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Practical Considerations in Piping Analysis

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FOREWORD

The piping design industry has witnessed an enormous increase in the sophistication of the approaches and methodologies used during the past decade. This increase in the sophistication is a direct result of both the complexity in the current generation of power plants and process facilities and is also due to the evermore stringent regulatory requirements.

In order to meet these new challenges, today's piping designers and engineers must apply innovative techniques to produce high quality work in a timely manner.

The papers assembled in this volume will assist the practising piping engineers in achieving these objectives since the material covers a broad spectrum from practical applications to theoretical investigations.

On behalf of the American Society of Mechanical Engineers, I would like to thank the authors and co-editors for their dedicated efforts to provide this new and exciting material. I also appreciate the long hours spent by Ms. Jane Gonzalez, who organized and coordinated the production of this volume. Their combined efforts will assure not only a successful session in Orlando, but also have resulted in a publication of lasting value.

Eric van Stijgeren
Cygna Energy Services

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STEAMHAMMER IN POWER PLANT PIPING EVALUATION AND RESTRAINT DESIGN OPTIMIZATION

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ABSTRACT

The design of power plant piping restraints for steamhammer loads ranges from nonexistent to gross overdesign. The analytical tools for determining steamhammer load are available and have been substantiated by actual test measurements. Structural dynamic analysis procedures are well established in the industry, but if the available computer codes are not used properly, the resulting calculated support and restraint loads may vary by an order of magnitude. This paper summarizes recent experiences in interpretation of unbalanced forces implied by the pressure and momentum transients analysis, and typical problems encountered in time history dynamic analysis of piping structures. A method of quick estimation of maximum support loads is given, based on steam temperature, pressure, and flow rate. The data can be used in estimating loads on structure at a very early stage in power plant design and for comparison with computer analysis results for reasonableness of the latter. Design experiences on several fossil and nuclear power plants are discussed.

NOMENCLATURE

C	=	acoustic velocity in fluid
dA	=	vector representing an outflow area element
dv	=	element of volume
F	=	the resultant of all forces acting on the fluid
g	=	gravitational acceleration constant
K	=	proportional constant
L_c	=	pipe length that a pressure wave travels within an effective valve operating time
m	=	mass
p	=	pressure
R	=	pipe reaction force
u	=	flow velocity
V	=	flow velocity from which the turbine is tripped
w	=	flow rate
ρ	=	mass density
τ	=	local shear force in the direction of fluid motion

INTRODUCTION

Rapid changes in flow conditions and fluid state in a steam piping system generate pressure and momentum transients, causing dynamic shock loading on the pipe. These shock loads are generally called steam hammer loads. Thermalhydraulic transient (steam hammer) loads on power plant steam piping affect primarily the main and reheat steam, turbine bypass, and safety relief valve (SRV) blowdown piping. These transients are caused by either quick closing of a valve such as a turbine stop valve, or by quick opening of a valve, such as a bypass valve or SRV.

Thermalhydraulic transient-induced loads have generally been addressed in nuclear power plant design because of great concern for public safety. In contrast, they have often been ignored in less conservatively-designed fossil power plants. The writer's experiences have shown, however, that steam hammer phenomena do occur frequently in fossil plant steam piping. The resulting problems, including high piping stresses and excessive reactions on equipment connections can be resolved by using current analytical tools (verified by measurements) in the design of appropriate restraints. The cost of upgrading the design of existing plants is usually realistic and affordable.

TURBINE TRIP STEAMHAMMER

When a turbine-generator unit trips at substantial load, the turbine-generator unit tends to accelerate. This is because the unit has zero output instantly, while its non-zero input remains unchanged. In order to prevent the turbine-generator from overspeeding, the turbine inlet valve must be closed quickly. However, the stoppage of fluid flow caused by quick valve action creates a pressure surge in the pipes. This pressure wave, traveling upstream in the pipe reaches different bends at different times causing time- and location-dependent unbalanced forces on the pipe. The pipe may be overstressed either by excessive pipe movement or by overpressure, or combination of both.

