

# 低比转数混流泵导叶内部压力脉动特性研究

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**摘要:** 对一比转数为 148.8 的设计混流泵进行试验和数值模拟研究, 比较不同流量工况下混流泵性能的试验与数值计算结果, 两者吻合较好。在流场内部设置监测点, 捕捉压力脉动由动静干涉无叶区向导叶出口的发展过程。分析不同工况下的混流泵各测点的压力脉动, 发现导叶内部各测点压力脉动主要受叶轮转动影响, 主频为叶频; 由动静干涉的无叶区到导叶出口, 平均压力逐渐增大, 而压力脉动的幅值强度越来越弱; 非设计工况下的压力脉动变化更复杂。上述结果可为混流泵进一步的优化设计提供参考。

**关键词:** 低比转数混流泵; 叶片扩压器; 压力脉动; 数值模拟

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## Internal Pressure Fluctuation Characteristic of Low Specific Speed Mixed Flow Pump

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**Abstract:** Experiment and numerical simulation research were conducted on a designed mixed flow pump with the low specific speed of 148.8. By comparing the mixed-flow pump performances at different flow rates, the numerical results and the test results were in good agreement. Monitoring points in the internal flow field were set to capture the pressure fluctuations from rotor-stator interface to the guide vane exit in both the experiment and the numerical simulation. Through analyzing the pressure fluctuations of mixed flow pump at the different flow rates, the pressure fluctuations in guide vane were mainly affected by impeller rotation, and the frequency was impeller blade frequency. From rotor-stator interface to the guide vane exit, the average pressure was increased gradually, and the amplitude of pressure fluctuations gradually became weaker. At the off-design points, the pressure fluctuations became more complex. At low flow rate, there were many low frequencies of the vortex fluctuation with low amplitude, which was consistent with flow field in the test. At high flow rate, the main frequency was also the impeller blade frequency, and there were some low frequencies of the vortex fluctuation with low amplitude, which were better than the low flow rate. These results can be useful for further optimization design of mixed flow pump.

**Key words:** low specific speed mixed flow pump; vane diffuser; pressure fluctuation; numerical simulation

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## 引言

导叶式混流泵兼有离心泵和轴流泵二者的优点,功率曲线较平、高效区较宽<sup>[1]</sup>。导叶是导叶式混流泵内部重要的组成部分,对机组整体性能、动静平衡、气蚀和噪声等有直接影响,因此研究混流泵导叶内部的非定常流动特性对混流泵导叶的设计,提高混流泵性能有着重要的指导意义。

以提高混流泵综合性能、稳定性和可靠性为目的,国内外学者对混流泵导叶设计、优化及内部流动分析开展了深入的研究。ARNDT等<sup>[2]</sup>通过试验研究了混流泵转轮与导叶间动静干涉下的压力脉动情况;席光等<sup>[3]</sup>设计了一半开式叶轮混流泵试验装置,采用PIV技术对叶轮内部流动进行了测量;郇浩等<sup>[4-6]</sup>给出了混流泵的正反问题设计方法,并设计了一系列不同导叶和轴面流道形状的混流泵,利用数值方法分析不同设计参数对混流泵性能的影响;MASAHIRO等<sup>[7]</sup>对低比转数导叶式混流泵进行了PIV测试,通过分析压力脉动研究了旋转失速的传播机理,认为从叶片扩压器向叶轮出口的大尺度突然回流是产生非定常流动的原因,且回流是扩压器进口处的强涡引起的;SANKAR等<sup>[8]</sup>在泵壁上靠近叶轮进口处开设“J”型槽来控制抑制流场非定常流动,分析了带有不同形状和数量射流槽的混流泵性能曲线以及有无射流槽时叶轮进口和无叶区非定常压力脉动特性;FERNANDEZ等<sup>[9-17]</sup>利用不同数值方法研究了混流泵和轴流泵叶轮和导叶内部水压力脉动性能,叶轮和导叶进口速度分布规律,得到了不同工况不同结构下叶栅区的压力脉动特性,采用反问题优化设计混流泵,双向流固耦合分析混流泵叶轮力学特性。

