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Instability Problem of Turbo-Machines with Radial Distortion Problems

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***Abstract*—**In the upstream we place a piece of ring and rotate it with 83Hz, 166Hz, 333Hz,and 666H to find the effect of the periodic distortion.In the experiment this type of the perturbation will not allow since the mechanical failure of any parts of the equipment in the upstream will destroy the blade system. This type of study will be only possible by CFD. We use two pumps NS32 (ENSAM) and three blades pump (Tamagawa Univ). The benchmark computations were performed without perturbation parts, and confirm the computational results well agreement in head-flow rate. We obtained the pressure fluctuation growth rate that is representing the global instability of the turbo-system. The fluctuating torque components were 0.01Nm(5000rpm), 0.1Nm(10000rmp), 0.04Nm(20000rmp), 0.15Nm(40000rmp) respectively. Only for 10000rpm(166Hz) the output toque was random, and it implies that it creates unsteady flow by separations on the blades, and will reduce the pressure loss significantly

***Keywords*—**inlet distorsion, perturbation, turbo-machine

I. Introduction

PROVIDED the unsteady inlet distortion, we studied the response function for the centrifugal compressor and pumps. In the most design of turbo-machine, several distortion tests are conducted with different grid patterns in the suction pipe since the efficiency of the turbo-machine is quite sensitive to the inlet condition (angle). In the perturbed flows, the wake of the rotating piece is the producer of the fine structure of the distorted flow: the patterns are vortex, and the various pattern of the vortex swallowed in the suction of the turbo-machine hit the blade at random or periodically. The flow passes at the angle of attack that is different from the design. That makes the efficiency lower. In the present work we simulate the pressure and torque change by the perturbed frequency with rotating body.

II. Configuration and Computational Conditions

The pump has three and five blades, which is shown in Fig.1. In the inlet, we put a piece of ring (30degree) in the upstream. We made perturbation by rotating the piece in 84Hz, 166Hz, 333Hz, and 666Hz.

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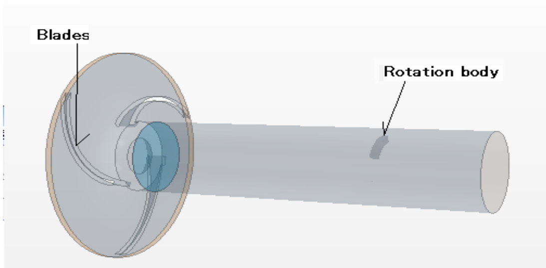

Fig.1 Configuration of Perturbation

III. Linear Transfer Function

The following linear matrix gives the property of the propagation of the angular perturbations:

$$P(t)=\mathrm{Matrix(Blade\text{-}passage)}\ A(e^{(ix/Uc)}e^{iwt})) \quad (1)$$

where A(f) is locally given by the variation of location of wake with artificial rotational speed ω , 5000rmp, 10000rpm, 20000rpm, 40000rpm, and U_c is the convective speed in the suction pipe. We use the natural perturbations instead of using the inlet velocity profile data for time variable input data: The perturbation is more natural than the vibratory boundary condition that is commonly used in CFD. In the computation, we detected the torque and the area averaged pressure fluctuation of the surface integral form as

$$P_{area}(t)= \iint p(t)\, ds$$

The time increment is from 0.001sec to 0.00033sec that is realistic for the simulation CPU time of PC.

IV. Bench mark Computational Results

Two pumps are tested for the benchmark computation to check boundary conditions and joints boundary condition and separated meshes. The mean mass balance error was 5 to 10%, and the difference between the experiment and computation (pressure) at the design point of NS32 was about 5 to 10 %.

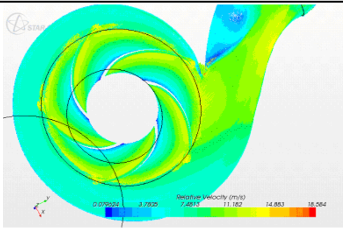

Fig2-A. NS32 ENSAM (Pump)(Undisturbed)

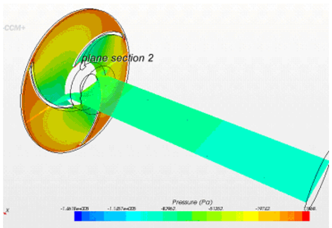

Fig.2-B. Tamagawa Univ. 3blades Pump(Un-disturbed)

