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蒸汽发生器传热管束弯管区流致振动试验

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摘要: [目的] 蒸汽发生器传热管在二次侧流体的冲刷作用下, 容易产生流致振动问题, 其中随机的湍流作用力是导致传热管振动的主要的机理之一。当流体的脉动压力频率与传热管固有频率相近时, 会导致结构共振, 长时间振动会导致传热管失效, 因此研究传热管在流体激励下的动力响应特点是非常必要的。[方法] 文章设计了节径比为 1.47 的传热管束弯管区流致振动试验装置, 用空气-水两相流模拟了二次侧流体工况, 分别测量了空泡份额为 0.7~0.98, 管间流速为 5~13 m/s 的流体脉动压力和管束振动加速度。[结果] 结果表明, 在低流速下, 脉动压力主频与传热管固有频率接近, 容易产生共振, 且共振时管束振幅会有所增大。随着管间流速增大, 管束受到的脉动压力也相应增大, 而随着空泡份额的增大, 脉动压力的变化是先增大后减小。在计算空气-水两相流工况下的脉动压力主频时, 经验公式中的系数需要进行适当调整。[结论] 本试验模拟了蒸汽发生器传热管束弯管区二次侧流体运行工况, 在试验本体设计上考虑了与原型的几何相似性与支撑、约束相似性, 相比以往的试验研究更接近实际情况, 可为工程应用提供设计参考。

关键词: 压水堆核电站; 蒸汽发生器传热管; 空气-水两相流; 脉动压力; 湍流激振

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论文二维码

Experimental Study on Flow-Induced Vibration in Bend Zone of Steam Generator Tube Bundle

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Abstract: [Introduction] Under the scouring impact of secondary side fluid, the steam generator tube is prone to flow-induced vibration. One of the main mechanisms leading to tube vibration is the random turbulent force. When the fluctuating pressure frequency of the fluid is close to the natural frequency of the tube, structural resonance will be caused and long-term vibration will lead to the failure of the tube. Therefore, it is necessary to study the dynamic response characteristics of steam generator tubes under fluid excitation. [Method] In this paper, a flow-induced vibration test device for the bend zone of tube bundle with a pitch-diameter ratio of 1.47 was designed. The secondary side fluid condition was simulated by air-water two-phase flow. The fluid fluctuating pressure and the vibration acceleration of tube bundle with a void fraction of 0.7~0.98 and a flow velocity between tubes of 5~13 m/s was measured. [Result] The results show that: the main frequency of fluctuating pressure is close to the natural frequency of tube at low flow velocity, which is easy to causes resonance; when resonance occurs, the amplitude of the tube bundle increases; with the increase of velocity between tubes, the fluctuating pressure on tube bundle increases correspondingly. With the increase of the void fraction, the fluctuation pressure first increases and then decreases. When calculating the main frequency of fluctuating pressure under the condition of air-water two-phase flow, the coefficient in

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the empirical formula can be adjusted appropriately. [Conclusion] This test simulated the working conditions of the secondary side fluid operation in the bend zone of steam generator tube bundle and considered the geometric similarity with the prototype and the similarity of support and constraint in the design of the model. This test was closer to the actual situation than the previous experimental research and can provide design references for engineering applications.

Key words: pressurized water reactor; steam generator tube; air-water two-phase flow; fluctuating pressure; turbulent excitation

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0 引言

核能作为一种清洁、低碳、高效的优质能源,备受世界各国瞩目。积极安全有序发展核电是中国优化能源结构、保障能源供给安全、实现“双碳”目标、应对气候变化的重要手段。基于此,国内多家企业对核能的发展策略进行了深入研究并探讨核能作为绿色能源更广泛地服务于设备能源需求的可能性^[1-2]。

蒸汽发生器是压水堆核电站中连接一回路与二回路的关键设备。蒸汽发生器U型传热管在二次侧流体的冲刷作用下,容易产生流致振动的问题^[3],该振动可引起传热管疲劳和磨损失效,进而威胁到核电站的安全运行^[4-6]。

