32.9% and decrease power consumption on 36.7%. Power consumption was measured by using special sensors.

The degree of thermodynamic efficiency refrigeration plant working with refrigeration cycle with economizer is higher than the degree of thermodynamic efficiency refrigeration plant working with single stage refrigeration cycle on 30.37%. Date was received during operation of real refrigeration plants.

Nomenclature

Q	heat (W)	q_o	specific mass cooling capacity (kJ/kg)
s	specific entropy (kJ/kg/K)	l_{min}	minimum specific work which is necessary for cold generation (kJ/kg)
h	specific enthalpy (kJ/kg/K)	l	specific compression work (kJ/kg)
T	temperature (K)	η_{therm}	the degree of thermodynamic efficiency
t	temperature (°C)	Δl	energetic losses (kJ/kg)
ε	COP		

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Research on Partial Reduction of Non-Essential Capabilities of Initiating Devices for Complex Mechanical Equipment Systems

By Volkov M. V., Pushko S. V., Rozhko A. A., Nadein I. O & Rozhko O. F.

Abstract- Currently, modern spacecraft use high-precision equipment that is sensitive to shock impacts. In this regard, there is a need to create initiating devices with reduced impact. This task determined the vector of development of electromechanical release devices that do not contain pyrotechnic means in their design. Along with new challenges, there are the constant problems of reducing weight, increasing holding force, and reducing response time. The paper presents an analysis of the possibility of upgrading initiating devices in order to improve its characteristics.A team of engineers conducts research using computational and experimental methods.During the work, samples of mechanisms were made and tested to confirm their performance.

Keywords: initiating device, nichrome wire, torsion spring, reduction.

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RESEARCHONPARTIALREDUCTIONOFNONESSENTIALCAPABILITIESOFINITIATINGDEVICESFORCOMPLEXMECHANICALEQUIPMENTSYSTEMS

Research on Partial Reduction of Non-Essential Capabilities of Initiating Devices for Complex Mechanical Equipment Systems

Volkov M. V. α, Pushko S. V. σ, Rozhko A. A. ρ, Nadein I. O Ѡ & Rozhko O. F. ¥

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I. Introduction

The most well-known method of holding transformable structural elements is using pyrotechnical devices installed outside the equipment area. This system results in high-impact forces.

This lack of pyrotechnical products led to the emergence of electromechanical initiating devices. [1, 2]

There are known designs of electromechanical devices [3] based on heating the structural elements of the devices when an electric current is passed through them. The most famous are devices based on:

- Materials that expand during heating, for example, paraffins.
- Fusible elements that lose their load-bearing capacity when heated;
- Heating knives that violate the integrity of load-bearing elements;

The EH-3525 is an initiator device developed by Sierra Nevada Corporation that uses a material that expands during heated. (Figure 1a)

The EH-3525 uses a heating element to melt the wax in the drive. The design diagram is shown in Figure 1b. Paraffin expands during heatingand hydraulic pressure pushes the rod out.The rod is moves back, when the paraffin cools.

Some characteristics of the EN-3525 drive are presented in Table 1.

The advantages of such devices include the high force created when the rod moves, which allows them to be used in more heavily loaded systems. However, the presence of a heated material and its transformation into a different state of aggregation forces a large amount of energy to be supplied to the device, and the response of such devices takes a long time.

Table 1: EH3525 Wax Drive Specifications

Parameter	Meaning
Weight, g	35
Response time, s	200
Nominaloutputforce, N	156

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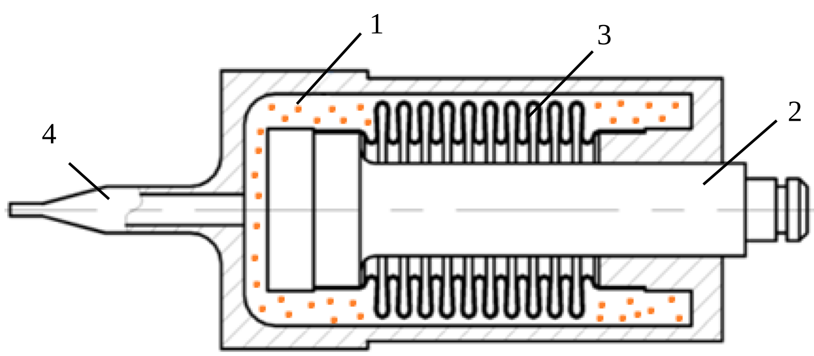
Author ρ Ѡ ¥: Design engineer, Limited Liability Company "Bureau 1440".

a



b

1 – paraffin; 2 – rod; 3 – bellows; 4 – cable

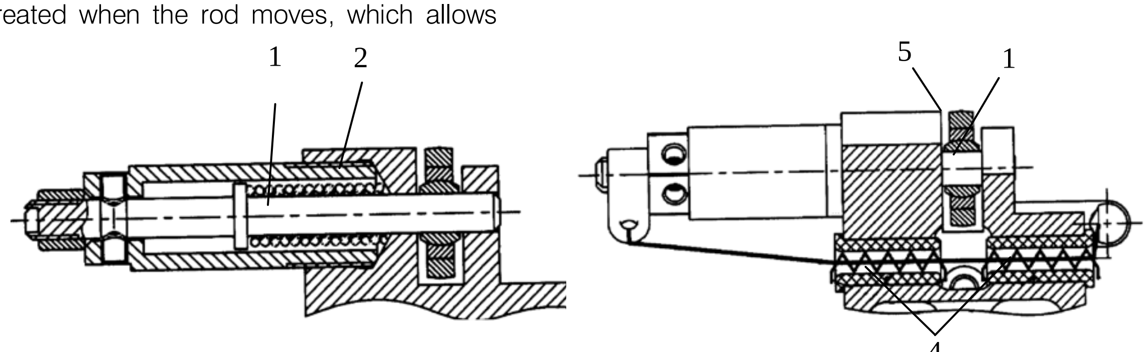
Figure 1: Initiating device based on an expanding structural element (paraffin drive EH3525)

An initiating device based on heating knives can include a device protected by patent RU 2307772 C2 (ZAO «KB «Polet») [4] (Figure 2).

The rod of the initiating device of this design is moved by a compression spring. The spring is compressed when the element being held is fixed.The rod is kept from moving using a high-strength thread, such as polyimide, Kevlar and Vectran. The device is triggered when the filament is destroyed/melted, using heating knives representing a wound incandescent spring.

The advantages of such devices include the high force created when the rod moves, which allows them to be used in more heavily loaded systems. In addition, such devices have a faster response/actuation time and consume less energy compared to initiating devices based on the principle of an expanding structural element.

However, excessive heat may cause the knife to melt. Then the initiating device may fail. This phenomenon determines high demands on knife coatings and precise regulation of the electric current supplied to the knives.



1 – Rod; 2 – Compression Spring; 3 – High-Strength Thread; 4 – Heating Knife; 5 – Retained Element

Figure 2: Initiating Device Based On Heating Knives

The characteristics of such a device, established by experimental and calculation methods, are presented in Table 2.

Table 2: Characteristics of Initiating Devices Based on Heating Knives

Parameter	Meaning
Weight, g	80
Response time, s	5
Nominaloutputforce, N	150-200

Initiating elements based on fusible elements include a device protected by patent RU 2707901 C1 (RESHETNEV), the design of which is shown in Figure 3 [5, 6].

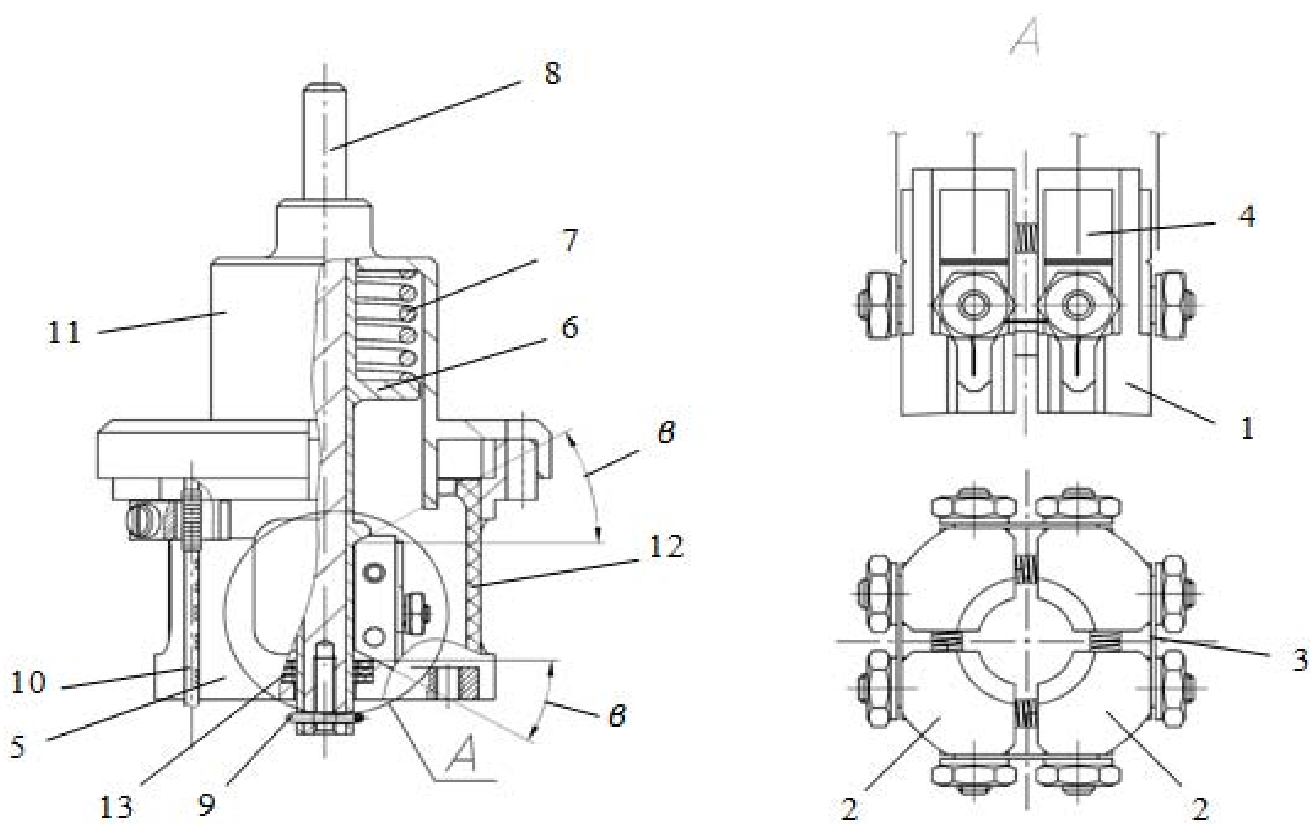
Fixation of the initiating device rod from axial movement under the influence of a compression spring is ensured by a split ring tightened with nichrome wire. When an electric current is applied through the nichrome wire, it heats up and melts/destroys, ensuring the mechanism operates. The design of the split ring ensures a reduction in the force coming from the spring, allowing the use of small-diameter nichrome wire.

The advantages of this type of mechanism include low electrical current consumption and fast response/ operation of the device.

The disadvantages of this design include the low force generated by the device, determined by the diameter of the fusible element, which is 0.14 mm. The characteristics of the initiating device are presented in Table 3.

Table 3: Characteristics of an Initiating Device Based on a Fusible Element

Parameter	Meaning
Weight, g	50
Response time, s	1
Nominaloutputforce, N	60



1 – split ring; 2 – segments; 3 – initiating element (wire); 4 – contacts; 5 – body; 6 – piston; 7 – spring; 8 – rod; 9 – wire; 10 – cable; 11 – cover; 12 – insulator; 13 – damper

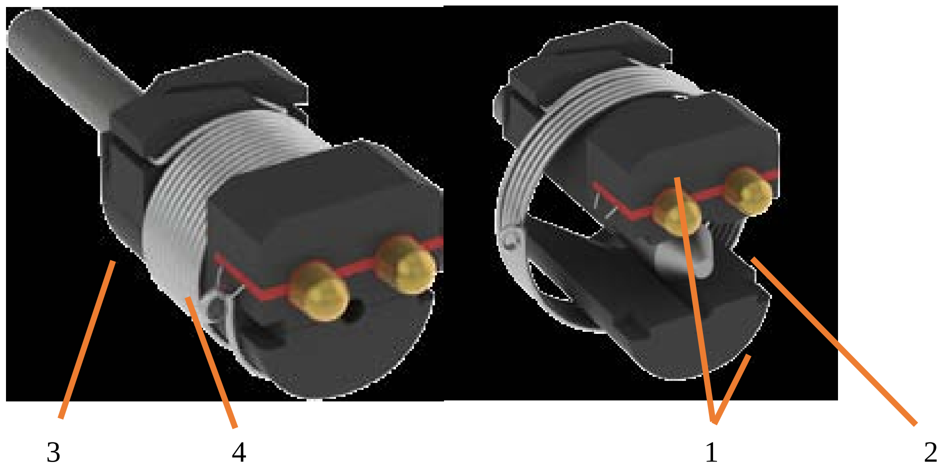
Figure 3: Design of the Initiating Device and Diagram of the Split Ring

Increasing requirements for structures determine the need for new technical solutions and combine the advantages of different systems.

The use of a fusible element makes it possible to obtain low current consumption.However, to increase the load-bearing capacity of the structure, it is necessary to increase the reduction of the system, which is held by the fusible element.

One of the solutions in this area was proposed by COOPER. Reducing the force on a small diameter wire (reduction) is achieved here by distributing the power over the coils of the tension spring.

The design of the initiating device developed by COOPER is shown in Figure 4.

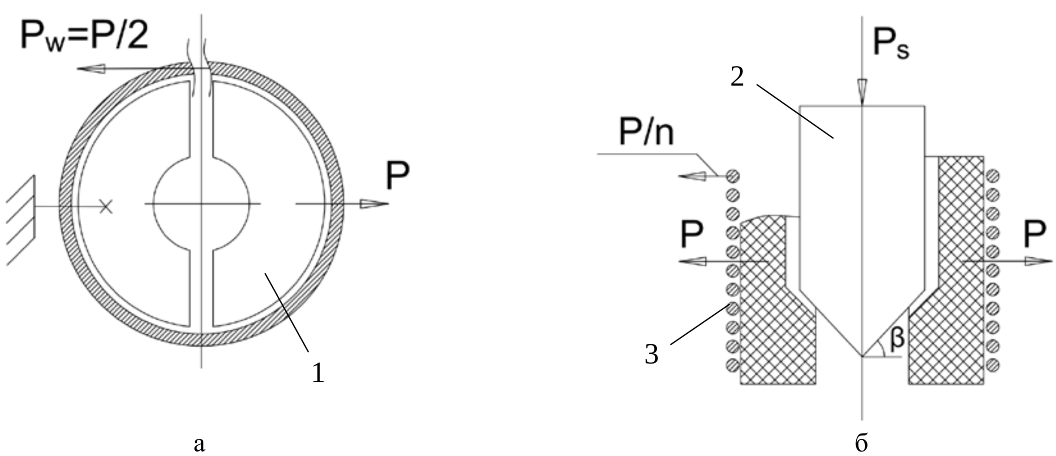


1 – body elements; 2 – rod; 3 – spring; 4 – nichrome wire

Figure 4: Design of the Initiating Device from COOPER

The body elements tightened by a torsion spring.The torsion spring is made ofa small diameter wire and is held by nichrome wire with a diameter of 0.1 mm.

Let's consider the kinematic diagram of the COOPER design [7,8]. Figure 5a shows the force distribution for one turn of the garter spring. Let us accept the assumption that the coil is a closed ring, then the reaction in the loopP_w willequal half the force *P* coming to the body element.



1 – Body Element; 2 – Rod; 3 – Spring

Figure 5: Kinematic Diagram of the COOPER Mechanism

For *n* turns, we accept the assumption that the load from the body elements is distributed evenly over all turns.Then, according to the diagram presented in Figure 5b, P_w will be calculated according to formula (1):

$$P_w = P/2n \qquad (1)$$

Let us determine the theoretical maximum permissible load on the rod P_s of a structure with a wire with a diameter of 0.1 mm made of X20N80 material. The tensile strength of the wire is 0.6 kg. When fixing the wire in like the COOPER design, we assume that the maximum force P_w will be 1.2 kg [9, 10, 11].

The maximum permissible load P_s will be calculated using formula (2):

$$P_s = 4n(P_w)tg(\beta) \tag{2}$$

According to Figure 4, the number of turns of the compressed spring is n =12, and the angle of inclination of the rod chamfer β=45°. Let us assume that there is no friction in the rod-body element pair. Thus, the theoretical maximum permissible load P_s on the device rod is 57.6 kg (≈560N).

Based on the calculation results, we calculate the device reduction coefficient k using formula (3):

$$k = P_s/P_w = 48 \tag{3}$$

The adopted design solutions and the kinematic diagram of the mechanism make it possible to significantly increase the nominal output force of the device, even in comparison with devices based on the principle of materials expanding during heating.At the same time, the electrical characteristics correspond to devices based on fusible elements.

However, to further increase the output force, a different kinematic scheme of the device was proposed and analyzed [12].

The principle on which the operation of the device is based is well known from navigation, namely from the possibilities of mooring ships (see Figure 6). When a vessel pulls on a rope wound around a bollard, the rope is pressed against the bollard, and frictional forces are generated, further holding the cord from being pulled out. Thus, minimal power is required from the free end of the have to hold it.

Figure 6: Ship Mooring

Let's consider the kinematic diagram of the developed mechanism, presented in Figure 7a, b, c.

The rod is kept from moving by wedges installed in the grooves of the body element. The wedges are tightened by a torsion spring. However, the torsion spring is wound on three sectors of the body element. Two sectors of the body element do not contain grooves for wedges. One sector contains grooves for wedges.This design allows the reduction process to be divided into two stages.

The firstreduction stage will be similar to the kinematics of the COOPER device (see Figure 7b). This calculation scheme will be valid for turns of torsion springs installed on wedges. The load P_w can be calculated using formula (4).

$$P_w = P_s/(4n_{wed} \bullet tg(\beta)), \tag{4}$$

where n_{wed} – is the number of elements installed along the end of the wedge.

The secondreduction stagecan be determined according to the scheme presented in Figure 7c [13].

