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METODOLOGIA DE EVALUACIÓN DE LA CONDICIÓN OPERATIVA DE LAS TUBERÍAS DE VAPOR PRINCIPAL EN LA CENTRAL TERMOELÉCTRICA ILO 1

INFORME DE SUFICIENCIA

PARA OPTAR EL TITULO PROFESIONAL

INGENIERO MECÁNICO

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Dedicatoria

Este trabajo lo dedico a mis padres, quienes siempre me inculcaron el amor al trabajo y al esfuerzo para poder alcanzar mis metas. Papá y Mamá les agradezco por todo el amor, cariño y paciencia que me dieron.

Todas sus enseñanzas siempre estarán junto a mí.

METODOLOGÍA DE EVALUACIÓN DE LA CONDICIÓN OPERATIVA DE LAS TUBERÍAS DE VAPOR PRINCIPAL EN LA CENTRAL TERMOELÉCTRICA ILO 1

TABLA DE CONTENIDO

PRÓLOGO

Las tuberías críticas de los sistemas de potencia en una planta de generación eléctrica son aquellas que operan por encima de los 500°F (260°C). Estas incluyen las tuberías de vapor principal desde la descarga de los calderos hasta el ingreso a las turbinas. La metodología de evaluación de la condición operativa que se establece en este informe podrá aplicarse para los componentes de las tuberías de vapor principal (tuberías, soportes y restricciones).

En una central termoeléctrica con más de 40 años de operación es importante la aplicación de esta metodología para evaluar la condición operativa y vida remanente de las tuberías de alta potencia como factores determinantes de un programa de extensión de vida de la planta. Estas tuberías críticas pueden dañarse o degradarse, incluso durante condiciones normales de operación de la planta, resultando en pérdida de potencia, costosas reparaciones y vida de servicio reducida. Algunas de las causas de daños en la tubería o degradación incluyen creep, fatiga, soportes dañados, inadecuada selección o ubicación de soportes, operación del sistema cerca de la temperatura de diseño o inadecuado diseño para las condiciones de operación requeridas.

Para realizar esta evaluación de la condición se establece una metodología: que consta de un programa sistemático de inspección y evaluación para establecer la condición de las tuberías y los soportes de una planta de generación eléctrica que emplea combustible fósil.

Esta metodología consiste en: 1) Revisión de la información técnica requerida para la evaluación; 2) Inspección de las líneas de vapor principal; 3) Etapa de evaluación; 4) Selección de los puntos de inspección y métodos de examinación; y 5) Evaluación por disponibilidad para el servicio según API RP 579.

El presente informe consta de 5 capítulos los cuales son los siguientes:

En el primer capítulo se desarrolla la introducción en la cual se detalla cuál es el objetivo del informe, alcances, limitaciones y la importancia que tiene este tema para ser desarrollado.

En el segundo capítulo se desarrolla la descripción de la planta en donde se detalla todas las características de la misma.

En el tercer capítulo se desarrolla el marco teórico, sobre el cual se fundamenta el funcionamiento de una central térmica a vapor, se explican las normas ASME B31.1 y API RP 579. Finalmente, se concluye con una exposición de los ensayos no destructivos (NDE) recomendados para realizar este trabajo.

En el cuarto capítulo se desarrolla la metodología de evaluación la cual empieza con la revisión de información requerida para la evaluación, inspección requerida para el sistema de vapor principal, análisis de

esfuerzos en las tuberías de vapor principal según ASME B31.1, ensayos no destructivos requeridos, evaluación por disponibilidad para el servicio según API RP 579 y la elaboración del reporte final.
En el quinto capítulo se desarrolla una valorización de la inversión que requiere la aplicación de esta metodología de evaluación y su impacto sobre el gasto que representaría la parada repentina de la planta debido a una falla en la tubería del sistema de vapor principal.

La evaluación por disponibilidad para el servicio es realizada para asegurarse que los equipos de las plantas de proceso, tales como recipientes a presión, tuberías, y tanques puedan operan de forma segura y confiable por un período futuro deseado. La Práctica Recomendada API 579 describe un procedimiento general para la evaluación por disponibilidad para el servicio. El procedimiento evalúa el esfuerzo remanente del equipo en su actual condición, debido a que este esfuerzo puede ser menor que su condición original.

Los mecanismos comunes de degradación incluyen corrosión, pitting, fragilización, fatiga, fluencia a alta temperatura y distorsión mecánica. Los métodos para evaluar el esfuerzo y la vida remanente de servicio de los equipos que tienen estos tipos de degradación están presentados y revisados.

CAPITULO 1

INTRODUCCIÓN

1.1 OBJETIVO

El objetivo del presente informe de suficiencia es establecer una metodología de evaluación de la condición operativa de las tuberías de vapor principal en la Central Termoeléctrica Ilo 1, debido a que tiene 48 años de servicio y se requiere que pueda seguir operando 20 años más, para ello se evaluará su condición actual.

Las tuberías del sistema de potencia sirven para transportar el vapor que produce el caldero, considerado como el corazón de la planta, debido a que en este equipo es donde todos los sistemas confluyen para producir el vapor que impulsa la turbina y permite producir energía. La falla o inadecuado funcionamiento de alguno de estos sistemas podrá impactar el funcionamiento del caldero y se reflejará en los costos de operación, costos de mantenimiento, la disponibilidad y la capacidad de la planta.

Debido a ello es importante conocer la condición de las tuberías de vapor principal y sus soportes ya que son indicativos de la operación y el mantenimiento de la planta. La presencia de soportes rotos, tubería fuera de su posición normal de funcionamiento o aislamiento dañado, puede indicar que un evento imprevisto pudo haber ocurrido. El cual pudo adicionar cargas inesperadas para las cuales el sistema de tuberías no fue diseñado.

1.2 ALCANCE

Este trabajo se enfocará a realizar una metodología para evaluar la condición del sistema de vapor principal de la Central Termoeléctrica Ilo 1. Las siguientes tuberías de vapor principal están incluidas:

- Descarga del Caldero N°1 hacia el cabezal de vapor
- Descarga del Caldero N°2 hacia el cabezal de vapor
- Descarga del Caldero N°3 hacia el cabezal de vapor
- Descarga del Caldero N°4 hacia el cabezal de vapor
- Ingreso de vapor de los calderos de recuperación al cabezal de vapor.
- Cabezal de vapor
- Cabezal de vapor hacia la Turbina N°1
- Cabezal de vapor hacia la Turbina N°2
- Cabezal de vapor hacia la Turbina N°3
- Cabezal de vapor hacia la Turbina N°4

1.3 LIMITACIONES

Este informe abarcará solo la elaboración del procedimiento requerido para realizar esta evaluación, debido a que esta metodología requiere la decisión de la gerencia de una planta termoeléctrica para ser aplicada.

1.4 IMPORTANCIA DEL TEMA

Este tema es importante ya que se propone una metodología para realizar la evaluación de la condición de las tuberías de vapor de una central termoeléctrica con más de 40 años de operación continua, es por ello que este informe puede sentar las bases para el desarrollo de este tema en nuestro país.

Por muchos años, los programas de mantenimiento en las centrales eléctricas han sido orientados a componentes con una expectativa de vida finita donde la degradación y la falla están asociadas con la fatiga y la fluencia. Componentes tales como los cabezales de transporte de vapor, tuberías de vapor principal y sobrecalentado, y turbinas a vapor están sujetos a una falla eventual del material debido a que operan a altas temperaturas y esfuerzos. Como una norma general, los calderos industriales así como sus redes de tuberías de vapor típicamente operan a temperaturas y presiones mucho más bajas. Como resultado, la vida de estos equipos no está definida necesariamente por la vida finita del material. De hecho hay varios ejemplos de calderos y tuberías de vapor muy antiguos (más de 50 años de operación) que todavía se encuentran operando y muchas veces son retirados por otras razones que no son confiabilidad ni seguridad.

Hoy la necesidad de establecer un programa para evaluar la condición de tuberías de vapor en plantas industriales está empezando a ser importante debido a que se buscan la reducción de los costos de operación y la optimización de la producción de la capacidad instalada.

Es por ello que partir de 1980 se han incrementado los programas para extensión de la vida útil de las plantas termoeléctricas. Con este método para la evaluación de la condición se podrá determinar la vida útil remanente de las tuberías de vapor.

Esta evaluación es un importante elemento para un programa de extensión de la vida útil de una planta termoeléctrica.

CAPITULO 2

DESCRIPCIÓN DE LA PLANTA

2.1 UBICACIÓN

La Central Termoeléctrica Ilo 1 se encuentra situada a 18 km al norte del puerto de Ilo, dentro del área de la Fundición de Southern Perú Copper Corporation (SPCC).

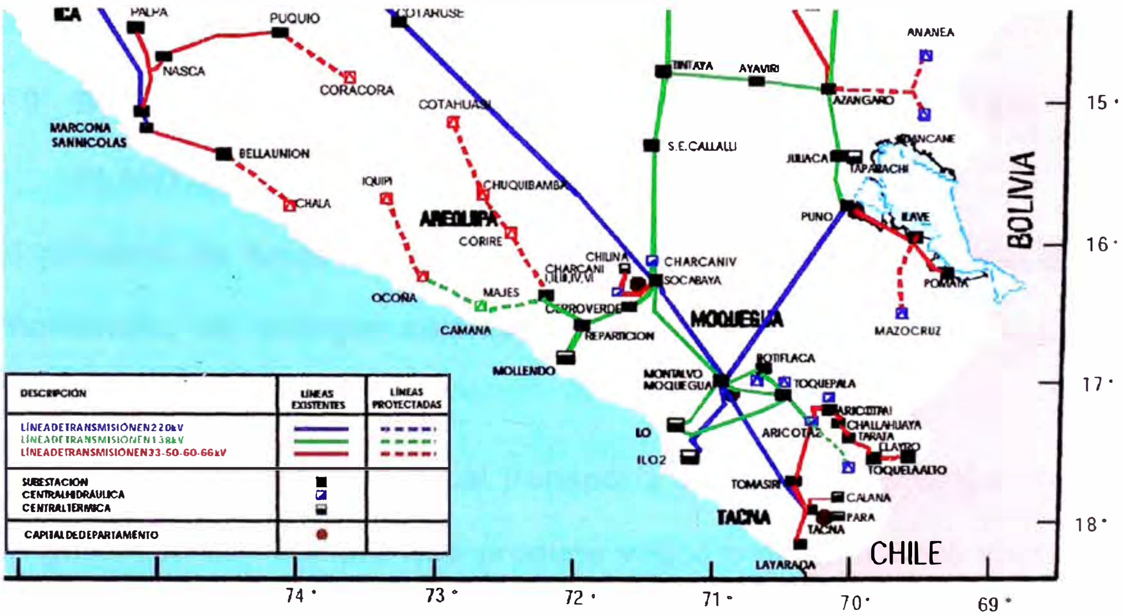


Figura N°2.1

2.2 CARACTERÍSTICAS TECNICAS DE LA PLANTA

El sistema de generación a vapor de la planta cuenta con 04 calderos acuotubulares, una tubería de descarga de vapor de los calderos de recuperación y 04 turbinas a vapor, los cuales tienen las siguientes características:

Equipo	Años de operación	Fabricante	Nominal Klb/h	Efectivo Klb/h
Caldero N°1	48	B & W	215	205
Caldero N°2	48	B & W	215	205
Caldero N°3	36	C-E	300	280
Caldero N°4	12	ABB	400	380
WHB			240	240
		Total:	**1370**	**1310**

Generadores	Años de operación	Fabricante	Nominal MW	Efectivo MW
Turbina - Generador N°1	48	BBC	22	22
Turbina - Generador N°2	48	BBC	22	22
Turbina - Generador N°3	28	GE	66	66
Turbina - Generador N°4	32	GE	66	66
		Total Vapor	**176**	**135***

*Máxima generación debido a la capacidad de los calderos (1310/9.7=135)

Tabla N°2.1

2.3 BREVE DESCRIPCIÓN DEL FUNCIONAMIENTO DE LA PLANTA

El principio de funcionamiento de la Central Térmica se basa en el intercambio de energía calórica en energía mecánica y luego en energía eléctrica.

Las tuberías de vapor principal transporta vapor a alta presión el cual es generado en caldera que produce vapor a presión, este vapor se aplica sobre los álabes de la turbina que convierte energía potencial (presión) en energía cinética que acciona al generador.

La energía potencial contenida en el combustible que emplea el caldero es denominada poder calorífico y se mide en BTU (British Thermal Units) por libra de peso. Estos valores se expresan comúnmente como BTU/lb. En el sistema métrico decimal se expresa en Calorías o

Kilo-Calorias/Kilogramo de peso: Kcal/Kg. La tabla que sigue muestra los valores típicos del poder calorífico contenido de los combustibles empleados:

Combustible	Gas Natural	Petróleo Diesel N°2	Petróleo Residual N°6	Carbón Antracita
Poder Calorífico BTU/lb	21.830	18.993	18.126	12.680
Equivalencia	1000 BTU/pie^3	137.000 BTU/gal	153.000 BTU/gal	---

Tabla N°2.2

Conociendo como se mide la energía tanto en el combustible como en el vapor y en el agua, se analizará un arreglo simplificado del ciclo de vapor en la planta de generación y también como cambia la energía conforme el agua y el vapor circulan a través de sus componentes.

Como ejemplo se empezará con 100,000 libras por hora de agua de alimentación entrando al tambor superior del caldero. La presión del agua es de 1000 psia y su temperatura de 360°F. Se debe tener en cuenta que la presión en el tambor superior es de 890 psia, por lo que la presión del agua debe ser necesariamente mayor para poder ingresar. Al absorber calor de los gases de combustión, el agua circula desde el tambor superior al tambor inferior por los tubos más alejados del calor de los quemadores y regresa al tambor superior por los tubos más cercanos llevando burbujas de vapor que suben a la superficie llenando la mitad superior del tambor con vapor saturado.

El vapor saturado sale del tambor superior a razón de 100,000 libras/hora e ingresa a los serpentines del sobrecalentador de donde sale a 875 psia y 900°F de temperatura. Cabe señalar que la presión disminuye en 15 psia, desde 890 psia hasta 875 psia, al pasar por los tubos del sobrecalentador. El vapor sobrecalentado, conducido por tuberías, llega a la Turbina pasando primero por las válvulas de control de velocidad. En la Turbina, el vapor fluye alternadamente a través de toberas estacionarias y una rueda de paletas, ambas constituyen una etapa. Conforme el vapor fluye en sucesión por toberas y paletas, va entregando energía a la turbina y tanto su presión como su temperatura van disminuyendo progresivamente. La turbina convierte la energía térmica del vapor en trabajo mecánico de rotación a 3,600 rpm que se transmite al generador para producir 10,000 kW de energía eléctrica.

