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Improving the Efficiency of Pumping Equipment by Enhancing the Cooling System of a Submersible Pump Motor

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ABSTRACT

Submersible pumping units used in the in-situ leaching (ISL) technology of the Kyzylkum region may be subject to electric motor overheating caused by insufficient fluid flow velocity around the motor, the formation of mineral deposits on the housing surface, and continuous operation in an aggressive environment. These factors lead to premature degradation of the winding insulation and reduced efficiency of the pumping equipment. This paper proposes a technical solution — a cooling jacket installed on the suction nozzle of a submersible pump — and develops a mathematical model of heat transfer in the annular channel between the motor housing and the jacket. The model is based on forced convection theory and accounts for the Reynolds and Nusselt numbers, the hydraulic diameter of the annular gap, and the dependence of the heat transfer coefficient on flow velocity. Numerical calculations were performed for two pump models, USK 408/42 and URN 6 25/14, operating in wells with diameters of 159 mm and 195 mm, respectively. The results show that the cooling jacket increases the fluid velocity along the motor of the USK 408/42 pump from a subcritical value (about 0.17 m/s) to 0.69 m/s, and provides velocities of 0.59–1.43 m/s for the URN 6 25/14 pump; at a fluid temperature of 40 °C, the calculated overheating of the USK 408/42 motor surface relative to the fluid decreases from 9.4 °C without a jacket to 2.4 °C with a jacket inner diameter of 113 mm. It is shown that the main function of the jacket is to ensure longitudinal flow along the motor, which may not be achieved without a jacket. Conjugate CFD simulation in SolidWorks Flow Simulation showed a reduction in the mean motor surface temperature of 12.6 °C (7.0 °C according to the analytical model) and in the maximum temperature inside the motor of 16.5 °C. Laboratory tests on the SGU-TsNS-012-6LR-PK test bench with a Chimp QDX3-20-0.55(Y) submersible pump and jackets of 220–240 mm diameter qualitatively confirmed that the motor surface temperature decreases as the jacket diameter decreases and the flow rate increases (overheating reduced from 14.2 to 0.5 °C). The capital cost of the jacket is about USD 37 per pump, which is negligible compared with the cost of motor repair; the effect of the jacket on energy consumption remains to be verified experimentally.

Keywords: submersible pump; motor cooling; cooling jacket; forced convection; heat transfer; Reynolds number; in-situ leaching; pump efficiency.

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Introduction

Pumping systems are among the largest consumers of electricity: they account for nearly 20% of the world's electrical energy demand, and energy costs typically represent the largest component of the life cycle cost of a pumping installation [1]. Moreover, the overall efficiency of pumping systems is usually considerably lower than that of individual pumps due to system losses [1]. In the in-situ leaching (ISL) operations of the Kyzylkum region, submersible pumping units are the main means of lifting pregnant solutions to the surface [2–4]. These units operate continuously under severe conditions that combine mechanical impurities, gas content in the pumped fluid, and the formation of deposits on the equipment [5, 6].

One of the main causes of submersible pump failure is motor overheating [5, 7–9]. Overheating occurs when the fluid velocity around the motor falls below the critical value of 0.2 m/s: with slow fluid movement, the motor heats the adjacent layer to temperatures that are dangerous for the winding insulation. Under such conditions, the winding temperature may exceed the level permissible for the insulation, which accelerates insulation degradation, increases the current drawn and disrupts normal pump operation [10–12]. An additional aggravating factor is the formation of mineral scale — mainly iron, manganese and calcium compounds — on the outer surface of the motor housing. Since the motor surface temperature is higher than that of the surrounding fluid, these substances oxidise and accumulate as a thermally insulating film that further intensifies heating [5, 13].

The consequences of motor overheating are both technical and economic. Loss of insulating properties leads to premature winding breakdown, unscheduled equipment downtime and high repair costs. Since submersible pumps in ISL installations operate almost continuously throughout the year (about 8,760 h), even short interruptions can disrupt the overall hydrometallurgical process. The economic significance of the problem is amplified by the energy intensity of pumping: at the nominal operating point of the USK 408/42 pump, the actual power consumption is about 7.4 kW, corresponding to an annual electricity consumption of approximately 55,100 kWh per unit.

The most technically rational solution to this problem is to install a cooling jacket on the suction nozzle of the submersible pump. The jacket restricts the cross-section of the annular channel around the motor housing, thereby accelerating the fluid flow and ensuring effective convective heat removal. The concept of flow-through cooling of submersible pump motors is not new: leading pump manufacturers recommend it as standard practice when the natural flow velocity in the well is insufficient [14, 15]. The influence of the motor–shroud gap on heat transfer has been studied numerically for electrical submersible pump systems in oil production [16, 17], and forced convection in annular channels has been considered, in particular, in [18]. In the authors' previous work [13], flow velocities were calculated for a range of jacket diameters and the velocity field was simulated in SolidWorks; however, the thermal aspect of the problem — the relationship between motor overheating and jacket diameter — was not considered. A systematic quantitative analysis of the optimal jacket diameter that accounts for heat transfer for the pump–well combinations typical of the Kyzylkum region has not been published previously. The present work fills this gap by developing a mathematical model of the cooling process, performing parametric calculations, comparing them with CFD simulations in SolidWorks and with laboratory tests, and providing a qualitative techno-economic assessment of the proposed solution.

The USK 408/42 and URN 6 25/14 pumps, the most widely used submersible pump models in the domestic ISL industry, were selected as the objects of study. The differences in their geometric and hydraulic parameters make it possible to treat them as representatives of two characteristic classes of installations and to formulate generalised engineering recommendations.

Problem Analysis and Operating Modes of Submersible Pumps

1.1. Main Causes of Submersible Pump Motor Overheating

The submersible pump and its drive motor form a mechanically integrated monoblock unit. The motor is cooled exclusively by the fluid flowing over its outer surface; unlike surface-mounted electric motors, the submersible unit has no independent air cooling system. The two pump models studied in this work — USK 408/42 (motor diameter $D_{\text{motor}} = 93$ mm) and URN 6 25/14 ($D_{\text{motor}} = 145$ mm) — are designed for production wells and rated for continuous operation according to the ISL process regulations, i.e. about 8,760 h per year.

The motor temperature rises under the following conditions: (1) the fluid velocity around the motor falls below 0.2 m/s; (2) the pump is installed in a large-diameter well or in an open reservoir, where natural convection is the only cooling mechanism; (3) the pumped fluid contains elevated concentrations of iron, manganese or calcium, leading to the formation of thermally insulating mineral scale on the motor housing. When the flow velocity falls below the minimum threshold, the winding temperature may approach the limits permissible for class F and H insulation, resulting in its accelerated ageing.

1.2. Effect of Heating on Pump Performance

From the standpoint of solid-state physics, an increase in metal temperature intensifies the vibrations of ions in the crystal lattice, shortening the mean free path of conduction electrons and increasing electrical resistance. For the motor as a whole, this manifests itself as increased power consumption, reduced motor efficiency η , and accelerated thermal ageing of the winding insulation. Each 10 °C increase in temperature approximately doubles the rate of insulation ageing according to the Arrhenius law [11, 12]. In addition, when the motor surface is covered with a layer of mineral scale acting as thermal insulation, the effective heat transfer coefficient drops sharply, creating local hot spots on the winding insulation that can cause premature electrical breakdown [5, 10].

The annual electricity consumption of a pumping unit is calculated using the standard energy balance formula:

$$W = P_{\text{act}} \cdot \text{EUF} \cdot t, \text{ kWh/year}; \quad (1)$$

where P_{act} is the actual power of the pumping unit, kW; $\text{EUF} = 0.85$ is the equipment utilisation factor; $t = 8,760$ h/year is the annual operating time.