Let the force $P_w(\varphi)$ be the tension force of the torsion spring, corresponding to its winding angle φ on the body element, N be the normal reaction of the body element to a section of the spring with length $\Delta S = r\Delta\varphi$. From the condition of equilibrium of the three forces $P_w(\varphi)$, N and $P_w(\varphi+\Delta\varphi)$, up to tiny values of the angle $\Delta\varphi$, the equalities follow:

$$P_w(\varphi+\Delta\varphi)\,\Delta\varphi = N, \tag{5}$$

$$P_w(\varphi+\Delta\varphi) - P_w(\varphi) = \Delta P_{fric,}\,, \tag{6}$$

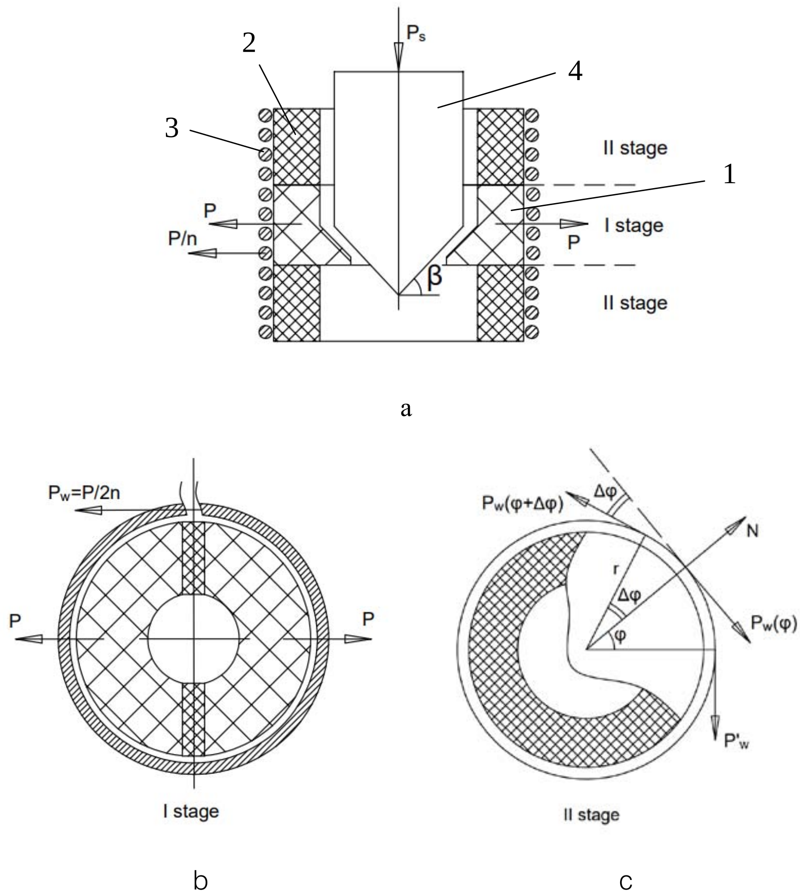
where ΔP_{fric}– is the frictional force acting on the specified element. According to the condition, $\Delta P_{fric} = \mu\Delta N$, from (5) and (6), we obtain the relation:

$$P_w(\varphi+\Delta\varphi) - P_w(\varphi) = \mu P_w(\varphi+\Delta\varphi)\,\Delta\varphi, \tag{7}$$

$$dP_w/d\varphi = \mu P_w, \tag{8}$$

$$P_w = P_w' e^{\mu\varphi}, \tag{9}$$

where P_w' – is the load of the fusible element (tension of the nichrome wire), μ – is the friction coefficient.



1 – cracker; 2 – body element; 3 – spring; 4 – rod

Figure 7: Kinematic diagram of the Mechanism being Developed

P_s is calculated by combining equations (4) and (9).

$$P_s = 4n_{wed}(P_w' e^{\mu\varphi})tg(\beta). \tag{10}$$

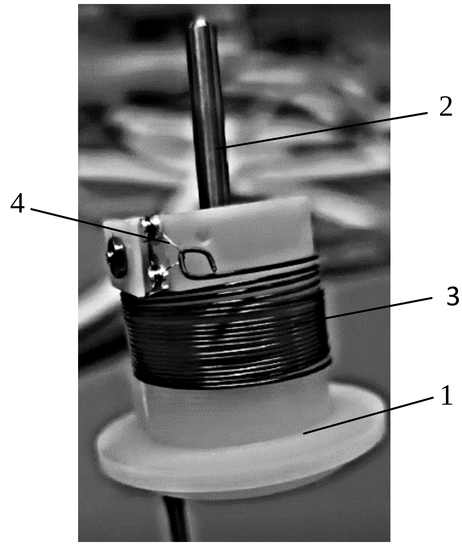
Calculate the theoretical maximum load Ps on the rod using Formula 10. We divide the distribution of the spring over the body of the device into three equal parts, then, provided the total number of turns is 12, $n_{wed} = 4$, and $\varphi = 8\pi$. The friction coefficient μ is taken equal to 0.3. The wire is fixed in a similar way to the COOPER initiating device. $P_w' = 1.2$ kg. Thus, the theoretical maximum permissible load P_s on the rod of the device being developed is 35986 kg (≈352671.6 N).

The reduction coefficient $k = P_s / P_w' = 29988$, which is 624 times higher than the kinematic diagram of the COOPER device, all other things being equal.

It was analytically determined that the selected kinematic scheme is promising regarding force reduction. The number of turns determines the degree of removal of the system while increasing the turns on the stationary part of the body element increases the reduction exponentially.

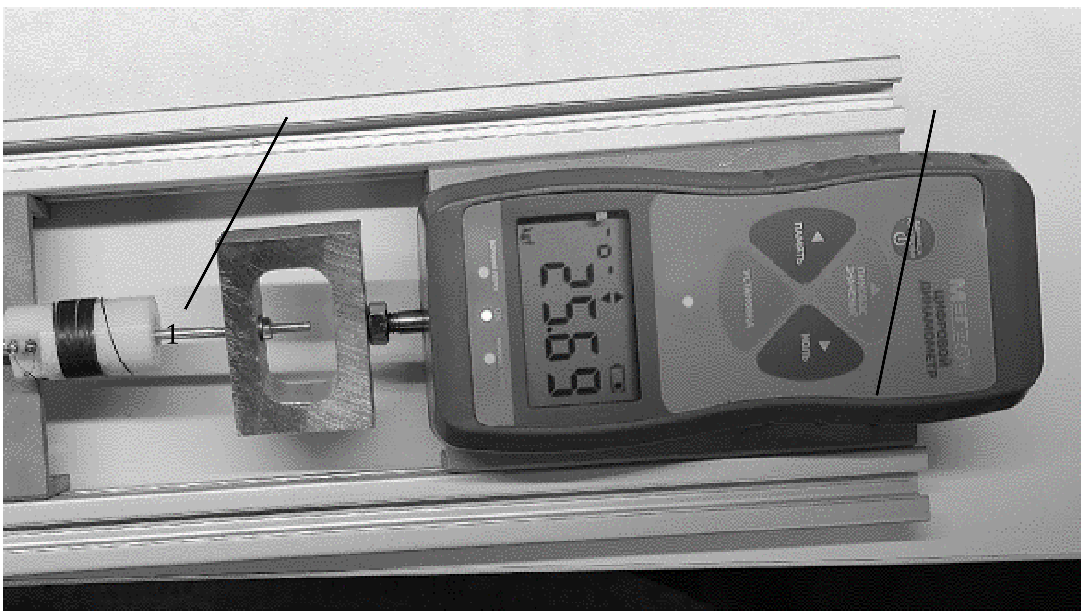
To confirm the calculations, samples of initiating devices were made, as shown in Figure 8. The spring was wound with 12 turns in the same way as in the design case.

The test scheme is shown in Figure 9 [14]. During the tests, the force that was applied to the rod of the initiating device was monitored - P_s (in the axial direction), as well as the force that was created at the end of the spring, secured with nichrome wire - P_w'.
The response time was also monitored during testing.



1 – Body; 2 – Rod; 3 – Spring; 4 – Nichrome Wire

Figure 8: Sample Initiating Device



1 – layout of the initiating device; 2 – dynamometer

Figure 9: Test Scheme for the Initiating Device Prototype

The test results are shown in Tables 4 and 5.

Table 4: Loading Results

Rodload P_s, kgf	Friction coefficient μ (wire-polyamide)	Design springtension	Calculatedspringreductionfactor	Actual spring tension P_w, kgf	Reduction ratio *k* devices actual	Safety factor of nichrome wire with a diameter of 0.1 mm
5	0,05	0,089	56	0,09	55	13
10		0,179		0,12	83	10
15		0,269		0,14	107	8,5
20		0,359		0,15	132	8
30		0,538		0,19	151	6,3

Table 5: Layout response time

Diameter of nichrome wire, mm	Response time, s		
	Electric currentstrength 2 A	Electric current 3 A	Electric current 4 A
0,1	0,2	0,1	0,05

The maximum load created on the experimental sample was 300 N, and there was no destruction of nichrome wire with a diameter of 0.1 mm (safety factor of the wire 6.3). The maximum response/activation time of the initiating devices is 0.2 s at a current of 2 A.

Figure 10 shows a graph of the increase in load on the wire. The design case is presented with a friction coefficient μ =0 between the wedges, rod, and body element (only the friction coefficient between the spring and the body element was considered). Actual data revealed a rise in the reduction coefficient, which is due to an increase in friction forces with increasing load; at low loads, the reduction coefficient corresponds to the design case.

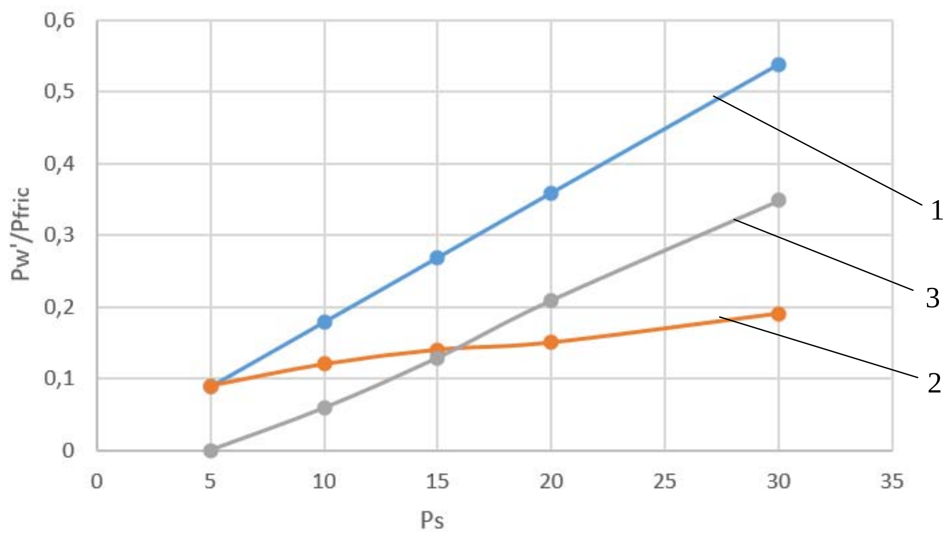


Figure 10: Dependence of the Load on the Wire on the Load on the Device Rod

A comparison of the characteristics of devices of different types is presented in Table 6.

Table 6: Comparison of Initiating Devices based on Heating of Structural Elements

Parameter	Meaning				
	Expandingelement (EN-3525 drive)	Heatingknives	Fusibleelement (RESHETNEV)	Fusibleelement (COOPER)	Promisingsample
Weight, g	35	80	50	35	35
Response time, s	200	5	1	0.2 orless	0.2 orless
Nominaloutputforce, N	156	150-200	60	560	morethan 560

Based on the test results, it was revealed that the mechanisms of this design have a high reduction coefficient. The kinematic scheme makes it possible to obtain a high nominal output force equal to or greater than initiating elements based on the principle of expanding structural elements and heating knives while maintaining the electrical characteristics inherent in initiating devices based on fusible elements. As the force on the retaining part of the structure increases, the reduction coefficient of the mechanism also increases due to an increase in the resistance forces between the rod and the wedges and between the wedges and the body elements.

The characteristics obtained from testing prototypes show that structures of this type are promising for use in devices for holding spacecraft mechanical systems.

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Entropic and Statistical Analysis of Refrigeration Plants for Retail Application at Design Stage

By M S Talyzin & A S Kochetov

Abstract- The entropic and statistical method of the analysis (ESMA) allows calculating losses in different refrigeration plant components and comparing them to define the elements that need measures to increase their operation efficiency.This paper presents results of analysis of refrigeration plants at design stage to choose appropriated system at energy efficiency and ecological influence point of view. Refrigeration plant with three working temperature levels (evaporation temperature -10℃, -18℃, -32℃) for retail application was analyzed.The degree of thermodynamic efficiency was calculated. This result could not be achieved using a traditional method of comparison refrigeration systems by using a coefficient of performance (COP) or seasonal efficiency as an only efficiency criterion.

GJRE-A Classification: *LCC: TP492*

ENTROPICANDSTATISTICALANALYSISOFREFRIGERATIONPLANTSFORRETAILAPPLICATIONATDESIGNSTAGE

Entropic and Statistical Analysis of Refrigeration Plants for Retail Application at Design Stage

M S Talyzin [α] & A S Kochetov [σ]

Abstract- The entropic and statistical method of the analysis (ESMA) allows calculating losses in different refrigeration plant components and comparing them to define the elements that need measures to increase their operation efficiency.This paper presents results of analysis of refrigeration plants at design stage to choose appropriated system at energy efficiency and ecological influence point of view. Refrigeration plant with three working temperature levels (evaporation temperature -10℃, -18℃, -32℃) for retail application was analyzed.The degree of thermodynamic efficiency was calculated. This result could not be achieved using a traditional method of comparison refrigeration systems by using a coefficient of performance (COP) or seasonal efficiency as an only efficiency criterion.

I. Introduction

The entropic and statistical method of analysis allows determining work which is necessary to compensate entropy production because of irreversibility and disequilibrium of real processes in the system components in order to identify an element or process requiring an increase in efficiency.

The aim of the modernization of the existed refrigeration plant was to increase energy efficiency and to use refrigerant with less global warming potential (GWP).

II. Entropic and Statistical Analysis

Refrigeration plant is located in Moscow and has three working temperature levels:

1. Evaporation temperature -10 ℃, cooling capacity 264.4 kW;
2. Evaporation temperature -18 ℃, cooling capacity 15.7 kW;
3. Evaporation temperature -32 ℃, cooling capacity 55.9 kW;

As a duty cycle, a single-stage compression cycle with a single throttle was used. One refrigeration unit for evaporation temperature -10 ℃ and one refrigeration unit for evaporation temperatures -18 ℃ and -32 ℃. The refrigerant was R404A.

For analysis, the following main dependencies were used:

1. Specific mass cooling capacity at evaporation temperature

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$$q_o^{tot} = \sum_{i=1}^{3} \left(q_{o_i} \times g_i \right) \tag{1}$$

where q_{o_i} - is the specific mass cooling capacity at the i temperature level; g_i - is the specific mass flow rate at the i temperature level.

The specific mass flow rate at the i temperature level is defined as the ratio of the mass flow rate G_i [kg/s] at the i temperature level to the mass flow rate G_l [kg/s] of the lowest temperature level.

$$g_i = \frac{G_i}{G_l} \tag{2}$$

2. Minimum specific work which is necessary for cold generation (electric power)

$$l_{\min}^{tot} = \sum_{i=1}^{3} \left(l_{\min_i} \times g_i \right) \tag{3}$$

where $l_{\min_i}$ - minimum specific work (electric power) for generating cold at i temperature level.

3. Adiabatic compression work

$$l_s^{tot} = \sum_{i=1}^{3} \left(l_{s_i} \times g_i \right) \tag{4}$$

where l_{s_i} - adiabatic compression work at i temperature level.

4. Actual specific compression work

$$l_{comp}^{tot} = \sum_{i=1}^{3} \left(l_{comp_i} \times g_i \right) \tag{5}$$

where l_{comp_i} - actual specific compression work at i temperature level.

5. Degree of thermodynamic efficiency

$$\eta_{therm}^{tot} = \frac{l_{\min}^{tot}}{l_{comp}^{tot}} \tag{6}$$

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6. COP at adiabatic compression

$$\varepsilon_s{}^{tot} = \frac{q_o{}^{tot}}{l_s{}^{tot}} \tag{7}$$

7. Actual value of COP

$$\varepsilon_{act}{}^{tot} = \frac{q_o{}^{tot}}{l_{comp}{}^{tot}} \tag{8}$$

Part of compression work, which is necessary for entropy production compensation in condenser (if there are separate condenser for each operating temperature level):

$$\Delta l_{cond}{}^{tot} = \sum_{i=1}^{3}\left(\Delta l_{cond\,i} \times g_i\right) \tag{9}$$

where $\Delta l_{cond\,i}$ - actual specific work, which is necessary for entropy production compensation in condenser at i temperature level.

Part of compression work, which is necessary for entropy production compensation in throttling process.

$$\Delta l_{thr}{}^{tot} = \sum_{i=1}^{3}\left(\Delta l_{thr\,i} \times g_i\right) \tag{10}$$

where $\Delta l_{thr\,i}$ - actual specific work, which is necessary for entropy production compensation in throttling process at i temperature level.

Part of compression work, which is necessary for compensation of entropy production in heat transferring processes from cooling object to refrigerant (evaporation):

$$\Delta l_{evap}{}^{tot} = \sum_{i=1}^{3}\left(\Delta l_{evap\,i} \times g_i\right) \tag{11}$$

where $\Delta l_{evap\,i}$ - actual specific work, which is necessary for entropy production compensation in heat transferring processes at i temperature level.

Estimated value of the adiabatic compression work is calculated:

$$l_{s.calc}{}^{tot} = l_{\min}{}^{tot} + \Delta l_{cond}{}^{tot} + \Delta l_{thr}{}^{tot} + \Delta l_{evap}{}^{tot} + \Delta l_{other}{}^{tot} \tag{12}$$

where $\Delta l_{other}{}^{tot}$ - losses in other processes (overheating, mixing, etc.)

Energetic Losses in Compressor:

$$\Delta l_{comp}{}^{tot} = l_{comp}{}^{tot} - l_{s.calc}{}^{tot} \tag{13}$$

Rated Compression Work:

$$l_{comp.calc}{}^{tot} = l_{s.calc}{}^{tot} + \Delta l_{comp}{}^{tot} \tag{14}$$

The results of the calculation of losses by system elements in percent (%) of the actual compression work expended are shown in Table 1. The results of the calculation of performance indicators are shown in Table 2.

Table 1: The Results of the Calculation of Losses by Elements of the System in % of the Actual Compression Work

	System 1	System 2	System a 3	System 4
Compressor	31,45	30,04	36,41	35,89
Condenser	13,98	14,01	13,05	15,29
Throttling	24,82	18,77	15,17	14,62
Economizer	-	1,98	-	-
Evaporator-condenser	-	-	1,15	1,38
Evaporator	6,86	7,97	9,63	8,52

Table 2: Performance Indicators of the Systems under Consideration

	System 1	System 2	System 3	System 4
η_{therm}^{tot}	0,23	0,27	0,25	0,25
ε_{act}^{tot}	1,74	1,97	2,49	2,50

During the analysis of the existing refrigeration system (System 1), it turned out that part of compression work, which is necessary for entropy production compensation in throttling process is on second place after energetic losses in compressor.

To reduce losses during throttling, it was decided to use separate refrigeration units for evaporation temperatures -18 ℃ and -32 ℃. The operating cycle was also changed to a cycle with an economizer with R404A as refrigerant (System 2). These measures allowed to reduce losses during throttling by 24.38%, increase the COP by 11.68% and the degree of thermodynamic efficiency by 14.81% (Table 2).

To reduce the negative impact to the environment, it was decided to use R744 as refrigerant. The most efficient cycle with R744 for similar applications is a cascade refrigeration system with R134a in the high cascade stage and R744 in the low cascade stage [2]. To increase efficiency, this unit was designed with separate compressors for evaporation temperatures of -18 ℃ and -32 ℃ (System 3). The schematic diagram of System 3 is shown on Fig. 1. The results of the analysis showed a decrease compression work which is necessary for entropy production compensation during throttling processes by 38.88% and an increase the COP by 30.12% compared to System 2.