En algunos puntos de su recorrido dentro de la turbina, hay conexiones que permiten "extraer" parte del vapor para calentar el agua de alimentación que retorna hacia el o los calderos.

El vapor de las extracciones fluye hacia los calentadores, donde cede su calor al agua que circula dentro de los tubos, y se condensa. El condensado del calentador de alta presión pasa por una trampa y se descarga en el calentador de baja presión. La trampa es como una válvula automática que permite pasar agua pero no permite pasar vapor.

El condensado recibe el vapor de escape de la turbina, lo condensa y lo colecta en el “pozo de condensado” o “Hotwell” junto con todo el condensado de los calentadores de agua. De esta manera, las 100,000 libras de vapor a 900°F que entraron a la turbina terminan como 100,000 libras de agua a 79°F en el pozo del condensador. De allí, las bombas de condensado y las bombas de alimentación impulsarán el condensado a través de los calentadores donde se calienta hasta 360°F y retorna nuevamente al caldero para iniciar un nuevo ciclo.

2.3.1 El Ciclo de Condensado en las Unidades Brown Boveri

2.3.1.1 Bombas de Condensado

Hay dos bombas verticales que reciben condensado del pozo Hotwell del condensador. Cada bomba tiene su propia línea de succión desde el Hotwell y capacidad para bombear 440 galones por minuto (gpm) a una presión de 300 psig. En operación normal, una bomba trabaja y la otra permanece en reserva o “stand-by”. Ambas se arrancan manualmente desde la sala de control. La bomba en stand-by arranca mediante un switch accionado por baja presión de condensado en la línea de descarga de las bombas. La temperatura promedio del condensado es de 90°F y en su recorrido desde la descarga de las bombas del pozo Hotwell hasta el domo del caldero, empezará a recoger calor del ciclo a través de los enfriadores de hidrógeno y los intercambiadores de calor.

2.3.1.2 Enfriadores de Hidrogeno

En la C.T. ILO 1, todos los generadores eléctricos son enfriados por hidrógeno el cual circula por su interior impulsado por dos ventiladores acoplados al eje mismo del generador. El hidrógeno a su vez es enfriado por el condensado que viene directamente de la descarga de las bombas Hotwell a una temperatura promedio de 91°F. Con este fin, los generadores tienen en su interior enfriadores tubulares en los cuales se transfiere calor del hidrógeno que circula por el exterior de los tubos, hacia el condensado que circula por el interior. De aquí el condensado continuará su recorrido dirigiéndose hacia los enfriadores de aceite.

2.3.1.3 Enfriadores de Aceite

Todo el aceite que utiliza la turbina pasa por enfriadores que son intercambiadores de calor del tipo de cascos y tubos, en los cuales también se utiliza el condensado como medio refrigerante. El condensado fluye por el interior de los tubos mientras que el aceite circula por el exterior. Cumplida su misión, el condensado continúa su recorrido, esta vez a través de los eyectores de aire.

2.3.1.4 Eyectores de Chorro a Vapor

Los eyectores son mecanismos diseñados para succionar aire y gases no condensables del interior del condensador principal. Utilizan vapor vivo, cuya presión se reduce desde 850 psig, mediante una válvula tipo aguja regulada por el operador, hasta 350 psig que es la presión normal de operación del eyector. El principio de funcionamiento de los eyectores se basa precisamente en convertir la presión del vapor en un chorro o "jet" de alta velocidad que a su paso por toberas especiales genera el efecto de succión o vacío necesario para evacuar aire y gases del condensador.

Aquí también el condensado del ciclo servirá como refrigerante para condensar el vapor que ya ha cumplido su función en los eyectores. Para este fin, el sistema de eyectores cuenta con cámaras de condensación en las cuales se consigue dos objetivos:

El vapor que ya ha trabajado se enfría y es recuperado como agua que se devuelve al sistema en el condensador.
El condensado principal actuando como refrigerante, absorbe calor lo cual mejora la eficiencia térmica del ciclo.

2.3.1.5 Calentadores de Baja Presión

El condensado que sale de los eyectores, pasa primero por el calentador N°1 y luego por el calentador N°2 de baja presión. Estos calentadores son también intercambiadores de calor de casco y tubos de 4 pasos en forma de U: El condensado fluye por el interior de los tubos y por el exterior vapor que se “extrae” de la turbina a través de conexiones en el extremo de baja presión. Por eso decimos que el calentador N°1 recibe vapor de la extracción 4 y el calentador N°2 recibe vapor de la extracción 3.

Mientras que el vapor se enfría y se convierte en agua que retorna al condensador, el condensado recibe calor del ciclo y aumenta su temperatura cada vez más.

En las líneas de vapor de extracción hacia los calentadores, hay dos clases de válvulas:

a) De bloqueo, que sirven para aislar el calentador en caso sea necesario sacarlo de servicio.
b) De no retorno, que impiden el flujo invertido de agua y/o vapor hacia la Turbina.

En la línea de condensado también hay válvulas de bloqueo a la entrada y salida. También una válvula de by-pass para sacar de servicio el calentador.

Se debe tener en cuenta que los calentadores de baja presión tienen líneas de “vent” por las cuales se descarga aire y gases no condensables de la cámara del calentador directamente al condensador. Esto es muy importante porque el aire y los gases pueden aislar las superficies de transferencia de calor y disminuir la eficiencia del ciclo.

En cada calentador el vapor de extracción se enfría y se convierte en agua condensada. En el caso del calentador N°2, que tiene más presión, el agua condensada fluye en cascada hacia el calentador N°1, si este calentador estuviera fuera de servicio, hay un by-pass que permite descargar este condensado directamente al condensador principal.

El calentador N°1 recibe, por un lado, vapor de extracción que se enfría y condensa, por otro lado el agua condensada del calentador N°2. Todo este condensado es descargado directamente al condensador principal.

2.3.1.6 Evaporador

Es un pequeño aparato “destilador” diseñado para producir agua de alto grado de pureza la cual es necesaria para reponer las pérdidas de líquido que inevitablemente se presentan como en todo proceso.

El evaporador es también un intercambiador de calor de casco y tubos: Recibe agua de baja calidad que fluye por el exterior de los tubos y, como fuente de calor, vapor de la extracción N°2 de la turbina que circula por el interior de los tubos. El agua recibirá calor y aumenta su temperatura hasta el punto de ebullición. Al hervir el agua se producirá vapor puro mientras que las impurezas quedan dentro del evaporador junto con la parte líquida que no ha evaporado.

El vapor de extracción que se usa como fuente de calor se enfría y condensa, luego es drenado continuamente a través de una trampa hacia el calentador N°2.
El vapor puro que se obtiene como producto de la evaporación del agua, pasa a un condensador denominado "condensador del evaporador".

2.3.1.7 Condensador del Evaporador

El condensado principal que sale del calentador de baja presión N°2, ingresa ahora como refrigerante por el interior de los tubos del condensador del evaporador. Por el exterior de los tubos fluye el vapor producido en el evaporador, el cual cede su calor al condensado principal y se enfría convirtiéndose en el agua líquida de gran pureza, que se requiere para reponer las pérdidas del ciclo. Este nuevo condensado recién obtenido pasa por diferencia de presión al calentador N°2 y si este no estuviera en servicio, pasa directamente al calentador N°1 o al condensador principal.

2.3.1.8 Calentadores de Alta Presión

El condensado principal sale del condensador del evaporador e ingresa en serie a los calentadores N°3 y N°4 de alta presión. Estos calentadores reciben esta denominación porque el vapor que reciben proviene de las extracciones de mayor presión de la Turbina. En estos calentadores se repite el mismo proceso que se ha definido para los calentadores de baja presión, solo que las condiciones de presión y temperatura son más altas.

El agua condensada del calentador N°4 fluye por diferencia de presión hacia el N°3 y si este no estuviera en servicio, al calentador N°2 o directamente al condensador.

El agua condensada del calentador N°3 fluye por diferencia de presión hacia el N°2 o hacia el N°1 o directamente hacia el condensador.

Los “vents” de estos calentadores descargan libremente a la atmósfera.

El condensado principal también tiene válvulas a la entrada y salida de cada calentador y en cada caso un by-pass que permite mantener el flujo de condensado aún cuando cualquiera de los calentadores estuviera fuera de servicio.

Al salir del último calentador, el condensado principal que tenía 91°F de temperatura al comienzo del ciclo, habrá adquirido el máximo de temperatura, unos 360°F, y será succionado por las bombas de alimentación. A partir de este punto, ya no se hablará de condensado sino únicamente de agua de alimentación.

2.3.1.9 Bombas de Agua de Alimentación

El condensado que sale del calentador N°4 de alta presión, fluye directamente a la línea de succión de las bombas de alimentación. Hay en este sistema tres bombas idénticas:

- La bomba N°1 dedicada exclusivamente al ciclo de la Turbina N°1
- La bomba N°2 dedicada exclusivamente al ciclo de la Turbina N°2
- La bomba N°3 instalada de tal modo que puede reemplazar a una cualquiera de las anteriores, requiriendo naturalmente direccional las válvulas de bloqueo tanto en la succión como en la descarga.

Cada bomba tiene seis etapas, capacidad de 500 gpm a 1200 psig. Tienen una válvula automática de recirculación con el fin de mantener un flujo mínimo de agua a través de la bomba para condiciones de baja demanda o baja "carga" en el grupo.

Asimismo hay una conexión que permite un pequeño flujo de agua caliente a través de la bomba que se encuentre de reserva o en "stand-by" con el fin de mantener sus componentes a una temperatura adecuada que le permita arrancar en cualquier momento.

2.3.1.10 Válvula de Distribución

Es uno de los componentes más importantes del circuito de control. Como se sabe la demanda de energía eléctrica es una variable que no se puede controlar porque depende únicamente de los usuarios de la red. La turbina por otro lado debe responder a las variaciones de la demanda mediante la modulación de sus válvulas de admisión que abren o cierran para admitir más o menos vapor en producción a la demanda eléctrica.

De aquí resulta la importancia de la válvula de distribución para controlar la cantidad de agua que impulsan las bombas de alimentación y mantener el balance de fluidos en el ciclo. Es decir, si en un momento dado la turbina recibe un flujo de vapor de 200,000 libras/hora, la válvula de distribución abrirá lo suficiente para descargar agua de alimentación a un régimen de

200,000 libras/hora del ciclo. Si luego aumenta o disminuye el flujo de vapor a la turbina, la válvula de distribución abrirá o cerrará para mantener siempre la relación agua-vapor constante.

Para que esta válvula pueda operar correctamente debe recibir señales de control equivalentes tanto al flujo de vapor entrando a la turbina como del agua de alimentación que sale por las bombas. Existe sin embargo un tercer elemento de control que es tanto o más importante: es la presión en el cabezal de distribución al cual están conectadas todas las bombas que hay en la planta y desde el cual se distribuye agua de alimentación a todos los calderos. Cada válvula de distribución en su respectivo ciclo, debe mantener la presión de agua de alimentación lo más estable posible a 1200 psig a fin de asegurar una buena alimentación a todos los calderos.

2.3.1.11 Caldero

El agua de alimentación fluye del cabezal de distribución a 360°F de temperatura y 1200 psig hacia la válvula de control de cada caldero y de allí al tambor o “domo” superior de donde se distribuye por todos los tubos de generación. Allí en los tubos es donde el agua recibe el calor de la combustión del petróleo y sube su temperatura hasta el punto de saturación correspondiente a la presión y se transforma en vapor saturado que se acumula en la mitad superior del domo.

El vapor saturado sale del domo superior hacia los serpentines de sobrecalentamiento dentro del mismo caldero. Allí recibe aún más calor de los gases de combustión hasta alcanzar la temperatura final con que sale hacia las Turbinas.

Los controles más importantes en la operación de cada caldero son los siguientes:

1. El control de nivel de agua en el domo superior: El domo superior es un cilindro de acero de 60 pulgadas de diámetro y 30 pies de largo. Es condición indispensable de operación que el nivel de agua se mantenga siempre a la mitad. Esta es la función primordial de la válvula de control de nivel, que como la válvula de distribución, tiene también un sistema de tres elementos: El flujo de vapor que sale del caldero debe ser igual al flujo de agua que ingresa a fin de mantener el balance de fluidos. El tercer elemento, el más importante, viene a ser el nivel propiamente dicho.

2. El control de temperatura del vapor sobrecalentado a 900°F: Esto se realiza mediante una válvula de control y un atemperador, mediante los cuales se inyecta agua de la línea

de alimentación finamente pulverizada, sobre el vapor sobrecalentado.

3. El control de presión de vapor: Este es un control más complejo por cuanto implica el control de todos los calderos en respuesta a las variaciones de la demanda eléctrica de la red. Hay un control principal o "Master" que recibe información de estas variaciones y envía señales a cada caldero para que su sistema de control aumente o disminuya el régimen de fuego a fin de aumentar o disminuir el régimen de generación de vapor de tal modo que la presión general del vapor en el cabezal principal que alimenta a las turbinas se mantenga estable bajo cualquier condición de la demanda, esta presión es 860 psig.

4. El control de combustión: Al recibir la señal del Master principal debe ajustar el flujo de aire y petróleo hacia los quemadores del caldero a fin de asegurar una respuesta inmediata y segura en cuanto al régimen de generación de vapor.

2.3.1.12 Turbina

Las turbinas Brown Boveri tienen una capacidad máxima de generación de 22,000 kW. Son turbinas de condensación del tipo horizontal, de un solo cilindro, y 4 extracciones de vapor no controladas. El vapor sobrecalentado de los calderos, ingresa a través de dos válvulas principales de parada de emergencia o válvulas STOP que constituyen la principal barrera de protección de la turbina. Cualquier condición de riesgo para la máquina, origina el cierre instantáneo de estas válvulas.

Luego el vapor pasa a las válvulas de admisión o válvulas de control que abren en forma secuencial y regulan e ingreso de vapor a fin de mantener constante la velocidad de rotación de la máquina en 3,600 RPM. Inmediatamente después de las válvulas de admisión, el vapor llega a la placa de toberas que dirige el flujo de vapor en el ángulo adecuado sobre la primera rueda de impulsión de la máquina. A continuación el vapor fluirá hacia las ruedas de paletas llamadas de reacción y distribuidas a lo largo de todo el cilindro hasta llegar finalmente al escape de donde fluye al condensador principal que se encuentra directamente debajo de la turbina y anclado al piso.

