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COUPLING THE EFFECTS OF RECIPROCATING COMPRESSOR VALVE DYNAMICS WITH PIPING ACOUSTIC RESPONSE

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ABSTRACT

Pulsations generated by flow through reciprocating compressor valves are transmitted throughout the attached piping system and frequently lead to excessive piping vibration. Simultaneously, acoustic piping response influences the motion of the compressor valves, thereby affecting flow capacity and valve life. This is particularly true at higher harmonics of compressor run speed. Therefore, to accurately predict pulsation at higher harmonics, the interaction between piping acoustics and valve dynamics must be considered.

This paper describes an efficient technique to link the results of frequency domain modelling of the piping acoustics with time domain modelling of the compressor valve dynamics. The validity of the technique is confirmed by good agreement between measured and predicted results. In addition, this paper confirms the feasibility of using computational fluid dynamics to determine compressor valve flow characteristics.

NOMENCLATURE

A_e = effective force area
A_f = valve flow area
A_l = valve plate area
A_p = piston area
c = damping coefficient
C_d = valve discharge coefficient
F = spring force
h = valve plate position
k = spring stiffness
L = connecting rod length
m = mass
n = polytropic exponent
p_i = total pressure pulsation at "i"
p_{ij} = pressure pulsation at "i" caused by source volume flow velocity at "j"
P = pressure
P_i = total pressure at "i"
P_{iave} = mean pressure at "i"
q = actual flow through valve
R = crank radius
Re = Reynolds number
t = time
v_j = volume flow velocity at valve "j"
V = cylinder volume
V_{cl} = cylinder clearance volume
$\bar{z}_{ij}$ = ratio of pressure pulsation at "i" to volume flow velocity at "j"
$[\bar{Z}]$ = equivalent system matrix
ΔP = pressure differential across valve
ρ = gas density
θ_p = pressure pulsation phase angle
ω = circular frequency

<u>Subsripts</u>

c = cylinder
d = discharge
max = values when valve fully open
o = values when valve closed
s = suction

INTRODUCTION

Reciprocating compressor valves not only alter the fluctuating flow produced by compressor piston motion, but also interact dynamically with the attached piping system. Numerous approaches dealing with the interaction of valve dynamics and piping response have been presented. Very simple piping systems have successfully been modelled using lumped parameter approaches[1,2]. For more complex piping systems, distributed parameter approaches have been used. For example, the impedance[3], transfer matrix[4], modal[5], and characteristics[6] methods have all been discussed previously. However, when using the previously presented approaches, computational effort increases dramatically with increasing piping complexity.

In this paper, the transfer matrix approach is used to model any arbitrarily complex piping network. Equivalent system matrices (having the units of acoustic impedance) characterize the response of an entire piping system with multiple compressor valve pulsation sources. The matrices are then used in the solution of the compressor valve dynamics, and hence account for accurate piping response during the modelling procedure. The paper describes this coupling technique in more detail.

Valve flow characteristics and effective force areas are critical in the simulation of the valve dynamics. These data are generally not available, and many analysts have attempted to generate them by experiment or numerically. For this paper, the flow coefficients and the effective force area at various valve openings were determined by computational fluid dynamics.

A reciprocating compressor station in NOVA's gas transmission system was modelled to demonstrate the validity of the proposed coupling technique. Good agreement was obtained between measured and predicted results.

