Dynamic behaviour of pump-turbine runner: From disk to prototype runner

X X Huang, E Egusquiza, C Valero and A Presas

Center for Industrial Diagnostics and Fluid Dynamics (CDIF), Universitat Politècnica de Catalunya, ETSEIB Building, Av. Diagonal 647 –08028 Barcelona (Spain)

E-mail: holaxxh@hotmail.com

Abstract. In recent decades, in order to increase output power of hydroelectric turbomachinery, the design head and the flow rate of the hydraulic turbines have been increased greatly. This has led to serious vibratory problems. The pump-turbines have to work at various operation conditions to satisfy the requirements of the power grid. However, larger hydraulic forces will result in high vibration levels on the turbines, especially, when the machines operate at off-design conditions. Due to the economic considerations, the pump-turbines are built as light as possible, which will change the dynamic response of the structures. According to industrial cases, the fatigue damage of the pump-turbine runner induced by hydraulic dynamic forces usually happens on the outer edge of the crown, which is near the leading edges of blades. To better understand the reasons for this kind of fatigue, it is extremely important to investigate the dynamic response behaviour of the hydraulic turbine, especially the runner, by experimental measurement and numerical simulation. The pump-turbine runner has a similar dynamic response behaviour of the circular disk. Therefore, in this paper the dynamic response analyses for circular disks with different dimensions and disk-blades-disk structures were carried out to better understand the fundamental dynamic behaviour for the complex turbomachinery. The influences of the pattern and number of blades were discussed in detail.

1. Introduction

In response to the oil crisis and global climate change, hydropower, as a renewable clean energy, is widely adopted in recent decades.

Hydraulic pumped-storage power plants, the largest-capacity form of grid energy storage now available, can store energy in the form of water by pumping water from a lower elevation reservoir to a higher elevation, with the excess generation capacity during periods of low electrical demand. At time of peak demand, the stored water is released through turbines to follow variations of the demand. In addition to balancing the load of the electric power system, pumped-storage power plants can also used for phase modulation, frequency modulation and incident spare to improve power quality, enhance power system stability and bring a good economic benefit at the same time.

The pump turbines have been well investigated and designed by the manufactories to achieve larger output power. With this tendency to raise the power concentration of the hydraulic unit, the design head, the fluid velocities and the corresponding hydraulic excitation forces have ascended remarkably. Especially, the machine, operating at the off-design conditions, will suffer high vibration levels caused by large hydraulic forces.
The dynamic responses of pump-turbines under the rising excitation forces, will affect seriously the stable operations of the hydroelectric units and the safety of power plants.

For the hydraulic turbines, when the frequencies range of the pressure pulsation caused by rotor-stator interaction (RSI) overlaps with same natural frequencies of the runner, and especially the vibratory mode of the excitation is the same as that of the runner, stronger vibrations and fatigue cracks will be induced on the turbines. Therefore, a good runner design should avoid the resonance.

In recent years, it is not uncommon that large fatigue cracks appear on some prototype pump-turbine runners [1,2].

Plenty of papers were focused on the dynamic response of Francis runners [3-5]. Nevertheless, the reversible pump-turbine runner has a quite different design and a more complex geometry than the Francis turbine runner.

In order to avoid fatigue damages on the pump-turbine runners, it is of prime importance to investigate the dynamic behavior of prototype runners. To better understand the fundamental dynamic response of these complex structures, the dynamic response analyses of geometrically similar disk-blades-disk structures should be carried out. And the influences of the pattern and number of blades on the dynamic response of the runner can also be investigated.

2. Dynamic response analysis of pump-turbine runners

2.1 Nodal diameter (ND) and nodal circle (NC)
Hydraulic turbine runner is a typical cyclically symmetric structure which is constructed by one basic sector around an axis. For such structure, it is well known that vibration mode can be characterized according to the number of its nodal diameters and nodal circles. A nodal diameter is a line that bisects the circle across the diameter and a nodal circle is a circumferential line. The mode with d ND and c NC can be denoted as mode (d, c).

2.2 Dynamic response analysis of circular disks
A copper circular disk (figure 1) was selected to conduct the modal analysis. The thickness, inner and outer radius of the disk are 3mm, 13mm and 72.5mm respectively. The density of this disk is 8300 kg·m$^{-3}$, the Young’s Modulus is 110 GPa, and the Poisson’s ratio is 0.34. The disk was treated as a free body in the simulation.

