PROBLEMS OF DESIGNING VANE PUMPS FOR PETROLEUM PRODUCTS

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Rotor vane pumps are widely used in positive-displacement hydraulic drives to create a high fluid pressure (up to 16 MPa) at low feed rates (up to 12 m³/h); they are well-described in the technical literature. At the same time, single-acting vane pumps for comparatively rather large feed rates at low pressures, which are finding increasing use abroad for pumping petroleum products, have not been completely studied up to now.

Analysis of the design and characteristics of foreign-built vane pumps has shown that these pumps are highly efficient self-priming pumps for a wide range of petroleum products. They are especially useful for replacing vortex, centrifugal-vortex, and centrifugal low-speed pumps, which are mounted on mobile units for pumping fuel and on trucks for delivering and transporting fuel (see Table 1). The economy of making such changes comes from reducing energy requirements, because the efficiency of the vane pump is 2-2.5 higher than for other comparable pumps.

However, up to now, our country has not developed vane pumps for petroleum products as it should have. For decades, we have produced only one brand of vane pump: the AZT-5 (feed rate of 3.5 m³/h) for service-station gas pumps, and these are not up to date.

When a basic manufacturer of these pumps— the Avtozapravochnaya Tekhnika Company (in Serpukhov)— tried designing or perfecting an efficient vane pump based on a commercial pump, without doing the required theoretical development and the critical analysis on the use of the proper materials, it failed.

The development of vane pumps in Russia is hindered primarily by insufficient scientific knowledge of the details of the hydrodynamic processes that occur in these pumps, and also by the absence of scientifically based methods for calculating their designs. As a result, pumps can only be developed based on using specific design curves and conclusions, which means the designs require a long experimental development. However, because a multitude of operational and design parameters affect pump operation, this approach does not always give satisfactory results. With no analytical models, it is impossible to evaluate and optimize a new pump design beforehand.

The experience of many foreign firms in designing vane pumps is reflected only in advertising information and marketing [1-4]. The monopolization of scientific and engineering knowledge on these pumps is fostered by both the absence of standards, not only in specific countries, but also within such organizations as Europump and the European Community. In the USA, a new rotor pump standard (API 676) was introduced only in 1980, while published handbooks contain only general technical information, usually for a whole family of rotor pumps.

The main theoretical problem in pump design, besides calculating positive-displacement losses and suction power, is establishing the profile of the stator working surface, which determines the kinematics and dynamics of the vanes, which in turn determine the overall lifetime of the pump. In domestic positive-displacement hydraulic drives, the stator is profiled only for multiple-action (dual-action) pumps and hydraulic motors [5].

Our research has shown that the basis of high-efficiency, single-stroke vane pumps is the shape of the stator. We established that the possibilities of improving pumps with a “round” stator are limited, and that such a design can be used only in the simplest pumps where improving efficiency is not an issue.

The nomogram (Fig. 1) can be used to estimate the increase in the feed rate $Q_\text{sh}$ of a pump with a shaped stator as compared with the feed rate $Q_\text{f}$ of a pump with a round stator, for otherwise equal conditions (diameter and width of the rotor, the number $z$ of vanes, and the pump rotor RPM).

One design constraint for a vane pump is that the vane must be in constant contact with the stator during the operation process. Separation of the vane from the stator leads to rapid wear of the contact surfaces of the vane and...
### TABLE 1

<table>
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<th>Domestic pumps</th>
<th>Foreign pumps</th>
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<tr>
<td></td>
<td>SVN-80A</td>
<td>STsL-20-24G</td>
<td>Alfons-Haar (Germany)</td>
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<tr>
<td>Pump type</td>
<td>Vortex</td>
<td>Centrifugal vortex</td>
<td>Vane</td>
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<tr>
<td>Feed rate, m³/hr</td>
<td>30-45</td>
<td>30-45</td>
<td>40-50</td>
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<tr>
<td>Head, m (Pressure, MPa)</td>
<td>20-40</td>
<td>20-40</td>
<td>(Up to 0.8)</td>
</tr>
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<td>RPM</td>
<td>1450-1700</td>
<td>1450-1700</td>
<td>1000-1200</td>
</tr>
<tr>
<td>Attainable positive-suction head, m</td>
<td>3.75</td>
<td>1.5-5</td>
<td>4.0</td>
</tr>
<tr>
<td>Power, kW</td>
<td>6.5</td>
<td>10-24</td>
<td>10.5</td>
</tr>
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<td>Efficiency, %</td>
<td>36</td>
<td>33</td>
<td>82</td>
</tr>
</tbody>
</table>

**Note:** Indices of foreign vane pumps are for rotation rates that give comparable indices at the same attainable positive-suction head.

The stator, and also to increased pump noise. If a custom-shaped stator is used, these drawbacks can be eliminated. Pascal’s limaçon, which is described by a known algebraic curve, is the best known stator profile that provides constant contact between the vane and the stator, thanks to the cosinusoidal variation of the stator radius. The calculated characteristics of this shape are close to the actual stator dimensions of pumps made by the Alfons Haar firm (Germany).

Research has established that the conditions which provide contact of the vanes to the surface of a symmetric stator are determined by the expression:

$$m\omega^2[r - r_\text{st} - (\pi^2/2\alpha^2)(R - r)] > mg(A - BF^*_\text{st})/(1 - F^*_\text{st}).$$

where $m$ is the mass of the vane; $\omega$ is the rotation rate of the pump rotor; $g$ is the acceleration due to gravity; $r$, $R$, $\alpha$, and $\varepsilon$ are the characteristics of the stator profile (radius of the profile and the aperture and contact angles of the profile); $l_\text{st}$ is the distance from the stator profile to the center of gravity of the vane; and $l_v$ is the vane width;

$$A = \mu_g(l_\text{st}/l_v)\sin(\varepsilon/2);$$

$$B = A[1 - (1/\mu_g\mu_\text{st})] + (1/\mu_\text{st})\sin(\varepsilon/2) + \cos(\varepsilon/2);$$

$\mu_g$ and $\mu_\text{st}$ are the friction coefficients of the vane in the rotor groove and along the stator profile;

$$F^*_\text{st} = \mu_g(l_v + l)(P_{\text{sp}} - P_\text{o})/[l_v + \mu_g\mu_\text{st}(l_v + l) - l]$$

is the friction force of the vane without considering the vane mass; $l = r - r_\text{st}$ is the difference between the minor radius of the profile and the radius of the rotor; and

$$P_{\text{sp}} - P_\text{o} = m[(\omega^2(\rho - l_\text{st}) - (d^2\rho/ds^2)]$$

is the contact force from the action of inertial spring forces, which is determined for each moment of time $\tau$ that the rotor rotates and for the current radius vector $\rho$ of the profile.

Equation (1) can be used to evaluate the effect of the vane mass, the rotation frequency, and the geometry of the stator profile on the contact force of the vane.

From the research results, we developed an analytical model and engineering calculation method that can be used to analyze the pump design and predict its characteristics. The analytical model is based on the analytical dependence of the feed