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后置灯泡贯流泵压力脉动特性数值模拟

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摘要:为研究不同工况下某南水北调泵站后置灯泡贯流泵叶轮导叶压力脉动规律,通过计算流体动力学(computational fluid dynamics,CFD)对偏流量、偏水位工况叶轮导叶区压力脉动进行计算与分析,结果表明:叶轮导叶区压力脉动时域图周期性明显,叶轮叶片个数对压力脉动主次频有一定影响,叶轮导叶区主次频均为整倍数叶频,叶轮导叶区压力脉动幅值整体从轮缘到轮毂呈减小趋势,叶轮区的压力脉动幅值明显大于导叶区。非设计水位工况下叶轮导叶区压力脉动幅值略大于设计工况,主次频未发生明显变化;非设计流量工况中小流量工况与大流量工况压力脉动幅值均大于设计工况,各个监测点的小流量工况压力脉动幅值为设计工况的 2~3 倍,且在此工况下低频脉动明显。可见非设计工况运行对机组压力脉动幅值的影响较大,长期在非设计工况下运行严重影响机组运行效率,泵站运行应尽量避免非设计工况运行的情况。研究结论可为泵站日常运维和研究异常水力振动提参考。

关键词:泵站;灯泡贯流泵;压力脉动;数值模拟;快速傅里叶变换

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灯泡贯流泵以效率高、流量大、水力性能优等优点,在南水北调东线等调水工程中得到了广泛的应用。大型泵站在非设计工况下运行时可能会诱发异常水力振动或涡激振动,异常水力振动会对泵站运行造成危害,因此,迫切需要对异常水力振动开展研究与防治。为了预防异常水力振动、提升水力性能,前人主要从泵装置和进水结构出发开展相关研究工作。泵装置方面:王福军等[1]研究了轴流泵叶轮、导叶以及不同流量下的水体压力脉动的规律,得出了压力脉动主要受叶轮转频控制等结论;陈世杰等[2]通过数值模拟对立式轴流泵叶片区水体压力脉动和不同流量工况下压力脉动特征进行了研究分析,揭示了一般情况下立式轴流泵叶片区压力脉动特征;张付林等[3]通过模型试验对超低扬程下的贯流泵水压力脉动进行了研究,得到了超低扬程下贯流泵扬程与叶片安放角与水压力脉动的关系;张德胜等[4]探究了叶轮与导叶叶片数对斜流泵水力性能的影响,得出了叶轮导叶叶片数、叶片厚度等对机组内水流压力脉动的影响机理;朱相源等[5]通过数值模拟与模型试验对比验证了导叶安放角度对离心泵内水流压力脉动的影响,得出了不同导叶安装角度对压力脉动的影响规律;方国材等[6]通过数值模拟分析了某泵装置内部水压力脉动规律与轴系模态规律,为贯流泵水力特性分析提供了一定参考。梁金栋等[7]研究了轴流泵装置出口速度环量规律。杨帆等[8]采用 RNG k-ε 模型针对“S”形卧式轴升贯流泵

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叶片区水流压力脉动进行了分析；苏文博等[9]通过RNG k-ε 模型研究了轴流泵叶片安放角对叶轮区域稳定性的影响；吕蕊蕊等[10]基于FBM湍流模型对轴流泵空化阶段压力脉动进行分析，得到轴流泵空化阶段叶轮导叶段压力脉动频域变化规律；周大庆等[11]通过DES方法对不同水头工况下的竖井贯流式水轮机叶轮、导叶、流道内的水压力脉动进行了研究。针对离心泵：前人[12-17]对离心泵内部与导叶处水体压力脉动特性与其影响因素进行了大量的研究；郑源等[18]、刘萌等[19]对混流泵压力脉动机理进行了研究。泵站进水结构方面：车晓红等[20]、施伟等[21]对不同流道水力优化设计方案进行了研究；焦伟轩等[22]采用RNG k-ε 模型对双向流道泵装置叶轮、导叶、流道区的水体压力脉动规律进行了研究分析；罗灿等[23]针对泵站整体进行水力性能优化，提出3种不同方案对某侧向进水泵站进行整流，与原方案对比流态显著改善，整流效果显著，泵站整体水力性能提升明显，为同类泵站运行提供了参考；还有学者[24-26]分析了隔墩对机组压力脉动的影响。国外学者[27-29]则主要关注的是泵装置压力脉动。

综上，前人围绕泵站的泵装置和进水结构开展了大量的研究，取得了丰硕的成果，然而泵装置压力脉动规律中多为单一流量或水位，缺乏不同流量和水位工况下压力脉动规律的分析，对进水结构的研究大多仅涉及进水池、流道、前池内流动特性的研究与水力性能的优化，缺少对压力脉动特性的研究分析。因此，为了探究不同流量和水位工况对机组压力脉动特性的影响，防止因异常水力脉动而诱发异常水力振动或涡激振动进而影响机组稳定运行，本文以某南水北调泵站为原型，针对灯泡贯流泵机组，采用非定常数值模拟技术分析灯泡贯流泵的压力脉动特性，可为同类型泵站预防异常水力振动，优化泵站运行效率提供一些参考。

1　数值模拟方法

1.1　计算模型

流动过程需满足质量守恒、动量守恒和能量守恒，由于无传热过程，所以本计算的控制方程为连续性方程与动量方程，湍流模型则采用RNG k-ε 模型。控制方程为

$$\frac{\partial u_i}{\partial x_i}=0 \tag{1}$$

$$\frac{\partial}{\partial (x_j)}(\rho u_i u_j)=-\frac{\partial p}{\partial x_j}+\frac{\partial}{\partial x_j}(\mu\frac{\partial u_i}{\partial u x_j})+\frac{\partial}{\partial x_j}(-\rho\overline{u'_i u'_j}) \tag{2}$$

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left(\alpha_k u_{\mathrm{eff}}\frac{\partial k}{\partial x_j}\right)+G_{\mathrm{k}}+G_{\mathrm{b}}-\rho\varepsilon-Y_{\mathrm{M}}+S_{\mathrm{k}} \tag{3}$$

