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DETERMINATION OF VOLUMETRIC LOSSES IN HYDRODYNAMIC PUMP USING
NUMERICAL MODELLING

STANOVENÍ OBJEMOVÝCH ZTRÁT V HYDRODYNAMICKÉM ČERPADLE S VYUŽITÍM
NUMERICKÉHO MODELOVÁNÍ

Abstract

This paper deals with the numerical modelling of the flow in the single-stage centrifugal pump. The main objective is to determine leakage losses through annular seals at the suction side of the pump. Leakage through a shaft seal is not included in the simulation. The amount of liquid that circulates from the impeller discharge back to suction of the pump is determined in dependence on the flow rate. Losses in the pump are further discussed as well as the possibility of their prediction.

Abstrakt

Článek se zabývá numerickým modelováním proudění v jednostupňovém hydrodynamickém čerpadle. Hlavním cílem je určení objemových ztrát skrze těsnicí kruhy v čerpadle v závislosti na průtoku. Průtok měkkou ucpávkou není v simulaci zahrnut. Předmětem výpočtu je pouze průtok mezerou mezi těsnícími kruhy na sání čerpadla, přičemž je stanoveno množství kapaliny, která recirkuluje mezi výtlakem a sáním čerpadla. To se projeví na velikosti ztrát, o kterých je také pojednáno.

1 INTRODUCTION

Leakage through annular seals of the centrifugal pump contributes to decreasing of the pump efficiency. The annular seal consists of the outer fixed casing ring and the inner ring rotating with the impeller. The radial clearance between those rings is very small in comparison with the radius of the rotating parts. The correct clearance design is of great importance, as the leakage flow increases approximately with the 1.5th power of the clearance [3]. Especially at low specific speeds, the deviation from the assumed dimensions has a remarkable effect on the efficiency.

There are many factors that influence the leakage flow through the annular seal:

- geometry of the annular seal (shape, clearance and length, sharpness of edges)
- pressure difference across the seal
- pump specific speed
- regime of the flow defined by *Re* number
- roughness of surfaces

To test the influence of individual factors, numerical modelling can be applied as a tool. This can help to obtain data that can be further used for the leakage prediction.

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In this study, numerical modelling has been applied to investigate the performance including the leakage flow in a single-stage centrifugal pump with horizontally mounted shaft and twisted blade. The design parameters are specified in table 1.

Tab. 1 Design parameters of modelled centrifugal pump.

Head	$H =$	80.9	[m]	Outlet diameter	$D_2 =$	244	[mm]
Flow rate (capacity)	$Q =$	7	[dm$^3 \cdot$s^{-1}]	Number of blades	$z =$	5	
Rotational speed	$n =$	2900	[min^{-1}]	Specific speed	n_b	0.0263	[1]

These parameters yield a very low value of non-dimensional specific speed

$$n_b = n \times \frac{Q^{0.5}}{Y^{0.75}} = \frac{2900}{60} \times \frac{0.007^{0.5}}{(80.9 \times 9.81)^{0.75}} = 0.0263, \qquad (1)$$

where:

n_b – specific speed [1], n – rotational speed [s^{-1}], Q_{V-} flow rate [m$^3 \cdot$s^{-1}], Y – specific energy [J$\cdot$kg^{-1}]

This value of n_b is below the specific speed range commonly considered for the radial type of impeller. It is caused by the high value of the head in connection with the low value of the flow rate. Such parameters can be reached by twisting the blades of the impeller, as was designed at the Victor Kaplan Department of Fluid Engineering, Energy Institute, Technical University of Brno.

2 NUMERICAL MODELLING OF THE FLOW IN CENTRIFUGAL PUMP

At the beginning of the simulation a 3-dimensional geometric model of the pump components was created which describes the calculation domain by coordinates. This calculation domain was then subdivided into a large number of cells, i.e. a grid is generated, the quality of which is essential for the reliable numerical solution.

