Steady Radial Forces

We now change the focus of attention back to pumps, and, more specifically, to the kinds of radial and rotordynamic forces which may be caused by the flow through and around an impeller. Unlike some of the devices discussed in the preceding sections, the flow through a pump can frequently be nonaxisymmetric and so can produce a mean radial force that can be of considerable importance. The bearings must withstand this force, and this can lead to premature bearing wear and even failure. Bearing deflection can also cause displacement of the axis of rotation of the impeller, that may, in turn, have deleterious effects upon hydraulic performance. The existence of radial forces, and attempts to evaluate them, date back to the 1930s (see Stepanoff's comment in Biheller 1965) or earlier.

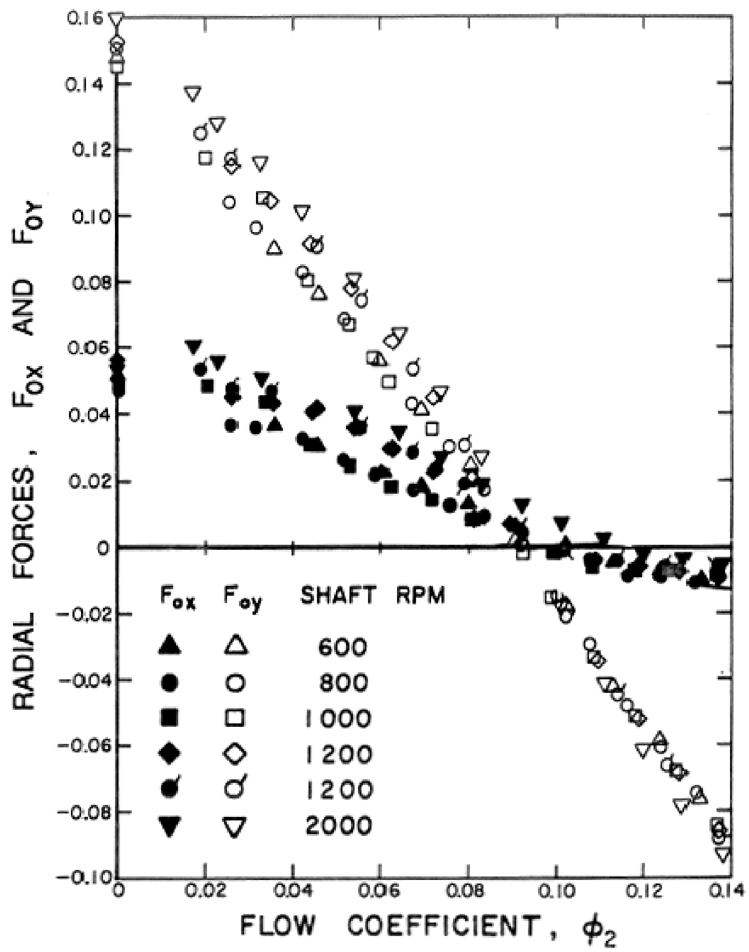

Figure 1: Radial forces for the centrifugal Impeller X/Volute A combination as a function of shaft speed and flow coefficient (Chamieh *et al.* 1985).

The nonaxisymmetries and, therefore, the radial forces depend upon the geometry of the diffuser and/or volute as well as the flow coefficient. Measurements of radial forces have been made with a number of different impeller/diffuser/volute combinations by Agostonelli *et al.* (1960), Iverson *et al.* (1960), Biheller (1965), Grabow (1964), and Chamieh *et al.* (1985), among others. Stepanoff (1957) proposed an empirical