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_j}\left(\alpha_\varepsilon u_{\mathrm{eff}}\frac{\partial\varepsilon}{\partial x_j}\right)+G_{1\varepsilon}\frac{\varepsilon}{k}(G_k+G_{3\varepsilon}G_{\mathrm{b}})-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}-R_\varepsilon+S_\varepsilon \tag{4}$$

式中：u_i、u_j 为各向上的流速，m/s，当 i、j 取1、2、3时分别对应 u、v、w；x_i、x_j 为三维方向，当 i,j 取1、2、3时分别对应 x、y、z；ρ 为流体的密度，kg/m^3；p 为流体压力，Pa；$-\rho\overline{u'_i u'_j}$ 为雷诺应力；$u_{\mathrm{eff}}=u+u_{\mathrm{t}}$，$u_{\mathrm{t}}=\rho Cu\frac{k^2}{\varepsilon}$；$C_{\mathrm{u}}=0.0845$；$\rho$ 为流体的密度，kg/m^3；$C_{1\varepsilon}=1.42$；$C_{2\varepsilon}=1.68$；$C_{3\varepsilon}$ 为常数；G_{k} 为平均速度梯度引起的湍流动能；G_{b} 为由浮力产生的湍流动能；Y_{M} 为可压缩湍流中波动膨胀对总耗散率的贡献；α_{k} 为 k 有效普朗特数的倒数；α_ε 为 ε 有效普朗特数的倒数；R_ε 为附加项，以适应应变率和流线曲率变化迅速流动的计算，S_{k}、S_ε 为用户定义的源项。

1.2　几何模型

构建包括进水延伸段、进水流道、前支撑片、叶轮、导叶、后支撑片、出水流道、出水延伸段和灯泡体等在内的灯泡贯流装置三维计算域，见图1。单机设计流量 $Q=37.5$ m^3/s，叶轮直径 $D=3.35$ m，叶轮转速 $n=115.4$ r/min，叶轮叶片数为3片，共3组支撑片，每组6片。泵站设计净扬程2.45 m。本模型设计工况下站下运行水位5.45 m，站上水位7.9 m。

图1　泵站灯泡贯流装置三维计算域

Fig. 1　Three-dimensional calculation domain of bulb tubular device in pumping station

为了分析灯泡贯流泵内部叶轮导叶区压力脉动规律，共布置18个脉动监测点，具体布置见图2。其中，在距离流道出口7.5 m处的叶轮进口断面设置了6个监测点，P_1～P_6 沿右侧轮缘到左侧轮缘每一侧均匀分布，同侧相邻监测点之间间隔为0.1D（D 为叶轮直径）。距离流道出口8.6 m的叶轮出口断面以同样的间隔设置了6个监测点，P_7～P_{12} 沿一侧轮缘到另一侧轮缘均匀分布。在距离流道出口10.3 m的导叶出口以相同方式布置，各监测点按照轴对称原则进行布置。

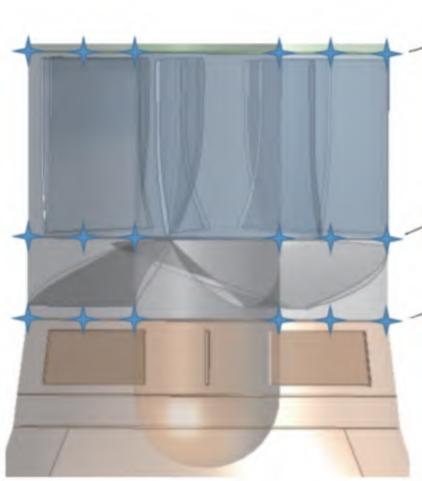


图 2　压力脉动监测点布置

Fig. 2　Layout of pressure fluctuation monitoring points

1.3　网格无关性分析

为了保证计算的高效性和结果的准确性,需对网格开展无关性分析。考虑到叶轮、导叶、流道结构均比较复杂,因此采用适应性较强的非结构化网格对模型进行网格划分。在 Ansys Mesh 平台下完成非结构化网格划分,网格数量分别为 15.4 万、20.9 万、70.1 万、128 万、255.7 万、497.4 万、640 万和 1 941.8 万个。分别计算对比不同网格数量下的 Δh 来选取合适的计算网格数量,公式为

$$\Delta h=\frac{P_{\rm in}-P_{\rm out}}{\rho g} \tag{5}$$

式中:Δh 为进水流道总水力损失,m;$P_{\rm in}$为进水流道进口处总压强,Pa;$P_{\rm out}$为进水流道出口处总压强,Pa;ρ 为水的密度,取 1×10^3 kg/m^3;g 为重力加速度,取 9.8 m/s^2。

图 3 为不同网格数量下的水力损失变化趋势,当网格数量超过 600 万个时,水力损失几乎没有变化,误差在±2%以内,此时网格整体质量在 0.8 以上,该网格满足计算要求。最终所选择的网格数量为 640 万个。

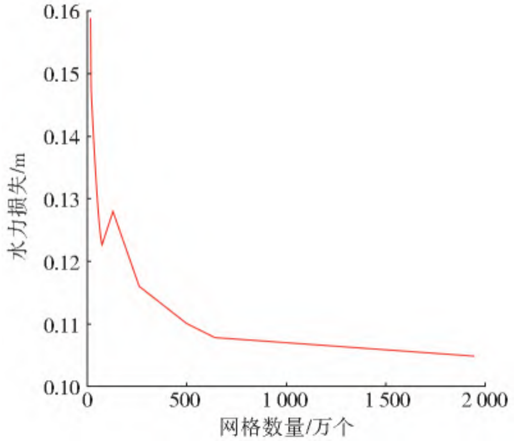


图 3　网格无关性变化趋势

Fig. 3　Trend of grid independence

采用 RNG k-ε 模型对网格 $y+$值要求保证在 30~100 以内,对计算结果影响较大的叶轮网格的 $y+$值进行分析发现,叶轮整体网格 $y+$值在 33.7~92.6,叶轮表面处网格 $y+$值在 37.5~76.8,满足计算模型要求。

