SIMPLIFIED ANALYSIS OF STEAMHAMMER PIPE SUPPORT LOADS

E. C. Goodling, Jr.
Gilbert/Commonwealth, Inc.
Reading, Pennsylvania

ABSTRACT

Rigorous analysis of steamhammer loads in power plant piping generally requires the use of computer mainframe programs to perform the dynamic transient analysis and time history analysis in order to quantify steamhammer forces and piping system response resulting from sudden stopvalve closure following a unit trip in a steam power plant. This paper offers a simplified procedure for approximate manual calculation of the momentary internal unbalanced forces due to steamhammer and the dynamic system response needed to properly design and analyze the supports and restraints on the affected pipe. The results are approximate but are of a degree of accuracy consistent with the input data required in the performance of rigorous computer-based analysis, and should therefore be adequate for the design of fossil plant piping and the non-safety piping in nuclear plants.

INTRODUCTION

The sudden closure of the stop valves on the main steam and reheat steam inlets to the turbines in a steam power plant can result in large momentary unbalanced forces in the steam piping as the system seeks to restore equilibrium. When the main steam stop valve closes, steam velocity at the valve entrance is quickly stopped in whatever time interval it takes for the valve to close. At the instant of closure, there is still a flow of steam into the boiler end of the pipe at the superheater connection. A wave of increasing pressure immediately begins forming in the steam at the turbine end and travels back upstream toward the boiler at sonic velocity [1][2]. At the same time, a similar event is taking place in the cold reheat piping except that the steam flow into the pipe at the high pressure turbine discharge is suddenly halted while steam flow continues into the reheater at the boiler end, resulting in a wave of decreasing pressure forming at the turbine and traveling at sonic speed toward the reheater. Meanwhile similar events are taking place in the hot reheat piping, with a wave of increasing pressure forming behind the closed reheater stop valve, traveling back upstream to the reheater outlet.

These traveling waves of increasing or decreasing pressure create momentary unbalanced forces in the piping along each longitudinal run of pipe, their magnitudes being primarily a function of initial rate of steam flow, run length and stop valve closure time [3]. As the pressure wave makes its pass through a pipe run and makes a (typically) right angle turn at the end of the run, the pipe and its restraints conserve momentum in either or a combination of two ways [8][10]:

1. By the pipe becoming set into motion in response to the impulse of the pressure wave and/or

2. Direct reaction of the wave-induced force in the pipe supports, restraints, and anchors.

The magnitude of the force in any given run of pipe reaches a momentary peak equal to the pressure differential existing over the length of the run multiplied times the cross-section area of the pipe. The response of the piping system is a function of its mass and stiffness; the stiffer and more massive a pipe is, the more sluggishly it responds to an impulse load such as steamhammer. A very heavy pipe such as the main steam pipe in a large power plant may respond very little to steamhammer, while the much thinner wall cold reheat piping, with much less mass, may jump a foot or more under an equivalent steamhammer load.

RIGOROUS ANALYSIS

The analysis of the steamhammer forces in power plant piping is typically carried out with the use of a computer program such as WAVENET [4] or RELAP [5], which calculate -

1. The shape and amplitude of the pressure wave generated by a rapidly closing valve in the steam line and

2. The unbalanced reactions due to the wave at different sections of the piping system where the changes in direction occur.

The total wave may be thought of as being made up of a series of superposed wavelets, each of which is generated by the valve closure process over an interval of time, typically .02 seconds. The following typical assumptions are necessary in developing the mathematical model for generating the pressure wave phenomena [6] (the assumptions vary for different programs, some may not be needed):

1. Homogeneous single phase one dimensional flow.
2. The change of steam properties caused by the pressure wave takes place isentropically.
3. The steam is treated as a perfect gas.
4. The temperature is constant.
5. Flow is considered frictionless.
6. Losses at valves, bends, reducers, etc. are neglected.

In addition to these, certain other assumptions must be made with regard to the boiler, i.e. zero feedwater flow into the economizer at the moment of stop valve closure, heat added to the boiler during wave propagation, behavior of the boiler as an infinite reservoir. Then the stop valve closing curve (flow area vs. time) must be modeled.

With all of these factors incorporated into the model, the computer program calculates a theoretical maximum amplitude of the pressure wave, and then determines the unbalanced reactions in every run of the piping system. If there is more than one pressure wave in a particular run of pipe, the program typically applies superposition to determine the combined effects of the multiple waves.

