Theoretical essentials for designing viscous liquid pumping systems in the conditions of the Arctic region

Konstantin Kim¹*, Sergey Ivanov², and Marat Khismatulin¹

¹Emperor Alexander I St. Petersburg State Transport University, Moskovsky Ave., 9, 190031, St. Petersburg Russia
²Komsomolsk-na-Amure State University, Lenin Ave., 27, building 1, Komsomolsk-na-Amure, 681013, Russia

Abstract. The purpose is the creation of a scientifically based solution for the design of hermetic electromechanical converters adapted to existing pipelines, as well as those under construction and development. These pipelines meet the energy efficiency requirements and operate in the conditions of the Arctic region. Methods: methods of mathematical analysis, numerical methods, and methods of mathematical programming were used as the main mathematical means. The CFD design package of Ansys Maxwell was used. Research results: we showed that an increase of the efficiency of the system of technological transportation of petroleum products at low temperatures is possible through the use of electromechanical devices that combine the functions of heating and pumping the working liquid. We solved the field problem based on the analytical calculations carried out by a computer. This allowed us to determine the distribution of eddy current density in the structural elements of these devices responsible for creating pressure and heating the pumped petroleum products. The importance of the electrical parameters of the secondary windings has been confirmed numerically and experimentally. The hydraulic characteristics, mechanical losses, volumetric losses and efficiency are determined. Taking into account the outcomes of theoretical studies the experimental sample of the sealed electromechanical converter with the specified physical and chemical characteristics was created at Emperor Alexander I St. Petersburg State Transport University. Practical relevance: the obtained analytical formulas can be used at the development of the systems for pumping viscous liquids at low temperatures using the sealed electromechanical converters which helps to reduce the installed power of energy equipment without degrading the values of technological indicators.

1 Introduction

The future of Russia is determined by the development of its territories of the northern latitudes which account for most of Russia. The consideration of the Arctic region as a
strategic trend for the long-term economic development of the northern territories involves the comprehensive development of transport and energy infrastructure. Its implementation requires a reliable power supply system focused on among other things, to local primary resources, primarily hydrocarbons, and products of their processing. This is due to its remoteness from centralized sources. The creation of the network of autonomous sources of electricity connected with the places of production, transportation, processing, storage and shipment is associated with the use of the technical reservoirs that ensure the uninterrupted operation of the main and backup electrical installations operating on liquid and gaseous fuels. At the same time, the topology of these objects is determined by the need for their maximum get closer to the corresponding technological objects (the enterprises of the main production, transport systems, draining and filling overpass operated at low temperatures).

Reducing the influence of the climatic factors and solving issues of energy saving at the transportation of petroleum products are especially important when the power of electricity sources is limited. This problem was previously considered by the authors [1]. The main normative document regulating the main stages of this process is the norms of technological design of enterprises for the provision of petroleum products. It sets the main design parameters (the kinematic viscosity, temperature, heating and draining time, etc.) during the cold period, the duration of which is determined by the rules for the transport of goods from October 15 to April 15.

An obvious way to increase the efficiency of transporting high-viscosity hydrocarbons is to heat them at intermediate stations or along the whole pipeline. That’s why we name such pipelines as “hot”. In the most common version, the pipeline contains two separate units (the heater and the pump unit).

Increasing the efficiency of the “hot” pipeline is traditionally provided by the following measures:
- improving the rheological properties of the pumped product by increasing its temperature;
- parallelization of the main pipeline (looping);
- optimization of the thermal insulation characteristics;
- increasing the number of thermal and/or pumping units.

The main task considered in this paper is to increase the efficiency of the system for technological transportation of petroleum products at low temperatures through the use of devices that combine the functions of heating and pumping the working liquid.

2 Theoretical basis

The additional task arises to determine the temperatures of the petroleum products at their pumping when we design the systems for transporting petroleum products operated in the conditions of northern territories. The optimization of the pipeline parameters requires setting the initial and boundary conditions (air and ground temperatures, throughput, characteristics of transported petroleum, etc.).