1.4　边界条件设置

模型进出口分别设置在进、出水延伸段的进口断面与出口断面,分别采用质量流量进口与平均静压出口,参考压力设置为 1 个大气压(1 atm),固体边壁条件设置为无滑移壁面,壁面函数采用可伸缩的壁面函数(scalable)。定常计算中叶轮与相邻部件交界面采用冻结转子法(frozen rotor)耦合,非定常计算中采用瞬态转子定子法(transient rotor stator)耦合,叶轮每旋转 6°保存一次样本,共计算 12 个周期。

2　压力脉动特性分析

2.1　设计工况下压力脉动时域特性

叶轮旋转 360°为 1 个周期,为消除初始不稳定数据的影响,取后 6 个周期的数据进行分析。为了统一不同位置监测点压力变化的尺度,引入压力系数 C_p,其定义为

$$C_p=(p-\bar{p})/(\frac{1}{2}\rho\bar{u}^2) \tag{6}$$

式中:p 为测点瞬时压力,Pa;$\bar{p}$ 为测点时均压力,Pa;ρ 为水的密度,取 1×10^3 kg/m^3;$\bar{u}$ 为叶轮旋转线速度,m/s。

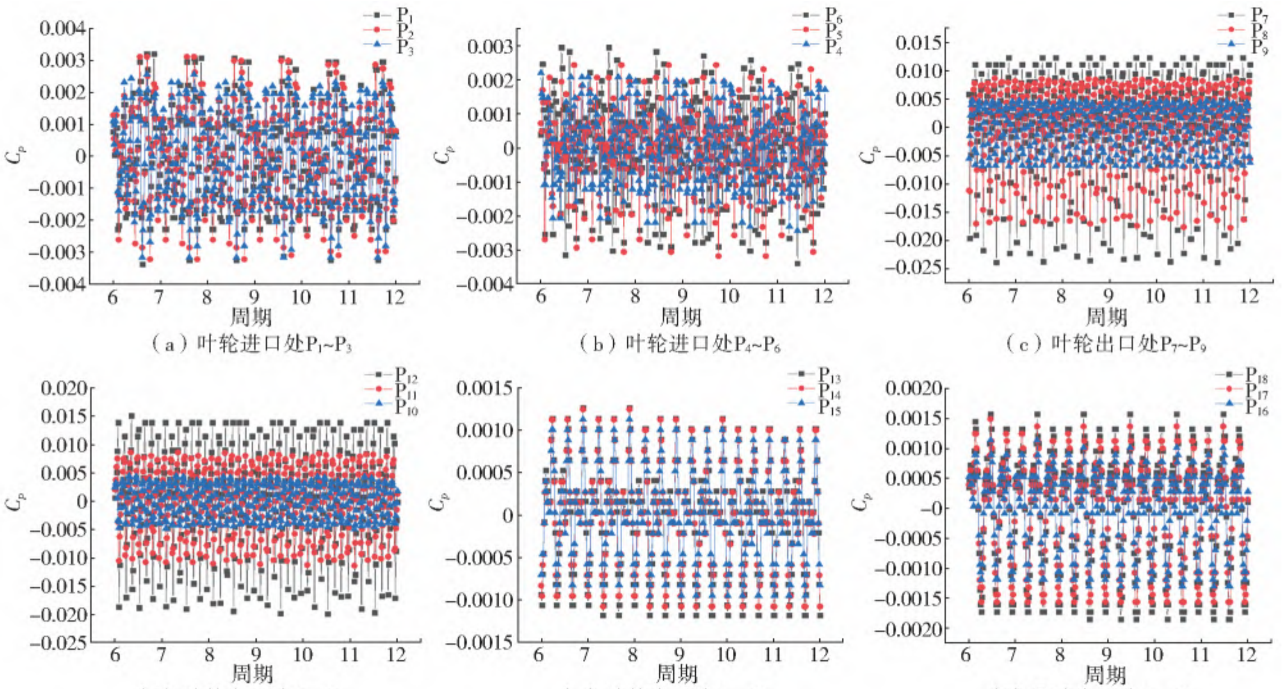
图 4 为各监测点的压力脉动时域图,可见各个断面从轮缘到轮毂监测点水压力脉动均具有一定的周期性,且沿泵轴线对称的两监测点脉动压力具有明显的相似性。

叶轮进口处水流压力脉动周期性明显,同一周期下压力脉动有多组波峰波谷,P_1~P_3 压力脉动幅值变化不大,P_4~P_6 压力脉动幅值呈渐增趋势,这是由于在叶轮进口前设置有 6 片支撑片,支撑片起支撑作用的同时也可以平顺水流,改善流态,与前置导叶作用相似,而支撑片与叶轮产生了动静干涉作用,无叶区产生的压力脉动对叶轮进口处压力脉动产生了影响。

叶轮出口水流压力脉动周期性亦很明显,同一周期内变化趋势比叶轮进口剧烈,P_7~P_9 压力脉动幅值呈逐渐减小的趋势,轮缘处压力脉动幅值约为轮毂处的 2~3 倍。

导叶出口水流压力脉动同一个周期内出现 3 个波峰 3 个波谷,这是因为叶轮叶片数也是 3 片,导叶

出口水流压力脉动仍受叶轮的影响。从轮缘到轮毂压力脉动亦呈逐渐减小的趋势。导叶出口水流由于远离脉动源且仍然具有一定的速度环量，压力脉动幅值整体呈减小或增大趋势。



（a）叶轮进口处P_1~P_3　（b）叶轮进口处P_4~P_6　（c）叶轮出口处P_7~P_9

（d）叶轮出口处P_{10}~P_{12}　（e）叶轮出口处P_{13}~P_{15}　（f）导叶出口处P_{16}~P_{18}

图 4　设计工况压力脉动时域图

Fig. 4　Time-domain diagram of pressure fluctuation under design conditions

整体看来，叶轮水体压力脉动周期性较强，从轮毂到轮缘压力脉动幅值变化明显，导叶区相对而言周期性没有叶轮区那么明显，但水流在叶轮导叶的共同作用下仍体现出一定的规律性，导叶区的水压力脉动幅值相对于叶轮区有明显的减小。