The output of the execution of a transient analysis is a set of forces acting on each node point of the piping model, with a different set of forces for each time step in the selected time interval. A typical interval of 2 seconds with a .02 second time step would result in 101 sets of forces to be read into the time-history analysis program model, which is then used to calculate the piping system response and the time-dependent forces on the pipe anchors and restraints.

Despite the abundance of computer codes available for transient analysis and time-history analysis of steamhammer effects on piping systems, the use of such codes requires that the analyst have sufficient knowledge and experience in assessing the reasonableness of the analysis results. A disturbing fact frequently encountered in analysis is that steamhammer loads shown in the outputs of different programs for the same problem may differ by a factor of 10 or more. This situation may be caused by inadequate theoretical development, incorrect interpretation of vague instructions in a poorly-written user's manual, incorrect selection of time step, or simply an input error or incorrect initial conditions. An inexperienced analyst may end up with fictitiously high loads without being able to recognize them as such. The result may be forces on supports and restraints beyond the capability of any reasonable hardware design [3].

SIMPLIFIED ANALYSIS

Steamhammer forces, piping system response, and restraint reactions may be estimated by the use of a relatively simple procedure to calculate the maximum momentary unbalanced forces in each run of the piping system. These forces can then be used to determine the approximate system response to the forces and to estimate the forces to be used in designing the flexible or rigid restraints needed to control the piping movements and keep pipe stresses within an acceptable range.

Calculation of Unbalanced Forces

The unbalanced forces in each pipe run may be calculated as follows:

1. Start with an isometric drawing of the system to be analyzed. From this, prepare a simplified isometric sketch showing the system idealized as follows:
   a. Treat the elbows and bends as sharp turns of negligible bend radius.
   b. Straighten out slight angular changes in direction; for example, if a pipe runs 30 feet in one direction, then turns 15 degrees before continuing for another 20 feet, treat the pipe segment as a straight 50-foot run.
   c. Show the lengths, directions, and diameters of each run in the idealized system, and label the elbows and bends in some convenient manner.

2. Calculate sonic velocity in the steam flow:

$$C = (144kgpv)^{1/2} \quad (1)$$

where C = sonic velocity, ft/sec
$k = c_p/c_v$
where c_p = specific heat at constant pressure
c_v = specific heat at constant volume
(Let $k = 1.27$ for superheated steam)
g = 32.2 ft/sec^2
p = pressure, psia
v = specific volume, cubic feet/pound

3. Determine the following:
   a. Steam flow rate W in pounds/second at full load
   b. Cross-section flow area A in square inches for each pipe diameter

c. Flow velocity V in ft/sec for each pipe diameter

$$V = 144vW/A \quad (2)$$

d. Effective valve closing time t_c in sec. This can be accomplished by approximating the valve closing curve with a straight line extending the steepest slope of the valve closing curve, and letting t_c equal the time interval between the intercept points of the straight line with the fully open line and the fully closed line.

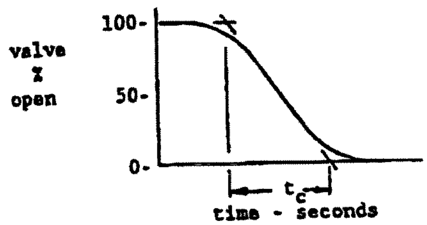


Fig. 1 Stop valve closing curve

If the valve closing curve is not available, t_c may be conservatively estimated at 0.04 second.

4. Calculate the maximum unbalanced axial force that can exist in any run of pipe [3]:

$$F_{max} = 1.05CW/g \quad (3)$$

where F_{max} = unbalanced force, pounds

Or, expressed another way:

$$F_{max} = A\Delta p \quad (4)$$

where Δp = pressure rise in the run, psia

= $1.05\,\rho VC/144$ where ρ = mass density of steam, pound sec^2/ft^4

= $1/vg$

This is the maximum force that can exist in a piping system which contains one or more runs long enough to intercept a full pressure wave, that is, to experience the full magnitude of the pressure rise before the advancing pressure wave encounters a change in direction.

5. Calculate the momentary unbalanced force in each pipe leg [3]:

$$F = 1.05WL/gt_c \quad (5)$$

where F = unbalanced force, pounds
L = length of straight pipe leg being analyzed, feet

Note: If $F > F_{max}$ from Step 4, then let $F = F_{max}$. Generally, $F \geq F_{max}$ means simply that the run is long enough to experience the full magnitude of the pressure rise.

