NONMANDATORY APPENDICES

NONMANDATORY APPENDIX II
RULES FOR THE DESIGN OF SAFETY VALVE INSTALLATIONS[1]

FOREWORD

ASME B31.1 contains rules governing the design, fabrication, materials, erection, and examination of power piping systems. Experience over the years has demonstrated that these rules may be reasonably applied to safety valve installations. Nevertheless, instances have occurred wherein the design of safety valve installations may not have properly and fully applied the ASME B31.1 rules. Accordingly, this nonmandatory Appendix to ASME B31.1 has been prepared to illustrate and clarify the application of ASME B31.1 rules to safety valve installations. To this end, Appendix II presents the designer with design guidelines and alternative design methods.

II-1 SCOPE AND DEFINITION

II-1.1 Scope

The scope of Appendix II is confined to the design of the safety valve installations as defined in para. 1.2 of this Appendix. The loads acting at the safety valve station will affect the bending moments and stresses in the complete piping system, out to its anchors and/or extremities, and it is the designer's responsibility to consider these loads. Appendix II, however, deals primarily with the safety valve installation, and not the complete piping system.

The design of the safety valve installation requires that careful attention be paid to

(A) all loads acting on the system

(B) the forces and bending moments in the piping and piping components resulting from the loads

(C) the loading and stress criteria

(D) general design practices

All components in the safety valve installation must be given consideration, including the complete piping system, the connection to the main header, the safety valve, valve and pipe flanges, the downstream discharge or vent piping, and the system supports. The scope of this Appendix is intended to cover all loads on all components. It is assumed that the safety valve complies with the requirements of American National Standards prescribed by ASME B31.1 for structural integrity.

This Appendix has application to either safety, relief, or safety-relief valve installations. For convenience, however, the overpressure protection device is generally referred to as a safety valve. The loads associated with relief or safety-relief valve operation may differ significantly from those of safety valve operation, but otherwise the rules contained herein are equally applicable to each type of valve installation. See para. II-1.2 for definition.

This Appendix provides analytic and nomenclature definition figures to assist the designer, and is not intended to provide actual design layout (drains, drip pans, suspension, air gaps, flanges, weld ends, and other design details are not shown). Sample problems have been provided at the end of the text to assist the designer in application of the rules in this Appendix.

II-1.2 Definitions (Valve Descriptions Follow the Definitions Given in Section I of the ASME Boiler and Pressure Vessel Code)

safety valve: an automatic pressure relieving device actuated by the static pressure upstream of the valve and characterized by full opening pop action. It is used for gas or vapor service.

relief valve: an automatic pressure relieving device actuated by the static pressure upstream of the valve that opens further with the increase in pressure over the opening pressure. It is used primarily for liquid service.

safety relief valve: an automatic pressure actuated relieving device suitable for use either as a safety valve or relief valve, depending on application.

[1] Nonmandatory appendices are identified by a Roman numeral; mandatory appendices are identified by a letter. Therefore, Roman numeral I is not used, in order to avoid confusion with the letter I.

power-actuated pressure relieving valve: a relieving device whose movements to open or close are fully controlled by a source of power (electricity, air, steam, or hydraulic). The valve may discharge to atmosphere or to a container at lower pressure. The discharge capacity may be affected by the downstream conditions, and such effects shall be taken into account. If the power-actuated pressure relieving valves are also positioned in response to other control signals, the control impulse to prevent overpressure shall be responsive only to pressure and shall override any other control function.

open discharge installation: an installation where the fluid is discharged directly to the atmosphere or to a vent pipe that is uncoupled from the safety valve. Figure II-1-2(A) shows a typical open discharge installation with an elbow installed at the valve discharge to direct the flow into a vent pipe. The values for l and m on Fig. II-1-2(A) are upper limits for which the rules for open discharge systems may be used. l shall be limited to a value less than or equal to $4D_o$; m shall be limited to a value less than or equal to $6D_o$ where D_o is the outside diameter of the discharge pipe. Open discharge systems which do not conform to these limits shall be evaluated by the designer for the applicability of these rules.

closed discharge installation: an installation where the effluent is carried to a distant spot by a discharge pipe which is connected directly to the safety valve. Figure II-1-2(B) shows a typical closed discharge system.

safety valve installation: the safety valve installation is defined as that portion of the system shown on Figs. II-1-2(A) and II-1-2(B). It includes the run pipe, branch connection, the inlet pipe, the valve, the discharge piping, and the vent pipe. Also included are the components used to support the system for all static and dynamic loads.

II-2 LOADS

II-2.1 Thermal Expansion

Loads acting on the components in the safety valve installation and the displacements at various points due to thermal expansion of the piping shall be determined by analyzing the complete piping system out to its anchors, in accordance with procedures in para. 119.

II-2.1.1 Installations With Open Discharge. For safety valve installations with open discharge, there will be no thermal expansion loads acting on the discharge elbow, the valve, or the valve inlet other than that from restraint to thermal expansion as described below. Restraint to thermal expansion can sometimes occur due to drain lines, or when structural supports are provided to carry the reaction forces associated with safety valve lift. Examples of such structural supports are shown in Fig. II-6-1, sketch (b). When such restraints exist, the thermal expansion loads and stresses shall be calculated and effects evaluated.

II-2.1.2 Installations With Closed Discharge. Loads due to thermal expansion and back pressure of a safety valve installation with a closed discharge can be high enough to cause malfunction of the valve, excessive leakage of the valve or flange, or overstress of other components. The loads due to thermal expansion shall be evaluated for all significant temperature combinations, including the cases where the discharge piping is hot following safety valve operation.

II-2.2 Pressure

Pressure loads acting on the safety valve installation are important from two main considerations. The first consideration is that the pressure acting on the walls of the safety valve installation can cause membrane stresses which could result in rupture of the pressure retaining parts. The second consideration is that the pressure effects associated with discharge can cause high loads acting on the system which create bending moments throughout the piping system. These pressure effects are covered in para. II-2.3.

All parts of the safety valve installation must be designed to withstand the design pressures without exceeding the Code allowable stresses. The branch connection, the inlet pipe, and the inlet flanges shall be designed for the same design pressure as that of the run pipe. The design pressure of the discharge system will depend on the safety valve rating and on the configuration of the discharge piping. The open discharge installation and the closed discharge installation present somewhat different problems in the determination of design pressures, and these problems are discussed in the paragraphs below.

II-2.2.1 Design Pressure and Velocity for Open Discharge Installation Discharge Elbows and Vent Pipes. There are several methods available to the designer for determining the design pressure and velocity in the discharge elbow and vent pipe. It is the responsibility of the designer to assure himself that the method used yields conservative results. A method for determining the design pressures and velocities in the discharge elbow and vent pipe for open discharge installation is shown below and illustrated in the sample problem.

(A) First, calculate the design pressure and velocity for the discharge elbow.

(A.1) Determine the pressure, P_1, that exists at the discharge elbow outlet (Fig. II-2-1).

$$P_1 = \frac{W}{A_1}\frac{(b-1)}{b}\sqrt{\frac{2(h_o-a)J}{g_c(2b-1)}}$$

Fig. II-1-2(A) Safety Valve Installation (Open Discharge System)

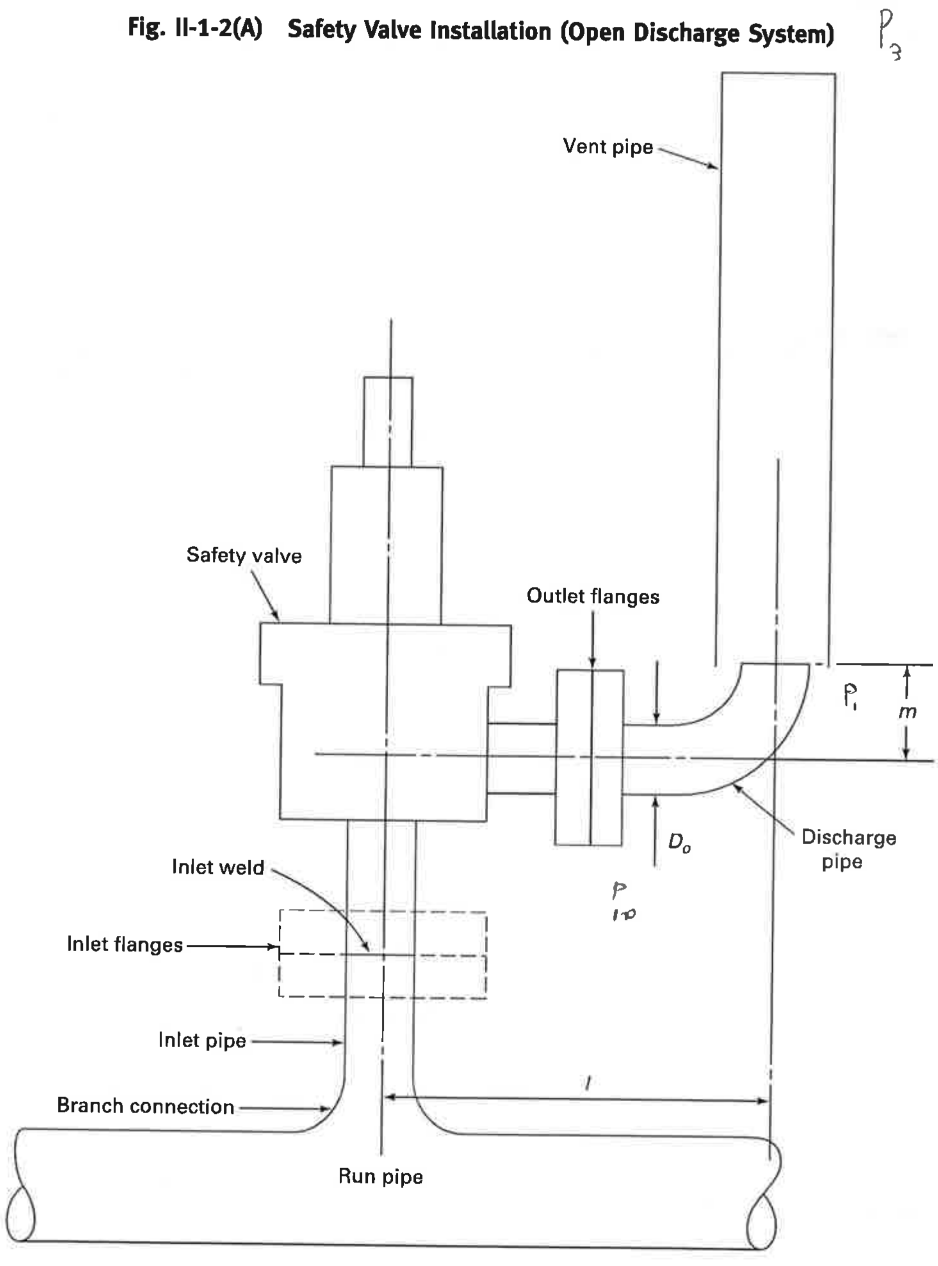


Fig. II-1-2(B) Safety Valve Installation (Closed Discharge System)