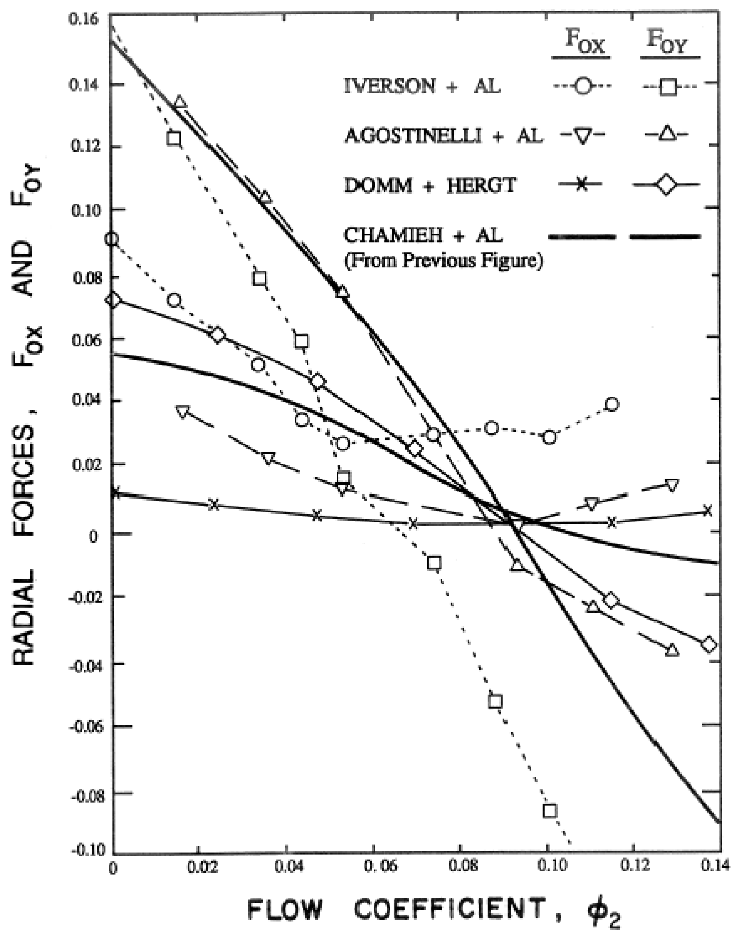

Figure 2: Comparison of the radial forces measured by Iverson, Rolling and Carlson (1960) on a pump with a specific speed, N_D, of 0.36, by Agostinelli, Nobles and Mockeridge (1960), on a pump with $N_D = 0.61$, by Domm and Hergt (1970), and by Chamieh *et al.* (1985) on a pump with $N_D = 0.57$.

formula for the magnitude of the nondimensional radial force,

$$|F_0| = (F_{0x}^2 + F_{0y}^2)^{\frac{1}{2}} = 0.229\psi \left\{1 - (Q/Q_D)^2\right\} \tag{Mcj1}$$

for centrifugal pumps with spiral volutes, and

$$|F_0| = 0.229\psi Q/Q_D \tag{Mcj2}$$

for collectors with uniform cross-sectional area. Both formulae yield radial forces that have the correct order of magnitude; however, measurements show that the forces also depend on other geometric features of the impeller and its casing.

Some typical nondimensional radial forces obtained experimentally by Chamieh *et al.* (1985) for the Impeller X/Volute A combination (see section (Mbbi)) are shown in figure 1 for a range of speeds and flow coefficients. First note that, as anticipated in the nondimensionalization, the radial forces do indeed scale with the square of the impeller speed. This implies that, at least within the range of rotational speeds used for these experiments, the Reynolds number effects on the radial forces are minimal. Second, focusing on Chamieh's data, it should be noted that the "design" objective that Volute A be well matched to Impeller X appears to be satisfied at a flow coefficient, ϕ_2, of 0.092 where the magnitude of the radial force appears to vanish.

