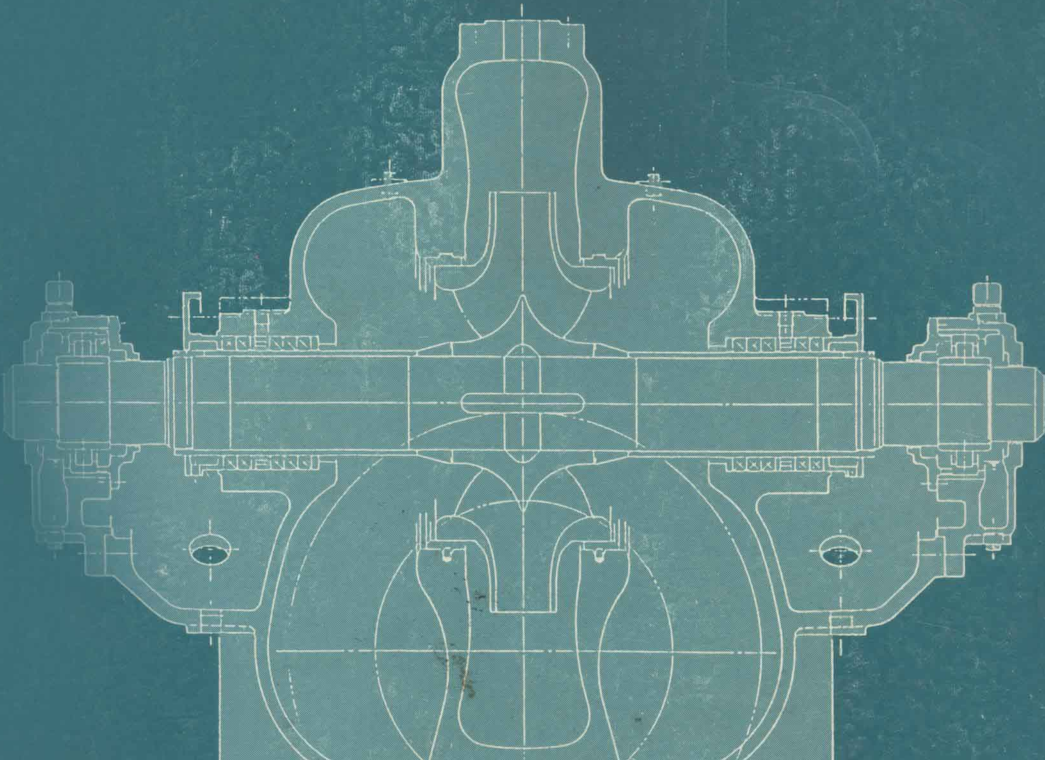
Radial loads and axial thrusts on centrifugal pumps



Papers presented at a Seminar organized by the Fluid Machinery Committee of the Power Industries Division of the Institution of Mechanical Engineers and held at the Institution of Mechanical Engineers on 5 February 1986

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The Institution of Mechanical Engineers
LONDON

First published 1986

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ISBN 0 85298 603 3

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Printed by Waveney Print Services Ltd, Beccles, Suffolk

RADIAL LOADS AND AXIAL THRUSTS ON CENTRIFUGAL PUMPS

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Review of parameters influencing hydraulic forces on centrifugal impellers

J GUELICH and **W JUD**
Sulzer Bros., Winterthur, Switzerland
S F HUGHES
Sulzer Bros., Leeds, UK

SYNOPSIS On the impeller of a centrifugal pump there are acting both radial and axial hydraulic forces. The radial forces are due to three different mechanisms: (1) radial thrust due to non-uniform pressure distribution around the impeller (2) labyrinth forces (3) impeller diffuser interaction forces. Nature, size and origin of these forces are discussed. Steady radial thrust depends on the specific speed of the pump, type of casing (single or double volute, diffuser) geometric parameters and operation conditions including cavitation.

The impact of these factors on radial loads is demonstrated and quantitative data are presented. Four different methods for radial thrust measurements are presented.

Axial thrust on pump impellers is extremely sensitive to tolerances and to the labyrinth flow rate. Two models for the prediction of axial thrust are presented and predictions are compared to measurements on a multi-stage boiler feed pump. A number of parameters influencing axial thrust are discussed.