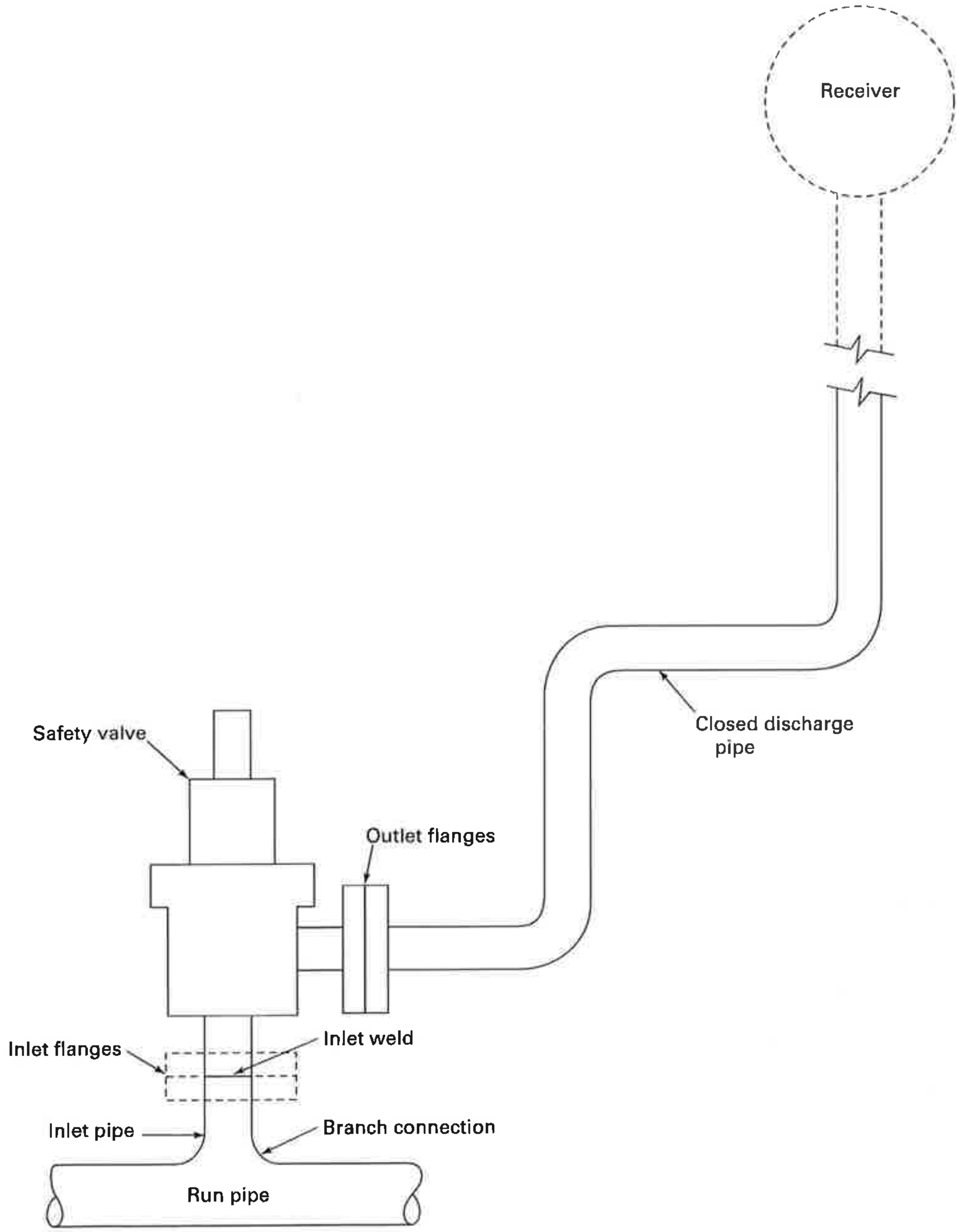
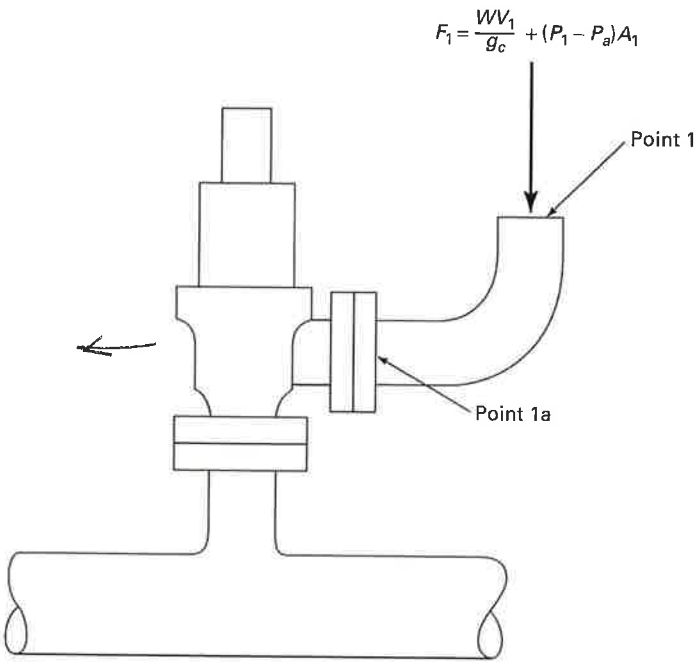


Fig. II-2-1 Discharge Elbow (Open Discharge Installation)



(A.2) Determine the velocity, V_1, that exists at the discharge elbow outlet (Fig. II-2-1).

$$V_1 = \sqrt{\frac{2g_cJ(h_o - a)}{(2b - 1)}}$$

where

A_1 = discharge elbow area, in.^{2}
g_c = gravitational constant
= 32.2 lbm-ft/lbf-sec^2
h_o = stagnation enthalpy at the safety valve inlet, Btu/lbm
J = 778.16 ft-lbf/Btu
P_1 = pressure, psia (lbf/in.^{2}, absolute)
V_1 = ft/sec
W = actual mass flow rate, lbm/sec

Common values of a and b are listed in Table II-2.2.1.

(A.3) Determine the safety valve outlet pressure, P_{1a}, at the inlet to the discharge elbow (Fig. II-2-1).

(A.3.1) Determine the length to diameter ratio (dimensionless) for the pipe sections in the discharge elbow (L/D)

$$L/D = \frac{L_{max}}{D}$$

(A.3.2) Determine a Darcy-Weisbach friction factor, f, to be used. (For steam, a value of 0.013 can be used as a good estimate since f will vary slightly in turbulent pipe flow.)

Table II-2.2.1 Values of *a* and *b*

Steam Condition	a, Btu/lbm	b
Wet steam, < 90% quality	291	11
Saturated steam, ≥ 90% quality, 15 psia ≤ P_1 ≤ 1,000 psia	823	4.33
Superheated steam, ≥ 90% quality, 1,000 psia < P_1 ≤ 2,000 psia[1]	831	4.33

NOTE:
(1) This method may be used as an approximation for pressures over 2,000 psi, but an alternate method should be used for verification.

(A.3.3) Determine a specific heat ratio (for superheated steam, k = 1.3 can be used as an estimate — for saturated steam, k = 1.1).

(A.3.4) Calculate $f(L_{max}/D)$.

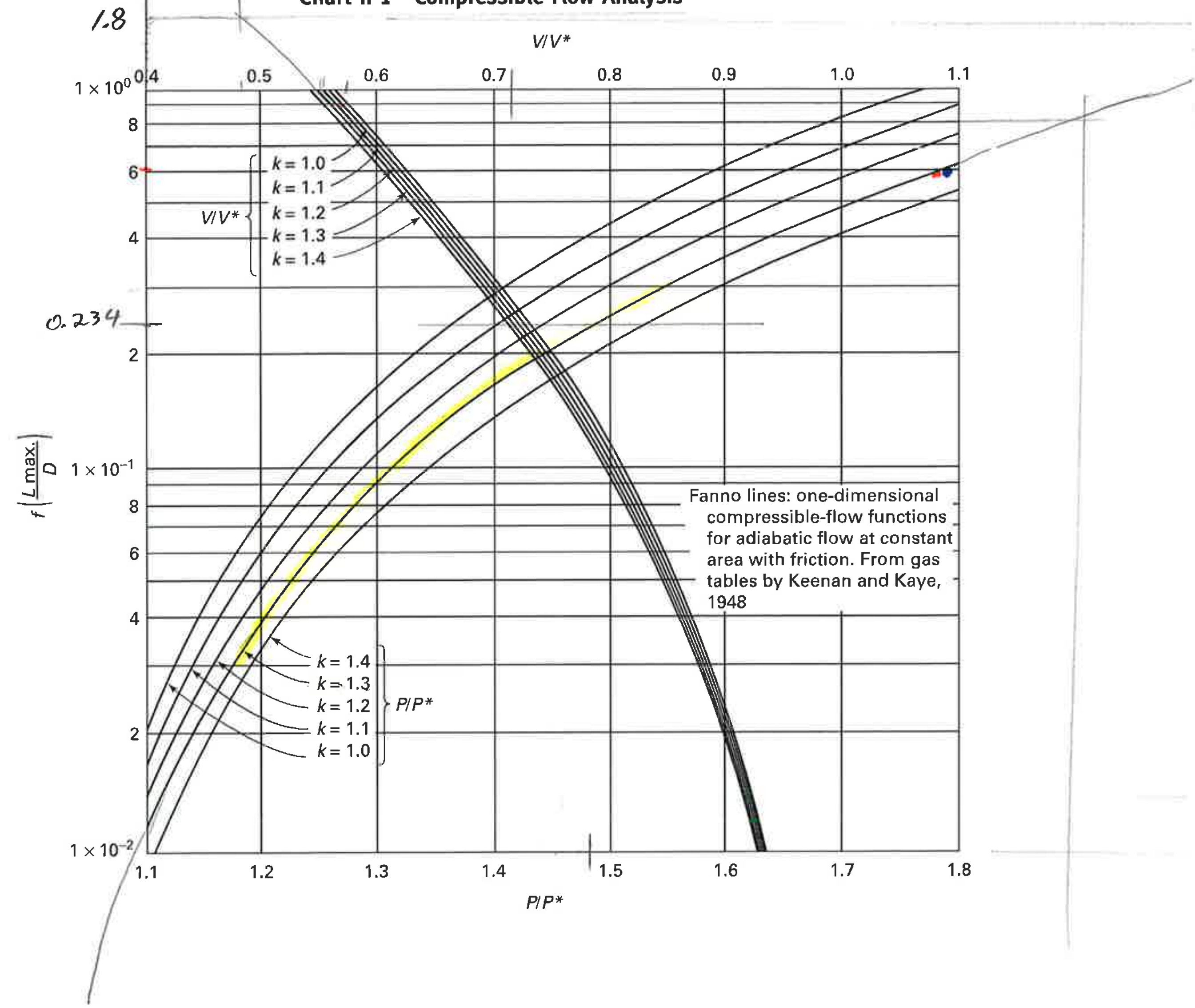
(A.3.5) Enter Chart II-1 with value of $f(L_{max}/D)$ and determine P/P^*.