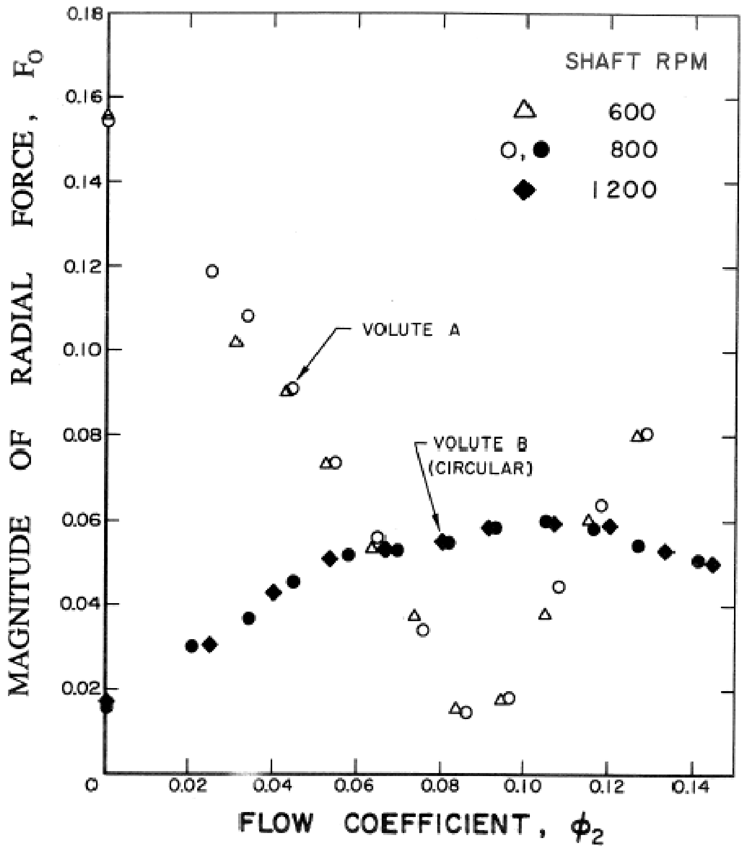

Figure 3: Comparison of the magnitude of the radial force (F_0) on Impeller X caused by Volute A and by the circular Volute B with a circumferentially uniform area (Chamieh *et al.* 1985).

Other radial force data are presented in figure 2. The centrifugal pump tested by Agostinelli, Nobles and Mockeridge (1960) had a specific speed, N_D, of 0.61, and was similar to that of Chamieh *et al.* (1985). On the other hand, the pump tested by Iversen, Rolling and Carlson (1960) had a much lower specific speed of 0.36, and the data of figure 2 indicates that their impeller/volute combination is best matched at a flow coefficient of about 0.06. The data of Domm and Hergt (1970) is for a volute similar to Volute A and, while qualitatively similar to the other data, has a significantly smaller magnitude than the other three sets of data. The reasons for this are not clear.

The dependence of the radial forces on volute geometry is illustrated in figure 3 from Chamieh *et al.* (1985) which presents a comparison of the magnitude of the force on Impeller X due to Volute A with the magnitude of the force due to a circular volute with a circumferentially uniform cross-sectional area. In theory, this second volute could only be well-matched at zero flow rate; note that the results do exhibit a minimum at shut-off. Figure 3 also illustrates one of the compromises that a designer may have to make. If the objective were to minimize the radial force at a single flow rate, then a well-designed spiral volute would be appropriate. On the other hand, if the objective were to minimize the force over a wide range of flow rates, then a quite different design, perhaps even a constant area volute, might be more effective. Of course, a comparison of the hydraulic performance would also have to be made in evaluating such design decisions. Note from figure 1, section (Mbej), that the spiral volute is hydraulically superior up to a flow coefficient of 0.10 above which the results are circular volute is superior.

As further information on the variation of the magnitude of the radial forces in different types of pump, we include figure 4, taken from KSB (1975), which shows how F_0/ψ may vary with specific speed and flow rate for a class of volute pumps. The magnitudes of the forces shown in this figure are larger than those

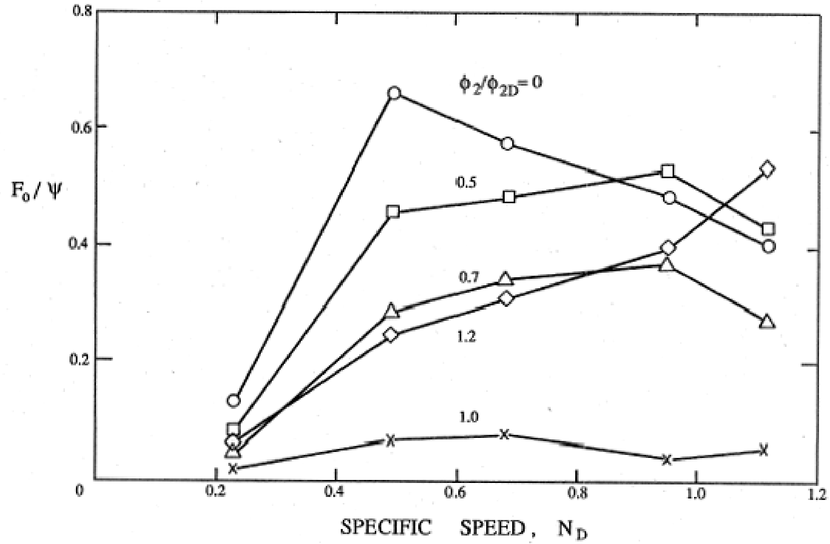

Figure 4: Variation of the radial force magnitude, F_0, divided by the head coefficient, ψ, as a function of specific speed, N_D, and flow for a class of volute casing pumps (adapted from KSB 1975).

of figure 2. We should also note that the results of Jery and Franz (1982) indicate that the presence of diffuser vanes (of typical low solidity) between the impeller discharge and the volute has relatively little effect on the radial forces.