For the USK 408/42 pump at its nominal operating point ($Q = 8$ m³/h), the actual power is $P_{\text{act}} = 7.4$ kW, whence $W = 7.4 \times 0.85 \times 8,760 = 55,100$ kWh/year. When the motor overheats, power consumption increases due to the higher winding resistance, which entails additional electricity consumption and accelerated insulation wear. Improving the energy efficiency of submersible pump installations has been addressed, in particular, in [19–22].

Technical Solution: Cooling Jacket for the Submersible Pump Motor

The proposed device is a cylindrical cooling jacket attached to the upper part of the pump suction nozzle; its length exceeds that of the motor housing by 20–50 cm. The difference between the inner diameter of the jacket and the outer diameter of the motor (i.e. the hydraulic diameter of the channel) is typically 20–30 mm. The jacket forms a directed flow channel, forcing the working fluid to pass in close proximity to the motor surface before entering the pump suction zone, thereby substantially increasing the local flow velocity and intensifying convective heat transfer.

The cooling jacket is recommended in the following cases: (1) when the natural fluid flow along the motor is insufficient to remove heat, including under current imbalance, dry running, overload or high ambient temperature; (2) when aggressive fluids are pumped, since the corrosion rate roughly doubles with every 10 °C rise in temperature. The design of the cooling jacket is shown in Figure 1.

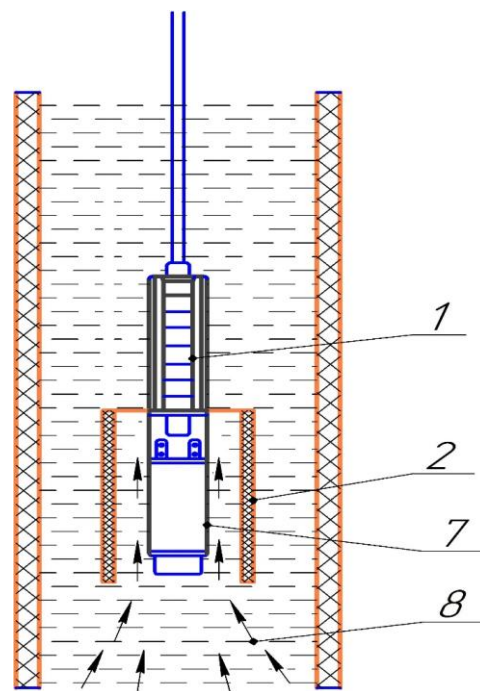
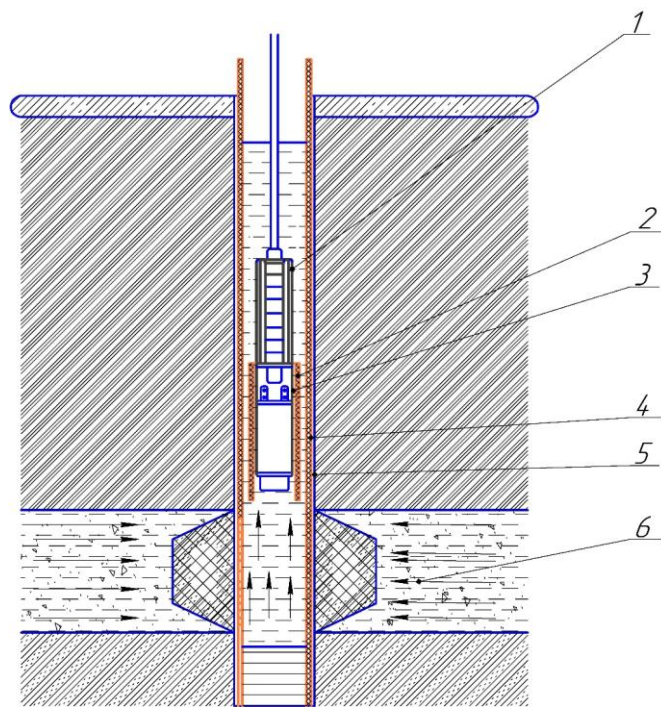


Figure 1. General view of a production well with a submersible pump and cooling jacket: a) cross-section of the well unit; b) operating principle of the cooling jacket. 1 — submersible pumping unit; 2 — cooling jacket; 3 — suction nozzle; 4 — casing; 5 — production well; 6 — productive formation; 7 — motor section of the pump; 8 — direction of fluid flow.

The flow velocity in the annular channel should be in the range of 0.2–3.0 m/s; this is the technical requirement of most leading manufacturers of well pumps for maintaining a normal thermal regime of the motor.

Mathematical Modelling of the Cooling Process

3.1. Flow Hydrodynamics in the Annular Gap

The motor is treated as a long cylinder in longitudinal forced flow within the annular channel between the motor housing (diameter D_{motor}) and the inner surface of the cooling jacket (diameter D_{jacket}). The fluid velocity V in this channel is determined from the continuity equation:

$$V = \frac{4 \cdot Q}{\pi \cdot (D_{\text{jacket}}^2 - D_{\text{motor}}^2)} \quad (2)$$

For practical calculations, with the flow rate Q in m^3/h and diameters in mm:

$$V = \frac{353.6 \cdot Q}{D_{\text{jacket}}^2 - D_{\text{motor}}^2} \quad (3)$$

where Q is the pump flow rate, m^3/h ; D_{jacket} is the inner diameter of the cooling jacket, mm; D_{motor} is the outer diameter of the motor, mm.

Calculations were performed for two pump–well combinations: (1) USK 408/42 in a 159 mm well ($D_{\text{motor}} = 93$ mm, $Q = 8$ m^3/h); (2) URN 6 25/14 in a 195 mm well ($D_{\text{motor}} = 145$ mm, $Q = 25$ m^3/h). These combinations correspond to the most common configurations at ISL sites in the Kyzylkum region.

3.2. Heat Transfer and Similarity Equations

The flow regime is characterised by the Reynolds number:

$$Re = \frac{V_{\text{liquid}} \cdot d_h}{\nu} \quad (4)$$

where $d_h = D_{\text{jacket}} - D_{\text{motor}}$ is the hydraulic diameter of the annular channel, m; ν is the kinematic viscosity of the fluid, m^2/s .

The intensity of heat transfer is expressed through the Nusselt number Nu , which relates the heat transfer coefficient α to the thermal conductivity of the fluid λ :

$$Nu = \frac{\alpha \cdot d_h}{\lambda} \quad (5)$$

For turbulent flow ($Re > 10^4$), the Dittus–Boelter equation is applied [23]:

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \quad (6)$$

Substituting the expression for Re gives the heat transfer coefficient as a function of flow velocity:

$$\begin{aligned} \alpha &= \frac{\lambda}{d_h} \cdot 0.023 \cdot \left(\frac{V \cdot d_h}{\nu} \right)^{0.8} \cdot Pr^{0.4} \\ \alpha &= 0.023 \cdot \frac{\lambda \cdot Pr^{0.4}}{\nu^{0.8} \cdot d_h^{0.2}} \cdot V^{0.8} \end{aligned} \quad (7)$$

This relationship shows that the cooling intensity is proportional to the flow velocity raised to the power of 0.8: $\alpha \sim V^{0.8}$. The non-linear nature of this dependence means that even a moderate increase in flow velocity yields a noticeable gain in heat removal.