(A.3.6) $P_{1a} = P_1\ (P/P^*)$.

(A.3.7) P_{1a} is the maximum operating pressure of the discharge elbow.

(B) Second, determine the design pressure and velocity for the vent pipe.

Chart II-1 Compressible Flow Analysis



(B.1) Determine the pressure, P_3, that exists at the vent pipe outlet (Fig. II-2-2)

$$P_3 = P_1 \left(\frac{A_1}{A_3}\right)$$

(B.2) Determine the velocity, V_3, that exists at the vent pipe outlet (Fig. II-2-2)

$$V_3 = V_1$$

(B.3) Repeat Steps (3.1) to (3.7) in the calculation of the discharge elbow maximum operating pressure to determine the maximum operating pressure of the vent pipe.

(B.4) Determine the velocity, V_2, and pressure, P_2, that exist at the inlet to the vent pipe (Fig. II-2-2).

(B.4.1) Enter Chart II-1[2] with value of $f(L_{max}/D)$ from Step (3.4) and determine values of V/V^* and P/P^*.

(B.4.2) Calculate V_2

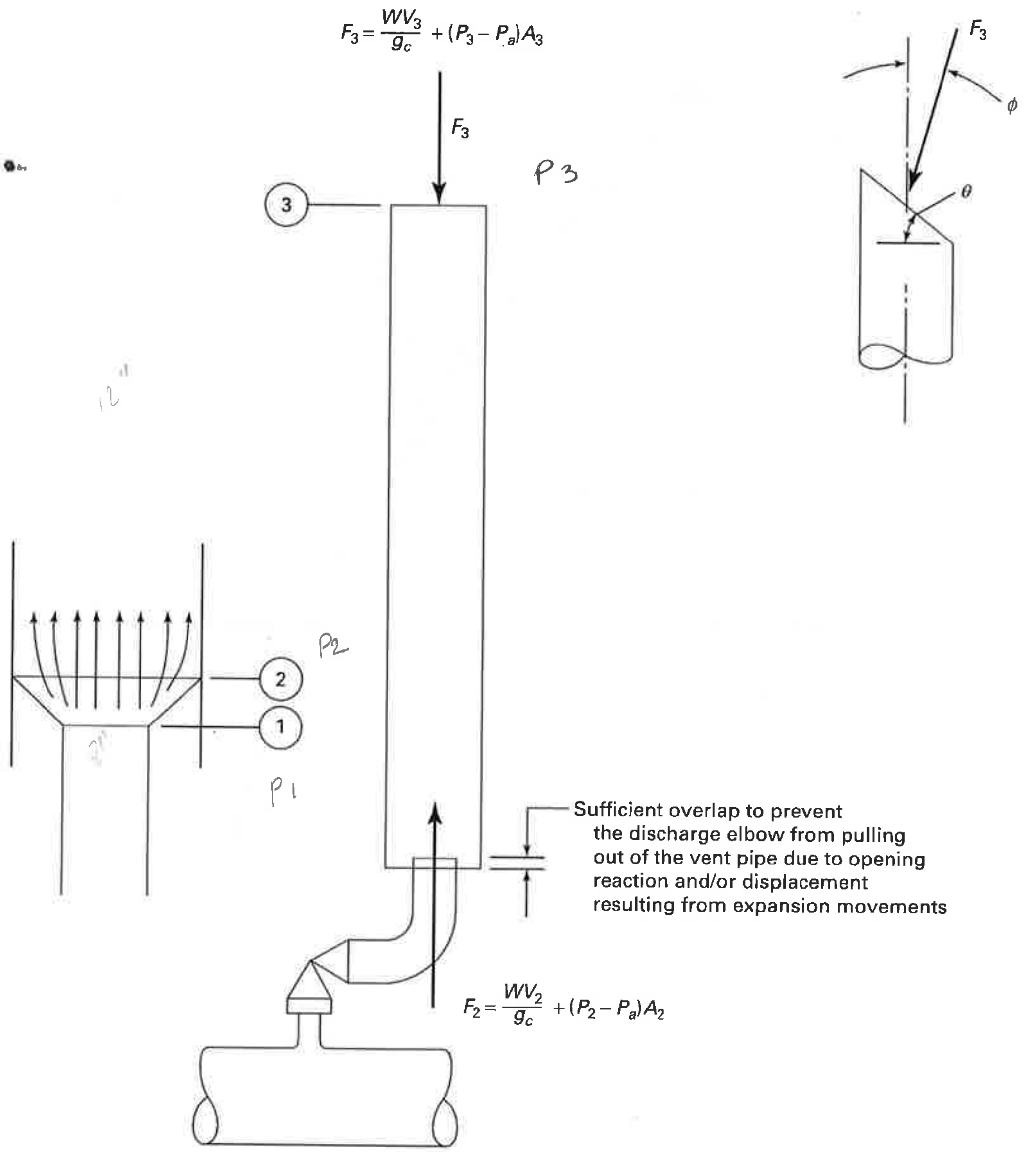
$$V_2 = V_3\,(V/V^*)$$

(B.4.3) $P_2 = P_3\,(P/P^*)$. This is the highest pressure the vent stack will see and should be used in calculating vent pipe blowback (see para. II-2.3.1.2).

II-2.2.2 Pressure for Closed Discharge Installations. The pressures in a closed discharge pipe during steady

[2] Chart II-1 may be extended to other values of $f\,(L_{max}/D)$ by use of the Keenan and Kaye Gas Tables for Fanno lines. The Darcy-Weisbach friction factor is used in Chart II-1, whereas the Gas Tables use the Fanning factor which is one-fourth the value of the Darcy-Weisbach factor.

Fig. II-2-2 Vent Pipe (Open Discharge Installation)



state flow may be determined by the methods described in para. II-2.2.1. However, when a safety valve discharge is connected to a relatively long run of pipe and is suddenly opened, there is a period of transient flow until the steady state discharge condition is reached. During this transient period, the pressure and flow will not be uniform. When the safety valve is initially opened, the discharge pipe may be filled with air. If the safety valve is on a steam system, the steam discharge from the valve must purge the air from the pipe before steady state steam flow is established and, as the pressure builds up at the valve outlet flange and waves start to travel down the discharge pipe, the pressure wave initially emanating from the valve will steepen as it propagates, and it may steepen into a shock wave before it reaches the exit. Because of this, it is recommended that the design pressure of the closed discharge pipe be greater than the steady state operating pressure by a factor of at least 2.

II-2.3 Reaction Forces From Valve Discharge

It is the responsibility of the piping system designer to determine the reaction forces associated with valve discharge. These forces can create bending moments at various points in the piping system so high as to cause catastrophic failure of the pressure boundary parts. Since the magnitude of the forces may differ substantially, depending on the type of discharge system, each system type is discussed in the paragraphs below.

II-2.3.1 Reaction Forces With Open Discharge Systems

II-2.3.1.1 Discharge Elbow. The reaction force, F, due to steady state flow following the opening of the safety valve includes both momentum and pressure effects. The reaction force applied is shown in Fig. II-2-1, and may be computed by the following equation:

$$F_1 = \frac{W}{g_c} V_1 + (P_1 - P_a) A_1$$

where

A_1 = exit flow area at Point 1, in.^{2}
F_1 = reaction force at Point 1, lbf
g_c = gravitational constant
= 32.2 lbm-ft/lbf-sec^2
P_1 = static pressure at Point 1, psia
P_a = atmospheric pressure, psia
V_1 = exit velocity at Point 1, ft/sec
W = mass flow rate (relieving capacity stamped on the valve × 1.11), lbm/sec

To ensure consideration of the effects of the suddenly applied load F, a dynamic load factor, DLF, should be applied (see para. II-3.5.1.3).

The methods for calculating the velocities and pressures at the exit point of the discharge elbow are the same as those discussed in para. II-2.2 of this Appendix.

II-2.3.1.2 Vent Pipe. Figure II-2-2 shows the external forces resulting from a safety valve discharge, which act on the vent pipe. The methods for calculating F_2 and F_3 are the same as those previously described. The vent pipe anchor and restraint system must be capable of taking the moments caused by these two forces, and also be capable of sustaining the unbalanced forces in the vertical and horizontal directions.

A bevel of the vent pipe will result in a flow that is not vertical. The equations shown are based on vertical flow. To take account for the effect of a bevel at the exit, the exit force will act at an angle, ϕ, with the axis of the vent pipe discharge which is a function of the bevel angle, θ. The beveled top of the vent deflects the jet approximately 30 deg off the vertical for a 60 deg bevel, and this will introduce a horizontal component force on the vent pipe systems.

The terms in the equations shown on Fig. II-2-2 are the same as those defined in para. II-2.3.1 above.

The vent pipe must be sized so that no steam is blown back at the vent line entrance. The criteria which may be used as a guide to prevent this condition are listed below.

$$\frac{W (V_1 - V_2)}{g_c} > (P_2 - P_a) A_2 - (P_1 - P_a)A_1$$

where

A = area, in.^{2}
g_c = gravitational constant
= 32.2 lbm-ft/lbf-sec^2
P_1, P_2 = local absolute pressure, psia
P_a = standard atmospheric pressure, psia
V = velocity, ft/sec
W = mass flow rate, lbm/sec

The inequality states that the momentum at Point 1 has to be greater than the momentum at Point 2 in order that air is educted into the vent pipe. If the momentum at Point 1 equalled the momentum at Point 2, no air would be educted into the vent pipe. If the momentum at Point 1 was less than the momentum at Point 2, steam would "blow back" from the vent pipe.

The educting effect of the vent pipe is especially important for indoor installation of safety valves. The steam being vented from the upper body during safety valve operation will be removed from the area through the vent pipe. For that reason, the fluid momentum at 1 should exceed the fluid momentum at 2, not just be equal.

If this inequality is satisfied, blowback will not occur. The pressures and velocities are those calculated in para. II-2.2.1.

II-2.3.2 Reaction Forces With Closed Discharge Systems. When safety valves discharge a closed piping system, the forces acting on the piping system under steady state flow will be self-equilibrated, and do not create significant bending moments on the piping system. The large steady state force will act only at the point of discharge, and the magnitude of this force may be determined as described for open discharge systems.

Relief valves discharging into an enclosed piping system create momentary unbalanced forces which act on the piping system during the first few milliseconds following relief valve lift. The pressure waves traveling through the piping system following the rapid opening of the safety valve will cause bending moments in the safety valve discharge piping and throughout the remainder of the piping system. In such a case, the designer must compute the magnitude of the loads, and perform appropriate evaluation of their effects.