MODELLING THE VALVE DYNAMICS

Equations describing the valve motion and the gas dynamics within the cylinder are solved simultaneously to determine the dynamic response of the compressor valves. Valve motion is modelled with the following second order differential equation (1):

$$m(h)\ddot{h} + c(h)\dot{h} + k(h)h + F_o = A_c(h)\Delta P(t) \tag{1}$$

Note the valve mass, damping, stiffness and effective force area are all functions of the valve position. The pressure differential, $\Delta P(t)$, across the valve is calculated using equations (2a) and (2b) for suction and discharge valves, respectively:

$$\Delta P_s(t) = P_s(t) - P_c(t) \quad ; \quad \Delta P_d(t) = P_c(t) - P_d(t) \tag{2 a,b}$$

The following equations are applied to solve various aspects of the cylinder gas dynamics:

Thermodynamics:

$$\dot{P}_c = \frac{nP_c(q_s - q_d - \dot{V})}{V} \tag{3}$$

Valve Flow Equations:

$$q_s = C_{ds}(h)A_{fs}(h)\sqrt{\frac{2\Delta P_s}{\rho_s}} \quad ; \quad q_d = C_{dd}(h)A_{fd}(h)\sqrt{\frac{2\Delta P_d}{\rho_c}} \tag{4 a,b}$$

Piston Kinematics:

$$V = A_p\left[L\left(1 - \sqrt{1 - (\frac{R}{L}\sin(\omega t))^2}\right) + R(1 - \cos(\omega t))\right] + V_{cl} \tag{5}$$

$$\dot{V} = A_p R\omega \sin(\omega t)\left[1 + \left(\frac{(\frac{R}{L}\cos(\omega t))}{\sqrt{1 - (\frac{R}{L}\sin(\omega t))^2}}\right)\right] \tag{6}$$

The above equations are formulated into an initial value problem, and solved numerically using a stabilized four stage Runge-Kutta procedure. Solution of the above equations is only possible if the suction and discharge pressures ($P_s(t)$, $P_d(t)$) are known a priori. The determination of these pressures is described later.

MODELLING THE PIPING ACOUSTIC RESPONSE

Acoustic response of the reciprocating compressor piping is predicted by the NOVA computer program PULS. In this program, transfer matrices relate pulsating pressures and volume flow velocities at the ends of each piping element. Different types of piping elements are modelled with unique transfer matrices[7,8,9]. For piping elements such as pipes and orifice plates, the effects of mean flow are included. The transfer matrices for all the piping elements are assembled into an overall system matrix. Boundary conditions, such as closed ends, infinite pipe lengths, and reciprocating compressor valves, are then defined for the model. The resulting system of equations is then solved for selected frequencies. For valve dynamics calculations, the frequencies of interest correspond to all significant harmonics of compressor run speed.

In the acoustic model, the pulsating pressure at each compressor valve is influenced by the pulsating volume flow velocity at all of the valves. The total pulsating pressure at each valve is therefore the sum of pulsating pressure resulting from each valve source volume flow velocity. The ratio of pulsating pressure at valve 'i' to the pulsating volume flow velocity at another valve 'j' is constant for a given piping layout, operating condition and frequency. This constant ratio has the units of acoustic impedance and will be referred to as:

$$\tilde{z}_{ij} = \frac{p_{ij}}{v_j} \tag{7}$$

In the above equation 'i' and 'j' represent valve locations. The total pulsation at valve 'i' is then the sum of the products of all $\tilde{z}_{ij}$ and valve source volume flow velocities:

$$p_i = \sum_{j=1}^{n} \tilde{z}_{ij} \times v_j \tag{8}$$

where n is the number of valve sources

For a given frequency, the total pressure pulsation for all valves can be obtained from the product of the n x n equivalent system matrix $[\tilde{Z}]$ and the vector of valve pulsating volume flow velocities:

$$[p_i] = [\tilde{Z}][v_j] \quad ; \quad [\tilde{Z}] = [\tilde{z}_{ij}] \tag{9}$$

where i and j vary from 1 to n valve sources

For a given compressor speed, the overall pressure time trace at a valve can be obtained by adding the pressure time traces of all significant harmonics. This makes it possible to completely characterize the acoustic response at the valves of an arbitrarily complex piping system by the relatively compact equivalent system matrices.