2.2　设计工况压力脉动频域特性

对上述所得压力脉动时域数据进行快速傅里叶变换(FFT)得到压力脉动频域图，通过压力脉动频域图分析压力脉动幅值和频率的变化特性。

为了寻求频域图中的频率与泵的特征频率之间的关系，定义了叶轮转频倍数，公式为

$$f=\frac{60N}{n} \tag{7}$$

式中：f 为转频倍数；N 为快速傅里叶变换后得到的实际频率，Hz；n 为叶轮转速，r/min。

图 5 为通过傅里叶变换后得到的各监测点压力脉动频域图。叶轮进口处主频为 6 倍转频，这是因为进口处水流受到了叶轮进口前支撑片的影响。次主频为 7 倍转频，进口处压力脉动幅值由轮缘到轮毂有小幅度波动但波动幅度不大，与之前的分析结果一致，这是因为水流受到了叶轮与支撑片动静干涉的影响，水流脉动较为紊乱，流场对称性较差。

叶轮出口处水流压力脉动幅值从轮缘到轮毂呈现出逐渐减小的趋势，主频为 7 倍转频，次主频为 14 倍转频，可见此处的水流主要受导叶与叶轮的共同影响。在叶轮一侧的 P_7、P_8、P_9 的主频幅值分别为 0.009 10、0.008 47、0.005 40，从轮缘到轮毂逐级递减，P_7 处压力脉动幅值约为 P_9 处压力脉动幅值的 1.69 倍。叶轮两侧对称监测点水压力脉动幅值差值最大的为轮毂两侧的点，相差达 26%，最小的为轮缘两侧测点，仅为 2.8%。这表明叶轮出口流场叶轮两侧对称点幅值基本相同，流场对称性良好。该处压力脉动仍然为低频脉动占据主导地位。

导叶出口处离脉动源较远，但水流脉动规律仍然受到叶轮与后支撑片的影响，主频为 3 倍转频，次频为 6 倍转频，导叶出口处的水流压力脉动仍然以低频脉动为主，分布相对于叶轮区较为分散。P_{13}～P_{18}的主频压力脉动幅值分别为 0.000 62、0.000 57、0.000 47、0.000 74、0.001 00、0.001 24，从轮缘到轮毂压力脉动主频幅值逐渐减小，轮缘处压力脉动约为轮毂处的 1.67～1.32 倍。导叶两侧对称点幅值相对差值最大达到了 50%，最小为 36%，可见该处流场对称性较差，这是因为导叶出口水流仍具有一定的速度环量，水流流态较为复杂。从图 5 可以看出叶轮区水流压力脉动幅值明显大于导叶区。

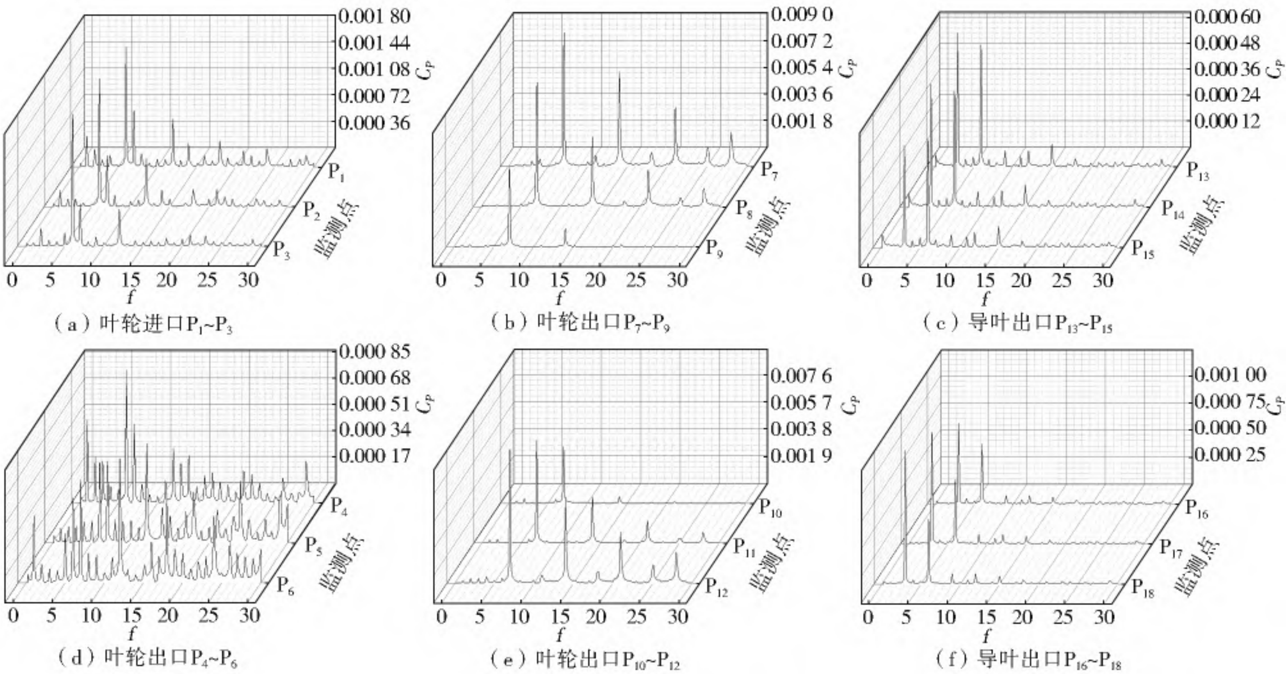


图 5　设计工况压力脉动频域图

Fig. 5　Frequency domain diagram of pressure fluctuation under design conditions

2.3　不同水位工况下的压力脉动特性

为了进一步对比分析机组非设计工况运行下的水力特性，取设计工况(下游水位 5.45 m，上游水位 7.90 m，上下游水位差 2.45 m)与运行工况(下游水位 6.30 m，上游水位 7.70 m，上下游水位差 1.40 m)两种工况下在 P_3(叶轮进口轮毂)、P_{11}(叶轮出口轮毂与轮缘中间点)和 P_{14}(导叶出口轮毂与轮缘中间点)的压力脉动频谱进行对比分析，见图 6。

