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ABSTRACT

The valves in reciprocating compressors use springs to control the timing of the valve closing. These springs are subjected to dynamic loading by the motion of the valve element. The element motion is periodic, but includes rapid acceleration and deceleration. It can therefore excite a wide range of frequencies in the spring. The resulting spring surge creates high stresses and is a major contributor to premature spring failure.

Valve failures are the most common reason for unscheduled compressor shutdowns. The valve springs are the most common source of these valve failures. Therefore, valve spring life is of utmost importance in attaining increased compressor reliability.

This paper discusses two approaches to calculating the spring stresses caused by the surge. The first is an approximate method that considers only torsional deflection of the spring wire and neglects end coil effects. The second is a more complete analysis that uses both kinematics and FEA simultaneously to replicate the spring's response to the dynamic loading. The results of the two methods are compared and their usefulness in preventing spring failures is discussed.

INTRODUCTION

The valves considered here are used in process compressors. Each application is different and the optimum valve design must be selected for every order. The valve type, the valve lift and the springing are strongly dependent on the gas molecular weight, the compressor speed, the cylinder size and the operating pressures. Each compressor may experience different operating conditions from day to day depending on process requirements. In addition, the gas is often dirty, abrasive, corrosive and wet. Selection of the spring material and design to meet the corrosion and endurance limits is often the most difficult part of the process.

The spring stress is frequently much higher than that predicted by a simple static analysis. To get accurate values, the spring surge caused by the rapid motion of the sealing element must be considered. This over stress has been calculated for valves in which the flat sealing element also acts as the spring, but is not normally calculated for the coil springs considered here (Adams; Futakawa; Moaveni; Woollatt). In many practical cases, the spring dynamics are such that adjacent spring coils clash, which can cause mechanical damage. Methods to predict the severity of this coil-to-coil contact are also required. It is also probable that the ends of the spring will temporarily leave their stops. This is a possible contributor to wear and should be predicted.

The calculation requirements listed above can be met by a modern finite element method that includes kinematics. However, as the natural frequency of the spring dynamics is high compared to the time period of interest, the finite element analysis can take a considerable amount of time to complete. It must be repeated for the suction and discharge valves of each cylinder under each operating condition. In practice, based on time and resources, it is not economical to perform such an in depth analysis for every order. Therefore a simpler, less accurate, method that can be combined with the valve dynamics prediction that is done for each application is needed. This simpler or approximate method of calculating spring stresses adds only a negligible amount of time to each run.

THEORY

Approximate Method

The approximate method for calculating spring dynamics considers only torsion of the wire and ignores the effects of closed end coils. It does take coil-to-coil contact into consideration and can allow the end coils to leave their stops.

With the assumption that only torsional deflection of the wire is important, the basic equation (Wahl 25-7) is:

$$\frac{\partial_2 y}{\partial t^2} = a^2 \frac{\partial_2 y}{\partial s^2} - b\frac{\partial y}{\partial t}$$

where y = Deflection of Wire in Direction of Spring Axis
t = Time
a = Speed of Torsion Wave in Wire
s = Distance along Wire
b = Damping Factor (Twice Wahl's definition)

From Wahl (5-18)

$$a = \frac{d}{D}\sqrt{\frac{G}{2\gamma}}$$

where d = Wire Diameter
D = Mean Coil Diameter
G = Torsional Modulus of Wire
γ = Density of Wire Material

The acceleration due to gravity is omitted on the assumption that consistent units are used.

The basic equation can be written as:

$$\frac{\partial}{\partial t}\left(\frac{\partial y}{\partial t}\right) = a^2 \frac{\partial}{\partial s}\left(\frac{\partial y}{\partial s}\right) - b\frac{\partial y}{\partial t} \text{...............}(1)$$

Now:

$$\frac{\partial}{\partial s}\left(\frac{\partial y}{\partial t}\right) = \frac{\partial}{\partial t}\left(\frac{\partial y}{\partial s}\right) \text{..............................}(2)$$

(1) + a (2) gives:

$$\frac{\partial}{\partial t}\left(\frac{\partial y}{\partial t}\right) + a\frac{\partial}{\partial s}\left(\frac{\partial y}{\partial t}\right) = a\left[\frac{\partial}{\partial t}\left(\frac{\partial y}{\partial s}\right) + a\frac{\partial}{\partial s}\left(\frac{\partial y}{\partial s}\right)\right] - b\frac{\partial y}{\partial t}$$