6. Plot the impulse function for each pipe run:

a. Calculate the critical length L'.

L' is the length in feet of a straight run of pipe at which the unbalanced force F_{max} occurs. Letting $F = F_{max}$ by combining Equations (4) and (5) and setting L' equal to L:

$$L' = Ct_c \quad (6)$$

b. Calculate the time t required for the unbalanced force to reach a maximum.

For $L > L'$, $t = t_c$

For $L < L'$, $t = t_c(L/L')$

c. Calculate the dwell time t_d, that is, the length of time a pipe run greater than L' in length experiences the full magnitude of F_{max}.

For $L > L'$, the pressure increases to its maximum of $P + \Delta P$ in t_c second, resulting in an unbalanced force of F_{max} which remains constant for a short time interval t_d until the advancing pressure wave reaches the bend and changes direction. With the passing of the pressure wave, the pressure differential along the pipe run drops, resulting in the unbalanced force dropping from F_{max} to zero. Thus, $t_d = t - t_c$.

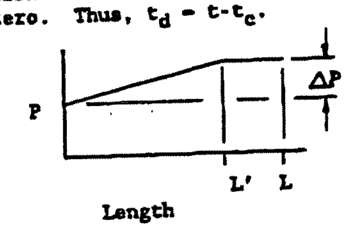


Fig. 2 Pressure differential causing maximum unbalanced force in a run where L > L'

For L = L', the pressure rises to its maximum of $P + \Delta P$ in t second, but the pressure differential along the run immediately begins its decline as the advancing front of the pressure wave reaches the end of the pipe run. Thus, $t_d = 0$.

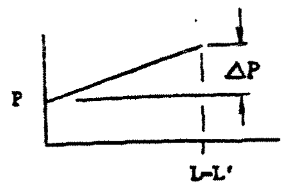


Fig. 3 Pressure differential causing maximum unbalanced force in a run where L = L'

For L < L', by the time the pressure has reached its maximum level of P + ΔP, the advancing front of the pressure wave has already reached the end of the run resulting in a decline in unbalanced force from its peak even before the pressure has reached its maximum in the run. Thus, $t_d = 0$.

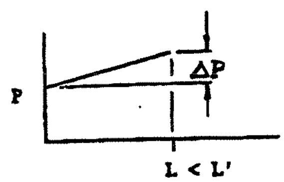


Fig. 4 Pressure differential for a run where L < L'

Determination of System Response

With the valve closing rate, the maximum unbalanced force, and the force dwell time all known for a specific pipe run, the forcing function is shown as a triangular pulse when the unbalanced force is plotted against time as in Fig. 5.

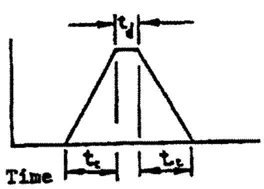


Fig. 5 Forcing function for a steamhammer wave

At this point it is necessary to determine the natural period T and the stiffness k of the system along the axis of the pipe run. This can be accomplished by the use of a PC-based pipe stress program or by manual approximations.

The next step is to determine the maximum displacement y of the pipe run when it is subjected to the triangular forcing function of a steamhammer. J.M. Biggs [7] presents a series of equations for calculating system response, but he makes the job easy for us by plotting dynamic load factor (DLF) against the ratio of loading time to natural period (t/T), so that we do not even have to solve the response equations (see Fig. 6 and 7).

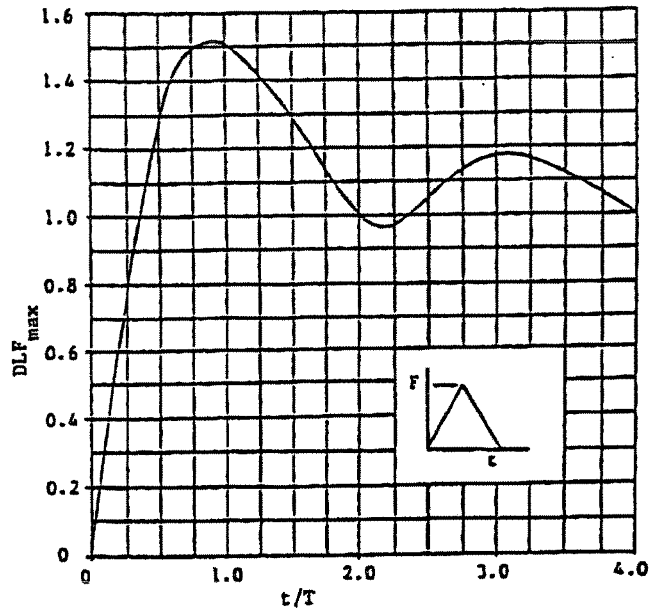


Fig. 6 DLF_{max} vs. t/T for triangular pulse [7]