COUPLING ACOUSTIC PIPING RESPONSE AND VALVE DYNAMICS

There are three steps involved in coupling the valve dynamics with the acoustic piping response. In the first step the equivalent system matrices $[\bar{Z}]$ are determined as discussed above. In the second step the valve dynamics calculations are performed. The effects of acoustic piping response are included in the valve dynamics calculations via the following equation:

$$P_i(t) = P_{iave} + [\sum_{k=1}^{nh} [\sum_{j=1}^{n} | \tilde{z}_{ij} \, x \, v_j | \cos(\omega t - \theta_p)]] \qquad (10)$$

where n is number of valve sources
nh is number of significant harmonics

Finally, the acoustic piping response is determined based on pulsating volume flow velocities from the valve dynamics calculations.

The first and last calculation steps are performed in the frequency domain by the acoustic simulator. The second step is performed in the time domain by the valve dynamics simulator. A flowchart of the calculation procedure is given in Figure 1.

EXAMPLE PROBLEM

Station Description and Operating Conditions

The station described in this paper is part of NOVA's gas transmission system, and has two single stage units. Only the second unit was operated during testing. The tested compressor has two double acting cylinders and all of the cylinder ends were loaded for the test. Therefore, all four cylinder ends were sources of pulsation for both the suction and discharge piping systems.

Figure 2 shows an isometric of the suction piping acoustic model, and Figure 3 shows an isometric of the discharge model. Model nodes corresponding to the testpoint locations are shown on the Figures. Tables 1-3 summarize the compressor geometry, gas composition and test operating conditions, respectively.

Valve Characteristics

The original compressor valves were Hoerbiger plate valves. For the test, all the valves of cylinder 1 were replaced by Dresser-Rand model 60B channel valves. The original plate valves remained in cylinder 2 during the test.

Both the suction and discharge channel valves have seven slots. Each slot contains an independent plate, spring and inlet flow path. The dimensions, weights and spring stiffnesses were measured for each slot and computational fluid dynamics modelling was used to determine the flow characteristics. For modelling purposes, the measured data from all the slots were combined into single equivalent values. Table 4 gives the equivalent dimensions and flow characteristic values used in the simulation.

The Hoerbiger plate valves are the older 157CGD style with a steel valve plate and two damping plates. The data shown in Table 5 was used to simulate the plate valves.

PREDICTED RESULTS VERSUS MEASURED DATA

Measured data was collected for compressor speeds ranging from 450 to 550 rpm. Predicted pulsations are compared with those measured at various locations within the cylinder passages and in the suction and discharge piping. Pulsating pressure spectra for both the suction and discharge systems are shown in Figures 4 and 5, respectively. Plots on the left hand side of each figure show the measured values, while those on the right hand side show the corresponding predicted values. The agreement between measured and predicted values is generally very good.

Trends in predicted pulsation are confirmed by comparison with measured data. The predicted pulsations for the suction system are very consistent with the measured data, except for the sixth harmonic where a predicted resonance results in higher predicted pulsations than measured. For the discharge system, predicted pulsation is generally higher for even harmonics and lower for odd harmonics than the measured data. The greater consistency for the suction system predictions is most likely due to the very stable suction temperature during testing. In contrast, the discharge temperature was sensitive to varying pipeline conditions during the test.

Overall, the agreement between predicted pulsation and measured data confirms the validity of both the coupling technique and the use of computational fluid dynamics to determine valve flow characteristics.

CONCLUDING REMARKS

1. A compact and efficient coupling technique has been developed to link time-domain valve dynamics calculations to frequency domain piping acoustics calculations. An equivalent system matrix was identified to effect this link for any arbitrarily complex piping system.

2. Computational fluid dynamics can be utilized to determine compressor valve characteristics that are difficult to obtain by direct measurement. An example of this is the effective force area of the valves.

