Centrifugal Pumps

Rotordynamic forces in a centrifugal pump were first measured by Hergt and Krieger (1969-70), Ohashi and Shoji (1984b) and Jery *et al.* (1985). Typical data for the dimensionless normal and tangential forces, F_n and F_t, as a function of the frequency ratio, ω/Ω, are presented in figure 1 for the Impeller X/Volute A combination. The curve for Impeller X is typical of a wide range of results at different speeds, flow coefficients, and with different impellers and volutes. Perhaps the most significant feature of these results is that there exists a range of whirl frequencies for which the tangential force is whirl destabilizing. A positive F_t at negative whirl frequencies opposes the whirl motion, and is, therefore, stabilizing, and fairly strongly so since the forces can be quite large in magnitude. Similarly, at large, positive frequency ratios, the F_t is negative and is also stabilizing. However, between these two stabilizing regions, one usually finds a regime at small positive frequency ratios where F_t is positive and therefore destabilizing.

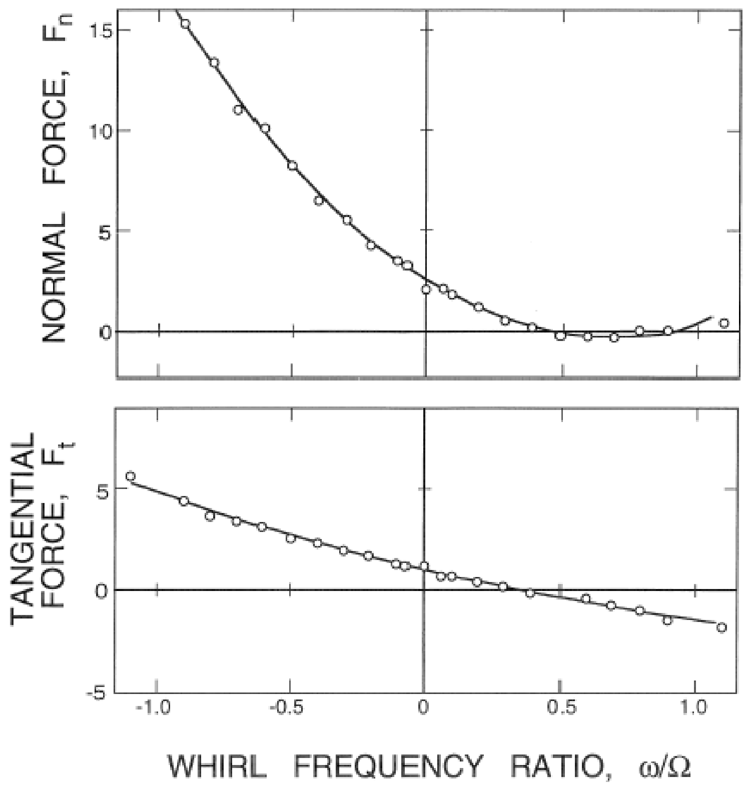

Figure 1: Typical rotordynamic forces, F_n and F_t, as a function of whirl frequency ratio, ω/Ω, for the Impeller X/Volute A combination running at 1000 *rpm* and a flow coefficient of $\phi_2 = 0.092$ (from Jery *et al.* 1985).

As is illustrated by figure 1, the variation of F_n and F_t with the whirl frequency ratio, ω/Ω, can be represented quite accurately by the quadratic expressions of equations (Mcb13) and (Mcb14) (this is not true for axial flow pumps, as will be discussed later). The rotordynamic coefficients, obtained from data like that of figure 1 for a wide variety of speeds, flow rates, and impeller, diffuser, and volute geometries, are given in table 1 (adapted from Jery *et al.* 1985). Note, first, some of the general characteristics of these coefficients. The direct stiffness, K, is always negative because of the Bernoulli effect (see section (Mcc)). The cross-coupled stiffness, k, is always positive, and is directly connected to the positive values of F_t at low positive whirl frequency ratios; consequently, k is a measure of the destabilizing effect of

Table 1: Rotordynamic coefficients for various centrifugal pump configurations (from Jery *et al.* 1985). Volute E is a 17-bladed diffuser with a spiral volute. Volutes D, F, G and H are spiral volutes fitted with zero, 6, 6 and 12 vanes respectively. Impeller Y is a 6-bladed impeller. Impeller S is a solid mass with the same external profile as Impeller X.