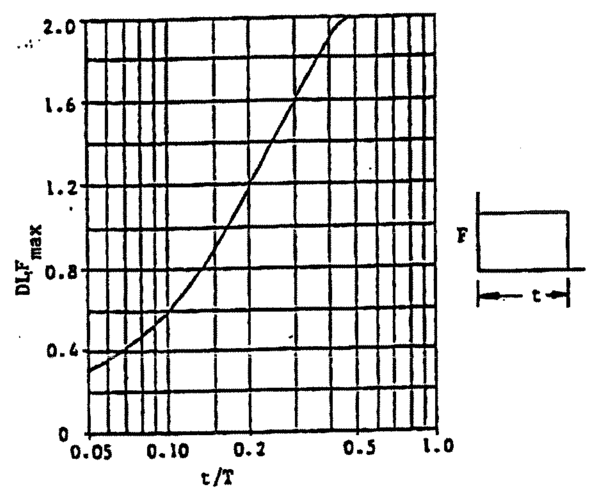


Fig. 7 DLF_{max} vs. t/T for rectangular pulse [7]

The Biggs plot in Fig. 6 is for a triangular impulse in a pipe where L≤L'. The actual function for a steamhammer acting on a pipe run with L slightly longer than L' is a triangle with a slight truncation at the top, representing the dwell time interval over which the peak unbalanced force acts. Such a function may be idealized, though, by simply extending the sides of the triangle to a pseudo peak that will represent a maximum force slightly higher than that which was actually calculated for the pipe run(See Fig. 10).

If the dwell time t_d is significant with respect to the valve closing time t_c, it may be more appropriate to approximate the forcing function as a rectangular impulse as shown in Fig. 7 From this, it is apparent that, for long runs (L>L'), t (and consequently t/T) becomes large and DLF reaches a maximum value of 2.0 as t/T exceeds 0.5. In those systems where a pipe run L is very long (L>>L') and the run is near the stop valve, the transient unbalanced force may behave more as a ramp function [7] as shown in Fig. 8. In such a run, the reflected wave from the boiler will hit the run in the opposite direction, causing a second dynamic transient after the primary wave has completely passed through the run. Biggs [7] notes that a peculiarity of this type of load pulse is the fact that , if the rise time is a whole multiple of the natural period, the response is the same as if F had been applied statically. In any case, the analyst should consider the possibility of a reflected wave acting in the opposite direction in any run in the system.

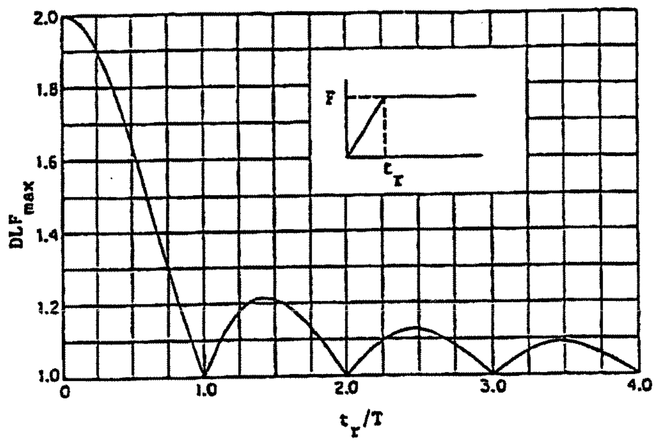


Fig. 8 DLF_{max} vs. t_r/T for ramp function [7]

The only equation to be solved then to determine response is:

$$y_{max} = (F/k)(DLF) \qquad (6)$$

where F = the maximum value from the actual or idealized plot of the impulse function
k = pipe system stiffness along the run axis
DLF = value read from Fig. 6 or 7

This value for y_{max} is therefore an approximation of how far a specific run flexible piping system will jump as a result of experiencing the momentary wave of unbalanced internal pressure resulting from the sudden closure of the stop valve to halt the flow of steam.