That is:

$$\frac{du}{dt} = a\frac{d\alpha}{dt} - bu \quad \text{or} \quad \frac{d}{dt}(u - a\alpha) = -bu \qquad \text{along} \qquad \frac{ds}{dt} = a \text{ ………………..}(3)$$

where $u = \frac{\partial y}{\partial t}$ = Velocity of a point on the wire.

and $\alpha = \frac{\partial y}{\partial s}$ = Angular deflection of a point on the wire. This is directly related to the stress in the wire.

Similarly, subtracting a times equation 2 from equation 1 gives:

$$\frac{d}{dt}(u + a\alpha) = -bu \quad \text{along} \quad \frac{ds}{dt} = -a \ldots\ldots\ldots\ldots\ldots\ldots(4)$$

As the wave speed is constant, equations 3 and 4 can be solved numerically using a rectangular grid. If the mesh dimensions are such that the distance dimension (Δs) is the wave speed (a) times the time dimension (Δt), no interpolation will be required. The values at position s and time t are obtained using equations 3 and 4 from those at time t – Δt and positions s – Δs and s + Δs.

At the s = 0 boundary condition, u + aα is known from equation 4 and at the other end u – aα is known. The boundary conditions, eg a known velocity u or a free end (α = 0), allow α and u to be calculated.

The stress can be calculated from the angular deflection using Wahl 19-5 and 19-14. The maximum stress in the wire for a statically deflected spring is:

$$\tau = \frac{GdK}{\pi D^2 n}\delta$$

where K = Wahl's stress correction factor. Several expressions resulting from different assumptions are given by Wahl. The results given here use the first terms of his equation 19-35 as given below.
δ = The static spring deflection.
n = The number of coils.

As used here:

$$K = 1 + \frac{1.25d}{D} + \frac{.875d^2}{D^2} + \frac{Tan^2\vartheta}{2}$$

where θ = Coil Angle

For a dynamically loaded spring the stress τ must be calculated from the local deflection as given by δ.

$$\delta = SCos(\theta)\alpha$$

where S = Total wire length

Using the above, the stress and velocity and hence the deflection at every mesh point can be calculated. If two or more coils contact each other, the position and velocity of all the coils are set to the average position. Thus the effects of coil-to-coil contact are approximated.

Both boundary conditions in this application are velocity inputs. The spring dynamics calculation is an integral part of the valve dynamics calculation. The spring calculation gives the force acting on the element at any instant, and the valve dynamics calculation gives the resulting velocity. At the other end of the spring, the input velocity is zero. At either end, the spring may leave its stop. When this happens, the boundary condition is that the stress is zero. The position of the end of the coil can then be calculated from its velocity. Once the spring returns to its stop, the boundary returns to a known velocity.

Mechanical Event Simulation (MES) Method

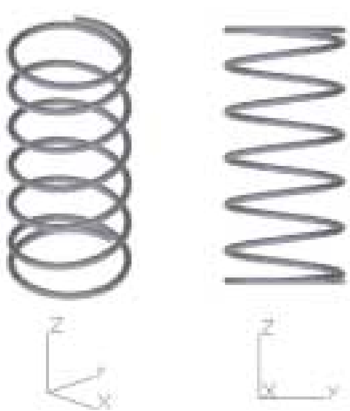
Mechanical event simulation software was used which combines the capabilities of kinematics and FEA to simultaneously replicate motion, flexing and stresses of the spring.

> Mechanical Event Simulation intrinsically calculates loads and stresses as motion takes place at each instant in time.
> The basic equation of event simulation is:
>
> $$[M]\{a\}+[C]\{v\}+[K]\{d\}=0$$
>
> where:
> [M] = the mass matrix
> {a} = the acceleration vector
> [C] = a constant matrix
> {v} = the velocity vector
> [K] = the stiffness matrix, and
> {d} = the displacement vector
> Note how the equation models the combination of motion, damping, and mechanical deformation.
> If the stresses are of interest, they can be calculated at any time during the analysis by application of the formula {s}=E{e}, where {e} (the strain vector) is easily obtained from the displacement vector {d}, and E is Young's modulus. (ALGOR)

DESCRIPTION AND RESULTS

The FEA model of the spring was an isotropic material model, which consisted of 5140 nodes and 3840 nonlinear 3-D brick elements. The free length, wire diameter and coil outside diameter of the FEA model of the spring was 0.639", 0.018" and 0.300" respectfully. The model consisted of 5.1 active coils and 2 inactive coils. The inactive coil at each end of the spring was squared (see Figure 1) to more closely simulate an actual spring, and provides for a better transfer of load.