1 INTRODUCTION

Non-uniform pressure distributions around an impeller cause radial forces on a pump rotor. Measurements of radial thrust in single volutes were published in ref. (1) for single entry impellers and in (2) for double entry impellers. Less information was published on double volutes and on the influence of different geometric or operational parameters on radial thrust. A general literature review is given in (27) but it contains no data.

It is the object of the present contribution to show the physical mechanisms causing radial forces on an impeller, to review and supplement the available data and to present them in a form readily usable for design purposes.

2 SYMBOLS/ABBREVIATIONS

BEP	best efficiency point
D	diffuser
DV	double volute
SV	single volute
C2u	circumferential component of absolute velocity at impeller outlet
B2*	impeller outlet width including shrouds (fig. 1)
D2	impeller outer diameter
fn	frequency of shaft rotation
FR	radial thrust
g	acceleration due to gravity
H	head
Hp	static head rise of impeller
K	ratio of liquid velocity in space between casing and impeller and circumferential impeller speed
KR	radial thrust coefficients, defined in equ. (2)
KRs	steady radial thrust coefficients
KRun	unsteady radial thrust coefficients (peak amplitude)
KRO	maximum steady radial thrust coefficient for single volutes
KR*	radial thrust coefficients of axial pumps
Qv	leakage flow rate
Q	flow rate
n	speed of pump shaft
nq	specific speed (n in rpm, Q in m3/s per eye of impeller, H in m per stage)

R	maximum radial thrust of double volute/maximum radial thrust of single volute
Re	Reynolds-number
y	defined by equ. (3)
Z2	number of impeller blades
Z3	number of diffuser blades
β_2	impeller outlet angle
ε_{Sp}	circumferential angle of inner volute
ν_D ν_I	mode numbers: 1, 2, 3, ...
ρ	density
ω	orbit frequency
Ω	angular frequency of impeller rotation

Subscripts

oo	best efficiency point
o	for $Q = 0$ or maximum radial thrust

3 ORIGIN AND MAGNITUDE OF RADIAL LOADS ON CENTRIFUGAL IMPELLERS

Apart from mechanical unbalance there are three types of radial hydraulic forces acting on a rotating impeller of a centrifugal pump:

a) Radial thrust
b) Labyrinth forces
c) Impeller/diffuser interaction forces

To understand the nature and origin of these forces will help the interpretation of radial thrust measurements and their application to pump design. Tables 1 and 2 give a summary of nature, origin and typical magnitude of radial forces acting on a centrifugal impeller.

3.1 Radial thrust

A radial load ("radial thrust") on a centrifugal impeller is caused by a non-uniform pressure distribution around the circumference of the impeller. The total radial thrust can be split up into steady and unsteady components.

The steady radial thrust can be caused by:

- non-uniform pressure distribution in single volutes. The non-uniformity increases strongly at off-design conditions.
- non-uniform pressure distribution in annular casings
- non-uniform pressure distribution in double volutes caused by the fact that outer and inner volute have different flow resistances
- static excentricity of impeller with respect to diffuser or volute
- geometrical tolerances in theoretically symmetric configurations such as twin volutes or diffusers
- Non-uniform flow distribution at the impeller inlet

The unsteady radial thrust can be caused by:

- geometrical tolerances of impeller (e.g. one blade has a larger outlet angle than the others). This phenomenon is sometimes termed "hydraulic unbalance" since this thrust component has the frequency of the rotational speed of the pump.
- pressure fluctuations (mostly low frequency) due to flow separation at part load (sometimes as rotating stall)
- unbalanced blade forces (9), if:

$$\nu_D \cdot Z_3 - \nu_I \cdot Z_2 = \pm 1 \qquad (1)$$

Both steady and unsteady radial thrust data given in this paper are normalized according to:

$$K_R = \frac{F_R}{\rho \cdot g \cdot H \cdot D_2 \cdot B_2^*} \qquad (2)$$

Unsteady radial thrust values given in this paper are to be understood as peak amplitudes in a frequency range between zero and impeller blade passing frequency.

3.2 Labyrinth forces

In a labyrinth there are radial and tangential forces which increase with the static excentricity and the vibration orbit of the shaft. While these forces are negligible for radial labyrinths, they are strong for axial flow labyrinths as used in most pumps. Therefore these forces should not be ignored when interpreting radial thrust measurements.