Impeller/Volute	rpm	ϕ_2	K	k	C	c	M	m
Imp.X/Volute A	500	0.092	-2.51	1.10	3.14	7.91	6.52	-0.52
	1000	0.092	-2.61	1.12	3.28	8.52	6.24	-0.53
	1500	0.092	-2.47	0.99	3.00	8.71	6.87	-0.87
	2000	0.092	-2.64	1.15	2.91	9.06	7.02	-0.67
Imp.X/Volute E	1000	0	-1.64	0.14	3.40	7.56	6.83	0.68
	1000	0.060	-2.76	1.02	3.74	9.53	6.92	-1.01
	1000	0.092	-2.65	1.04	3.80	8.96	6.60	-0.90
	1000	0.145	-2.44	1.16	4.11	7.93	6.20	-0.55
Imp.X/none	1000	0.060	-0.55	0.67	1.24	3.60	4.38	1.68
Imp.X/Volute D	1000	0.060	-2.86	1.12	2.81	9.34	6.43	-0.15
Imp.X/Volute F	1000	0.060	-3.40	1.36	3.64	9.51	6.24	-0.72
Imp.X/Volute G	1000	0.060	-3.34	1.30	3.42	9.11	5.75	-0.39
Imp.X/Volute H	1000	0.060	-3.42	1.33	3.75	10.34	7.24	-0.65
Imp.Y/Volute E	1000	0.092	-2.81	0.85	3.34	8.53	5.50	-0.74
Imp.S/Volute A	1000		-0.42	0.41	1.87	3.81	6.54	-0.04

the fluid. The direct damping, C, is positive, but usually less than half of the value of the cross-coupled damping, c. Note that the value of k/C is usually a fairly accurate measure of the whirl frequency ratio corresponding to the upper bound of the destabilizing interval of whirl frequency ratios. From table 1 the values of k/C, for actual impellers with volutes and with nonzero flow, range from 0.25 to 0.40, so the range of subsynchronous speeds, for which these fluid forces are destabilizing, can be quite large. Resuming the summary of the rotordynamic coefficients, note that the cross-coupled added mass, m, is small in comparison with the direct added mass, M, and can probably be neglected in many applications. Note that, since the direct added mass is converted to dimensional form by $\pi\rho R_{T2}^2 B_2$, it follows that typical values of the added mass, M, are equivalent to the mass of about six such cylinders, or about five times the volume of liquid inside the impeller.

Now examine the variations in the values of the rotordynamic coefficients in table 1. The first series of data clearly demonstrates that the nondimensionalization has satisfactorily accounted for the variation with rotational speed. Any separate effect of Reynolds number does not appear to occur within the range of speeds in these experiments. The second series in table 1 illustrates the typical variations with flow coefficient. Note that, apart from the stiffness at zero flow, the coefficients are fairly independent of the flow coefficient. The third series utilized diffusers with various numbers and geometries of vanes inside the same volute. The presence of vanes appears to cause a slight increase in the stiffness; however, the number and type of vanes do not seem to matter. Note that, in the absence of any volute or diffuser, all of the coefficients (except m) are substantially smaller. Ohashi and Shoji (1984b) made rotordynamic measurements within a much larger volute than any in table 1; consequently their results are comparable with those given in table 1 for no volute. On the other hand, Bolleter, Wyss, Welte, and Sturchler (1985, 1987) report rotordynamic coefficients very similar in magnitude to those of table 1.

The origins of the rotordynamic forces in typical centrifugal pumps have been explored by Jery *et al.* (1985) and Adkins and Brennen (1988), among others. In order to explore the effect of the discharge-to-suction leakage flow between the shroud and the casing, Jery *et al.* (1985) compared the rotordynamic forces generated by the Impeller X/Volute A combination with those generated in the same housing by