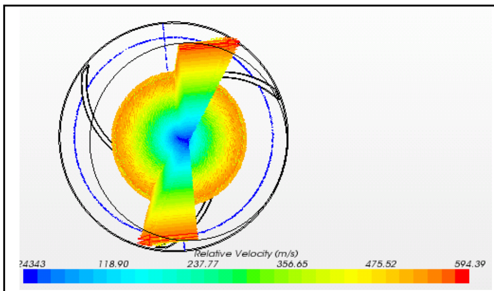

Fig.3-A. Perturbed Pattern

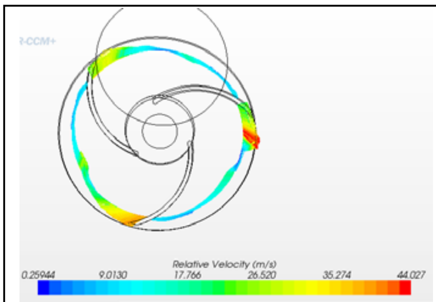

Fig. 3-B. Distortion effect can bee see velocity field F=166HZ

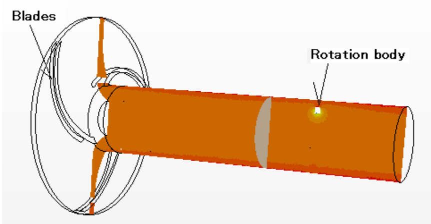

Fig. 3-C. Set rotational body(Tama3blades)

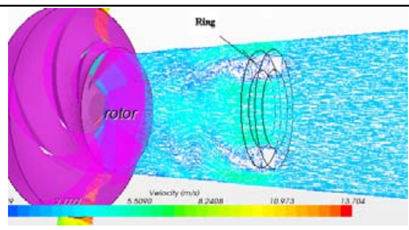

Fig.3-D. Ring upstream (Ns32)

V. PERTURBED COMPUTATIONAL RESULTS

We observed that the torque and pressure responded to the perturbation of frequency: we have observed at the outlet pressure. We observed the max response is the frequency of the perturbed frequency (166HZ, 0.006sec) near three-blade frequency (71Hz, 14sec) in Fig.4-B, where we can see the irregular frequency patterns. This means that at 10000rpm(166Hz), there is a resonance.

At 83Hz there are dips with Toque 0.01Nm. and the higher frequency at 20000rpm(333Hz, 0.003sec), the perturbed frequency is superposed on the blade frequency.

VI. CONCLUSION

At 166Hz we observed random motion in torque. This implies there is separation caused by the perturbation. There are interesting futures in the Fig.4-A, 4-C, 4-D. The perturbation frequency is superposed.

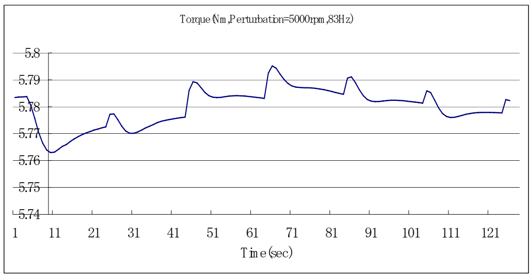

Fig.4-A. Time variation of Torque(5000rpm,Nm)

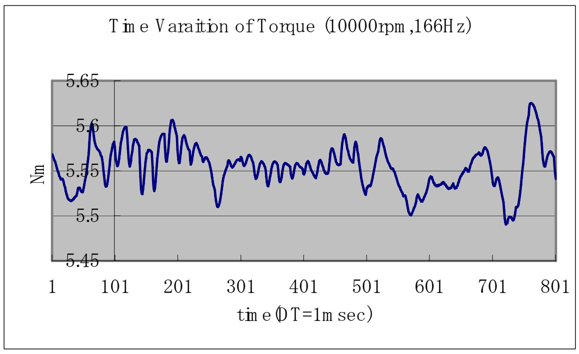

Fig.4-B Time variation of Torque(10000rpm)

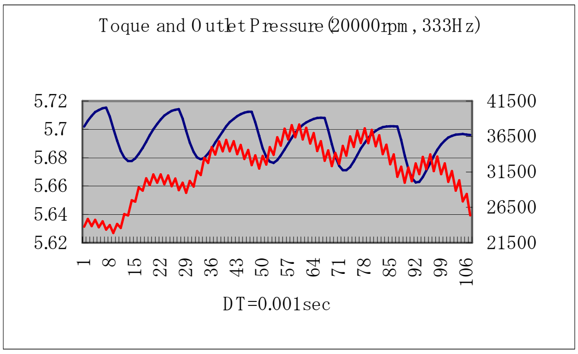

Fig.4-C Time variation of Torque and Pressure 20000rpm

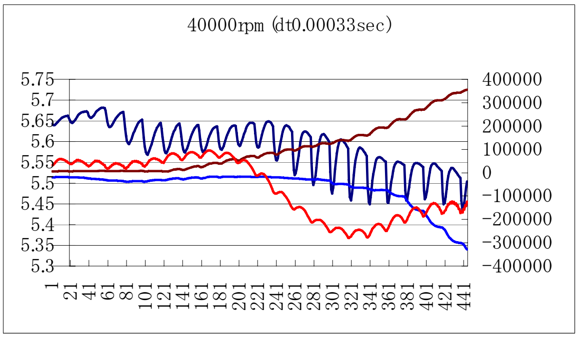

Fig.4-D Time variation of Torque and Pressure

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