The movement y_{max} can then be used to calculate the upset stresses and piping reactions at connected equipment for comparison with piping code stress criteria and recommended levels for boiler and turbine connections. If it turns out that stresses and/or reactions are too high, the system response y_{max} can be reduced to an acceptable level by increasing the value of k. This can often be accomplished by adding a relatively stiff spring restraint [8] to the pipe run to resist the unbalanced force without completely reacting it. A new value for y_{max} can then be calculated, evaluated for acceptability, and then used to calculate the force experienced by the variable spring restraint.

Of course, if the pipe run is fully restrained along its axis, k is high and the displacement y is negligible. In this case there is no system response, and the full magnitude of the force F is reacted directly by the restraint.

EXAMPLE PROBLEM

A main steam pipe in a relatively small powerplant is subject to steamhammer following stop valve closure due to unit trip. It is desired to calculate -

1. The steamhammer load on an existing rigid hanger.
2. The maximum axial deflection (response) of the longest unrestrained horizontal run during steamhammer loads.
3. The effect on response of adding a flexible restraint to the long horizontal run to control axial movement.
4. The maximum load in the added axial restraint.

The piping is low CrMo steel, 10" NPS Schedule 160. The full load operating conditions are as follows:

Pressure = 1400 psia
Temperature = 955 degrees F
Steam flow = 450,000 pounds per hour

Calculations

1. Draw an isometric sketch of the portion of the piping system containing the runs to be analyzed. (See Fig. 9)

2. Calculate sonic velocity in the steam flow.

$$C = (144kgpv)^{1/2} \qquad (1)$$

$k = 1.27$
$g = 32.2$ ft/sec^2
$p = 1400$ psia
$v = 0.5566$ ft^3/lb

$$C = [(144)(1.27)(32.2)(1400)(0.5566)]^{1/2} = 2142 \text{ ft/sec}$$

3. Determine the following:

a. Steam flow rate, lb/sec

$W = (450,000$ lb/hr$)/3600$ sec/hr $= 125$ lb/sec

b. Flow area of pipe

$A = 56.74$ in^2 for 10" Sch 160 pipe

c. Flow velocity in pipe

$$V = 144vW/A = (144)(0.5566)(125)/56.74 = 176.6 \text{ ft/sec} \qquad (2)$$

d. Stop valve closing time

$t_c = 0.040$ sec (idealized) See Fig 1.