A similar problem arises during the operation of existing “hot” pipelines but the part of the conditions changes (ambient temperature, volume, petroleum characteristics) in this case. This leads to the difference between the calculated (design) heating temperature and the operating temperature.

The optimal temperature is determined by the fact that with an increase of the heating temperature, viscosity decreases, pressure losses in the pipeline and transportation costs but at the same time, the cost of heating petroleum increases. The objective function corresponds to the minimum sum of costs for heating and transporting petroleum. The additional restrictions are imposed by the list of petroleum products approved for the transportation by the filling in the tank cars, sea and river ships and by road, as well as the preparation of
vehicles for loading and transportation. For example, the initial filling of the tank with flammable and combustible petroleum products belonging to hazard classes 1 and 2 is carried out at a speed of no more than 1 m / sec.

The rheological characteristics of liquids in dynamics are described by the viscosity coefficient, speed or its gradient in the designed section and dimensional ratios of the pipeline.

The effect of temperature on the coefficient of dynamic viscosity $\mu$ for Newtonian liquids (light, low-viscosity petroleum products with a low content of paraffin) is different for non-Newtonian waxy and highly tarry ones. At high temperatures there is the presence of paraffin in the dissolved state and at low temperatures there are the structured paraffin crystals in the free or bound state. In addition, the properties of paraffinic petroleum products also depend on the mechanical influences (mixing). To ensure the suboptimal characteristics during transportation, we must both heat liquid and mix it.

There are no analytical dependencies determining the optimal temperatures and pumping rates for the given pipeline geometry due to a large number of non-deterministic factors. So the values of the determining parameters are determined experimentally.

In practice, there are several empirical equations to determine the temperature-viscosity dependence of the transported liquid with acceptable accuracy. The influence of temperature is taken into account by introducing the functions of viscosity, density, heat capacity and thermal conductivity on temperature into the calculation equations.

For example, Walther formula (ASTM) establishes the relationship between viscosity and temperature:

$$\lg \lg (\nu + 0.8) = a + b \cdot \lg \Theta,$$

where $a = \lg \lg (\nu_1 + 0.8) - b \cdot \lg \Theta_1$; $b = \lg [\lg (\nu_1 + 0.8)/\lg (\nu_2 + 0.8)] \lg (\Theta_1/\Theta_2)$.

For the transition from the kinematic to the dynamic viscosity the density $\rho$ is calculated using the formula:

$$\rho = \rho_{293} - (1.825 - 0.001315 \cdot \rho_{293}) (\Theta - 293).$$

In the temperature range of 273...470 K the specific heat capacity $C_p$ is calculated using Krego formula:

$$C_p = 31.569(762 + 3.39 \Theta)/\sqrt{\rho_{293}}.$$  

The thermal conductivity coefficient $\lambda$ is approximately estimated using Crego-Smith formula:

$$\lambda = 156.6(1 - 0.00047\Theta)/\rho_{293}.$$  

The determining rheological parameter of the liquid is the viscosity. The density, heat capacity and coefficient of thermal conductivity can be set constant in preliminary calculations. Taking into account the viscosity-temperature dependence on numerous losses we can evaluate the efficiency of heating the liquid during its transportation.

The hydraulic losses and pressure reduction in the straight sections $\Delta P_i$ and $\Delta h_i$ can be calculated using the formulas:
$\Delta P_{tr} = k_{hydr} \frac{\rho l}{2d} W_m^2$, $\Delta h_{tr} = k_{hydr} \frac{1}{2dg} W_m^2$,

where $k_{hydr}$ is the coefficient of the hydraulic resistance determined for the laminar mode ($Re_{\text{eq}} \leq 1000$) according to Stokes formula: $k_{hydr} = \frac{64}{Re_{\text{eq}}}$; for the turbulent mode ($Re_{\text{eq}} \geq 2200$) according to Blasius formula: $k_{hydr} = \frac{0.3164}{Re_{\text{eq}}^{0.25}}$, where $l$, $d$ are the length and diameter of the straight section of the channel; $w_m$ is the average speed of liquid movement.