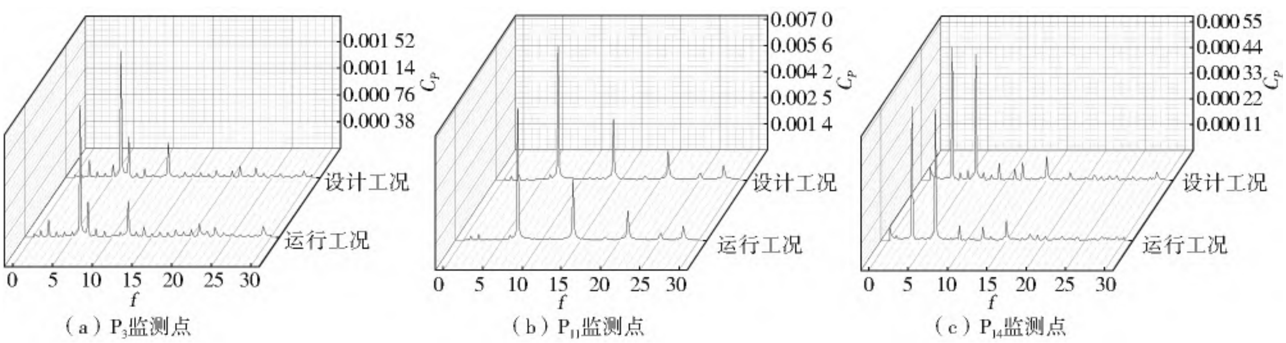


图 6　不同水位工况压力脉动频域图

Fig. 6　Frequency domain diagram of pressure fluctuation under different water level conditions

图 6 显示，不同水位下泵机组 3 个监测点压力脉动主次频均未有明显改变，P_3、P_{11}、P_{14} 监测点主频仍然分别为 6 倍转频、7 倍转频和 3 倍转频。运行工况处主次频压力脉动幅值略大于设计工况，由此可见上下游水位差会对机组压力脉动幅值产生影响。

2.4　不同流量工况下的压力脉动特性

分别对 0.5Q、1.0Q、1.5Q 这 3 种流量工况下的压力脉动数据进行快速傅里叶变换，分别取 P_5(叶轮进口轮缘轮毂中间点)，P_{12}(叶轮出口轮缘点)和 P_{16}(导叶出口轮毂点)的压力频谱图进行对比分析，见图 7。

由图 7 可以看出，各特征工况主次频未有明显变化，P_5 监测点的水压力脉动主频均为 6 倍转频。设计工况下主频幅值最小，小流量工况主频幅值最大，小流量工况下主频幅值为设计工况的 2.55 倍，此时可观察到小流量工况下存在大量低频脉动。P_{12} 监测点，3 种特征工况主频均为 7 倍转频，小流量工况主频幅值最大，为设计工况的 1.95 倍。P_{16} 监测点 3 种工况主频均为 3 倍转频，小流量工况的主频幅值仍最大，约为设计工况主频处幅值的 2.24 倍，小流量工况下还存在低频脉动，随着流量的增大，低频脉动逐渐减弱，大流量工况下该低频脉动已基本消失。

可以看到流量是影响压力脉动的一个重要因素，小流量工况下压力脉动幅值基本为设计工况的2倍左右，大流量工况下压力脉动幅值也基本大于设计工况，这说明偏离设计流量时压力脉动主频基本不变，其幅值则会发生较大波动。

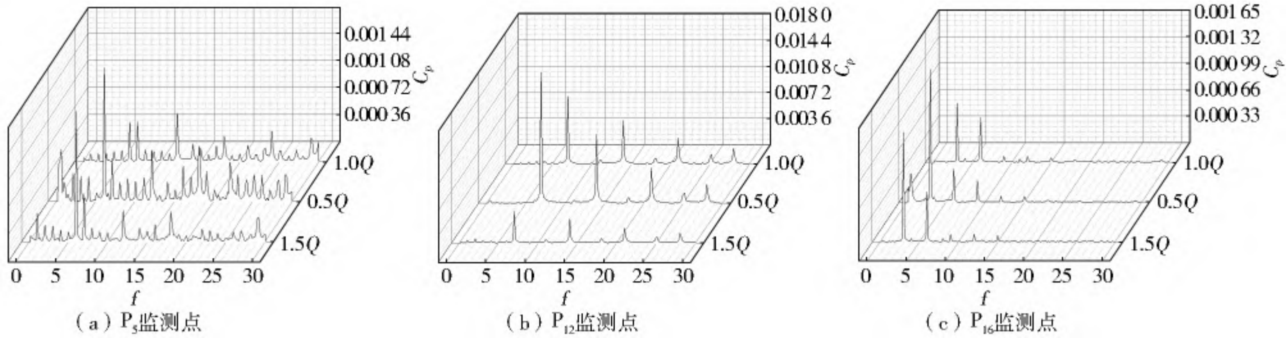


图7　不同流量工况压力脉动频域图

Fig. 7　Frequency domain diagram of pressure fluctuation under different flow conditions

3　结　论

本文通过数值模拟的方法对灯泡贯流泵机组叶轮导叶区水流压力脉动进行了计算分析，对比不同流量、水位工况下压力脉动时频域特性，对可能影响泵站正常运行的工况进行了具体分析，为同类泵站避免异常水力振动、安全高效运行提供了一定的参考，具体结论如下：

设计工况下各断面水压力脉动时域图均有明显的周期性，一个周期内脉动压力规律受叶轮导叶叶片数影响，导叶区水流压力脉动幅值整体小于叶轮区。设计工况下叶轮进出口、导叶出口水流压力脉动主次频均受到叶轮、导叶、前后支撑片影响，为整数倍转频。压力脉动分布规律受到叶轮与导叶或支撑片间的动静干涉作用影响，叶轮进口处压力脉动幅值从轮缘到轮毂变化不大，轮毂处压力脉动幅值略大于轮缘，叶轮出口与导叶出口水流压力脉动幅值从轮缘到轮毂逐渐减小。叶轮区流场对称性较强，对称点相对幅值差值最大为26%、最小仅为2.8%，导叶区受速度环量影响对称性较弱。