Theoretically, the closing time of the turbine inlet valve must be optimally determined and field-adjusted to obtain an acceptable compromise between the speed rise of the turbine and the pressure rise in the pipe. This has been a common practice in hydroelectric power plant design for decades. In contrast, in a steam power plant, the turbine stop valves are designed solely to protect the turbines, without consideration for their impact on the pipes. It becomes the piping engineer's responsibility to identify this impact and to advise the client that these loads should be considered in the piping support/restraint design for safe and economical operation of the plant.

SAFETY RELIEF VALVE BLOWDOWN STEAMHAMMER

The steamhammer load on piping due to pressurizer safety relief valve blowdown, poses a problem entirely different from the one just described. A spring-loaded safety valve pops open in less than 40 milliseconds when it is subjected to overpressure. The high-energy fluid accelerates as it flows through the discharge pipe filled with air and tends to cause air shock preceding the flow. The problem is more complicated if the piping upstream of the valve contains water in the loopseal. In particular, if the temperature of the loopseal water is in the 200° F (93° C) range, water may remain in the liquid state after being discharged through the valve. This water slug, accelerated by the steam following behind, may result in severe loads on the pipe. This phenomenon, as well as the analytical methods to predict piping loads, is currently being investigated under the direction of Electric Power Research Institute (1).*

HISTORY OF DESIGN FOR STEAMHAMMER LOADS

Until about 15 years ago, steamhammer effects in steam power plants were generally ignored by designers. In old, small fossil units, piping runs were not long, and stop valve closure was not as rapid. Consequently, an occasional unit trip produced little in the way of pipe movement to alarm the operators. However, the larger units that began to go into service in the 1960's had much taller boilers, and much longer and more flexible runs in the main and reheat steam piping. Boiler capacities reaching 5,000,000 pounds per hour or more and the incorporation of fast operating stop valves on the high-pressure and intermediate-pressure turbine inlets resulted in large steamhammer forces following full load unit trip, as evidenced by sudden large pipe movements. This was especially true in the relatively thin-wall, cold-reheat piping carrying heavy steam. However, after one or two experiences of witnessing pipe response to a unit trip, many operators came to accept this as characteristic of these large new units. Coverage of the steamhammer phenomenon in the piping codes is sparse, and is typically mentioned only incidentally as one of the conditions to be considered in the analysis for occasional upsets.

In 1966, Coccio (2) presented a method for calculating the pressure waves following rapid valve closure in a steam supply line. However, this procedure

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*Underlined numbers in parentheses indicate reference at end of paper.

yielded only the momentary unbalanced forces inside the pipe, but not the actual pipe load history itself. A factor, usually 2.0, was used to account for the dynamic nature of the unbalanced force, and this, coupled with the notorious overestimation of pipe movements perceived visually, again yielded rather conservative but still tentative results.

Also, about this time, computer codes became available for calculating both the time-dependent fluid-transient forcing functions caused by the steamhammer phenomenon and the piping system response to these transient forces. The usual procedure now is to generate tables of unbalanced forces for some suitable time increment, generally 0.001 to 0.005 second, for a time duration long enough to ensure coverage of at least one fully reflected wave. The tables thus generated can then be incorporated directly to the time history system response computer program used for piping analysis.

PHYSICAL EVIDENCE

In power plants which do not have adequate steamhammer restraints on the main and reheat steam piping, a variety of evidence due to steamhammer following stop valve closure concerning system response to the unbalanced forces has been collected. Following are some examples:

1. Insulation damage - dented and abraded lagging, crushed insulation at penetrations or near steel, other pipes or pipe hanger rods. Three-dimensional reconstruction of the actual movements is sometimes possible with careful measurement of location and size of damage.