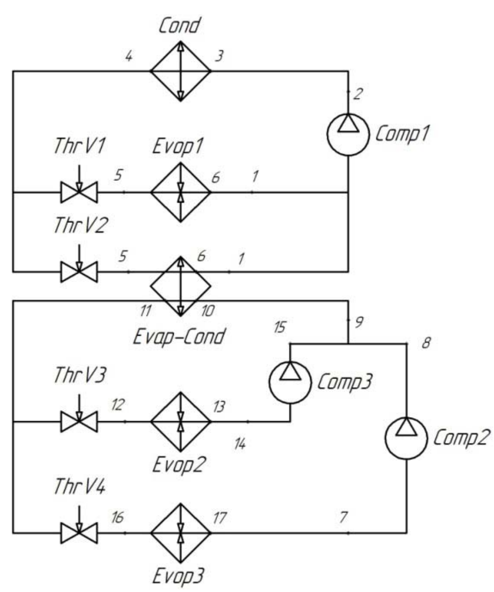

Figure 1: Schematic diagram of the System 3. Comp1, Comp2, Comp3 - compressors; Evap1, Evap2, Evap3 - evaporators; Cond - condenser, ThrV1, ThrV2, ThrV3, ThrV4 - throttle valves; Evap-Cond - evaporator-condenser.

To reduce negative impact to the environment it was offered to use R450A instead of the R134a (System 4). According to the results of the analysis, the performance indicators of a cascade refrigeration plant with R450A in the high cascade stage are at a comparable level with System 3, and the GWP of the refrigerant R450A is lower than R134a (601 versus 1300).

The results of the analysis are shown on Fig. 2.

III. Conclusions

1. The application of the entropic and statistical method of the analysis allows to determine energylosses in the elements of the system and take measures to increase the efficiency at the design stage;
2. To increase the efficiency of refrigeration systems, it is necessary to reduce part of compression work, which is necessary for entropy production compensation in throttling process, for example, by using cycle with an economizer, cascade cycle or alternative refrigerant;
3. The COP of a cascade refrigeration system with separate compressors for evaporation temperatures -18 ℃ and -32 ℃ exceeds COP of refrigeration system operating on a single-stage compression

cycle with a single throttling and two refrigeration units (one unit for evaporation temperature -18 ℃, one for evaporation temperature -18 ℃ and -32 ℃) by 30.4%, the degree of thermodynamic efficiency exceeds the same value by 8%;

4. R450A refrigerant is a full replacement of R134a refrigerant in the high cascade stage due to comparable performance indicators (COP and degree of thermodynamic efficiency) and a lower value of the GWP.

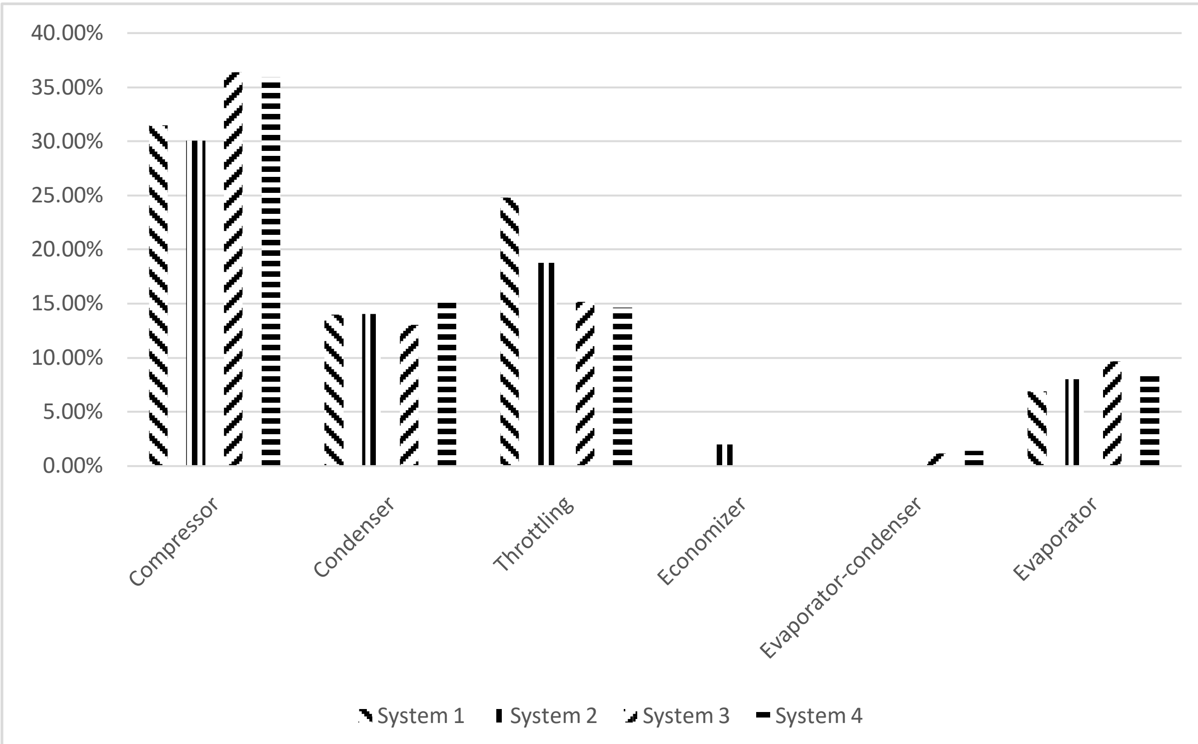

Figure 2: Comparison of Energy Losses by System Components in % of the Actual Compression Work Expended.

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The use of Universal (Control) Characteristics of Vane Blowers for an Objective Assessment of the Energy Efficiency of Pumping Systems

By Valentin G. Nikolaev & Georgy V. Nikolaev

Abstract- This article describes the methodology for assessing the energy efficiency of pumping systems equipped with a variable electric drive and operating with variable load, which was elaborated by the authors of the article. The analysis of traditional methods of recalculating the characteristics of the efficiency of pumping units when changing the frequency of rotation of the impeller using the formulas of theories of hydrodynamic similarity of blade blowers is presented. It is shown that in the derivation of the similarity formulas assumptions were made, which were not fully confirmed later when testing operating pumps. An algorithm and a technique developed by the authors are presented that allow one to read the actual values of the efficiency from the pre-digitized universal characteristics of vane blowers The results of comparison of efficiency values calculated by different methods are presented.

Keywords: mathematical model, pumping system, pipeline network, universal (control) characteristics of blade blowers, energy efficiency.

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Keywords: mathematical model, pumping system, pipeline network, universal (control) characteristics of blade blowers, energy efficiency.

I. Introduction

A considerable part of the energy-efficient technological processes used in the national economy are quasi-stationary. They include, for instance: cold and hot water supply, water disposal, heat supply, etc. Since these processes are, as a rule, of accidental and random nature, the governing parameters (flow, head, and pressure) are subject to significant changes in the course of time. This presents great challenges for the proper selection of vane blowers (such as rotodynamic vane pumps, fans, smoke exhausters, and air blowing machinery) as well as complicates their control noticeably.

The existing regulatory and technical literature [1-7] recommends that the selection of parameters for rotodynamic vane blowers operating with variable load should be done using extreme parameters arising at peak loads. This approach is based on the reliability principle as the characteristics of the installed equipment covering the peak load can ensure coverage of any other current load with a considerable reserve. Processing of the statistical data which characterize the operation of equipment with reliable load in engineering systems shows that the probability of arising of a peak load within a year is insignificant while its time duration amounts to approximately 2–3%. That is why the equipment operates most (major part) of the time under the modes which differ substantially from the extreme ones.

The processing of the statistical data, which characterize the operation of equipment with reliable load in engineering systems, shows that the probability of arising of a peak load within a year is rather insignificant while its time duration amounts to approximately 2–3%. That is why, the equipment operates most (major part) of the time under the modes which differ substantially from the extreme ones.

In compliance with the traditional methodology for selecting parameters of rotodynamic vane blowers, their flow under the optimal mode of operation Q_{opt} is assumed to be equal to the maximal load value Q_{max}; that is the following condition shall be fulfilled: $Q_{max}=Q_{opt}$. The maximal value of efficiency corresponds to the optimal flow Q_{opt}, so with the traditional technique for selecting such rotodynamic vane blowers, the following condition shall be satisfied: the maximal value of efficiency shall correspond to the optimal flow. Since all the current flow values in this case are found to be lower than the maximal (peak) load, their coverage corresponds to the rotodynamic vane blowers' operation in the underload modes. Herewith, there is a displacement of the vane blower's operation modes into the region of low values of efficiency, which leads to an excessive energy consumption and, consequently, to a low energy efficiency of operation of the blowing system on the whole.

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Testing and mathematical simulation for the existing pumping systems indicate that efficient operation of the rotodynamic vane blowers under variable load can be achieved if two conditions are met simultaneously: firstly, high efficiency values of the pump itself and, secondly, maximal binding of its characteristics to the parameters of the system where it will be installed. Researches show that the efficient operation of the blowing system under variable load will be composed from a set of efficiencies of its operation in separate modes that compound the process. The basic parameters that form the actual operation modes are as follows: the flow rate is Q, head — H, and efficiency — η. If the water consumption value is set by the system, then the head H_j and efficiency η_j (at the flow Q_j) depend in great measure on the validated method of control of the pumping unit, as well as on the parameters of the technology process supported by the rotodynamic vane blower.

Analysis of the energy efficiency of the blower operation under the current operation modes requires determination of the head (pressure), efficiency, useful and consumed power in each of the statistical intervals described by the process histogram. To obtain the needed accuracy of estimating the head and efficiency values that form the values of the consumed power in any considered statistical interval, it is necessary to determine their number correctly to ensure adequacy of describing the real physics processes by a mathematical model that replaces them. In order to achieve the accuracy of estimating the head and efficiency demanded for practical calculations, it is essential, as the studies have shown, that the gradients of these parameters ($\Delta H/\Delta Q$ and $\Delta\eta/\Delta Q$) in the limits of each considered statistical interval be not greater than 1.5–2%.

The on-site investigations of the water supply and heat supply systems carried out by the SPL “Energosit” show that the efficiency of the rotodynamic vane blowers operating with variable load is affected greatly by a wide range of factors, namely: the range of consumptions used by the system and distribution of their probabilities within this range, the position of the optimal flow Q_{opt} and efficiency of the rotodynamic vane supercharge towards the placement of the load (flow) range, as well as the Q_{opt} position relative to the flow at the blower’s operating point. Besides, when the proportional method of control is used and at minimizing the exceeded heads, the energy efficiency of the rotodynamic vane pump operation is affected significantly by the hydraulic exponent of the system (i.e. the relation $a = H_{st}/H_{req}$, where H_{st} *and* H_{req} are the static and required heads of the system).

Do the accepted traditional methods of selecting rotodynamic vane blowers take into account the impact made by the above factors on the selection? For sure, they do not, as binding of the pump characteristics to the system parameters is done using one single point with the coordinates Q_{opt} and H_{opt}, which is true only when a rotodynamic vane blower with a steady load or close to this is operating. Extrapolation of the traditional technique for selection of rotodynamic vane superchargers equipped with VFD and operating with variable load does not permit us to make a maximal binding of the rotodynamic vane blower characteristics to the system parameters, without sufficient substantiation and correction (or without replacement of this technique to a new one). In the absence of such binding, one of the two conditions described above and being required will ensure the best efficiency of pumping systems.

The Scientific and Practical Laboratory “Energosit” has been performing the researches on the enhancement of energy efficiency of rotodynamic vane blowers operating with variable load in water supply, water disposal and heat supply systems for more than ten years. The research is being done in frames of the energy service contracts when payment for the fulfilled work is made from the resources obtained through power saving that is achieved as a result of modernization of the operating pumping systems by way of selecting the optimal characteristics of the installed rotodynamic vane blowers relative to the parameters of the operating systems as well as by way of selecting the most efficient techniques of their control. The work under the energy service contracts requires a thorough instrumental examination of the validating operating systems having the purpose of obtaining objective and reliable data on the evaluation of the existing power saving potential.

Since mistakes in defining the power saving potential may lead to economically unsubstantiated decisions and unjustified investment, the authors of this study have worked out and tested in practice the technique for evaluation of the energy efficiency of operation of pumping systems. As for most of the technology processes supported by rotodynamic vane blowers the change in the load (flow) takes place rather slowly, the duration of the process can be split into separate static time intervals, in the limits within which it is allowed to consider, with a reasonable degree of accuracy, the flow, head and efficiency as being constant. This permits one to determine the energy consumption of the pumping unit through numerical integration for any considered time period, for instance, one year:

$$S_w = \sum_{j=1}^{m} \frac{\rho g Q_j H_j}{1000\,\eta_{jpum}\,\eta_{j\,mot}\,\eta_{j\,dr}} p_j T \qquad (1)$$

where

ρ is the density of the liquid, kg/m³; g is the free-fall acceleration, m/s²;

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Q_j, H_j are the flow and head of the rotodynamic vaneblower in the j-th interval of flows, m^3/s, m.w.c;

η_{jpum}, $\eta_{j\,mot}$, $\eta_{j\,dr}$ are the efficiencies of the rotodynamic vane blower, electric motor and electric drive in the j-th statistical interval;

p_j is the probability of the flow Q_j in one year;

T is the time of the pump operation during one year (hour);
$m$is the number of members in a statistical series.

For the pumps equipped with a variable-frequency drive (VFD), the head characteristic and efficiency characteristic may be represented in the form:

$$H_j = A_n Q_j^2 + K_T B_n Q_j + K_T^2 C_n \qquad (2),$$

$$\eta = D_n Q_j^2 + E_n Q_j + F_n \qquad (3),$$

where

A_n, B_n, C_n, D_n, $E_n$and $F_n$are the approximation coefficients of the pump characteristics;

$K_T = \frac{n_C}{n_N}$ is the coefficient of the rotation frequency change;

n_C and n_N are the current and nominal rotation frequencies of the impeller of the rotodynamic vane pump.

In the case when a VFD is used, recalculation of the characteristics of rotodynamic vane blowers in terms of the current frequency of the impeller rotation, which is different from the nominal one, is done using the formulas received from the theory of hydrodynamic similarity of rotodynamic vane blowers. The formulas for the recalculation of the flow, head and power have the following form in such a case:

$$\frac{Q_C}{Q_N} = K_T \qquad (4),$$

$$\frac{H_C}{H_N} = K_T^2 \qquad (5),$$

$$\frac{N_C}{N_N} = K_T^3 \qquad (6),$$

where

Q_C, H_C, N_C are the current values of the flow, head and supplied power for the current frequency of the impeller rotationn_c;

Q_N, H_N, N_N are the same for the nominal frequency of the impeller rotation n_C.

The use of formulas (4) and (5) permits one to determine with a high degree of accuracy the values of flow and head when the frequency of the impeller rotation is changed, which allows one to obtain a new position for the head characteristic of the pump in the H-Q coordinates (Fig.1).

Figures for this Study (including captions)

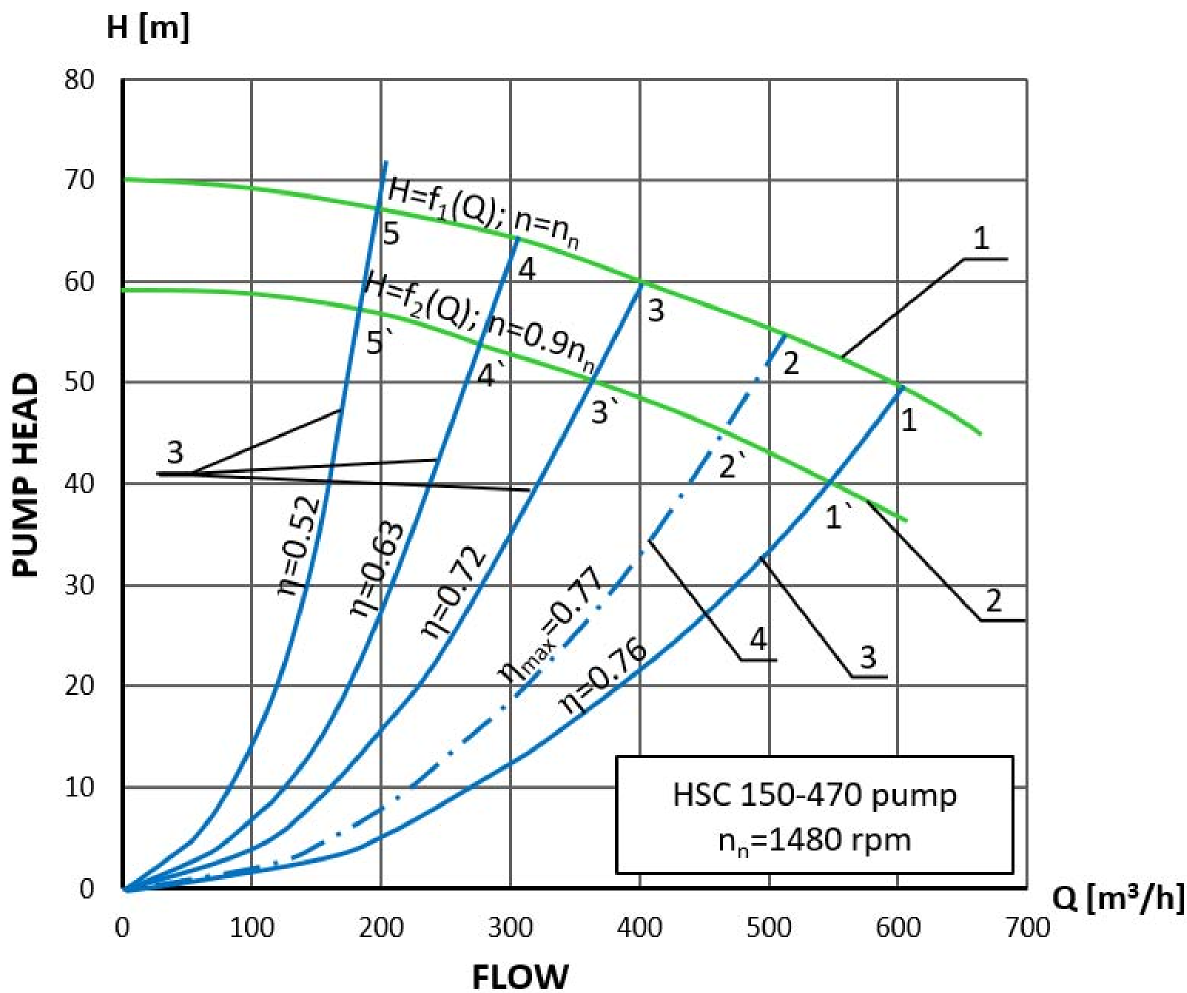

Fig. 1: For the Recalculation of the Characteristics of a Rotodynamic Pump using the Formulas of Hydrodynamical Similarity

1. The head characteristics of the pump for the nominal rotation frequency of the impeller (n_n=1480 rpm);
2. The head characteristics of the pump at the reduction of the rotation frequency ($n=0{,}9n_n$=1330 rpm);
3. The curves of similar modes (CSM);
4. A curve of similar modes of the maximal value of efficiency.

