A method is proposed for selecting the best values for the geometrical and working parameters of a spiral impeller with constant pitch, which gives the maximal cavitation performance in pumps in which it is used either as a preliminary pumping facility (centrifugal pump) or the inlet section of an axidiagonal pump.

Centrifugal pumps, including major oil pumps (MOP) of type NM have large permissible cavitation margins $\Delta h_{\text{per}}$, which increase with the nominal output to 97 m for the NM 10000-210 pump with interchangeable rotor for an output of 12500 m$^3$/h. When oil is passed from pump to pump, the pressure at the inlet to the centrifugal pump produced by the previous oil-pumping station (OPS) limits the throughput of the major pipeline (MP). If there is a part in the MP with minimal head, which is equal to the limiting head in the pipeline, the throughput of the MP can be raised by reducing the heads in a series of OPS as defined in the hydraulic calculation.

It has been shown [1] that the throughput of an MP can be increased by 1.2% by reducing the permissible cavitation margin in the major pumps to 10 m. That reduction can be done either by installing pressurizing pumps at intermediate OPS or by installing a spiral stage before the centrifugal one. In that case, we have what is called a spiral centrifugal pump (SCP). The transition from a centrifugal pump to an SCP can be provided either by installing the spiral on existing major centrifugal pumps (modifying the centrifugal pumps) or by setting up new major SCP.

It is found [2] that installing such a spiral in a model NM 7000-210 centrifugal pump (speed coefficient $n_s = 196$) increases the critical cavitation coefficient by 29% (from $C_{cr} = 1214$ to $C_{cr} = 1566$). Such a spiral is used in aerospace booster pumps or in an SCP to provide $C_{cr} = 4000–5000$ [3]. However, such spirals with optimum cavitation parameters are rather uneconomical: efficiency $\eta = 0.4–0.55$, which is due to the high nonuniformity patterns at the inlet and outlet (particularly on light loading, where reverse flows occur at the edge of the inlet pipe, while countercurrents occur beyond the spiral at the sleeve), and also there are elevated hydraulic losses in the spiral itself and in the output (in the case of a booster pump).

The best form of MOP is a spiral diagonal pump (SDP), which combines the good economy of typical NM 2500 – NM 10000 pumps ($\eta = 0.86–0.89$) with the good cavitation performance of the SCP used in aerospace engineering ($C_{cr} > 5000$).

The rotor in an SDP can be divided into input, transitional, and output (pressurized) parts [4]. The cavitation performance is determined by the inlet part, which has geometrical parameters approximately corresponding to a spiral of constant pitch with a cylindrical sleeve. The output section provides the required head and high efficiency of the pump, while the transition region provides a smooth transition from the inlet part to the outlet (pressurized) one.

To attain the maximum cavitation performance of the SCP and SDP, one should optimize the geometrical and working parameters of the constant-pitch spiral with cylindrical sleeve.
A method has been given [5] for choosing the outside diameter of the spiral or the working parameter (flow angle at the inlet) from the viewpoint of cavitation performance by examining the turning point in the empirical equation relating the critical cavitation margin to the basic geometrical parameters and spiral diameter coefficient $K_D$ (flow angle at inlet), but this is incorrect because the initial equation lacks the angle of attack $\alpha$, which is one of the parameters determining the critical cavitation margin.

Models have been designed [6, 7] for the detached flow on a planar grid of plates or a grid of blades in the spiral of constant pitch, which has given theoretical relationships for determining the best $K_D$ and $\alpha$ in relation to the sleeve ratio $d$, as well as to the sizes and shapes of the input edges of the blades [6], including cylindrical ones [7]. However, analysis shows that the best $K_D$ and $\alpha$ obtained in those papers do not correspond to a local turning point in $C_{cr}$, which is a function of two independent variables: $K_D$ (flow angle at inlet) and angle of attack $\alpha$, subject to the condition that $d$ and the radius of the inlet edge of a blade $r^*$ are chosen as the minimum possible from design considerations and strength.

Visual observations on the cavitation show that in the spiral there is characteristically profile cavitation of jet form at the inlet side of a blade in the spiral and starting from the inlet edge [4]. The limiting or third critical condition (supercavitation mode) occurs with completely developed detached flow on the blades of the spiral with a profile cavity propagating beyond the spiral, which determines the characteristic governing the working limit. The third critical condition is preceded by the second critical one (or presupercavitation one), when the cavity reaches approximately the throat of the grid (i.e., the density of the grid occupied by the cavity is approximately one). If there is a slight pressure reduction at the inlet, that cavity enlarges stepwise and emerges beyond the grid, i.e., there is a transition from the second critical state to the third one, with a sharp fall in the pressure provided by the spiral, which leads to cavitation breakaway in the centrifugal pump or outlet (pressurized) section of the SDP, and thus of the pump as a whole. Then cavitation in the SCP or SDP corresponds to the second critical state for the spiral in the case of SCP or inlet part for the SDP.

As the thermophysical properties of oil and water differ considerably, the cavitation erosion in the flow part of the MOP is hardly evident [8]. Therefore, the preceding spirals working with developed cavitation can be used as parts of oil SCP or SDP whose working life to major repair is about 60 thousand hours.

We use the spirals with grating density $\tau \geq 2$ and small radial gap, there is characteristically a steeply falling part of the cavitation characteristic, so the cavitation margins for the second critical state ($\Delta h_{II}$) and the third ($\Delta h_{III}$) are almost the same, i.e., $\Delta h_{II} = \Delta h_{III}$.

In supercavitation (detached flow), the reverse flows at the inlet and the countercurrent at the outlet are absent [4], and to determine the best spiral parameters one can use a two-dimensional flow model on the assumption that the meridian velocity and specific flow energy before the spiral are uniformly distributed over the radius.

We now consider the detached flow for a blade in relation to the plane of an elementary cylindrical grating as cut to the mean diameter of the spiral $D^* = 0.5D_s(1 + d)$, in which $D_s$ is the outside diameter of the spiral and $d$ is the sleeve ratio (Fig. 1). The asterisk denotes quantities relating to the mean spiral diameter.