2.3.1.13 Condensador

Es también un intercambiador de calor que recibe todo el vapor de escape de la turbina, lo condensa, lo libera de aire y gases no condensables y lo acumula en el pozo o Hotwell de donde lo succionarán las bombas de condensado para iniciar nuevamente el ciclo. El fluido refrigerante para la condensación del vapor es el agua de mar que circula por un haz de tubos distribuidos en dos pases secuenciales.

Sus principales funciones son las siguientes:

- Disminuir la presión en el escape de la turbina para incrementar la eficiencia térmica.
- Condensar el vapor para retornarlo al caldero por el sistema de alimentación.
- Remover el exceso de oxigeno del condensado.

2.3.2 El Ciclo de Condensado en las unidades General Electric

El ciclo de condensado de las unidades G.E. es similar al de las Brown Boveri, con algunas diferencias que se derivan de la mayor capacidad de las turbinas, los principales equipos son los siguientes:

2.3.2.1 Extracciones de Vapor y Calentadores de Agua de Alimentación

El ciclo G.E. tiene 5 extracciones: dos de baja presión para los calentadores 1 y 2, una extracción de presión intermedia para el desaereador y dos extracciones de alta presión para los calentadores 4 y 5.

2.3.2.2 *Drenaje de Condensado de los Calentadores de Alta Presión*

El control de nivel del calentador 5 tiene dos válvulas que operan en secuencia: una descarga el condensado hacia el calentador y cuando esta no se abastece para mantener el nivel, la otra abre para descargar condensado directamente al condensador. Igualmente el control de nivel del calentador 4 tiene dos válvulas que operan en secuencia: una descarga condensado al desaereador y la otra hacia el calentador de baja presión N°2.

En condiciones normales de operación, el condensado producto de las extracciones de alta presión irá al desaereador y de allí a través de las bombas de alimentación hacia los calderos. De esta manera se mejora la eficiencia térmica del ciclo porque se aprovecha todo el calor contenido en los drenajes de condensado de alta.

2.3.2.3 *Drenaje de Condensado de los Calentadores de Baja Presión*

El control de nivel del calentador 2 descarga el condensado a través de dos válvulas: una hacia el calentador 1 y la otra directamente al condensador.

El calentador 1 tiene un tanque que recibe todos los drenajes de condensado provenientes de las extracciones de baja presión y eventualmente de las extracciones de alta. Este tanque recibe el nombre de tanque de drenaje y cuenta con una bomba centrífuga llamada también bomba de drenaje que succiona el condensado y lo descarga a través de una válvula de control de nivel hacia la línea principal de condensado que va al desaereador.

De esta manera se obtiene una mejor eficiencia térmica del ciclo ya que se recupera todo el calor que contienen los drenajes de los calentadores. Cabe indicar que en las unidades Brown Boveri, este condensado se descarga al condensador donde se pierde calor. Solamente si la bomba de drenaje no estuviera en condiciones de operar, hay otra válvula que permite dirigir el condensado hacia el condensador principal.

2.3.2.4 Enfriadores de Hidrogeno y Enfriadores de Aceite

A diferencia de las unidades Brown Boveri, los enfriadores de hidrógeno del generador y de aceite de la turbina no utilizan condensado sino agua tratada con bicromato de potasio como inhibidor de corrosión del circuito de enfriamiento general de equipos (bearing cooling water).

2.3.2.5 Bombas de Condensado

La única diferencia es que estas tienen sellos mecánicos en vez de empaquetaduras en el eje, por lo tanto necesitan un suministro continuo de agua limpia (condensado) para enfriamiento de estos sellos.

2.3.2.6 Desaereador

Sirve para eliminar el aire y los gases corrosivos del agua de alimentación y, al mismo tiempo, para precalentar el agua antes de enviarla al caldero. Eliminando los gases corrosivos, principalmente el oxigeno y el CO_2, se protege el caldero y sus equipos auxiliares contra la corrosión. Además cumple las siguientes funciones:

1. Mantiene una reserva de agua caliente y desaireada para casos de cambios súbitos en la demanda de vapor. La cantidad de agua que se alimenta al caldero, es igual a la cantidad de vapor que se produce, por tanto, la reserva de agua es proporcional a la capacidad del caldero. Un criterio recomendable para determinar la reserva de agua y la capacidad del tanque es: Almacenar una cantidad suficiente para sostener la evaporación del caldero durante 20 minutos.

2. Eliminando el aire, el vapor mantiene su temperatura y la máxima eficiencia en la transferencia de calor hacia los procesos industriales en los cuales se utiliza.

3. Se reduce el costo de aditivos químicos necesarios para neutralizar el oxígeno y CO_2 del agua de alimentación. Los aditivos químicos, por otro lado, aumentan el contenido de sólidos disueltos que contiene el agua, los cuales se eliminan mediante "la purga". Reduciendo la adición de químicos, se reduce la cantidad y frecuencia de la purga del caldero, la cual siempre será un mal necesario, porque ayuda a controlar los sólidos incrustantes pero significa también pérdidas de agua, de calor y de aditivos.

4. Proporciona un lugar adecuado donde retornar el condensado de los diferentes sistemas o procesos que utilizan vapor generalmente a diferentes niveles de presión.

5. Reduce la corrosión y costos de mantenimiento, reparación o limpieza en tuberías, válvulas, bombas, trampas de vapor, etc.

6. Permite recuperar calor que normalmente se pierde, de los tanques de evaporación "flash", escape de turbinas auxiliares, vents, trampas, etc., ayudando a calentar del agua de alimentación con este calor. Es importante notar que un desaereador ahorra 1% del costo del combustible por cada 10°F que aumenta la temperatura del agua de alimentación.

7. Calentando el agua, el desaereador reduce el mantenimiento del caldero, porque al acercar las temperaturas del agua de alimentación con la de operación normal del caldero, se reducen esfuerzos indebidos producidos por el "shock" térmico cuando se alimenta agua fría al caldero caliente.

2.3.2.7 *Control de Nivel del Desaereador*

En el ciclo Brown Boveri todo el vapor que ingresa a la turbina termina en forma de condensado en el Hotwell del Condensador principal. En el ciclo G.E. en cambio todo el vapor que ingresa a la turbina, termina en el desaereador. De allí es captado por las bombas de alimentación que lo envían a través de la válvula de distribución hacia los calderos.

El control de nivel del desaereador cuenta con los siguientes componentes:

1. Una válvula de control de flujo que además de controlar el nivel, mantiene el balance entre el flujo de condensado que ingresa y el agua de alimentación que sale y cuenta con tres elementos de control:
   a) El flujo de agua de alimentación que sale del desaereador hacia las bombas.
   b) El flujo de condensado principal que entra al desaereador.
   c) El nivel del desaereador propiamente dicho.

2. Una válvula de control de muy alto nivel, que descarga el condensado hacia el tanque de almacenamiento que está debajo del desaereador

3. Las bombas de transferencia de condensado que cuentan con un circuito automático de control de muy bajo nivel. En este caso de nivel muy bajo, las bombas succionan condensado del tanque y lo envían al desaereador.

Las bombas de transferencia en control manual también suministran agua para el sello mecánico de las bombas de condensado durante el arranque inicial de la planta. Igualmente en control manual sirven para llenar el condensador para pruebas hidrostáticas.

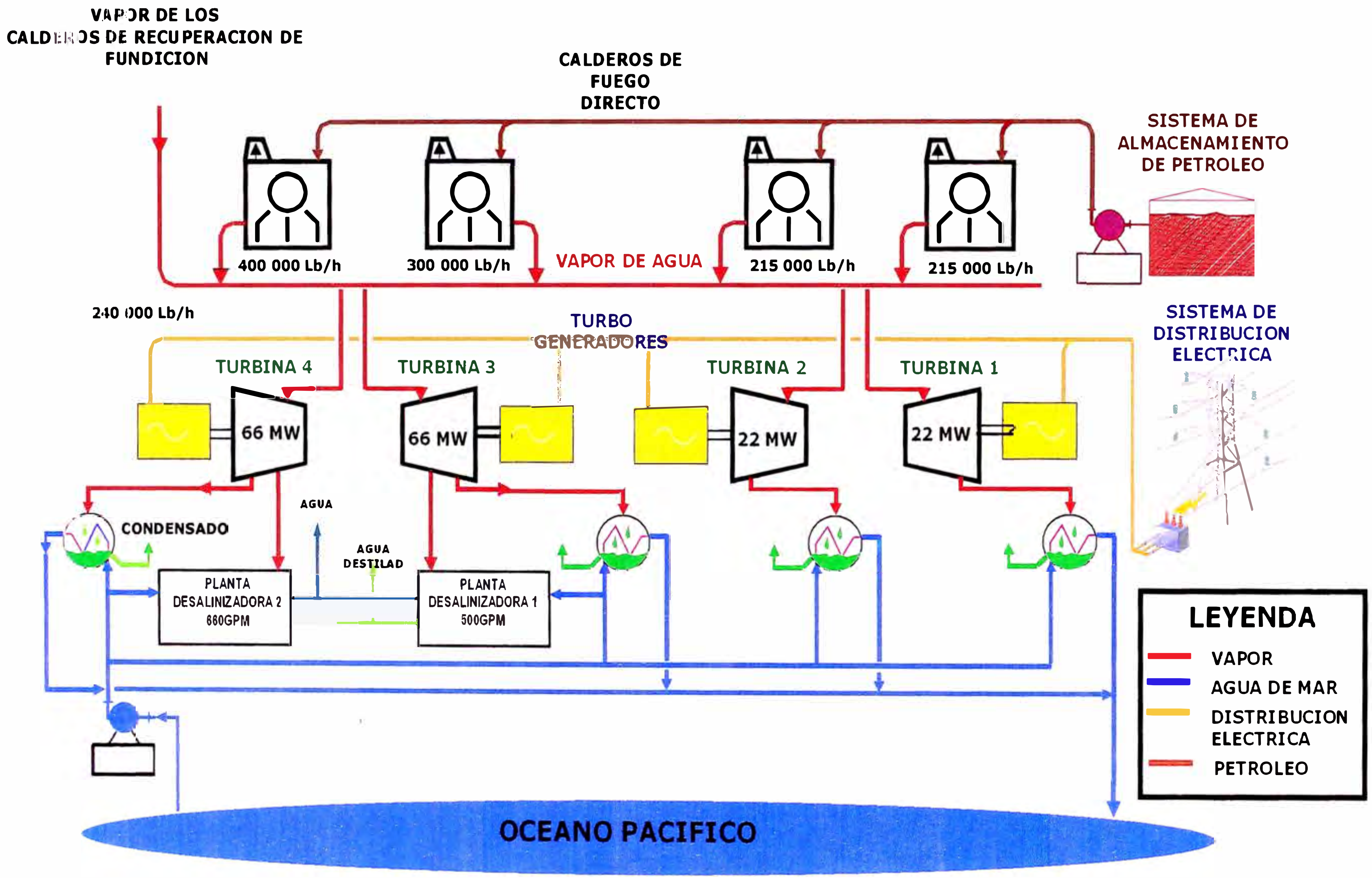

VAPOR DE LOS
CALDEROS DE RECUPERACION DE
FUNDICION
CALDEROS DE
FUEGO
DIRECTO
SISTEMA DE
ALMACENAMIENTO
DE PETROLEO
400 000 Lb/h
300 000 Lb/h
VAPOR DE AGUA
215 000 Lb/h
215 000 Lb/h
240 000 Lb/h
TURBO
GENERADORES
SISTEMA DE
DISTRIBUCION
ELECTRICA
TURBINA 4
TURBINA 3
TURBINA 2
TURBINA 1
66 MW
66 MW
22 MW
22 MW
AGUA
CONDENSADO
AGUA
DESTILAD
PLANTA
DESALINIZADORA 2
660GPM
PLANTA
DESALINIZADORA 1
500GPM
LEYENDA
VAPOR
AGUA DE MAR
DISTRIBUCION
ELECTRICA
PETROLEO
OCEANO PACIFICO

Figura N°2.2

2.4 IDENTIFICACIÓN DE LOS EQUIPOS GENERADORES DE VAPOR

2.4.1 Caldero N°1

Fabricante:	BABCOCK & WILCOX
S/N o Contrato N°:	FH – 2665
Tipo:	Caldero Acuotubular
Año de Fabricación:	1958
Capacidad:	215,000 LB/Hr
Tipo de Vapor:	Sobrecalentado
Presión de Diseño:	1000 psig
Temperatura de Diseño:	910°F
Combustible Empleado:	Petróleo Residual 500

Este caldero está ubicado fuera del edificio de turbinas en el lado sureste de la planta y provee vapor a un cabezal común de vapor principal, para uso en las turbinas a vapor de la planta termoeléctrica ILO 1. Fue diseñado para emplear 6 quemadores de petróleo de los cuales 5 son atomizados mecánicamente. Este caldero es inspeccionado semestralmente y se realizan reparaciones según se requiera.

2.4.2 Caldero N°2

Fabricante:	BABCOCK & WILCOX
S/N o Contrato N°:	FH – 2665
Tipo:	Caldero Acuotubular
Año de Fabricación:	1958
Capacidad:	215,000 LB/Hr
Tipo de Vapor:	Sobrecalentado
Presión de Diseño:	1000 psig
Temperatura de Diseño:	910°F
Combustible Empleado:	Petróleo Residual 500

Este caldero está ubicado fuera del edificio de turbinas en el lado sureste de la planta y provee vapor a un cabezal común de vapor principal, para uso en las turbinas a vapor de la planta termoeléctrica ILO 1. Fue diseñado para emplear 6 quemadores de petróleo de los cuales 5 son atomizados mecánicamente. Este caldero es inspeccionado semestralmente y se realizan reparaciones según se requiera.

Figura N°2.3 – Calderos N°1 y N°2

2.4.3 Caldero N°3

Fabricante:	COMBUSTION ENGINEERING
S/N o Contrato N°:	Contrato N° 22169
Tipo:	Caldero Acuotubular
Modelo:	VU - 60
Año de Fabricación:	1970
Capacidad:	300,000 LB/Hr
Tipo de Vapor:	Sobrecalentado
Presión de Diseño:	1000 psig
Temperatura de Diseño:	910°F
Combustible:	Petróleo Residual 500

Este caldero fue construido en la planta termoeléctrica ILO 1 en el año de 1970 y está ubicado fuera del edificio de turbinas en el lado este de la planta. Provee vapor a un cabezal común de vapor principal, para uso en las turbinas a vapor de la planta termoeléctrica ILO 1. Fue diseñado para emplear 6 quemadores de petróleo de los cuales 4 son atomizados con vapor.

Es inspeccionado semestralmente y se realizan reparaciones según se requiera.

