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INTRODUCTION

Gas pulsations and oscillations are inherent to the suction and discharge processes of compressors. For this reason many compressor designers have been concerned with their control and optimization. In many cases trial and error methods which involve compressor component variations are employed to alleviate undesirable pulsation problems. However, this approach may sometimes be quite expensive and seldom leads to an optimum solution. In the following, research efforts are reviewed which constitute a positive step away from the trial and error procedure.

When valves open and gas rapidly enters or leaves one of the compressor plenums, the pressure in the plenum and associated piping is altered. The net effect of this process is a pressure distribution in the plenum and piping which is a function of both time and specified spatial coordinates. Basically this pressure distribution can be interpreted as the superposition of a pressure pulse and high frequency pressure oscillations which are indicative of acoustic resonances in the piping system. Both pressure pulsations and oscillations are dependent on valve mass flow and plenum and piping geometry. Valve mass flow is in turn affected by pressure pulsations and oscillations in the plenum. Thus, an accurate prediction of pressure pulsations and oscillations cannot be expected without first including valve response in the calculation procedure. To do this, a mathematical simulation of the compressor and piping system is generally required and will be discussed herein.

Compressor gas pulsations and oscillations may excite piping vibration, influence thermodynamic performance, and serve as a source for noise radiation. Compressor piping vibrations are dependent upon pulsations; however, the reverse is generally not true. To a large extent, piping vibrations may be minimized by insuring that no mechanical resonances of the piping system occur at frequencies corresponding to multiples of the compressor running frequency.

The importance of dynamic supercharging in the suction line of a reciprocating compressor was investigated experimentally by Czaplinski (7). In his results, as well as those of others cited by him, compressor cyclic mass flow was increased by as much as 30% by varying the length of the compressor suction line. Similarly, Stein and Eibling (20) showed improved compressor performance by tuning the discharge side of the compressor. Thus, significant improvements in compressor thermodynamic performance may be obtained by an optimized design of the piping system.

High frequency gas oscillations which execute several or more cycles during the open time of the valve, have little effect on the thermodynamic performance of the compressor. They may, however, be significant contributors to compressor sound radiation. In this regard Johnson (14), from his experimental investigation of noise sources in a rotary vane compressor, concluded that gas oscillations in the compressor discharge plenum may be significant contributors to compressor radiated sound. Particularly this is true if the gas oscillations occur at a frequency corresponding to a strong harmonic of the cyclic volume velocity flow through the valve.

GAS OSCILLATION THEORY

Many investigations have, in the past, been concerned with the mathematical prediction of pressure pulsations in compressors. Considerable research related to this problem has also occurred in the internal combustion engine and pneumatic control industries. Virtually all this research has been based on one-dimensional models for the compressor piping system. Thus, in the following the pulsation or unsteady flow theory discussed will be restricted to one dimension.

In selecting an appropriate theory for predicting pressure pulsations, consideration should first be given to the magnitude of the pressure pulse relative to the mean pressure of the piping system, $\Delta P/P_o$. For $(\Delta P/P_o) << 1$, linear or plane wave acoustic theory may be employed in the calculations. For larger values of $\Delta P/P_o$, nonlinear terms in the one-dimensional continuity and momentum equations may be required. The value of $\Delta P/P_o$ for which acoustic theory is no longer valid is not well defined and generally is dependent on how much accuracy is required. However, Groth (11) in citing other investigators states that acoustic theory may be employed for values of $\Delta P/P_o$ as great as 0.20. Elson (8) also has recently obtained good computed-experimental correlation for compressor discharge plenum pressure oscillations when $\Delta P/P_o$ was as large as 0.15. Based on these results, acoustic theory appears to be valid for many compressor pulsation calculations.

Compressor pulsation levels have been calculated by both transient and steady state means. The transient approach is the most versatile in that the gas oscillation equations may be readily coupled with compressor valve equations. This allows a simultaneous solution for valve displacement, instantaneous volume velocity flow through the valve, and pressure pulsation. A disadvantage of this approach is that it requires the compressor piping system to be broken into a series of elements or stations at which pressure pulsation levels can be calculated. For a complicated piping system, this lumped parameter approach may require a prohibitive number of stations to achieve desired accuracy at high frequencies. The steady state approach allows the piping system to be replaced by an equivalent impedance function which relates system pressure harmonics to the harmonics of the volume velocity flow through the valve. It is restricted by the fact that the input volume velocity function must be known before pulsation amplitudes can be calculated. For best results the volume velocity flow through the valve should be calculated simultaneously with pulsation pressure. Many investigators have, however, assumed simplified forms for the volume velocity function in order to employ steady state theory (1,12,15).