本文对一低比转数设计混流泵进行试验和数值模拟,并对比数值计算性能曲线与试验性能曲线的吻合情况。同时对不同工况下导叶内部不同测点压力脉动进行监测,比较分析试验和数值结果中各测点的频谱特性,以揭示导叶内部水压力脉动规律。

## 1 试验装置

本文研究的混流泵由日本久保田公司设计,其比转数为148.8,几何参数和设计工况性能参数为:转轮直径 $D = 150 \text{ mm}$ ,设计流量 $Q_d = 1.39 \text{ m}^3/\text{min}$ ,设计扬程 $H = 14.59 \text{ m}$ ,设计转速 $n = 2\,000 \text{ r/min}$ 。混流泵的测试试验在哈尔滨大电机研究所完成,图1为试验台,图2为混流泵试验图,图3为试验压力脉动监测点的具体位置。



图1 混流泵试验装置

Fig. 1 Photo of mixed flow pump test apparatus



图2 混流泵试验图

Fig. 2 Photo of mixed flow pump test model

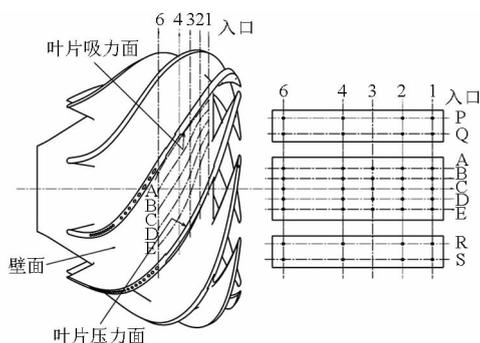


图3 压力脉动监测点位置分布

Fig. 3 Distribution of monitoring points in test

## 2 数学模型和网格

根据设计尺寸,采用UG NX 6.0对混流泵全流道进行了三维建模,几何模型如图4所示。



图4 混流泵全流道三维模型

Fig. 4 3D configuration of the mixed flow pump

计算域分成4部分,分别为进口管、叶轮区、导叶区、出口管,采用TurboGrid和ICEM将4个计算域划分为结构化网格,叶轮和导叶壁面网格的 $y^+ = 20 \sim 30$ ,其中 $y^+$ 表示无量纲壁面距离。叶轮和导叶网格划分如图5所示。采用定常数值模拟对网格数做无关性检验,最终选取约398万全流道计算网格

数进行非定常数值模拟。



图5 叶轮与导叶区的计算网格

Fig.5 Computational meshes for impeller and guide vanes

本文运用 ANSYS CFX 对该混流泵全流道进行数值模拟。应用全隐式耦合方法求解方程组,本文选择 SST  $k-\omega$  模型来封闭方程组,进口给定均匀流速,出口流动充分发展,近壁区采用标准壁面函数、无滑移条件。

为研究混流泵导叶内部压力脉动特性,结合试验监测点,在数值计算中设置了多个监测点,如图6所示。除导叶流道内,还在无叶区和导叶出口处设置监测点。

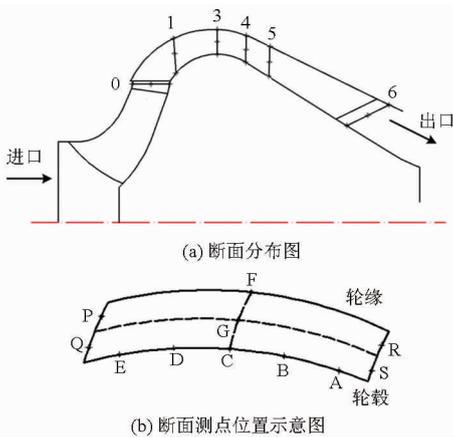


图6 数值模拟监测点分布图

Fig.6 Distribution of monitoring points in computation

### 3 试验与数值结果分析

#### 3.1 混流泵性能分析

试验测试了设计转速下不同流量的混流泵性能,从中选取了8个流量工况点进行数值模拟。

图7为混流泵试验与数值计算的性能对比曲线图,图中  $Q/Q_d$  表示计算流量与设计流量的比值。从图中可以看出,功率、扬程和效率数值计算与试验性能曲线基本吻合,并且设计工况为最优工况(效率最高),仅在大流量工况下,效率数值计算值比效率试验值稍高。对比结果证明了采用本文数值方法来预测混流泵性能的可靠性。