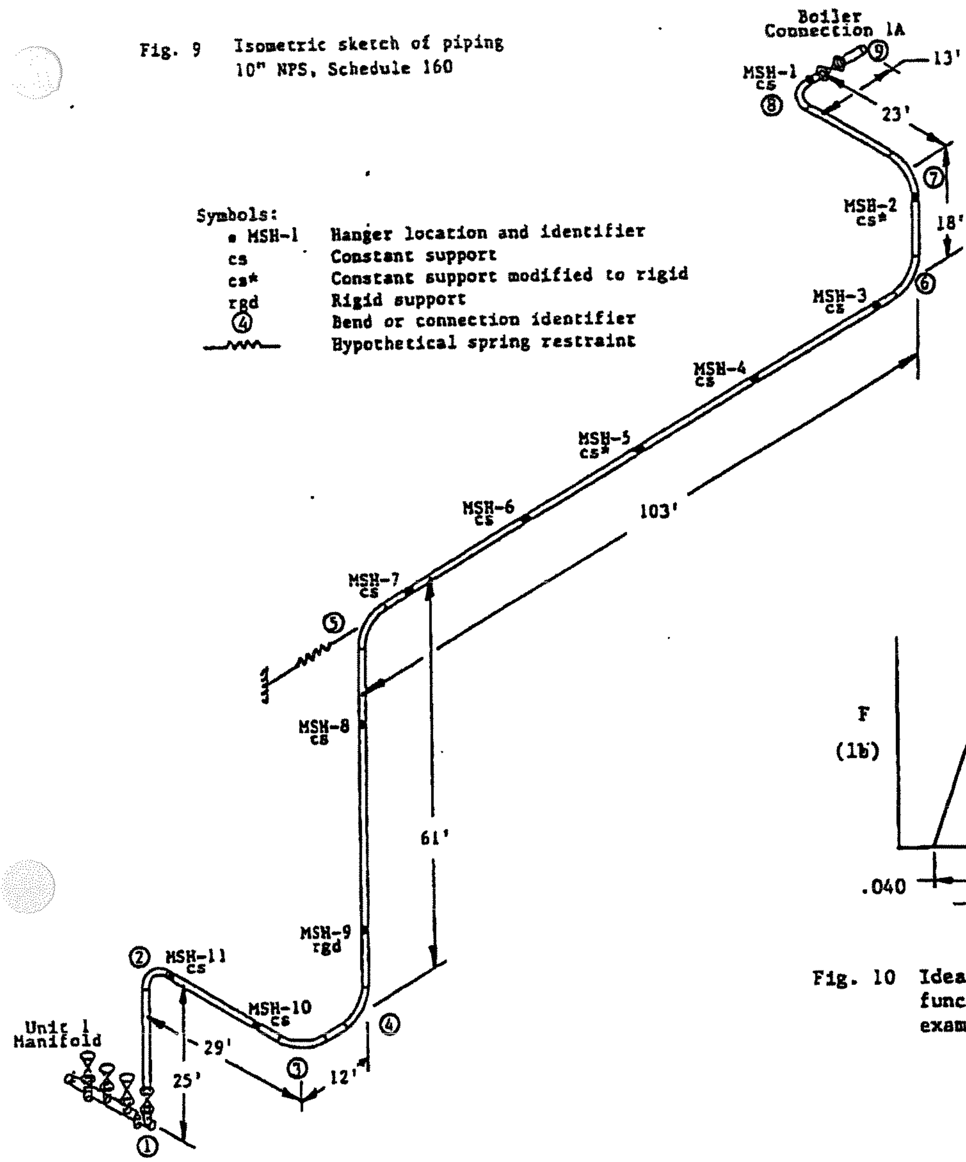


Fig. 9 Isometric sketch of piping
10" NPS, Schedule 160

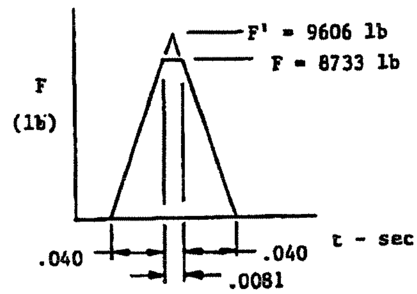


Fig. 10 Idealized triangular impulse function for Run 5-6 in the example problem

4. Maximum possible unbalanced force.

$$F_{max} = 1.05\ CW/g \qquad (3)$$
$$= (1.05)(2142)(125)/32.2$$
$$= 8731\ lb$$

or, alternatively -

$$F_{max} = A\Delta p \quad (4)$$

$$\Delta p = 1.05 \rho VC/144$$

$$\rho = 1/vg$$
$$= 1/(0.5566)(32.2)$$
$$= 0.0558\ lb\ sec^2/ft^4$$

$$\Delta p = (1.05)(0.0558)(176.6)(2142)/144$$
$$= 154\ psi$$

$$F_{max} = (54.76)(154)$$
$$= 8733\ lb$$

5. Momentary unbalanced force in the pipe runs.

$$F = 1.05WL/gt_c \qquad (5)$$

a. For the vertical run 4-5

$L = 61$ ft

$$F = (1.05)(125)(61)/(32.2)(0.04)$$
$$= 6216\ lb$$

Since F_{max} = 8733 lb, it is apparent that the run is not long enough to intercept a full pressure wave. Therefore the unbalanced force in this run is less than F_{max}.

b. For the horizontal run 5-6

$L = 103'$

$$F = (1.05)(125)(103)/(32.2)(0.04)$$
$$= 10{,}496\ lb$$

But since $F_{max} = 8733$ lb, it is apparent that the run is long enough to intercept a full pressure wave. Therefore, the full unbalanced force is limited to F_{max}; then -

$F = 8733$ lb

6. The time-dependent impulse functions.

a. Critical length L' required for a full pressure wave to develop in a pipe run:

$$L' = Ct_c \qquad (6)$$
$$= (2142)(0.04)$$
$$= 85.7 \text{ ft}$$

b. Time t_r required for the unbalanced force to reach a maximum:

For Pipe Run 4-5, L<L' (61'<85.7') and -

$$t_r = t_c(L/L')$$
$$= 0.04(61/85.7)$$
$$= 0.0285 \text{ sec}$$

For Pipe Run 5-6, L>L' and so -

$$t_r = t_c = 0.040 \text{ sec}$$

c. Dwell time t_d for F_{max} in a run where L>L':

For Pipe Run 4-5

$t_d = 0$ since L<L'.