It is also important to recognize that small changes in the location of the impeller within the volute can cause large changes in the radial forces. This gradient of forces is represented by the hydrodynamic stiffness matrix, $[K]$ (see section (Mcb)), for which data will be presented in the context of the rotordynamic coefficients. The dependence of the radial force on the impeller position also implies that, for a given impeller/volute combination at a particular flow coefficient, there exists a particular location of the axis of impeller rotation for which the radial force is zero. As an example, the locus of zero radial force positions for the Impeller X/Volute A combination is presented in figure 5. Note that this location traverses a distance of about 10% of the impeller radius as the flow rate increases from zero to a flow coefficient of 0.14.

Visualizing the centrifugal pump impeller as a control volume, one can recognize three possible contributions to the radial force. First, circumferential variation in the impeller discharge pressure (or volute pressure) will clearly result in a radial force acting on the impeller discharge area. A second contribution could be caused by the leakage flow from the impeller discharge to the inlet between the impeller shroud and the pump casing. Circumferential nonuniformity in the discharge pressure could cause circumferential nonuniformity in the pressure within this shroud-casing gap, and therefore a radial force acting on the exterior of the pump shroud. For convenience, we shall term this second contribution the leakage flow contribution. Third, a circumferential nonuniformity in the flow rate out of the impeller would imply a force due to the nonuniformity in the momentum flux out of the impeller. This potential third contribution has not been significant in any of the studies to date. Both the first two contributions appear to be important.

In order to investigate the origins of the radial forces, Adkins and Brennen (1988) (see also Brennen *et al.* 1986) made measurements of the pressure distributions in the volute, and integrated these pressures to evaluate the contribution of the discharge pressure to the radial force. Typical pressure distributions for the Impeller X/Volute A combination (with the flow separation rings of figure 7 installed) are presented in figure 6 for three different flow coefficients. Minor differences occur in the pressures measured in the front sidewall of the volute at the impeller discharge (front taps) and those in the opposite wall (back taps).

The experimental measurements in figure 8 are compared with theoretical predictions based on an analysis that matches a guided impeller flow model with a one-dimensional treatment of the flow in the

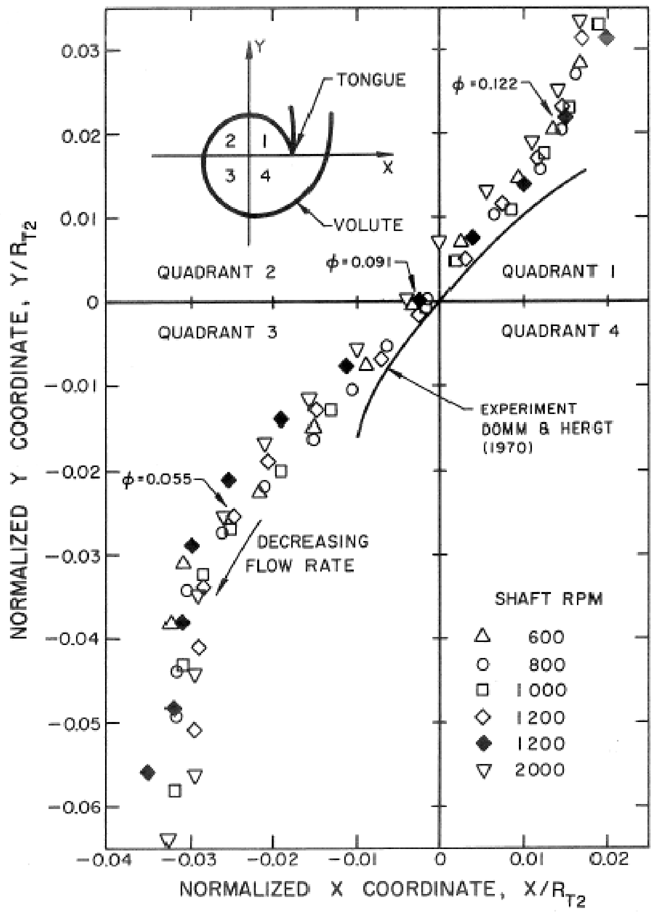

Figure 5: Locus of the zero radial force locations for the Impeller X/Volute A combination (Chamieh *et al.* 1985) compared with that from the data of Domm and Hergt (1970).