目前认为管束流致振动激励机制主要有4种:(1)周期性漩涡脱落;(2)湍流激振;(3)流体弹性失稳;(4)声共振^[7]。传热管在受到二次侧流体冲刷时,由于随机脉动的流场压力会使得传热管发生振动,即湍流抖振。当流体的激振力频率与传热管的固有频率相近时,传热管会发生共振,从而使振幅增大。如果经历长时间的反复振动,管子将产生疲劳,甚至与支撑板管孔接触的管壁会被磨穿^[8-9],因此,在设计中需要避免传热管发生共振。

为了研究蒸汽发生器传热管在实际运行过程中的振动性能,各国开展了大量试验和计算仿真^[10-12]工作。韩国Chu等^[13]建立了一套以空气-水两相流来模拟二次侧流体冲刷的试验装置,测试了传热管弯管区域的动态特性并对流弹失稳的经验公式进行了验证。印度英迪拉·甘地原子能研究中心^[14]建立了一套钠冷快堆蒸汽发生器试验装置,并开展了一系列关于蒸汽发生器管板、法兰接头设计、传热管流弹失稳的试验。加拿大Sawadog等^[15-16]针对转置三角形排布的直管段建立了空气-水两相流试验装置,得到了不同空泡份额和节径比下的管束动态响应特性。中国核动力研究设计院^[17]对管束流致振动

的关键力学问题进行了讨论,分别对华龙一号蒸汽发生器^[18]和ZH-65型蒸汽发生器^[19]传热管束流致振动问题进行了仿真及试验验证。上海核工程研究设计院、中广核工程设计公司的研究者^[20-22]发现在蒸汽发生器中对管束起支撑、约束作用的抗振条对传热管流致振动特性有较大影响,而往往在试验研究中是忽略掉抗振条的,工程计算中也多采用经验公式^[23],应用范围有限。除此之外,研究人员发现蒸汽发生器的管子支撑板开孔形式影响管束局部流场分布,从而影响传热管流致振动及微动磨损行为^[24-25]。还有学者通过试验研究讨论了泵致脉动压力与传热管动态特性和疲劳失效的关系^[26]。

文章建立了包含63根传热管弯管区和抗振条的试验本体,通过气水两相流模拟蒸汽发生器二次侧流体,对传热管束弯管区的流体脉动压力、管束振动特性进行了测量与分析。

1 试验概述

1.1 试验系统

试验系统主要由试验本体、气水分离器、供水系统、供气系统、测量系统等组成。本次试验主要模拟空泡份额 ϵ_g 从0.7~0.98,管间流速 V_{cross} 从5~13 m/s变化,经计算供水系统的设计总流量为25~3040 m³/h,可在该范围内连续调节。供气系统用于向试验本体提供稳定的不同流量压缩空气,压缩空气流经试验本体后,通过气水分离器排空。供气系统的运行压力为0.75 MPa,设计流量800~10200 Nm³/h。因试验规模及场地所限,试验本体采用局部等比例模型,截取实际蒸汽发生器传热管的中间9排。试验本体装置如图1所示,包括63根内侧的U型传热管束、4组抗振条、1组支撑板、管板、导流罩以及支座。传热管外径 D 为17.5 mm,壁厚 t 为1.01 mm,弯管半径 R 为510~660 mm,每排管束之间均安装有抗振条。传热管按照三角形排布方式成14排、