Fig. 1 shows how radial thrust and labyrinth forces determine the rotor position and influence the apparent radial thrust: We assume that a radial thrust F_{RT} acts on the impeller and pushes the impeller from its concentric position M to the excentric position O_1 which would be observed with a radial labyrinth. An axial flow labyrinth, however, would give a restoring radial force F_{LX} pushing the rotor somewhat back and a tangential force F_{LY}. The observed rotor position would therefore be O_2 and the measured radial thrust F_{RM} differs by magnitude and direction from the true hydraulic thrust F_{RT}. A non-vibrating rotor under the assumed radial load would rotate around O_2 and a vibrating rotor would show some orbit around O_2. In practice every rotor will be subject to some vibrations.

The labyrinth forces discussed above can be estimated from numerous publications, e.g. ref. (3), but reliable data for high Reynolds-numbers and different types of labyrinths are still scarce. In the present context it is sufficient to note that the radial and tangential forces increase with the head of the pump or the square of the speed and the damping forces increase with the speed of the pump. There is also an apparent mass associated with the acceleration of liquid in the gap of the labyrinth, but this effect can be ignored for the discussion of radial thrust. In the literature the labyrinth forces are treated as stiffness, damping and mass coefficients. The order of magnitude of the stiffness effect (calculated with half the labyrinth clearance) can be seen from table 1, where the labyrinth forces were normalized according to equ. (2) in order to provide a comparison with other radial forces. The negative sign indicates that it is a restoring force acting in opposite direction of the radial displacement of the rotor.

3.3 Impeller diffuser interaction

If a rotor vibrates with respect to the casing (volute or diffuser) the pressure distribution around the impeller circumference changes due to the varying distance between impeller and casing. The resultant forces can be described as radial and tangential stiffness, damping and mass coefficients in a similar manner as for the labyrinth forces, ref. (4). These forces can not be readily separated from the measured radial thrust but they are much lower than the labyrinth forces, since the gap between impeller and diffuser is much larger than the labyrinth clearance. They can be neglected in radial thrust considerations.

4 MEASUREMENT OF RADIAL THRUST

Four methods are available to measure the radial thrust. Detailed discussion on related problems can be found in (7) and (8).

4.1 Measurement of pressure distribution

The pressure distribution around the circumference of the impeller is measured by a number of pressure tappings and the resultant radial force is determined by integration of the pressure profile measured (5), (6).

Advantages:

- This method is simple, since no sophisticated equipment is needed.
- No resonance with flexible measuring elements endangers the accuracy of the results.

Disadvantages:

- The measurement of a multitude of pressure tappings and the integration is cumbersome
- The accuracy is low
- Instationary thrust components are not recorded.

The interpretation of the results is difficult because the pressure distribution on the impeller shroud and hub and their possible contribution to the force on the impeller are unknown. It is therefore not surprising to find that radial forces integrated from the measured pressure distribution in ref. (6) are lower than the forces on the bearing since the integration was only done over the impeller outlet width.

4.2 Measurement of bearing forces

The forces transmitted from the rotor to the stator can be measured if the bearing is supported by a device comprising brackets which are equipped with strain gauges. This measuring assembly can be calibrated by an unbalance and/or by static loading. In order to get good signals from the strain gauges the brackets must be dimensioned accordingly. This results in a system with low stiffness and low resonant frequencies. It is therefore necessary to run a test with a mechanical unbalance at variable speed in order to check that the bearing forces increase with the square of the speed and there is no resonant amplification at the test speed. Contrary to the pressure integration technique the bearing measurements can only give the resultant of all forces acting on the rotor (i.e. true hydraulic thrust <u>and</u> labyrinth).

Other means of force measurement can be employed instead of strain gauges.

4.3 Measurement of shaft deflection

The forces acting on the rotor can also be determined by measuring the shaft deflection by proximity probes, if the shaft is calibrated by a known mechanical unbalance and/or static load.

Advantages:

- relatively inexpensive and simple tests
- often only small modifications of the test pump are necessary

Disadvantages:

- The bearing clearances affect the readings of the proximity probes
- accuracy moderate
- run-out problems

4.4 Measurement of shaft stresses

The stresses produced by the forces acting on the impeller can be measured by strain gauges on the shaft as described in ref. (4). As with measurement of the bearing forces resonances must be avoided and the measured forces include also the labyrinth forces unless a radial labyrinth (see fig. 1) is employed. This technique clearly is much more complex than the measurement of the bearing forces and is only justified in special cases.