2.1 Definition of the computational geometry

Dimensions of hydraulic parts were obtained from the Victor Kaplan Department of Fluid Engineering. Complete technical documentation was provided by Sigma Group Lutín. Based on the drawings, complex geometry of the rotor and stator was defined, including the space between the impeller and casing. The model is presented in Fig.1.
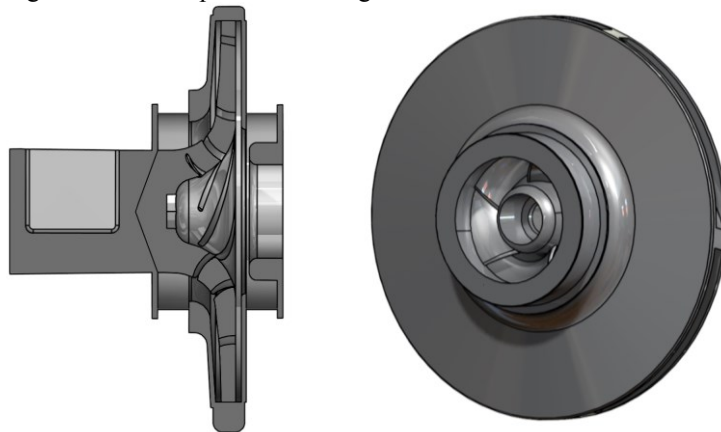
Fig. 1 Computational geometry of the pump.

The geometry was divided into six volumes (inlet part, impeller, volute with impeller sidewall gaps, clearances between impeller and casing on the front and rear side and the space behind the rear

shroud under the annular seal). Unstructured mesh was generated; the number of cells was about 4300000.

2.2 Numerical modelling

ANSYS FLUENT release 13[1] was applied for numerical modelling of the flow through the pump. The incompressible, isothermal, turbulent flow was modelled in complex 3D geometry. The problem involves multiple moving parts as well as stationary surfaces. Multiple Reference Frame model (MRF) was applied to model such case. MRF approach is steady-state approximations and differs primarily in the manner in which conditions at the interfaces are treated. In this case unsteady problem is transferred to the role of stationary due to the rotating coordinate system. Acceleration of the liquid is incorporated into the model as a new term in the momentum equation. This approach does not allow solving the interaction between the stator and rotor; results are given for one position of blades of the impeller against volute. Simulations take less of a time but dynamic effects of flow are neglected.

Turbulent model k-ω SST was applied that is recommended for the flow in rotating geometries [3]. This model employs the k-ε model in the core flow, the k-ω model near wall and it modifies the relation for the eddy viscosity.

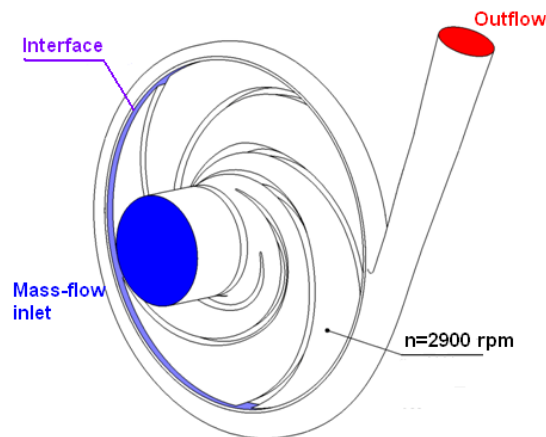Definition of boundary conditions is illustrated on the Fig. 2.

Fig. 2 Definition of boundary conditions

The convergence of the steady solution can be observed by evaluation of residuals and also by the monitoring of the static pressure at the outlet.

## 2.3   Losses in hydrodynamic pump

Losses resulting from the transformation of mechanical energy into hydraulic energy can be expressed in the steady state as a power loss:

$$\sum P_v = P - P_u \qquad\qquad (2)$$

where: $\sum P_v$ is the sum of all losses, P is the power supplied at the pump shaft, P_u is the useful power transferred to fluid $P_u = \rho \times g \times H \times Q$.

The power dissipation in the pump is converted to heat dissipation.

Balance of power flow and power loss in the pump shows the following figure (Fig. 3**Chyba! Nenalezen zdroj odkazů.**) [3].