Figura N°2.4 – Caldero N°3

2.4.4 Caldero N°4

Fabricante:	ABB CE
S/N o Contrato N°:	Contrato N° 60392
Tipo:	Caldero Acuotubular
Año de Fabricación:	1993

Fecha de Comisión:	Junio de 1994
Capacidad:	400,000 LB/Hr
Tipo de Vapor:	Sobrecalentado
Presión de Diseño:	1100 psig
Temperatura de Diseño:	900°F
Combustible:	Petróleo Residual 500

Este caldero es llamado un caldero modular, pero fue ensamblado en sitio. La instalación y diseño de la unidad fue provista por BECHTEL, con la supervisión de la construcción del caldero bajo la dirección de ABB-CE. Este caldero fue puesto en operación comercial en Junio de 1994. El caldero está ubicado fuera del edificio de turbinas en el lado Norte de la planta. El caldero provee vapor a un cabezal común de vapor principal, para uso en las turbinas a vapor de la Planta Termoeléctrica ILO 1. El caldero fue diseñado para emplear 6 quemadores de petróleo de los cuales 2 son atomizados con vapor.

Figura N°2.5 – Caldero N°4

2.4.5 Calderos de Recuperación de Calor

Estos son equipos que pertenecen a la Fundición de Cobre de la empresa Southern Perú Copper Corporation (SPCC) en donde se realiza el muestreo, descarga, molienda y fundición del concentrado de Cobre.

El proceso de fundición se realiza en hornos de reverberos, equipados con 7 quemadores los cuales están diseñados para quemar hasta 27 GPM de petróleo residual N°6; la atomización se realiza mediante vapor seco con una presión máxima de 60 lbs/pulg^2 y la combustión se produce con aire precalentado hasta 800°F, bajo estas condiciones el concentrado alimentado se funde por efecto de la alta temperatura que existe dentro del horno (2400°F a 2500°F).

Los gases producidos por la combustión y la fusión de concentrado arrastran consigo algo de polvos y salen de los hornos a una temperatura de 2200°F. Estos gases calientes llegan a los cuatro calderos (dos por cada horno) diseñados para producir vapor a 865 lbs/pulg^2 de presión y a una temperatura de 910°F. La producción normal de vapor de agua es de 50,000 lbs/hr por caldero con una recuperación de calor que representa el 56% del combustible inyectado a los hornos. La temperatura de los gases que salen, de los calderos, promedia los 700°F.

Todo este vapor producido por los calderos de la fundición es enviado a la planta termoeléctrica donde es transformado en energía eléctrica. La energía eléctrica producida por la planta termoeléctrica es capaz de satisfacer todas las necesidades de consumo de toda la Mina Toquepala.

Otra ventaja de los calderos de la fundición es bajar la temperatura de los gases de manera que ellos puedan ser conducidos por los ductos metálicos hasta una planta de precipitación electrostática compuesta por ocho unidades precipitadoras, donde más del 90% de los sólidos que acompañan a los gases son recuperados. El elemento valioso en estos sólidos es el cobre que es regresado a los hornos, mezclado con el concentrado. Esta recuperación significa una gran economía para la fundición.

Finalmente, los gases que abandonan la planta de precipitación electrostática continúan por un ducto hasta una chimenea de concreto reforzado de 342’ de altura, de donde salen al ambiente así totalmente limpios.

Figura N°2.6 – Descarga de los Calderos de Recuperación

2.5 IDENTIFICACIÓN DE LOS EQUIPOS CONSUMIDORES DE VAPOR

2.5.1 Turbina a Vapor N°1

Fabricante:	BROWN BOVERI
Turbina N°:	S/N: B33236 Tipo: DSQ 2f, 42 BK
Capacidad de la Turbina	22,000 KW
Generador N°:	S/N: M32994 Tipo: WTH652d
Capacidad del Generador:	27,058 KVA
Fecha de Fabricación:	1958

Está ubicada en el lado sur del piso de turbinas de la planta termoeléctrica, con el condensador Hotwell extendiéndose a través del piso en su nivel inferior. La turbina esta soportada en una cimentación de concreto, tiene cuatro extracciones de vapor, para uso en los calentadores de agua de alimentación y en la planta desalinizadora.

La instalación de esta turbina no fue diseñada con un desareador, por ello los gases no condensables en el condensador son removidos en el pozo Hotwell.

A esta turbina se le realizó una inspección y un Overhaul Major en 1991.

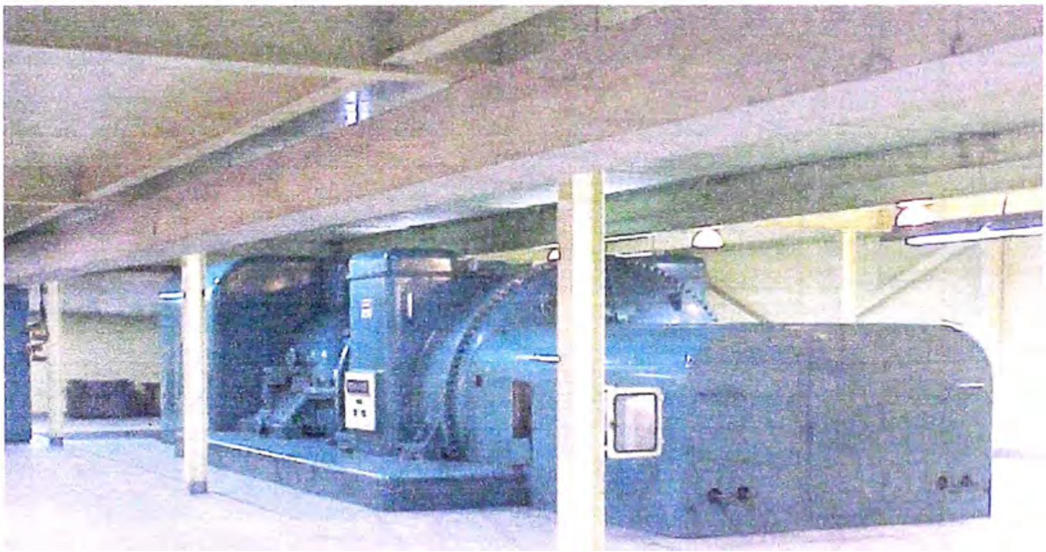


Figura N°2.7 – Turbina N°1

BROWN BOVERI TURBINE N°1 - STEAM CYCLE

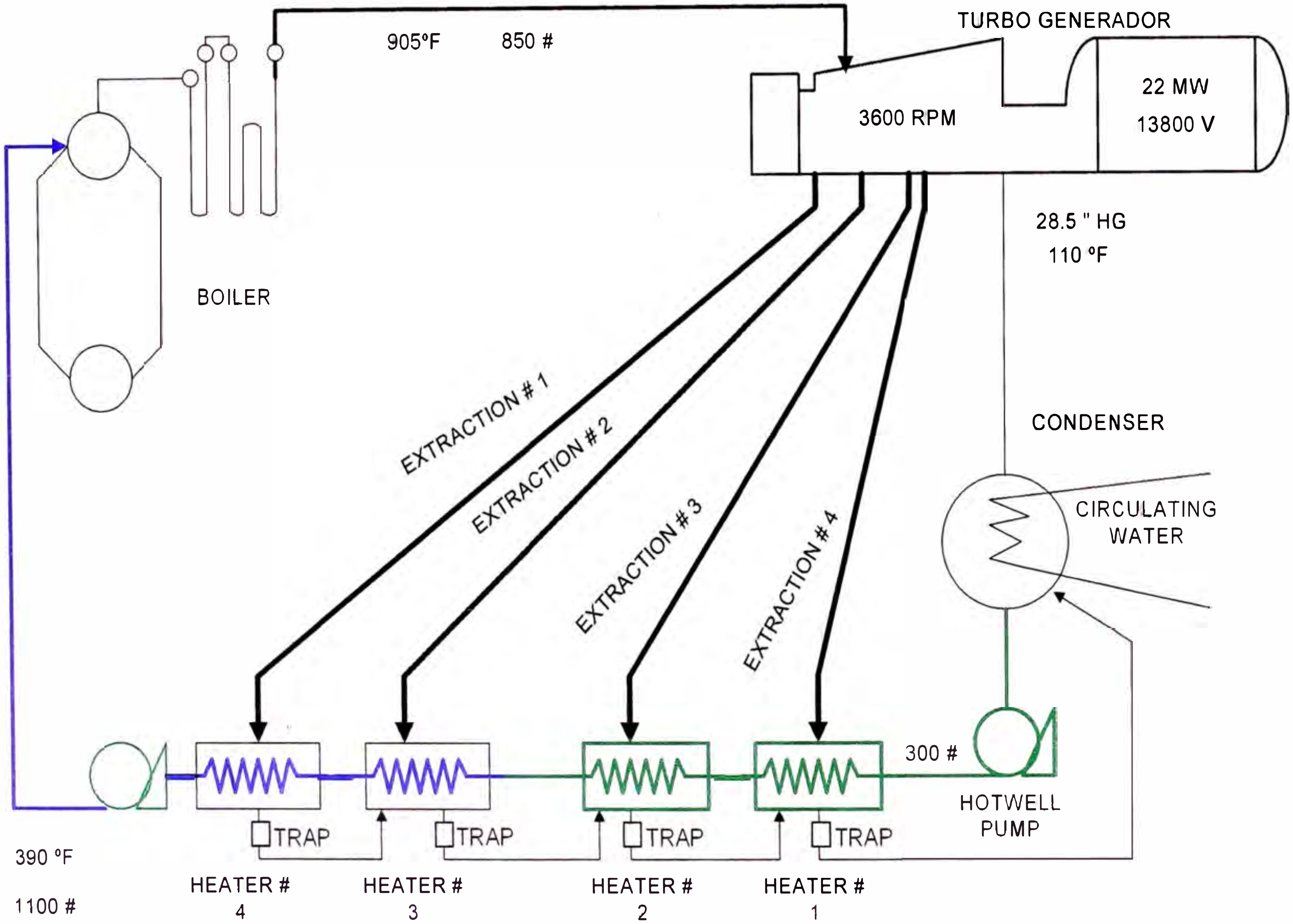


Figura N°2.8

2.5.2 Turbina a Vapor N°2

Fabricante:	BROWN BOVERI
Turbina N°:	S/N: B33185 Tipo: DSQ 2f, 42 BK
Capacidad de la Turbina	22,000 KW
Generador N°:	S/N: M32956 Tipo: WTH652d
Capacidad del Generador:	29,411 KVA
Fecha de Fabricación:	1958

Está ubicada en el lado sur del piso de turbinas de la planta termoeléctrica, con el condensador Hotwell extendiéndose a través del piso en su nivel inferior. Esta soportada en una cimentación de concreto, tiene cuatro extracciones de vapor, para uso en los calentadores de agua de alimentación y en la planta desalinizadora.

La instalación de esta turbina no fue diseñada con un desareador, por ello los gases no condensables en el condensador son removidos en el pozo Hotwell.

Figura N°2.9

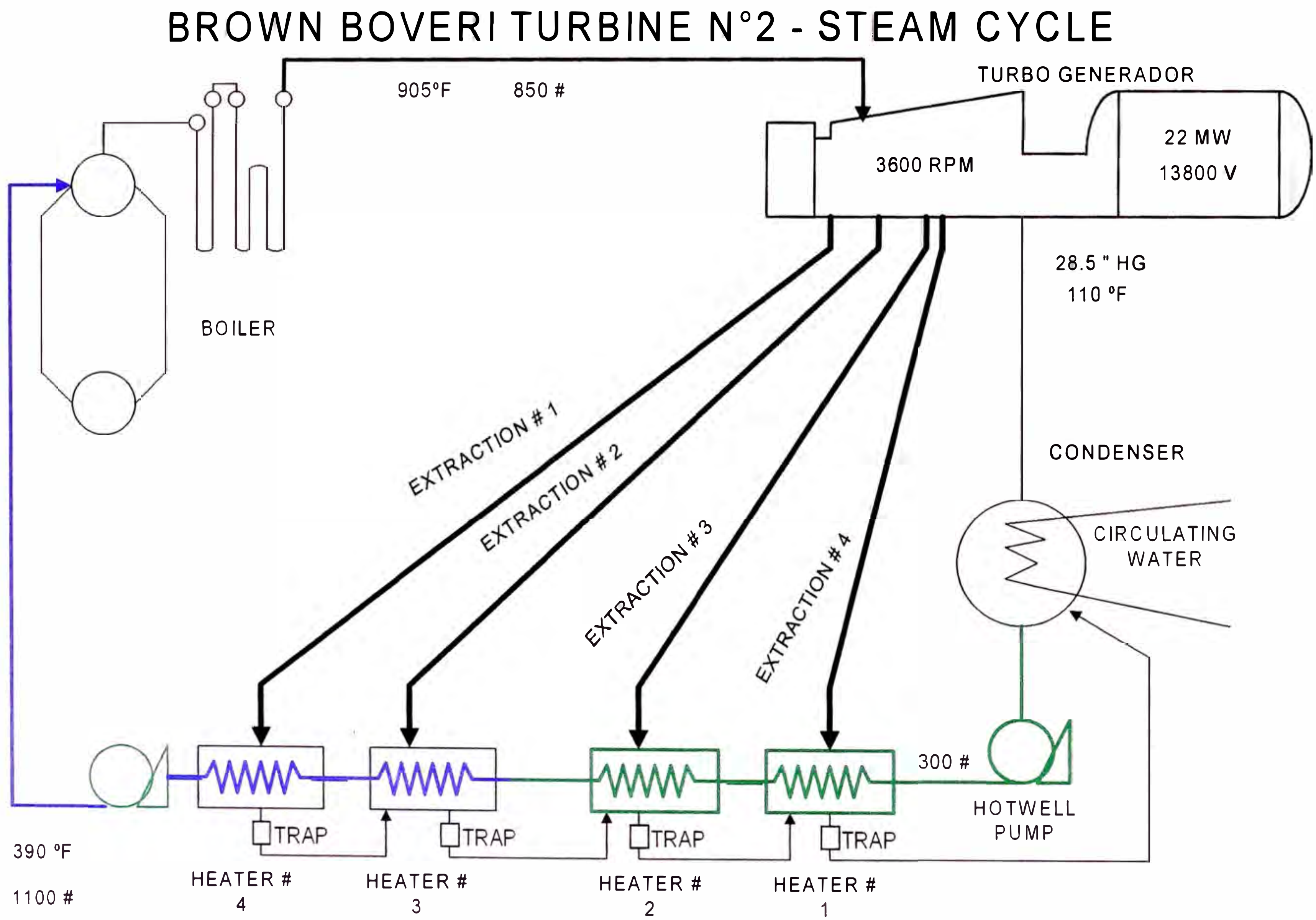
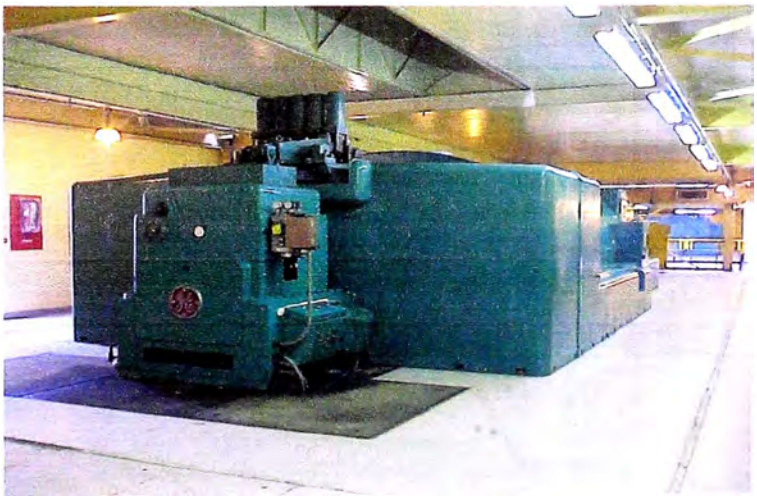

