1974

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THE PIPELINE VIBRATION PROBLEM

A typical large natural gas compressor installation may employ multiple reciprocating compressors with total horsepower requirements in excess of 50,000 horsepower. Such installations also include a complex array of pipes and valves, as well as accessory equipment such as pumps, coolers and tanks. Equipment components in these systems are generally hard mounted to massive concrete reaction masses, but may in some cases be vibration isolated from the concrete masses by rubber pads.

When operated within certain limits dictated by various service and safety constraints, these installations may be reasonably quiescent. However, experience has shown that structural vibrations of significant magnitude can occur in the pipelines (loading lines) under various operating conditions, including normal design operation.

If these high amplitude vibrations are sustained, they often lead to problems such as misalignment, equipment malfunction, and structural failures which are very costly in terms of down-time and component replacement. As such, structural vibrations of natural gas compressor installations are highly undesirable and have a significant influence on the operational reliability, maintenance and safety of these installations. In the extreme, vibration induced failures of compressor components have been catastrophic in a few cases.

The technical literature is replete with design methodologies for reducing the likelihood of severe vibrations and the associated failures. However, these methods are less than totally successful, partly because of the inherent complexities of piping systems, and partly because some of their dynamic parameters (such as the boundary conditions and the structural damping) cannot be predicted reliably. In view of the limited reliability of the design methods, and also because actual installations generally do not conform precisely to the well analyzed initial design, one still often encounters significant loading line vibrations in the field.

LIMITATIONS OF PULSATION DAMPERS

Because of the oscillatory piston action of reciprocating compressors, pressure pulses are superimposed on the flow of gas from the compressor into the loading lines. Since the piston motion is not strictly a sinusoidal motion, these pressure pulses contain a band of frequencies. Additional frequencies within the gas result from pressure wave reflections at valves, pipe bends and other impedance discontinuities in the flow. The resulting oscillatory gas flow generates forced oscillations of the loading lines; and when the frequencies of the pressure oscillations coincide with, or are nearly equal to, resonance frequencies of the piping system, resonant buildup of pipe vibrations can occur. Resonant amplitudes of the pipe are even higher when driven by resonances within the gas.

In order to minimize the amplitudes of pulsation-induced pipe vibrations, it is common practice to insert pulsation dampers in the loading lines. These dampers act essentially like acoustic filters which remove a specific narrow band of frequencies from the troublesome gas pressure pulses, thus decreasing the amplitudes of the pressure pulses and the attendant pipe vibrations. Pulsation dampers must be designed to match the pressure pulse characteristics, operating speeds, and gas pressures and temperatures. For a given installation, these operational parameters are often variable because of changes in gas pressures at the source, environmental temperature fluctuations, and consumer demands. In fact, reciprocating compressors are often used because of their flexibility with regards to operating modes. Thus, a damper that performs well for one set of operating conditions may be less effective, or even ineffective, for other sets of operating conditions.

MODIFICATION AND ADDITION OF PIPING SUPPORTS

Many situations are encountered wherein

pulsation-induced loading line vibrations are significant, even if pulsation dampers are present. The modification of existing pipe supports and/or the introduction of additional supports then is usually the most practical means for achieving the desired reduction in vibrations. Towards this goal, it is necessary to determine design requirements for modifying or adding piping supports. These requirements are usually specified in terms of support stiffness and mass, support location, and direction(s) of piping constraint. Unfortunately, these requirements are not easily obtained. Computer modeling of the piping system can be only of limited help because of the aforementioned limited quantitative information regarding key parameters of the system. Realistic values for certain of these parameters can be obtained only from dynamic field measurements on the actual system.

In the field it often turns out that the existing operational conditions do not permit the engineer to obtain the compressor operating modes required for appropriate dynamic testing and diagnosis of troublesome vibration problems. It is then necessary to make the best estimate derivable on the basis of whatever data can be obtained under the existing operational conditions - an approach that is less than desirable if the first solution to be implemented must be adequate. It is the purpose of the following discussion to suggest an alternate approach to obtaining the dynamic data required for an adequate diagnosis and resolution of the vibration problem of interest.

NEW DYNAMIC TEST APPROACH

Equipment

The suggested dynamic test approach makes use of an external source of vibrations instead of the compressor, together with suitable sensors and electronic signal processing equipment. Until recently, there have been available no field-portable vibration generators that can be controlled to provide the required force amplitudes and frequencies; however, such a system with verified capabilities is now in hand.* Although such tests may be performed more easily if the compressor is not in operation, it is possible to carry out the desired measurements while the compressor is running - without interfering with operation of the system. In the latter case, electronic filtering and correlation techniques may be employed to distinguish between the vibrations induced by the compressor and those due to the external vibration generator; simple control systems can be used to ensure that the vibrations induced in the test never reach potentially dangerous levels.

Procedure

The test is performed by attaching the vibration generator to the piping system at some convenient, relatively unconstrained location. (Since the electrodynamic vibrator needs no structure or foundation against which to push, because it pushes against its own housing mass, suitable locations usually can readily be found.) The vibration generator then is programmed to sweep through the entire frequency range of interest - usually from 2 to 100 Hz - at a constant force amplitude, which is adjusted to be just high enough to yield easily resolvable signals from accelerometers mounted at several points on the piping system. From this initial test, resonance frequencies of the system can be obtained by noting where there occur peaks in the accelerometer signals.