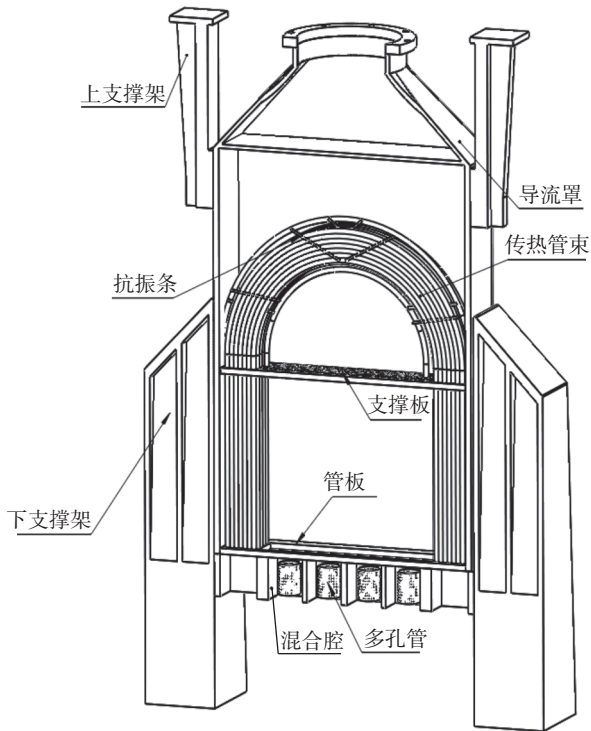


图 1 试验本体装置图

Fig. 1 Test model

9 列, 传热管通过带有三叶梅花孔的支撑板, 固定在底端的管板上。试验本体的结构设计满足与工程样机的几何相似性及刚度相似性^[27]。

1.2 测量系统及测试工况

试验本体上共布置了 12 个三向加速度计和 12 个脉动压力传感器。如图 2 所示, 分别布置在传热管束弯管区 90°弯与 45°弯位置上。加速度计通过卡箍固定在管 5-1(第 5 排第 1 列, 下同)、管 5-2、管 6-1、管 6-4、管 6-7 和管 5-7 内部, 脉动压力安装在传热管 4-1、管 4-4、管 4-7、管 5-3、管 5-6 和管 6-2 的外壁面。通过调整气、水的体积流量来模拟空泡份额 ε_g 从 0.7~0.98, 管间流速 V_{cross} 从 5~13 m/s 变化的二次侧流工况。

2 传热管固有频率计算

传热管在空气中与气水两相流中的固有频率是不同的。因此考虑动水附加质量^[28], 利用有限元软件分别计算了空泡份额在 0.7、0.8、0.9、0.98 共 4 种工况下传热管束的固有频率。在传热管与管板连接处作固支边界条件, 在与支撑板接触处作简支边界条件处理。抗振条与管束接触的部位, 则采用接地弹簧

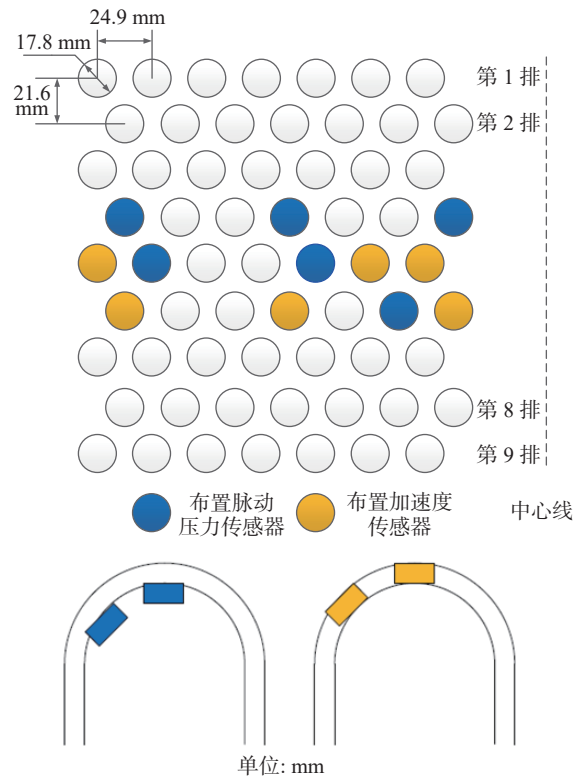


图 2 传感器分布图

Fig. 2 Distribution of sensors

来进行模拟^[20]。其中: 附加流体质量计算公式^[28]为:

$$m_A = \left(\frac{\rho \pi D^2}{4} \right) \frac{(D_e/D)^2 + 1}{(D_e/D)^2 - 1} \quad (1)$$