3.3. Motor Cooling Equation

In steady-state operation, the thermal power dissipated by the motor equals the heat removed by the surrounding fluid. The heat flow P through the motor surface of area $S = \pi \cdot D_{\text{motor}} \cdot L_{\text{motor}}$ is determined from the motor efficiency:

$$P = P_{\text{nom}} \cdot (1 - \eta) \quad (8)$$

According to Newton's law of cooling:

$$P = \alpha \cdot S \cdot \Delta T \quad (9)$$

The motor overheating ΔT is then expressed as:

$$\Delta T(V) = \frac{P \cdot v^{0.8} \cdot d_h^{0.2}}{0.023 \cdot \lambda \cdot Pr^{0.4} \cdot S \cdot V^{0.8}} \quad (10)$$

Equation (10) is the main result of the mathematical model. It relates motor overheating to the fluid velocity and the geometric parameters of the annular channel, thereby providing a rational basis for selecting the cooling jacket diameter. Since $\Delta T \propto V^{-0.8}$, doubling the flow velocity reduces overheating by a factor of approximately 1.74.

The input data for the calculation using Equation (10) for the USK 408/42 pump were as follows: rated power input $P_{nom} = 7.5$ kW; motor efficiency $\eta = 0.65$, the lower limit of the rated range of 0.65–0.85, which gives a conservative (overestimated) heat release; length of the cooled motor surface $L_{motor} = 1.0$ m. Hence $S = 0.292$ m² and $P = 2625$ W. The base case assumes a well fluid temperature of $T_{liquid} = 40$ °C, with the thermophysical properties of water at this temperature [23]: $\lambda = 0.632$ W/(m·K), $\nu = 0.662 \cdot 10^{-6}$ m²/s, $Pr = 4.34$. To assess the most unfavourable conditions, a case with $T_{liquid} = 70$ °C ($\lambda = 0.662$ W/(m·K), $\nu = 0.410 \cdot 10^{-6}$ m²/s, $Pr = 2.53$) was also considered.

The flow pattern and temperature distribution in the annular channel of the cooling jacket are shown in Figure 2.

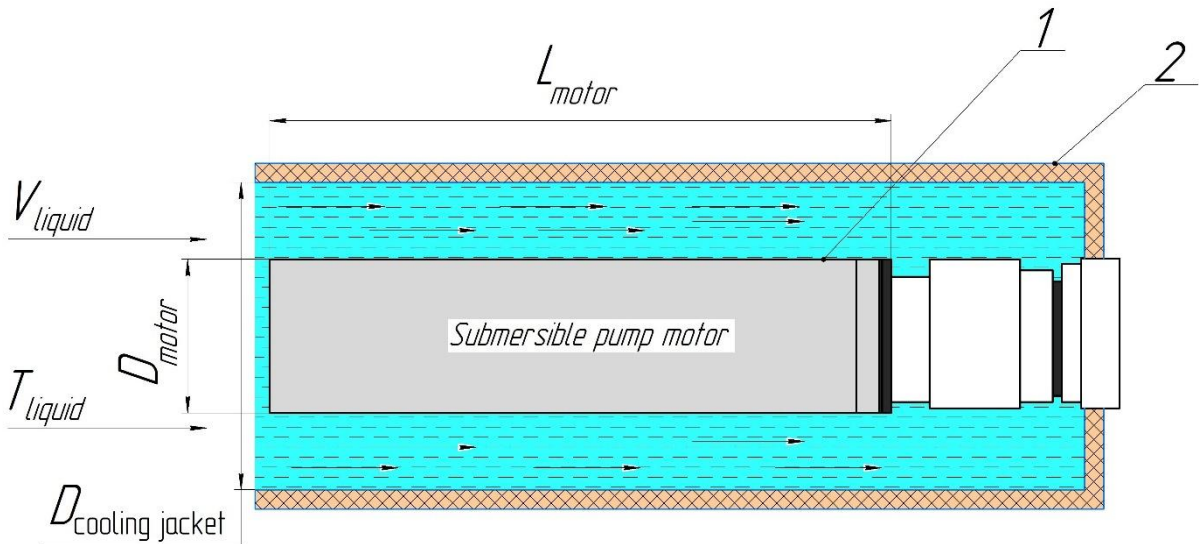


Figure 2. Flow pattern and temperature distribution in the annular channel of the motor cooling jacket:

1 — submersible pump motor; 2 — cooling jacket; T_{liquid} — fluid temperature, °C; V_{liquid} — fluid velocity, m/s; D_{jacket} — inner diameter of the cooling jacket, mm; D_{motor} — motor diameter, mm; L_{motor} — motor length, m.

Results and Discussion

4.1. Selection of the Optimal Jacket Diameter: Parametric Calculations

The dependence of the fluid velocity on the inner diameter of the cooling jacket was calculated using Equations (2) and (3) for both pump–well configurations. The results are summarised in Table 1 and shown in Figures 3 and 4.

Table 1. Determination of the optimal cooling jacket diameter for both pump models

Pump model	Q, m ³ /h	D _{well} , mm	D _{motor} , mm	D _{jacket} , mm	V, m/s
USK 408/42	8	159	93	113	0.69

Pump model	Q, m ³ /h	D _{well} , mm	D _{motor} , mm	D _{jacket} , mm	V, m/s
				123	0.44
				133	0.31
				143	0.24
				153	0.19
URN 6 25/14	25	195	145	165	1.43
				170	1.12
				175	0.92
				180	0.78
				185	0.67
				190	0.59

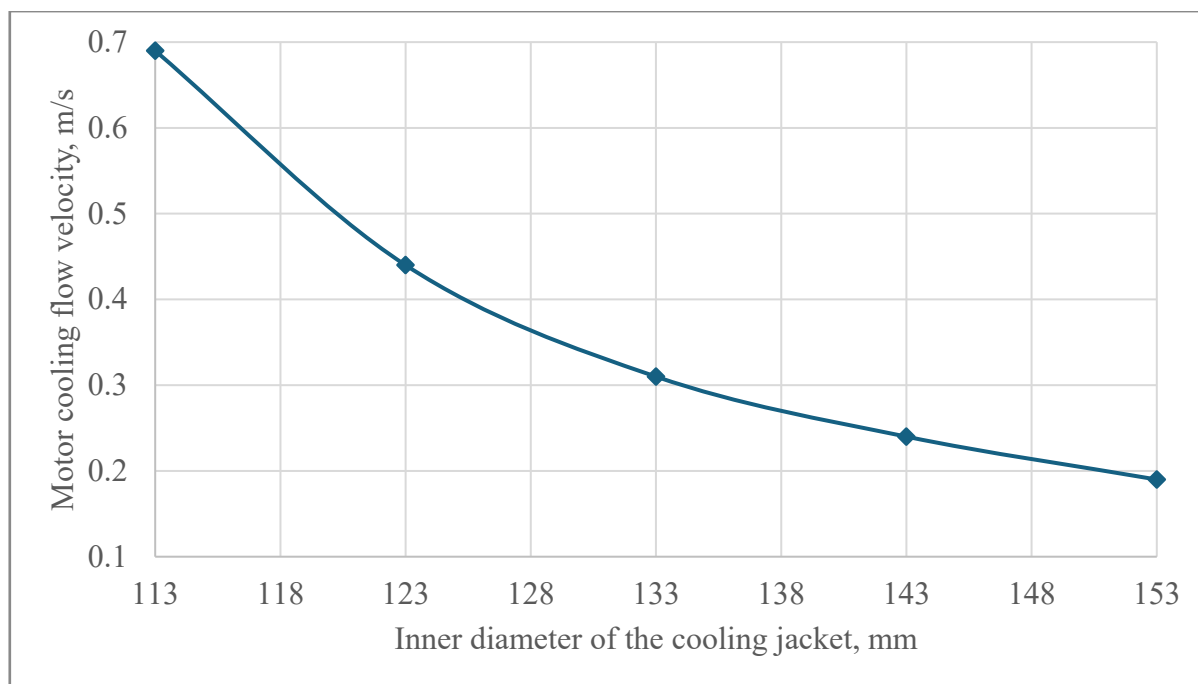


Figure 3. Theoretical dependence of the coolant flow velocity on the inner diameter of the jacket for the USK 408/42 pump ($D_{\text{well}} = 159$ mm, $D_{\text{motor}} = 93$ mm, $Q = 8$ m³/h).