The top of the spring was constrained in the Tx and Ty degrees of freedom. The base of the spring was constrained in the Tx, Ty and Tz degrees of freedom. A prescribed displacement was applied to the top of the spring. This prescribed displacement, shown in Figure 2, represents a typical compressor valve opening lift diagram for t > 0.008 sec. The prescribed displacement was also used to compress the spring from its free length to its initial installed length (t=0 to t=0.008 sec). To minimize the dynamic effects at the start of the comparison, a



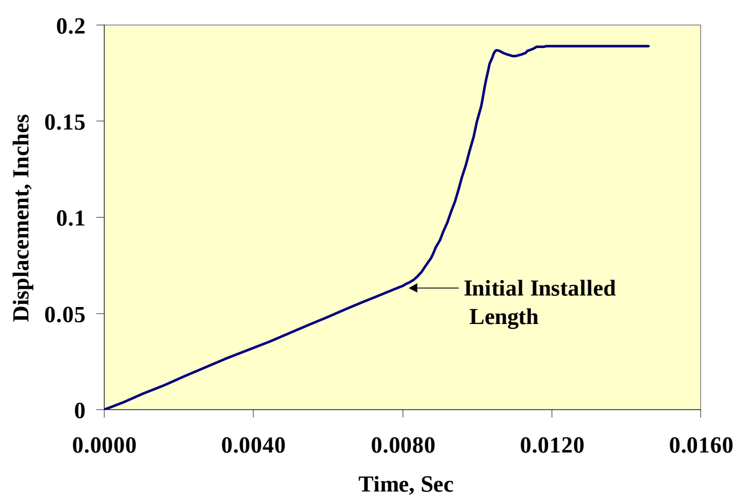


Figure 1 FEA Model of Spring Showing Squared Ends and Global Coordinate System.

Figure 2 Prescribed Displacement Applied to Top of Spring (Displacement vs. Time).

much lower velocity was used to bring the top of the spring to its' initial installed length. The material specified for the analysis was stainless steel (AISI 302). The time step size was 0.0001 sec. The convergence criteria was displacement with a tolerance of 1e-15 inches. To compare stress values between the two methods the Tresca stress was plotted for the MES method.

In comparing the two methods, six data points were identified where data would be collected. As shown in Figure 3, point 1 was located at the start of the first active coil and point 6 was located at the end of the fifth active coil. Intermediate points were spaced one coil apart.

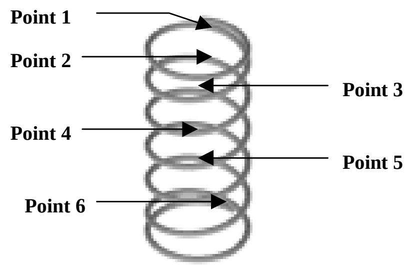


Figure 3 Model of Spring Showing Data Comparison Points.

In the following graphs of stress versus time (Figures 4a – 9b) points 1-6 refer to the locations where data was taken and compared between the two methods. All points from the MES method were taken on the inside radius of the FEA model where stresses were the highest.

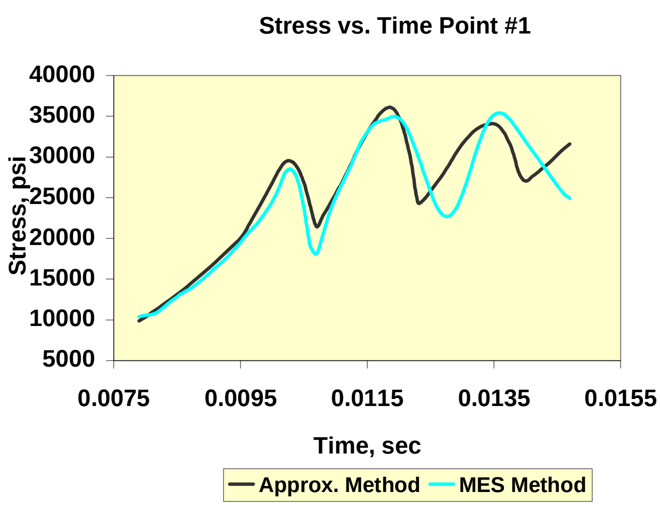


Figure 4a Original Results.