BROWN BOVERI TURBINE N°2 - STEAM CYCLE
905°F
850 #
TURBO GENERADOR
3600 RPM
22 MW
13800 V
28.5 " HG
110 °F
BOILER
EXTRACTION # 1
EXTRACTION # 2
EXTRACTION # 3
EXTRACTION # 4
CONDENSER
CIRCULATING WATER
300 #
HOTWELL PUMP
TRAP
TRAP
TRAP
TRAP
390 °F
1100 #
HEATER # 4
HEATER # 3
HEATER # 2
HEATER # 1

Figura N°2.10

2.5.3 Turbina a Vapor N°3

Fabricante:	GENERAL ELECTRIC
Turbina N°:	197791
Capacidad de la Turbina	66,000 KW
Generador N°:	316X287
Capacidad del Generador:	81176 KVA
Fecha de Fabricación:	1978

Esta turbina fue instalada en 1979, pero las tuberías de vapor y de agua de alimentación fueron instaladas en 1968. Esta ubicada en la parte central del piso de turbinas de la planta termoeléctrica, con el condensador Hotwell extendiéndose a través del piso en el nivel inferior. Esta soportada en una cimentación de concreto, tiene cinco extracciones de vapor, para uso en los calentadores de agua de alimentación y en la planta desalinizadora.



GENERAL ELECTRIC TURBINE N°3 - STEAM CYCLE

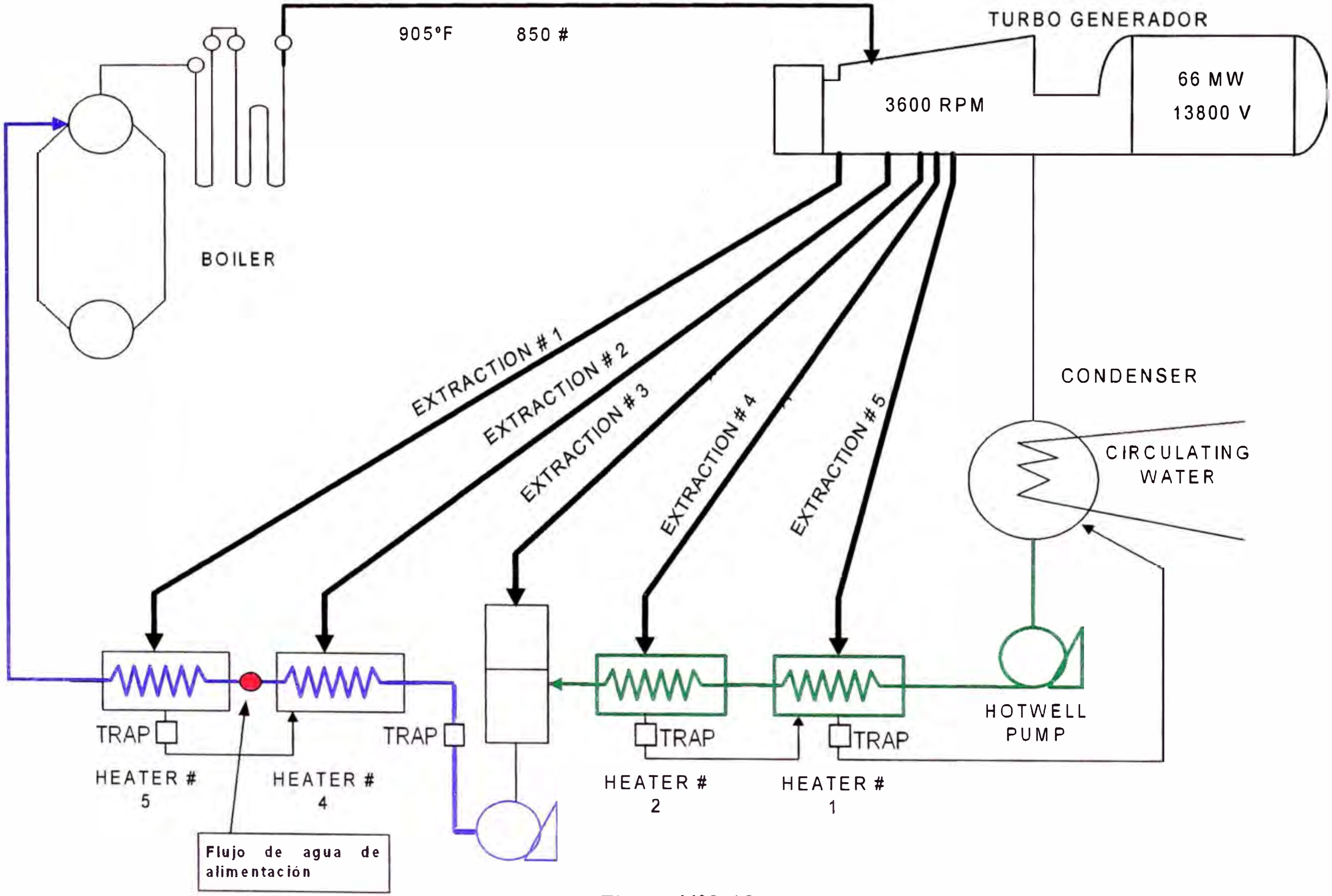


Figura N°2.12

2.5.4 Turbina a Vapor N°4

Fabricante:	GENERAL ELECTRIC
Turbina N°:	197791
Capacidad de la Turbina	66,000 KW
Generador N°:	316X287
Capacidad del Generador:	81176 KVA

Esta turbina fue instalada en 1976, esta ubicada en el lado Norte del piso de turbinas de la planta termoeléctrica, con el condensador Hotwell extendiéndose a través del piso en el nivel inferior. Esta soportada en una cimentación de concreto, tiene cinco extracciones de vapor, para uso en los calentadores de agua de alimentación y en la planta desalinizadora.

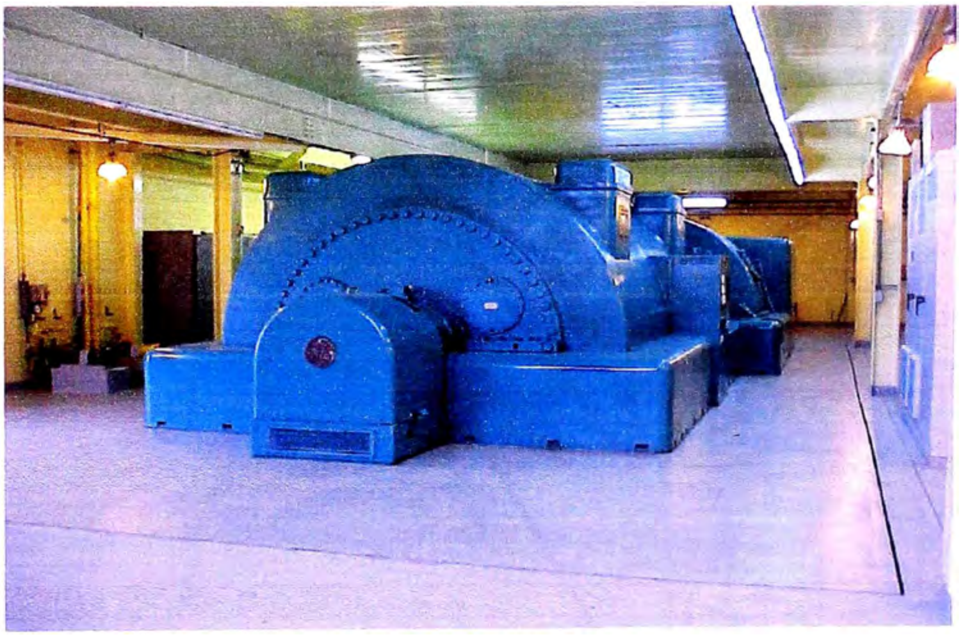


Figura N°2.13

GENERAL ELECTRIC TURBINE N°4 - STEAM CYCLE

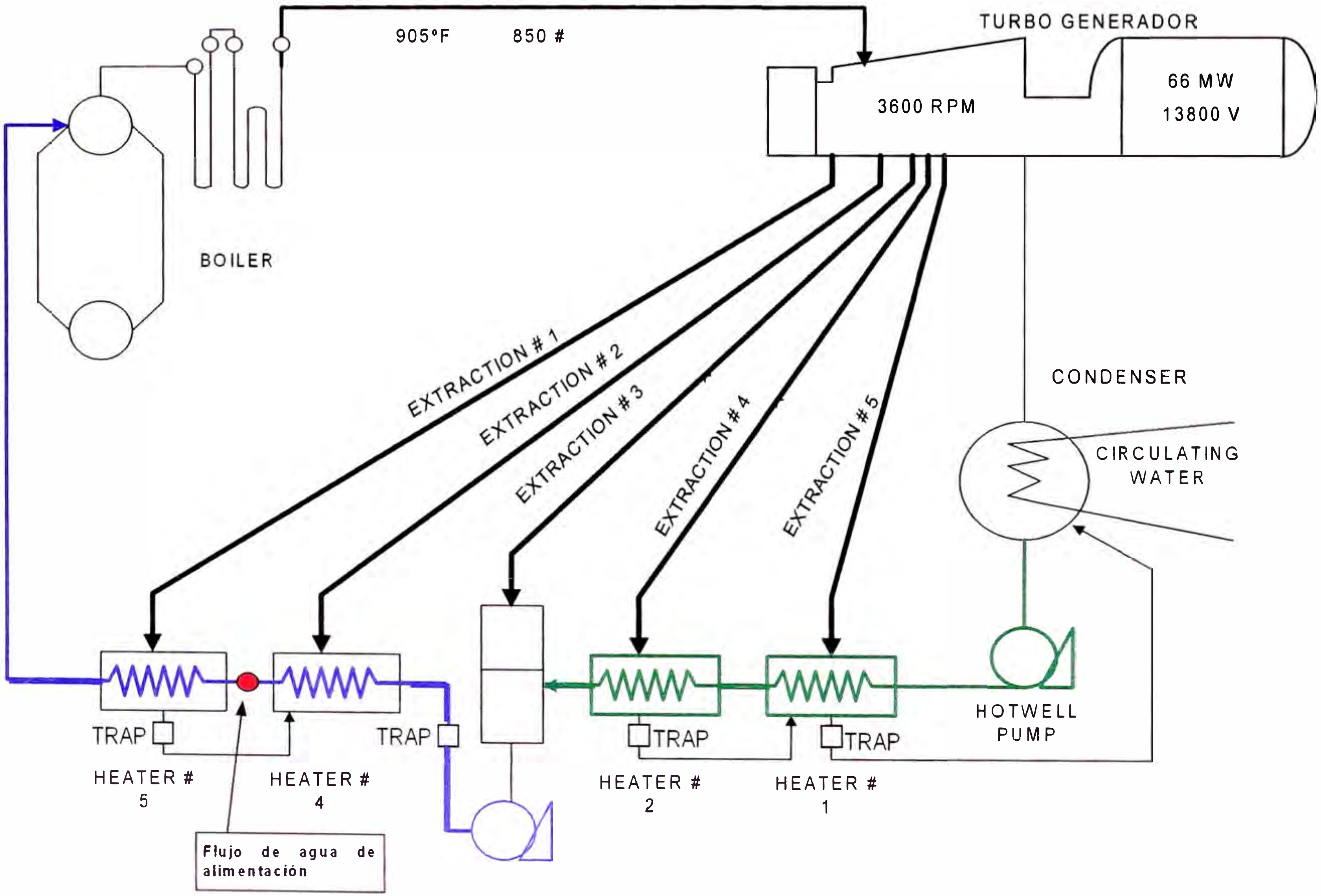


Figura N°2.14

CAPITULO 3

MARCO TEÓRICO

3.1 FUNCIONAMIENTO DE LA CENTRAL TÉRMICA A VAPOR

Las plantas de generación eléctrica a partir del vapor de agua basan su procedimiento de transformación de energía en los principios establecidos para las máquinas térmicas, las que tienen su idealización en la máquina de Carnot.

- Ciclo Carnot con Vapor:

Si se emplea vapor de agua como sustancia de trabajo, desarrollando un ciclo de Carnot, este podría constituir un ciclo de comparación para las máquinas térmicas a vapor.

Sea que se trabaje con vapor húmedo o con vapor sobrecalentado, los ciclos serían como los que muestran a continuación:

Si el ciclo trabaja con vapor húmedo, se logra que las isotermas y las isóbaras coincidan durante los procesos de transferencia de calor qA y qB. Sin embargo se puede indicar dos dificultades notables:

a) La compresión de 1 a 2 es difícil de realizar en la práctica. El trabajo de compresión es grande puesto que se comprime una sustancia pseudogaseosa.

b) La temperatura T_A queda limitada por la temperatura crítica.