式中:

D ——传热管外径(mm);

ρ ——空气-水两项流密度(kg/m^3);

D_e ——管道的等效直径(mm)。

对于三角形排布的管束^[29]而言。

$$\frac{P}{D} = \frac{24.9}{17.8} = 1.42 \quad (2)$$

$$\frac{D_e}{D} = \left(0.96 + 0.5 \frac{P}{D} \right) \frac{P}{D} = 2.37 \quad (3)$$

式中:

P ——管束排布间距(mm)。

通过对传热管的固有频率分析发现前两阶振动方向均为面外方向, 如图 3、图 4 所示为传热管典型 1 阶、2 阶阵型图。由表 1 可知, 随着弯管半径的增加, 管束的 1 阶固有频率是逐渐减小的。同时随着空泡份额的增大, 附加动水质量变小, 因此, 每根传热管的固有频率均随着空泡份额增大而增大。

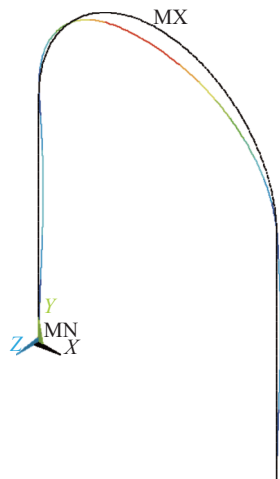


图 3 传热管典型 1 阶阵型图
Fig. 3 Typical first modal shape of tube

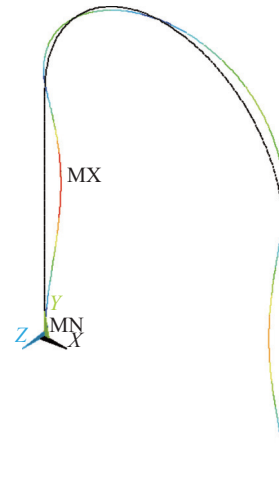


图 4 传热管典型 2 阶阵型图
Fig. 4 Typical second modal shape of tube

3 实验结果分析

3.1 测点位置的影响

在空泡份额为 0.7, 管间流速为 5 m/s 工况下, 各个测点的脉动压力主频均一致为 43.457 Hz, 脉动压力功率谱密度值随着弯管半径的增大而减小, 90°位置与 45°位置处所受流体激振力基本一致, 如下图 5 所示。表 2 中分别选取了管束内侧、中间和外侧的 3 组测点, 内侧的脉动压力功率谱密度峰值是外侧的 2.7 倍, 说明了流体激振力在流体出口处相比流体入口处有较大衰减。结合管束固有频率的分析, 在空泡份额 0.7, 管间流速为 5 m/s 工况下, 流体脉动压力频率与传热管束一阶固有频率相近, 结构易发生共振。

3.2 管间流速的影响

图 6 显示了空泡份额 0.7 时, 传热管 4-1-90°所受到的脉动压力随管间流速的增大而增大, 5 m/s 时

有明显的振动主频, 与一阶管束固有频率相近, 易发生共振。其他工况下均显示在 10 ~ 100 Hz 的宽频范围内分布。其他测点在不同管间流速下脉动压力分布规律与管 4-1-90°相似。

3.3 空泡份额的影响

如图 7, 在管间流速 5 m/s 时, 管 4-1-90°位置处脉动压力功率谱密度峰值随空泡份额的变化是先增大后减小, 在空泡份额为 0.8 时存在最大峰值。此时脉动压力频率与传热管束固有频率相近, 结构易发生共振。

3.4 经验公式讨论

流体湍流中脉动变化的压力和速度场使管束振动。对湍流抖振频率计算, 目前公认采用的计算公式为 Owen 得到的半经验公式^[30]:

$$f_t = \frac{V_{\text{cross}} D}{LT} \left[3.05 \left(1 - \frac{D}{T} \right)^2 + 0.28 \right] \quad (4)$$