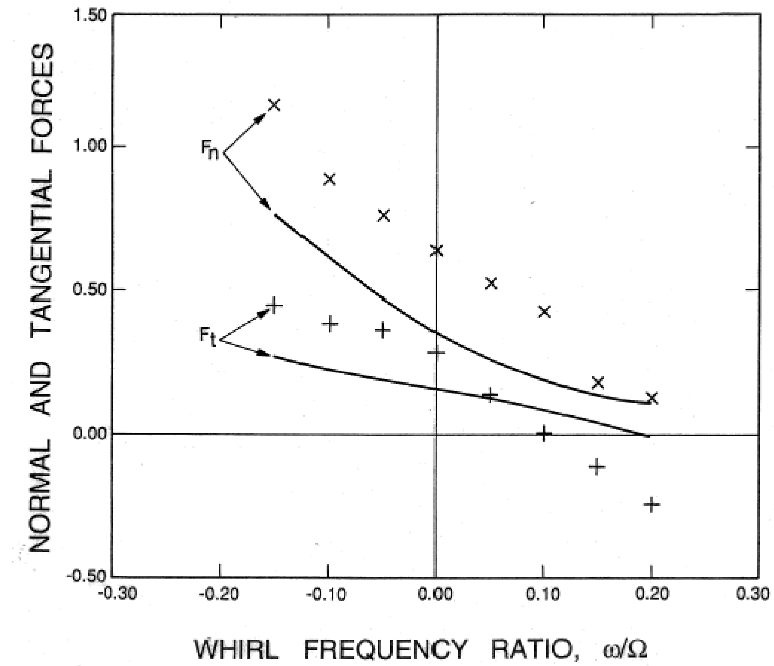

Figure 2: Comparison of the rotordynamic force contributions due to the impeller discharge pressure variations as predicted by the theory of Adkins and Brennen (1988) (solid lines) with experimental measurements using Impeller X and Volute A (at $\phi_2 = 0.092$) but with the casing surrounding the front shroud removed to mimimize the leakage flow contributions.

a dummy impeller (Impeller S) with the same exterior profile as Impeller X. A pressure difference was externally applied in order to simulate the same inlet to discharge static pressure rise, and, therefore, produce a leakage flow similar to that in the Impeller X experiments. As in the case of the radial forces, we surmise that unsteady circumferential pressure differences on the impeller discharge and in the leakage flow can both contribute to the rotordynamic forces on an impeller. As can be seen from the coefficients listed in table 1, the rotordynamic forces with the dummy impeller represented a substantial fraction of those with the actual impeller. We conclude that the contributions to the rotordynamic forces from the unsteady pressures acting on the impeller discharge and those from the unsteady pressures in the leakage flow acting on the shroud are both important and must be separately investigated and evaluated.

We focus first on the impeller discharge contribution. Adkins and Brennen (1988) used an extension of the theoretical model described briefly in section (Mcj) to evaluate the rotordynamic forces acting on the impeller discharge. They also made measurements of the forces for an Impeller X/Volute A configuration in which the pump casing structure external to the shroud was removed in order to minimize any contributions from the leakage flow. The resulting experimental and theoretical values of F_n and F_t are presented in figure 2. First note that these values are significantly smaller than those of figure 1, indicating that the impeller discharge contributions are actually smaller than those from the leakage flow. Second note that the theory of Adkins and Brennen (1988) provides a reasonable estimate of the impeller discharge contribution to the rotordynamic forces, at least within the range of whirl frequencies examined.

Using the Impeller X/Volute A configuration, Adkins and Brennen also made experimental measurements of the pressure distributions in the impeller discharge flow and in the leakage flow. These measurements allowed calculations of the stiffnesses, $K = F_n(0)$ and $k = F_t(0)$. The results indicated that the leakage flow contributes about 70% of K and about 40% of k; these fractional contributions are similar to those expected from a comparison of figures 1 and 2.

About the same time, Childs (1987) used the bulk-flow model described in section (Mcg) to evaluate the contributions to the rotordynamic forces from the discharge-to-suction leakage flow. While his results exhibit some peculiar resonances not yet observed experimentally, the general magnitude and form of

Childs results are consistent with the current conclusions. More recently, Guinzberg *et al.* (1990) have made experimental measurements for a simple leakage flow geometry that clearly confirm the importance of the rotordynamic effects caused by these flows. They also demonstrate the variations in the leakage flow contributions with the geometry of the leakage path, the leakage flow rate and the swirl in the flow at the entrance to the leakage path.

It is important to mention previous theoretical investigations of the rotordynamic forces acting on impellers. A number of the early models (Thompson 1978, Colding-Jorgensen 1979, Chamieh and Acosta 1981) considered only quasistatic perturbations from the mean flow, so that only the stiffness can be evaluated. Ohashi and Shoji (1984a) (see also Shoji and Ohashi 1980) considered two-dimensional, inviscid and unseparated flow in the impeller, and solved the unsteady flow problem by singularity methods. Near the design flow rate, their results compare well with their experimental data, but at lower flows the results diverge. More recently, Tsujimoto *et al.* (1988) have included the effects of a volute; their two-dimensional analysis yielded good agreement with the measurements by Jery *et al.* (1985) on a two-dimensional impeller.

Finally, in view of the significant effect of cavitation on the radial forces (section (Mcj)), it is rather surprising to find that the effect of cavitation on the rotordynamic forces in centrifugal pumps seems to be quite insignificant (Franz *et al.* 1990).