Fig. 1 displays the head characteristic $H_1=f_1(Q)$ for the nominal (n_N=1480 rpm) and decreased (current) frequency of rotation (n_C=0.9, n_N=1330 rpm) of the impeller $H_2=f_2(Q)$. The initial position of the head characteristic for the nominal frequency of the impeller rotation H-Q (see Fig. 1, pos. 1) and its position at decreased rotation frequency are given in the figure (see Fig. 1, pos. 2). Fig. 1 also presents the curves of similar modes (CSM), i.e., the equal level curves of efficiency (see pos.3). In compliance with the theory of hydrodynamic similarity, as the rotation frequency reduces, point 5 (see Fig. 1) shifts to position 5' (4 to 4'), etc. Displacement of each point of head characteristics 1, 2, 3, 4, 5 to positions 1', 2', 3', 4' and 5' (see Fig.1) occurs by the parabola $H_j=K_jQ^2$, that is by the curve of similar modes (CSM). When formula (6) is used for recalculation of the consumed power, it is supposed that the efficiency values along the parabola of similar modes do not change, i.e. $\eta_5/\eta_{5'}=1$, $\eta_4/\eta_{4'}=1$, etc.

It is known that the actual efficiency values at the change of the impeller rotation frequency in the rotodynamic vane blower can be received only by way of performing bench tests at different frequencies of its rotation.

This allows to obtain the so-called universal (control) characteristics of the rotodynamic vane blower. As an example, the control characteristic of the WiloCronoNormNL-125/200 (n_N=2900 rpm) vane blower is given in Fig. 2.

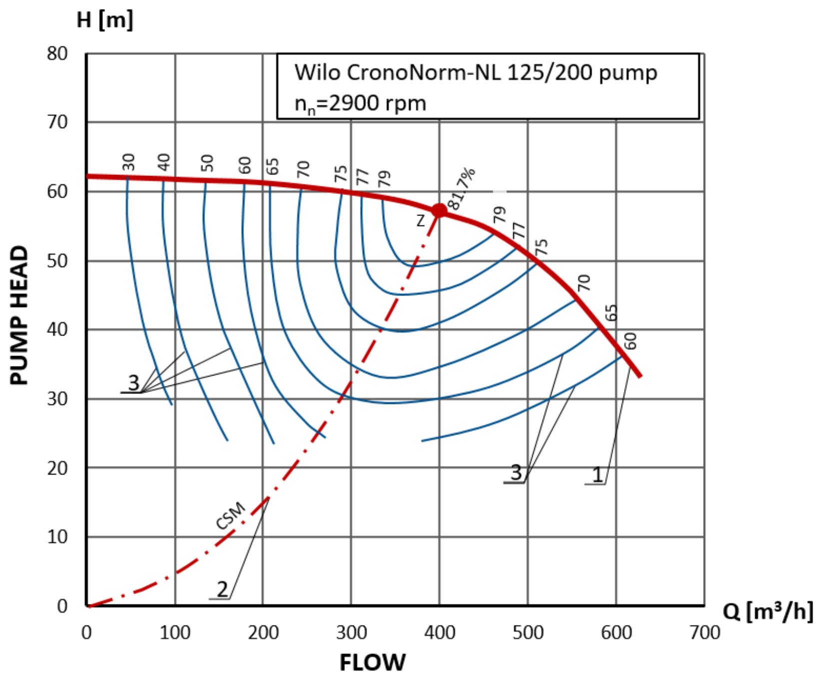

Fig. 2: The Universal (Control) Characteristics of the Pump

1. The head characteristics of the pump ($n_c=n_n$);
2. A curve of similar modes of the maximal value of efficiency;
3. The efficiency equal level curves.

Fig. 2 displays the head characteristic of the pump (see pos.1) for the nominal rotation frequency of the impeller, the curve of similar modes (CSM) of the maximal value of efficiency (see pos.2) as well as other curves of similar modes (see pos.3), each of which corresponds to a certain level of efficiency.

A visual comparison of the geometrical shapes of the equal level curves of efficiency, which are presented in Fig. 1 and 2, indicates their significant difference. Whereas, the CSM curves received basing on the theory of hydrodynamic similarity (see Fig. 1) present a family of curves (parabolas) which start at the origin of the coordinates; so the equal level curves of efficiency at the universal characteristic are concentric elliptic-and-oval curves with a maximal value (vertex) of efficiency at the Z point. During the shift along the curve of similar modes ($\eta=\eta_{max}$) from the Z point to the O point (the origin of the coordinates), the actual values of efficiency descend continually (see Fig. 2), while the efficiency along each CSM line remains constant, according to the theory of hydrodynamic similarity (see Fig. 1). An analogous comparison of the curve shapes for pumping units manufactured by various companies reveals a substantial divergence both in the shape and degree of decrease in the maximal values of efficiency along them. The main distinction between the compared characteristics for different pumping units consists only in the degree of intensity of the efficiency reduction along the curves of similar modes for a common value of deviation of the current values of rotation frequency from the nominal one, i.e. the rotation frequency coefficient κ_T.

A steady exceedence of the efficiency values calculated using the formulas of hydrodynamic similarity relative to the practical values obtained with the help of the universal characteristics is explained by the violation of the conditions of similarity arising at the deviation of the current rotation frequency of the impeller from the nominal one. This circumstance was pointed to by prof. K. Pfleiderer, one of the founders of the theory of vane blowers, in his monography back in the 30-s of the previous century [8]. He stressed out that "testing of the real pumps does not confirm fully the derived law of efficiency constancy along the parabolas of the constant flow coefficient. The curves of the constant efficiency value usualy have an ellipsoidal shape". K. Pfleiderer proposed an empirical formula for taking into account the efficiency reduction along the curves of similar modes:

$$\eta_{ch} = 1-(1-\eta_n)\mathrm{x}\left(\frac{n_n}{n_{ch}}\right)^{0,2}, \qquad (7)$$

where

η_N and η_{ch} are the efficiencies for the nominal and changed frequencies of the impeller rotation; n_H and n_{ch} are the values of the nominal and changed rotation frequency.

The calculations made in compliance with formula (7) for the efficiency change at deviation of the current values of rotation frequency of the impeller from the nominal and a comparison between them and the actual ones received using the universal characteristics for the curves of the maximal efficiency values have shown their substantial difference. At the same time, the decrease in the efficiency values along the curve of similar modes calculated according to formula (7) is considerably lower than the values obtained using the universal characteristics of the pumping unit. Because of the divergence of the calculated efficiency values calculated along the curves of similar modes, as compared to the efficiency values determined with the help of the universal characteristics, some of the authors have attempted to do extrapolation of the exponent in formula (7) by way of extending the range of its values from 0.1 to 0.25. However, such correction has not permitted us to obtain a generalized formula that could describe fairly adequately the decrease of efficiency relative to the universal characteristics for pumping units of various types from different manufacturers.

Due to the accepted proposal concerning a slight reduction of the efficiency values along the CSM, calculated in compliance with the formulas of the similarity theory, the Euro-zone standard [2], as well as a number of authors [9, 10], have admitted the possibility of substantial broadening of the region of possible operating modes of rotodynamic vane blowers where the reduction of efficiency can be disregarded. Thus, Euro-zone standard [2], for instance, permits one to disregard the efficiency reduction with the decrease of rotation frequency by 20% relative to the nominal one; and the authors of a series of studies [9,10] recommend that the reduction of efficiency at the decrease of rotation frequency up to 33% should not be taken into account.

The significant dispersed opinions and quite contradictory recommendations made by different authors about the technique of determination of the efficiency of rotodynamic vane blowers at variable rotation frequency, as well as the influence on its values exerted by the degree of deviation of the current rotation frequency from the nominal one, are explained by the insufficient attention of the scientific community to this problem. During a fairly long period of time, a rotodynamic vane blower drive has been triggered using asynchronous electric motors operating with a constant frequency of rotation of the rotor shaft. That is why the problem of the impact of the impeller rotation frequency on the efficiency of the rotodynamic vane blower presented a purely theoretical interest without being popular at the market of technologies forrotodynamic vane blowers. The appearance of the variable-frequency drive at the market at the end of the previous century has led to its wide application and transfer of the studied problem from the theoretical to the practical plane.

To study the above problem, the following tasks have been set in the present research that can be relatively divided into two sets:

1. Investigation of the degree of deviation of the current frequency of the impeller rotation from the nominal and its influence on the pump efficiency in a wide range of load variation, characteristics of pipeline systems, and control methods, in particular, for:

- The range of changes in the pump flow from 0.25 to $1.3Q_{opt}$;
- Correlation of the statistical component of the required head H_{st} and total head H_n (of the hydraulic factor of pipeline systems) in the limits: $0 \leq a \leq 1$;
- The method of selecting the parameters of the pumping equipment and control method;

2. The study of the possibility of the use of universal (control) characteristics for calculation of the efficiency values of pumps operating with a variable drive, as well as investigation of an algorithm for their determination, which requires the following:

- To apply software for automated digitization of complex curves to a family of isolines of constant values of efficiency which are obtained in Q-H coordinates at the variable rotation frequency of the impeller (the so-called universal characteristics of the pump);
- To work out the algorithms of approximate calculation of the pump efficiency through pre-digitizing of its universal characteristics in the region of supposed modes of the pump operation and specified head characteristics for the nominal rotation frequency of the impeller ($n_C=n_N$), as well as for the rotation frequency that exceeds the nominal by 20% ($n_C=1{,}2\ n_N$), and at its reduction up to 50-55% from the nominal;
- To test the elaborated algorithm by comparing the pump efficiency values obtained with its help, with the efficiency values calculated according to the formulas basing on the theory of hydrodynamic similarity of rotodynamic vane blowers.
- To show the impact of the pump efficiency calculation method on the quality of the estimation of pumping systems' operation.

To solve the above mentioned tasks, the methodology and LAB-MZ computer program for selection of the parameters of the pumping equipment and assessment of the energy efficiency of the pumping equipment were used, which had been elaborated earlier at the SPL "Energosit" [11-14]. This methodology was worked out basing on mathematical modelling of the "pumping unit – variable drive – pipeline network" system.

The carried out comparison and analysis of the methodology made by the authors as well as the method of estimation of the energy efficiency using the EEI index (elaborated in accordance with the concept of ENPA) have shown a variety of advantages of our mathematical model which has been developed by us [15–17].

The key feature of the used mathematical model is its sufficiently complete adequacy to the physical process of the water flow in the piping system that provides obtaining insignificant gradients of basic parameters which define the energy efficiency of the pumping work in the limits of every interval of statistical sampling. These include the useful and consumed power, efficiency, the frequency of rotation of the impeller etc. This circumstance permits one to obtain not only the integral estimation of the efficiency of the pumping work in the limits of the whole range of load change within the defined period of time but also to calculate, with the reasonable for the practical calculations accuracy (1–2%), such parameters as efficiency, the frequency of rotation of the impeller and others for each statistical interval.

The mentioned applicability of methodology of the estimation of pumping systems' energy efficiency worked out by the authors and application of the computer program permit us to use them for solving of the earlier formulated problem of applying the universal (control) characteristics as well, related with its solution task for determining the actual values of efficiency of rotodynamic vane blowers during their operation in the variable load.

When performing the study of the dependence of the deviation of the current frequency of rotation of the impeller from the nominal one for different values of the hydraulic factor, the mathematical model of the WILO-CronoNorm-NL-125/200 pump was used. The values of the statistical head were accepted as follows: 0, 5, 20, 26, 32, 36и 41m, which corresponded to the following values of the hydraulic factor: 0, 0.11, 0.43, 0.56, 0.69, 0.77, and 0.88.

The parameters of the pump under the optimal mode were: $Q_{opt}=442 m^3/h$; $H_{opt}=56{,}7$m; $\eta_{opt}=0{,}809$; and at the operating point: $Q_{op}=618 m^3/h$; $H_{op}=45{,}2 m^3/h$; $\eta_{op}=0{,}695$. The flow rate in the pipeline system changed within the range from $Q_{min}=140 m^3/h$to $Q_{max}=600 m^3/h$; the distribution of heads was taken in accordance with the actual data of inspections bythe VNS-2 (Balashikha city, Moscow region). The pressure value of the stabilization in the pressure header of the pump was accepted as $P_{staв}=4{,}6$t.a. The results of the calculation of the$\kappa_т$frequency rate of rotation for different values of the hydraulic factor and methods of control are presented in Table 1.

Table 1: A List of Tables for this Study: Table □ 1 Dependence of the Coefficient of Changes in the Rotation Frequency of the Impeller for Different Values of the Hydraulic Factor of a Pipeline System and Control Methods

№	The value of the coefficient of changes in the rotation frequency of the impeller $\kappa_т$ as a function of the adopted pump control and hydraulic factor of the system	Units	Correlation of the static component and total head of the pump at the operating point (hydraulic factor of the system)							
			0	0.11	0.43	0.56	0.69	0.77	0.88	
1	Throttling of a pipeline system: κ_{th}		1.0	1.0	1.0	1.0	1.0	1.0	1.0	
2	Pressure stabilization in the pressure header of a pumping station (Hstaв=46m): $\kappa_{staв}$	-//-	$0.856 \le \kappa_{staв} \le 0.996$							
3	Minimization of excessive heads in a pipeline system (proportional control) at selection of the parameters of pumps by the traditional technique:									
	- minimal K_{minh}^{min}	-//-	0.235	0.321	0.582	0.655	0.721	0.768	0.821	
	- maximalK_{minh}^{max}	-//-	0.951	0.952	0.976	0.986	0.991	0.996	0.998	
4	Minimization of exces-sive heads in a pipeline system at selection of the parameters of a pump by the method-logy of the SPL "Energosit" (using a virtual pump):									
	- minimalK_{opt}^{min}	-//-	0.392	0.445	0.732	0.818	0.854	0.879	0.878	
	- maximalK_{opt}^{max}	-//-	1.22	1.178	1.149	1.117	1.088	1.103	1.032	

It is seen from Table 1 that at the throttling of the pipeline system, the pump operates with a constant frequency of rotation of the impeller ($n_C=n_N$; $K_т=1$) which is independent from the hydraulic factor of the pipeline system (a). At the pressure stabilization in the discharge header of the pumping unit, the deviation of the current rotation frequency of the pump's impeller was also non-dependent on the characteristic of the pipeline system, and its maximal value amounted to no more than 15% from the nominal one.

Because the most significant deviations of the working frequency of rotation from the nominal one manifest themselves in the pipeline systems with an insignificant statistical component of the required head, it is of interest to consider such a control method as minimizing of the excessive heads, or its modification – proportional regulation. As our research has shown, the chosen method of selection of the optimal pump parameters produces a considerable influence on the value of the $K_т$ coefficient. Therefore, when making this research, two variants of selection of the equipment parameters were considered, namely: the traditional approach for the maximal (peak) load and the authors' method for the most probable load using the mathematical model of a virtual pump.

In a general case, the value of the coefficient $K_т$ of the change of the rotation frequency is a function of the change between two parameters: $K_т=f(a, Q_j)$, where: «a» is the hydraulic factor of the pipeline system; Q_j is the pump head. That is why the following algorithm for performing our research was accepted: for the fixed value of the hydraulic factor the value of the $K_т$ coefficient was defined on the whole possible range of the varying pump head from the maximal $Q_j=Q_{max}$ (the right boundary of the head range) to the minimal $Q_j=Q_{min}$ (left boundary), then other values of the hydraulic factor were accepted for making recalculations. The range of the factor (a) variation covered practically all variants taking place in practice: $0 \le a \le 0.88$. The summary of our research is given in Table 1. Of practical interest is to consider the results obtained for the boundary values of the hydraulic factor, i.e., for a=0 and a=0.88.

For the traditional selection of the optimal parameters of the pump, for the maximal (peak) load at the value of the hydraulic factor of the system (a=0), the values of the $K_т$ coefficient of the varying rotation frequence were: $K_{minh}^{max} = 0.951\,(Q_j = Q_{max})$ and $K_{minh}^{min} = 0.235\,(Q_j = Q_{min})$. The amplitude of variation of $K_т$ for the fixed value (a) was: $\Delta K_{minh} = 0.716$ (where: $\Delta K_{minh} = K_{minh}^{max} - K_{minh}^{min}$). For a=0.88, the values K_{minh}^{max} and K_{minh}^{min} are equal to 0.998 and 0.821, correspondingly; the amplitude of variation of the factor $K_т$ was: $\Delta K_{minh} = 0.117$.

For the case of selection of the optimal pump parameters for the most probable load, the following results of determining the coefficient of change of the current frequency of rotation $K_т$ were obtained for the hydraulic factor (a=0):

$$K_{opt}^{max} = 1.22\,(Q_j = Q_{max}) \text{ and } K_{opt}^{min} = 0.392\,(Q_j = Q_{min})$$

The amplitude of variation of $K_т$ was: $\Delta K_{opt} = 0.828$, where: $\Delta K_{opt} = K_{opt}^{max} - K_{opt}^{min}$. For a=0.88, the values K_{opt}^{max} and K_{opt}^{min} are equal to 1.032 and 0.878, correspondingly; and the amplitude of variation of the $K_т$ coefficient was $\Delta K_{opt} = 0.154$.

According to the results of our research (see Table 1), Fig 3. Shows diagrams with the dependencies of the current rotation frequency coefficients K_{minh} and K_{opt} on the hydraulic factor of the pipeline systems using different methods of selection of the pump equipment. A comparison was made for a wide spectrum of characteristics of the pipeline systems ($0 \le a \le 0.88$) and possible operation modes of the pumps (head: from 0.25 to 1.2 Q_{opt}; the impeller rotation frequency is from 0.235 to 1.22 $n_н$). This covers almost all characteristics of pipeline systems that are of practical interest and operation modes of the pump equipment with a variable-speed drive. It is seen from the given diagrams in the Fig that the area covered by the curves 1and 2, shifts equidistantly in parallel to the X axis (see the are a covered by curves 3 and 4). This circumstance leads to the fact that for one and the same value of the coefficient (a) of the pipeline system, the coefficient of deviation of the current frequency of rotation of the impeller from the nominal one is reduced significantly (by 15–20%). Mathematical simulation and on-site investigations of the pipeline systems have shown that the decrease of deviation of the current frequency of rotation of the impeller from the nominal one at other equal conditions leads to an increase of the pump efficiency, and consequently, to growth of the efficiency of its operation.