表 1 传热管在两相流中的固有频率

Tab. 1 Natural frequency of tubes in two-phase flow

空泡份额	阶数	频率/Hz(传热管按照弯管半径由小到大排列)					
		管6-1	管5-1	管5-2	管6-4	管6-7	管5-7
0.7	1阶	44.197	44.087	43.891	43.508	43.322	43.130
	2阶	50.181	49.865	49.712	49.510	49.332	49.187
0.8	1阶	44.932	44.844	44.643	44.217	43.509	43.284
	2阶	51.016	50.759	50.413	50.150	49.903	48.955
0.9	1阶	45.706	45.577	45.301	45.052	44.877	44.511
	2阶	51.894	50.964	50.534	50.210	49.920	49.330
0.98	1阶	46.521	46.254	45.920	45.750	45.510	45.119
	2阶	52.819	52.579	52.113	51.904	51.723	51.249

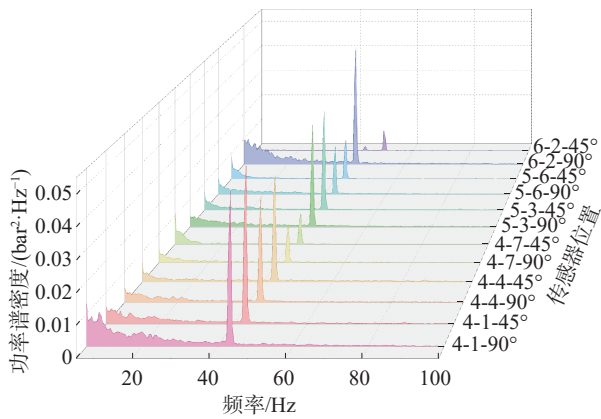


图 5 空泡份额 0.7, 管间流速 5 m/s 时不同测点的脉动压力功率谱密度图

Fig. 5 Power spectral density of fluctuating pressure at different measuring points with $\epsilon_g=0.7$ and $V_{cross}=5$ m/s

表 2 空泡份额 0.7, 管间流速 5 m/s 下不同测点部位的脉动压力主频及对应功率谱密度值

Tab. 2 Main frequency of fluctuating pressure and power spectral density at different measuring points with $\epsilon_g=0.7$ and $V_{cross}=5$ m/s

管束位置(按弯管半径从大到小排列)	4-1-90°	4-1-45°	5-3-90°	5-3-45°	5-6-90°	5-6-45°
脉动压力主频/Hz	43.457	43.457	43.457	43.457	43.457	43.457
功率谱密度值/($\text{bar}^2 \cdot \text{Hz}^{-1}$)	0.048	0.050	0.037	0.036	0.018	0.015

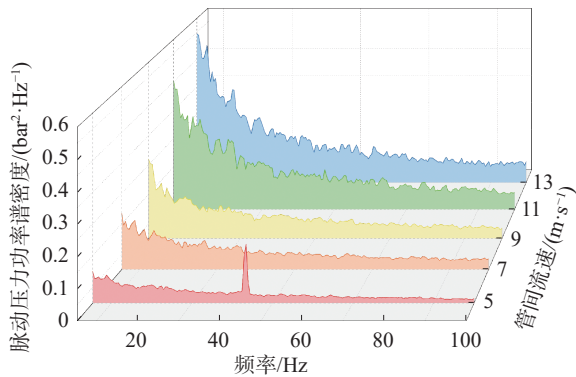


图 6 空泡份额 0.7, 在不同管间流速下管 4-1-90°测点处的脉动压力幅值图

Fig. 6 Amplitude of fluctuating pressure at measuring point 4-1-90° with $\epsilon_g=0.7$ and different V_{cross}

式中:

L ——纵向传热管中心距(mm);