In view of the above discussion, a review of gas oscillation research contributions in piping systems will be given under the general categories of transient and steady state.

Transient

Grover (12) applied finite difference techniques to two linear partial differential equations which represented continuity and momentum conditions in a section of piping and were equivalent to the acoustic wave equation. To account for wave attenuation due to pipe friction, a velocity proportional term, indicative of a quasi-steady laminar flow assumption, was included in the momentum equation. After generating a set of first order ordinary differential equations for his piping system, a transient solution was obtained which became a steady state solution after several cycles of an assumed input volume velocity function. It should be noted that for many high speed compressors, quasi-steady turbulent flow should be assumed and in this case a friction term proportional to the square of velocity should be included in the momentum equation.

If large pressure pulsations are present such that acoustic theory can no longer be assumed, nonlinear forms of the one-dimensional continuity and momentum equations must be employed. Funk and Robe (9), in their investigations of large amplitude pressure pulses in pneumatic transmission lines, use a finite difference scheme similar to that used by Grover except that a spatial variation of density term and a convective acceleration term have been included respectively in the continuity and momentum equations. Also a general pipe friction term dependent on Reynolds' number is included to account for both laminar and turbulent flow. A series of ordinary differential equations are obtained which may then be solved by standard integration routines for first order differential equations. An alternate approach to solving for large amplitude pressure pulses may be found in the method of characteristics. Benson (2), in a review of non-steady flow research, shows an application of this method to a centrifugal compressor.

Steady State

Steady state analyses of compressor piping systems include those of Wallace (21), Miller and Hatten (15), and Abe, Fujikawa, and Ito (1). Wallace employed mainly lumped parameter theory in his analysis of the intake process of large, low speed reciprocating compressors. Both single and double expansion chambers were investigated as well as the use of venturis as low loss acoustic impedances. Miller and Hatten compared lumped parameter and distributed element methods for calculating dynamic attenuation for both the suction and discharge mufflers of a refrigeration compressor. The lumped parameter approach was shown to be inadequate for determining high frequency resonances which may contribute to noise radiation. In the distributed element calculations, a steady state wave equation solution was used to relate pressure and volume velocity at various stations in the compressor piping system. Muffler attenuation was obtained relative to an experimental compressor system rather than relative to anechoic exits as is often done in muffler analysis (13). A distributed element approach was also used by Abe, Fujikawa, and Ito (1); however, a damping term was added to the wave equation to account for pipe friction. This modified the assumed wave motion by the factor, $e^{\pm \alpha x}$, where α represented an attenuation constant. Binder's modification (3) to the classical Kirchhoff attenuation theory was employed by these investigators to account for turbulent effects associated with large amplitude pressure waves. A variation of the transfer matrix method was also used to relate pressure and volume velocity at various stations along the compressor piping system. With this method, only 12 coefficients were required whereas 16 are needed with the usual transfer matrix method. References 1 and 15 should be consulted in evaluating the relative merits of these two methods.

Gatley and Cohen (10) present a general method for the design of mufflers in small refrigeration systems. Plane wave acoustic theory is combined with measured reflection and transmission factors to predict muffler attenuation relative to an operating compressor system. An attenuation factor, α, is employed in the calculations to account for pipe friction. An important feature of their method is that experimental techniques are employed in determining reflection and transmission factors for mufflers with complicated internal geometries. Plane wave theory is generally unacceptable for this type of muffler analysis due to the presence of three dimensional effects. In this regard Igarashi and Toyama (13) and Miwa and Igarashi (16) present an experimental technique for measuring "terminal constants" which relate input and output volume velocities and pressures for muffler elements. Differences obtained by these investigators between measured "terminal constants" and those estimated from plane wave theory illustrate the necessity of experimentally determining the attenuation characteristics of some muffler elements. Thus, both the experimental techniques discussed above may be extremely important to realistic muffler design.

COMPRESSOR SIMULATIONS

The important influence of valve motion on the resulting pressure oscillations in compressor piping systems was previously emphasized. In the following, compressor simulations will be discussed which provide for the simultaneous solution of valve displacement and pressure oscillation.