If resonance problems can not be avoided the test pump can be treated as a vibrating system consisting of one mass, stiffness and damping (8).

As in the methods of 4.2 and 4.3 the system must be checked for resonances, since the shaft must be flexible enough to permit accurate readings.

5 RADIAL THRUST

5.1 Radial thrust in volute pumps

5.1.1 Volute pressure distribution and radial thrust

At part load there is flow separation at the suction side of the cutwater since the flow angle is smaller than the geometric angle of the tongue (see fig. 2a). Consequently in the stalled region there is little pressure recovery, flow deceleration takes place in the second half of the volute, and the radial thrust is directed to the stalled region, i.e. somewhat downstream of the tongue.

Putting in an additional rib opposite the cutwater (i.e. part of a double volute) produces a second stalled region and increases the symmetry of flow and pressure distribution considerably (see fig. 2b). Tests reported in (16), which demonstrate this effect very clearly are reproduced in fig. 3.

If the tongue is cut back to form a concentric casing (e.g. over 90° of the circumference) as shown in fig. 2C the stalled region is eliminated since the geometric tongue angle has been decreased and radial thrust at part load is reduced accordingly. The radial thrust measurements of ref. (1) (15) (18) show clearly this reduction of thrust at part load.

At flow rates in excess of the design point the flow separation is (due to the high flow angle) on the pressure side of the tongue (i.e. in the pressure nozzle). Consequently there is a pressure build-up on the suction side of the tongue (see fig. 2d) and the radial thrust is directed more or less opposite the direction at part load. Since the volute throat velocity is much larger than the impeller outlet velocity, there is an acceleration and an associated drop in static pressure near the cutwater.

The mechanisms described above are confirmed by measured pressure distributions (17) (5) (19) (20) and directions of the radial thrust. However non-uniform pressure distribution at off-design conditions occurs already without flow separation. The direction of the radial thrust in single volutes is given in fig. 2a and 2d.

For double volutes the steady radial thrust at part load can be very sensitive to small geometrical variations, which impair the symmetry of the pressure distribution in the inner and outer volute. This is demonstrated also by fig. 6 which shows that small deviations from a 180° - volute cause an appreciable rise in radial thrust.

It appears that the low radial thrust of a double volute depends on a rather delicate equilibrium of pressures in the inner and outer volute that can be destroyed easily by unsymmetries of the cutwaters and the shape of the volute. In this respect radial thrust in single volutes is probably less sensitive to small modifications.

5.1.2 Design data for single and double volutes

In the absence of specific test data fig. 4 can be used to estimate the maximum steady radial thrust for single volutes with single or double entry impellers. K is defined by equ. (2) and y by equ. (3):

$$y = a \cdot nq \qquad (3)$$

$a = 1$ for single entry impellers
$a = \sqrt{2}$ for double entry impellers

Basis for fig. 4 are the data of (1) and (2) the band of scatter is estimated from available test data. Curve 2 is a rough estimate from tests with large clearances, which are confirmed by ref. (28).

Curve 1 in fig. 4 is valid for clearances according to ref. (10).

The excessive leakage associated with large clearances (i.e. clearance above 2- to 3-times the API 610 values)

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has an impact on the pressure distribution on the impeller shrouds. Nevertheless it is suggested that curve 2 should be used too for radial labyrinths with any type of clearance.

The maximum steady thrust given by fig. 4 occurs mostly at Q=0, in some cases for $0 < Q < 0.5\ Q_{oo}$. Having determined K_{Ro} from fig. 4 the thrust curve $K_R = f\ (Q)$ can be estimated from the formula proposed by Stepanoff (30).

$$K_R = K_{Ro} \left[1 - \left(\frac{Q}{Q_{oo}} \right)^2 \right] \tag{4}$$

Contrary to the above formula the steady radial thrust is not zero at BEP but has a finite value in the range of $0.03 \leqslant K_R \leqslant 0.08$ which we found independent of the specific speed in the range of $10 \leqslant nq \leqslant 100$. This force is caused mainly by the final thickness of the cutwater.