2. Damage to structure - bent structural steel members adjacent to a moving pipe or its support hardware.

3. Damage to penetrations - bent or torn architectural siding or curtain walls through which pipe passes.

4. Restraint damage - structural damage to lateral and axial restraints designed for thermal reactions but not for steamhammer loads.

5. Support damage - damage or failure of deadload supports overloaded as a result of the added steamhammer effect. Rigid supports are more susceptible, but a bottomed-out constant or variable spring support may also become overloaded. Damage in vertical linear supports occurs primarily in tension due to downward steamhammer loads. It also occurs in compression, but more rarely, since the upward force which occurs

during the initial or reflected wave is greatly offset by the weight of the pipe.

6. Eyewitness reports - reports from operators or others who have observed a moving pipe and can recall approximate magnitude and direction of movements.

MEASURED DATA

Several actual tests were performed, in which suitably instrumented operating units were purposely tripped at various power settings. These included tests on two 800 MW nuclear units, reported by Lee and Muldoon (3) and on a 660 MW oil-fired unit, reported by Ying and Shah (4). Pressure rise was measured in the main steam piping and corresponded well with pressure rise predicted analytically.

In addition to correlation of measured and calculated pressures, a combined pressure wave/time history analysis was recently performed for a 900 MW coal-fired unit following the discovery of a complete failure of a principal rigid support on the cold reheat piping. Further, an operator stated that while standing next to a long horizontal run of the cold reheat pipe during a full load unit trip, he witnessed an axial movement of 'a foot and a half'. The computer-aided analysis not only predicted reactions on the rigid hanger sufficient to cause failure, but it also predicted a movement of approximately 15 inches in the run where the movement was witnessed. These results closely correspond with the physical evidence.

NEED FOR ANALYSIS

Analysis of steamhammer effects in power plant piping is performed for these three purposes:

1. To determine upset stresses in the pipe and reactions on the terminal connections. The stresses can then be compared with the allowable levels, to determine compliance with the Piping Code requirements. The terminal reactions can be evaluated to determine their acceptability with respect to the turbine and boiler/steam generator connection allowables.

2. To predict movements of the piping in order to verify clearances or to predict possible interferences with nearby piping or structure.

3. To establish design loads for pipe supports and restraints. It is necessary for the initial design or subsequent design verification of the

adequacy of rigid supports and any restraints installed primarily for control of thermal movements. Additional rigid or dynamic restraints required in order to react steamhammer forces are also designed on the basis of results of analysis in which these restraints are modeled.

ANALYTICAL TOOLS

Analysis of piping responses caused by the steamhammer phenomenon consists of three stages. These are: (1) thermalhydraulic analysis to predict the flow conditions and the states of the fluid; (2) calculations of the time and location dependent force on the pipe caused by fluid in the pipe; and (3) structural analysis of the piping system subjected to the time dependent force induced by fluid. Many computer codes include the thermalhydraulic analysis and the forcing function generation in a single package, while other thermalhydraulic analysis codes require the forcing functions be generated by a separate postprocessor. For transients involving a single phase of fluid, such as turbine trip steamhammer, or waterhammer in liquid filled ines, most of the commercially available codes such as WHAM (5), WAVENET (6), etc. are adequate. Measured data taken at 800 MW class nuclear power plants (3) show that the classical waterhammer analysis methods are practically acceptable. Computer codes, however, can generate time dependent forcing functions in more detail for dynamic structural analysis of the piping system. The most recent version of RELAP5/MOD1 seems to predict two phase flow fluid behavior in the SRV discharge lines reasonably well. The conclusions are yet to be verified by the EPRI test program (7).

Despite the abundance of computer codes available for the thermalhydraulic transient analysis, and the power and sophistication of some of them, the use of such codes requires that the users have sufficient knowledge in interpreting the reasonableness of the analysis results. Furthermore, the validity of thermalhydraulic analysis results is not apparent until the forcing functions generated from them are judged to be reasonable.