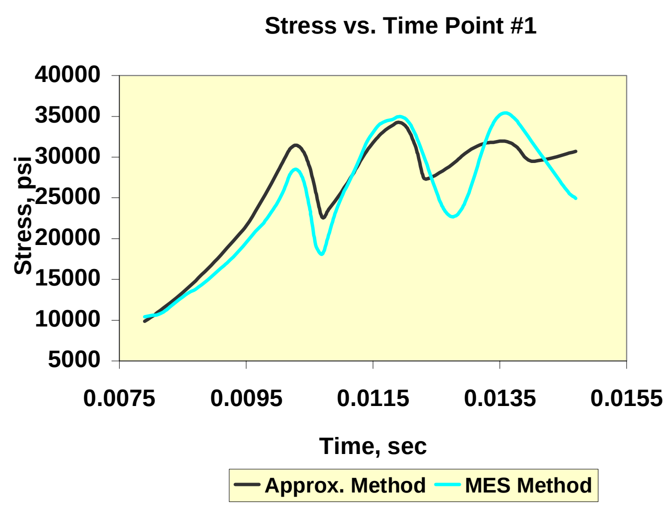


Figure 4b Approx. Method with 2X Damping.

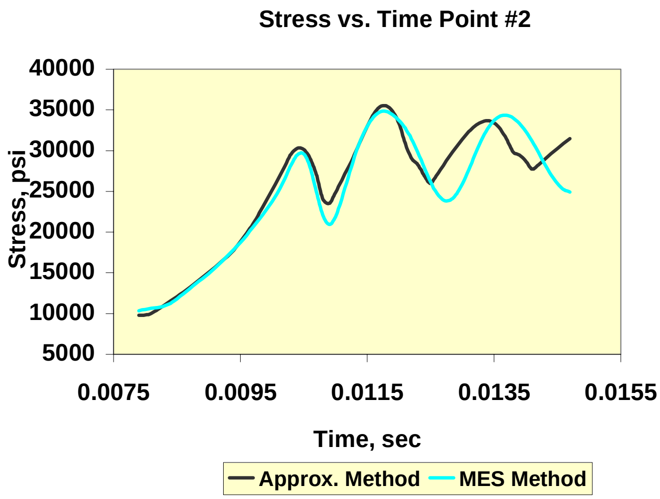


Figure 5a Original Results.

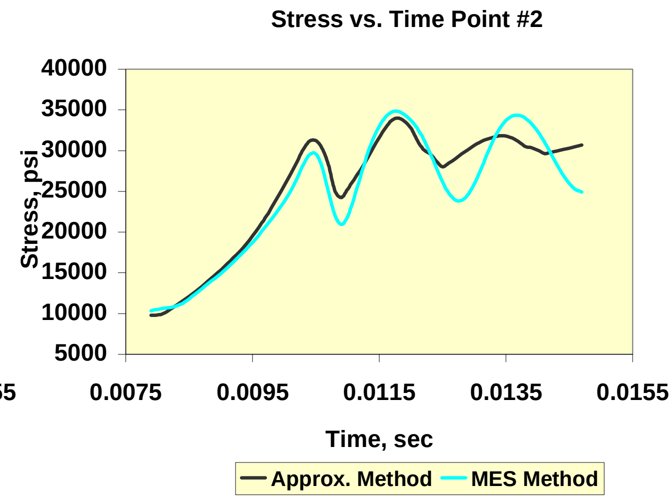


Figure 5b Approx. Method with 2X Damping.