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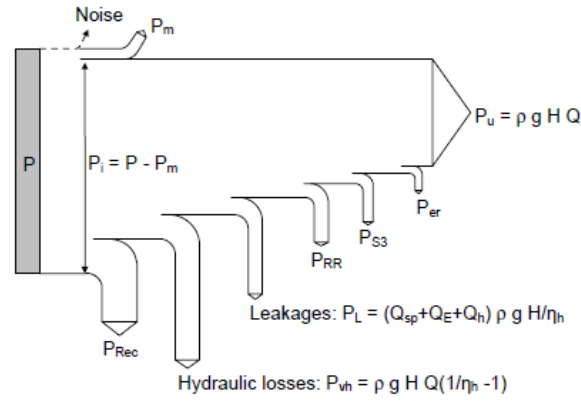
Fig. 3 Balance of the power flow [3]

In the scheme (Fig.3) the following sources of loss are considered [3]:

P_m – *mechanical losses* in the bearings and shaft seals. These losses usually do not lead to heating of the liquids and are determined as external loss.

P_{er} – *friction losses* created by the components of axial thrust balance devices (balance piston or disk).

P_{s3} – *throttling losses* which occur in the annular seal separating two pump stages. These leakages do not flow through the impeller and so they consume only a part of the power; they are evaluated with only about 40% of the head.

P_{RR} – *disk friction losses* which are generated on rear and front shrouds of the impellers.

P_L – *volumetric losses* connected with the leakages.

Volumetric losses belong to secondary losses. Generally they include:

- the leakage (Q_{sp}) through the annular seal at the impeller inlet
- the leakage (Q_E) through the device for axial thrust balancing
- additional amount of fluid (Q_h) which may be circulated within the pump for auxiliary purposes such as sealing or cooling, or feeding a hydrostatic bearing.

To pump all of these leakages, the power P_L must be supplied by the impeller where η_v is the volumetric efficiency.

$$P_L = \rho \times g \times H_{th} \times \left(Q_{sp} + Q_E + Q_h\right) = \rho \times g \times H_{th} \times Q / \eta_v \qquad (3)$$

where:

$$H_{th} = \frac{H}{\eta_h} \qquad (4)$$

The volumetric losses can be expressed by means of volumetric efficiency:

$$\eta_v = \frac{Q}{Q + Q_S + Q_E + Q_P} = \frac{1}{1 + \dfrac{Q_S}{Q} + \dfrac{Q_E}{Q} + \dfrac{Q_P}{Q}} \qquad (5)$$

P_{Vh} – *hydraulic losses* due to friction and turbulent dissipation in all components between suction and discharge nozzle are covered by the hydraulic efficiency η_h. The dissipated power due to hydraulic losses is then defined:

$$P_{Vh} = \rho \times g \times H \times Q \times \left(\frac{1}{\eta_h} - 1\right) \qquad (6)$$

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P_{Rec} – *losses due to fluid recirculation* in case of part load. They should be zero near and above the best efficiency point. They cause the greatest part of the power consumption, when operating against a closed valve or at a low flow rate.

All types of losses can be estimated by the help of empirical formulas, but uncertainties of ±20 % to 30 % can be expected. Loss analysis is very important for the pump design as well as for control of the pump performance. Numerical modelling can be a very useful tool for this purpose. In this paper volumetric losses will be investigated connected with the leakage Q_{sp} through the annular seal at the impeller inlet.

2.4 Leakage through the annular seal

The straight plain annular seal with constant clearance was assumed. For the leakage calculation, the pressure difference across the seal must be defined. This pressure difference depends both on the pump head and the pressure generated by the friction force in a gap between the stator and rotating disks. The distribution of static pressure is illustrated in Fig. 3.
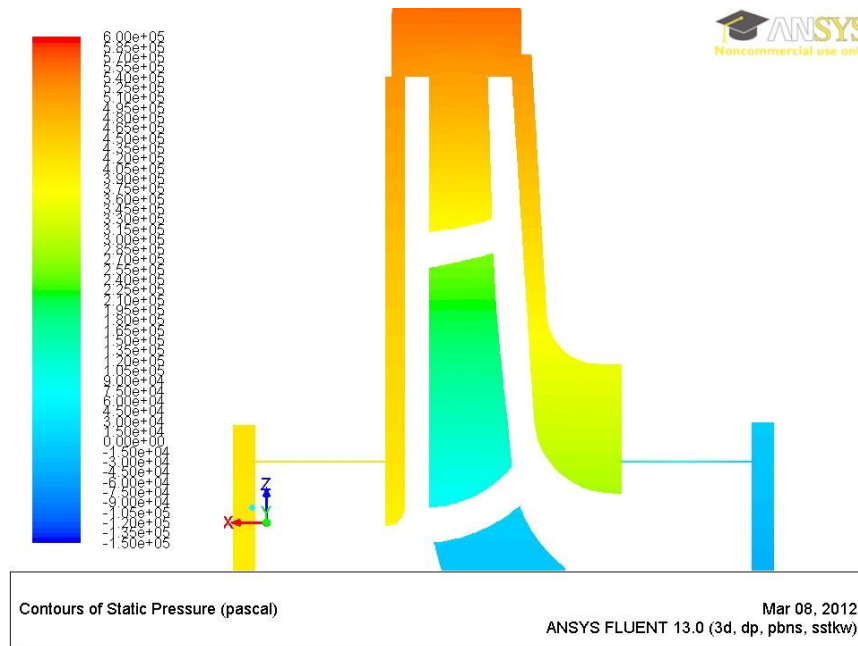
Fig. 3 Contours of static pressure in the impeller and gaps between impeller and rotor.