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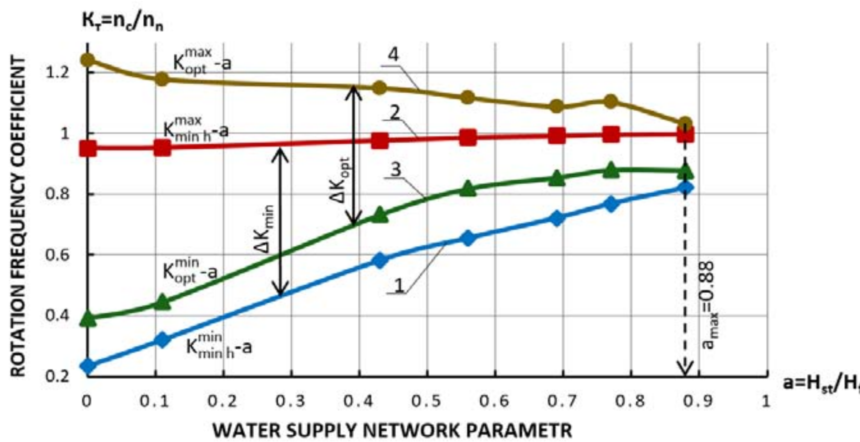

Fig. 3: The dependence of the coefficient of changing the frequent rotation of the impeller ($K_т$) on the hydraulic factor (a) at minimizing of the excessive heads for different methods of selection of the optimal parameters of a pumping unit

The method of the selection of the pump parameters: 1, 2 — for the traditional method of selection of the optimal parameters of the pumping unit; 3,4 — for method NPL "ENERGOSIT" at the most possible load

In such a case, curve 3 (see Fig. 4) shifts in respect to the X axis to position 4. The value of the Y axis of the curve shift ΔK is defined by the value of the hydraulic factor of the pipeline system (see Fig. 3). At shifting to position 4 the curve ABC crosses line 1 at the pointBwhere the frequency of rotation of the impeller of the virtual pump is equal to the nominal ($K_т$=1) and its head is equal to the optimal ($Q_j = Q_{opt}^{virt}$). That is why all the head modes of the pump when operating within the AB area ($Q_{min} \leq Q_j \leq Q_{opt}^{virt}$) will be underloading, and it is required to reduce the frequency of rotation of the impeller (n_C<n_N) for their support, although overloading under operation in the BC area ($Q_{opt}^{virt} \leq Q_j \leq Q_{max}$), for which the increase of the frequency is required (n_C>n_N). The study of geometric shapes of the universal characteristics of the rotodynamic vane blowers in a wide range of varying rotation of the impeller from 0.5 to 1.3 n_N and heads from 0.25 to 1.5 Q_{opt} has shown that the deviation of the current frequency of rotation from the nominal towards a greater or lesser value within the limits±15% (0.85≤ $K_т$≤ 1.15) leads to the decline of the maximal value of efficiency no more than 3–5%. At the traditional selection of the pump parameters for the maximal (peak) load ($Q_{opt} = Q_{max}$) all the operation modes of the pump relative to Q_{max} are underloading, and as the heads go down to Q_{min} a more significant decrease of the current frequency of rotation of the impeller is required. When selecting the pump parameters according to the most probable load while reducing costsof the virtual pump from Q_{max} to Q_{opt}^{virt}, there is an increase of the efficiency value from η_j to η_{max}^{virt}, and only when $Q_j < Q_{opt}^{virt}$, its decrease begins. The shift of the value of optimal efficiency, with a maximal head inside the range of loads, i.e. in the area of the most probable heads, allows one (see Fig. 4) to reduce the deviation of the current rotation frequency of the impeller from the nominal one significantly. In the example we are examining (see Table 2) the decrease of the deviation of the current frequency from the nominal coefficient across the entire range of the load variation reduced from 19.6 (at Q_{min}) up to 20.3 (at Q_{max}). This circumstance provides apossibilityfor increasing the energy efficiency of the pumping systems' operation.

The mathematical model of the pumping system with LOWARA NSC-150-470 unit was used for the investigation of the dependence of the frequency rotation coefficient $K_т$on the pump head for different variants of the pumping unit control. The range of variation in the flow rate of the pumping system was accepted as from$Q_{min} = 140\, m^3/h$to$Q_{max} = 600\, m^3/h$; and the probability distribution of the flow rate was accepted the same as in the example with theWILO-CronoNorm-NL-125/200 pump. The statistical head in the pipeline system was taken to be as$H_{st} = 20m$, the hydraulic factoras a=0,45, and the system hydrodynamical resistance as $\beta = 6.596 \cdot 10^{-5} (h^2/m^5)$. The pump parameters selected according to the traditional methodology for the characteristic modes were: at the operation point$Q_{op} = 645\, m^3/h$, $H_{op} = 47.5m$, $\eta_{op} = 0.754$; at the optimal head: $Q_{opt} = 540\, m^3/h$, $H_{opt} = 54m$, $\eta_{opt} = 0.781$. The parameters of a virtual pump when selecting it according to the most probable load are as follows:$Q_{opt}^{virt} = 464\, m^3/h$, $H_{opt}^{virt} = 34m$, $\eta_{opt}^{virt} = 0.781$. The results of the $K_т$coefficient calculation are shown in Table 2.

Table № 2: Dependence of the Coefficient of Changes in the Rotation Frequency of the Impeller (K_c) on the Flow of the Pump for Different Control Methods

№ of Parameter	1	2	3	4	5	6	7	8	9	10	11	12
Q, m^3/h	140	180	240	280	320	360	400	440	480	520	560	600
K_{other}	1	1	1	1	1	1	1	1	1	1	1	1
K_{stab}	0.827	0.835	0,849	0.860	0.870	0.883	0.897	0.911	0.927	0.944	0.961	0.980
K_{minh}	0.560	0.580	0.618	0.646	0.677	0.711	0.746	0.784	0.823	0.863	0.905	0.947
K_{opt}	0.756	0.775	0.812	0.840	0.871	0.904	0.941	0.979	1.019	1.040	1.105	1.150

In compliance with the data from Table 2, Fig. 4 shows diagrams with dependencies of $K_т = f(Q)$ for different ways of control. When throttling of the pipeline system takes place, the pump operates with the constant rotation frequency $K_т = 1$ (seecurve 1), and when stabilizing the pressure in the pressure header (see curve 2), the values of the $K_т$ coefficient were changing within the range from 0.827 to 0.98.

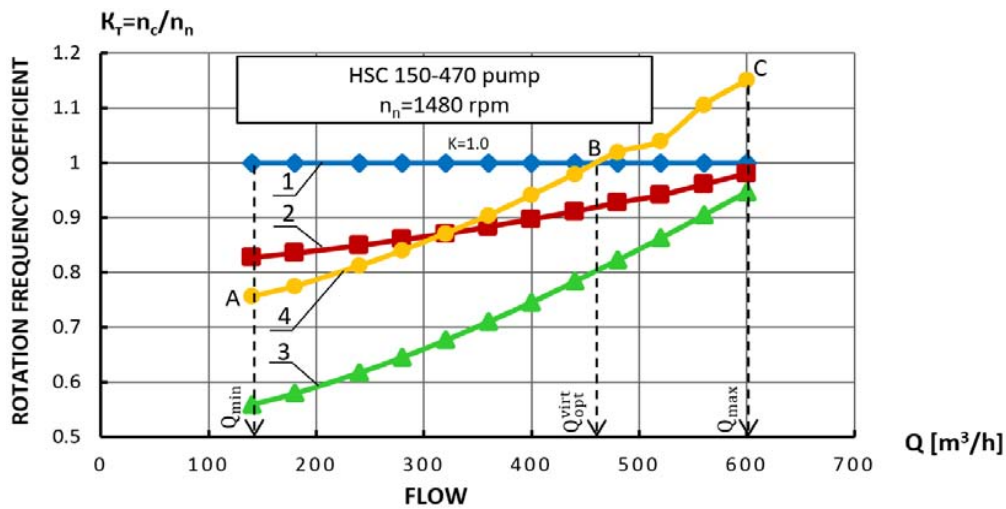

Fig. 4: Dependence of the Coefficient of Changes in the Current Rotation Frequency of the Impeller of the Pump on the Flow for Different Ways of Pump Control with Applying the Variable Frequency Drive

Control method:

1. Throttling of a pressure header;
2. Stabilization of the pressure in a pressure header;
3. Minimization of the excessive heads;
4. Optimization of the pump parameters using the model of the virtual pump.

The dependencies of the $K_т$ coefficient for two different cases of the selection of the pump equipment are presented in Fig. 4. In the case of selecting the pump parameters by a traditional way at minimization of excessive heads (see Fig. 4, curve 3), the value of the $K_т$ coefficient varied from 0.56 (at $Q_{min} = 140\, m^3/h$) to 0.947 (at $Q_{max} = 600\, m^3/h$). When selecting the parameters at the most probable load with using a virtual pump, the values of the $K_т$ coefficient changed from 0.756 (at Q_{min}) to 1.15 (at Q_{max}).

In regard to the above mentioned, of interest is the study of the pump efficiency dependence on the degree of deviation of the current rotation frequency of the impeller from the nominal one, i. e. on the K_c coefficient. The results of this research with the use of a mathematical model describing a pumping system with HSC-150-470 unit are given in Table 3.

Table № 3: Dependence of the Pump Efficiency on the Coefficient of Changes in the Rotation Frequency at Minimization of the Excessive Heads.

№ of Parameter	1	2	3	4	5	6	7	8	9	10	11	12
Q, $m^3/ч$	140	180	240	280	320	360	400	440	480	520	560	600
$K_т$	0.560	0.581	0.618	0.646	0.577	0.711	0.747	0.784	0.823	0.863	0.905	0.947
η	0.387	0.463	0.553	0.600	0.540	0.673	0.700	0.721	0.739	0.752	0.764	0,768

Fig. 5 displays a diagram of the pump efficiency dependence on the K_T coefficient, i. e. $\eta=f(K_T)$ with the use of data from Table 3.

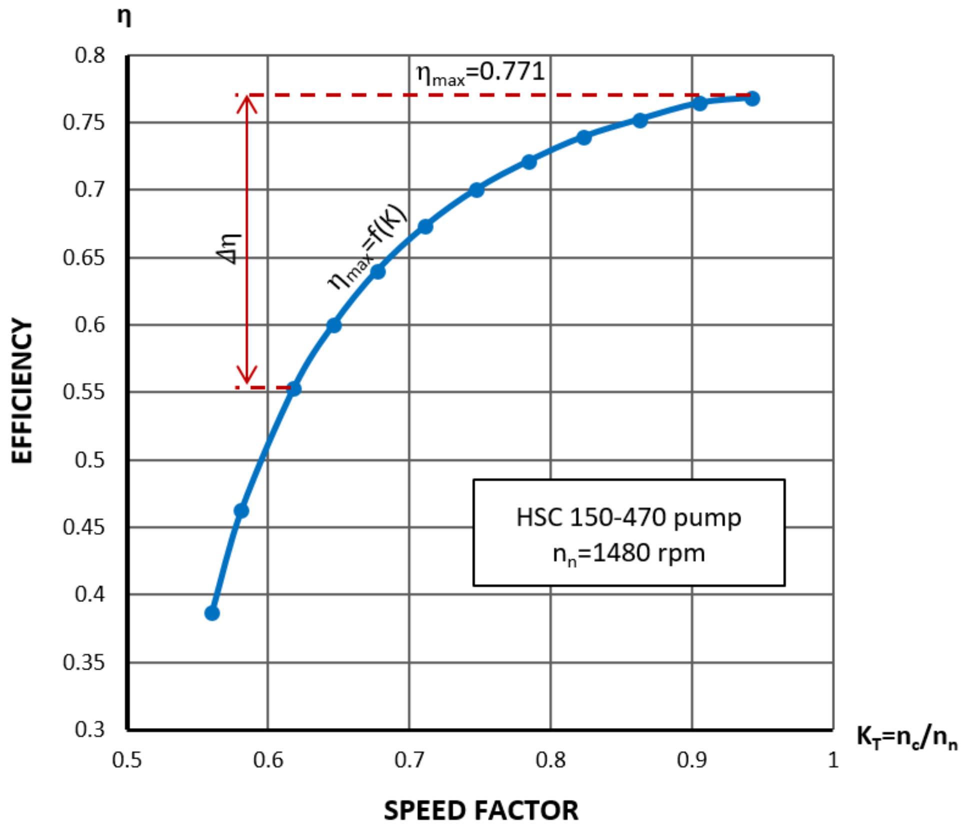

Fig. 5: Dependence of the Pump Efficiency from the Coefficient of the Rotation Frequency of the Impeller ($K_T=n_c/n_n$)

From the given Figure and Table 3 it can be seen that with a slight reduction of the impeller rotation frequency ($0.75 \leq K_T \leq 1$) the dependence $\eta = f(K_T)$ is close enough to the linear one, but with a subsequent decrease of the rotation frequency; the obtained dependence becomes exponentially vanishing. For a more detailed analysis of the efficiency characteristics dependent on the K_T coefficient, consideration of the data presented in Table 4 is of certain interest.

Table № 4: Dependence of Intensity of Decrease of the Pump Efficiency on the Range of Changes in the Coefficient of Rotation Frequency of the Impeller.

№ п/п	Range of changes in the coefficient of rotation frequency of the impeller $K_T=n_C/n_N$	Current value of the rotation frequency of the impeller, $n_C=K_T \cdot n_N$	Reduction of efficiency (%) in the considered range of the coefficient K_T	Summary reduction of efficiency (%) for the current frequency n_C
1	$K_T=1$	$n_C=n_N$	$\eta_T=\eta_{max}=0.771$	-
2	0.9 ÷ 0.95	(0.9 ÷ 0.95) n_N	1.0	1.0
3	0.85 ÷ 0.90	(0.85 ÷ 0.90) n_N	2.0	3.0
4	0.80÷ 0.85	(0.80÷ 0.85) n_N	2.0	5.0
5	0.75÷ 0.80	(0.75÷ 0.80) n_N	2.0	7.0
6	0.70÷ 0.75	(0.70÷ 0.75) n_N	4.0	11.0
7	0.65÷ 0.7	(0.65÷ 0.7) n_N	4.0	15.0
8	0.60÷ 0.65	(0.60÷ 0.65) n_N	10.0	25.0
9	0.55÷ 0.60	(0.55÷ 0.60) n_N	14.0	39.0

It is seen from Table 4 that the entire range of changes in the K_T coefficient was split into equal intervals with its variation in the limits of each interval equal to $0.05\% \cdot K_T$. The Table presents for each interval the values of the current rotation frequency, values of efficiency reduction in the specified intervals, as well as summary reduction of efficiency at deviation of the current value of rotation frequency from the nominal. As it has been pointed out above, at the linear segment of the curve (see Fig. 5) decrease of efficiency with variation of κ_T by 20% (in the limits from 0.95 to 0.75) amounted approximately to 7%, while its further reduction by the same value (from 0.75 до 0.55) led to a decrease in the efficiency value by 32%; and the summary reduction of efficiency over the whole range of changes in κ_T amounted to 39%.

Thus, the parameters of pumping systems changed within a wide range when performing an investigation of the possible deviations of the impeller rotation frequency from the nominal as well as when controlling the modes of the pumps' operation. So, the possible range of current values of the Q_T flow, for instance, was in the limits from 0.25 up to 1.4,Q_{opt}; the variation of the hydraulic factor of the pipeline systems ($a=H_{st}/H_п$) covered practically the whole range of its possible values, that is from 0 to 0.88. At the same time, research was conducted into a broad range of variation of the impeller rotation frequency, which changed in a wide interval of possible operating modes of the pump from 0.55 up to 1.25 n_N. The κ_T coefficient of the rotation frequency variation is an auxiliary factor which allows one to judge indirectly about the efficiency of the pump operation, while the key factor that determines the energy efficiency of the pump operation is its efficiency coefficient. That is why the study of the certainty of determination of the efficiency values for the studied range of variation of the κ_T rotation frequency coefficient is of considerable interest. The unit for determination of the pump efficiency that is a part of a computer program for estimation of the energy efficiency of the LAB-MZ pumping systems allowed us to calculate the efficiency values for different operating modes using the formulas of the hydrodynamic similarity theory and curves of similar modes (CSM) which are efficiency's equal value lines. However, in practice, as it was pointed out earlier, the efficiency constancy condition (η=const) along the CSM is fulfilled not in full, which leads to deviation of the real efficiency values from the calculated ones. Here, the values of the indicated divergence increase with the growth of the deviation of the current rotation frequency from the nominal, i.e. from the magnitude of the κ_T coefficient value.

The only source of obtaining trustworthy values of efficiency under the deviation of the current rotation frequency of the impeller from the nominal one is its universal characteristics, which are received when doing bench tests of a pump with a variable rotation frequency of the impeller. Since the universal characteristics of the pump present a family of curves of complex geometric form, their practical use was most difficult up to a certain moment. The main impediment on the way to their wide application for the analysis of energy efficiency of pumping systems and control in the online mode was their transfer from the geometric form to the analytical one. The appearance over the last years of a series of specialized computer programs for digitizing curves of complex geometric form opens up a way to their practical application.

However, the availability of modern computer software for automated digitizing of single curves of complex geometric forms does not allow one to read the efficiency value from the universal characteristics representing a whole family of such curves. At the same time, the operating point of the pump shifts into the area of possible modes of its operation along the trajectory set by the selected control method.

The trajectory intersects a considerable part of the isolines, each of which corresponds to a certain level of the efficiency value. Therefore, when analyzing the online modes of the pump operation, its greater part may appear in the intervals between the digitized isolines, between which the efficiency gradient can be quite significant, especially for the isolines placed at the periphery off the center with the maximal value of efficiency.

This circumstance has required development of a specialized algorithm and compiling a computer program on its basis for calculating the efficiency with the help of pre-digitized universal characteristics of the pump. Setting up a problem is based on the visual study of examples of the universal and head characteristics of rotodynamic vane blowers, which are presented in the catalogues of manufacturing companies. For the analytical expression of the head characteristic $H=f_1(Q,K)$ formula 2 for the nominal rotation frequency of the impeller ($\kappa_T=1$) was used, which was extrapolated for broadening of the region of possible modes of the pump by way of enhancing the rotation frequency by 20%, that is $H_2=f_2(Q,K)$, where $\kappa_{max}=1.2$. The worked out algorithm was applied to the mathematical model of the WiloCronoNormNL-125/200 pump, the universal characteristics of which are given in Fig. 2. At the same time, the following conditions of the pump operation were set: the parameters of the pump at the operating point ($Q_{op}=618m^3/h$, $H_{op}=45.2$m) of the optimal mode ($Q_{opt}=442m^3/h$, $H_{opt}=56.7$m, $\eta_{opt}=0.809$). The statistic head of the pipeline system was taken as equal to $H_{st}=26$m, the hydraulic factor $a=0.56$. The stabilization pressure in the pressure header of the system was taken as follows: $P_{stab}=4.6$ t.a. The range of load fluctuation was in the limits from $Q_{min}=140m^3/h$ up to $Q_{max}=600m^3/h$, the probabilities of flow distribution in the limits of the range were taken according to the statistical data.

Because of the limited volume of this publication, we will confine ourselves to a brief description of the worked out algorithm for determining the pump efficiency using the universal characteristics. The algorithm envisages the following sequence of implementing operations:

- The dimensions of the area of the set-up problem are defined visually at the universal characteristics:

Q_{max}, Q_{min} and H_{min}. Building of the area of determination begins «from the inside» from the point Z (see Fig. 2), where the maximal value of efficiency (Q_{opt}, H_{opt}) is achieved. Its coordinates are used for setting up appropriate boundaries of the region from the point of view of possible operation modes of the pump. For the pump used in the example, at specified conditions of its operation the following restrictions were set: by the X-axis — $Q_{min}=0{,}2Q_{opt}$; $Q_{max}=1{,}6Q_{opt}$, by the Y-axis — $H_{min}=0{,}2H_{opt}$. The value H_{max} was set by its maximal value in the interval (Q_{max}; Q_{min}) according to the head characteristic of the pump (see formula 2) at $K_T=1.2$.

- The minimal distance between the isolines is set up by the X-axis Δ. As a grid spacing the expression (grid*Δ) is used, where the fineness parameter is chosen in the interval 0<grid<1. In the considered algorithm the grid value was taken as equal to 2/5; consequently, this means that between two any isolines at least two grid nodes will be placed, which leads to a reduction in the error of the approximate calculation of efficiency.
- The number of the nodes along the X-axis is calculated according to the formula $N=1+(Q_{max}-Q_{min})/grid*\Delta$, with the round-off of the result up to an integer number.
- The position of the coordinate axes is set for digitization.