For Pipe Run 5-6

$$t_d = t_c[(L/L')-1]$$
$$= 0.04[(103/85.7)-1]$$
$$= 0.0081 \text{ sec}$$

d. Total time duration t of the load impulse.

$$t = 2t_r + t_d$$

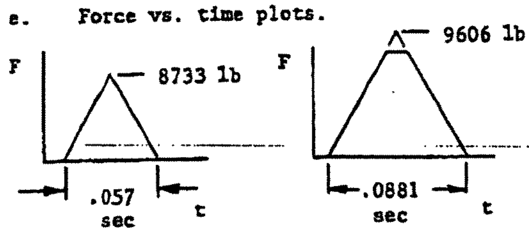
For Pipe Run 4-5 -

$$t = 2(0.0285) + 0$$
$$= 0.057 \text{ sec}$$

For Pipe Run 5-6 -

$$t = 2(0.040) + 0.0081$$
$$= 0.0881$$

e. Force vs. time plots.



Pipe Run 4-5 Pipe Run 5-6

Note that the plot for Run 5-6 has been idealized slightly to turn it into a triangular impulse function as described by Biggs [7]. This results in a slight conservative increase in the maximum force from F of 8733 lb to F' of 9606 lb, which is used in the remainder of the calculations for Run 5-6. If F' were substantially higher than F, say 30% or more, as it would for higher values for L and td, it may be more appropriate to represent the steamhammer load function for Run 5-6 as a rectangular impulse as in Fig. 7 [7][10] or as a ramp function (see [7] Fig. 2.9).

7. Stiffness k and the natural period T for the piping system.

Pipe Run 4-5 is supported rigidly along its axis by Hanger MSH-8. It therefore has a very high stiffness k and very small natural period. For a k of 10^6 lb/in, its natural period would be about 0.002 sec.

Pipe Run 5-6 is unrestrained along its axis; it therefore has been found to have a stiffness k of about 350 lb/in along its axis and a natural period T of 3 sec.

8. Dynamic Load Factor DLF.

Noting that the force vs. time plots for both pipe runs are essentially triangular functions, DLF for both can be determined from Fig. 6.

For Pipe Run 4-5, t/T is >>4.0; therefore

$$DLF \simeq 1.0$$

For PIPE Run 5-6, t/T is about 0.03; therefore

$$DLF \simeq 0.1$$

9. Maximum deflection due to the dynamic transient force F.

For Pipe Run 4-5, the high stiffness results in essentially zero deflection.

For Pipe Run 5-6, maximum deflection z_{max} is calculated by -

$$z_{max} = (F'/k)(DLF) \qquad (7)$$
$$= (9606/350)(0.1)$$
$$= 2.74 \text{ in}$$

10. Maximum transient load in rigid hanger MSH-8.

Since there is essentially zero deflection in the vertical run 4-5, and since the natural period is high resulting in DLF of 1.0, the maximum transient load in Hanger MSH-8 is F of 6216 lb. However, if the net support stiffness, considering the combined flexibility of the building structure and other components in the support assembly results in a value of ~ 0.9 for t/T, the

DLF could reach a peak of 1.55. Biggs [7] observes that for the triangular loading function, the maximum dynamic effect occurs when the duration of loading is approximately equal to the natural period of the system. It would therefore be advisable to consider this peak value for DLF in analyzing or designing a rigid support.

11. Effect of adding an axial restraint to Pipe Run 5-6.

If a rigid restraint were installed to react the axial steamhammer load in this run, it would be subject to the full magnitude of the load F' of 9606 lb, subject to the DLF effects cited by Biggs [7] for pipe runs with a natural period that approximates the duration of loading.