In case of the straight plain annular seal, the leakage can be calculated according to [5] using the empirical formula given by (7). The variables in this formula are illustrated ion Fig. 4.

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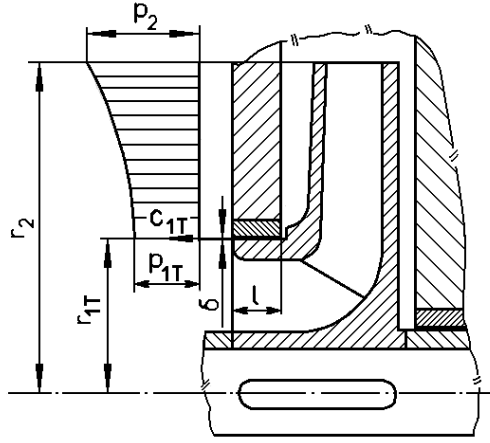
Fig. 4 Scheme of the leakage losses through annular seals at the suction side of the pump [5].

$$\frac{p_{1T}}{\rho} = Y \times \left(1 - \frac{Y}{2u_2^2\eta_h^2}\right) - \frac{\omega^2}{8} \times \left(r_2^2 - r_{1T}^2\right) \tag{7}$$

The velocity in a gap between the sealing rings:

$$c_{1T} = \mu \times \sqrt{2 \times \frac{p_{1T}}{\rho}} \tag{8}$$

where μ is the discharge coefficient defined by:

$$\mu = \frac{1}{\sqrt{\frac{\lambda \times l}{2 \times \delta} + 1.5}} \tag{9}$$

where: λ [1] is a friction coefficient dependent on the shape of the sealing ring,
δ [m] is the clearance between the rings,
l [m] is the length of the length of the sealing gap.
Friction coefficient is determined experimentally and for smooth annular seals and turbulent flow its value is in the range $\lambda = (0.035 - 0.05)$.

The leakage flow through the annular seal can be calculated from:
$$Q_{sp} = 2 \times \pi \times r_{1T} \times \delta \times c_{1T} \tag{10}$$

This theoretical prediction can be applied in the case of best efficiency point. Besides the pump optimal operational conditions, numerical modelling can be applied to investigate leakage flow. In this case leakage flow has been investigated for the flow rate ranging from 0 dm^{3}s^{-1} to 10 dm^{3}s^{-1}. The results obtained from theoretical prediction and numerical simulation is compared in Fig. 5.