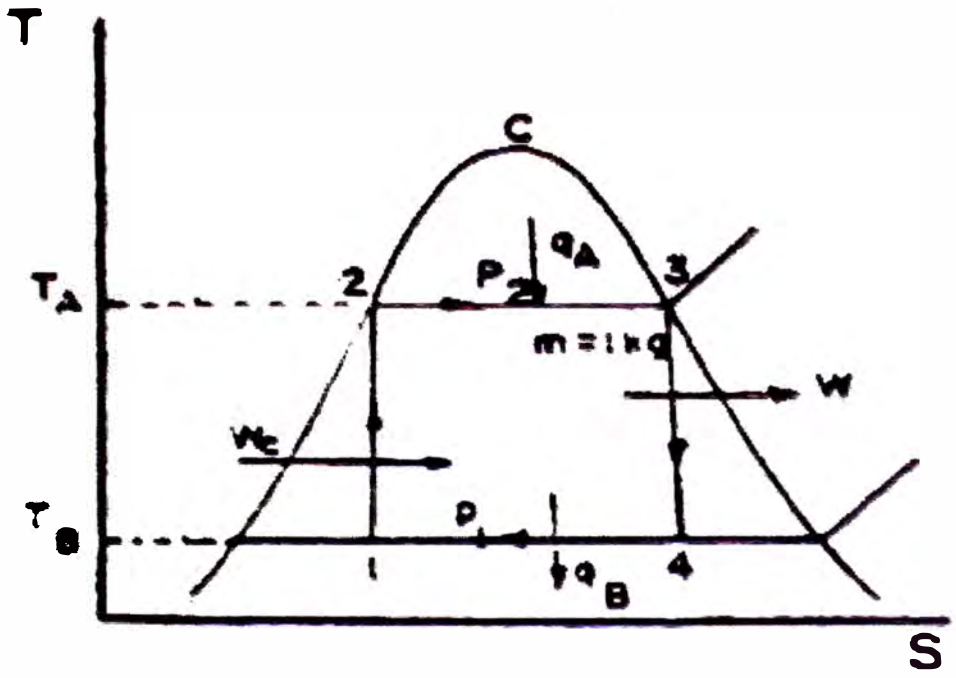


Figura N°3.1 – Ciclo de Carnot con Vapor Húmedo

Si el ciclo trabaja con vapor sobrecalentado, se observará lo siguiente:

a) La compresión tendría que ser hasta presiones supercríticas, requiriendo de una gran potencia para comprimir el agua.

b) El calor qA se transfiere a temperatura constante pero la presión varía, haciendo complicado este proceso, y por lo tanto impracticable.

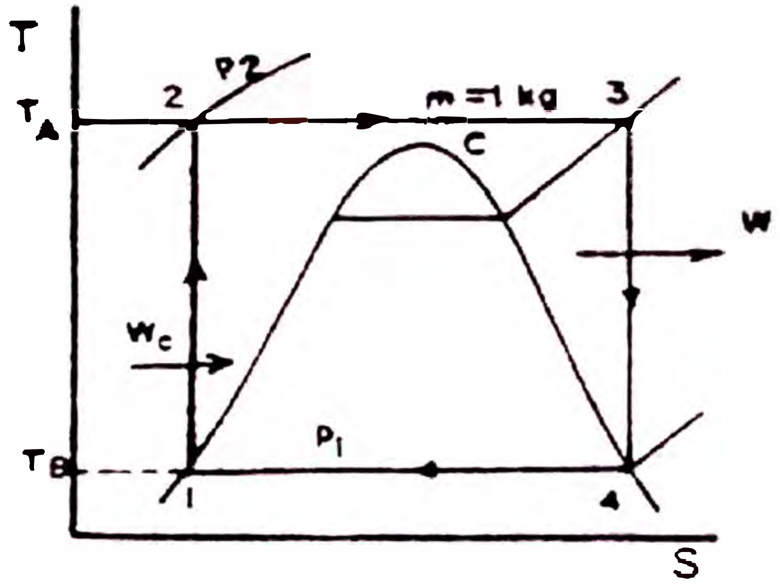


Figura N°3.2 – Ciclo de Carnot con Vapor Sobrecalentado

En resúmen es necesario recurrir a otro ciclo cuyos procesos sean realizables en la práctica, cuando se utiliza vapor de agua, en el que las dificultades de orden técnico sean mínimas.

Se ha observado que un ciclo formado por procesos isoentrópicos e isobáricos es más realizable. Uno de estos ciclos es el ciclo Clausius-Rankine, que constituye el ciclo de comparación para las máquinas térmicas que trabajan con vapor.

- Ciclo Clausius – Rankine:

El ciclo termodinámico básico que emplea la planta de generación es el Ciclo Clausius-Rankine, el cual consta de dos procesos isoentrópicos y dos procesos isobáricos. Este ciclo se muestra en las siguientes figuras:

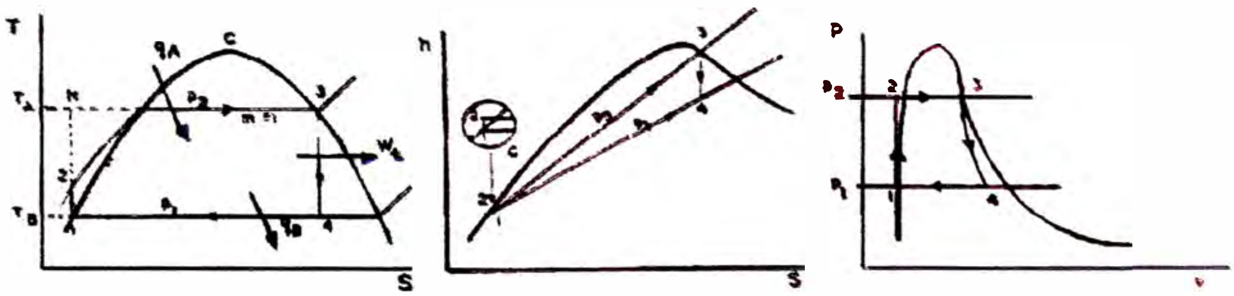


Gráfico T-s   Gráfico h-s   Gráfico p-v

Figura 3.3

El ciclo se realiza en una máquina térmica o planta de potencia ideal cuyo esquema se muestra en la siguiente figura:

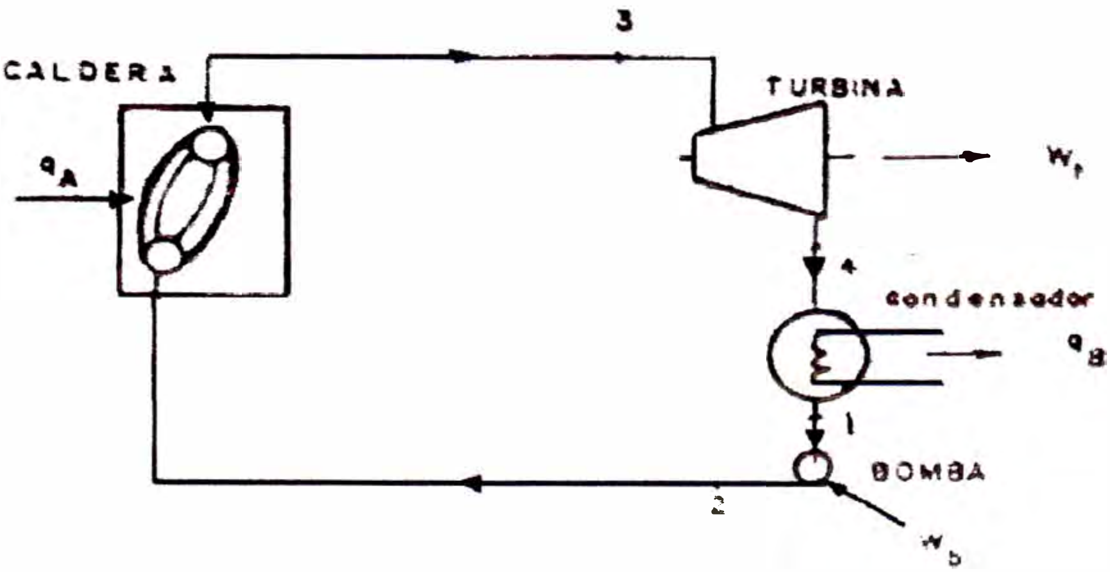


Figura 3.4 – Máquina Térmica Ideal a Vapor

Los procesos que integran el ciclo Clausius-Rankine son:

- Proceso 1-2: Bombeo de líquido, isoentrópico.
- Proceso 2-3: Calentamiento y evaporación a presión constante, en la caldera.
- Proceso 3-4: Expansión isoentrópica en la turbina a vapor.
- Proceso 4-1: Condensación del vapor a presión constante en el condensador.

Se considerará para una masa unitaria:

q_A: Calor transferido al ciclo: ${}_2q_3$

q_B: Calor transferido al sumidero: ${}_4q_1$

wt: Trabajo producido por la turbina a vapor: ${}_3w_4$

wb: Trabajo suministrado a la bomba de agua: ${}_1w_2$

El ciclo Clausius-Rankine carece de pérdidas internas, pero externamente es irreversible, puesto que no recibe el calor q_A a temperatura constante. Por lo tanto, su eficiencia será menor que la del ciclo Carnot, entre las mismas temperaturas T_A y T_B.

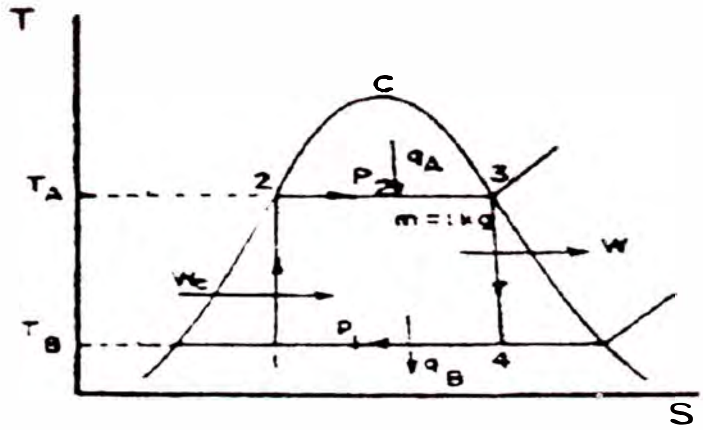


Figura 3.5 – Ciclo de Carnot con Vapor Húmedo

Parámetros característicos

- Presión de vapor: Es la presión del vapor en el ingreso de la turbina.
- Presión de descarga: Es la presión de vapor en la descarga de la turbina.
- Temperatura de vapor: Es la temperatura del vapor en el ingreso a la turbina. Se le conoce también como temperatura máxima.
- Eficiencia (η_R): Considerando despreciables ΔEc y ΔEp, y teniendo en cuenta que los procesos son del tipo denominado de flujo y estado estables (FEES):

En la turbina:

$_3q_4 = {}_2w_4 + h_4 - h_3$ pero: $_3q_4 = 0$

$\mathbf{wt = {}_3w_4 = h_1 - h_4}$ (turbina adiabática)

En el condensador:

$_4q_1 = {}_4w_1 + h_1 - h_4$

pero: $_4w_1 = \int_4^1 -v\,dp = 0$ (proceso isobárico)

$\mathbf{q_B = {}_4q_1 = h_1 - h_4}$

En la bomba:

$_1q_2 = {}_1w_2 + h_2 - h_1$ pero: $_1q_2 = 0$

$\mathbf{wb = -{}_1w_2 = h_2 - h_1}$ (bomba adiabática)

En el caldero:

$_2q_3 = {}_2w_3 + h_3 - h_2$ pero: $_2w_3 = 0$

$q_A = {}_2q_3 = h_3 - h_2$ (caldero isobárico)

Entonces el rendimiento será:

$$\eta_R = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2} = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_1) - (h_2 - h_1)}$$

$q_A = {}_2q_3 = h_3 - h_2$ (caldero isobárico)

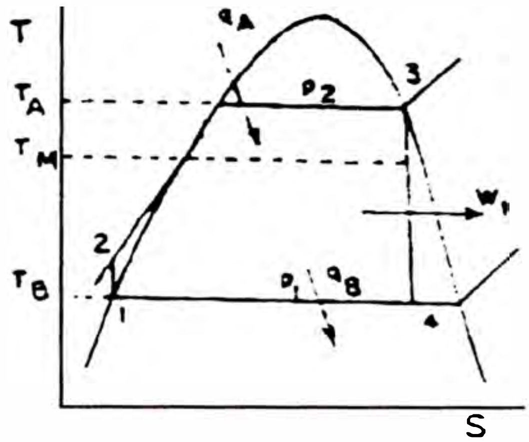


Figura 3.6 – Ciclo Clasius-Rankine

El trabajo de la bomba es $h_2 - h_1$, y puede calcularse por:

$$h_2 - h_1 = -\int_1^2 v\,dp$$

Aquí el volumen puede considerarse constante e igual al volumen del líquido saturado v_1, que ingresa a la bomba.

La bomba por comprimir líquido (fluido prácticamente incompresible), requiere de una cantidad muy pequeña de trabajo en comparación con el trabajo producido por la turbina, por lo que para algunos cálculos se le considera, en términos relativos, despreciable.

Una expresión aproximada de la eficiencia, para cálculos rápidos, sería:

$$\eta_R = \frac{(h_3 - h_4)}{h_3 - h_1}$$

El ciclo de Clausius-Rankine descrito puede ser modificado para mejorar la eficiencia, como también para resolver algunas dificultades de orden técnico que trae consigo este ciclo básico. De la expresión:

$$\eta_R = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2} = \frac{(h_3 - h_2) - (h_4 - h_1)}{(h_3 - h_2)} = 1 - \frac{h_4 - h_1}{h_3 - h_2}$$

Resulta en:

$$\eta_R = 1 - \frac{q_B}{q_A}$$

Considerando la temperatura media de transferencia de calor al ciclo:

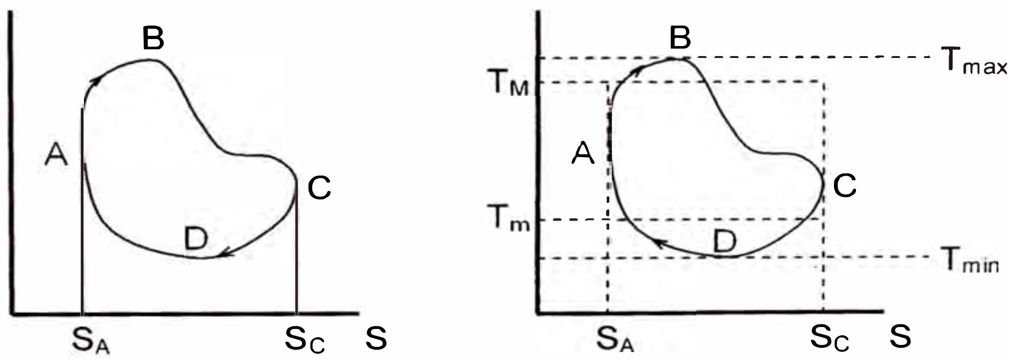


Figura 3.7 – Temperaturas Medias Termodinámicas

Durante los procesos ABC y CDA la temperatura de la sustancia no permanece constante sino que va variando a lo largo del proceso, pero se puede pensar en una temperatura hipotética, constante, tal que durante un proceso a esa temperatura que involucre el mismo cambio de entropía $s_A - s_B$ se transfiera el mismo calor que corresponde al proceso. Esta temperatura así definida se denomina "Temperatura media termodinámica de transferencia de calor" y se designa como:

T_M: La temperatura media termodinámica de transferencia positiva de calor.