II-2.4 Other Mechanical Loads

Other design mechanical loads that must be considered by the piping designer include the following:

II-2.4.1 Interaction loads on the pipe run when more than one valve opens.

II-2.4.2 Loads due to earthquake and/or piping system vibration (see para. II-3.4).

II-3 BENDING MOMENT COMPUTATIONS

II-3.1 General

One of the most important considerations related to the mechanical design and analysis of safety valve installation is the identification and calculation of the moments at critical points in the installation. If the bending moments are not properly calculated, it will not be possible to meet the loading and stress criteria contained in ASME B31.1. As a minimum, the following loads, previously discussed in para. II-2 of this Appendix, should be considered in determining these moments:

(A) thermal expansion
(B) dead weight
(C) earthquake
(D) reaction force from valve discharge
(E) other mechanical loads

The analysis of the safety valve installation should include all critical sections, such as intersection points, elbows, transition sections, etc., and any related piping, vessels, and their supports that may interact with the safety valve installation. It is often most appropriate to model the safety valve installation and its related piping as a lumped mass system joined by straight or curved elements.

II-3.2 Thermal Expansion Analysis

There are many standard and acceptable methods for determination of moments due to thermal expansion of the piping installation. The thermal expansion analysis must comply with the requirements in para. 119. The safety valve installation often presents a special problem in that there may be a variety of operational modes to consider where each mode represents a different combination of temperatures in various sections of the piping system. The design condition shall be selected so that none of the operational modes represents a condition that gives thermal expansion bending moments greater than the design condition.

The design of the safety valve installation should consider the differential thermal growth and expansion loads, as well as the local effects of reinforcing and supports. The design should also consider the differential thermal growth and expansion loads existing after any combination of safety valves (one valve to all valves) operates, raising the temperature of the discharge piping.

II-3.3 Dead Weight Analysis

The methods used for determination of bending moments due to dead weight in a safety valve installation are not different from the methods used in any other piping installation. If the support system meets the requirements in para. 121, the bending moments due to dead weight may be assumed to be $1,500Z$ (in.-lb) where Z is the section modulus (in.^{3}) of the pipe or fitting being considered. However, bending moments due to dead weight are easily determined and should always be calculated in systems where stresses exceed 90% of the allowable stress limits in meeting the requirements of eqs. (15) and (16) of para. 104.8.

II-3.4 Earthquake Analysis

Seismic loads must be known to calculate bending moments at critical points in the safety valve installation. If a design specification exists, it should stipulate if the piping system must be designed for earthquake. If so, it should specify the magnitude of the earthquake, the plant conditions under which the earthquake is assumed to occur, and the type earthquake analysis to be used (equivalent static or dynamic). If a design specification does not exist, it is the responsibility of the designer to determine what consideration must be given to earthquake analysis. It is beyond the scope of this Appendix to provide rules for calculating moments due to earthquake. The literature contains satisfactory references for determining moments by use of static seismic coefficients and how to perform more sophisticated dynamic analyses of the piping system using inputs in such form as time histories of displacement, velocity, and acceleration or response spectra where displacement, velocity, or acceleration is presented as a function of frequency.

Two types of seismic bending moments occur. One type is due to inertia effects and the other type is due to seismic motions of pipe anchors and other attachments. As will be shown later, the moments due to inertia effects must be considered in eq. (16), para. 104.8, in the kS_h category. Moments due to seismic motions of the attachments may be combined with thermal expansion stress and considered in eq. (17), para. 104.8 in the S_A category. For this reason, it may sometimes be justified for the designer to consider the moments separately; otherwise both sets of moments would have to be included in the kS_h category.

II-3.5 Analysis for Reaction Forces Due to Valve Discharge

II-3.5.1 Open Discharge Systems

II-3.5.1.1 The moments due to valve reaction forces may be calculated by simply multiplying the force, calculated as described in para. II-2.3.1.1, times the distance from the point in the piping system being analyzed, times a suitable dynamic load factor. In no case shall the reaction moment used in para. II-4.2 at the branch connection below the valve be taken at less than the product of

$$(DLF)\,(F_1)\,(D)$$

where

D = nominal O.D. of inlet pipe
DLF = dynamic load factor (see para. II-3.5.1.3)
F_1 = force calculated per para. II-2.3.1.1

Reaction force and resultant moment effects on the header, supports, and nozzles for each valve or combination of valves blowing shall be considered.

II-3.5.1.2 Multiple Valve Arrangements. Reaction force and moment effects on the run pipe, header, supports, vessel, and connecting nozzles for each valve blowing, and when appropriate, for combinations of valves blowing should be considered. In multiple valve arrangements, each valve will open at a different time, and since all valves may not be required to open during an overpressure transient, several possible combinations of forces can exist. It may be desirable to vary the direction of discharge of several safety valves on the same header to reduce the maximum possible forces when all valves are blowing.

II-3.5.1.3 Dynamic Amplification of Reaction Forces. In a piping system acted upon by time varying loads, the internal forces and moments are generally greater than those produced under static application of the load. This amplification is often expressed as the dynamic load factor, *DLF*, and is defined as the maximum ratio of the dynamic deflection at any time to the deflection which would have resulted from the static application of the load. For structures having essentially one degree-of-freedom and a single load application, the *DLF* value will range between one and two depending on the time-history of the applied load and the natural frequency of the structure. If the run pipe is rigidly supported, the safety valve installation can be idealized as a one degree-of-freedom system and the time-history of the applied loads can often be assumed to be a single ramp function between the no-load and steady state condition. In this case, the *DLF* may be determined in the following manner:

(A) Calculate the safety valve installation period T using the following equation and Fig. II-3-1:

$$T = 0.1846 \sqrt{\frac{Wh^3}{EI}}$$

where

E = Young's modulus of inlet pipe, lb/in.^{2}, at design temperature
h = distance from run pipe to centerline of outlet piping, in.
I = moment of inertia of inlet pipe, in.^{4}
T = safety valve installation period, sec
W = weight of safety valve, installation piping, flanges, attachments, etc., lb

(B) Calculate ratio of safety valve opening time to installation period (t_o/T), where t_o is the time the safety valve takes to go from fully closed to fully open, sec, and T is determined in (A) above.

(C) Enter Fig. II-3-2 with the ratio of safety valve opening time to installation period and read the *DLF* from the ordinate. The *DLF* shall never be taken less than 1.1.

If a less conservative *DLF* is used, the *DLF* shall be determined by calculation or test.

II-3.5.1.4 Valve Cycling. Often, safety valves are full lift, pop-type valves, and are essentially full-flow devices, with no capability for flow modulation. In actual pressure transients, the steam flow required to prevent overpressure is a varying quantity, from zero to the full rated capacity of the safety valves. As a result, the valves may be required to open and close a number of times during the transient. Since each opening and closing produces a reaction force, consideration should be given to the effects of multiple valve operations on the piping system, including supports.

II-3.5.1.5 Time-History Analysis. The reaction force effects are dynamic in nature. A time-history dynamic solution, incorporating a multidegree of freedom lumped mass model solved for the transient hydraulic forces is considered to be more accurate than the form of analysis presented in this Appendix.

II-3.5.2 Closed Discharge Systems. Closed discharge systems do not easily lend themselves to simplified analysis techniques. The discussions on pressure in para. II-2.2.2 and on forces in para. II-2.3.2 indicate that a time-history analysis of the piping system may be required to achieve realistic values of moments.

II-3.5.3 Water Seals. To reduce the problem of steam or gas leakage through the safety valve seats, the valve inlet piping may be shaped to form a water seal below each valve seat. If the valves are required to open to prevent overpressure, the water from the seal is discharged ahead of the steam as the valve disk lifts. The subsequent flow of water and steam through the discharge piping produces a significant pressure and momentum transient. Each straight run of discharge piping experiences a resulting force cycle as the water mass moves from one end of the run to the other.

For most plants that employ water seals, only the first cycle of each occurrence has a force transient based on water in the seal. The remaining cycles of each occurrence would be based on steam occupying the seal piping, and the transient forces would be reduced in magnitude.

II-4 LOADING CRITERIA AND STRESS COMPUTATION

II-4.1 Loading Criteria

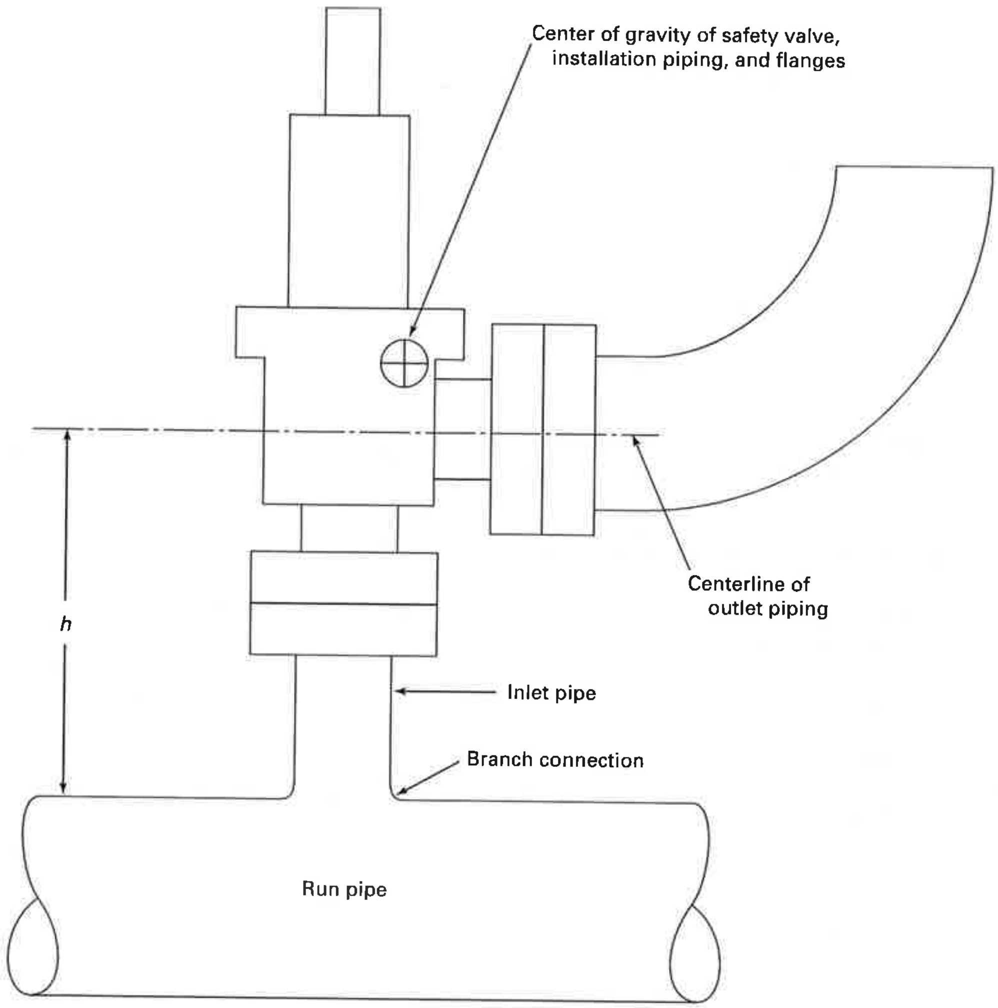
All critical points in the safety valve installation shall meet the following loading criteria:

$$S_{lp} + S_{SL} \le S_h \tag{1}$$

$$S_{lp} + S_{SL} + S_{OL} \le kS_h \tag{2}$$

$$S_{lp} + S_{SL} + S_E \le S_A + S_h \tag{3}$$

Fig. II-3-1 Safety Valve Installation (Open Discharge System)



where

S_E = bending stresses due to thermal expansion
S_{lp} = longitudinal pressure stress
S_{OL} = bending stresses due to occasional loads, such as earthquake, reaction from safety valve discharge and impact loads
S_{SL} = bending stresses due to sustained loads, such as dead weight

S_h, k, and S_A are as defined in ASME B31.1.