T ——横向传热管中心距(mm)。

但是其关键经验系数 3.05 仅是对气体试验数据(即空泡份额为 1)拟合所得。当空泡份额为 0.7、

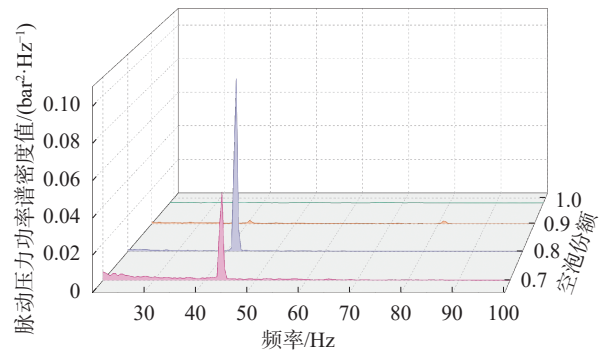


图 7 管间流速 5 m/s, 不同空泡份额下管 4-1-90°测点处的脉动压力功率谱密度图

Fig. 7 Power spectral density of fluctuating pressure at measuring point 4-1-90° with $V_{cross}=5$ m/s and different ϵ_g

0.98 时, 采用该公式进行计算与试验值有差距, 因此文章中通过调整经验系数值进行计算, 与试验结果实现了较好的拟合。如表 3 和表 4 对空泡份额 0.7

表 3 空泡份额 0.7, 湍流抖振频率计算值与试验值对比

Tab. 3 Comparison between the calculated value and the experimental value of turbulence vibration frequency at $\epsilon_g=0.7$.

管间流速/($\text{m} \cdot \text{s}^{-1}$)	试验值/Hz	计算值/Hz	
		经验系数=3.05	经验系数=2.8
5	43.457	44.822	43.017
6	45.410	47.511	45.598
7	49.316	52.441	50.330
8	59.248	61.854	59.363
9	64.453	67.232	64.525
10	68.359	71.266	68.397
11	93.750	98.608	94.637
12	98.632	102.193	98.079
13	99.450	103.986	99.799

表 4 空泡份额 0.98, 湍流抖振频率计算值与试验值对比

Tab. 4 Comparison between the calculated value and the experimental value of turbulence vibration frequency at $\epsilon_g=0.98$.

管间流速/($\text{m} \cdot \text{s}^{-1}$)	试验值/Hz	计算值/Hz	
		经验系数=3.05	经验系数=2.85
5	43.296	44.822	43.378
6	45.410	47.511	45.981
7	50.781	52.441	50.752
8	59.082	61.854	59.861
9	64.941	67.232	65.069
10	68.847	71.266	68.971
11	94.238	98.608	95.431
12	98.145	102.193	98.902
13	100.098	103.986	100.637

和 0.98 的计算值及试验值进行了汇总, 当经验系数为 3.05 时, 计算值平均比试验值大 4%, 当调整经验系数后, 计算值与试验值符合较好。说明在二次侧是两相流时, Owen 公式中的经验系数不再适用, 需要根据试验进行适当调整。

3.5 管束振动特性分析

对加速度计信号进行处理, 通过两次积分得到传热管的位移数据。如图 8 所示, 管束在 X 方向的振动有效值随管间流速的变化呈现出先增大到一定

峰值再降低随后又继续增大的特点。总体上, 空泡份额越大, 管束振动的有效值峰值越小。空泡份额 0.7~0.98 时, 管束振动有效值峰值分别出现在 6 m/s、10 m/s、10.2 m/s 和 11 m/s。随着空泡份额的增大, 出现峰值所对应的管间流速也越来越大。对应上述对脉动压力的分析, 可以看到出现峰值时对应的工况均是发生了管束共振, 当脉动压力频率与管束固有频率不一致时, 传热管又恢复了由流体激振为主导的强迫振动。