Because of the high number of variables it seems so far impossible to establish a correlation with less scatter than in fig. 5 by introduction of specific geometric design parameters.

Fig. 5 gives steady radial thrust coefficients for 50 % of BEP flow. The upper scatter band is even larger than in fig. 4 since often maximum thrust does not occur at shut-off but at part load up to 50 %: The maximum thrust occurs when the impeller blade loading becomes excessive (i.e. the maximum local pressure in the volute is limited by the impeller blade loading).

For double volutes fig. 6 together with fig. 4 can be used to determine the maximum steady radial thrust. The maximum steady radial thrust of a double volute is determined by:

$$K_{R\text{-}DV} = R \cdot K_{Ro} \tag{5}$$

For $\varepsilon_{Sp} = 180°$ the curve $F_R = f\ (Q)$ can be assumed very flat, i.e. for design purposes the F_R calculated from fig. 6 and 4 must be assumed for all flow rates. With decreasing ε_{Sp} the relation $F_R = f\ (Q)$ assumes more and more the shape given by equation (4) for single volutes. Fig. 6 is based on a number of tests with different types of pumps and different specific speeds. It is confirmed by data from (21) and (29).

Unsteady radial thrust for single and double volutes can be estimated from table 4.

5.1.3 Geometric parameters

5.1.3.1 Casing clearance

Large clearances between casing and impeller shrouds foster the equilization of the pressure around the impeller circumference and hence reduce radial thrust. In ref. (21) the difference between a (unusually) small clearance of $0.006\ D_2$ and a wide clearance of $0.035\ D_2$ brought an increase of $\Delta K_R \cong 0.1$ in maximum radial thrust for nq=23. For a pump of nq=45 the increase was about $\Delta K_R \cong 0.04$ and for a double volute with nq=23 the increase was $\Delta K_R \cong 0.034$. Pressure distribution measurements in (17) showed a similar effect.

5.1.3.2 Cutwater distance

Tests of (21) and Sulzer data show that the distance between cutwater and impeller has little effect (in the order of 5 to 10 %) on steady radial thrust. Un-steady thrust is more affected but this effect is included in the broad band of scatter given in table 4.

For double volutes an increase in radial thrust must be expected if the cutwaters of the inner and outer volute are appreciably different, see (21).

5.1.3.3 Shape of the volutes

Single volutes designed according to constant velocity (Stepanoff) or constant momentum (Pfleiderer) give nearly the same radial thrust. Other designs gave an increase in thrust by $\Delta K_R = 0.05$, ref. (22).

Differences in the shape of the outer and inner volute of a double volute casing can cause a high increase in thrust at part load.

5.2 Radial thrust in diffuser pumps

5.2.1 Design data

The mechanisms, which can cause radial thrust in a diffuser pump, were discussed in section 2. Some test data were published in ref. (11) to (15). Published and Sulzer test data do not permit to establish any relation between radial thrust coefficients and specific speed or other parameters. The available test data gathered from 12 tests over a wide range of pumps are summarized in table 3:

Table 3: Radial thrust of diffuser pumps				
Q	K_R (steady)		K_R (unsteady)	
	average	range	aver.	range
BEP	0.037	0.01-0.06	0.033	0.01-0.09
0	0.05	0.02-0.09	0.1	0.04-0.16

5.2.2 Geometric parameters

5.2.2.1 Influence of impeller tolerances

If one impeller channel has a larger outlet width than the rest of the channels or one blade has a higher outlet angle than the others the head produced by the disturbed channel is expected to be higher than by the other channels.

Consequently a hydraulic force rotating with the frequency of the shaft is expected. Fig. 7 and 8 show test results where the outlet angle of one impeller blade was increased from 36° to about 51° (test pump: nq=47, 7 impeller vanes, 9 diffuser vanes).

As expected the unsteady thrust component with the frequency of shaft rotation increased clearly over the entire load range, whereas components with other frequencies were not changed. The reduction in static thrust came rather unexpectedly.

The variation of 15° is much higher than any manufacturing tolerances would be. A first test, which was done with an increase of 4° of the impeller blade outlet angle showed virtually no influence on radial thrust. This finding is confirmed by ref. (11). From the tests so far available it appears that geometric tolerances of the impeller outlet do not contribute significantly to the (unsteady) radial thrust.