The three loading criteria defined above are represented by eqs. (15) and (16) in para. 104.8.

II-4.2 Stress Calculations

II-4.2.1 Pressure Stresses. The Code does not require determination of the pressure stresses that could cause failure of the pressure containing membrane. Instead, the Code provides rules to ensure that sufficient wall thickness is provided to prevent failures due to pressure. It is not necessary to repeat these rules in this Appendix; however, some of the more important are listed below for reference.

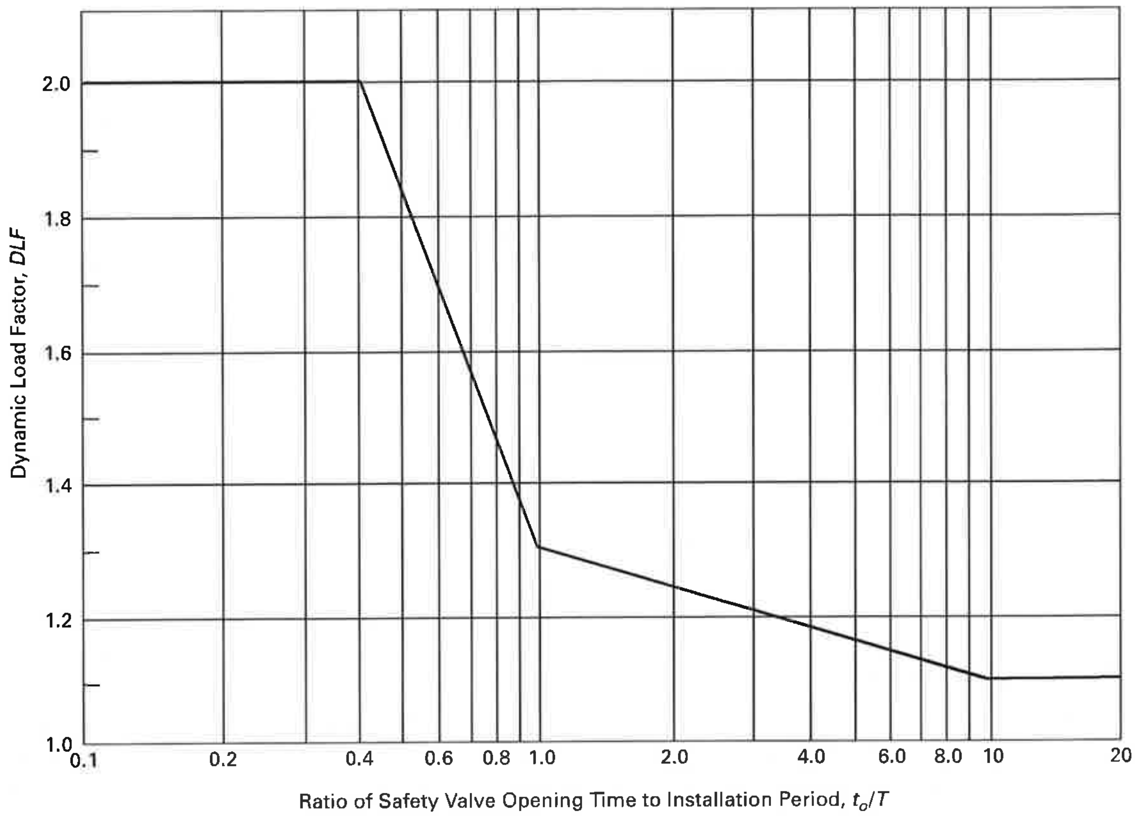
(A) All pipe (plus other components) must satisfy the minimum required wall thickness of eq. (7) of para. 104.1.2. In addition, wall thickness must be adequate to satisfy eqs. (15) and (16) in para. 104.8. These two equations may govern determination of wall thickness in low pressure systems.

(B) No minimum wall thickness calculations are needed for components purchased to approved standards in Table 126.1.

(C) Pipe bends must meet the requirements of eq. (1) above *after* bending.

(D) Branch connections that do not meet the requirements of eq. (2) above must meet the area replacement requirements of para. 104.3.

Fig. II-3-2 Dynamic Load Factors for Open Discharge System



GENERAL NOTE: This Figure is based on curves from *Introduction to Structural Dynamics*, J. M. Biggs, McGraw-Hill Book Co., 1964.

II-4.2.2 Pressure Plus Bending Stresses. To guard against membrane failures (catastrophic), prevent fatigue (leak) failures, and to ensure shakedown, the equations in para. 104.8 must be satisfied. These equations apply to all components in the safety valve installation and will not be repeated here. However, some additional explanation of these equations in regard to the very critical points upstream of the safety valve are in the paragraphs below.

II-4.2.2.1 Additive Stresses at Branch Connection. For the purposes of eqs. (15), (16), and (17) in para. 104.8, the section modulus and moments for application to branch connections, such as safety valve inlet pipes, are as follows:

(A) For branch connections, the Z should be the effective section modulus for the branch as defined in para. 104.8. Thus,

$$Z = Z_b = \pi r_b^2 t_s \text{ (effective section modulus)}$$

where

r_b = mean branch cross-sectional radius, in.
t_s = lesser of t_r and it_b, where
t_r = nominal thickness of run pipe
i = the branch connection stress intensification factor
t_b = nominal thickness of branch pipe

(B) Moment terms shall be defined as follows:

$$M_B = \sqrt{M_{x3}^2 + M_{y3}^2 + M_{z3}^2}$$ page 33

where M_B, M_{x3}, M_{y3}, and M_{z3} are defined in para. 104.8.

(C) Where the D_o/t_n of the branch connection differs from the D_o/t_n header or run, the larger of the two D_o/t_n values should be used in the first term of eqs. (15) and (16), where D_o and t_n are defined in paras. 104.1 and 104.8, respectively.

II-4.2.2.2 Additive Stresses in Inlet Pipe. Equations (15), (16), and (17) in para. 104.8 may be applied to the inlet pipe in the same manner as described above for the branch connection, except that the values for D_o/t_n and Z should be for the inlet pipe and the stress intensification factor used will be different. It should be noted that the values D_o, t_n, and Z should be taken from a point on the inlet pipe such that D_o/t_n will have a maximum and Z a minimum value for the inlet pipe.

II-4.2.3 Analysis of Flange. It is important that the moments from the various loading conditions described in para. II-4.2.2 do not overload the flanges on the safety valve inlet and outlet. One method of doing this is to convert the moments into an equivalent pressure that is then added to the internal pressure. The sum of these two pressures, P_{FD}, would be acceptable if either of the following criteria are met:

(A) P_{FD} does not exceed the ASME B16.5 flange rating.

(B) S_H, S_R, and S_T should be less than the yield stress at design temperature, where S_H, S_R, and S_T are as defined in 2-7 of ASME Section VIII, Division 1 with the following exceptions:

(B.1) P_{FD} should be used in the ASME Section VIII, Division 1 equations instead of the design pressure.

(B.2) S_H should include the longitudinal pressure stress at the flange hub.

II-4.2.4 Analysis of Valve. The allowable forces and moments which the piping system may place on the safety valves must be determined from the valve manufacturer. In some cases, the valve flanges are limiting rather than the valve body.

II-5 DESIGN CONSIDERATIONS

II-5.1 General

The design of safety valve installations shall be in accordance with para. 104 except that consideration be given to the rules provided in the following subparagraphs. These rules are particularly concerned with that portion of the piping system attached to and between the safety valve and the run pipe, header, or vessel that the valve services and includes the branch connection to the run pipe, header, or vessel.

II-5.2 Geometry

II-5.2.1 Locations of Safety Valve Installations. Safety valve installations should be located at least eight pipe diameters (based on I.D.) downstream from any bend in a high velocity steam line to help prevent sonic vibrations. This distance should be increased if the direction of the change of the steam flow is from vertical upwards to horizontal in such a manner as to increase density of the flow in the area directly beneath the station nozzles. Similarly, safety valve installation should not be located closer than eight pipe diameters (based on I.D.) either upstream or downstream from fittings.

II-5.2.2 Spacing of Safety Valve Installation. Spacing of safety valve installations must meet the requirements in Note (10)(c), Appendix D, Table D-1.

II-5.3 Types of Valves and Installations

II-5.3.1 Installations With Single Outlet Valves. Locate unsupported valves as close to the run pipe or header as is physically possible to minimize reaction moment effects.

Orientation of valve outlet should preferably be parallel to the longitudinal axis of the run pipe or header.

Angular discharge elbows oriented to minimize the reaction force moment shall have a straight pipe of at least one pipe diameter provided on the end of the elbow to ensure that the reaction force is developed at the desired angle. Cut the discharge pipe square with the centerline. Fabrication tolerances, realistic field erection tolerances, and reaction force angle tolerances must be considered when evaluating the magnitude of the reaction moment.

The length of unsupported discharge piping between the valve outlet and the first outlet elbow [Fig. II-1-2(A), distance l] should be as short as practical to minimize reaction moment effects.

II-5.3.2 Installations With Double Outlet Valves. Double outlet valves with symmetrical tail-pipes and vent stacks will eliminate the bending moment in the nozzle and the run pipe or header providing there is equal and steady flow from each outlet. If equal flow cannot be guaranteed, the bending moment due to the unbalanced flow must be considered. Thrust loads must also be considered.

II-5.3.3 Multiple Installations. The effects of the discharge of multiple safety valves on the same header shall be such as to tend to balance one another for all modes of operation.