非设计水位工况下压力脉动幅值大于设计工况，上下游水位差会影响压力脉动幅值。非设计工况下运行，无论小流量还是大流量工况，各断面监测点压力脉动幅值均大于设计工况，且小流量工况下会产生频繁的低频脉动。因此，非设计工况下，随着压力脉动幅值增大，低频脉动增多，将会影响机组运行效率，特别在小流量工况下，流场呈现出不稳定性，低频脉动激增，易引发异常水力振动，影响泵站安全运行，泵站运行过程中应尽量避免非设计工况运行特别是小流量工况运行。

参考文献(References)：

[1] 王福军，张玲，张志民. 轴流泵不稳定流场的压力脉动特性研究[J]. 水利学报，2007(8)：1003-1009.（WANG F J，ZHANG L，ZHANG Z M. Analysis on pressure fluctuation of unsteady flow in axial-flow pump[J]. Journal of Hydraulic Engineering，2007(8)：1003-1009.（in Chinese））DOI：10.3321/j.issn：0559-9350.2007.08.019.

[2] 陈世杰，杨帆，黄辉，等. 立式轴流泵装置叶片区压力脉动数值分析[J]. 流体机械，2019，47(2)：17-22.（CHEN S J，YANG F，HUANG H，et al. Numerical analysis of pressure fluctuation of the blade region in vertical axial-flow pumping system[J]. Fluid Machinery，2019，47(2)：17-22.（in Chinese））DOI：10.3969/j.issn.1005-0329.2019.02.004.

[3] 张付林，郑源，孙奥冉，等. 超低扬程贯流泵模型试验的压力脉动研究[J]. 中国农村水利水电，2017(11)：173-176.（ZHANG F L，ZHENG Y，SUN A R，et al. Research on the pressure pulsation of model tests of super low head tubular pump[J]. China Rural Water and Hydropower，2017(11)：173-176.（in Chinese））DOI：10.3969/j.issn.1007-2284.2017.11.036.

[4] 张德胜，施卫东，王川，等. 斜流泵叶轮和导叶叶片数对压力脉动的影响[J]. 排灌机械工程学报，2012，30(2)：167-170.（ZHANG D S，SHI W D，WANG C，et al. Influence of impeller and guide vane blade number on pressure fluctuation in mixed-flow pump[J]. Journal of Drainage and Irrigation Machinery Engineering，2012，30(2)：167-170.（in Chinese））DOI：10.3969/j.issn.1674-8530.2012.02.009.

[5] 朱相源，江伟，李国君，等. 导叶与隔舌相对位置对离心泵非定常压力脉动影响的数值模拟[J]. 排灌机械工程学报，2015，33(10)：829-834.（ZHU X Y，JIANG W，LI

G J, et al. Numerical investigation into influence of installation location between diffuser vane and volute tongue on unsteady pressure fluctuations in centrifugal pump with vane diffuser[J]. Journal of Drainage and Irrigation Machinery Engineering, 2015, 33(10): 829-834. (in Chinese)) DOI: 10.3969/j.issn.1674-8530.15.0075.

[6] 方国材,郭绘娟,胡大明,等.大型竖井贯流泵压力脉动及轴系模态优化[J].南水北调与水利科技(中英文),2021,19(4):822-832.(FANG G C, GUO H J, HU D M, et al. Pressure pulsation and shaft system modal optimization of a large shaft tubular pump[J]. South-to-North Water Transfers and Water Science & Technology, 2021, 19(4): 822-832. (in Chinese)) DOI: 10.13476/j.cnki.nsbdqk.2021.0086.

[7] 梁金栋,陆林广,徐磊,等.轴流泵装置导叶出口水流速度环量对出水流道水力损失的影响[J].农业工程学报,2012,28(1):55-60.(LIANG J D, LU L G, XU L, et al. Influence of flow velocity circulation at guide vane outlet of axial-flow pump on hydraulic loss in outlet conduit[J]. Transactions of the Chinese Society of Agricultural Engineering, 2012, 28(1): 55-60. (in Chinese)) DOI: 10.3969/j.issn.1002-6819.2012.01.011.

[8] 杨帆,刘超,汤方平,等.S形下卧式轴伸贯流泵装置叶片区压力脉动特性研究[J].流体机械,2015,43(1):16-22.(YANG F, LIU C, TANG F P, et al. Pressure pulsations of the blade region in S-shaped shaft-extension tubular pumping system[J]. Fluid Machinery, 2015, 43(1): 16-22. (in Chinese)) DOI: 10.3969/j.issn.1005-0329.2015.01.004.

[9] 苏文博,刘卫东,周大庆,等.轴流泵叶片安放角增大过渡过程特性[J].南水北调与水利科技(中英文),2020,18(6):164-169.(SU W B, LIU W D, ZHOU D Q, et al. Characteristics of transition process of increasing axial flow pump's blade angle[J]. South-to-North Water Transfers and Water Science & Technology, 2020, 18(6): 164-169. (in Chinese)) DOI: 10.13476/j.cnki.nsbdqk.2020.0128.

[10] 吕蕊蕊,郑源,张德虎,等.基于FBM模型的轴流泵空化流动特性CFD分析[J].南水北调与水利科技(中英文),2020,18(1):150-165.(LYU R R, ZHENG Y, ZHANG D H, et al. CFD analysis of cavitation flow for axial-flow pump based on FBM model[J]. South-to-North Water Transfers and Water Science & Technology, 2020, 18(1): 150-165. (in Chinese)) DOI: 10.13476/j.cnki.nsbdqk.2020.0017.