An off-set of the hydraulic from the geometric center or blade inlet angle tolerances might have a larger impact than the outlet angle tolerance.

5.2.2.2 Influence of diffuser tolerances

If one channel of a vaned diffuser differs in geometry from the other channels a static radial thrust is expected. Fig. 9 and 10 show thrust measurements where one of nine diffuser channels had a throat area reduced by about 30 % from the design value (same test pump as in previous section). As expected the steady thrust increases whereas the unsteady thrust is not much affected, except near shut-off.

5.3 Influence of cavitation on radial thrust

Data in ref.(2) and (28) showed no increase of steady or unsteady radial thrust with increasing cavitation. Tests of ref. (6) showed an increase of radial thrust only at flow rates above BEP and at high degrees of cavitation. Fig. 11 confirms this for the total (i.e. steady and unsteady) radial thrust of a diffuser pump (nq=47). It is concluded that cavitation up to 3 % head drop will not cause, in general, an increase in steady or unsteady radial thrust neither in volute nor in diffuser pumps.

5.4 Radial thrust in axial pumps

Radial thrust of axial propeller pumps is caused mainly by a non-uniform velocity distribution at the impeller inlet and/or outlet. Some test data are given in ref (15). Although these data were derived for a pump with one bend downstream of the diffuser and one bend shortly up-stream of the impeller they might be useful (in the absence of other data) to estimate the radial thrust:

Static thrust: $K_{RS}{}^* = 0.02$) for
Unsteady thrust: $K^*_{Run} = 0.01$) $Q = Qoo$

For Q > Qoo static and unsteady thrust increase. Note that K_R^* is defined differently from radial impellers:

$$K_R^* = \frac{F_R}{\rho\, g\, H\, D_2^2}$$

It is also possible to estimate the radial thrust caused by an uneven inlet velocity distribution by calculating the difference in head produced by the differences in local incidence.

5.5 Unsteady radial thrust

As with steady radial thrust it is as yet impossible to correlate unsteady radial thrust in a general fashion with geometric and operational data: Amplitudes and frequencies depend on flow separation and geometric tolerances which disturb the often delicate flow symmetry in the pump.

From our experience the following assessment can be made:

* Subsynchronous thrust components, which show broad band spectrum behaviour, amount normally to up to 20 % of the total unsteady thrust. Any clearly defined frequencies in the subsynchronous range such as reported in (13) could lead to a vibration problem in high-speed machinery.

* The synchronous thrust component is expected to amount to 10 to 20 % of the unsteady thrust.

* The high portion of blade passing frequency thrust shown in fig. 8 and

10 is rather unusual. Normally thrust components of multiples of the rotation frequency of the rotor are quite low and do not contribute much to the vibration level of the rotor if blade numbers of impeller and diffuser are selected properly.

5.6 Radial thrust design data summary

For single and double entry impellers radial thrust can be predicted according to table 4.

Table 4: Radial thrust design data			
Type of pump	$\frac{Q}{Qoo}$	Radial thrust coeff. K_R steady	unsteady
Single	0-0.5	fig. 4	0.07 0.12
volute	1.0	0.03 to 0.08	0.01 to 0.05
Double volute		figs. 4, 6 equ. 5	as single volute
Diffuser		Table 3	

Generally the measured unsteady thrust increases with increasing labyrinth clearance because of the reduced damping.

The scatter of the radial thrust data of different sources and for different pump types is caused by:

- Different hydraulic designs of different manufacturers
- Influence of labyrinth forces (type of labyrinth, length, clearance, Reynolds-number)
- Geometric tolerances of volutes, diffusers, impellers (unsteady thrust)
- Differences in testing and data reduction procedures. For example if a test is carried out with a very stiff shaft the bearing forces measured represent the true hydraulic thrust (little labyrinth forces since small deflection).

6 COMMENTS ON AXIAL THRUST IN MULTI-STAGE PUMPS

Basically there is an empirical and a theoretical approach to predict axial thrust. In practice, however, both methods are distinguished only by the degree to which they rely on experimental data.

6.1 Empirical Method

The empirical method as discussed in (23) can be characterized as follows:

- A "theoretical axial thrust" is calculated on the basis of more or less crude assumptions as to the rotative velocity of the liquid between casing and impeller and as to the static pressure rise of the impeller.
- From model or prototype tests with the hydraulic components in question the difference between actual thrust and theoretical thrust is determined and an appropriate dimensionless corrective term is derived.
- This corrective term can subsequently be used to predict the thrust of pumps of different size, speed and number of stages but with geometrically similar hydraulic components.
- The corrective term must be known as a function of Q/Qoo, Reynolds-number and labyrinth clearance.