II-5.4 Installation Branch Connections

Standard branch connections shall as a minimum meet the requirements of para. 104.3. It should be noted that branch connections on headers frequently do not have sufficient reinforcement when used as a connection for a safety valve. It may be necessary to provide additional reinforcing (weld deposit buildup) or special headers that will satisfactorily withstand the reaction moments applied.

Material used for the branch connection and its reinforcement shall be the same or of higher strength than that of the run pipe or header.

It is strongly recommended that branch connections intersect the run pipe or header normal to the surface of the run pipe or header at $\alpha = 90$ deg, where α is defined as the angle between the longitudinal axis of the branch connection and the normal surface of the run pipe or header. Branch connections that intersect the run pipe or headers at angles,

$$90 \text{ deg} > \alpha \geq 45 \text{ deg}$$

should be avoided. Branch connections should not in any case intersect the run pipe or header at angles,

$$\alpha < 45 \text{ deg}$$

II-5.5 Water in Installation Piping

II-5.5.1 Drainage of Discharge Piping. Drains shall be provided so that condensed leakage, rain, or other water sources will not collect on the discharge side of the valve and adversely affect the reaction force. Safety valves are generally provided with drain plugs that can be used for a drain connection. Discharge piping shall be sloped and provided with adequate drains if low points are unavoidable in the layout.

II-5.5.2 Water Seals. Where water seals are used ahead of the safety valve, the total water volume in the seals shall be minimized. To minimize forces due to slug flow or water seal excursion, the number of changes of direction and the lengths of straight runs of installation piping shall be limited. The use of short radius elbows is also discouraged; the pressure differential across the cross section is a function of the elbow radius.

II-5.6 Discharge Stacks

If telescopic or uncoupled discharge stacks, or equivalent arrangements, are used then care should be taken to ensure that forces on the stack are not transmitted to the valve discharge elbow. Stack clearances shall be checked for interference from thermal expansion, earthquake displacements, etc. Discharge stacks shall be supported adequately for the forces resulting from valve discharge so that the stack is not deflected, allowing steam to escape in the vicinity of the valve. In addition, the deflection of the safety valve discharge nozzle (elbow) and the associated piping system when subjected to the reaction force of the blowing valve shall be calculated. This deflection shall be considered in the design of the discharge stacks slip-joint to ensure that the discharge nozzle remains in the stack, preventing steam from escaping in the vicinity of the valve.

To prevent blowback of discharging steam from inlet end of vent stack, consider the use of an antiblowback device that still permits thermal movements of header.

II-5.7 Support Design

Supports provided for safety valves and the associated piping require analysis to determine their role in restraint as well as support. These analyses shall consider at least the following effects:

(A) differential thermal expansion of the associated piping, headers, and vessels.

(B) dynamic response characteristics of the support in relation to the equipment being supported and the structure to which it is attached, during seismic events and valve operation. Maximum relative motions of various portions of the building and structures to which supports are attached resulting from seismic excitation must be considered in selecting, locating, and analyzing support systems.

(C) capability of the support to provide or not provide torsional rigidity, per the support design requirements.

II-5.7.1 Pipe Supports. Where necessary, it is recommended that the support near the valve discharge be connected to the run pipe, header, or vessel rather than to adjacent structures in order to minimize differential thermal expansion and seismic interactions.

Each straight leg of discharge piping should have a support to take the force along that leg. If the support is not on the leg itself, it should be as near as possible on an adjacent leg.

When a large portion of the system lies in a plane, the piping, if possible, should be supported normal to that plane even though static calculations do not identify a direct force requiring restraint in that direction. Dynamic analyses of these systems have shown that out-of-plane motions can occur.

II-5.7.2 Snubbers. Snubbers are often used to provide a support or a stop against a rapidly applied load, such as the reaction force of a blowing valve or the pressure-momentum transient in a closed piping system. Since snubbers generally displace a small distance before becoming rigid, the displacement must be considered in the analysis. In addition, if the load is applied to the snubber for a relatively long time, the snubber performance characteristics shall be reviewed to ensure that the snubber will not permit motion during the time period of interest, or the additional displacement must be considered in the analysis. The snubber performance shall also be reviewed for response to repetitive load applications caused by the safety valve cycling open and closed several times during a pressure transient.

II-5.8 Silencer Installation

Silencers are occasionally installed on safety valve discharges to dissipate the noise generated by the sonic velocity attained by the fluid flowing through the valve.

Silencers must be properly sized to avoid excessive backpressure on the safety valve causing improper valve action or reducing relieving capacity.

Safety valve discharge piping, silencers, and vent stacks shall be properly supported to avoid excessive loading on the valve discharge flange.

II-6 SAMPLE DESIGNS

Examples of various safety valve installations that a designer may encounter in practice are presented in Figs. II-1-2(A) and II-6-1.

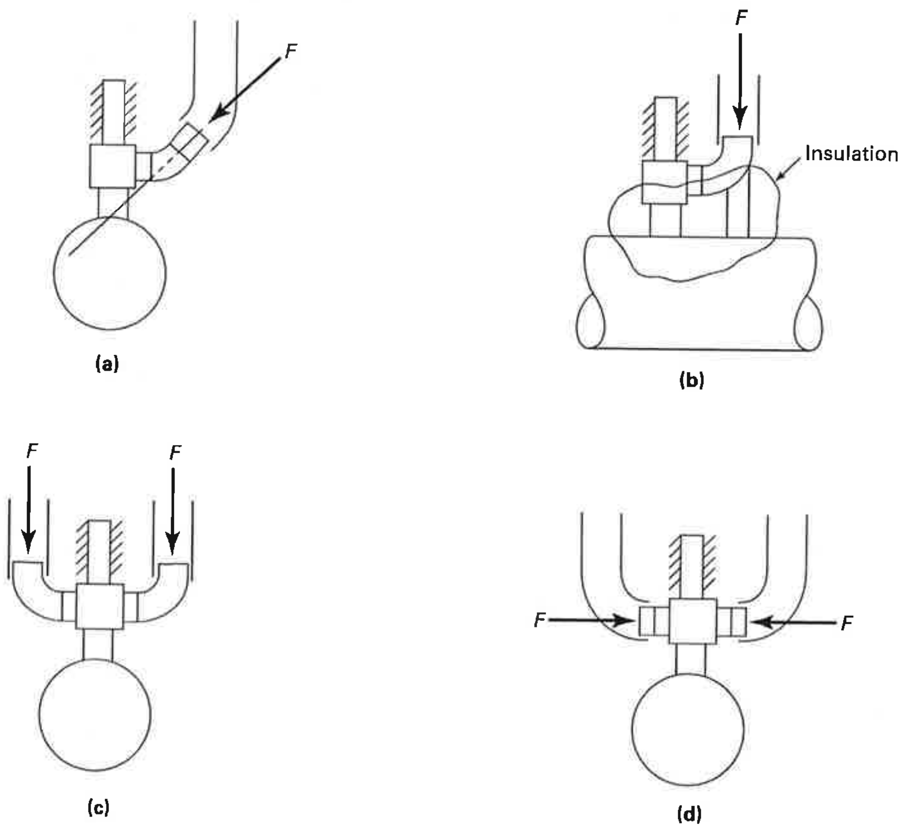
II-7 SAMPLE PROBLEM (SEE FIGS. II-7-1 AND II-7-2)

II-7.1 Procedure

(A) Determine pressure and velocity at discharge elbow exit.

(B) Calculate maximum operating pressure for discharge exit.

Fig. II-6-1 Examples of Safety Valve Installations



F = reaction force

(C) Calculate reaction force at discharge elbow exit.

(D) Calculate bending moments of Points (1) and (2) from reaction force and seismic motion.

(E) Determine stress intensification factors at Points (1) and (2).

(F) Calculate predicted stresses at Points (1) and (2) and compare with allowable stress.

(G) Calculate maximum operating pressure for vent pipe.

(H) Check for blowback.

(I) Calculate forces and moments on vent pipe.

II-7.1.1 Pressure and Velocity at Discharge Elbow Exit (Para. II-2.2.1)