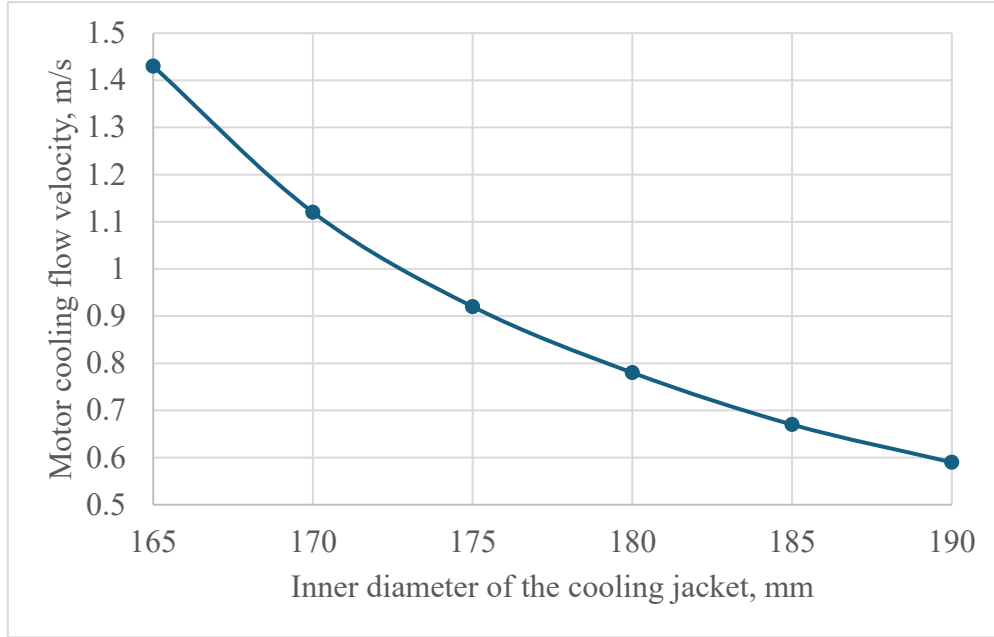


Figure 4. Theoretical dependence of the coolant flow velocity on the inner diameter of the jacket for the URN 6 25/14 pump ($D_{\text{well}} = 195$ mm, $D_{\text{motor}} = 145$ mm, $Q = 25$ m³/h).

For the USK 408/42 pump, the minimum required velocity of 0.2 m/s is ensured with jacket diameters up to 153 mm; however, the recommended target velocity of ≥ 0.5 m/s is achieved only with diameters of no more than ≈ 120 mm (113 mm among those considered). At $D_{\text{jacket}} = 113$ mm, the flow velocity is 0.69 m/s, more than three times the critical threshold. Each subsequent 10 mm increase in jacket diameter noticeably reduces the velocity, highlighting the high sensitivity of this parameter to the choice of jacket size. For the URN 6 25/14 pump, considerably higher velocities (0.59–1.43 m/s) are achieved over the entire range of jacket diameters studied, owing to the higher pump flow rate (25 m³/h) and the larger well diameter. It should be noted that for the URN 6 25/14 pump, the mean velocity in the well annulus is about 0.52 m/s even without a jacket; therefore, for this configuration the jacket is advisable primarily where the incoming fluid does not pass along the motor (for example, when the inflow into the well is located above the motor).

Of fundamental importance is the non-linear shape of the velocity curve in both cases (Figures 3 and 4). At small gaps, a slight change in jacket diameter leads to a large change in velocity, whereas at wide gaps the velocity is less sensitive to geometric variations. This behaviour is a direct consequence of the quadratic dependence of the annular cross-sectional area on diameter. In practice, this means that the choice of cooling jacket diameter must be accompanied by careful control of manufacturing tolerances, especially for small-diameter wells.

4.2. Results of the Motor Overheating Calculation

The motor overheating ΔT was calculated using Equation (10) as a function of jacket diameter for the USK 408/42 pump. Without a jacket, the mean flow velocity in the well annulus is about 0.17 m/s, i.e. below the critical value of 0.2 m/s. In all cases considered, the Reynolds number is $1.7 \cdot 10^4$ – $2.1 \cdot 10^4$ at 40 °C and $2.7 \cdot 10^4$ – $3.4 \cdot 10^4$ at 70 °C, i.e. the flow is turbulent and the use of Equation (6) is justified. The results for the base case ($T_{\text{liquid}} = 40$ °C) are presented in Table 2.

Table 2. Calculated overheating of the USK 408/42 pump motor for different cooling jacket diameters at a fluid temperature of 40 °C

Pump model	Q, m ³ /h	D _{well} , mm	D _{motor} , mm	ΔT without jacket, °C	D _{jacket} , mm	V, m/s	ΔT, °C
USK 408/42	8	159	93	9.4*	113	0.69	2.4
					123	0.44	3.8
					133	0.31	5.2
					143	0.24	6.7
					153	0.19	8.4

* Estimated using Equation (10) for the mean velocity in the well annulus without a jacket ($V \approx 0.17$ m/s), assuming that the entire flow passes uniformly over the motor; the actual overheating may be higher (see Section 4.3).

For a correct interpretation of the data in Table 2, the physical meaning of ΔT needs to be clarified. Within the adopted model, ΔT is the excess of the motor surface temperature over the temperature of the surrounding fluid: $T_{\text{motor}} = T_{\text{liquid}} + \Delta T$. At the base fluid temperature of 40 °C, the motor surface temperature is ≈ 42.4 °C at $D_{\text{jacket}} = 113$ mm, ≈ 48.4 °C at $D_{\text{jacket}} = 153$ mm and ≈ 49.4 °C without a jacket. In the most unfavourable case considered ($T_{\text{liquid}} = 70$ °C), the overheating decreases slightly owing to the lower viscosity of water and amounts to 2.0 °C at $D_{\text{jacket}} = 113$ mm, 6.8 °C at $D_{\text{jacket}} = 153$ mm and 7.6 °C without a jacket; the motor surface temperature is then 72.0–77.6 °C. Thus, with uniform flow over the motor, its surface temperature is determined primarily by the well fluid temperature, while the jacket reduces overheating by 5–7 °C; in all cases the temperature remains below the maximum operating temperature of the motors of about 120 °C. Consequently, the main value of the cooling jacket lies not so much in reducing overheating under uniform flow as in ensuring longitudinal flow along the motor in the first place: as the simulation results show (Section 4.3), without a jacket a substantial part of the flow bypasses the motor and a quasi-stagnant zone forms at its surface, for which the calculation by Equation (10) is not applicable and the actual overheating may be considerably higher. In addition, reducing overheating by several degrees increases the thermal margin of the insulation and slows its ageing [11, 12], while the higher flow velocity hinders the formation of deposits on the motor housing. The condition $V \geq 0.5$ m/s at $Q = 8$ m³/h and $D_{\text{motor}} = 93$ mm is satisfied for jacket inner diameters of no more than ≈ 119.7 mm (Equation (3)); a jacket of 123 mm ($V = 0.44$ m/s) no longer meets this condition. Therefore, the rational range of jacket inner diameters for the USK 408/42 pump in a 159 mm well is ≈ 113 –120 mm. A further reduction of the gap is limited by the increase in the hydraulic resistance of the annular channel, which was not evaluated in this work (see Section 4.5).

The dependence $\Delta T \propto V^{-0.8}$ is clearly evident in the calculation results. The transition from $V = 0.19$ m/s ($D_{\text{jacket}} = 153$ mm, $\Delta T = 8.4$ °C) to $V = 0.69$ m/s ($D_{\text{jacket}} = 113$ mm, $\Delta T = 2.4$ °C) corresponds to a 3.6-fold increase in velocity and a 3.5-fold reduction in overheating. Of this reduction, a factor of about 2.8 is due to the increase in velocity ($3.6^{0.8} \approx 2.8$), and the remainder is due to the decrease in the hydraulic diameter d_h from 60 to 20 mm (factor $(60/20)^{0.2} \approx 1.25$), which is consistent with Equation (10).