Since the labyrinth leakage increases with Reynolds-number and the leakage has an important impact on the pressure distribution on the impeller shrouds, the influence of Re is strong. For this reason prototype tests are preferred over low-speed model tests.

If test data of high quality are available the accuracy of prediction is entirely satisfactory. For new configurations or for the investigation of the influence of individual parameters a finer calculation model is required.

6.2 Theoretical Method

The theoretical approach is based on the following procedure:

- The static pressure rise of the impeller on the shroud and hub streamline is calculated from a more or less sophisticated model (or taken from test data).
- The circumferential absolute velocity at the impeller outlet (C2u) is calculated (or estimated from the Euler equation).
- The labyrinth flow rates are calculated.
- Based on the above results the velocity and the pressure distributions on the hub and shroud are calculated as a function of geometrical data, labyrinth flow rate and direction and C2u. For this a procedure similar to (24) (25) can be followed. Since the labyrinth flow rate depends on the pressure distribution an interative process is needed.
- Finally the pressure distribution is integrated to give the forces on hub and shroud and the resultant axial thrust.

6.3 Parameters influencing Axial Thrust

To assess the merits and limitations of the methods discussed abcve the parameters influencing axial thrust are examined.

6.3.1 Main Parameters

The following parameters are correctly taken into account by both methods:

- Diameters of impeller, labyrinths including balancing device and steps of shaft subject to different pressures
- Head of pump and number of stages
- Difference between normal and last stage of a pump
- Change of momentum of flow through the impeller

6.3.2 Static Pressure Rise of Impeller

Although in general the hub and shroud streamline will be intended by design to give a constant static pressure at the impeller outlet, this condition is not necessarily always achieved neither at part load nor even at the BEP. It is obvious that a difference in static pressure between hub and shroud has a direct impact on the axial thrust. Test data have shown that a given impeller can produce a constant pressure rise over the outlet width with one diffuser but a non-uniform one with another diffuser of moderately different design. Considering the difficulties of predicting part load behaviour of pumps the above fact confirms that an accurate prediction of the static pressure at the impeller outlet is not an easy matter.

6.3.3 Pressure Distribution on Impeller Shroud and Hub

Ancient pump technology assumed K = 0.5 (K being the ratio of fluid velocity in the space between impeller and casing to the circumferential velocity of the impeller). The deviation of reality from the above assumption is the main reason for the failure to predict axial thrust in multi-stage pumps by any simple procedure: in reality on the shroud K is greater than 0.5 because of the inward leakage and on the hub K is less than 0.5 because of the outward leakage. Both deviations act in the same direction to increase the axial thrust to the suction side.

K and the resultant pressure distribution depend on a number of quantities:

- Width of space between casing and impeller shroud
- Magnitude and direction of labyrinth leakage hence Reynolds-number as well as type and clearance of labyrinth.

* On the (suction side) impeller shroud leakage is radially inward and K increases with leakage because the flow at the impeller outlet has a circular momentum (proportional to QL · C2u) which even increases when flowing inwardly.
* On the (pressure side of a normal stage) impeller hub leakage is radially outward and K decreases with increasing leakage.

- The circular component of absolute velocity C2u at impeller outlet has an influence on the velocity and pressure distribution for radially inward leakage but negligible impact if leakage is radially outward. At part load flow recirculation occurs at the impeller outlet. It starts mostly on the hub side where consequently C2u is much reduced. On the shroud side C2u increases, K increases accordingly and the axial thrust increases. The changing flow pattern at the impeller outlet at part load must therefore to a large extent be responsible for the increase of axial thrust at partload (apart from the increase in leakage rate and head) any hysteresis or other unexpected forms of the axial thrust curve.
- Roughness of impeller (K increases with increasing roughness)
- Roughness of casing (K decreases with increasing of roughness)
- Reynolds-number

From the above discussion it is inferred that accurate prediction of axial thrust by analytical models, for the time being, is only possible if supplemented by or adapted to experimental data.