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Table 1: Compressor Geometry

Stroke, 2R [m]	0.2286
Connecting rod length, L [m]	0.7366
Pushrod diameter [m]	0.1143
Bore [m]	0.3302
Decimal clearance: Head End	0.3574
Crank End	0.2251

Table 2: Gas Composition

COMPONENT	PERCENTAGE
Methane	91.96
Ethane	1.42
Propane	0.1
Butane+	0.2
Nitrogen	5.81
Carbon Dioxide	0.51

Notes 1 The dimensions are the same for both cylinders.
2 The crank and cylinder arrangement is such that both pistons are fully extended into the head ends of their respective cylinders at the same time.

Table 3: Test Operating Conditions

Spead [RPM]	Gauge Pressure		Temperature	
	Suction [kPa]	Discharge [kPa]	Suction [°C]	Discharge [°C]
450	2249	6690	7.0	107
470	2246	6725	7.0	107
490	2235	6840	7.0	109
510	2354	6626	7.0	100
520	2413	6568	7.0	100
530	2433	6552	7.0	100
550	2372	6494	7.0	100

Table 4: Channel Valve Characteristics

Valve Position	Suction Valve		Discharge Valve	
	Closed	Open	Closed	Open
Spring deflection, h [m]	0.004322	0.006761	0.004518	0.006554
Valve mass[1], m [kg]	0.188	0.250	0.186	0.248
Spring stiffness[1], k [N/m]	24751	24795	25328	25384
Damping, c [kg/s]	1% of critical damping			
Normalized Valve Position[2]	Suction Valve		Discharge Valve	
	C_{ds}	A_{es}/A_{ls}	C_{dd}	A_{ed}/A_{ld}
0.2	0.0760	0.717	0.0944	0.724
0.4	0.1759	0.674	0.1881	0.665
0.6	0.2641	0.712	0.2819	0.702
0.8	0.3564	0.789	0.3637	0.753
1.0	0.4126	0.874	0.4238	0.825
Flow area, A_f [m^2]	0.006734		0.006814	
Plate area, A_l [m^2]	0.0116		0.0097	

Notes 1 Values for intermediate valve positions are obtained by linear interpolation.
2 Normalized Position = $(h-h_o)/(h_{max} - h_o)$

Table 5: Hoerbiger Plate Valve Characteristics

Valve Position	Suction Valve		Discharge Valve	
	Closed	Open	Closed	Open
Spring deflection, h [m]	0.00512	0.00738	0.00512	0.00738
Valve mass[1], m [kg]	0.210	0.606	0.207	0.606
Spring stiffness[1], k [N/m]	34136	112302	34850	115706
Flow coefficient[2,3], Cd	0	0.3	0	0.3
Damping, c [kg/s]	1% of critical damping			
Force area, A_c [m^2]	0.00750			
Flow area[3], A_f [m^2]	0.00521			

Notes 1 Values change when the valve is in the 68% open position.
2 Values for intermediate valve positions are obtained by linear interpolation.
3 These values are based on Vendor supplied information.