The linear momentum equation for the fluid in a control volume, based on Newton's second law for a system, is:

$$F = \frac{d}{dt}(mu) = \frac{\partial}{\partial t}\int_{c.v.} \rho u dv + \int_{c.s.} \rho u u . dA \qquad (1)$$

where F = the resultant of all forces acting on the fluid

m = total mass

u = velocity of the center of mass of the fluid

dA = vector representing an outflow area element

dv = element of volume

ρ = mass density

and c.v. and c.s. refer to control volume and control surface.

The forces exerted on the fluid in the control volume are comprised of force due to pressure and friction forces on the wetted walls of the container, force due to pressure of the adjacent fluid (a.f.) element and the body force.

$$F = -\int_{wet.} p dA - \int_{wet.} \tau dA - \int_{a.f.} p dA - \int_{c.v.} \rho g dv \qquad (2)$$

where τ denote the local shear force in the direction of fluid motion. The resultant reaction force on the container is comprised of the first two terms of Eq. (2). From the above two equations, the component of reaction force in direction i is:

$$R_i = -\left\{\frac{\partial}{\partial t} \int_{c.v.} i.u(\rho dv) + \int_{af} i.(p + \rho uu) .dA + \int_{c.v.} i.g(\rho dv)\right\} \qquad (3)$$

The unbalanced force on a bounded pipe segment omitting the gravity is:

$$R_i \simeq \frac{-\partial}{\partial t} \int_{c.v.} i.\rho u dv \qquad (4)$$

which is the time rate of change of momentum inside the control volume. It should be noted that, at steady-state, the net unbalanced force diminishes to zero.

The unbalanced force on the pipe can be computed for each straight pipe segment or for each bend. Each force can be given by components, either in global coordinates or in local coordinate of pipe. Scalar, time-dependent net unbalanced forces on straight pipe segments, bounded at both ends by bends or tees in the axial directions of the pipe segments, are the easiest ones to generate and most convenient data to apply. This method gives the analyst the clearest physical insight to the nature and the magnitude of the forces, while, at the same time, requiring the minimum file length for forcing function data storage. For some piping anaylsis computer codes that do not allow a force to be applied in local coordinates of pipe, the scalar force in the axial direction of the pipe can be projected on global coordinates by using direction cosines as scaling factors.

STEAMHAMMER LOAD ESTIMATION

No matter how extensive the calculations one chooses to perform, it is always necessary to know a correct approximation of the key solutions either from experience or from use of other simple means. Despite the speed and accuracy of the digital computer, the computed results are only as good as the ability of the coded program to reflect the intent of the user. On the other hand, early stages of plant design require correct estimation of piping loads on the building structure. Therefore, it is important for an engineer to know a ball-park figure at different stages of design, to assure that major design parameters are in line.

For turbine trip steamhammer, the pressure surge can be calculated by:

$$\Delta p = 1.05 \frac{\rho VC}{144} \qquad (5)$$

where Δp = pressure surge, psi

ρ = mass density of steam, $\frac{\text{lb sec}^2}{\text{ft}^4}$

C = acoustic velocity of steam, ft./sec.

V = flow velocity of steam, ft./sec., from which the turbine is stripped.

The five percentile factor used is usually sufficient to cover the change in density with respect to pressure.

The maximum unbalanced force on the pipe is proportional to the flow rate being stopped.

$$\Delta F = \frac{1.05}{32,2\text{x}3.6} CW = KW \qquad (6)$$

where ΔF = maximum unbalanced force, Kips

W = flow rate, million pounds per hour

The proportional constant, K, for typical steam conditions are tabulated below.