![Figure 1. Geometry and finite element model of the circular disk.](image)

The disk was suspended with flexible ropes in the measurement. The ropes have no influence on the modal behaviour of the disk, since their natural frequencies are much lower than those of the disk. The rowing hammer technique has been used for the measurements. The hammer has been moved around the structure to excite the DOFs needed to well characterize the dynamic behaviour meanwhile the accelerometers have been fixed to the structure. The numerical simulation of the circular disk can be used to identify the frequency band of interest and the regions with largest deformations for the expected mode-shapes. One piezoelectric low mass accelerometers was located on the circumference of the disk to measure the impulsive force excitations, and the selected 80 points were impacted by an instrumented force hammer. The signals of the impulse excitation and the response were acquired and recorded. The averaged FRFs for all the excitation positions have been calculated in order to extract the modal parameters.
Taking into account of repeated roots for the cyclically symmetric structure, the mean value of natural frequencies is adopted to describe the simulation and experiment results. Frequency deviation ratio (FDR) $\Delta$ is used to check the accuracy of the result between Sim. ($R_{\text{Sim.}}$) and Exp. ($R_{\text{Exp.}}$). The value of $\Delta(\%)$ is calculated by the following equation:

$$\Delta(\%) = 100 \times \left(\frac{R_{\text{Sim.}} - R_{\text{Exp.}}}{R_{\text{Exp.}}}\right).$$  \hspace{1cm} (1)

The natural frequencies and FDR are shown in table 1. The FDR values of all the obtained mode-shapes remain in a range of ±2.5%.

**Table 1.** The natural frequencies and FDR of the disk.

| Mode-shape | Sim.   | Exp.   | $\Delta(\%)$ |
|------------|--------|--------|--------------|
| (2,0)      | 1031.55| 1050   | -1.76        |
| (0,0)      | 1747.60| 1790   | -2.37        |
| (3,0)      | 2434.15| 2440   | -0.24        |
| (1,1)      | 4012.05| 3985   | 0.68         |
| (4,0)      | 4252.30| 4210   | 1.00         |
| (5,0)      | 6481.85| 6360   | 1.92         |
| (2,1)      | 6796.90| 6780   | 0.25         |
| (0,1)      | 7671.20| 7730   | -0.76        |

Figure 2 illustrates the mode-shapes estimated by simulation and experiment, which serve to characterize the vibration modes. For all the modes, the experimental and numerical results agree with each other very well, which validates the simulation method.

![Mode-shapes](image)

**Figure 2.** Mode-shapes of the disk obtained by Sim. and Exp.

With the same material parameters, dynamic response analyses of 9 disks with different radius and thickness (table 2) was performed. The natural frequencies for the corresponding mode-shape of the disks are shown in figure 3.

**Table 2.** The dimensions of the disks.

| (mm) | No.1 | No.2 | No.3 | No.4 | No.5 | No.6 | No.7 | No.8 | No.9 |
|------|------|------|------|------|------|------|------|------|------|
| h    | 3    | 6    | 12   | 3    | 6    | 12   | 3    | 6    | 12   |
| r    | 6.5  | 6.5  | 6.5  | 13   | 13   | 13   | 26   | 26   | 26   |
| R    | 36.25| 36.25| 36.25| 72.5 | 72.5 | 72.5 | 26   | 26   | 26   |
Figure 3. Natural frequencies of the investigated disks.

The corresponding mode-shapes of different disks are similar with large deformation in radial direction. For any specified mode-shape, the natural frequencies for the disks with the same radius dimensions increase with the increasing thickness, and the corresponding frequencies increase almost twice with the doubled thickness. The natural frequencies for the disks with the same thickness decrease with the increasing radius dimensions, and the corresponding frequencies are reduced to a quarter of the former one when the radius dimension is doubled.

2.3 Dynamic response analysis of model and prototype pump-turbine runner

With the same procedure, the dynamic response analysis of the scaled model runner and prototype runner were carried out by experiment and simulation. The obtained mode-shapes are illustrated in figure 4. The FDR and the corresponding natural frequencies of modal runner obtained from simulation and experiment are listed in table 3. The experimental and numerical results have a rather good agreement. The first several modes of the model and prototype runners give the same mode-shapes as the disks.

Figure 4. Mode-shapes of the modal runner(Up) and the prototype runner (down).