6.3.4 Tolerances

The axial position of the rotor, which is known to have a significant influence on axial thrust, has an impact on both the static pressure distribution at the impeller outlet (because the impeller/diffuser position is influenced) and on the velocity and pressure distribution between casing and impeller.

The same is true for the geometric tole- rances of impeller and diffuser.

6.4 Accuracy of Thrust Prediction

Naturally one would tend to assess the accuracy of thrust prediction by comparing the calculated residual thrust to the measured one. To do this on a percentage basis can be quite meaningless: for a highly balanced multi-stage pump even a small discrepancy between predicted and measured thrust could show up as an error of 100 % or more, because the residual thrust is a small difference of two large figures (i.e. the hydraulic thrust and the balance thrust).

Any deviation between predicted and measured residual thrust should therefore be compared to the hydraulic thrust in order to get a meaningful assessment of the accuracy of the prediction, which can be used subsequently to determine the necessary thrust bearing margin.

6.5 Axial thrust tests of a boiler feed pump

Fig. 12 shows the residual thrust measured with a 4-stage boiler feed pump. The results of 5 different units lay within the scatter band of curve 1. The scatter is due to unavoidable manufacturing and assembly tolerances (presumably mainly labyrinth clearances). The predicted residual thrust for both clearances is given on fig. 12 too (prediction was based on prototype tests according to the empirical method). The width of the scatter band corresponds to about 8 % of the hydraulic thrust at BEP and to about 11 % at 25 % load.

How some tolerances do affect the different axial forces is demonstrated by table 5 for 25 % load: The error of the residual thrust is roughly 14-times greater than the error of the predicted force on the shroud (compare columns 2 and 4).

Table 5: Influence of tolerances on axial thrust

Tolerance		Increase of thrust in %		
Type	Size	Force on shroud	Imp. thrust	Residual thrust
Static pressure rise at hub/shroud streamline	7 %	11	19	50
Fluid rotation on shroud (K)	10 %	3.3	5.8	46
Circumferential component of absolute impeller outlet velocity	15 %	2.7	4.4	35

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TABLE 1

STEADY RADIAL FORCES ON CENTRIFUGAL IMPELLERS

FORCE	TYPE	MECHANISM/ORIGIN/REMARKS	Q/Qoo	$\Delta K_R = \frac{\Delta H_p}{H}$
STEADY RADIAL THRUST	D V	ROTATIONAL UN-SYMMETRIES AT IMPELLER INLET. ΔF_R INCREASES WITH Q/Qoo	≥ 0.5	0.02 TO 0.05
	D	ROTATIONAL UN-SYMMETRIES CAUSED BY PUMP OUTLET ΔF_R INCREASES WITH Q/Qoo	≥ 0.5	0.01 TO 0.02
	D	DIFFUSER TOLERANCES ΔF_R INCREASES WITH Q/Qoo	≥ 0.5	0.01
	D	PART LOAD FLOW SEPARATION IN DIFFUSER * CAUSED BY α_3-TOLERANCES (?) * THE EQUILIBRUM OF A SYSTEM OF Z_3 PARALLEL CHANNELS IS VERY SENSITIVE TO SMALL PERTURBATIONS	< 0.5	0.01 TO 0.02
	D V	STATIC EXCENTRICITY OF IMPELLER WITH RESPECT TO DIFFUSER 1)		0.01 TO 0.02
	SV	LARGE-SCALE NON-UNIFORM PRESSURE DISTRIBUTION	< 0.5	0.2 TO 0.4
	DV	OUTER VOLUTE HAS HIGHER RESISTANCE THAN INNER VOLUTE WHICH GIVES NON-UNIFORM PRESSURE DISTRIBUTION AROUND IMPELLER		0.05 TO 0.15
LABYRINTH FORCE	ALL	STEADY PRESSURE DISTRIBUTION IN EXCENTRIC LABYRINTH. FORCE: * INCREASES WITH ε_{STAT} * DECREASES WITH SERRATIONS * ε_{STAT}=RELATIVE EXCENTRICITY	ALL	- 0.02 TO - 0.05 1)
DEAD WEIGHT		TO BE CONSIDERED IN TEST DATA EVALUATION (MIND ALSO BUOYANCY)	-	-

1) ASSUMED VIBRATION AMPLITUDE: HALF LABYRINTH CLEARANCE

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