Journal of Fluids and Structures 45 (2014) 216–234

Contents lists available at ScienceDirect

Journal of Fluids and Structures

journal homepage: www.elsevier.com/locate/jfs

Pressure pulsations in piping system excited by a centrifugal turbomachinery taking the damping characteristics into consideration

I. Hayashi [a,*], S. Kaneko [b]

[a] Chiyoda Corporation, Minato Mirai Grand Central Tower 4-6-2, Minatomirai, Nishi-ku, Yokohama 220-8765, Japan
[b] University of Tokyo, 7-3-1, Hongo, Bunkyo-ku, Tokyo 113-8656, Japan

ARTICLE INFO

Article history:
Received 27 April 2011
Accepted 25 November 2013
Available online 21 January 2014

Keywords:
Pressure pulsation
Centrifugal compressor
Centrifugal fan
Centrifugal pump
Resonance
Blade passing frequency
Damping

ABSTRACT

Pressure pulsations excited by a centrifugal turbomachinery such as compressor, fan or pump at the blade passing frequency may cause severe noise and vibrations in piping system. Therefore, the practical evaluation method of pressure pulsations is strongly recommended. In particular, the maximum pressure amplitude under the resonant conditions should be appropriately evaluated. In this study, a one-dimensional excitation source model for a compressor or pump is introduced based on the equation of motion, so as to incorporate the non-linear damping proportional to velocity squared in the total piping system including the compressor or pump. The damping characteristics of the compressor or pump are investigated by using the semi-empirical model. It is shown that the resistance coefficient of the compressor or pump depends on the Reynolds number that is defined using the equivalent velocity of the pulsating flow. The frequency response of the pressure amplitude and the pressure distribution in the piping system can be evaluated by introducing the equivalent resistance of the compressor or pump and that of piping system. In particular, the relation of the maximum pressure amplitude in piping system to the location of the excitation source under resonant conditions can be evaluated. Finally, the reduction of the pressure pulsations by use of an orifice plate is discussed in terms of the pulsation energy loss.

1. Introduction

Severe noise and vibrations in piping systems are caused by pressure pulsations from a compressor or fan. In particular, the reduction of pressure pulsations at the blade passing frequency is important for noise and vibration control of a centrifugal compressor or fan. Extensive research (Embelton, 1963; Ploner and Herz, 1969; Weidemann, 1971; Smith et al., 1974; Neise, 1975; Ohta et al., 1987) has been reported since early times regarding blade passing sound/pulsation generated in a centrifugal fan, and similarity laws have been proposed (Neise, 1975). Regarding the properties of pressure pulsations from a centrifugal pump, which has the same excitation source as a centrifugal compressor, Morgenroth and Weaver (1998) showed the relation between the radius of the tongue and the generation of the sound. Parrondo-Gayo et al. (2002) reported that the pressure pulsation in the centrifugal pump becomes minimum, when the pump is operated at the design point on the performance curve. For the aforementioned research, the generation characteristics of the pressure pulsations by a

* Corresponding author. Tel.: +81 45 225 7198.
E-mail address: itsuro.hayashi@ykh.chiyoda.co.jp (I. Hayashi).

0889-9746/$ - see front matter
http://dx.doi.org/10.1016/j.jfluidstructs.2013.11.012

Nomenclature

A	cross-sectional area of a pipe [m^2]
A_c	equivalent area in a compressor or pump [m^2]
c	speed of sound [m/s]
D	pipe diameter [m]
D_{inp}	impeller diameter [m]
E	damping energy [N m]
f	frequency [Hz]
F	excitation force [N]
ℓ	length of a piping element [m]
$\tilde{m}$	mass flow rate [kg/s]
$\overline{m}$	average mass flow rate [kg/s]
m	fluctuation component of a mass flow rate [kg/s]
$\hat{m}$	equivalent mass flow rate [kg/s] (refer to Matsuda and Hayama, 1985)
p	fluctuation component of pressure [Pa]
p_a^*	pressure amplitude non-dimensionalized by dynamic pressure based on U [dimensionless]
Δp	pressure loss [Pa]
R	equivalent resistance [kg/m^2 s]
Re	Reynolds number $=\hat{u}D/\nu$ [dimensionless]
t	time [s]
T	cycle ($T=2\pi/\omega$) [s]
U	tip velocity of an impeller [m/s]
$\tilde{u}$	fluid velocity [m/s]
$\overline{u}$	average fluid velocity [m/s]
u	fluctuation component of a fluid velocity [m/s]
$\hat{u}$	equivalent velocity $=\hat{m}/\rho A$ [m/s]
x	coordinate along pipe [m]
Δx	length of an excitation part in a compressor or pump [m]
X_{pum}	pump position from a pipe inlet [m]
δ	acceptable calculation error [dimensionless]
ε	damping energy per area [N/m]
ζ	damping coefficient of concentrated resistance [dimensionless]
λ	friction factor for a pipe flow [dimensionless]
ρ	fluid density [kg/m^3]
ω	angular velocity [rad/s]

Subscripts

a	amplitude
c	excitation part in a compressor or pump
ca	amplitude at excitation part in a compressor or pump
d	discharge side of an excitation part in a compressor or pump
exp	experiment
img	imaginary part
p	one piping element
pa	amplitude at one piping element
pip	piping system which is composed of piping elements
r	concentrated resistance of pipe inlet, pipe outlet, valves and orifices
ra	amplitude at concentrated resistance of pipe inlet, pipe outlet, valves and orifices
$real$	real part
s	suction side of an excitation part in a compressor or pump
sim	simulation
t	total system including compressor or pump piping
α, β	condition of piping arrangement

compressor or fan itself have been clarified in terms of the design parameter of the equipment. However, the amplitude of the pressure pulsations in the piping systems needs to be evaluated by taking the response characteristics in the piping into consideration.

In order to evaluate the pressure response in the piping systems, the mechanism of the sound generation in a compressor or fan and the characteristics of the dynamic response in the piping systems have to be clarified. Chandrashekhara (1971) showed that the noise source from an axial fan in the piping systems is dominated by the dipole source, based on the empirical data. Margetts (1987) confirmed experimentally that the phase difference between the discharge side and the suction side of an axial fan is 180°, by using the anechoically terminated ducts. Regarding the simulation model of a fan, Cremer (1971) proposed Two-port model which is composed of the scattering matrix. Terao and Sekine (1992) and Lavrentzev et al. (1995) developed a method to determine the scattering matrix and the noise source for an axial fan in the ducts experimentally.

Regarding the properties of pressure pulsations from a centrifugal pump, which has the same excitation source as a centrifugal compressor, pressure pulsations are evaluated using a Two-port model including the transfer matrix, if the transfer matrix in the pump is known experimentally (Desmet and Barrand, 1986; Goto, 1988a,b; Bolleter, 1993). Rzentkowski and Zbroja (1997, 2000a,b) showed that the Two-port model is suitable for the analysis of actual pump–piping systems in industry. Although Two-port model is one of the most practical models to evaluate the pressure pulsations in the compressor–piping systems, the characteristics of the pressure response under resonant conditions cannot be accurately evaluated unless the damping characteristics of the piping systems including the compressor is specified.

For example, regarding the relation between pressure pulsations and pump position in a piping system, Sano (1984) reported that the pressure pulsation reaches a maximum when an excitation source is close to the node of the pressure distribution of the resonant mode. However, this phenomenon cannot be evaluated by the Two-port model unless the effect of the piping system is appropriately incorporated.