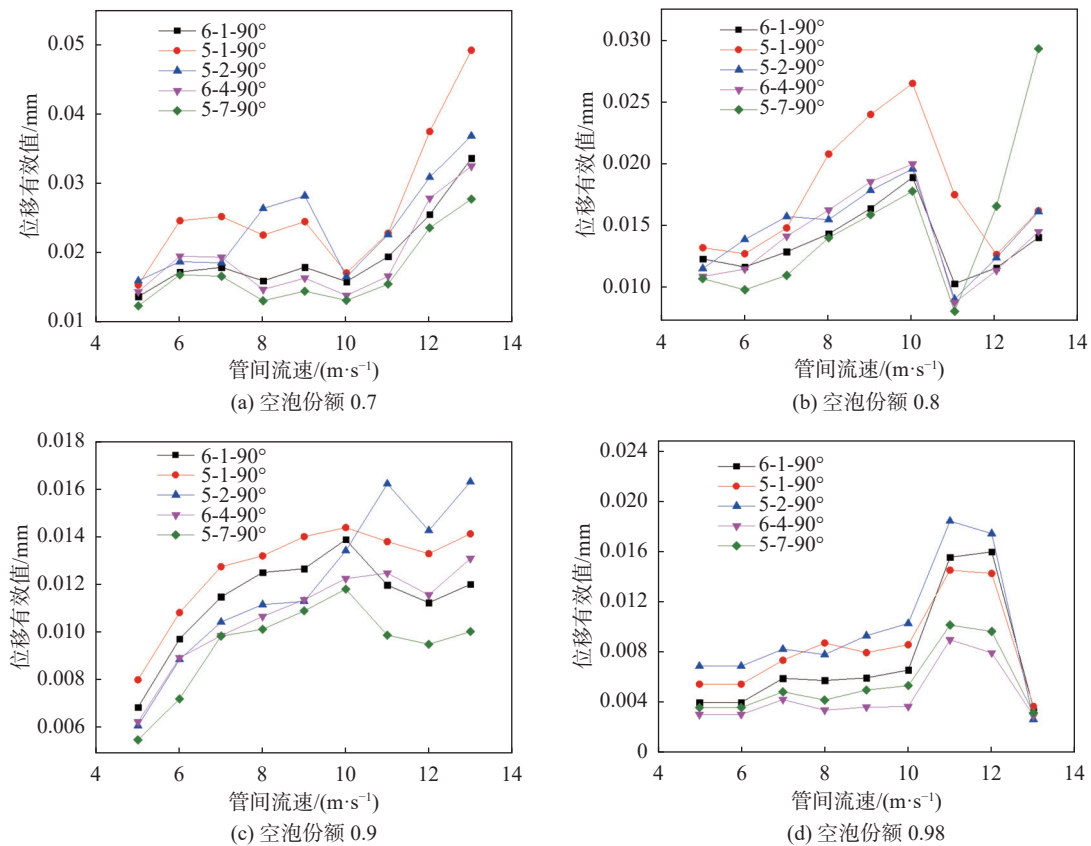


图 8 不同空泡份额下传热管束 X 方向(面内)振动位移有效值随管间流速变化曲线图

Fig. 8 Effective vibration displacement of tube bundle in X direction (in-plane) with different ε_g and V_{cross}

4 结论

文章设计了蒸汽发生器传热管束弯管区流致振动的试验装置, 利用空气-水两相流体模拟蒸汽发生器传热管束二次侧的不同的运行工况。测量了空泡份额从 0.7~0.98、管间流速从 5~13 m/s 变化时不同管束位置受到的流体脉动压力, 得到了管束振动特征, 并进行了详细分析, 得到如下结论:

1) 传热管束外侧处的流体激励相比内侧处有较

大衰减。

2) 在 5~7 m/s 的流速范围内, 流体脉动压力主频与管束的固有频率接近, 传热管易发生共振。

3) 流体脉动压力的频率分布范围比较宽, 随着管间流速的增大, 脉动压力幅值逐渐变大。

4) 采用经验公式计算得到的流体脉动压力主频比试验得到的值偏大, 调整了其中的经验系数后, 计算值与试验值更相符。

5) 管束的振幅随着空泡份额增大而减小, 发生

共振时管束的振幅会有较大的增长。

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