volute. This same theory was used to calculate rotordynamic matrices and coefficients presented in section (Mcm). In the present context, integration of the experimental pressure distributions yielded radial forces in good agreement with both the overall radial forces measured using the force balance and the theoretical predictions of the theory. These results demonstrate that it is primarily the circumferential nonuniformity in the pressure at the impeller discharge that generates the radial force. The theory clearly demonstrates that the momentum flux contribution is negligible.

The leakage flow from the impeller discharge, between the impeller shroud and the pump casing, and back to the pump inlet does make a significant contribution to the radial force. Figure 7 is a schematic of the impeller, volute, and casing used in the experiments of Chamieh *et al.* (1985) and Adkins and Brennen (1988), as well as for the rotordynamic measurements discussed later. Adkins and Brennen obtained data with and without the obstruction at the entrance to the leakage flow labelled “flow separation rings”. The data of figures 6 and 8 were taken with these rings installed (whereas Chamieh’s data was taken without the rings). The measurements showed that, in the absence of the rings, the nonuniformity in the impeller discharge pressure caused significant nonuniformity in the pressure in the leakage annulus, and, therefore, a significant contribution from the leakage flow to the radial force. This was not the case once the rings were installed, for the rings effectively isolated the leakage annulus from the impeller discharge nonuniformity. However, a compensating mechanism exists which causes the total radial force in the two cases to be more or less the same. The increased leakage flow without the rings tends to relieve some of the pressure nonuniformity in the impeller discharge, thus reducing the contribution from the impeller discharge pressure distribution.

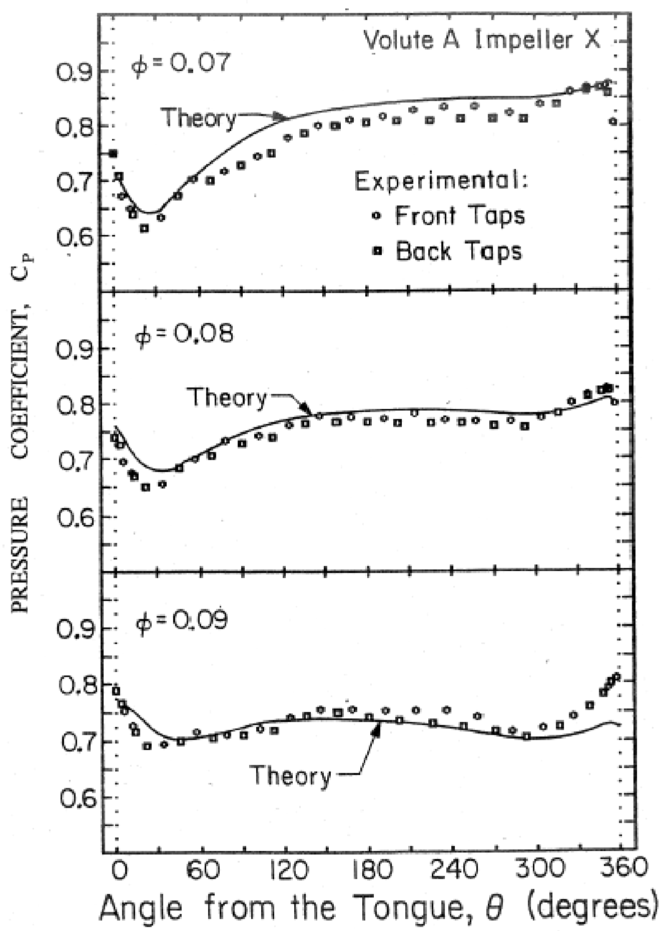

Figure 6: Circumferential pressure distributions in the impeller discharge for the Impeller X/Volute A combination at three different flow rates. Also shown are the theoretical pressure distributions of Adkins and Brennen (1988).