Consider now the effect of adding a stiff spring restraint along the axis of this run with a stiffness k_r of 10,000 lb/in. The following changes in the dynamic characteristics of the system occur:

a. The stiffness of the system with the new spring restraint becomes -

$$\begin{aligned} k' &= kr + k \\ &= 10{,}000 + 350 \\ &= 10{,}350 \text{ lb/in} \end{aligned}$$

b. The natural period then changes because of the added stiffness -

$$\begin{aligned} T' &= T(k/k')^{1/2} \\ &= 3(350/10{,}350)^{1/2} \\ &= 0.55 \text{ sec} \end{aligned}$$

c. The value for t/T' then becomes -

$$\begin{aligned} t/T' &= 0.0881/0.55 \\ &= 0.16 \end{aligned}$$

and DLF from Fig. 6 is read off as 0.6. Compare this with DLF of 0.1 for this run with no restraint.

d. The maximum deflection of the modified system under the transient steamhammer load becomes -

$$\begin{aligned} z_{max} &= (9606/10{,}350)(0.6) \\ &= 0.557 \text{ in} \end{aligned}$$

Compare this with 2.74 in for this run with no restraint.

e. The maximum load F_r in the restraint itself becomes -

$$\begin{aligned} F_r &= k_r z_{max} \\ &= (10{,}000)(0.557) \\ &= 5570 \text{ lb} \end{aligned}$$

This implies that it takes only 5570 lb to restrain the transient load of 9606 lb and bring the pipe to a halt in 0.557 in, showing that much of the energy in the transient was dissipated in setting the pipe in motion with the remainder absorbed in compressing the spring restraint.

Summary

The example problem demonstrates how the following can be calculated through the use of the simplified approach outlined in this paper:

1. Maximum design value for steamhammer force in any run of pipe.
2. Maximum theoretical response (deflection under transient load) in any run.
3. Maximum design load in rigid or flexible restraints.
4. Effects of modifying the system by adding rigid or flexible restraints.

The actual details of calculating piping system stiffness and natural period are not covered here, as they are best handled by desk-top computer-based general purpose ~~structural or~~ pipe stress programs or by approximate methods outlined by Biggs [7], Kellogg [9], or Snow [10].

CONCLUSIONS

Unbalanced forces and piping system response to steamhammer can be rigorously calculated with the aid of computer codes which perform fluid transient and time-history analyses. However, the simplifying assumptions made with regard to many of the input parameters and their effect on the accuracy of the results of the rigorous analysis raises a question of whether the trouble and expense of such analysis is justified. This paper suggests a simplified alternative method of performing a steamhammer analysis of a piping system, and gives results which are believed to be conservative but reasonable. The use of a PCbased pipe stress program is helpful in calculating piping system stiffness and natural period, but reasonable approximations may be obtained by manual calculations alone.

ACKNOWLEDGMENTS

The author expresses his appreciation to Dr. M. Z. Lee of Litton Industries, R. L. Brinker of Pennsylvania Electric Company, and R.L. Heisler and P. A. Schmitzer of Gilbert/Commonwealth for their helpful comments in the preparation of this paper.

REFERENCES

1. Lee, M.Z., and Muldoon, J.B., "Recorded Pressure Surges after Turbine Trips", Fluid Transients and Acoustics in the Power Industry, edited by C. Papadakis and H. Scarton, ASME 1978, pp. 17-23

2. Ying, S.P., and Shah, V.J.,"Transient Pressure in Boiler Steam Lines", Fluid Transients and Acoustics in the Power Industry, edited by C. Papadakis and H. Scarton, ASME 1978, pp. 13-16

3. Lee, M.Z., and Goodling, E.C., "Steamhammer in Power Plant Piping Restraint Design and Optimization", ASME Pressure Vessels and Piping Conference, Orlando, 1982, PVP-Vol. 69, pp. 1-14.

4. R.T. Bradshaw, WAVENET - Waves in Fluid Networks, Program User Guide 1.0, 1976

5. Ranson, V.H., et al, RELAP5/MOD1 Code Manual, EG&G-Idaho, Inc., Idaho Falls, March 1982

6. Rochino, A.P., unpublished procedures for steamhammer analysis

7. Biggs, J.M., Introduction to Structural Dynamics, McGraw-Hill, Inc., New York, pp. 229-230

8. Goodling, E.C.,"The Disc Spring Pipe Restraint", ASME Pressure Vessels and Piping Conference, Pittsburgh, 1988, PVP-Vol. 144, pp. 187-192.

9. Kellogg, M.W. Co., Design of Piping Systems, 2nd Edition, Wiley, New York, 1956, pp. 260-266.

10. Snow, R.A., "Use of Dynamic Load Factors Less than 1.0 for Short Term Transient Loads", Gilbert/Commonwealth, Inc., Dec. 1988.