$$P_1 = \frac{W}{A_1}\frac{(b-1)}{b}\sqrt{\frac{2(h_o - a)J}{g_c(2b-1)}}$$

$$V_1 = \sqrt{\frac{2g_c J(h_o - a)}{(2b-1)}}$$

where

A_1 = 50.03 in.^{2}

a = 823 Btu/lbm for $15 \le P_1 \le 1{,}000$ psia and $h_o \le$ 1,600 Btu/lbm

b = 4.33 for $15 \le P_1 \le 1{,}000$ psia and $h_o \le$ 1,600 Btu/lbm

g_c = 32.2 lbm-ft/lbf-sec^2

h_o = stagnation enthalpy for steam at 925 psia, 1,000°F

= 1,507.3 Btu/lbm

J = 778 ft-lbf/Btu

P_1 = 118 psia

V_1 = 2,116 ft/sec

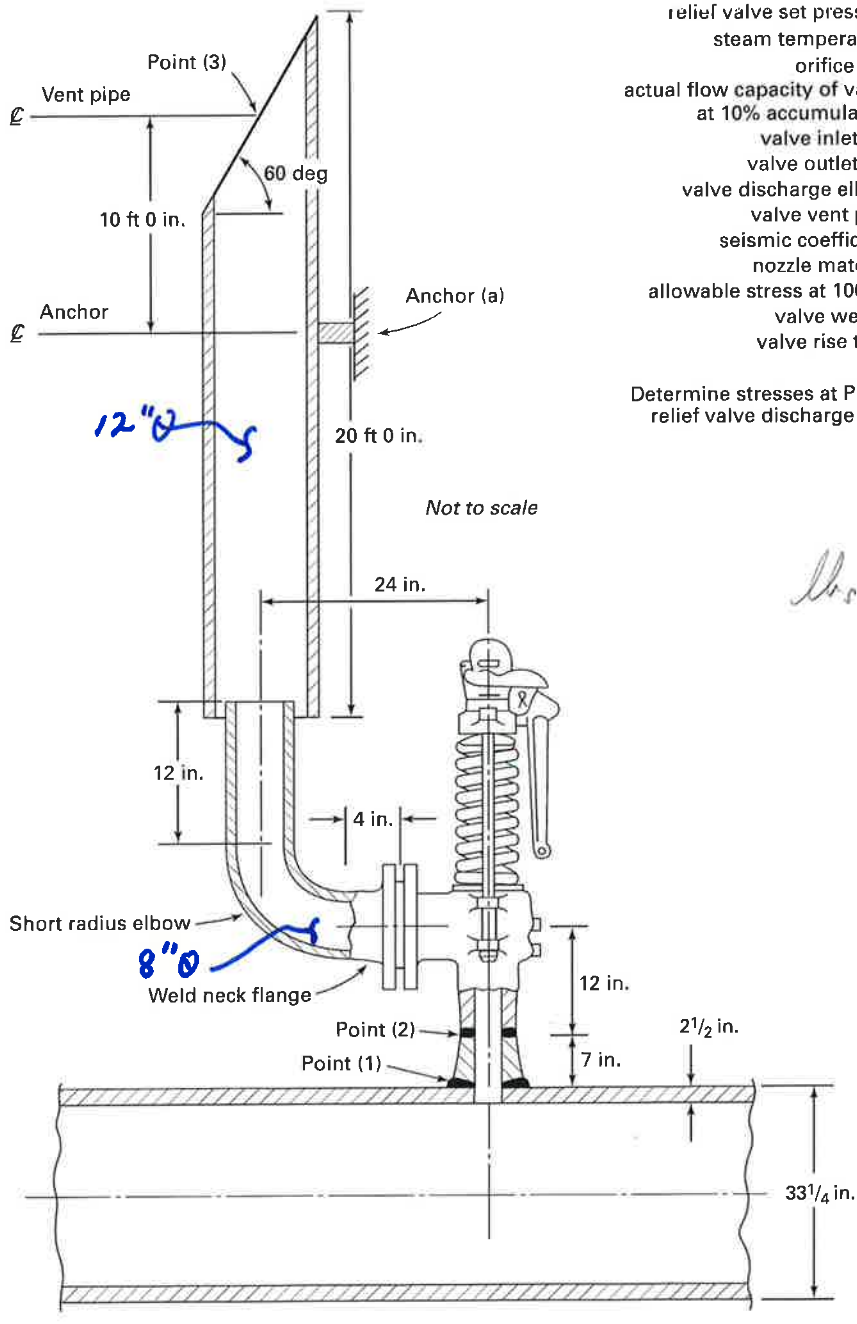
W = flow rate

= 116.38 lbm/sec

II-7.1.2 Discharge Elbow Maximum Operating Pressure. For 8 in. Class 150 ASME weld neck flange,

$$\frac{L}{D} = \frac{4 \text{ in.}}{7.981 \text{ in.}} = 0.5$$

Fig. II-7-1 Sample Problem Figure 1



relief valve set pressure = 910 psig
steam temperature = 1,000°F
orifice size = 11.05 in.^{2} (Q orifice)
actual flow capacity of valve
at 10% accumulation = 418,950 lbm/hr
valve inlet I.D. = 6 in.
valve outlet I.D. = 8 in.
valve discharge elbow = 8 in. SCH 40
valve vent pipe = 12 in. SCH 30
seismic coefficient = 1.5*g*
nozzle material = ASTM A 335 P22 2 1/4Cr–1Mo
allowable stress at 1000°F = 7,800 psi
valve weight = 800 lb
valve rise time = 0.040 sec

Determine stresses at Points (1) and (2) due to seismic and relief valve discharge loads only.

$lb_f = lb_m \times 32.2$

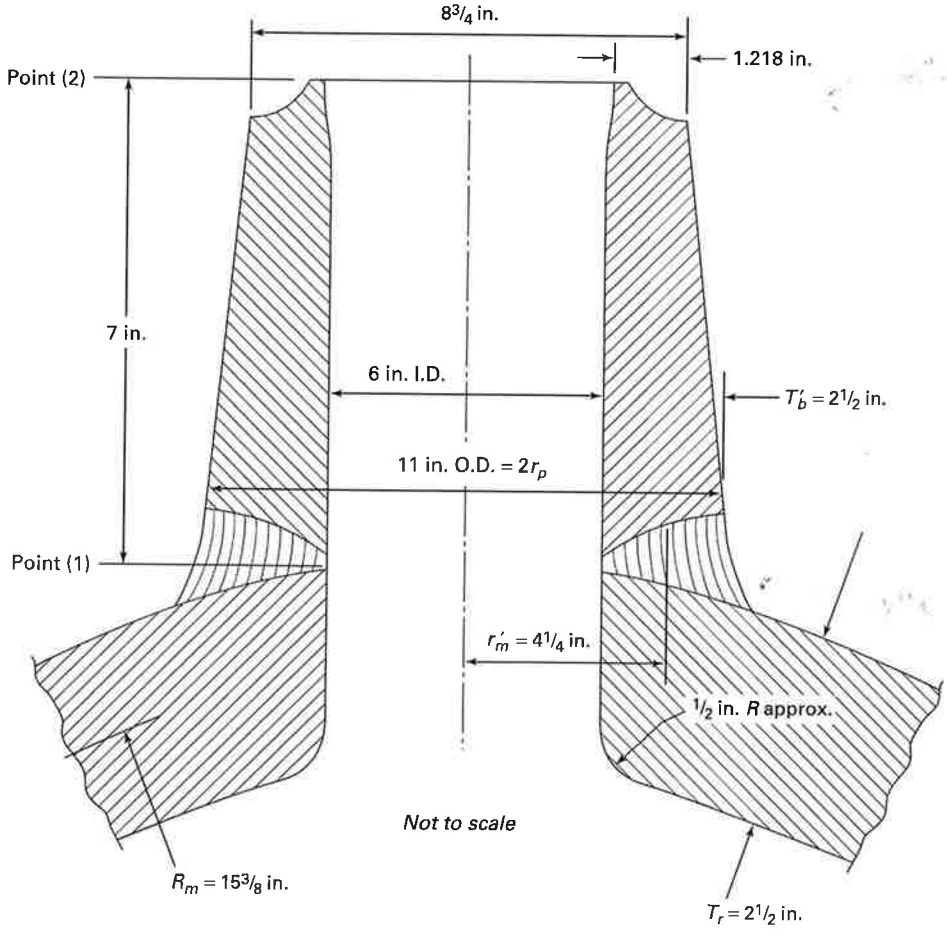
Fig. II-7-2 Sample Problem Figure 2

$$i = 1.5\left(\frac{R_m}{T_r}\right)^{2/3}\left(\frac{r'_m}{R_m}\right)^{1/2}\left(\frac{T'_b}{T_r}\right)\left(\frac{r'_m}{r_p}\right)$$

R_m, T_r, r'_m, T'_b, and r_p are shown in sketch below:

$$i_{(1)} = 1.5\left(\frac{15.375}{2.5}\right)^{2/3}\left(\frac{4.25}{15.375}\right)^{1/2}\left(\frac{2.5}{2.5}\right)\left(\frac{4.25}{5.5}\right)$$

$$i_{(1)} = 2.05$$



$L = \frac{\pi \times 16}{4 \times 8}$

For 8 in. SCH 40 short radius elbow,

$$\frac{L}{D} = 30$$

$\frac{L}{D} = 15 \times 5$

For 12 in. of 8 in. SCH 40 pipe,

$$\frac{L}{D} = \frac{12 \text{ in.}}{7.981 \text{ in.}} = 1.5$$

$$\Sigma\left(\frac{L}{D}\right) = \left(\frac{L_{max}}{D}\right) = 0.5 + 30 + 1.5 = 32.0$$

$$f = 0.013$$

$$k = 1.3$$

$$f\left(\frac{L_{max}}{D}\right) = 0.416$$

$\frac{96.56 \times 12}{10} = 115.87$

75

From Chart II-1, $P/P^* = 1.647$.

$$P_{1a} = P_1\,(P/P^*) = 194 \text{ psia}$$

II-7.1.3 Reaction Force at Discharge Elbow Exit. Reaction force,

$$F_1 = \frac{WV_1}{g_c} + (P_1 - P_a)\,A_1$$

where

W = 116.38 lbm/sec
V_1 = 2,116 ft/sec
g_c = 32.2 lbm-ft/lbf-sec^2
P_1 = 118 psia
P_a = 15 psia
A_1 = 50.03 in.^{2}
$(P_1 - P_a)$ = 118 − 15 = 103 psig
WV_1/g_c = 7,648 lbf
$(P_1 - P_a)\,A_1$ = 5,153 lbf
F_1 = 12,801 lbf

II-7.1.4 Bending Moments at Points (1) and (2)

(A) Bending Moment at Points (1) and (2) Due to Reaction at Point (1)

$M_{1(1)} = M_{1(2)}$
$= F_1 \times L \times DLF$
L = moment arm
= 24 in.
DLF = dynamic load factor

To determine DLF, first determine the safety valve installation period T:

$$T = 0.1846\sqrt{\frac{Wh^3}{EI}}$$

where

E = Young's modulus of inlet pipe at design temperature
$= 23 \times 10^6$ psi
h = distance from run pipe to centerline of outlet piping
= 19 in.
I = moment of inertia of inlet pipe
$= \frac{\pi}{64}\,(D_o^4 - D_i^4)$

Use average O.D. and I.D. to determine I. D_o = 9.875 in. avg.; D_i = 6 in. avg.

= 403.2 in.^{4}
T = 0.00449 sec
W = weight of valve
= 800 lb

For a valve rise time of 0.040 sec = t_o, the ratio t_o/T is 8.9. From Fig. II-3-2, DLF = 1.11.

Using F_1 = 12,801 lbf, L = 24 in., and DLF = 1.11,

$$M_{1(1)} = M_{1(2)} = 341{,}018 \text{ in.-lb}$$

(B) Bending Moments at Points (1) and (2) Due to Seismic Loading

Seismic force,

$$F_S = \text{mass} \times \text{acceleration}$$

$$= \left[\frac{800 \text{ lbm}}{32.2 \text{ lbm-ft/lbf-sec}^2}\right] \times 1.5(32.2 \text{ ft/sec}^2)$$

$$= 1{,}200 \text{ lbf}$$

Moment arm for Point (1) = 19 in.

$$M_{S(1)} = 1{,}200 \text{ lbf (19 in.)} = 22{,}800 \text{ in.-lb}$$

Moment arm for Point (2) = 12 in.

$$M_{S(2)} = 1{,}200 \text{ lbf (12 in.)} = 14{,}400 \text{ in.-lb}$$

(C) Combined Bending Moments at Points (1) and (2)

$$M_{(1)} = M_{1(1)} + M_{S(1)} = 363{,}819 \text{ in.-lb}$$

$$M_{(2)} = M_{1(2)} + M_{S(2)} = 355{,}419 \text{ in.-lb}$$

II-7.1.5 Stress Intensification Factors at Points (1) and (2)

(A) At Point (1), Branch Connection

$$i_{(1)} = 2.05$$

(B) Stress Intensification Factors at Point (2), Butt Weld

$$i_{(2)} = 1.0$$

II-7.1.6 Predicted Stresses at Points (1) and (2)

(A) Predicted Stresses at Point (1), Branch Connection

$$\text{Predicted stress} = \frac{PD_o}{4t_n}$$

$$\frac{D_o}{t_n} \text{ for run pipe} = \frac{33.25 \text{ in.}}{2.5 \text{ in.}} = 13.3$$

$$\frac{D_o}{t_n} \text{ for branch pipe} = \frac{11 \text{ in.}}{2.5 \text{ in.}} = 4.4$$

Use larger value with $P = 910$ psig.