4.3. Simulation in SolidWorks

The pump assembly with and without the cooling jacket was designed in KOMPAS-3D, and the velocity distribution in the annular channel was calculated in SolidWorks Flow Simulation. Two

configurations were analysed: (I) $D_{\text{well}} = 159 \text{ mm}$, $D_{\text{motor}} = 93 \text{ mm}$; (II) $D_{\text{well}} = 195 \text{ mm}$, $D_{\text{motor}} = 145 \text{ mm}$.

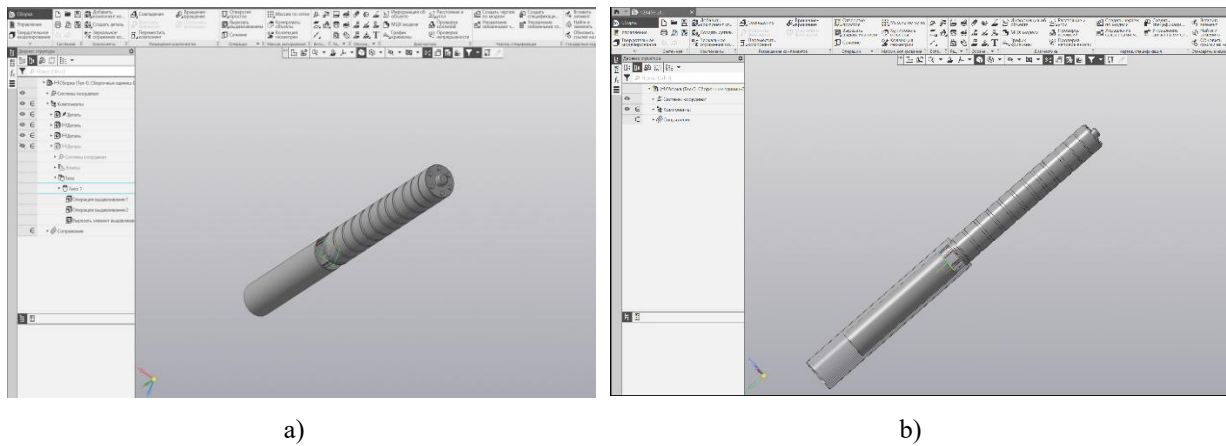


Figure 5. Three-dimensional model of the submersible pumping unit: a) pump without a cooling jacket; b) pump with a cooling jacket (KOMPAS-3D).

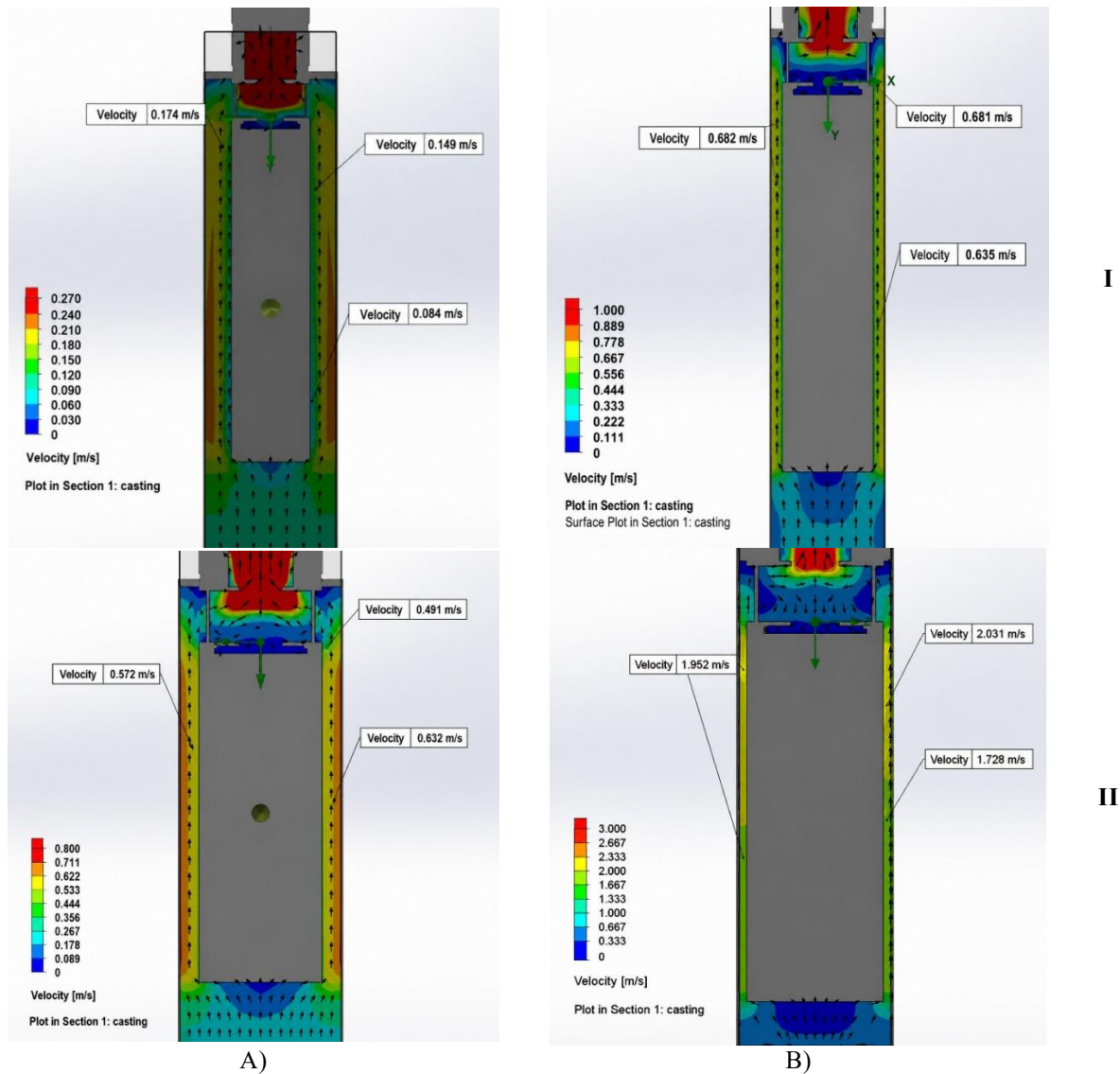


Figure 6. Results of calculations in SolidWorks Flow Simulation: A) submersible pump without a cooling jacket; B) submersible pump with a cooling jacket. I — $D_{\text{well}} = 159 \text{ mm}$, $D_{\text{motor}} = 93 \text{ mm}$; II — $D_{\text{well}} = 195 \text{ mm}$, $D_{\text{motor}} = 145 \text{ mm}$.

In configuration A (without a jacket), the velocity field shows nearly stagnant flow at the motor surface: most of the fluid bypasses the motor through the wide annular gap of the well without providing any significant cooling. In configuration B (with a jacket), the flow is directed into the narrow annular gap, forming a uniform high-velocity profile along the entire length of the motor. This result confirms that the cooling jacket works as intended: it forces the entire flow entering the pump suction to pass in direct contact with the motor surface. The simulation results for both well diameters are in qualitative agreement with the analytical predictions.

The model yields the following practically significant consequences. Temperature reduction: since $\Delta T \propto V^{-0.8}$, an increase in flow velocity V leads to a non-linear decrease in motor surface overheating. Cooling intensity: the heat transfer coefficient α grows with velocity, allowing the same power P to be removed at a smaller temperature difference. Effect of geometry: reducing the gap d_h at a constant flow rate Q increases the velocity V much faster than the coefficient decreases due to the scale factor $d_h^{0.2}$, which confirms the effectiveness of cooling jackets.