Tm: La temperatura media termodinámica de transferencia negativa de calor.

$$T_M = \frac{\int_A^C T ds}{s_C - s_A} = \frac{q_A}{s_C - s_A}$$ Entonces: $q_A = T_M(s_C - s_A)$

$$Tm = \frac{\int_C^A T ds}{s_A - s_C} = \frac{q_B}{s_A - s_C}$$ Entonces: $q_B = Tm(s_C - s_A)$

La eficiencia del ciclo puede expresarse como: $\eta = 1 - \frac{Tm}{T_M}$

Es decir que la eficiencia de un ciclo cualquiera en función de las temperaturas medias termodinámicas tiene así la misma expresión formal que la correspondiente al ciclo de Carnot que es:

$$\eta = 1 - \frac{T_B}{T_A}$$

En donde T_A y T_B son las temperaturas constantes, a las que se efectúan los procesos de aportación y rechazo de calor. Se debe tener en cuenta que T_A y T_B representan las temperaturas máxima y mínima del ciclo respectivamente.

La definición de las temperaturas medias termodinámicas es importante para efectos de comparación ya que la eficiencia de un ciclo e mayor mientras mayor es su temperatura media T_M y mientras menor es su temperatura media Tm.

En otros términos si se quiere aumentar la eficiencia de un ciclo se debe incrementar su temperatura media de transferencia positiva de calor T_M y disminuir su temperatura media de transferencia negativa de calor Tm.

Existen procedimientos fundamentales, practicados en los ciclos de las plantas térmicas a vapor en la actualidad, tendientes a mejores eficiencias además de otras ventajas técnicas.

No obstante estos procedimientos, tienen inevitables limitaciones de diversa índole. Por ello, es necesario analizar para cada caso, su influencia en el ciclo, así como sus limitaciones. En este análisis se considerará una masa unitaria.

Los procesos fundamentales pueden resumirse en:

1. Sobrecalentamiento

Consiste en elevar la temperatura del vapor saturado que sale de la caldera hasta una temperatura T, a presión constante.

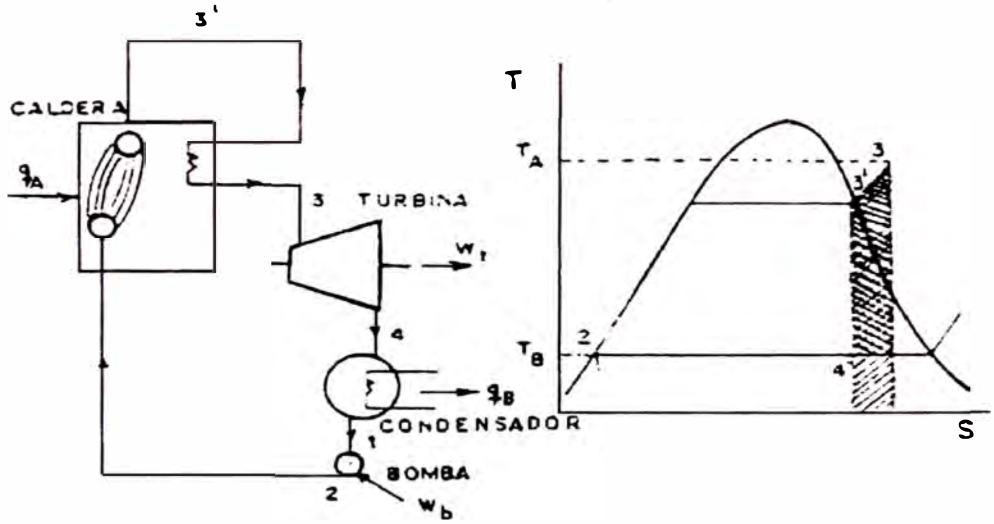


Figura 3.8 – Ciclo Clausius-Rankine con Sobrecalentamiento

El vapor saturado que sale de la caldera se hace pasar por un sobrecalentador en el que recibirá una parte del calor q_A, para elevar su temperatura hasta $T_3=T_A$, a presión constante (3' a 3).

Siendo la temperatura máxima T_3 mayor que T'_3, la temperatura media T_M será mayor. Por otra parte, la temperatura T_3 no podrá sobrepasar el límite metalúrgico permisible de las partes en contacto con el vapor sobrecalentado.

Se logra también que la expansión de 3 a 4, ocurra en su mayor parte con vapor sobrecalentado (vapor seco). Esto favorece a la turbina pues se la preserva del golpeteo permanente de pequeñas gotas de líquido que originan la erosión de los álabes. En la práctica se permite, sin embargo, que el vapor descargado por la turbina tenga una humedad que no exceda del 12%.

2. Influencia de la presión de vapor (pv)

Considerando constantes la temperatura máxima T_3 y la presión de descarga p_1, se podrá variar la presión de vapor pv.

Para el análisis se considera despreciable la influencia del trabajo de la bomba en la eficiencia, al variar la presión pv. Al elevar p_2 se observa que:

a) Se eleva la temperatura media T_M
b) Disminuye el calor perdido en el condensador (q_B)
c) La humedad del vapor en la descarga aumenta (esta es una desventaja)

A medida que la presión es mayor (incluyendo presiones supercríticas) el incremento de la eficiencia es menor. Habrá una presión máxima permisible que dependerá de la combinación de la resistencia mecánica y térmica de las partes en contacto con el vapor.

La influencia de la presión de vapor en la eficiencia se puede apreciar en la siguiente figura:

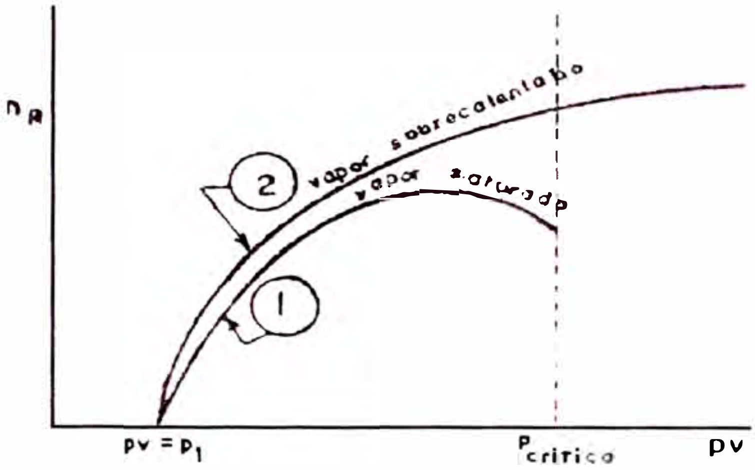


Figura 3.9 – Influencia de la Presión de Vapor

en la Eficiencia para T_A y p_1

3. Influencia de la presión de descarga

Considerando un ciclo con una presión de vapor p_2 y una temperatura T_3 constantes.

Siendo la condensación un proceso a temperatura constante, al variar la presión pd, T_B variará de igual forma.

Al disminuir la presión pd, disminuye T_B, lo que implica transferir menos calor q_B.

Esto queda confirmado con el análisis de la ecuación:

$$\eta = 1 - \frac{Tm}{T_M}$$

Al disminuir la presión pd, aumenta la humedad del vapor en la descarga. La presión de descarga pd queda limitada por:

a) La humedad en la descarga (máximo 12%)

b) La temperatura del sumidero al cual se transfiere q_B.

En las plantas a vapor, el sumidero lo constituye el agua de un depósito, un rio, el mar, etc. que se encuentra aproximadamente a la temperatura ambiente (To). Por lo tanto T_B estará limitado a un valor mínimo por encima de To.

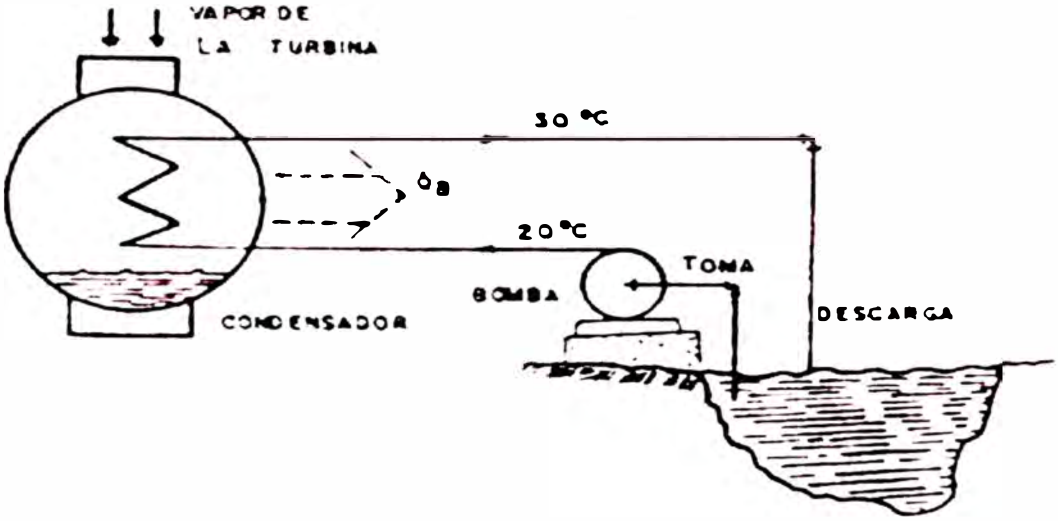


Figura 3.10 – Presión del Condensador

4. Regeneración

Se le denomina al intercambio regenerativo de calor, a las transferencias de calor efectuadas desde y hacia la propia sustancia de trabajo, en algún proceso, sin intervención de la fuente ni del sumidero. Se consigue evidentemente elevar la temperatura media de transferencia de calor al ciclo.

- Ciclo regenerativo con extracciones de vapor

El calentamiento regenerativo del agua de alimentación se puede conseguir extrayendo pequeñas cantidades de vapor, en varios puntos de la turbina durante la expansión del vapor.

Estas pequeñas masas son conducidas a calentadores de agua dispuestos en serie tal como se muestra.

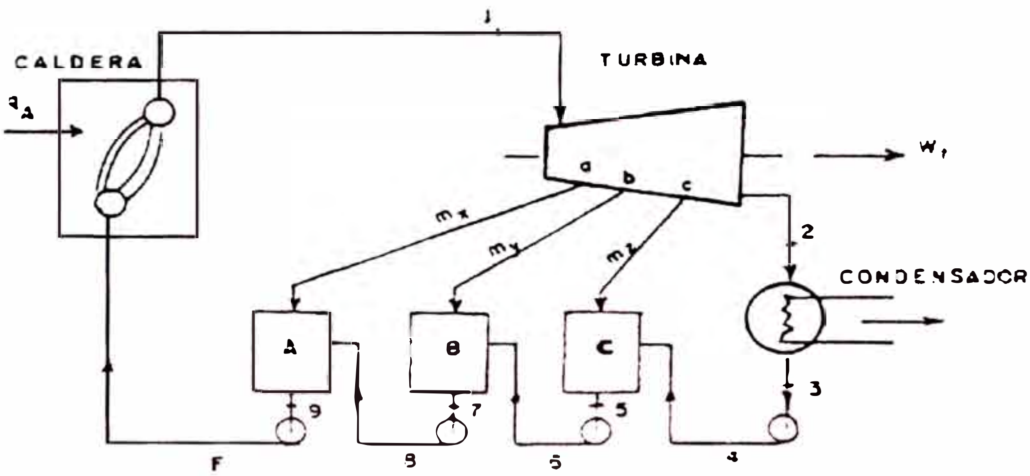


Figura 3.11 – Planta a Vapor de Ciclo Regenerativo

Durante la expansión del vapor, de 1 a 2, en los puntos a, b y c se hacen extracciones de vapor, cuyas masas son m_x, m_y y m_z; las que son conducidas a los calentadores A, B y C respectivamente.

En los calentadores, el vapor se condensa cediendo calor al agua de alimentación. La mayor parte del vapor continúa expandiéndose para producir trabajo. Estos pueden ser de dos tipos:

a) Calentadores de contacto directo

   Se les denomina también calentadores de mezcla o "abiertos". En ellos, el vapor extraído se condensa al ponerse en contacto físicamente con el agua de alimentación a la caldera, dando como resultado una mezcla saturada líquido-vapor. Las condiciones de funcionamiento para un régimen estabilizado de la planta (proceso FEES en el calentador) se muestra a continuación:

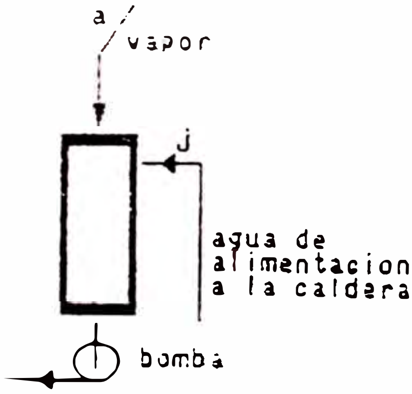


Figura 3.12 – Calentador de Contacto Directo

En este tipo de calentamiento de agua se requiere de una bomba de agua independiente para cada calentador. La temperatura de la mezcla T_K es aproximadamente igual a la temperatura de saturación a la presión de vapor ingresante pa.

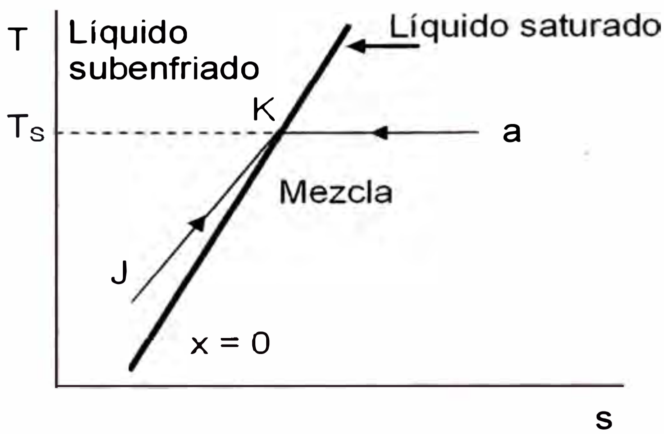


Figura 3.13 – Diagrama T-s calentador abierto

Con frecuencia se les utiliza para la eliminación de los gases no condensables en el circuito de vapor (siempre que la presión sea mayor que la atmosférica).

b) Calentadores de contacto indirecto

Se les denomina también calentadores cerrados o de superficie. En este tipo de calentadores, el vapor extraído se condensa al ponerse en contacto con los tubos dentro de los cuales circula agua de alimentación a menor temperatura.