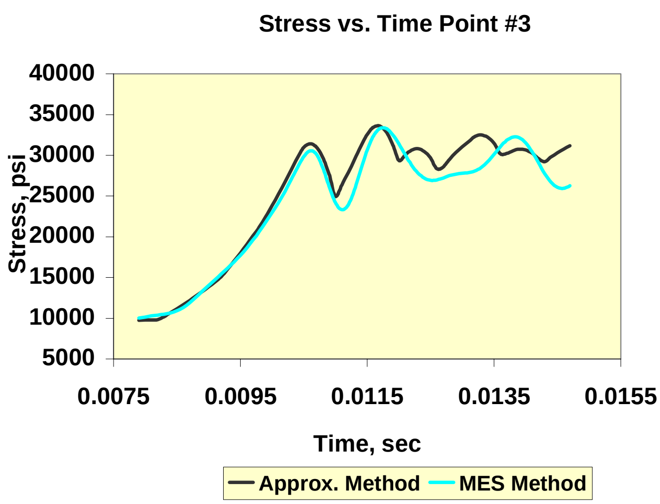


Figure 6a Original Results.

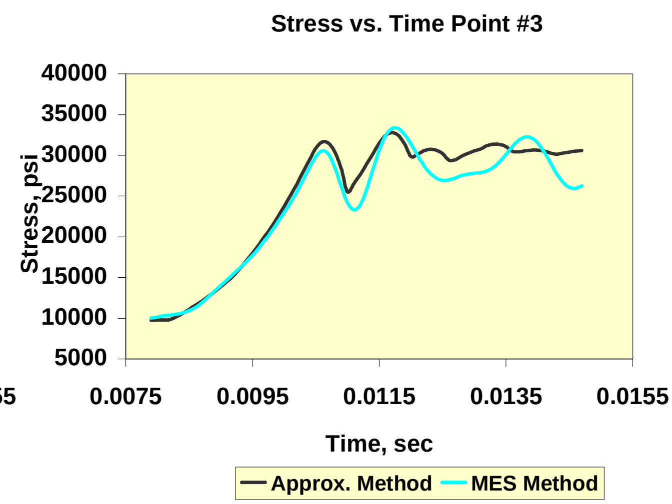


Figure 6b Approx. Method with 2X Damping.

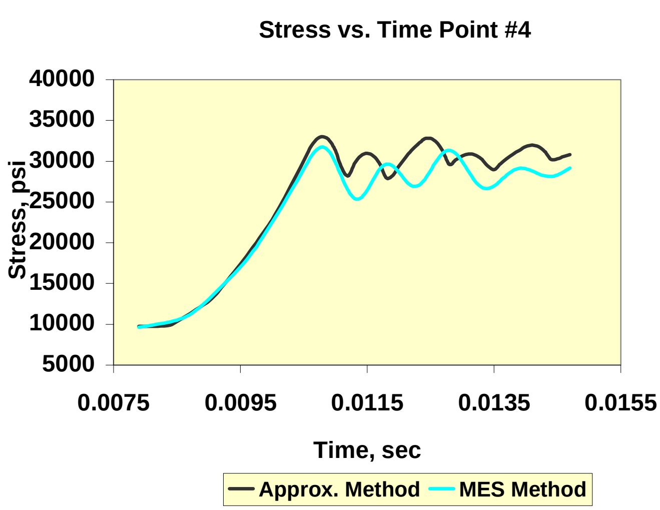


Figure 7a Original Results.

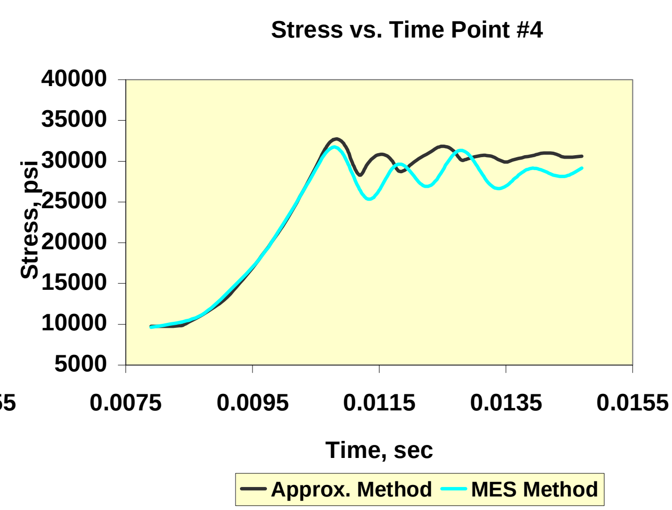


Figure 7b Approx. Method with 2X Damping.