These equations use the effective Reynolds number

$Re_{\text{eq}} = \frac{Re}{\left(1 + \frac{\tau_0 D}{\rho w_m v}ight)^{\frac{1}{2}}}$,

where $\tau_0$ is the initial yield strength; $A$ is the dimensionless empirical coefficient taking into account the properties of the pumped liquid.

The pressure and head losses ($\Delta P_m$, $\Delta h_m$) in the short sections with complex geometry are calculated quite accurately using Weisbach formula:

$\Delta P_m = \xi \frac{\eta}{2v} W_m^2$; $\Delta h_m = \xi \frac{w_m^2}{2g}$,

where $\xi$ is the coefficient of local hydraulic resistance depending on the flow regime, type and design of the channel; $\eta$, $v$ are the respectively the dynamic and kinematic viscosities.

The average logarithmic temperature is taken as the calculated one:

$\Theta_m = \Theta_0 + (\Theta_1 - \Theta_2)/\ln ((\Theta_1 - \Theta_0)/(\Theta_2 - \Theta_0))$.

The analysis of the above equations shows that the increase of the temperature and the decrease of the dynamic viscosity of the transported liquid make it possible to make the pumping process energy efficient.

### 3 Existing heating systems

The heating of petroleum and petroleum products and the reduction of petroleum viscosity to a technologically feasible level are carried out by the various devices. We use the serpentine or sectional tubular heaters to preheat the stationary tanks. At the same time the heating temperature is lower than the calculated one. This is connected with the large convective losses of the bounding surfaces into the environment. The main type of coolant is water vapor. To reduce the heat losses, tanks can be equipped with thermal insulation. To heat the pumping liquid to the design temperature we use additionally the steam or fire heaters with the multi-pass heat exchangers. The transported liquid passes through the pipeline; the steam is supplied into the inter-tube space.

The petroleum heating plant is a complex technical device including the heat exchange chamber, the fan assembly for forced air supply, input and output pipes and other equipment (shutoff and control valves, instrumentation and protection equipment, piping). The fan assembly includes an electric motor loaded through a V-belt transmission to a high-pressure centrifugal fan.

The use of existing heating plants not only reduces the fire safety and influences the environment but also limits the making of the pipeline. Partially, these shortcomings can be
eliminated by installing the smaller diameter pipelines with hot water parallel to the main line.

The high-frequency electrical heating based on the skin effect is more economical. The source of thermal power is a high-frequency current running through the copper conductor with the cross-section of 8...60 mm². This conductor locates inside the pipe with the diameter of 6...40 mm, welded to the main pipeline.

The use of electric heating (heating pads, heating cables, etc.) during the technological operations provides for the thermal control of the heating system by the recommendations for the integrated electric heating of viscous petroleum products at the petroleum depots. The heating temperature of the viscous petroleum products should not exceed 90°C, the heating temperature in a closed crucible should be at least 25°C lower than the flash point of petroleum product vapors.

The electric heating devices must be explosion-proof and automatically turn off when the maximum allowable temperatures or levels are reached.

4 Materials and methods

An analysis of the existing designs and requirements for the plants of heating viscous liquids shows that one of the promising technical solutions in this area is the use of the pumping devices based on the heat-generating electromechanical converters [2-5].

Traditionally, the industrial pumping units, whose power reaches some tens of megawatts, are energy-consuming equipment. Their efficiency does not exceed 60%. One of the reasons is the aggregate layout, here the pump and the electric motor are connected by a belt drive, a clutch, and a gearbox and the losses in these elements are dissipated into the environment.

Increasing the energy efficiency of the transportation process is achieved by combining the motor and pump into one device excluding intermediate gears and using all heat generations to heat the transported liquid. Compliance with technical requirements ensures the hermetic design of the installation.

One of the possible designs of the sealed electromechanical converter combining the functions of the pump and the heater is shown in Fig. 1.

The main elements of the device are an outer casing (1), the magnetic core with the mains winding (2), the working gap (3), the fixed short-circuited winding (4), the rotating short-circuited winding (5), the pressure blades (6), the axial channels (7), the input pipe (8), the output pipe (9), the plain bearing (10), the outer cylinder (11), the inner cylinder (12), the guide cover (13), the spacer sleeve (14).