$$\text{Pressure stress}_{(1)} = 3{,}030 \text{ psi}$$

$$\text{Flexure stress}_{(1)} = \frac{0.75i\, M_{(1)}}{Z_{(1)}}$$

$$Z_{(1)} = \pi r_b^2\, t_s$$

$$t_S = \text{lesser of } t_r \text{ or } (i)\, t_b$$

$$t_R = 2.5 \text{ in.}; (i)\, t_b = (2.05)\ 2.5 \text{ in.}$$

$$t_S = 2.5 \text{ in.}$$

$$r_b = 4.25 \text{ in.}$$

$$Z_{(1)} = 142 \text{ in.}^3$$

$$i_{(1)} = 2.05;\ M_{(1)} = 363{,}819 \text{ in.-lb}$$

$$\text{Flexure stress}_{(1)} = 3{,}939 \text{ psi}$$

$$\begin{aligned}\text{Combined stress}_{(1)} &= \text{pressure stress}_{(1)} \\ &\quad + \text{flexure stress}_{(1)} \\ &= 6{,}969 \text{ psi}\end{aligned}$$

(B) Predicted Stresses at Point (2), Butt Weld

$$\text{Pressure stress} = \frac{PD_o}{4t_n}$$

$$P = 910 \text{ psig}$$

$$D_o = 8.75 \text{ in.}$$

$$t_n = 1.218 \text{ in.}$$

$$\text{Pressure stress}_{(2)} = 1{,}635 \text{ psi}$$

$$\text{Flexure stress}_{(2)} = \frac{0.75\, i\, M_{(2)}}{Z_{(2)}}$$

$$Z_{(2)} = \frac{\pi}{32}\, \frac{D_o^4 - D_i^4}{D_o}$$

$$D_o = 8.75 \text{ in.}$$

$$D_i = 6 \text{ in.}$$

$$Z_{(2)} = 51.1 \text{ in.}^3$$

$$i_{(2)} = 1.0$$

$$M_{(2)} = 355{,}419 \text{ in.-lb}$$

$$\text{Flexure stress}_{(2)} = 6{,}955 \text{ psi}$$

(Note that $0.75i$ is set equal to 1.0 whenever $0.75i$ is less than 1.0, as in this case.)

$$\begin{aligned}\text{Combined stress}_{(2)} &= \text{pressure stress}_{(2)} \\ &\quad + \text{flexure stress}_{(2)} \\ &= 8{,}590 \text{ psi}\end{aligned}$$

(C) Comparison of Predicted Stress With Allowable Stress. Allowable stress of nozzle material at 1,000°F is

$$S_h = 7{,}800 \text{ psi}$$

$$k = 1.2$$

$$kS_h = 9{,}360 \text{ psi}$$

$$\text{Combined stress}_{(1)} = 6{,}969 \text{ psi}$$

$$\text{Combined stress}_{(2)} = 8{,}590 \text{ psi}$$

II-7.1.7 Calculate the Maximum Operating Pressure for Vent Pipe

$$P_3 = P_1 \left(\frac{A_1}{A_3}\right) = 118 \text{ psia} \left(\frac{50.03 \text{ in.}^2}{114.80 \text{ in.}^2}\right)$$

$$= 51.4 \text{ psia}$$

fL/D for 20 ft 0 in. of 12 in. SCH 30 pipe = 19.85.

$$\Sigma(L/D) = \left(\frac{L_{max}}{D}\right) = 19.85$$

$$f = 0.013$$

$$k = 1.3$$

$$f\left(\frac{L_{max}}{D}\right) = 0.258$$

From Chart II-1, $P/P^* = 1.506$.

$$P_2 = P_3\,(P/P^*) = 77.4 \text{ psia}$$

II-7.1.8 Check for Blowback From Vent Pipe. Calculate the velocity V_2 that exists at the inlet to the vent pipe (para. II-2.2.1.4).

$$f\left(\frac{L_{max}}{D}\right) = 0.258 \text{ from Step (7)}$$

$$V_3 = V_1 = 2{,}116 \text{ ft/sec}$$

From Chart II-1, $V/V^* = 0.7120$.

$$V_2 = V_3\,(V/V^*) = 1{,}507 \text{ ft/sec}$$

Check the inequality from para. II-2.3.1.2.

$$\frac{W(V_1 - V_2)}{g_c} > (P_2 - P_a)A_2 - (P_1 - P_a)A_1$$

$$\frac{116.38\,(2{,}116 - 1{,}507)}{32.2} > (77.4 - 14.7)(114.8) - (118 - 14.7)(50.03)$$

$$2{,}201 > 2{,}030$$

The inequality has been satisfied but the designer may require a design margin that would make 14 in. SCH 30 more acceptable. If a larger vent pipe is chosen, then the vent pipe analysis would have to be repeated for the 14 in. SCH 30 pipe.

II-7.1.9 Calculate Forces and Moments on Vent Pipe Anchor

$$F_2 = \frac{WV_2}{g_c} + (P_2 - P_a)A_2$$

$$= \frac{(116.38)(1{,}507)}{32.2} + (77.4 - 14.7)(114.8)$$

$$= 5{,}447 + 7{,}198.0 = 12{,}645 \text{ lbf}$$

$$F_3 = \frac{(116.38)(2{,}116)}{32.2} + (51.4 - 14.7)(114.8)$$

$$= 7{,}648 + 4{,}213 = 11{,}861 \text{ lbf}$$

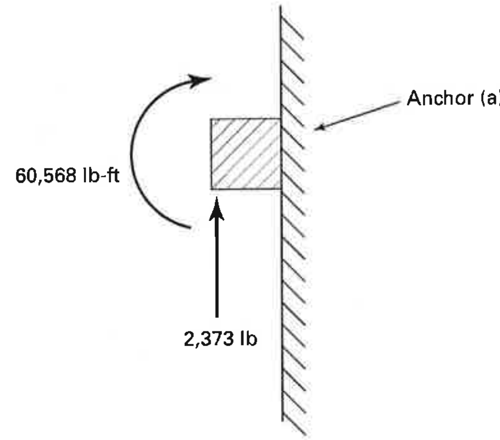
Assume a 30 deg jet deflection angle for vent pipe outlet. Vertical component of F_3

$$F_{3V} = F_3 \cos 30 \text{ deg} = 10{,}272 \text{ lbf}$$

Horizontal component of F_3

$$F_{3H} = F_3 \sin 30 \text{ deg} = 5{,}931 \text{ lbf}$$

Fig. II-7-3 Sample Problem Figure 3



Net imbalance on the vent pipe in the vertical direction is

$$F_2 - F_{3V} = 2{,}373 \text{ lbf}$$

Moment on vent pipe anchor

$$\Sigma M = (F_2 - F_{3V})\frac{D_o}{2} + F_{3H} \times [\text{distance from (a) to Point (3)}]$$

$$= (2{,}373)\left(\frac{1.06}{2}\right) + (5{,}931)(10.0)$$

$$= 60{,}568 \text{ ft-lb}$$

The vent pipe anchor would then be designed for the loads shown in Fig. II-7-3 for safety valve operation.

II-7.1.10 Conclusion. Branch connection stresses at Points (1) and (2) due to seismic and relief valve discharge are within 1.2 S_h. Blowback will not occur with the 12 in. standard weight vent pipe. The vent pipe anchor loads have been identified.

NONMANDATORY APPENDIX III
RULES FOR NONMETALLIC PIPING AND PIPING LINED WITH NONMETALS

FOREWORD

ASME B31.1 contains rules governing the design, fabrication, materials, erection, and examination of power piping systems. Experience in the application of nonmetallic materials for piping systems has shown that a number of considerations exist for the use of these materials that are not addressed in the current body of the Code. To address these, the requirements and recommendations for the use of nonmetallic piping (except in paras. 105.3, 108.4, 116, and 118) have been separately assembled in this nonmandatory Appendix.

III-1 SCOPE AND DEFINITION

III-1.1 General

III-1.1.1 This Appendix provides minimum requirements for the design, materials, fabrication, erection, testing, examination, and inspection of nonmetallic piping and metallic piping lined with nonmetals within the jurisdiction of the ASME B31.1 Power Piping Code. All references to the Code or to Code paragraphs in this Appendix are to the Section B31.1 Power Piping Code. In this Appendix, nonmetallic piping shall be limited to plastic and elastomer based piping materials, with or without fabric or fibrous material added for pressure reinforcement. Metallic piping lined with nonmetals shall be limited to factory-made plastic-lined ferrous metal pipe, fittings, and flanges produced to one of the product standards for plastic-lined piping materials listed in Table III-4.1.1.

III-1.1.2 Standards and specifications incorporated in this Appendix are listed in Table III-4.1.1. The effective date of these documents shall correspond to the date of this Appendix.

III-1.1.3 The provisions in Chapters I through VI and in Appendices A through F are requirements of this Appendix only when specifically referenced herein.

III-1.2 Scope

III-1.2.1 All applicable requirements of para. 100.1 and the limitations of para. 105.3 shall be met in addition to those in this Appendix.

III-1.2.2 Use of this Appendix is limited to

(A) water service.

(B) nonflammable and nontoxic liquid, dry material, and slurry systems.

(C) reinforced thermosetting resin pipe in buried flammable and combustible liquid service systems [refer to para. 122.7.3(F)].

(D) polyethylene pipe in buried flammable and combustible liquid and gas service. Refer to paras. 122.7.3(F) and 122.8.1(G).

(E) metallic piping lined with nonmetals. If used in accordance with para. 122.9 for conveying corrosive liquids and gases, the design of the lined piping system shall meet the requirements of para. 104.7.

III-1.2.3 Nonmetallic piping systems shall not be installed in a confined space where toxic gases could be produced and accumulate, either from combustion of the piping materials or from exposure to flame or elevated temperatures from fire.

III-1.3 Definitions and Abbreviations

III-1.3.1 Terms and definitions relating to plastic and other nonmetallic piping materials shall be in accordance with ASTM D 883. The following terms and definitions are in addition to those provided in the ASTM standard.

adhesive: a material designed to join two other component materials together by surface attachment (bonding).

adhesive joint: a bonded joint made using an adhesive on the surfaces to be joined.

bonder: one who performs a manual or semiautomatic bonding operation.

bonding operator: one who operates a machine or automatic bonding equipment.

bonding procedure: the detailed methods and practices involved in the production of a bonded joint.

Bonding Procedure Specification (BPS): the document that lists the parameters to be used in the construction of bonded joints in accordance with the requirements of this Code.

butt-and-wrapped joint: a joint made by applying plies of reinforcement saturated with resin to the surfaces to be joined.

chopped roving: a collection of noncontinuous glass strands gathered without mechanical twist. Each strand is made up of glass filaments bonded together with a finish or size for application by chopper gun.

chopped strand mat: a collection of randomly oriented glass fiber strands, chopped or swirled together with a binder in the form of a blanket.