Next, the dynamic amplification factor associated with each of the resonances is obtained by determining the "half power points". For a given resonance, the half power points are those frequencies above and below the resonance frequency at which the mean-square acceleration is one half of that obtained at the resonance frequency, provided that the force amplitude is held constant. The dynamic amplification factor, which indicates the importance of a given resonance, is equal to the ratio of the resonance frequency to the difference between the two half-power-point frequencies.

Then the vibration generator is set to dwell at one of the important resonance frequencies. By scanning along the piping system with one or more vibration sensors, the corresponding mode shape can be mapped, thus indicating the system's nodes and anti-nodes for that mode. By repeating this process for all resonance frequencies of interest and with the exciting force

*This recently developed system uses an electrodynamic vibrator capable of generating force amplitudes up to 30 lb at frequencies between 2 and 30 Hz, and somewhat lower forces at frequencies up to several hundred Hertz. The entire vibrator weighs less than 100 lb. In use, it need not be fastened to any massive reaction structure, since it uses its own casing as a reaction mass.

applied in all principal planes of the system, it is possible to amass a collection of valuable resonant response (modal) data. This data, in conjunction with information about the compressor operational characteristics, then permits well-founded engineering decisions concerning the design requirements for support modifications and/or additional supports.

Excessive motion of any existing pipe supports at any of the resonance frequencies of the pipe network may necessitate modification of the stiffnesses and/or masses of such supports, with or without the construction of additional supports. Design requirements for support modifications can be developed from a knowledge of the support impedances which can be calculated from measured vibration data. Support impedance is defined as the ratio of the amplitudes of the oscillatory force acting on the support and the velocity response of the support. At each pipe resonance, the velocity of the support can be measured directly, or can be calculated from the measured acceleration. The force on the support is the constraining reaction force between the support and the attached pipe. Generally, this force cannot be measured directly, but can be calculated from the resonant inertia load distribution along the pipe, which in turn can be calculated as the unit length weight of the pipe and the measured acceleration distribution of the pipe. For highly redundant piping systems, conservative approximations may be made regarding the inertia load reacted by each support; or alternatively, finite-element methods may be used to calculate reaction loads from pipe stiffnesses and deflection amplitudes. Unless the support is resonant within the frequency range of interest (which is highly unlikely), the impedances of a support at various pipe resonance frequencies should be proportional to the effective stiffness or mass of the support. Increasing this effective stiffness or mass will increase support impedance and constraint of the pipe. The above calculations tell the designer how much stiffness or mass must be added to reduce support motion to acceptable levels.

If it is deemed necessary to construct additional supports, then the locations of these supports should be selected as the anti-nodes of the measured mode shapes of the pipe network. Compromises in these locations must be made if the network displays more than one significant mode for which these is no common anti-nodal location and if constraints along different principal axes are required. Consideration must also be given to the resonance frequencies and mode shapes of any new pipe resonances introduced by the added support. Once a candidate location has been selected for a support, the vibratory force generator should be positioned at this location, with appropriate orientation. A frequency sweep through the frequency range of interest, and an evaluation of the mechanical input impedance at this location, will readily define the stiffness and mass characteristics that the added support must have in order to be effective.

The entire process described above usually can be accomplished in an eight to twelve hour period for a typical (approximately 150 ft long) piping system. The lead time required to prepare and assemble the required equipment rarely is longer than ten days - thus, it is possible to obtain accurate data for support positioning within two or three weeks after the problem is recognized.

COMBINING ANALYSIS AND TEST

Refinement of Mathematical Models

Computerized mathematical models are commonly used to evaluate various design configurations of gas compressor installations, and to optimize the final design so that vibration response levels are minimized within practical economic considerations for a broad range of operating conditions. Both analog and digital computer (finite element) models have been used by the industry. These models are designed to include the principal structural stiffness and inertial characteristics of the piping system, supports and related mechanical equipment as well as the principal sources of excitation.

Because of idealizations made in modeling of system components, theoretical predictions of dynamic characteristics usually differ somewhat from actual characteristics measured in the field. However, when the computer program is designed with sufficient flexibility, experimental data of the type discussed above can be used to refine the models so that more realistic predictions of system response characteristics can be made. The refined models can then be used to optimize the design and placement of additional pipe supports, to modify the designs of existing pipe supports, and to predict the effects of these modifications on system vibrations.

Development of a Data Bank

Generally, measured vibration data obtained for a given piping system has value beyond the immediate vibration problem under consideration. When properly evaluated in terms of system parameters such as pipe size, support impedance, damping, etc., these data can often be generalized to form empirical models of vibration characteristics of components and subsystems commonly used in other gas compressor installations. In fact, the use of proven subsystem designs for which vibration characteristics are known and can be described mathematically and empirically aids the designer in establishing a reliability envelop for the various modes of operation of a new installation in the planning and design phase. A continual state-of-the-art advance in the gas compressor industry could therefore be achieved by conducting vibration survey field tests on a variety of typical installations, and then incorporating these data in appropriate empirical formats into a comprehensive data bank that would be available to the industry for designing new installations. This systematic approach should result in both immediate and long-term benefits in developing effective methods for control of structural vibrations in gas compressor loading lines.

CONCLUDING REMARKS

The new test method discussed above should provide practical cost effective and timely solutions not only to vibration problems after these problems occur in operating systems, but to verify the adequacy of existing systems before they actually come on line. In either case, considerable savings in time and money may be realized. Furthermore, the combination of testing and analysis, along with the development of a data bank of valuable test results, should lead to a general state-of-the-art advance in the gas compressor industry.