The operation principle of the converter is described in [7]. The thermal power used to heat the transported liquid consists of 1) the electrical losses released in the primary, fixed short-circuited and rotating secondary windings, 2) the magnetic losses, 3) the mechanical losses, 4) the hydraulic losses, 5) the additional losses.

To define the electrical losses in the fixed short-circuited winding which is the main source of liquid heating, we solved the field problem of finding the current density distribution in Ansys Maxwell package. The results are shown in Fig. 2.
When modeling the active resistance of the secondary winding is given in the relative units, the rated value corresponds to the phase resistance of the stator winding. The inductance of the secondary winding is not taken into account.

The simulation results show that the analytical equations for calculating the temperature and pressure of the pumped liquid can be obtained under several assumptions:

- only the main spatial harmonic of the magnetizing force is taken into account;
- the field strength is taken into account only by the normal component;
- the size of the gap is significantly less than the outer diameter of the fixed short-circuited winding;
- the current density is constant across the thickness of the winding;
- there is no magnetic resistance of the ferromagnetic sections.

Maxwell's equations in the current and without current areas are solved using the vector potential. The field characteristics (vector potential and eddy current density) are found from the solution of Laplace's equation for the without current area and Poisson's equation for the current area [8-10].

The real part of Poynting vector determines the thermal (electrical) power of the fixed short-circuited winding $P_{el,n}$:

$$P_{el,n} \approx 4pB^2\tau^3f^2l_{fw}y_{fw}\Delta_{fw} \left(1 - 1.273 \frac{\tau}{l_{fw}\left(c\text{th }0.785l_{fw}/\tau\right)}\right),$$

where $p$ is the number of pairs of poles; $B$ is the magnetic induction; $\tau$ is the pole division; $f$ is the frequency of the supply network; $l_{fw}$ is the active length of the fixed short-circuited winding; $y_{fw}$ is the specific electrical conductivity; $\Delta_{fw}$ is the thickness.

The thermal power of the rotating winding $P_{el,v}$ depends on the slip:

$$P_{el,v} \approx 4pB^2\tau^3s^2f^2l_{rw}y_{rw}\Delta_{rw} \left(1 - 1.273 \frac{\tau}{l_{rw}\left(c\text{th }0.785l_{rw}/\tau\right)}\right),$$

where $l_{rw}$ is the active length of the rotating winding; $y_{rw}$ is the specific electrical conductivity; $\Delta_{rw}$ is the thickness; $s$ is the slip.

From the above equations you can get a formula for calculating the liquid temperature $\Theta_g$: 
where $l_{lob}$ is the length of the end parts of the winding; $\Delta \zeta$ is the reference coefficient depending on the ratio of the length and diameter of the pipeline; $r_{fw}$ is the fixed short-circuited winding radius; $r_{rw}$ is the radius of the rotating winding.

5 Outcomes and discussion

A preliminary calculation shows that the heating power can be changed in the range of 0.15...0.85 of the input power, for example, by varying the material of the secondary windings, their dimensions, frequency and value of the supply voltage. In this case, the power used for pumping liquid will be from 0.85 to 0.15 of the power consumed by the plant. The significance of the electrical characteristics of the secondary windings is confirmed both by the outcomes of calculations and experimentally. At the constant value of magnetic induction, the losses for the windings of equal thickness, made of stainless steel and iron (their specific electrical resistance varies from $1.15 \cdot 10^6$ Ohm·m to $0.98 \cdot 10^6$ Ohm·m), differ by almost an order of magnitude. The value of the heat flow for low-magnetic stainless steel can be 6...8% and for ferromagnetic steel this value can be 70...80% of the input power. The quantitative ratio between thermal and mechanical powers determines the predominant nature of the plant mode: heating, pumping or combined.

When designing, along with the thermal characteristics, it becomes necessary to find the hydraulic ones (pressure, productivity).