An analog computer was used by Brunner (6) to simulate a large reciprocating compressor used in the gas industry. In addition to kinematic, thermodynamic, and valve motion equations, his model included a simulation of pressure pulsations in the associated compressor piping. For this purpose a lumped parameter representation of the acoustic wave equation was utilized in which a velocity proportional damping term was added. His valve modeling, however, assumed valve lift to be a function only of the pressure drop across the valve and as such did not include the dynamics of the valve. Compressor pulsations were shown by this model to be important to compressor efficiency.

Nimitz (17) discusses a compressor simulation system in which the physical behavior of individual compressor components is simulated with equivalent electrical circuits. The details of the mathematical modeling are not given; however, reference is given to previous technical descriptions.

Brablik (4,5) states that good agreement between theory and experiment can only be obtained when the calculation of valve motion is obtained simultaneously with a solution for gas pulsations in piping. No equations are given for his simulation of valve motion; however, a finite difference approximation for the acoustic wave equation is used to simulate the gas piping system. A damping term proportional to the square of velocity is added somewhat artifically to an equation relating pressure at various stations in the piping. The effects of pulsations on valve life are discussed.

Compressor simulations involving the modeling of plenum pressure fluctuations have been performed recently by Padilla (18), Schwerzler (19), and Elson (8). In all these investigations, a digital computer was necessary to simulate various nonlinear functions associated with the valve models employed. The two cylinder reciprocating compressor used by Padilla contained three interconnected discharge plenums; one for each cylinder and one common to both cylinder plenums. A superposition of two phenomena was assumed in calculating the pressure variations in the plenums. The first involved pressure changes due to quasi-steady mass flow in and out of the plenums. The second involved dynamic pressure variations due to mass oscillation in the interconnecting passageways. A lumped parameter model which represented the plenums and interconnecting passageways as equivalent springs and masses was used here.

Schwerzler, in his investigation of a 50hp, three cylinder ammonia compressor, modeled pressure variations in suction and discharge plenums which were common to all three cylinders. To include heat transfer effects in the plenums, both a conservation of mass equation and the first law of thermodynamics were used to define pressure variations. To control the mass flow into the suction plenum and out of the discharge plenum, a hypothetical orifice was included at the suction plenum inlet and at the discharge plenum outlet. In this way, system components external to the compressor were replaced by an effective impedance.

A reciprocating compressor with a simplified discharge geometry was used by Elson to allow the use of one-dimensional theory in calculating pressure fluctuations in a discharge plenum. Two types of mathematical models were considered for the discharge plenum. One type combined a single volume approximateion for the discharge plenum with a steady state solution to the acoustic wave equation. The single volume approximation was used to calculate an approximate volume velocity flow through the valve while the steady state solution allowed the calculation of pressure harmonics in the plenum in terms of harmonics of the volume velocity function. With the other model, a finite difference approximation to the acoustic wave equation was used to directly calculate pressure oscillations in the plenum. Both methods predicted pressure harmonics which agreed well with corresponding experimental results. Slightly better accuracy was obtained with the finite difference approach but the single volume method was more efficient.

CONCLUSIONS

1. The calculation of gas pulsations and oscillations in compressor piping systems should involve the simultaneous solution for valve motion and volume velocity flow through the valve (4,8).
2. Linear or small pertubation theory provides sufficient accuracy for the calculations of pressure fluctuations in most compressor piping systems (8,11).
3. Compressor simulations to date have utilized either lumped parameter or finite difference techniques to simulate gas oscillations (4,5,6,8,18,19). These methods have the disadvantage of requiring numerious elements to properly simulate most physical systems.
4. Steady state methods allow the efficient computation of compressor pressure fluctuations if the input volume velocity flow through the valve is known. Several experimental techniques have been developed to measure steady state characteristics of compressor piping elements (10,16). For complicated muffler elements experimental methods may be the best approach for determining steady state characteristics.
5. Steady state methods with the exception of (8) have not been employed in compressor simulations because the nonlinearities of valve simulation require a transient solution approach. If may, however, be possible to converge on pressure harmonics by using an approach similar to that used in (8). In this regard an estimate of volume velocity flow through the valves might first be obtained by a compressor simulation employing simplified lumped parameter models for the compressor piping systems. Then steady state impedance functions may be developed (or determined experimentally) and used to convert volume velocity harmonics into pressure harmonics at the compressor valves. Pressure-time functions may next be calculated and used in the next compressor simulation cycle to calculate new volume velocity functions for the valves. This process could then be repeated until calculated pressure harmonics no longer change. Whether or not convergence can be obtained by this technique is presently being investigated for the simplified compressor system described in (8).

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