For a quantitative check of the analytical model, a conjugate heat transfer problem was additionally solved in SolidWorks Flow Simulation for the USK 408/42 pump ($D_{\text{well}} = 159 \text{ mm}$, $D_{\text{motor}} = 93 \text{ mm}$) without a jacket and with a jacket of $D_{\text{jacket}} = 113 \text{ mm}$. The input data correspond to the base case of Section 3.3: $T_{\text{liquid}} = 40 \text{ }^{\circ}\text{C}$; flow rate $Q = 8 \text{ m}^3/\text{h}$, supplied upward along the motor towards the pump suction; motor heat release $P = 2625 \text{ W}$, specified as a volumetric heat source over the entire motor volume. The motor was modelled as a homogeneous body made of AISI 321 stainless steel. The built-in k - ε turbulence model of Flow Simulation with wall functions was used. The computational mesh was generated automatically (initial mesh level 5) with a minimum gap size of 0.01 m and refined channel resolution; for the jacket case it contained about 2.76 million cells (1.93 million in the fluid and 0.83 million in the solid). The calculations were terminated upon convergence of the specified goals (242 and 389 iterations for the cases without and with the jacket, respectively). A preliminary calculation on a coarser mesh (minimum gap $\approx 0.06 \text{ m}$) gave a similar mean surface temperature for the case without a jacket (difference of $0.7 \text{ }^{\circ}\text{C}$) but a value $2.8 \text{ }^{\circ}\text{C}$ higher for the jacket case, indicating the need to resolve the narrow annular gap with the mesh; the results obtained on the refined mesh are reported below. The temperature was determined at three monitoring points near the motor housing surface, located along its length (Figure 7). The results are compared with the calculation by Equation (10) in Table 3.

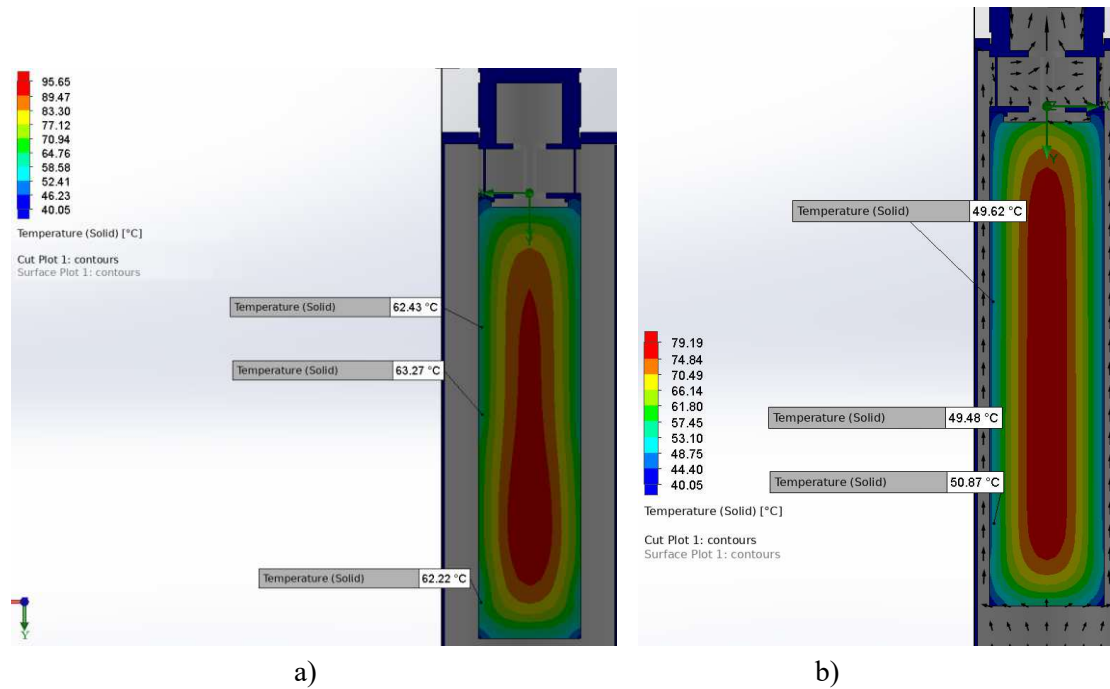


Figure 7. Temperature field in the USK 408/42 pump motor (SolidWorks Flow Simulation; $T_{\text{liquid}} = 40\text{ }^{\circ}\text{C}$, $Q = 8\text{ m}^3/\text{h}$, $P = 2625\text{ W}$): a) without a cooling jacket; b) with a cooling jacket of $D_{\text{jacket}} = 113\text{ mm}$. The labels show the solid temperature at the monitoring points.

Table 3. Comparison of CFD simulation results with calculation by Equation (10) for the USK 408/42 pump ($T_{\text{liquid}} = 40\text{ }^{\circ}\text{C}$, $P = 2625\text{ W}$)

Parameter	Without jacket	With jacket, $D_{\text{jacket}} = 113\text{ mm}$	Change
Temperature at monitoring points (CFD), $^{\circ}\text{C}$	62.4 / 63.3 / 62.2	49.6 / 49.5 / 50.9	—
Mean surface temperature (CFD), $^{\circ}\text{C}$	62.6	50.0	−12.6
Mean overheating ΔT (CFD), $^{\circ}\text{C}$	22.6	10.0	−12.6 (−56 %)
Temperature spread along the motor (CFD), $^{\circ}\text{C}$	1.1	1.4	+0.3
Maximum temperature in the motor (CFD), $^{\circ}\text{C}$	95.7	79.2	−16.5
Overheating ΔT by Equation (10), $^{\circ}\text{C}$	9.4	2.4	−7.0

As Table 3 shows, the cooling jacket reduces the mean motor surface temperature by $12.6\text{ }^{\circ}\text{C}$ (from 62.6 to $50.0\text{ }^{\circ}\text{C}$) and the maximum temperature inside the motor by $16.5\text{ }^{\circ}\text{C}$. Equation (10) predicts a reduction in overheating of $7.0\text{ }^{\circ}\text{C}$, i.e. it correctly reflects the direction and order of magnitude of the effect but underestimates it. The absolute overheating values from the CFD simulation are 2.4–4.2 times higher than the analytical ones. This discrepancy is explained by the fact that the analytical model accounts only for the convective resistance to heat transfer under uniform flow, whereas the CFD model accounts for heat conduction inside the motor and for non-uniform flow; moreover, the monitoring points are located in the solid near the housing surface, and the motor is represented as a homogeneous stainless-steel body with volumetric heat release. Thus, Equation (10) gives a conservative estimate of both the absolute motor temperature and the benefit of the jacket, and it is appropriate for comparative assessment of design options. The temperature distribution along the motor is close to uniform in both configurations (spread at the monitoring points of 1.1 – $1.4\text{ }^{\circ}\text{C}$). A $12.6\text{ }^{\circ}\text{C}$ reduction in surface temperature is significant for insulation life: according to the rule derived from the Arrhenius law, each 8 – $10\text{ }^{\circ}\text{C}$ reduction in operating temperature

approximately halves the rate of thermal ageing of insulation [11, 12], which corresponds to an approximate 2–3-fold increase in its service life.

4.4. Experimental Verification on a Laboratory Test Bench

To verify experimentally the trends obtained by calculation, tests were carried out on the upgraded laboratory test bench “Centrifugal Pumps” SGU-TsNS-012-6LR-PK (Figures 8 and 9). The base bench (maximum head 20 m, maximum flow rate 60 L/min, main tank volume 90 L) was additionally equipped with an auxiliary tank simulating a well, a submersible pump and a control panel, and the pumping unit was connected to a measuring circuit with a flow meter and a pressure sensor. Fluid from tank B1 enters tank B2, flows through the annular channel between the motor housing and the jacket along the motor to the pump intake strainer, and returns to tank B1 via the discharge line.