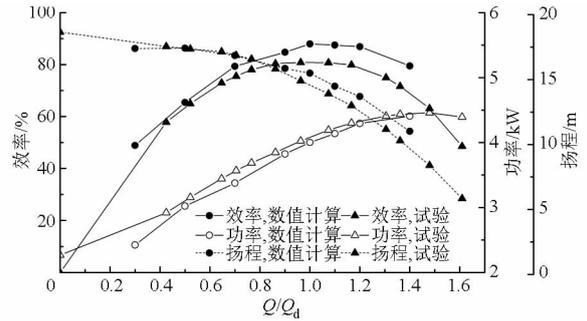


图7 数值与试验结果的对比

Fig.7 Comparison of numerical simulation and experiment results

#### 3.2 导叶段压力脉动

试验中对不同转速下不同流量工况进行了导叶段压力脉动监测,压力与电信号的转换公式为

$$p = 10^{-6} \alpha \Delta e \quad (1)$$

式中  $p$ ——压力

$\alpha$ ——校正系数,取值为  $1.88 \times 10^5$

$\Delta e$ ——电信号输出值

采用97%的置信度对最后5个周期进行分析,表1给出部分工况部分监测点的压力脉动结果,其中C1、C3和C5分别表示1断面、3断面和5断面轮毂上的监测点C。从导叶进口到出口段压力均值逐渐增大,压力脉动的幅值逐渐减小。

对设计转速下  $0.5Q_d$ 、 $0.75Q_d$ 、 $Q_d$  和  $1.25Q_d$  4个不同流量工况进行非定常数值模拟,并与试验结果比较。采用计算10个周期的定常计算结果为初始值,时间步长为  $0.0002\text{ s}$ ,计算总时长为15个周期。从数值结果看,当计算5个周期后压力脉动的周期性比较明显,同样采用97%的置信度对最后5个周期进行分析。压力脉动系数为

表1 压力脉动部分试验结果分析

Tab.1 Experiment results analysis of pressure fluctuation at mixed flow pump vane

工况		C1		C3		C5	
转速/( $\text{r} \cdot \text{min}^{-1}$ )	流量/( $\text{kg} \cdot \text{s}^{-1}$ )	压力均值/kPa	峰峰值/kPa	压力均值/kPa	峰峰值/kPa	压力均值/kPa	峰峰值/kPa
1200	7.38	85.49	8.27	138.83	4.51	143.09	5.26
	14.55	77.05	5.26	135.84	3.95	139.46	3.57
1500	9.19	105.34	10.90	146.67	6.01	151.22	7.89
	18.74	90.86	6.77	141.44	4.70	144.97	4.70
2000	12.19	147.93	22.75	163.76	9.02	168.64	13.72
	24.63	123.16	9.40	154.63	6.21	157.76	8.46

$$C_p = \frac{\bar{p} - \bar{p}}{\rho g H} \times 100\% \quad (2)$$

式中  $\bar{p}$ ——平均压力, Pa      $\rho$ ——密度,  $\text{kg}/\text{m}^3$   
 $g$ ——重力加速度,  $\text{m}/\text{s}^2$   
 $H$ ——水泵扬程, m

表2给出C1和C3监测点压力脉动系数的数值计算与试验结果对比。从表中看出,数值计算与试验结果基本一致,从导叶进口段到出口段,随着压力增加,压力脉动系数的幅值逐渐减小。导叶进口处的压力脉动幅值最大,在优化设计时需要注意导

表2 混流泵导叶段压力脉动系数

Tab.2 Pressure fluctuation coefficient at mixed

工况	C1		C3	
	试验	数值	试验	数值
0.5 $Q_d$	8.90	12.3	6.97	9.22
0.75 $Q_d$	-	11.5	-	11.4
$Q_d$	3.53	5.28	2.68	4.17
1.25 $Q_d$	-	9.13	-	6.69