[11] 周大庆,张蓝国,郑源,等.基于脱体涡模型的竖井贯流式水轮机组压力脉动数值分析[J].水利水电科技进展,2013,33(3):29-33.(ZHOU D Q, ZHANG L G, ZHENG Y, et al. Numerical analysis of pressure pulsation of pit water turbines based on detached-eddy model[J]. Advances in Science and Technology of Water Resources, 2013, 33(3): 29-33. (in Chinese)) DOI: 10.3880/j.issn.1006-7647.2013.03.007.

[12] 曹卫东,刘光辉,施卫东,等.多级离心泵内部非定常压力分布特性[J].农业工程学报,2014,30(14):64-70.(CAO W D, LIU G H, SHI W D, et al. Distribution of unsteady pressure in multistage centrifugal pump[J]. Transactions of the Chinese Society of Agricultural Engineering, 2014, 30(14): 64-70. (in Chinese)) DOI: 10.3969/j.issn.1002-6819.2014.14.009.

[13] 成科,陈启明,崔宝玲,等.多级离心泵扬程预测及内部流场数值模拟[J].流体机械,2021,49(3):60-67.(CHENG K, CHEN Q M, CUI B L, et al. Multi-stage centrifugal pump head prediction and numerical simulation of internal flow field[J]. Fluid Machinery, 2021, 49(3): 60-67. (in Chinese)) DOI: 10.3969/j.issn.1005-0329.2021.03.010.

[14] 王玉川,谭磊,曹树良,等.离心泵叶轮区瞬态流动及压力脉动特性[J].机械工程学报,2014,50(10):163-169.(WANG Y C, TAN L, CAO S L, et al. Characteristics of transient flow and pressure fluctuation in impeller for centrifugal pump[J]. Journal of Mechanical Engineering, 2014, 50(10): 163-169. (in Chinese)) DOI: 10.3901/JME.2014.10.163.

[15] 周强,李宏坤,仲作文,等.离心泵导叶流道进口处压力脉动研究[J].工程热物理学报,2020,41(7):1679-1684.(ZHOU Q, LI H K, ZHONG Z W, et al. Investigation on pressure pulsation at the inlet of guide vane in a centrifugal pump[J]. Journal of Engineering Thermophysics, 2020, 41(7): 1679-1684. (in Chinese)).

[16] 王文杰,邓起凡,裴吉,等.双吸离心泵压力脉动特性数值模拟及试验研究[J].振动与冲击,2020,39(19):107-113,120.(WANG W J, DENG Q F, PEI J, et al. Numerical simulation and tests for pressure fluctuation characteristics in a double-suction centrifugal pump[J]. Journal of Vibration and Shock, 2020, 39(19): 107-113, 120. (in Chinese)) DOI: 10.13465/j.cnki.jvs.2020.19.016.

[17] 杨从新,杨藤,刘满.叶轮直径对双蜗壳离心泵压力脉动特性的影响[J].中国农村水利水电,2020(5):139-144,149.(YANG C X, YANG T, LIU M. The influence of impeller diameter on pulsation characteristics of double volute centrifugal pumps[J]. China Rural Water and Hydropower, 2020(5): 139-144, 149. (in Chinese)) DOI: 10.3969/j.issn.1007-2284.2020.05.028.

[18] 郑源,陈宇杰,毛秀丽,等.混流泵压力脉动特性及其

对流动诱导噪声的影响[J]. 农业工程学报,2015,31(23):67-73.(ZHENG Y,CHEN Y J,MAO X L,et al. Pressure pulsation characteristics and its impact on flow-induced noise in mixed-flow pump[J]. Transactions of the Chinese Society of Agricultural Engineering,2015,31(23):67-73.(in Chinese)) DOI:10.11975/j. issn. 1002-6819. 2015. 23. 009.

[19] 刘萌,姜永明,陈婷. 叶片-导叶间隙对混流泵非稳态流场特性的影响[J]. 水动力学研究与进展(A辑),2019,34(6):734-739.(LIU M,JIANG Y M,CHEN T. Influence of the blade-guide vane clearance on the unsteady performance of a mixed-flow pump[J]. Journal of Hydrodynamics,2019,34(6):734-739.(in Chinese)) DOI:10. 16076/j. cnki. cjhd. 2019. 06. 005.

[20] 车晓红,张帝,成立. 立式混流泵站进出水流道水力优化[J]. 南水北调与水利科技(中英文),2020,18(5):144-150.(CHE X H,ZHANG D,CHENG L. Hydraulic optimization of inlet and outlet passages of vertical mixed flow pumping station[J]. South-to-North Water Transfers and Water Science & Technology,2020,18(5):144-150.(in Chinese)) DOI:10. 13476/j. cnki. nsbdqk. 2020. 0103.

[21] 施伟,刘军,陆林广,等. 前置竖井式贯流泵装置进出水流道水力设计标准化[J]. 南水北调与水利科技(中英文),2020,18(5):115-122.(SHI W,LIU J,LU L G,et al. Hydraulic design standardization for inlet and outlet conduit of tubular pump system with front-positioned shaft[J]. South-to-North Water Transfers and Water Science & Technology,2020,18(5):115-122.(in Chinese)) DOI:10. 13476/j. cnki. nsbdqk. 2020. 0099.

[22] 焦伟轩,成立,颜红勤,等. 超低扬程双向流道泵装置压力脉动特性研究[J]. 水力发电学报,2019,38(6):101-112.(JIAO W X,CHENG L,YAN H Q,et al. Pressure fluctuation characteristics of bidirectional pumping system under super-low head operation[J]. Journal of Hydroelectric Engineering,2019,38(6):101-112.(in Chinese)) DOI:10. 11660/slfdxb. 20190611.

[23] 罗灿,雷帅浩,袁尧,等. 小型闸站式侧向泵站进水流态数值模拟研究[J]. 排灌机械工程学报,2021,39(8):797-803.(LUO C,LEI S H,YUAN Y,et al. Numerical simulation research on inlet flow pattern of small sluice station lateral pumping station[J]. Journal of Drainage and Irrigation Machinery Engineering (JDIME),2021,39(8):797-803.(in Chinese)) DOI:10. 3969/j. issn. 1674-8530. 21. 0010.