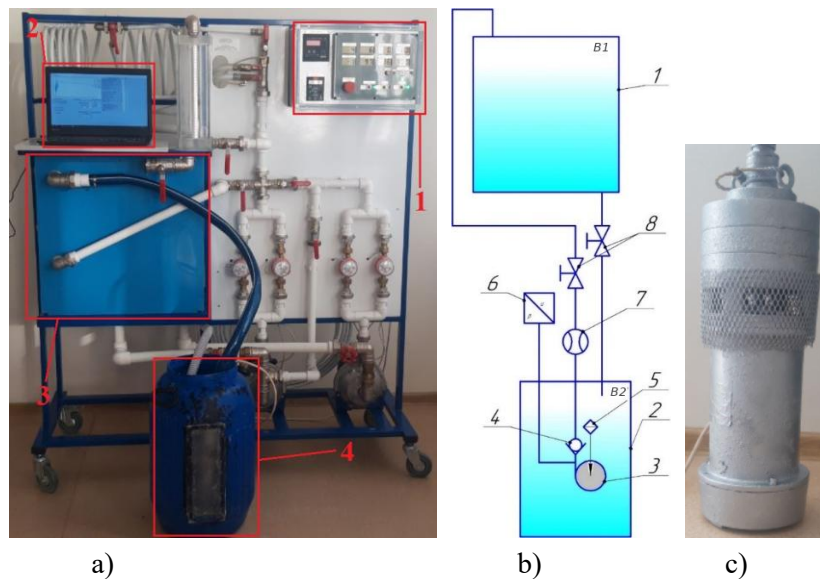


Figure 8. Experimental setup: a) upgraded test bench “Centrifugal Pumps” SGU-TsNS-012-6LR-PK (1 — control panel; 2 — PC; 3 — main tank; 4 — well simulator); b) schematic hydraulic diagram (1 — tank B1; 2 — tank B2 simulating the well; 3 — submersible pump; 4 — check valve; 5 — submersible pump strainer; 6 — pressure sensor; 7 — flow meter; 8 — gate valves); c) the tested Chimp QDX3-20-0.55(Y) submersible pump.



Figure 9. Experimental work on the upgraded laboratory test bench “Centrifugal Pumps” SGU-TsNS-012-6LR-PK.

The test object was a Chimp QDX3-20-0.55(Y) submersible pump (dimensions 210×380 mm, motor power 550 W, efficiency 0.65). Three cooling jackets 400 mm long with inner diameters of 220, 230 and 240 mm were used, corresponding to hydraulic diameters of the annular channel of $d_h = 0.01$, 0.02 and 0.03 m. The motor surface temperature was measured with a three-wire Pt100 resistance thermometer attached to the motor housing with a hose clamp and connected to an OWEN TRM meter. The coolant temperature was kept constant at $T_{\text{liquid}} = 30.0$ °C. The flow rate was controlled by a proportional valve and set to $Q = 2, 2.5$ and 3 m³/h; after each mode was set, a 10 min holding time was allowed for the readings to stabilise. The pump power consumption at $Q = 2.5$ and 3 m³/h was 452 and 547 W, respectively. The flow velocity in the channel was calculated using Equation (3), and the Reynolds number with $\nu = 0.801 \cdot 10^{-6}$ m²/s. Temperature readings were recorded with a resolution of 0.1 °C; the uncertainty of ΔT is estimated at ± 0.5 °C, excluding the systematic error associated with the sensor attachment method. The results are presented in Table 4 and Figure 10.

Table 4. Results of laboratory tests of the Chimp QDX3-20-0.55(Y) pump ($T_{\text{liquid}} = 30.0$ °C)

Q, m ³ /h	D _{jacket} , mm	d _h , m	V, m/s	Re	T _{surf} , °C	ΔT, °C
2	220	0.01	0.165	2050	33.6	3.6
2	230	0.02	0.080	2010	39.8	9.8
2	240	0.03	0.052	1960	44.2	14.2
2.5	220	0.01	0.206	2570	32.4	2.4
2.5	230	0.02	0.100	2510	37.8	7.8
2.5	240	0.03	0.065	2450	41.7	11.7
3	220	0.01	0.247	3080	30.5	0.5
3	230	0.02	0.121	3010	35.3	5.3
3	240	0.03	0.079	2940	40.8	10.8

The experiment confirms the main trend established by calculation: at a constant flow rate, reducing the jacket diameter increases the flow velocity in the channel and lowers the motor surface temperature, and for each jacket diameter an increase in flow rate also lowers it. The highest overheating ($\Delta T = 14.2$ °C) was obtained for the 240 mm jacket at $Q = 2$ m³/h, and the lowest ($\Delta T = 0.5$ °C, i.e. within the measurement uncertainty) for the 220 mm jacket at $Q = 3$ m³/h.

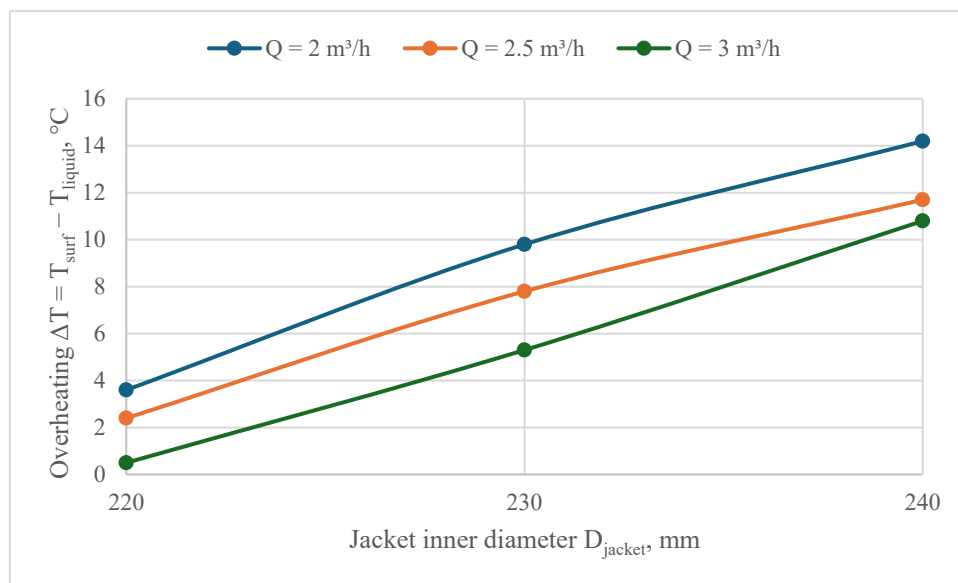


Figure 10. Measured overheating of the Chimp QDX3-20-0.55(Y) pump motor surface as a function of jacket inner diameter at different flow rates ($T_{\text{liquid}} = 30.0$ °C).

At the same time, under laboratory conditions the Reynolds number is $1.96 \cdot 10^3 - 3.08 \cdot 10^3$, i.e. the flow is laminar or transitional, whereas Equation (10) was derived for turbulent flow ($Re > 10^4$), which is typical of industrial wells (Section 4.2). Therefore, the experiment confirms the qualitative trend but does not constitute a quantitative verification of Equation (10) for the USK and URN pumps. For fully developed laminar flow, the Nusselt number is approximately constant and the heat transfer coefficient is inversely proportional to d_h [23], so that at constant heat flux the overheating should increase roughly in proportion to d_h . At $Q = 2 \text{ m}^3/\text{h}$, the measured ΔT values (3.6, 9.8 and 14.2°C at $d_h = 0.01, 0.02$ and 0.03 m) agree with this estimate in order of magnitude. The noticeable decrease in ΔT with increasing flow rate indicates the influence of the hydrodynamic and thermal entrance regions, and also that the clamped sensor partly senses the temperature of the surrounding fluid and may underestimate the surface overheating; this probably also explains the practically zero overheating at $Q = 3 \text{ m}^3/\text{h}$, $D_{\text{jacket}} = 220 \text{ mm}$.