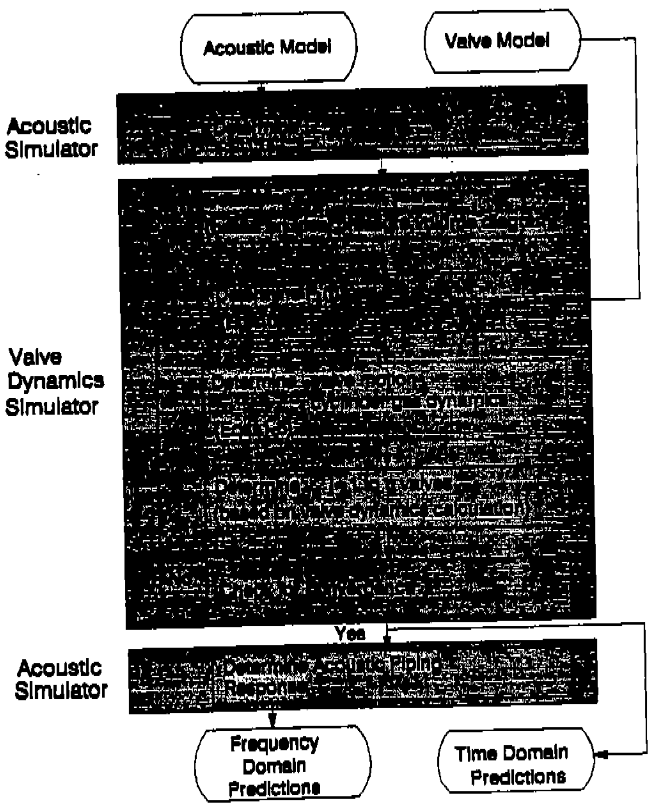

Figure 1: Flowchart of Calculation Procedures

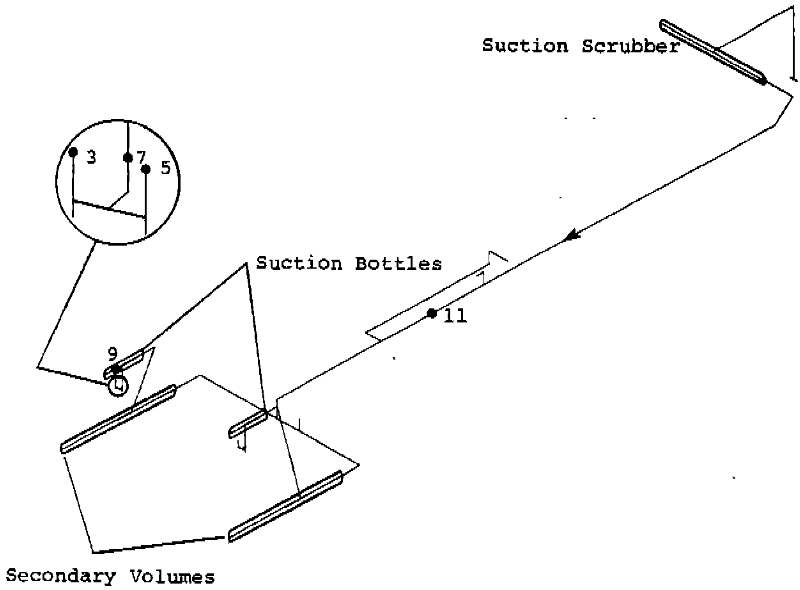

Figure 2: Isometric Drawing of Suction Piping Acoustic Model