A necessary condition for the possibility of performing the hydraulic calculation based on an ideal movement model is obtaining the required intensity of liquid movement. This condition is ensured by the design of the plant, namely, the presence of the bladed actuating element with adjustable pressure. The model assumes a piston nature of the liquid flow, similar to the tubular heat exchangers with turbulent flow and a uniform speed profile. The use of such a model makes it possible to separate two conditionally independent processes: thermal (heating) and hydraulic (movement) and establish the energy relationships between them. Obviously, the hydraulic process is the main one, since it determines not only the productivity but also the intensity of heat removal from the heat power sources.

It should be noted that finding an accurate description of the processes of movement and heating of the transported liquid is extremely difficult and requires a special study of the laws of liquid flow using the modern CFD technologies in the field of hydro-gasdynamics, for example, CFX, Fluent, Flotran and corresponding hardware resources.

At the simplified calculations, the liquid flow can be theoretically characterized by the volumetric flow rate and the speed averaged over the pipeline section. During the movement of real liquids, the part of the mechanical energy of movement is converted into the thermal energy. It represents the pressure losses, head losses and energy losses due to the molecular and turbulent viscosity of the liquid.

The ideal movement model assumes that the circulation of the liquid flow leads to two processes with the different rates: the heat removal from the active surface of the secondary windings and the liquid displacement in the axial direction.

We can find the hydraulic characteristics, using a single-stage (single-flow) model, after selecting the defining parameters. The accurate determination of the most significant factors is a separate problem even using the methods of experiment planning and statistical analysis. At the first stage of the calculation the simple geometric model was adopted. It consists of the unlimited number of the straight blades of unit thickness and the end rings connecting
them. The number of blades, their dimensional ratios, profile, the input angles / the output angles of the liquid flow and other parameters are approximately taken into account by the hydraulic, volumetric, and mechanical coefficients used in the design of vane machines.

Euler formula was adopted as the initial calculation equation. It relates the hydrodynamic characteristics with the help the difference of the scalar products of the velocities $u$ and $v$.

This formula, written for the adopted system of blades rotating about the longitudinal axis with a frequency $n$, looks like this:

$$gH = (u_2, v_2) - (u_1, v_1),$$

where $H$ is the pressure; $u_1$ is the tangential component of the liquid speed at the input; $v_1$ is the absolute speed of the coolant at the input; $u_2$ is the tangential component of the liquid speed at the output; $v_2$ is the absolute speed of the coolant at the output.

In the accepted design model the left side of the equation represents the specific energy transmitted by the actuating element without taking into account heating. For an incompressible liquid and an infinitely large number of blades in it, it is enough to simply take into account the profile of the blade:

$$2gH = u_2^2 - u_1^2 - v_2^2 + v_1^2,$$

where $\gamma_1, \gamma_2$ are the cutting angles of the lower and upper edges of the blade, respectively; $v_1, v_2$ are the radial components of the absolute speed of the pumped liquid at the input and output; $\beta_1, \beta_2$ are the angles of attack of the blade at the input and output.

The values of the input and output radial and tangential components are determined by the dimensional ratios and the speed of rotation of the blade:

$$u_1 = \pi D_1 n; \quad u_2 = \pi D_2 n; \quad v_1 = Q/(\pi D_1 b_1); \quad v_2 = Q/(\pi D_2 b_2),$$

where $Q$ is the performance (flow rate); $D_1, D_2$ are the theoretical diameters formed by the inner and outer edges of the blades, respectively; $b_1, b_2$ are the widths of the blade at the input and output of the flow.

Substituting the speed values into the equation of specific energy allows to obtain an equation relating the pressure, head, transport speed and dimensional ratios of the blade:

$$H = (\pi D_2 n g^{0.5} \sin \gamma_2)^2 - (\pi D_1 n g^{0.5} \sin \gamma_1)^2 +$$

$$+ Q n g b_2^{-1} \sin^2 \gamma_2 \cdot \cot \beta_2 - Q n g b_1^{-1} \sin^2 \gamma_1 \cdot \cot \beta_1.$$

The last equation needs to be adjusted to take into account the actual number of the pressure blades $z$. This number, depended on the scope of existing structures, can range from 3 to 72.