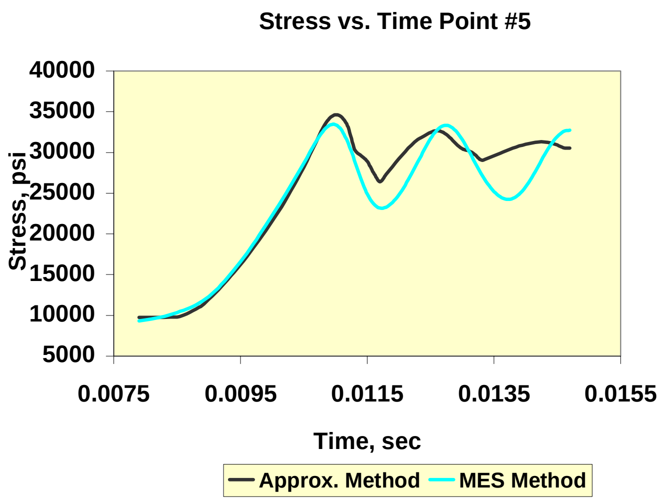


Figure 8a Original Results.

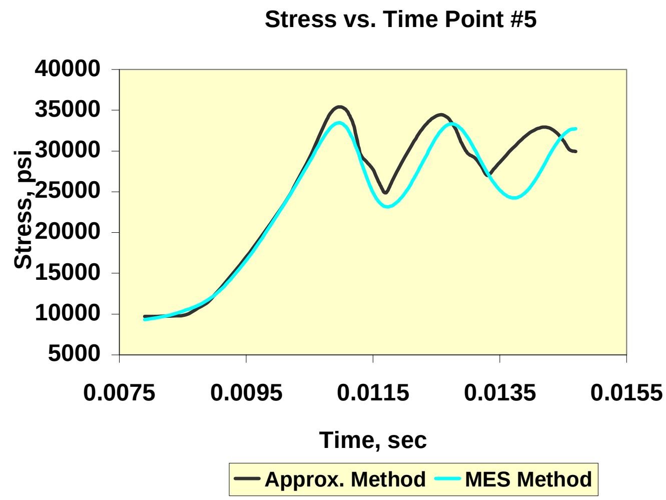


Figure 8b Approx. Method with 2X Damping.

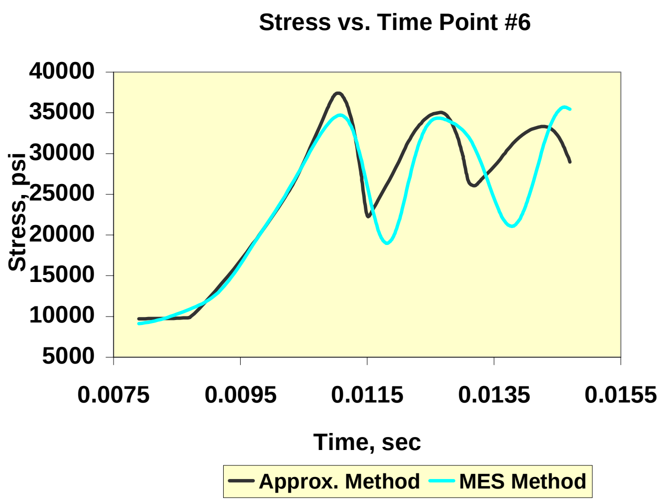


Figure 9a Original Results.