For a specific operation of the algorithm for calculation of the distances and angles on the XY plane, it is required that the grid spacing along the Y-axis be equal to the grid spacing along the X-axis. This demanded renormalization of the Y-axis with the introduction of scaling parameters: $S=(Q_{max}-Q_{min})/H_{max}-H_{min}$. The introduction of the scaling parameter S is equivalent to nondimensionalization of the system of coordinates. Renormalized coordinates $Y_{min}=SH_{min}$ and $Y_{max}=SH_{max}$ were used as the Y-axes for digitizing the isolineswhich were set up using the GetData Graph Digitizer program [18]

After this digitizing of the isolines in a certain order is performed, where the vectors are columns with the values for the isoline coordinates X, Y, a special matrix called "lines" can be filled in. Results of automated digitizing of the universal characteristics of the WiloCronoNormNL-125/200 pump are given in Fig.6. The result of automated digitizing is derived using the Get Data Graph Digitizer program with the distance between the points selected far smaller than the grid spacing. In the given example, the number of points, for instance, along line 1, which are placed most closely to the Z point of the maximal efficiency, is equal to 17. It should be noted that at the maximal achievable accuracy of manual digitization of the same curve at the diagram of the A4 format, the number of points equals 6, which is by three times less.

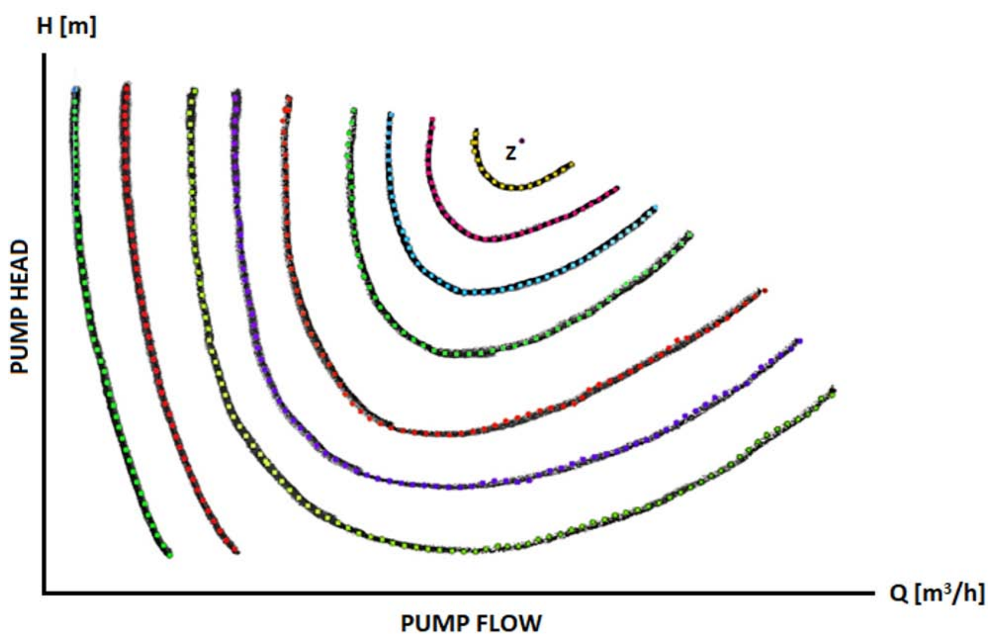

Fig. 6: Automatical Digitizing of Universal Characteristics of the Pump WILO-CronoNorm-NL125/200 (see Fig 2) using GetData Graph Digitizer Program

Calculation of the efficiency values in the area of possible operating modes of the pump was done by the method of the "fastest descent" between the isolines of the universal characteristics. In the method of the "fastest descent", it is required to find for a given node the distance from this node to the nearest isolines between which this node is located. If the distance up to the closest isolines with the efficiencies η_1 and η_2 are

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equal to d_1 and d_2, respectively, then the value of efficiency can be calculated according to the formula:

$$\eta = \frac{\eta_1 d_1 + \eta_2 d_1}{d_1 + d_2} \qquad (8)$$

The obtained algorithm, firstly, is a linear one and, secondly, a local one as it uses the information only about two neighbouring isolines. Herein lies its advantage, especially in the terms of this task when the number of isolines is insignificant by itself; moreover, the difference in the efficiency values between the neighbouring isolines can be quite inhomogeneous. This circumstance dramatically increases errors at the periphery when any grid algorithms are used. Formula 8 works best in the case if isolines are approximately parallel. The elaborated algorithm permits to obtain a perspective view of the surface of the efficiency values for the studied model of the pump in the R programming environment with an open source code (see Fig. 7).

The developed algorithm and computer program for reading the efficiency values with universal characteristics of the pump were included into a computation unit for its values using the LAB-MZ program for evaluation of the energy efficiency of pumping systems. Introduction of efficiency calculation into the efficiency computation unit in the LAB-MZ program for the algorithm of efficiency determination using the universal characteristics has broadened its possibilities substantially. This circumstance allows one to calculate the pump efficiency, which is necessary for the analysis of the energy efficiency of operation of pumping systems by two independent methods, namely: the traditional one using the formulas of the hydrodynamic similarity theory for vane blowers, as well as by way of pre-digitizing and reading of the actual values of efficiency from the universal characteristics of the studied pump.

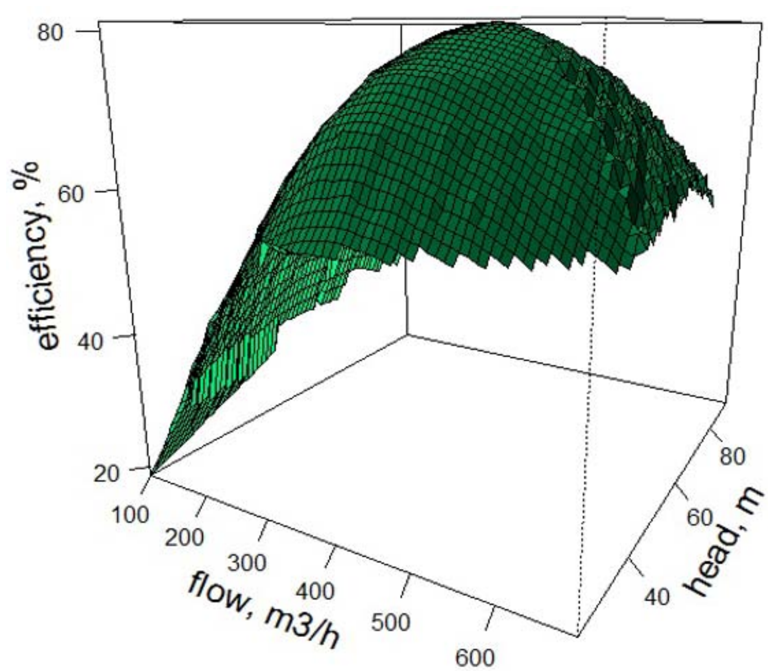

Fig. 7: A Perspective View of the Efficiency Surface

In this connection, a correlation of the efficiency values calculated by two different methods is of interest, as well as other important indices for determination of the energy efficiency during mathematical modelling of the WiloCronoNormNL-125/200 pumping unit in the case of using different control methods. Table 5 presents the energy efficient values for the following control ways: throttling of a pipeline system (pos.1), pressure stabilization in a pressure header (pos.2), minimization of excessive heads (the so-called its modification - proportional control) (pos.3), as well as minimization of the excessive heads using the universal characteristics (pos.4) by way of the transfer of the optimum of its efficiency to the point with the optimal parameters determined at the most probable load using a virtual pump (Q_{opt}=438m^3/h; H_{opt}=36m).

Table 5 presents the results of calculation of the pump efficiency with the help of the formulas of hydrodynamic similarity of vane superchargers (η_{csm}) and digitizing and reading the efficiency from the universal characteristic (η_{ucn}), as well as their difference: $\Delta\eta=\eta_{csm}-\eta_{ucn}$. In the Table, the values of the coefficient of deviation of the current rotation frequency of the impeller from the nominal one and some other indices are given also.

Table № 5: Correlation of the Values of the Pump Efficiency Calculated using the Formulas of Hydrodynamic Similarity and by Way of Pre-digitizing and Reading its Values from the Universal Characteristics for Different Control Methods

Compared parameters for different control methods	Units	Pump flow Q, m^3/h												
		140	180	220	260	300	340	380	420	460	500	540	580	600
1. Throttling of the pipeline system														
Head, H_{th}	m	62.6	62.9	62.7	62.2	61.5	60.4	59.1	57.5	55.6	53,5	51,0	48.3	46.5
Required head, H_{rq}	m	27.0	27.6	28.4	29.4	30.5	31.8	33.3	34.9	36.7	38,6	40,7	43,0	44.2
Supplied power	kW	55.0	58.4	61.7	65.2	68.9	72.7	76.7	81.1	85.9	91,1	97,1	104,0	107.9
Theoretical power of the coef. of rotation frequency $K_т$	kW	12.7	17.6	28.4	29.4	30.5	31.8	33.3	34.9	36.7	38,6	40,7	43.0	44.2
	-//-	1	1	1	1	1	1	1	1	1	1	1	1	1
Coefficiency, η_{csm}	%	43.6	52.9	60.9	67.5	72.9	76.9	79.7	81.1	82.1	79.9	77.3	73.5	71.1
Coefficiency, η_{hist}	%	48.1	57.9	64.0	69.7	72.6	78.0	80.3	80.9	80.9	80.3	76.9	73.7	70.9
$\Delta\eta$	%	+4.5	+5.0	+3.1	+2.2	-0.3	+1.1	+0.6	-0.2	-1.2	+0.4	-0.4	+0.2	-0.2
Energy consumed over one year	kW·h	S_w^{th} =691,4thous. kW·h												
2. The pressure stabilization in the pressure header														
Coef. of changes in the rotation frequency $K_т$	-//-	0.854	0.856	0.859	0.865	0.873	0.884	0,896	0,910	0,976	0,942	0,962	0,983	0,993
Coef, η_{csm}	%	48.3	57.9	65.6	71.6	76.0	78.8	80.8	80.5	79.6	77.9	75.4	72.3	70.5
Coef, η_{hist}	%	42.3	52.4	59.9	66.5	71.9	75.5	76.8	77.3	76.7	75.7	73.4	71.4	70.1
$\Delta\eta$	%	-6.0	-5.5	-5.7	-5.1	-4.0	-3.3	-4.0	-3.2	-2.9	-2.2	-2.0	-0.9	-0.4
Energy consumed over one year	kW·h	S_w^{stab} =557,5thous. kW·h												
3. Minimization of excessive heads (proportional control)														
Coef. of changes in the rotation frequency $K_т$	-//-	0.655	0.667	0.683	0,703	0,727	0,754	0,783	0,815	0,849	0,884	0.922	0.960	0.980
Coef, η_{csm}	%	52.6	61.4	67.8	72,2	74.6	76.3	76.8	76.4	75.4	74.1	72.4	70.4	69.4
Coef, η_{hist}	%	27.8	39.1	48.7	55.6	60.6	64.6	67.0	69.2	69.0	69.5	69.6	69.5	69.3
$\Delta\eta$	%	-24.8	-22.3	-19.1	-16.6	-14.3	-11.7	-9.8	-7.2	-6.4	-4.7	-2.8	-0.9	-0.1
Energy consumed over one year	kW·h	$S_w^{min\,h}$ =443,8 thous. kW·h												
4. Minimization of excessive heads using the universal characteristics of the pump by way of transfering its optimum of the efficiency to the point with optimal parameters determined for a virtual pump ($Q_{opt}^{virt} = 438m^3/h;\ H_{opt}^{virt} = 36m$)														
Coef. of changes in the rotation frequency $K_т$	-//-	0.818	0.832	0.850	0.869	0.891	0.915	0.942	0.970	1.00	1.031	1.644	1.100	1,116
Coef, η_{csm}	%	52.6	61.4	67.8	72.2	74.9	76.3	76.8	76.4	75.4	74.1	72,4	70.4	69,4
Coef, η_{hist}	%	44.2	53.6	61.9	69.0	73.9	77.7	80.0	81.1	80,6	78.7	75,1	70.9	68,5
$\Delta\eta$	%	-8.4	-7.8	-5.9	-3.2	-1.0	+1.4	+3.2	+4.7	+5.2	+4.6	+2,7	+0.5	-0,9
Energy consumed over one year	kW·h	$S_w^{opt} = 434$ thous. kW · h												

Note: η_{csm} is the value of the pump efficiency calculated using the formulas of the hydraulic similarity theory for vane pumps; η_{ucm} is the values of the pump efficiency computed using the universal (control) characteristics for the traditional method of selection of parameters of the pump; $\eta_{ухн}^{opt}$ is the same for the selection of optimal parameters with the help of a virtual pump; $\Delta\eta\eta_{csm} - \eta_{ucm}$.

Using the data from Table 5, Fig. 8 presents diagrams with the dependences of the pump efficiency on the flow for different methods of its determination in the case when the pump is controlled by way of minimization of the excessive pressure in the pressure header of the pumping station.

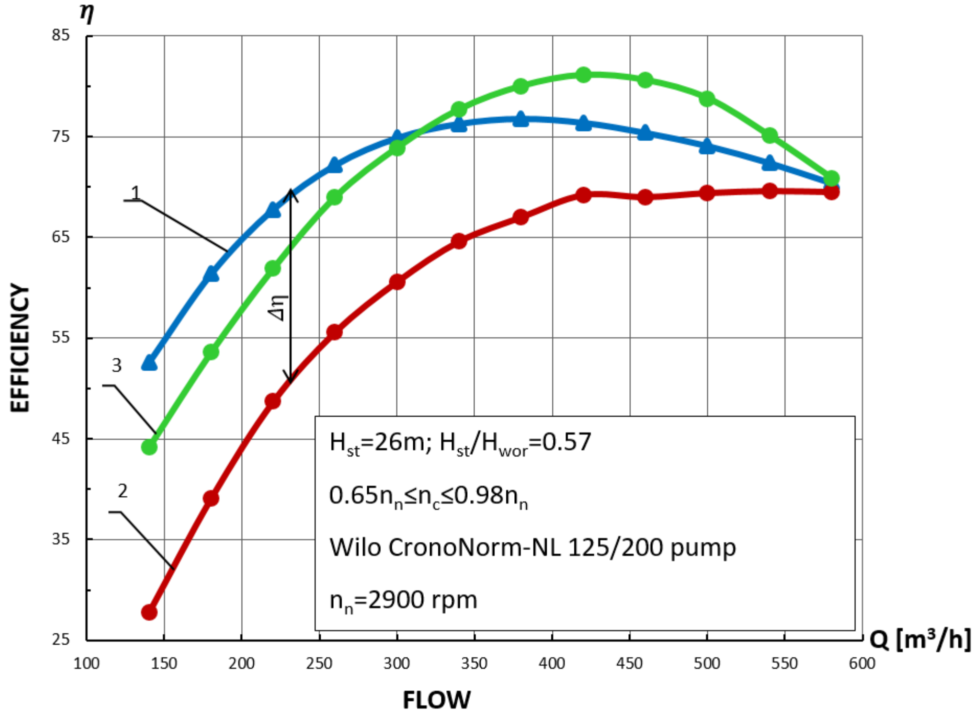

Fig. 8: Dependence of the pump efficiency on the flow for different methods of its determination when controlling the pump by way of minimization of excessive pressure in the pressure header of a pumping station (the so-called proportional control)

1. by the traditional method using the formulas of the hydrodynamic similarity theory for vane blowers;
2. using the universal (control) characteristics of the rotodynamic vane blower and traditional method of selection of its parameters;
3. using the universal characteristics of the pump by way of transfer of its optimum to the point with the parameters (Q_{opt}^{virt} and H_{opt}^{virt}) determined for a virtual pump.

The most substantial exceedance of efficiency calculated using the traditional technique (see Fig. 8, curve 1), from the efficiency values derived using the universal characteristics (see Fig. 8, curve 2), was obtained during minimization of excessive (above-normal) values of pressure in a pipeline system. Here, the efficiency exceedance across the entire range of changes in the flow amounted to from 1% up to 23.8%. It should be noted that the deviation of the current rotation frequency from the nominal was, respectively, in the range from 0.98 up to 0.65 from n_H. The hydraulic factor of the pipeline system was equal to 0.57. It is evident that for the pumping systems with a smaller value of the hydraulic factor (a<0.57) the deviation of the current rotation frequency from the nominal will be more considerable and, consequently, divergence of the efficiency values computed using the formulas of similarity and with the help of the universal characteristics will be more substantial.

In Table 5 and Fig.8 (curve 3), dependence of the efficiency values read from the digitized universal characteristics of the pump is given for the case of selection of the pump parameters for the existing pipeline system using preliminary calculation of the parameters of a virtual pump ($Q_{opt}^{virt} = 438m^3/h$;$H_{opt}^{virt} = 36m$). From the data provided in Table 5 (pos.4) and in Fig.8 (curve 3) it is seen that deviation of the current frequency from the nominal at coverage of the underload modes was of the order 18% (K_T=0.818), and efficiency exceedance computed using the traditional methodology was not more than 8.4% at underload modes (Q_T=340m^3/h). At coverage of loads in the range from 340 up to 600m^3/h the pump efficiency exceeds substantially the efficiency of a pump selected using the traditional technique (up to 5.2%) at the selection of its parameters by the most probable load. The peak load is then covered here by way of increasing the rotation frequency of the impeller by 12% relative to the nominal (K_T=1.116).

The determining factor for making a decision concerning the reasonability of investments into the reconstruction of the acting or newly projected pumping systems is the objective evaluation of their energy efficiency. Since the pump efficiency calculation is the most significant factor that affects the trustworthiness of the value of energy efficiency of operation of pumping systems, application of the traditional methodology for its determination using the formulas of hydrodynamic similarity can lead to a significant overestimation of their energy efficiency and, consequently, to making wrong decisions about the reasonability of their construction or reconstruction.

In this connection, of interest is the correlation of the indices of energy efficiency of the modelled pumping system with the NL-125/200 pumping unit for different control methods and techniques of selecting its parameters. The energy efficiency of the pumping system on the whole and factors that determine (form) it was calculated using the LAB-MZ computer program. At the same time, the calculation of the energy efficiency was conducted taking into account the use of two variants of calculating the efficiency values, namely, using the formulas of the hydrodynamic similarity theory and by way of pre-digitizing and reading the actuall values of efficiency from the universal characteristics of the pump. Results of calculation of the energy efficiency of operation of a pumping system over one year for two compared variants at different control methods are given in Table 6.

Table № 6: Comparison of the Energy Consumed by the Pump for Different Methods of Determination of its Efficiency and Control Methods

№	Method of control of a pumping unit and method of selection of the optimal parameters	Consumed electric power for different methods of calculation of the pump efficiency S_w, thous. kW·h		Increase of the value of consumed electric power at calculation of efficiency using the universal characteristics	
		by the formula of the hydrodynamic similarity theory	using the universal characteristics	thous. kW·h	%
1	Throttling of the pipeline system	691.4	691.4	0	0
2	Pressure stabilization in the pressure header of the pumping station	557.5	581.2	23.7	4.2
3	Minimization of excessive heads (proportional control) for different methods of selection the optimal parameters: - using the traditional method of selection by the maximal (peak) load	443.8	500.2	56.4	12.7
	- Optimization of the parameters of the selected equipment using a virtual pump	420.1	434.4	14.4	3,4
4	Theoretical minimally possible energy consumption (minimum of the energy functional)	405.7	405.7	0	0

It is visible from the data provided in Table 6 that the use of the formulas of hydrodynamical similarity for the calculation of efficiency of the pump leads to an overestimation of the expected energy efficiency of its operation. Thus, for instance, at pressure stabilization in the pressure header the exceedance of energy efficiency calculated using the formulas of hydrodynamic similarity was 4.2%, while at minimization of the excessive heads the exceedance of the calculated energy efficiency increased up to 12.7%. At pressure stabilization, as has been pointed out earlier, deviation of the current rotation frequency from the nominal was insignificant and amounted to 15%, increased up to 35% at minimization of the excessive heads, which entailed overestimation of the calculated energy efficiency up to 12.7%. That is why, broadening of the flow range (at pressure stabilization) or further decrease of the static component of the required head of the pipeline system (at the minimization of the excessive heads) will lead to a more substantial exceedance of the calculated efficiency relative to the actual one obtained through computation of the efficiency by way of using the universal characteristics of the pump.