Pressure (psi)	Temp. (^{o} F)	C (ft/sec)	K (kips/10^{6}#/hr)
700	570	1570	14.2
900	560	1500	13.6
1100	600	1520	13.8
2300	1000	1900	16.4

It is not unusual to find one or two pipe segments in a main steam system that are long enough to intercept a full pressure wave and be subjected to the maximum unbalanced force. The majority of pipe segments are not long enough to intercept a full wave, and the unbalanced force on them can be computed by using unbalanced force per unit length of pipe. The latter can be estimated by

$$\frac{\Delta F}{\Delta L} = \frac{1.05}{32.2\text{x}3.6} \frac{W}{t_c} \qquad (7)$$

where t_c is the effective valve closing time, the specifics of which will be discussed shortly. Combination of Eqs 6 and 7 results in the relationship of the ramp function for the maximum unbalanced force in a piping segment, in terms of segment length with slope change to constant force at L_c as shown in Fig. 1. L_c is a pipe length which a pressure wave travels within an effective valve closing time. 'Valve closing time' is a term loosely used. It may include signal transmission time, dead band time, and valve stroke time. Only valve stroke time is essential to steamhammer load, and only the rate of change in flow rate affects the piping load. The effective valve closing time is usually 1/3 to 1/4 of the total stroke time.

FICTITIOUS STEAMHAMMER LOAD

A disturbing fact frequently encountered in analysis is that steamhammer loads shown in different computer outputs for the same problem may differ by a factor of ten or more. This situation is usually caused by inadequate theoretical development, poor numerical algorithm used for solution of theoretical equations, incorrect interpretation of vague instructions in the user's manual, or simply an input error. For example, the correct use of a certain network wave code may require that each pipe section be an integer multiple of the wave-step chosen, but this requirement is not spelled out in the user's manual. An innocent analyst, not knowing this limitation of the code, is likely to see fictitiously high loads in the solution.

The numerical computation of force from an equation as simple as Eq. 4 may easily inflate the solution beyond what any realistic support hardware can accommodate. When the low-temperature loopseal of a PWR SRV is discharged, the density, velocity, and other fluid states fluctuate violently, with respect to time and space. The magnitude of the momentum under the integration of Eq. 4 may fluctuate in a solution time step as small as 10^{-5} second. Because of the short time step used, the time derivative would be highly oscillatory. What is worse is that the magnitude of the solution may vary by a factor of ten or more, depending on the time step chosen for force calculation. It is always necessary to review the reasonableness of all assumptions and the load solution before any hardware design is undertaken.

PIPE ANALYSIS

Time domain analysis is usually required for prediction of piping behavior under steamhammer load. There are basically two time domain analysis methods: modal superposition, and direct integration. For analysis of quick transients, the direct integration method is usually preferable.

Support attachment eccentricities should be modeled to account for the moments created by the load acting at the centerline of a pipe. Ignoring this eccentricity may result in overlooking critically high stresses in a pipe. Support stiffness should be considered in the analysis to more realistically predict the support load and piping stress. This is especially important for a support which is critical for protecting an equipment connection. For extremely large and rapid forces with short duration, such as those caused by collapse of steam bubbles, or pressure wave induced from a pipe rupture, etc., pipe restraint loads may differ substantially if the flexibility of the restraint is considered. This is because the flexibility of a support, as a result of filtering effect, decreases a dynamic load factor for certain force characteristics.

Damping factor is not important for fast and short transients such as those caused by steamhammer load. However, damping factor may affect the responses of branch lines not directly subjected to steamhammer loads.

Theoretically, it is ideal to model a piping system from anchor to anchor to reflect the overall character of a piping system structure. However, a system may be too large for a computer code to handle, or considerable expense may be incurred for using a large amount of direct access memory in a computer. Proper and improper models of the same piping system can result in the execution costs of a steamhammer analsis differing by a factor of twenty or more. It should be noted that the size of an analysis model is quite different from the physical size of the piping system being analyzed. An analyst should also be aware of the fact that a computer analysis model for a steamhammer piping dynamic analysis could be quite different from that of static and seismic analysis. In other words, an analyst may be able to use a considerably smaller number of nodes in the steamhammer analysis to cut the computer cost without compromising the accuracy of results. This, however, requires thorough understanding and a good feel for structure dynamics, as well as an understanding of the behavior of fluid transient forces.