Table 3. Natural frequencies and FDR of the modal runner in air.[6-9]

| Mode-shape | Model runner | Prototype runner | Δ(%) | Model runner | Prototype runner | Δ(%) |
|------------|--------------|------------------|------|--------------|------------------|------|
| (2,0)      | 825.26       | 838.00           | -1.52| 250.93       | 250.41           | -0.21|
| (0,0)      | 1247.40      | 1111.00          | 12.28| 320.96       | 337.04           | 5.01 |
| (3,0)      | 1479.75      | 1463.00          | 1.14 | 435.65       | 439.89           | 0.97 |
| (1,1)      | 1605.55      | 1439.00          | 11.57| 424.30       | 431.81           | 1.77 |
3. Dynamic response analysis disk-blades-disk structures

In order to gain a better understanding of the complicated dynamic response of pump-turbine runners, it is better to investigate the corresponding simplified models. Since the pump-turbine runner is consist of three parts: crown, blades and band, it can be simplified as disk-blades-disk structure.

Dynamic response analyses of disk-blades-disk structures were performed to understand the influences of the basic dimensions (such as shape and number of blades) on the dynamic response the structures. The investigated disk-blades-disk structures consist of a couple of circular disks and several curved or straight blades (figure 5). The circular disks have the same dimensions as the previously described disk No.5. The height and width of the blades are 13.5mm and 3mm.

![Geometries of disk-blades-disk structures.](image)

Figure 5. Geometries of disk-blades-disk structures.

The modal analysis of the disk-blades-disk structures with 7-9 curved and straight blades was carried out. The natural frequencies of the same mode-shapes are listed in table 4.

| Mode-shape | Curved blades | Natural frequencies | Straight blades |
|------------|---------------|---------------------|-----------------|
|            | 6 blades  | 7 blades | 8 blades | 9 blades | 6 blades | 7 blades | 8 blades | 9 blades |
| (2,0)      | 2633.70   | 2754.60 | 2850.05 | 2931.40 | 2223.70 | 2330.50 | 2409.30 | 2471.90 |
| (0,0)      | 3289.50   | 3533.70 | 3740.80 | 3933.90 | 4790.90 | 5226.50 | 5542.10 | 5776.30 |
| (3,0)      | 4542.80   | 4668.90 | 4939.15 | 5151.30 | 3698.00 | 3893.50 | 4132.00 | 4304.50 |
| (1,1)      | 5032.60   | 5235.50 | 5414.75 | 5578.40 | 4624.00 | 4830.10 | 5010.30 | 5173.50 |

Since the corresponding mode-shapes of the structures with different number of blades are similar, the mode-shape of the disk-blades-disk with 9 curved and straight blades are shown in figure 6.

![Mode-shapes of the disk-blades-disk structures with curved blades (up) and straight blades (down).](image)

Figure 6. Mode-shapes of the disk-blades-disk structures with curved blades (up) and straight blades (down).
Although the number and the shape of the blades are changed, the disk-blades-disk systems give the same mode-shapes. The mode-shapes like (2,0), (0,0), (3,0) and (1,1) can be found for the structures with curved and straight blades, but the mode-shape (0,0) for the structure with straight blades swaps the positions with (3,0) and (1,1).

Various number and shape of the blades will modify the mass and stiffness of the structure, therefore the corresponding natural frequencies of the disk-blades-disk structures will be changed. For the same mode-shape, the more blades the structure has, the higher natural frequency is. For mode-shapes (2,0), (3,0), (1,1), the structure with straight blades, compared with the one with the same number of curved blades, has lower natural frequencies. It is opposite for the mode-shape (0,0).

4. Conclusions
The dynamic response behaviours of the structures, from disk to prototype type pump-turbine runner, have been analyzed numerically and experimentally. The comparison between the experiment and simulation gives a good agreement.

The model and prototype runners give the same mode-shape as the disks and disk-blades-disk structures. The vibration modes represent the typical mode-shapes for the cyclic geometry (2,0), (0,0), (3,0), (1,1) etc.

For the specified mode-shape of the disks, the natural frequencies increase with the increasing thickness and decreasing radius.

The number and shape of the blades will not change the mode-shapes of the disk-blades-disk structures but change the natural frequencies. For the specified mode-shape, the structure with more blades has a higher natural frequency. The shape of blades can increase or decrease the natural frequencies of the structure depending on the mode-shapes.

So the designer can change the number or the shape of the blades to modify the natural frequencies and/or the exciting mode of the runner. From the industrial point of view, the modal parameters obtained from these simplified models can be adopted to speed-up the design process and build-up a realistic turbine with enough accuracy to decrease the vibration levels and avoid resonance and fatigue problems.

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