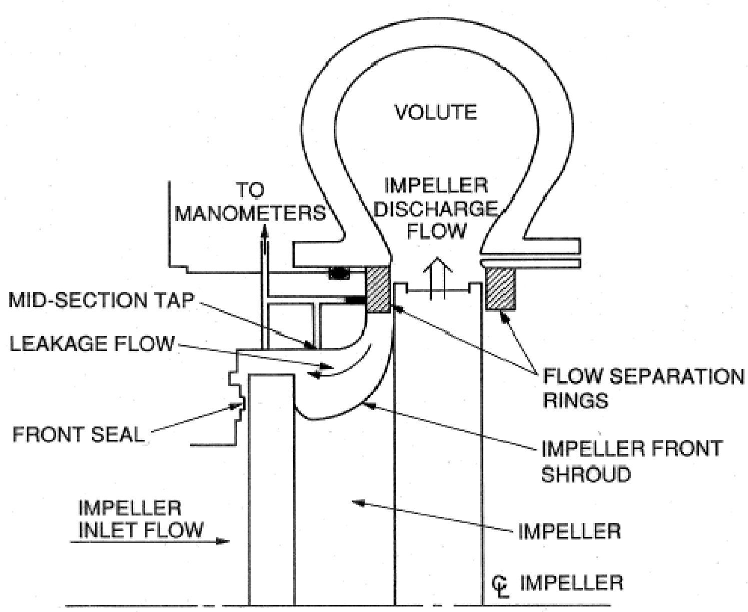

Figure 7: Schematic of the Impeller X/Volute A arrangement used for the experiments of Chamieh *et al.* (1985) and Adkins and Brennen (1988).

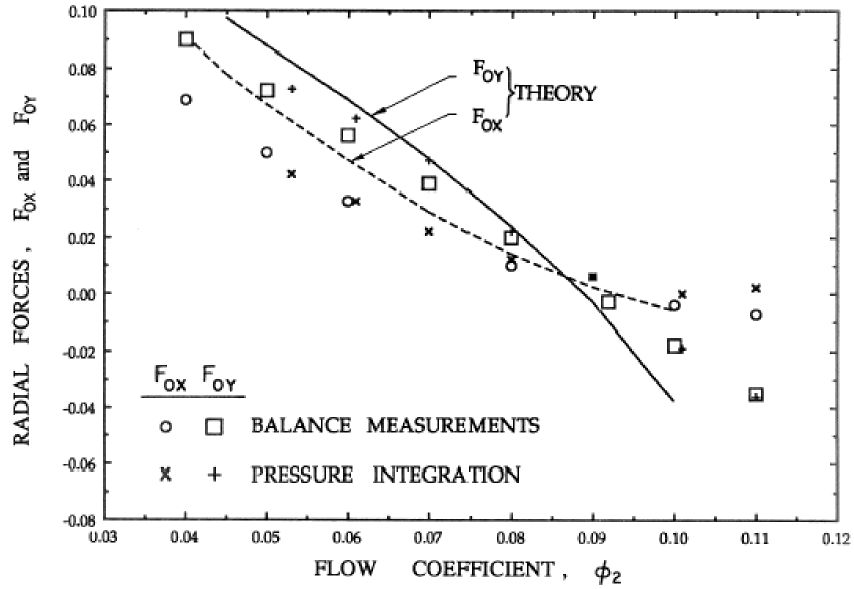

Figure 8: Comparison of radial forces from direct balance measurements, from integration of measured pressures, and from theory for the Impeller X/Volute A combination (from Adkins and Brennen 1988).

A number of other theoretical models exist in the literature. The analysis of Lorett and Gopalakrishnan (1983) is somewhat similar in spirit to that of Adkins and Brennen (1988). Earlier analyses, such as those of Domm and Hergt (1970) and Colding-Jorgensen (1979), were based on modeling the impeller by a source/vortex within the volute and solutions of the resulting potential flow. They represent too much of a departure from real flows to be of much applicability.

Finally, we note that the principal focus of this section has been on radial forces caused by circumferential nonuniformity in the discharge conditions. It must be clear that nonuniformities in the inlet flow due, for example, to bends in the suction piping are also likely to generate radial forces. As yet, such forces have not been investigated. Moreover, it seems reasonable to suggest that inlet distortion forces are more likely to be important in axial inducers or pumps than in centrifugal pumps.