RESTRAINT DESIGN CONSIDERATIONS

Steamhammer pipe restraints can be classified into three main types: rigid, spring, and dynamic. There are special design conditions involved for each, which must be considered, as follows:

1. Rigid restraints - These must be designed to accommodate thermal loads as well as steamhammer loads, and must be placed so that thermal reactions on the terminal connections will not be adversely affected. Rigid restraints can be of either the linear type (struts with self-aligning close tolerance pinned end fittings) or of the fabricated structural type. This latter is usually in the form of an integral trunnion welded directly to the pipe, with a close-fitting double acting stop attached to building structure.

2. Spring restraints - Double acting spring struts can be very useful in reducing the structural reactions to steamhammer forces. This is a linear type of restraint which incorporates a series of Belleville (disk) springs with a combined stiffness which is typically about 10 to 20 kips/inch. The restraint compresses as the momentary load is applied dynamically, absorbing a substantial amount of the energy in the pipe. This permits the pipe to move slightly (perhaps an inch or two) during steamhammer, but greatly reduces the net force necessary to control the movement of the pipe. In order for the spring restraint to be equally effective during the initial and reflected wave, it must be placed where axial thermal movement is minimal. Modeling restraints as stiff dynamic springs in the time history steamhammer analysis can show a reduction of as much as 50% in the magnitude of the restraint reactions, in comparison with the restraint reactions in the rigid restraint model. Spring-type restraints may be used only on relatively flexible runs, where the movements permitted will not overload the equipment connections.

3. Dynamic restraints or shock arrestors (snubbers) - These are used where thermal movement of the pipe at the point of restraint must be permitted. Snubbers are approximately linear in nature and are of two types:

hydraulic and mechanical. The choice of one or the other is generally based on the personal preferences and experiences of the owner or designer. Hydraulic snubbers require periodic servicing or they will go dry and become ineffective in a passive sense (offering no resistance to movement unless they bottom out). Mechanical snubbers, on the other hand, are relatively maintenance-free; however, a failure of this type to function may turn them into rigid struts that inappropriately act as thermal restraints.

Once the location, type, and design load for a steamhammer restraint have been established, and the restraint itself has been designed, the pipe attachment and steel attachment must be carefully designed. Load stresses in the pipe wall must be analyzed, and reinforcement provided, if necessary, to keep combined stresses within the limits permitted by the piping codes. The building attachments and structure must be analyzed to determine if they can react and distribute the restraint loads appropriately. Supplementary steel members must be sized adequately, and particular attention must be paid to the method of attaching them to the primary building structure, to permit forces to be transferred efficiently across the connections.

CODE TREATMENT

Until 1980, Section III of the Boiler and Pressure Vessel Code stated that the longitudinal pressure stress, sustained bending stress, and bending stress due to occasional loads (including those resulting from pressure and flow transients) shall be combined and must meet the requirements of Equation 9 in Subsection NC-3650. This combination of stresses must not exceed 1.2 S_h where S_h is the allowable stress for Class 2 and Class 3 components. (In 1981, the Code was revised to use the Class I approach for Class 2 and Class 3 systems, resulting in higher calculated stresses and higher allowables.)

The ANSI B31.1 Code for non-nuclear power cycle piping contains a similar requirement in paragraph 104.8.2, which states that the combined stresses shall not exceed kS_h, with the k factor prescribed as 1.2 for occasional loads acting less than 1% of operating time.

Despite the implied 20% increase in the allowable stress over the stress level S_h, permitted for sustained loads in both Section III and ANSI B31.1, there is growing sentiment among Code committee members that the 1.2 factor should not be permitted when the overload is dynamic in nature or when it is applied suddenly as an impact load. Steamhammer forces are generated rather suddenly, and it may be that specific Code clarification of this point may be needed.