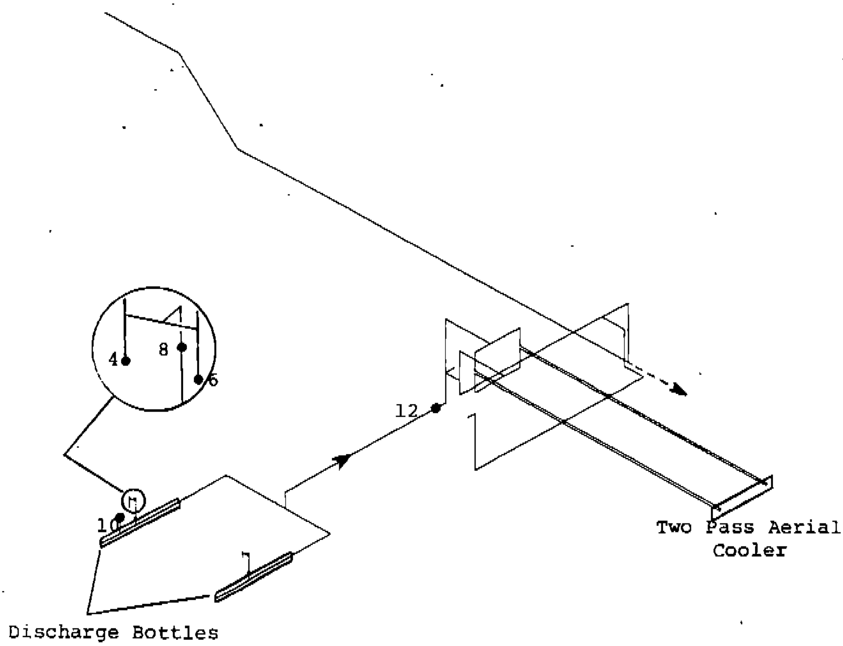

Figure 3: Isometric Drawing of Discharge Piping Acoustic Model

Figure 4: Suction Pulsation Spectra

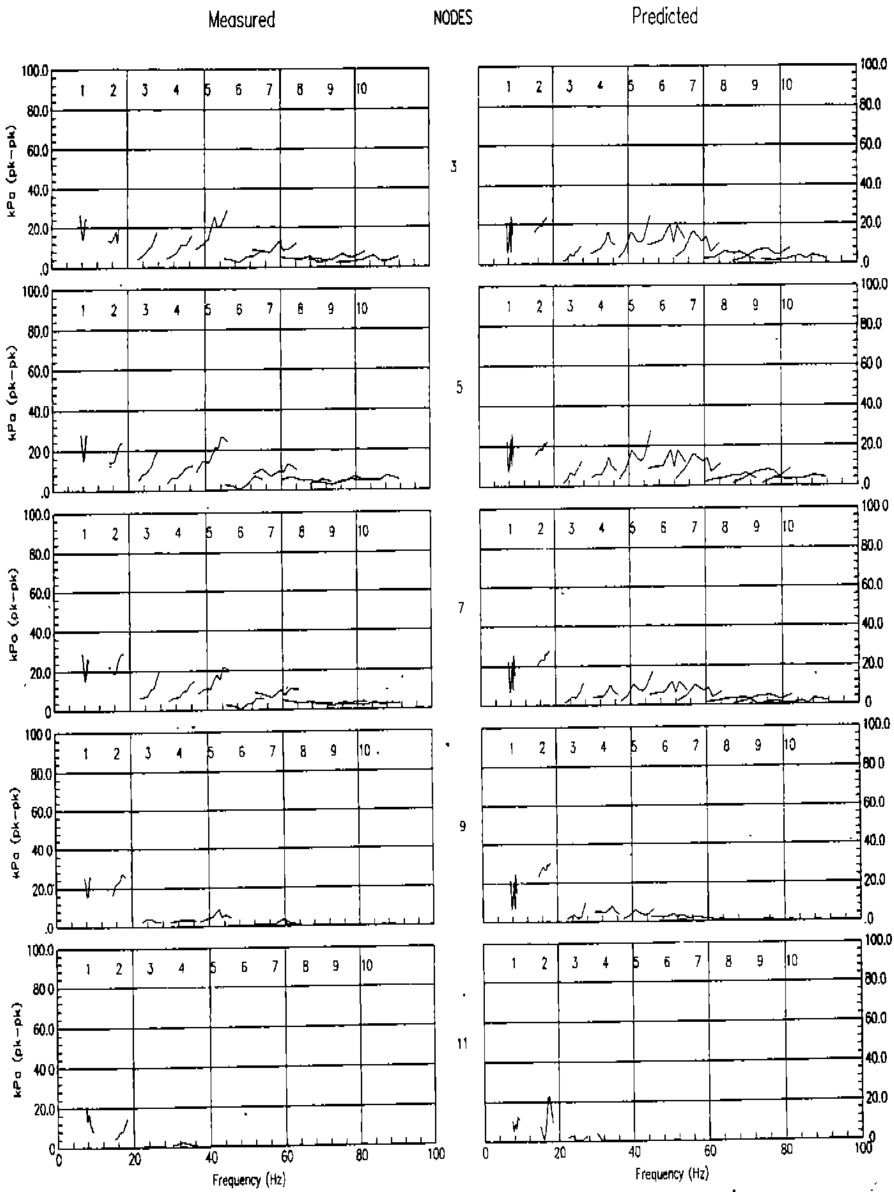

Figure 5: Discharge Pulsation Spectra