4.5. Limitations of the Study

The results should be considered in light of the following limitations. First, the Dittus–Boelter equation was derived for fully developed turbulent flow in circular tubes; for annular channels and entrance regions it gives an approximate estimate with an error of up to $\pm 25\%$ [23]. Second, the model assumes a uniform flow distribution over the cross-section of the annular channel and constant thermophysical properties of the fluid; the effects of mineralisation, gas content and mechanical impurities of the well solution were not considered. Third, the thermal resistance of mineral deposits on the motor housing was not taken into account, although in real conditions it may substantially increase overheating. Fourth, the motor efficiency was taken at the lower limit of the rated range ($\eta = 0.65$), which overestimates heat release: at $\eta = 0.85$ the overheating would be about 2.3 times lower; the base fluid temperature of 40°C also needs to be refined by measurements. Fifth, the additional hydraulic resistance created by the jacket at small gaps and its effect on the pump head and efficiency were not evaluated. Sixth, Equation (10) underestimates the absolute overheating and the effect of the jacket compared with the CFD simulation (Section 4.3), so it should be used for comparative assessment of design options; the CFD model, in turn, represents the motor as a homogeneous stainless-steel body with volumetric heat release, without accounting for its internal structure. Seventh, the laboratory verification (Section 4.4) was performed on a smaller pump in the laminar and transitional regimes, without a no-jacket configuration and without repeated measurements; the temperature sensor attached with a clamp may underestimate the surface overheating. Finally, the effect of the jacket on the energy consumption of the pumping unit was not quantified. Further experimental studies are planned to include measurements in the turbulent regime, a no-jacket configuration, repeated measurements and direct measurement of active power.

Techno-Economic Assessment

5.1. Capital Costs

The capital costs associated with the purchase, delivery and installation of a cooling jacket for one USK 408/42 pumping unit are presented in Table 5.

Table 5. Capital costs per USK 408/42 submersible pump

№	Cost item	Cost, USD
1	Cooling jacket	29.2
2	Installation (17%)	5.0

№	Cost item	Cost, USD
3	Delivery to Navoi (10%)	2.9
Total:		37.1

The total capital investment is about USD 37 per unit (UZS 445 thousand at the Central Bank of Uzbekistan exchange rate of $\approx 12,000$ UZS/USD, 2026). This amount is negligible compared with the cost of rewinding the motor or replacing the pump.

5.2. Qualitative Assessment of the Economic Effect

A quantitative estimate of electricity savings is not provided in this work. According to the results of Section 4, the jacket reduces the motor surface temperature by 7 °C according to the analytical model and by 12.6 °C according to the CFD simulation; the associated reduction in winding resistance and electrical losses is small, and its reliable estimation requires direct measurements of power consumption. Such verification is a subject of further experimental research (Section 4.5).

It should be emphasised that well fluid with a high content of iron, manganese and lime promotes the formation of thermally insulating deposits on the motor surface, since the fluid temperature near the motor surface is higher than in the suction zone [5, 13]. When the flow is too slow, this layer of oxidised minerals and metals creates overheated areas on the winding insulation, reducing its durability and causing the permissible current to be exceeded. By creating a directed higher-velocity flow along the motor, the cooling jacket intensifies heat removal; however, the thermal resistance of deposits already formed is not accounted for in the present model.

The main economic effect of the jacket is associated with improved reliability of the pumping unit: a lower motor failure rate, reduced repair and downtime costs, and longer intervals between overhauls. Since ISL well pumps operate almost continuously and usually in remote areas where unscheduled maintenance is particularly costly, preventing even one unscheduled motor repair far outweighs the cost of the jacket. A quantitative assessment of this effect requires failure statistics for the pumping equipment and is a subject of further research.

Conclusion

This work addressed the problem of electric motor overheating in submersible pumping units used in ISL technology in the Kyzylkum region. The technical feasibility of the cooling jacket as a means of solving this problem has been substantiated. The main results are as follows.

A mathematical model of convective heat transfer in the annular cooling channel has been developed on the basis of forced convection theory (Equations 2–10). The model establishes an analytically transparent relationship between the fluid velocity, controlled by the inner diameter of the jacket, and motor overheating: $\Delta T \propto V^{-0.8}$. This dependence provides a physically sound criterion for selecting the jacket diameter at the engineering assessment stage; its accuracy is limited by the model assumptions (Section 4.5).

Parametric calculations for the USK 408/42 pump ($Q = 8$ m³/h, $D_{\text{motor}} = 93$ mm, $D_{\text{well}} = 159$ mm) show that the rational inner diameter of the cooling jacket is ≈ 113 –120 mm: at larger diameters the flow velocity falls below 0.5 m/s. At a fluid temperature of 40 °C and uniform flow, the calculated overheating of the motor surface relative to the fluid decreases from 9.4 °C without a jacket to 2.4 °C at $D_{\text{jacket}} = 113$ mm, and at 70 °C from 7.6 to 2.0 °C. Since under uniform flow the motor temperature remains within permissible limits even without a jacket, the main function of the jacket is to guarantee longitudinal flow along the motor, which according to the simulation may not be ensured without a

jacket, as well as to increase the thermal margin of the insulation and to prevent the formation of deposits.

The SolidWorks Flow Simulation results confirmed the transition from quasi-stagnant flow at the motor surface (configuration without a jacket) to a uniform high-velocity profile along its surface (configuration with a jacket). The conjugate thermal calculation on the refined mesh showed that a jacket with $D_{\text{jacket}} = 113$ mm reduces the mean motor surface temperature by 12.6 °C and the maximum temperature inside the motor by 16.5 °C, whereas Equation (10) predicts a reduction in overheating of 7.0 °C. Equation (10) correctly describes the direction and order of magnitude of the jacket effect but gives a conservative estimate of both the absolute overheating and the benefit of the jacket.

Laboratory tests of the Chimp QDX3-20-0.55(Y) pump with jackets of 220–240 mm diameter at flow rates of 2–3 m³/h confirmed the qualitative trend: the motor surface overheating decreases with decreasing jacket diameter and increasing flow rate (from 14.2 to 0.5 °C). Since the flow under laboratory conditions is laminar or transitional ($Re \approx 2 \cdot 10^3 - 3 \cdot 10^3$), these results do not constitute a quantitative verification of Equation (10) for industrial conditions.

The capital cost of manufacturing, delivering and installing the cooling jacket is about USD 37 per pump, which is negligible compared with the cost of rewinding the motor or replacing the pump. The main economic effect is associated with improved reliability and service life of the motor; the effect of the jacket on energy consumption requires experimental verification.

The developed model and design recommendations are directly applicable to both pump models studied and can be extended to other configurations of submersible pumping units by substituting the corresponding geometric and hydraulic parameters into the governing equations. Further research should focus on experimental verification of the predicted temperature reductions under real field conditions, as well as on evaluating the long-term effect of the cooling jacket on the dynamics of mineral deposit formation.

Conflict of interest. On behalf of all authors, the corresponding author states that there is no conflict of interest.

Data availability. All data used in this study shall be made available upon request.

Author contributions: Atakulov L.N.: conceptualisation, methodology, supervision. Kurbonov O.M.: software, visualisation, computational and experimental studies. Zokhidov O.U.: validation, data curation.

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