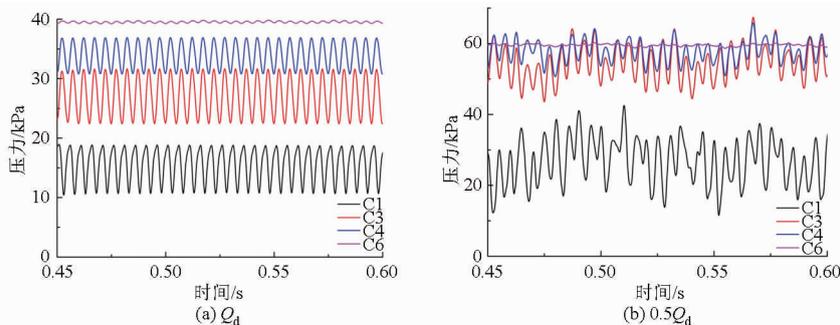


图8 不同监测点的压力脉动时域图

Fig.8 Time-domain figures of pressure fluctuation at different monitoring points

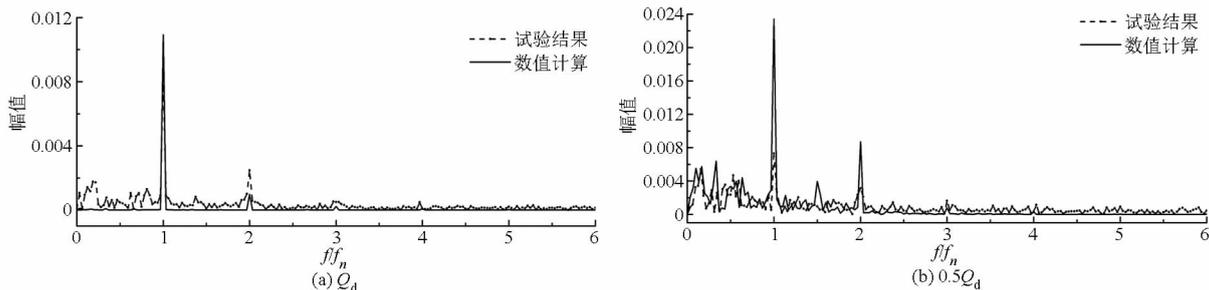


图9 试验与数值计算压力脉动对比

Fig.9 Comparison of experimental and numerical pressure fluctuations

## 4 结论

(1)该比转数为148.8的设计混流泵综合性能的数值计算与试验结果吻合较好,数值模拟可以为下一步优化设计提供参考。

(2)该混流泵在设计流量工况下,导叶内部各测点压力脉动主要受叶轮转动影响,主频为叶频。且由导叶进口到出口,平均压力增加,但压力脉动幅值减小。数值计算与试验结果一致。

叶的进口安放角;设计工况的压力脉动系数幅值最小;小流量工况随流量减小压力脉动增加;大流量工况随流量增加压力脉动增加。

图8给出了给出了 $Q_d$ 和 $0.5Q_d$ 工况下,不同监测点的压力脉动时域图。从图中可以看出: $Q_d$ 工况下各监测断面的压力脉动周期变化很好,从导叶进口到出口,平均压力增大,压力脉动幅值减小; $0.5Q_d$ 工况下各监测断面的压力脉动变化相对复杂,从导叶进口到出口,平均压力增大,导叶进口的压力脉动幅值最大,出口的压力幅值最小。对比图8a、8b,同一监测点在 $0.5Q_d$ 工况下的压力脉动幅值更大。

图9给出监测点C1在 $Q_d$ 和 $0.5Q_d$ 工况下,数值计算和试验结果的压力脉动系数FFT分析图,图中 $f/f_n$ 表示压力脉动频率与叶频的比值。从图中可以看出,数值计算和试验结果的主频一致,为1倍叶频。 $Q_d$ 工况下的幅值基本吻合,而 $0.5Q_d$ 工况下数值结果偏大,但低频幅值与试验结果吻合。

(3)小流量工况下,由导叶进口到出口,平均压力增加,但压力脉动变化相对复杂。各测点的主频与设计工况下相同,但出现了很多低频率低幅值的涡频,流量越小流场越不稳定,这与试验中流场内部出现很多细小的涡团相符。

(4)大流量工况下,各测点的主频也与设计工况下相同,但低频率的幅值较小,流动不稳定性也增加,但相比小流量工况要好。

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