As can be seen from Table 6, the largest energy effect is achieved in the case of the maximal binding of the characteristics of the installed pump to the parameters of the network by way of selection of the pump by the most probable load using a virtual pump. The transfer of the optimum of the efficiency of the selected pump to the point with optimal parameters obtained for a virtual pump (Q_{opt}^{virt} and H_{opt}^{virt}) allows one to achieve the maximal yearly energy efficiency (S_w=434 thousand kW*h) which ensures the maximal (up to 90%) extraction of the energy saving potential, which is not permitted by any of the known methods.

II. Main Conclusions

1. The question about reduction of the pump efficiency at deviation of the current rotation frequency of the impeller (n_T) from the nominal (n_H) had theoretical interest during a long time and was brought into the practical plane only with broad application of the variable-frequency drive. Regulation of the operating modes of the pump by way of changing the rotation frequency of the impeller leads to a change of its parameters: head, flow and efficiency, which requires recalculation of its characteristics from the nominal to the current rotation frequency. In connection with the violations of the conditions of hydrodynamic similarity at change of the frequency rotation, the greatest difficulties appear after recalculation of the efficiency of the rotodynamic vane blower, for which formulas of hydrodynamic similarity are traditionally used. The requirements of the Euro-zone standards to the minimally allowed value of the efficiency index (MEI) cover only a rather narrow part of the area of possible modes of the pump operation in the range of flows that are close to the optimal (from 0.75 up to 1.1 Q_{opt}). Due to the narrowness of the range of modes covered by the standard, the deviations of the current frequency from the nominal in this range will be in significant (no more than 10-15%), but in the frequently occurred in practice cases of the substantial widening of the load range (from 0.25 to 1.2 Q_{opt}) as well as in the case of applying more energy efficient ways of the pumping unit control (such as minimization of the exceeded heads or its variety – proportional regulation) will bring more significant deviations of the current rotation frequency from the nominal one. This circumstance will lead to the substantial decrease of the pump efficiency at the underload modes (Q_j< 0,75 Q_{opt}) that can level the efficiency of the pump operation entirely at covering the whole range of heads.
2. In order to obtain numerical values of the impact of deviation of the current frequency of the impeller from the nominal ($K_T = n_C/n_N$) on the pump efficiency, the specialized LAB-MZ computer program was used, which was developed by the authors at the SPL "Energosit" (Moscow) earlier and then substantially complemented. This computer software has been designed for evaluation of the energy efficiency value of the pumping systems operating with variable load. One of the distinctive features of the LAB-MZ program is the mathematical model of the water flow process in the pipeline system. This model consists of a number of statistical intervals with distribution of the probabilities of the flows in the statistical selection, which ensures a rather full adequateness of the model and physical process of the water flow. The presence of a sufficient number of statistical intervals provides a possibility to obtain minor gradients in each of them for such important parameters forming the current value of the energy efficiency as: efficiency, head, effective and supplied power, rotation frequency of the impeller, and its deviation from the nominal. This circumstance allows one to calculate the current values of the above listed parameters, as well as the energy efficiency of the pump operation in each interval. The total index of the energy efficiency of the system is considered as a sum of its values in the entire range of the load change. Besides, LAM-MZ contains an optimization unit, which executes selection of the pump equipment not by the traditional comparison of the manufactured pumps in the industry but by way of calculating the parameters of the virtual pump model by the most probable load. Then, basing on the parameters obtained for the virtual pump, a real pump with the parameters that are most close to the virtual one is selected. This is done to achieve a rather full compliance of the pump characteristics and parameters of the system, which ensures its maximal energy efficiency that cannot be achieved with the traditional method of selecting the parameters by the single operating point specified by the consumer.
3. The studies of the influence of deviation of the current rotation frequency from the nominal have been conducted on the basis of mathematical modelling of a pumping system with the NL-125/200(WILO) unit for a broad flow range (from 0.25 up to 1.4 Q_{opt}) and a full spectrum of variation of the hydraulic factor a (from 0 up to 0.88, where $\alpha = H_{st}/H_{op}$).Research has been done for different methods of control of a pumping unit. The values of deviation of the current frequency from the nominal for pressure stabilization in the pressure header were in the range from 0.856 up to 0.996n_N. A full-scale study of a whole number of objects of water supply systems has shown that in practice there are often cases when the deviation of the current frequency from the nominal at stabilization is more substantial, from 0.55 up to 0.6n_N, which leads to the pump operation with the efficiency not more than 20-30%. Of the greatest interest are the results of the study of the deviation of the current frequency from the nominal at the minimization of excessive (above the normative) heads (the so-called proportional control). Here, the most considerable in its value deviation is observed, especially in the area of underload modes at an insignificant statistical constituent of the required head H_{st}. At minimization of excessive heads the deviation of the current frequency from the nominal was in the limits from

$0.235n_N$ (at a=0) up to $0.821n_N$ (at a=0.88), while during selection of a pump by the most probable load the minimal value of deviation was from $0.392n_N$ (at a=0) up to 0.878 (at a=0.88). Since at determination of parameters using a virtual pump underload modes were covered by reduction of the rotation frequency, and maximal flows by way of its increasing in respect to the nominal, the value of the current frequency for the virtual pump was from $1.22n_{N)}$ (at a=0) up to 1.032 n_N (at a=0.88). At the minimization of excessive heads for the HSC-150-470 pump, the pump efficiency dependence on the degree of deviation of the impeller frequency from the nominal was obtained, which was in the interval from 0.56 up to $0.942n_N$ in the tests. Analysis of the dependence $\eta = f(K_T)$ has shown that at insignificant deviations of K_T in the range from 1.0 up to $0.8n_N$ it is practically linear, while at further increase of the deviation the dependence becomes considerably nonlinear. If at decrease of the rotation frequency by 20% (from 1.0 up to $0.8n_N$) the efficiency reduction amounted to 7% only, while at its further decrease by the same value (20%) the efficiency reduction was 32%, and the total reduction of efficiency in the linear and nonlinear areas amounted to 39% (at n_T=$0.56n_N$).

4. The only source of trustworthy information about the actual values of the efficiency of the pump operating with variable rotation frequency of the impeller is its universal (control) characteristics obtained by way of conducting bench tests of pumps at different rotation frequencies of the impeller. The isolines of efficiency of equal value that are given at the universal characteristics of the pump are elliptic and oval curves located concentrically relative to the point with the maximal value of efficiency in the H-Q coordinate system. A simple visual analysis of the efficiency characteristics by way of comparing the curves of similar modes presenting a family of parabolas, which follow from the formulas of hydrodynamic similarity, and the isolines of efficiency of equal level given at the universal characteristics of the pump allow one to find out the substantial qualitative difference for the same types of pumps. This circumstance required performing special research aimed at deriving numerical difference of efficiency values obtained using the formulas of similarity and read out from the universal characteristics of the pumps for a broad spectrum of variation of the parameters of water supply systems and methods of controlling the pumping equipment. Making the required comparison is possible only in the presence of special mathematical models, algorithms and computer programs, which allow us to obtain calculated data for such a comparison.
5. The main impediment on the way to a wide application of the universal characteristics for the mathematical modelling of pumping systems with the goal of determination of their energy efficiency and the use of control of pumping units in the systems is their transfer from the geometric form to the analytic one. The appearance in the last years of a number of specialized computer programs for automatized digitizing of curves of geometric shapes opens up a way to their practical application. However, the availability of such programs does not yet ensure a possibility of reading out the efficiency values from the universal characteristics. Solving this task has required elaboration of a special algorithm and compiling a computer program that permits to read out the efficiency values independently of the trajectory of motion of the operating point of the pump in the area of possible modes of its operation. For automatized digitizing we used the GetData Graph Digitizer program. Calculation of the efficiency values with reading its values from the universal characteristics is performed using the developed algorithm at the displacement of the operating point in the H-Q coordinates for any trajectory specified by the selected method of controlling the pump. The elaborated algorithm allows one to obtain a perspective view of the surface of the efficiency of the studied pump model in the R programming environment (programming language).
6. The elaborated algorithm and computer program for reading the efficiency values from the universal characteristics of the pumps were included into the unit for computing efficiency values using the LAB-MZ program that was considered earlier. This allowed us to calculate the efficiency of the pumps and to evaluate the energy efficiency of operation of pumping systems by two independent methods – using formulas of hydrodynamic similarity and by way of reading the actual values of efficiency from the universal characteristics after their pre-digitizing. Correlation of the efficiency values and energy efficiency indices was done by mathematical modelling of the Wilo-CronoNormNL-125/200 pumping unit for different control methods and technological conditions of its operation. The analysis of comparison of the efficiency values calculated by two different methods has shown a steady tendency towards exceeding the efficiency values derived according to the formulas of hydrodynamical similarity and by way of reading its values from the universal characteristics. Since exceedance of the efficiency values calculated based on the formulas of similarity over the values read out from the universal characteristics achieves its maximal value at the pump operation in underload modes and with a relatively insignificant

static constituent of the required head, consideration of these modes with considerable deviation of the current rotation frequency of the impeller from the nominal presents the greatest interest. So, for instance, at pressure stabilization in the pressure header deviation of the current rotation frequency from the nominal amounted to $n_C=0.85n_N$, and exceedance of efficiency by the similarity formulas was 5%. At the minimization of excessive (above norm) heads the deviation of the current frequency from the nominal was $n_T=0.65n_H$ and exceedance by the similarity formulas achieved 23.8%. At the same time, the value of the hydraulic factor was taken as equal to a=0.57. It is evident that for pumping systems with the least value of the hydraulic factor the required exceedance will be more substantial. Overestimation of the energy efficiency value of the pumping system on the whole due to the overestimation of efficiency at pressure stabilization amounted to 5.3% and to 12.7% at minimization of excessive heads. The obtained values of overestimation of the energy efficiency when traditional methods of efficiency determination are used testify to the necessity of using for estimation of the energy efficiency of pumping systems the efficiency values read out from the universal characteristics. This will allow us to objectively assess the energy efficiency of the pumping equipment proposed for installation (or existing) one in order to prevent adoption of insufficiently justified and wrong decisions about reasonability of investment both into reconstruction of the existing and construction of new pumping systems.

7. In the current time the universal characteristics of rotodynamic vane blowers are used chiefly for manual correction of efficiency at correction of the impeller diameter in the selected pump. The use of the universal characteristics for compiling universal algorithms for automatized reading the efficiency values at the evaluation of the energy efficiency of the pumping unit operation and selection of efficient methods of control of pumping systems revealed the necessity of enhancing the quality of the universal (control) characteristics, namely:

- Broadening of the area of coverage by the isolines of the efficiency of equal value, which are currently concentrated around the centre with the coordinate (Q_{opt} and H_{opt}). Broadening at the flow shall be from 0.25 up to $1.4Q_{opt}$; by the head — from 0.25 up to $1.2H_{opt}$, by the impeller rotation frequency — from 0.5 up to $1.25n_N$.
- Decrease of distance between the isolines placed at the periphery from the centre with the coordinates (Q_{opt} and H_{opt}); the isolines of the constant efficiency value shall cover the area of possible modes of the pump operation fully enough and evenly; differences of the efficiency gradients between two neighbouring isolines shall not differ substantially at the periphery and at the centre of the characteristics.
- The complete set of the pumping equipment released by a manufacturing company shall contain the universal characteristics, requirements to the quality that are to be determined by the standard. This standard must be worked out in the nearest time.

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Global Journals follows the definition of authorship set up by the Open Association of Research Society, USA. According to its guidelines, authorship criteria must be based on:

1. Substantial contributions to the conception and acquisition of data, analysis, and interpretation of findings.
2. Drafting the paper and revising it critically regarding important academic content.
3. Final approval of the version of the paper to be published.

Changes in Authorship

The corresponding author should mention the name and complete details of all co-authors during submission and in manuscript. We support addition, rearrangement, manipulation, and deletions in authors list till the early view publication of the journal. We expect that corresponding author will notify all co-authors of submission. We follow COPE guidelines for changes in authorship.

Copyright

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Appealing Decisions

Unless specified in the notification, the Editorial Board's decision on publication of the paper is final and cannot be appealed before making the major change in the manuscript.

Acknowledgments

Contributors to the research other than authors credited should be mentioned in Acknowledgments. The source of funding for the research can be included. Suppliers of resources may be mentioned along with their addresses.

Declaration of funding sources

Global Journals is in partnership with various universities, laboratories, and other institutions worldwide in the research domain. Authors are requested to disclose their source of funding during every stage of their research, such as making analysis, performing laboratory operations, computing data, and using institutional resources, from writing an article to its submission. This will also help authors to get reimbursements by requesting an open access publication letter from Global Journals and submitting to the respective funding source.

Preparing your Manuscript

Authors can submit papers and articles in an acceptable file format: MS Word (doc, docx), LaTeX (.tex, .zip or .rar including all of your files), Adobe PDF (.pdf), rich text format (.rtf), simple text document (.txt), Open Document Text (.odt), and Apple Pages (.pages). Our professional layout editors will format the entire paper according to our official guidelines. This is one of the highlights of publishing with Global Journals—authors should not be concerned about the formatting of their paper. Global Journals accepts articles and manuscripts in every major language, be it Spanish, Chinese, Japanese, Portuguese, Russian, French, German, Dutch, Italian, Greek, or any other national language, but the title, subtitle, and abstract should be in English. This will facilitate indexing and the pre-peer review process.

The following is the official style and template developed for publication of a research paper. Authors are not required to follow this style during the submission of the paper. It is just for reference purposes.

Manuscript Style Instruction (Optional)

- Microsoft Word Document Setting Instructions.
- Font type of all text should be Swis721 Lt BT.
- Page size: 8.27" x 11'", left margin: 0.65, right margin: 0.65, bottom margin: 0.75.
- Paper title should be in one column of font size 24.
- Author name in font size of 11 in one column.
- Abstract: font size 9 with the word “Abstract” in bold italics.
- Main text: font size 10 with two justified columns.
- Two columns with equal column width of 3.38 and spacing of 0.2.
- First character must be three lines drop-capped.
- The paragraph before spacing of 1 pt and after of 0 pt.
- Line spacing of 1 pt.
- Large images must be in one column.
- The names of first main headings (Heading 1) must be in Roman font, capital letters, and font size of 10.
- The names of second main headings (Heading 2) must not include numbers and must be in italics with a font size of 10.

Structure and Format of Manuscript

The recommended size of an original research paper is under 15,000 words and review papers under 7,000 words. Research articles should be less than 10,000 words. Research papers are usually longer than review papers. Review papers are reports of significant research (typically less than 7,000 words, including tables, figures, and references)

A research paper must include:

a) A title which should be relevant to the theme of the paper.
b) A summary, known as an abstract (less than 150 words), containing the major results and conclusions.
c) Up to 10 keywords that precisely identify the paper’s subject, purpose, and focus.
d) An introduction, giving fundamental background objectives.
e) Resources and techniques with sufficient complete experimental details (wherever possible by reference) to permit repetition, sources of information must be given, and numerical methods must be specified by reference.
f) Results which should be presented concisely by well-designed tables and figures.
g) Suitable statistical data should also be given.
h) All data must have been gathered with attention to numerical detail in the planning stage.

Design has been recognized to be essential to experiments for a considerable time, and the editor has decided that any paper that appears not to have adequate numerical treatments of the data will be returned unrefereed.

i) Discussion should cover implications and consequences and not just recapitulate the results; conclusions should also be summarized.
j) There should be brief acknowledgments.
k) There ought to be references in the conventional format. Global Journals recommends APA format.

Authors should carefully consider the preparation of papers to ensure that they communicate effectively. Papers are much more likely to be accepted if they are carefully designed and laid out, contain few or no errors, are summarizing, and follow instructions. They will also be published with much fewer delays than those that require much technical and editorial correction.

The Editorial Board reserves the right to make literary corrections and suggestions to improve brevity.

Format Structure

It is necessary that authors take care in submitting a manuscript that is written in simple language and adheres to published guidelines.

All manuscripts submitted to Global Journals should include:

Title

The title page must carry an informative title that reflects the content, a running title (less than 45 characters together with spaces), names of the authors and co-authors, and the place(s) where the work was carried out.

Author details

The full postal address of any related author(s) must be specified.

Abstract

The abstract is the foundation of the research paper. It should be clear and concise and must contain the objective of the paper and inferences drawn. It is advised to not include big mathematical equations or complicated jargon.

Many researchers searching for information online will use search engines such as Google, Yahoo or others. By optimizing your paper for search engines, you will amplify the chance of someone finding it. In turn, this will make it more likely to be viewed and cited in further works. Global Journals has compiled these guidelines to facilitate you to maximize the web-friendliness of the most public part of your paper.

Keywords

A major lynchpin of research work for the writing of research papers is the keyword search, which one will employ to find both library and internet resources. Up to eleven keywords or very brief phrases have to be given to help data retrieval, mining, and indexing.

One must be persistent and creative in using keywords. An effective keyword search requires a strategy: planning of a list of possible keywords and phrases to try.

Choice of the main keywords is the first tool of writing a research paper. Research paper writing is an art. Keyword search should be as strategic as possible.

One should start brainstorming lists of potential keywords before even beginning searching. Think about the most important concepts related to research work. Ask, "What words would a source have to include to be truly valuable in a research paper?" Then consider synonyms for the important words.

It may take the discovery of only one important paper to steer in the right keyword direction because, in most databases, the keywords under which a research paper is abstracted are listed with the paper.

Numerical Methods

Numerical methods used should be transparent and, where appropriate, supported by references.

Abbreviations

Authors must list all the abbreviations used in the paper at the end of the paper or in a separate table before using them.

Formulas and equations

Authors are advised to submit any mathematical equation using either MathJax, KaTeX, or LaTeX, or in a very high-quality image.

Tables, Figures, and Figure Legends

Tables: Tables should be cautiously designed, uncrowned, and include only essential data. Each must have an Arabic number, e.g., Table 4, a self-explanatory caption, and be on a separate sheet. Authors must submit tables in an editable format and not as images. References to these tables (if any) must be mentioned accurately.

Figures

Figures are supposed to be submitted as separate files. Always include a citation in the text for each figure using Arabic numbers, e.g., Fig. 4. Artwork must be submitted online in vector electronic form or by emailing it.

Preparation of Eletronic Figures for Publication

Although low-quality images are sufficient for review purposes, print publication requires high-quality images to prevent the final product being blurred or fuzzy. Submit (possibly by e-mail) EPS (line art) or TIFF (halftone/ photographs) files only. MS PowerPoint and Word Graphics are unsuitable for printed pictures. Avoid using pixel-oriented software. Scans (TIFF only) should have a resolution of at least 350 dpi (halftone) or 700 to 1100 dpi (line drawings). Please give the data for figures in black and white or submit a Color Work Agreement form. EPS files must be saved with fonts embedded (and with a TIFF preview, if possible).