The finite number of blades can be taken into account using the coefficient $k_H$:

$$k_H = 1 - u_2^2 \frac{\pi}{zH} (1 - \frac{D_1^3}{D_2^3}) \sin \beta_2.$$

It shows that the decrease of pressure compared to the ideal model ($z \to \infty$) discussed earlier is $10...30\%$.

We take into account the mechanical losses using the speed factor $n_s$: 

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\[ n_s = 3.65n_s \frac{Q^2 N^3}{H^3}, \]

where \( n \) is the number of stages of the blade system (a single-stage model is adopted above).

The efficiency factor taking into account mechanical losses is calculated by the formula:
\[ \eta_m = 1/(1 + 820n_s^2) \]
and has values of the order of 0.92...0.96.

The volume losses are found through the volumetric compression coefficient:
\[ n_{ob} = 1/(1 + 0.68n_s^{-0.66}). \]

The value of this coefficient is 0.85...0.95.

In general form the formula for the power without taking into account the hydraulic losses on the direct and local resistances of the pipeline is written as:
\[ P_m = \frac{pgH_dQ_d}{kHn_mn_{ob}}, \] (1)

where \( H_d \) and \( Q_d \) are the updated values of pressure and flow rate taking into account the main design parameters.

The substitution of equations \( k_H, n_m, n_{ob} \) in (1) allows obtaining the formula for the mechanical power required for the liquid transportation.

The power \( P_{tem} \), necessary to heat the liquid to the temperature \( \Theta_g \) during transportation, is:
\[ P_{tem} \approx 0.076(\Theta_g \lambda_g \nu_g)^{1.25} d^2 Q_g^0.54 \left( \frac{q_g}{q_0} \right)^{0.31} \xi_1^{1.25}, \]
\[ q = \nu \rho c_p / \lambda, \] (2)

where \( \lambda, \nu, \rho, c_p \) are the thermal conductivity, kinematic viscosity, density, specific heat of the liquid, reduced to the temperatures of the liquid (subscript “\( g \)” or initial (environment) (subscript “\( 0 \)”); \( \xi_1 \) is the reference coefficient depended on the ratio of the length \( l \) and diameter \( d \) of the pipeline.

The system of the equations (1) and (2) determines the coefficient of thermal-mechanic use of the drive. Their use allows to solve the inverse problem (ensuring the required energy ratios, for example, by choosing the dimensional ratios of the blades using the approximating models of the axial component of the absolute speed in the pipeline).

In conclusion, it is necessary to evaluate the screening effect of the secondary windings, which is not taken into account when deriving equations (1) and (2). We defined earlier the relative values of the resistances. The influence of these resistances on the starting, maximum and rated torques during design can be neglected.

On the other hand, at the small relative resistances of the secondary windings (first of all, fixed short-circuited winding), the influence of eddy currents circulating in them can lead to the deterioration of the mechanical characteristics.

To assess this effect we can proceed from the fact that at the nominal speed of the electric motor, the frequency of eddy currents in the rotating heating element does not exceed 5...10 Hz. Therefore, first of all, it is required to take into account the electromagnetic flux connected with the main harmonic of the linear current load. This flux passes through the
non-magnetic fixed short-circuited winding. The approximate calculation, taking into account the accepted geometry and material (the screen thickness is equal to $0.2 \times 10^{-3}$ m, the electrical conductivity is equal to $5.59 \times 10^7$ Sm/m, magnetic permeability is equal to $4\pi \times 10^{-7}$ H/m), showed that the value of the screening coefficient is close to one (0.98). The fixed short-circuited winding has no significant shielding effect.

6 Conclusions

The theoretical evaluation of the efficiency of the use of devices combining the functions of a pump and a heater in systems of transporting viscous liquids at low temperatures shows that their use makes it possible to reduce the installed power of energy equipment without deteriorating technical indicators. The given analytical ratios allow designing the sealed electromechanical converters adapted to existing and under-construct pipelines that meet energy efficiency requirements.

The obvious complexity of the considered research task associated with the non-uniformity of the flow speed profile, taking into account discontinuous flows, stagnant zones and other factors, suggests further research, for example, based on volume-element or diffusion models.

References