It can be seen that leakage flow is dependent on the pump flow rate (capacity). The maximum of leakage was obtained for zero flow rate which is corresponds to the maximum of the pump head. With increasing flow rate the leakage is decreasing and reaches its minimum in BEP. Behind BEP it is increasing again.
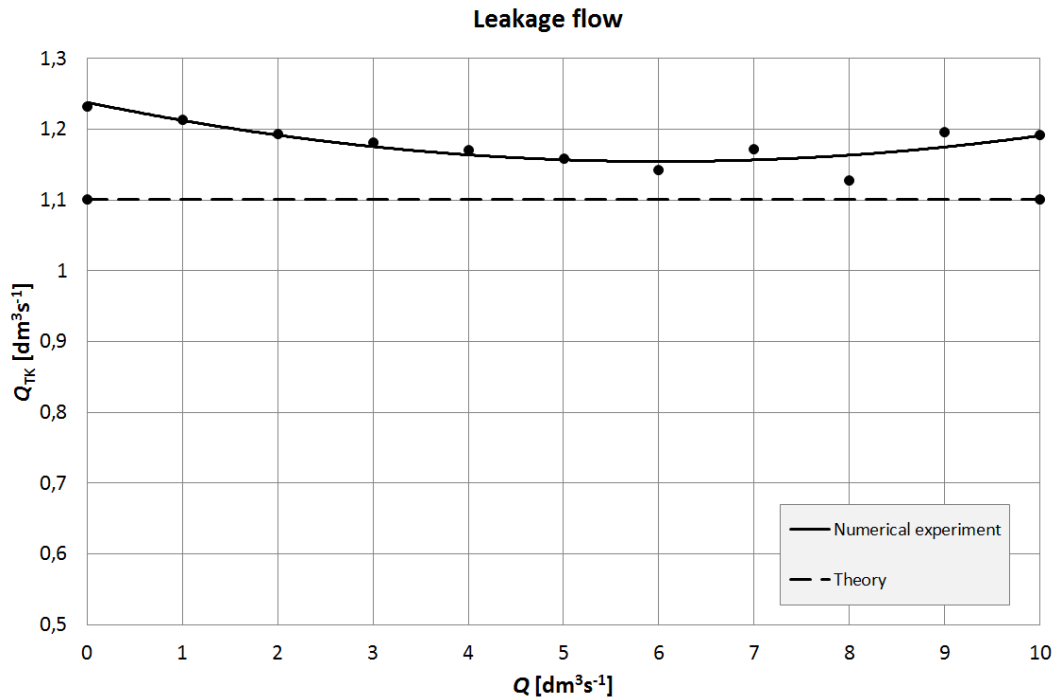
Fig. 5 Comparison of results obtained from theoretical prediction and numerical simulation.

Comparison of results from the theory and numerical modelling shows that for the preliminary estimation of annular seal leakage a simple calculation based on empirical relations can be applied. However, if we want to determine leakage depending on the pump flow rate, it is possible to use numerical modelling tools. This can be of great benefit especially in the stage of pump design, when the shape, clearance and other important features of the annular seal can be tested. In this modelled case the annular seal leakage substantially exceeds volume loss specified in the literature (about 3% - 5% of rated flow). In this case it would be useful to use the labyrinth or different type of grooves in order to increase the flow resistance.

3 CONCLUSIONS

It can be concluded, that the annular seal leakage can be predicted by empirical formulas derived from measurements. For the investigation of the annular seal leakage in dependence on the pump flow rate, numerical modelling can be applied. Numerical modelling can provide deeper insight into fluid flow in a centrifugal pump. Close interaction exists between leakage flow, pressure distribution in the impeller sidewall gap, disk friction and main flow. The results obtained with numerical modelling can lead to change of the seal type or modification of its geometry. Numerical investigation carried out for alternative shape of the annular seal can help to evaluate whether the change in geometry is reflected in the pump parameters.

Another advantage of numerical modelling may be the fact that the theoretical relationships are derived only for some standard shapes of sealing rings. In case of non-standard sealing rings and the settings of the flow, the theoretical assumptions may fail. In this field, numerical modelling can again be applied as a helpful tool.

REFERENCES

[1]   FLUENT 6.3.26 – *User´s guide*. Fluent Inc. 2003 [online]. Datum poslední revize 3.10.2006
       [cit. 2008-12-08]. Dostupné z
       < http://spc.vsb.cz/portal/cz/documentation/manual/doc.vsb.cz/Aplikacni%20software/FlueFl_
       6.3.26/pdf/ug/flug.pdf >.

[2]   BLAHA, J.; BRADA, K. *Hydraulické stroje.* 1st ed. Praha: SNTL, 1992. 747 pp. ISBN 80-03-
       00665-1.

[3]   GÜLICH, JOHANN, F. *Centrifugal Pumps*, Second Edition: Springer, 2010. 964 pp. ISBN
       978-3-642-12823-3.

[4]   TUZSON, J. *Centrifugal pump design* 2000. 320 pp. ISBN 0-471-36100-3.

[5]   PACIGA, A., STRÝČEK, O., GANČO, M. *Čerpacia technika*. Praha: SNTL, 1984. 224 s.