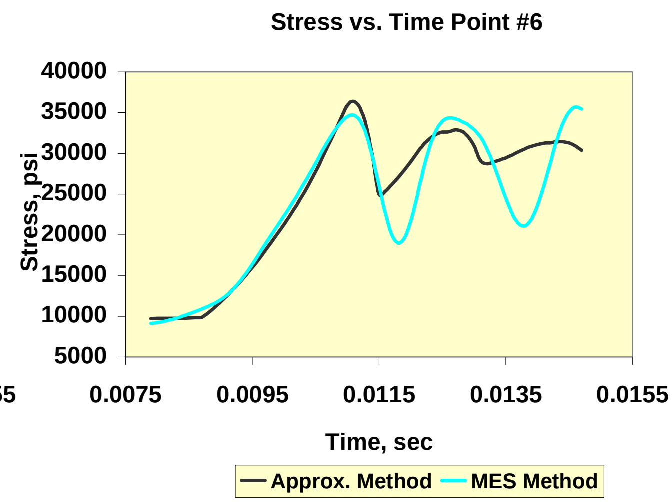


Figure 9b Approx. Method with 2X Damping.

DISCUSSION

As the stresses in a spring of this size in operation in a compressor are difficult to measure, we assume that the FEA method is accurate and use it to check the accuracy of the approximate method. We have no way of knowing what really happens when adjacent coils impact, so the case used here is for relatively mild motion that does not cause coil-to-coil contact. The strongest surge occurs following impact of the moving element on the stop at the conclusion of the opening process. The curves show the results for the time period around maximum stress. This represents a small part of the total valve event. The agreement between the two methods as shown in Figs 4a, 5a, 6a, 7a, 8a and 9a is good, giving credibility to the approximate method.

The effective damping for a spring installed in a valve is unknown. The effect of doubling the damping in the approximate calculation is shown on Figs 4b, 5b, 6b, 7b, 8b and 9b. The damping value initially chosen agrees best with the results of the FEA calculation. The damping in the MES method was based on the time integration method used. The parameter entered for this method allowed for the most damping of high frequencies without loss of high temporal integration accuracy. Additional mass and stiffness related damping such as Rayleigh damping was not included. The initial damping in the approximate method was 0.01 /sec. The agreement of the two methods when this value is used should not be taken to imply that it is closer to reality. Additional testing will be required to determine the damping of a valve spring. Results for a calculation of the valve spring stresses in a compressor are shown in Figure 10. The stresses calculated by the quasi-static method and by the approximate dynamic method are shown. Note the high natural frequency of the spring and the large excess in stress caused by the spring surge. Figure 10 for a typical valve spring shows the maximum calculated dynamic stress anywhere in a spring compared to the stress that would occur if the spring did not surge. It will be seen that the actual stress is about 50% higher than the static stress and, more important for failures due to fatigue, the actual stress variation is about twice that for the ideal case with no surge. It is probable that the damping in a valve spring is less than the value used in this calculation. If this is true, the vibrations caused by the valve opening will still have high amplitude when the valve closes. The interaction of the opening and closing vibrations will have results that are very difficult to predict. As the highest stresses occur at valve opening, prediction of the opening surge may be adequate to ensure reliability even though damping values are not known.

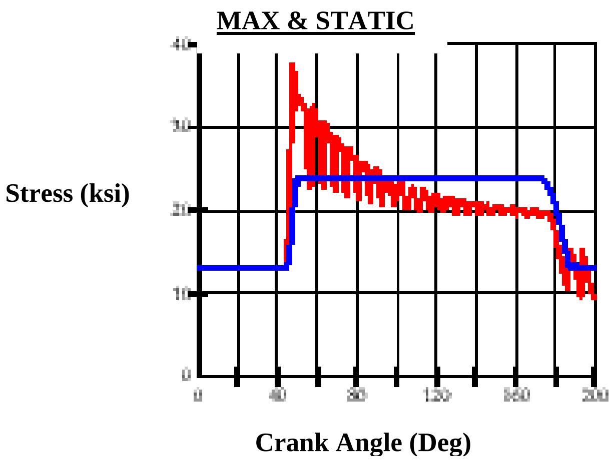


Figure 10 Difference Shown Between Quasi-Static and Approximate Dynamic Method

CONCLUSIONS

A simple method of predicting the actual stresses in a valve spring that can be used interactively with a valve dynamics calculation has been developed. It has been shown to give good agreement with a more detailed finite element method analysis. The use of this method in practical examples shows that the actual stresses in a valve spring are much greater than those predicted by quasi-static methods.

Additional MES analysis will show the benefits of damping materials used in conjunction with springs. Results from this could then be incorporated into the approximate method. These analyses will lead to lower stress spring designs, which translates into increased fatigue life. The end result is increased compressor reliability.

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