For scanned images, the scanning resolution at final image size ought to be as follows to ensure good reproduction: line art: >650 dpi; halftones (including gel photographs): >350 dpi; figures containing both halftone and line images: >650 dpi.

Color charges: Authors are advised to pay the full cost for the reproduction of their color artwork. Hence, please note that if there is color artwork in your manuscript when it is accepted for publication, we would require you to complete and return a Color Work Agreement form before your paper can be published. Also, you can email your editor to remove the color fee after acceptance of the paper.

Tips for Writing a Good Quality Engineering Research Paper

Techniques for writing a good quality engineering research paper:

1. Choosing the topic: In most cases, the topic is selected by the interests of the author, but it can also be suggested by the guides. You can have several topics, and then judge which you are most comfortable with. This may be done by asking several questions of yourself, like "Will I be able to carry out a search in this area? Will I find all necessary resources to accomplish the search? Will I be able to find all information in this field area?" If the answer to this type of question is "yes," then you ought to choose that topic. In most cases, you may have to conduct surveys and visit several places. Also, you might have to do a lot of work to find all the rises and falls of the various data on that subject. Sometimes, detailed information plays a vital role, instead of short information. Evaluators are human: The first thing to remember is that evaluators are also human beings. They are not only meant for rejecting a paper. They are here to evaluate your paper. So present your best aspect.

2. Think like evaluators: If you are in confusion or getting demotivated because your paper may not be accepted by the evaluators, then think, and try to evaluate your paper like an evaluator. Try to understand what an evaluator wants in your research paper, and you will automatically have your answer. Make blueprints of paper: The outline is the plan or framework that will help you to arrange your thoughts. It will make your paper logical. But remember that all points of your outline must be related to the topic you have chosen.

3. Ask your guides: If you are having any difficulty with your research, then do not hesitate to share your difficulty with your guide (if you have one). They will surely help you out and resolve your doubts. If you can't clarify what exactly you require for your work, then ask your supervisor to help you with an alternative. He or she might also provide you with a list of essential readings.

4. Use of computer is recommended: As you are doing research in the field of research engineering then this point is quite obvious. Use right software: Always use good quality software packages. If you are not capable of judging good software, then you can lose the quality of your paper unknowingly. There are various programs available to help you which you can get through the internet.

5. Use the internet for help: An excellent start for your paper is using Google. It is a wondrous search engine, where you can have your doubts resolved. You may also read some answers for the frequent question of how to write your research paper or find a model research paper. You can download books from the internet. If you have all the required books, place importance on reading, selecting, and analyzing the specified information. Then sketch out your research paper. Use big pictures: You may use encyclopedias like Wikipedia to get pictures with the best resolution. At Global Journals, you should strictly follow here.

6. Bookmarks are useful: When you read any book or magazine, you generally use bookmarks, right? It is a good habit which helps to not lose your continuity. You should always use bookmarks while searching on the internet also, which will make your search easier.

7. Revise what you wrote: When you write anything, always read it, summarize it, and then finalize it.

8. Make every effort: Make every effort to mention what you are going to write in your paper. That means always have a good start. Try to mention everything in the introduction—what is the need for a particular research paper. Polish your work with good writing skills and always give an evaluator what he wants. Make backups: When you are going to do any important thing like making a research paper, you should always have backup copies of it either on your computer or on paper. This protects you from losing any portion of your important data.

9. Produce good diagrams of your own: Always try to include good charts or diagrams in your paper to improve quality. Using several unnecessary diagrams will degrade the quality of your paper by creating a hodgepodge. So always try to include diagrams which were made by you to improve the readability of your paper. Use of direct quotes: When you do research relevant to literature, history, or current affairs, then use of quotes becomes essential, but if the study is relevant to science, use of quotes is not preferable.

10. Use proper verb tense: Use proper verb tenses in your paper. Use past tense to present those events that have happened. Use present tense to indicate events that are going on. Use future tense to indicate events that will happen in the future. Use of wrong tenses will confuse the evaluator. Avoid sentences that are incomplete.

11. Pick a good study spot: Always try to pick a spot for your research which is quiet. Not every spot is good for studying.

12. Know what you know: Always try to know what you know by making objectives, otherwise you will be confused and unable to achieve your target.

13. Use good grammar: Always use good grammar and words that will have a positive impact on the evaluator; use of good vocabulary does not mean using tough words which the evaluator has to find in a dictionary. Do not fragment sentences. Eliminate one-word sentences. Do not ever use a big word when a smaller one would suffice.

Verbs have to be in agreement with their subjects. In a research paper, do not start sentences with conjunctions or finish them with prepositions. When writing formally, it is advisable to never split an infinitive because someone will (wrongly) complain. Avoid clichés like a disease. Always shun irritating alliteration. Use language which is simple and straightforward. Put together a neat summary.

14. Arrangement of information: Each section of the main body should start with an opening sentence, and there should be a changeover at the end of the section. Give only valid and powerful arguments for your topic. You may also maintain your arguments with records.

15. Never start at the last minute: Always allow enough time for research work. Leaving everything to the last minute will degrade your paper and spoil your work.

16. Multitasking in research is not good: Doing several things at the same time is a bad habit in the case of research activity. Research is an area where everything has a particular time slot. Divide your research work into parts, and do a particular part in a particular time slot.

17. Never copy others' work: Never copy others' work and give it your name because if the evaluator has seen it anywhere, you will be in trouble. Take proper rest and food: No matter how many hours you spend on your research activity, if you are not taking care of your health, then all your efforts will have been in vain. For quality research, take proper rest and food.

18. Go to seminars: Attend seminars if the topic is relevant to your research area. Utilize all your resources.

19. Refresh your mind after intervals: Try to give your mind a rest by listening to soft music or sleeping in intervals. This will also improve your memory. Acquire colleagues: Always try to acquire colleagues. No matter how sharp you are, if you acquire colleagues, they can give you ideas which will be helpful to your research.

20. Think technically: Always think technically. If anything happens, search for its reasons, benefits, and demerits. Think and then print: When you go to print your paper, check that tables are not split, headings are not detached from their descriptions, and page sequence is maintained.

21. Adding unnecessary information: Do not add unnecessary information like "I have used MS Excel to draw graphs." Irrelevant and inappropriate material is superfluous. Foreign terminology and phrases are not apropos. One should never take a broad view. Analogy is like feathers on a snake. Use words properly, regardless of how others use them. Remove quotations. Puns are for kids, not grunt readers. Never oversimplify: When adding material to your research paper, never go for oversimplification; this will definitely irritate the evaluator. Be specific. Never use rhythmic redundancies. Contractions shouldn't be used in a research paper. Comparisons are as terrible as clichés. Give up ampersands, abbreviations, and so on. Remove commas that are not necessary. Parenthetical words should be between brackets or commas. Understatement is always the best way to put forward earth-shaking thoughts. Give a detailed literary review.
22. Report concluded results: Use concluded results. From raw data, filter the results, and then conclude your studies based on measurements and observations taken. An appropriate number of decimal places should be used. Parenthetical remarks are prohibited here. Proofread carefully at the final stage. At the end, give an outline to your arguments. Spot perspectives of further study of the subject. Justify your conclusion at the bottom sufficiently, which will probably include examples.
23. Upon conclusion: Once you have concluded your research, the next most important step is to present your findings. Presentation is extremely important as it is the definite medium though which your research is going to be in print for the rest of the crowd. Care should be taken to categorize your thoughts well and present them in a logical and neat manner. A good quality research paper format is essential because it serves to highlight your research paper and bring to light all necessary aspects of your research.

Informal Guidelines of Research Paper Writing

Key points to remember:

- Submit all work in its final form.
- Write your paper in the form which is presented in the guidelines using the template.
- Please note the criteria peer reviewers will use for grading the final paper.

Final points:

One purpose of organizing a research paper is to let people interpret your efforts selectively. The journal requires the following sections, submitted in the order listed, with each section starting on a new page:

The introduction: This will be compiled from reference matter and reflect the design processes or outline of basis that directed you to make a study. As you carry out the process of study, the method and process section will be constructed like that. The results segment will show related statistics in nearly sequential order and direct reviewers to similar intellectual paths throughout the data that you gathered to carry out your study.

The discussion section:

This will provide understanding of the data and projections as to the implications of the results. The use of good quality references throughout the paper will give the effort trustworthiness by representing an alertness to prior workings.

Writing a research paper is not an easy job, no matter how trouble-free the actual research or concept. Practice, excellent preparation, and controlled record-keeping are the only means to make straightforward progression.

General style:

Specific editorial column necessities for compliance of a manuscript will always take over from directions in these general guidelines.

To make a paper clear: Adhere to recommended page limits.

Mistakes to avoid:

- Insertion of a title at the foot of a page with subsequent text on the next page.
- Separating a table, chart, or figure—confine each to a single page.
- Submitting a manuscript with pages out of sequence.
- In every section of your document, use standard writing style, including articles ("a" and "the").
- Keep paying attention to the topic of the paper.

- Use paragraphs to split each significant point (excluding the abstract).
- Align the primary line of each section.
- Present your points in sound order.
- Use present tense to report well-accepted matters.
- Use past tense to describe specific results.
- Do not use familiar wording; don't address the reviewer directly. Don't use slang or superlatives.
- Avoid use of extra pictures—include only those figures essential to presenting results.

Title page:

Choose a revealing title. It should be short and include the name(s) and address(es) of all authors. It should not have acronyms or abbreviations or exceed two printed lines.

Abstract: This summary should be two hundred words or less. It should clearly and briefly explain the key findings reported in the manuscript and must have precise statistics. It should not have acronyms or abbreviations. It should be logical in itself. Do not cite references at this point.

An abstract is a brief, distinct paragraph summary of finished work or work in development. In a minute or less, a reviewer can be taught the foundation behind the study, common approaches to the problem, relevant results, and significant conclusions or new questions.

Write your summary when your paper is completed because how can you write the summary of anything which is not yet written? Wealth of terminology is very essential in abstract. Use comprehensive sentences, and do not sacrifice readability for brevity; you can maintain it succinctly by phrasing sentences so that they provide more than a lone rationale. The author can at this moment go straight to shortening the outcome. Sum up the study with the subsequent elements in any summary. Try to limit the initial two items to no more than one line each.

Reason for writing the article—theory, overall issue, purpose.

- Fundamental goal.
- To-the-point depiction of the research.
- Consequences, including definite statistics—if the consequences are quantitative in nature, account for this; results of any numerical analysis should be reported. Significant conclusions or questions that emerge from the research.

Approach:

- o Single section and succinct.
- o An outline of the job done is always written in past tense.
- o Concentrate on shortening results—limit background information to a verdict or two.
- o Exact spelling, clarity of sentences and phrases, and appropriate reporting of quantities (proper units, important statistics) are just as significant in an abstract as they are anywhere else.

Introduction:

The introduction should "introduce" the manuscript. The reviewer should be presented with sufficient background information to be capable of comprehending and calculating the purpose of your study without having to refer to other works. The basis for the study should be offered. Give the most important references, but avoid making a comprehensive appraisal of the topic. Describe the problem visibly. If the problem is not acknowledged in a logical, reasonable way, the reviewer will give no attention to your results. Speak in common terms about techniques used to explain the problem, if needed, but do not present any particulars about the protocols here.

The following approach can create a valuable beginning:

- o Explain the value (significance) of the study.
- o Defend the model—why did you employ this particular system or method? What is its compensation? Remark upon its appropriateness from an abstract point of view as well as pointing out sensible reasons for using it.
- o Present a justification. State your particular theory(-ies) or aim(s), and describe the logic that led you to choose them.
- o Briefly explain the study's tentative purpose and how it meets the declared objectives.

Approach:

Use past tense except for when referring to recognized facts. After all, the manuscript will be submitted after the entire job is done. Sort out your thoughts; manufacture one key point for every section. If you make the four points listed above, you will need at least four paragraphs. Present surrounding information only when it is necessary to support a situation. The reviewer does not desire to read everything you know about a topic. Shape the theory specifically—do not take a broad view.

As always, give awareness to spelling, simplicity, and correctness of sentences and phrases.

Procedures (methods and materials):

This part is supposed to be the easiest to carve if you have good skills. A soundly written procedures segment allows a capable scientist to replicate your results. Present precise information about your supplies. The suppliers and clarity of reagents can be helpful bits of information. Present methods in sequential order, but linked methodologies can be grouped as a segment. Be concise when relating the protocols. Attempt to give the least amount of information that would permit another capable scientist to replicate your outcome, but be cautious that vital information is integrated. The use of subheadings is suggested and ought to be synchronized with the results section.

When a technique is used that has been well-described in another section, mention the specific item describing the way, but draw the basic principle while stating the situation. The purpose is to show all particular resources and broad procedures so that another person may use some or all of the methods in one more study or referee the scientific value of your work. It is not to be a step-by-step report of the whole thing you did, nor is a methods section a set of orders.

Materials:

Materials may be reported in part of a section or else they may be recognized along with your measures.

Methods:

- Report the method and not the particulars of each process that engaged the same methodology.
- Describe the method entirely.
- To be succinct, present methods under headings dedicated to specific dealings or groups of measures.
- Simplify—detail how procedures were completed, not how they were performed on a particular day.
- If well-known procedures were used, account for the procedure by name, possibly with a reference, and that's all.

Approach:

It is embarrassing to use vigorous voice when documenting methods without using first person, which would focus the reviewer's interest on the researcher rather than the job. As a result, when writing up the methods, most authors use third person passive voice.

Use standard style in this and every other part of the paper—avoid familiar lists, and use full sentences.

What to keep away from:

- Resources and methods are not a set of information.
- Skip all descriptive information and surroundings—save it for the argument.
- Leave out information that is immaterial to a third party.

Results:

The principle of a results segment is to present and demonstrate your conclusion. Create this part as entirely objective details of the outcome, and save all understanding for the discussion.

The page length of this segment is set by the sum and types of data to be reported. Use statistics and tables, if suitable, to present consequences most efficiently.

You must clearly differentiate material which would usually be incorporated in a study editorial from any unprocessed data or additional appendix matter that would not be available. In fact, such matters should not be submitted at all except if requested by the instructor.

Content:

- Sum up your conclusions in text and demonstrate them, if suitable, with figures and tables.
- In the manuscript, explain each of your consequences, and point the reader to remarks that are most appropriate.
- Present a background, such as by describing the question that was addressed by creation of an exacting study.
- Explain results of control experiments and give remarks that are not accessible in a prescribed figure or table, if appropriate.
- Examine your data, then prepare the analyzed (transformed) data in the form of a figure (graph), table, or manuscript.

What to stay away from:

- Do not discuss or infer your outcome, report surrounding information, or try to explain anything.
- Do not include raw data or intermediate calculations in a research manuscript.
- Do not present similar data more than once.
- A manuscript should complement any figures or tables, not duplicate information.
- Never confuse figures with tables—there is a difference.

Approach:

As always, use past tense when you submit your results, and put the whole thing in a reasonable order.

Put figures and tables, appropriately numbered, in order at the end of the report.

If you desire, you may place your figures and tables properly within the text of your results section.

Figures and tables:

If you put figures and tables at the end of some details, make certain that they are visibly distinguished from any attached appendix materials, such as raw facts. Whatever the position, each table must be titled, numbered one after the other, and include a heading. All figures and tables must be divided from the text.

Discussion:

The discussion is expected to be the trickiest segment to write. A lot of papers submitted to the journal are discarded based on problems with the discussion. There is no rule for how long an argument should be.

Position your understanding of the outcome visibly to lead the reviewer through your conclusions, and then finish the paper with a summing up of the implications of the study. The purpose here is to offer an understanding of your results and support all of your conclusions, using facts from your research and generally accepted information, if suitable. The implication of results should be fully described.

Infer your data in the conversation in suitable depth. This means that when you clarify an observable fact, you must explain mechanisms that may account for the observation. If your results vary from your prospect, make clear why that may have happened. If your results agree, then explain the theory that the proof supported. It is never suitable to just state that the data approved the prospect, and let it drop at that. Make a decision as to whether each premise is supported or discarded or if you cannot make a conclusion with assurance. Do not just dismiss a study or part of a study as "uncertain."

Research papers are not acknowledged if the work is imperfect. Draw what conclusions you can based upon the results that you have, and take care of the study as a finished work.

- You may propose future guidelines, such as how an experiment might be personalized to accomplish a new idea.
- Give details of all of your remarks as much as possible, focusing on mechanisms.
- Make a decision as to whether the tentative design sufficiently addressed the theory and whether or not it was correctly restricted. Try to present substitute explanations if they are sensible alternatives.
- One piece of research will not counter an overall question, so maintain the large picture in mind. Where do you go next? The best studies unlock new avenues of study. What questions remain?
- Recommendations for detailed papers will offer supplementary suggestions.

Approach:

When you refer to information, differentiate data generated by your own studies from other available information. Present work done by specific persons (including you) in past tense.

Describe generally acknowledged facts and main beliefs in present tense.

The Administration Rules

Administration Rules to Be Strictly Followed before Submitting Your Research Paper to Global Journals Inc.

Please read the following rules and regulations carefully before submitting your research paper to Global Journals Inc. to avoid rejection.

Segment draft and final research paper: You have to strictly follow the template of a research paper, failing which your paper may get rejected. You are expected to write each part of the paper wholly on your own. The peer reviewers need to identify your own perspective of the concepts in your own terms. Please do not extract straight from any other source, and do not rephrase someone else's analysis. Do not allow anyone else to proofread your manuscript.

Written material: You may discuss this with your guides and key sources. Do not copy anyone else's paper, even if this is only imitation, otherwise it will be rejected on the grounds of plagiarism, which is illegal. Various methods to avoid plagiarism are strictly applied by us to every paper, and, if found guilty, you may be blacklisted, which could affect your career adversely. To guard yourself and others from possible illegal use, please do not permit anyone to use or even read your paper and file.

Criterion for Grading a Research Paper (Compilation) by Global Journals

Please note that following table is only a Grading of "Paper Compilation" and not on "Performed/Stated Research" whose grading solely depends on Individual Assigned Peer Reviewer and Editorial Board Member. These can be available only on request and after decision of Paper. This report will be the property of Global Journals.

Topics	Grades		
	A-B	**C-D**	**E-F**
Abstract	Clear and concise with appropriate content, Correct format. 200 words or below	Unclear summary and no specific data, Incorrect form Above 200 words	No specific data with ambiguous information Above 250 words
Introduction	Containing all background details with clear goal and appropriate details, flow specification, no grammar and spelling mistake, well organized sentence and paragraph, reference cited	Unclear and confusing data, appropriate format, grammar and spelling errors with unorganized matter	Out of place depth and content, hazy format
Methods and Procedures	Clear and to the point with well arranged paragraph, precision and accuracy of facts and figures, well organized subheads	Difficult to comprehend with embarrassed text, too much explanation but completed	Incorrect and unorganized structure with hazy meaning
Result	Well organized, Clear and specific, Correct units with precision, correct data, well structuring of paragraph, no grammar and spelling mistake	Complete and embarrassed text, difficult to comprehend	Irregular format with wrong facts and figures
Discussion	Well organized, meaningful specification, sound conclusion, logical and concise explanation, highly structured paragraph reference cited	Wordy, unclear conclusion, spurious	Conclusion is not cited, unorganized, difficult to comprehend
References	Complete and correct format, well organized	Beside the point, Incomplete	Wrong format and structuring

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