[24] 徐磊,夏斌,施伟,等. 中隔墩长度对斜式轴伸泵装置出水流道水力特性的影响[J]. 农业工程学报,2020,36(1):74-81.(XU L,XIA B,SHI W,et al. Influence of middle pier lengths on hydraulic characteristic of outlet conduit in pump system with slanted extension shaft[J]. Transactions of the Chinese Society of Agricultural Engineering,2020,36(1):74-81.(in Chinese)) DOI:10. 11975/j. issn. 1002-6819. 2020. 01. 009.

[25] 梁金栋,陆林广,刘荣华,等. 中隔墩对大型泵站出水流道水力性能的影响[J]. 排灌机械工程学报,2011,29(1):77-81.(LIANG J D,LU L G,LIU R H,et al. Influence of middle division pier on hydraulic performance in pumping station outlet conduit[J]. Journal of Drainage and Irrigation Machinery Engineering,2011,29(1):77-81.(in Chinese)) DOI:10. 3969/j. issn. 1674-8530. 2011. 01. 016.

[26] 杨帆,陈世杰,刘超,等. 隔墩对轴流泵装置直管式出水流道内流及脉动的影响[J]. 农业机械学报,2018,49(5):212-217.(YANG F,CHEN S J,LIU C,et al. Influence of division pier on pressure fluctuation and internal flow pattern in outlet conduit of axial-flow pumping system[J]. Transactions of the Chinese Society for Agricultural Machinery,2018,49(5):212-217.(in Chinese)) DOI:10. 6041/j. issn. 1000-1298. 2018. 05. 024.

[27] JIN Y,HE X,ZHANG Y,et al. Numerical and experimental investigation of external characteristics and pressure fluctuation of a submersible tubular pumping system[J]. Processes,2019,7(12):949. DOI:10. 3390/pr7120949.

[28] KHALIFA A E,AL-QUTUB A M,BEN-MANSOUR R. Study of pressure fluctuations and induced vibration at blade-passing frequencies of a double volute pump[J]. Arabian Journal for Science and Engineering,2011,36(7):1333-1345. DOI:10. 1007/s13369-011-0119-8.

[29] WANG C,HE X,SHI W,et al. Numerical study on pressure fluctuation of a multistage centrifugal pump based on whole flow field[J]. AIP Advances,2019,9(3):035118. DOI:10. 1063/1. 5049196.

Numerical simulation of pressure fluctuation in postpositional bulb tubular pump

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Abstract: Bulb tubular pump is widely used in South-to-North Water Transfer Project because of high efficiency, large flow rate, and excellent hydraulic performance. Abnormal hydraulic resonance could cause when large pumping stations operate under off-design conditions, and the operation of pumping stations could disturbed by abnormal hydraulic resonance. Therefore, it is urgent to study and prevent abnormal hydraulic resonance due pressure fluctuation of impeller and guide vane of postpositional bulb tubular pump in South-to-North Water Transfer pumping station under different working conditions.

The pressure fluctuation in the vane area of the impeller under the condition of partial flow and water level was calculated and analyzed by CFD method. To analyze the pressure fluctuation law of impeller and guide vane area in bulb tubular pump, 18 pressure fluctuation monitoring points were arranged. The frequency-domain diagram of pressure fluctuation was obtained by fast Fourier transform of time-domain data of pressure fluctuation. The variation characteristics of amplitude and frequency of pressure fluctuation were analyzed by frequency domain diagram of pressure fluctuation.

The results showed that the time-domain diagram of pressure fluctuation in the impeller and guide vane was periodical. The primary and secondary frequency of pressure fluctuation was influenced by the number of impeller blades. The primary and secondary frequencies of the impeller and guide vane area were all integral multiples, and the pressure fluctuation amplitude of the impeller and guide vane area was decreased from the rim to the hub. The pressure fluctuation amplitude in the impeller area was significantly greater than that in the guide vane area. Under the partial water level condition, the amplitude of pressure fluctuation in the impeller and guide vane area was slightly larger than that under the design condition, and the primary and secondary frequencies were not changed. The pressure fluctuation amplitude of small and large flow conditions under partial flow conditions was greater than the design condition. The pressure fluctuation amplitude of the small flow condition at each monitoring point was about 2-3 times of the design condition. The low-frequency fluctuation was obvious under the small flow condition. The pressure fluctuation amplitude of the pumping units was greatly influenced by the operation under the partial working condition. Pumping unit operation efficiency was seriously affected by the long-term off-design operation.

Under the design condition, the time-domain diagram of pressure fluctuation of each section was obvious periodicity. The law of fluctuating pressure in a cycle was affected by the number of impeller guide vanes, and the pressure fluctuation amplitude in the guide vane area was less than that in the impeller area as a whole. The primary and secondary frequency of pressure fluctuation at the inlet and outlet of the impeller and the outlet of the guide vane were affected by the impeller, guide vane and front and rear support plates, which was an integral multiple of the conversion frequency. The distribution law of pressure fluctuation was affected by the dynamic and static interference between the impeller and the guide vane or support plate. The pressure fluctuation amplitude at the impeller inlet changed little from the rim to the hub, the pressure fluctuation amplitude at the hub was slightly larger than the rim, and the pressure fluctuation amplitude at the impeller outlet and guide vane outlet was decreased gradually from the rim to the hub. The pressure fluctuation amplitude under partial water level conditions was large than that under design conditions, and the pressure fluctuation was affected by the amplitude of the water level difference between upstream and downstream. Under the off-design conditions, the pressure fluctuation amplitude of each section monitoring point was large than that of the design condition both under small flow rate and large flow rate, and frequent low-frequency fluctuation occurred under small flow rate. Abnormal hydraulic resonance was easy to occur under off-design conditions, which aggravated the cavitation of the pump impeller and generated adverse flow patterns. During the operation of the pumping station, off-design conditions, especially small flow conditions, should be avoided as far as possible.

Key words: pumping station; bulb tubular pump; pressure fluctuation; numerical simulation; fast Fourier transform