Como consecuencia de esta transferencia de calor, el agua de alimentación eleva su temperatura. Las condiciones de funcionamiento para el régimen FEES, son las siguientes:

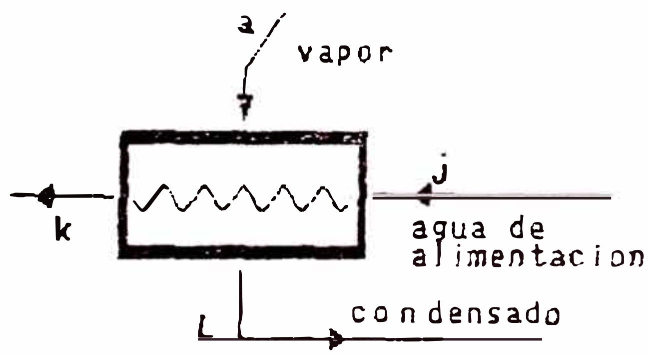


Figura 3.14 – Calentador de Contacto Indirecto

En este caso no se requiere necesariamente de una bomba de agua por cada calentador, pudiéndose emplear una bomba para dos o más calentadores en serie.

La temperatura T_K es menor que la temperatura del condensado T1, en 5 o 6°C aproximadamente, solo en el caso de intercambio de calor ideal, $T_K = T_1$.

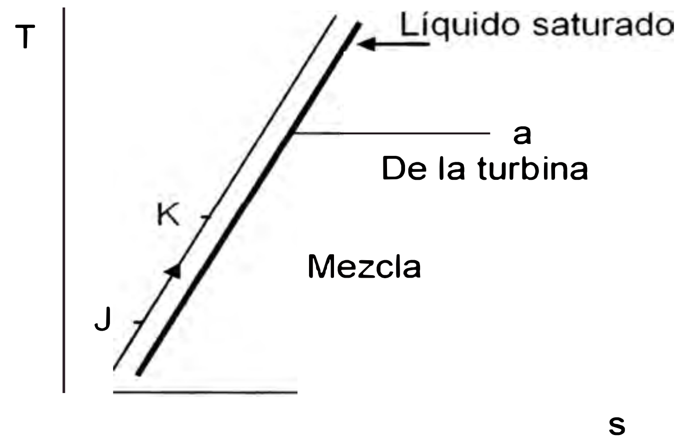


Figura 3.15 – Diagrama T-s calentador cerrado

Para que se cumpla con la condición de continuidad y no estando en contacto el vapor con el agua de alimentación, la masa m_1=m_a debe ser extraída del calentador y conducida al circuito de agua de alimentación (a no ser que se le de otro uso en la planta). Siempre será recomendable su recuperación, y para el efecto se puede proceder de dos maneras:

Enviando el condensado a una región de menor presión, pudiendo ser a otro calentador o al condensador. En estos casos se requiere reducir la presión, estrangulando.

Enviando al condensado a una región de mayor presión pudiendo ser directamente a la línea de agua de alimentación. En este caso será necesario utilizar una bomba de agua.

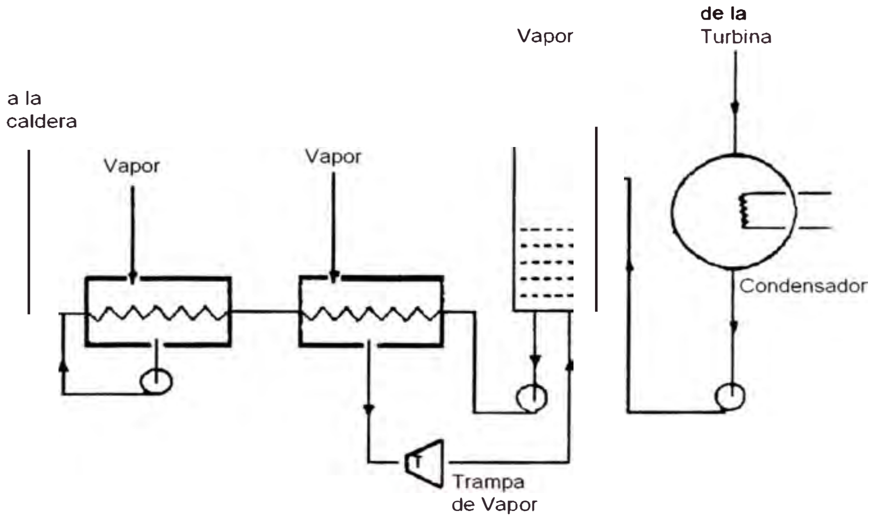


Figura 3.16 – Recuperación del Condensado en los Calentadores de Contacto Indirecto

El elemento designado como trampa de vapor es una válvula automática que permite controlar el paso del fluido hacia una presión menor, dejando ingresar únicamente el líquido, el mismo que sufre un proceso de estrangulamiento hasta la presión de salida.

La disposición de las bombas, como los calentadores es muy variada dependiendo de otras consideraciones además de las ya analizadas.

Lógicamente, el trabajo total producido por la turbina disminuirá, pero también disminuye el calor transferido al ciclo y en mayor proporción que el trabajo. Esto ocasiona un incremento de la eficiencia del ciclo, hasta un valor máximo, para luego decrecer.

El máximo valor de T_F (ver figura 3.11) queda por debajo de la temperatura de saturación, debido a las irreversibilidades en los calentadores. Sin embargo, se observará que la eficiencia del ciclo se hace igual a la de Carnot cuando el número de calentadores es infinito.

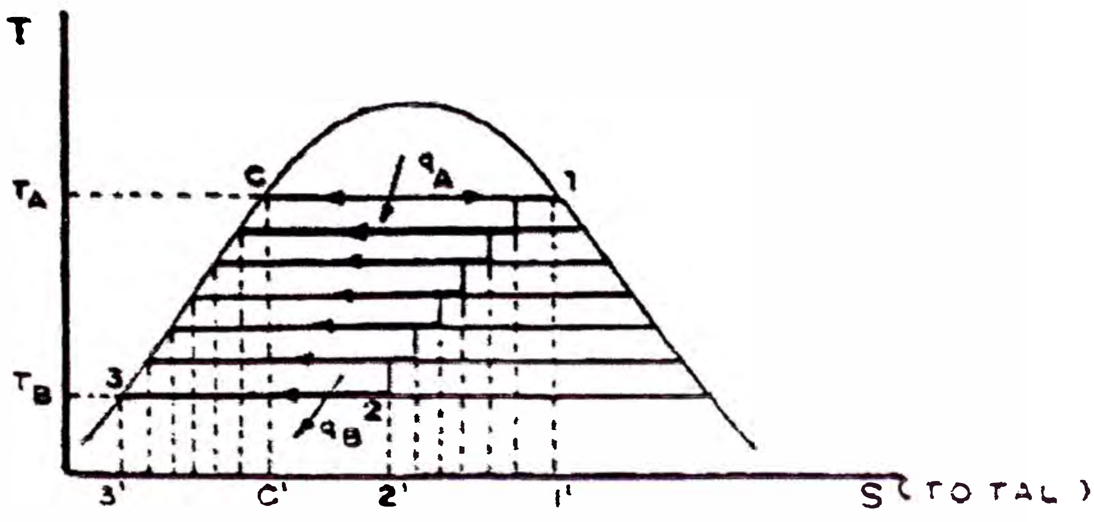


Figura 3.17 – Ciclo Regenerativo con "n" Calentadores

La eficiencia del ciclo regenerativo será:

$$\eta = \frac{q_A - q_B}{q_A} = \frac{Wn}{q_A}$$

$$q_A = T_A(s_1 - s_C)$$

$$Wn = T_A(s_1 - s_C) - \sum_{1}^{2}(T\Delta s) + \sum_{3}^{C}(T\Delta s) - T_B(s_1 - s_C)$$

Luego:

$$\eta = \frac{T_A(s_1 - s_C) - \sum_{1}^{2}(T\Delta s) + \sum_{3}^{C}(T\Delta s) - T_B(s_1 - s_C)}{T_A(s_1 - s_C)}$$

Pero:

$$\sum_{1}^{2}(T\Delta s) = \sum_{3}^{C}(T\Delta s)$$

Finalmente:

$$\eta = 1 - \frac{T_B}{T_A} = \eta_{CARNOT}$$

Es importante tener presente que:

Ciclo regenerativo: Cuando las extracciones de vapor son destinadas únicamente para calentar el agua de alimentación a la caldera.

Ciclo de extracción: Se refiere a los casos en que se realiza una o más extracciones de vapor en la turbina para diversos usos, como calentamiento de agua, procesos, calefacción, etc.

3.2 CODIGOS Y NORMAS EMPLEADOS

Los códigos por lo general establecen requerimientos para el diseño, materiales, fabricación, pruebas e inspección de sistemas de tuberías. Las normas son procedimientos recomendados para la evaluación de sistemas de tuberías. Se requiere cumplir sus requerimientos para asegurar la seguridad de los trabajadores y del público en general tal como recomienda el INDECOPI (Instituto Nacional de Defensa de la Competencia y de la Protección de la Propiedad Intelectual) a través de la NTP (Norma Técnica Peruana) en la reconoce el uso de los códigos y normas internacionales tales como ASME (The American Society of Mechanical Engineers) y API (American Petroleum Institute).

ASME es una de organización líder en el mundo que desarrolla códigos y normas sobre calderos de vapor, recipientes a presión, tuberías a presión, entre otros.

API publica especificaciones, boletines, prácticas recomendadas, normas y otras recomendaciones como ayuda para estandarizar equipos y materiales.

De las cuales según la aplicación se decidió emplear el código ASME B31.1 y la norma API RP 579 para establecer la metodología de evaluación de la condición en este informe.

3.2.1 Código ASME B31

Empezó como el proyecto B31 en Marzo de 1926, la primera edición de este estándar fue publicado en 1935. En vista de continuos desarrollos en la industria y el incremento en diversas necesidades a través de los años, se decidió publicar varias secciones del código para tuberías a presión. Desde Diciembre de 1978, el proyecto B31 de la ANS (American Nacional Standards) fue reorganizado como el código ASME B31 para tuberías a presión bajo los procedimientos desarrollados por ASME y acreditados por ANSI.

Actualmente, las siguientes secciones del código ASME B31 para tuberías a presión están publicadas:

ASME B31.1	Power Piping
USAS B31.2	Fuel Gas Piping
ASME B31.3	Process Piping
ASME B31.4	Liquid Transportation Systems for Hydrocarbons, Liquid Petroleum Gas, Anhydrous Ammonia, and Alcohol
ASME B31.5	Refrigeration Piping
ASME B31.8	Gas Transmission and Distribution Piping Systems
ASME B31.9	Building Services Piping
ASME B31.11	Slurry Transportation Piping Systems

- ASME B31.1: Código para tuberías de sistemas de potencia

Este código establece los requerimientos para el diseño, material, fabricación, construcción, pruebas e inspección de tuberías para sistemas de potencia y de servicios auxiliares para estaciones de generación eléctrica en plantas industriales y centrales termoeléctricas, sistemas de calentamiento geotérmico, y sistemas de calentamiento y enfriamiento. Este código no es aplicable para sistemas de tuberías cubiertas por otras secciones del código para tuberías a presión, y otras tuberías que están excluidas del alcance de este código.

La tubería exterior que no pertenece al caldero (a partir de la primera unión soldada en la descarga en el domo del caldero) debe ser construida de acuerdo al código ASME B31.1.

En adición a los sistemas de tuberías cubiertos por otras secciones de ASME B31, el código para tuberías a presión, ASME B31.1 no cubre los siguientes:

- Componentes cubiertos por el código ASME para calderos y recipientes a presión (excepto la tubería conectada no cubierta por el código ASME para calderos y recipientes a presión debe cumplir con los requerimientos del ASME B31.1).
- Tuberías para distribución de vapor en edificios diseñadas para soportar 15 psig o menos, o sistemas de tuberías para agua caliente diseñados para soportar 30 psig o menos.

- Tuberías para drenajes de techo y piso, alcantarillas, y regadoras y otros sistemas de protección.
- Tuberías para herramientas hidráulicas, neumáticas y sus componentes luego de la primera válvula de bloqueo luego del cabezal del sistema de distribución.
- Tubería para instalaciones marinas y otras bajo control federal.
- Tuberías cubiertas por otras secciones del ASME B31 y ASME sección III.
- Tuberías de gas combustible dentro del alcance de ANSI Z 223.1, código nacional de gas combustible.
- Tubería para combustible pulverizado dentro del alcance de la NFPA.

Los requerimientos de este código se aplican para sistemas centrales de calefacción para distribución de vapor y agua caliente lejos de las centrales ya sea subterránea o donde fuera, y sistemas de tuberías geotérmicas de vapor y agua caliente juntas y desde el cabezal del pozo.

Para tuberías de sistemas de potencia (diferentes de los sistemas relacionados a la seguridad en las centrales nucleares) construidas y nuevos sistemas de tuberías construidos bajo el código ASME B31.1, las siguientes pautas deben ser usadas para determinar la edición efectiva y adenda del código ASME B31.1:

“Las ediciones son efectivas y deben ser usadas en o después de la fecha de publicación impresa en la primera página. Las adendas son efectivas y deben ser usadas en o después de la fecha de publicación impresas en la primera página.”

La última edición y adenda, publicada antes de la fecha del contrato original para la primera fase de la actividad que involucra un sistema de tuberías debe ser el documento que regula las actividades de diseño, selección de materiales, fabricación, construcción, inspección, y pruebas para los sistemas de tuberías hasta el cumplimiento del trabajo y operación inicial. Excepto que el acuerdo sea específicamente extendido entre las partes contratantes, ninguna edición del código y/o adenda debe ser retroactiva.

Los casos del código pueden ser usados después de haber sido aprobados por el consejo del ASME. Las provisiones de un caso del código pueden ser usados incluso después de su expiración o retiro, provisto el caso de código que fue efectivo en la fecha del contrato original y fue usado para la construcción original o fue adoptado antes del cumplimiento del trabajo y las partes contratantes acordaron su uso.

No se usa las revisiones o casos del código que son menos restrictivos que los requerimientos formales sin haberse asegurado que hayan sido autorizados por la autoridad competente donde la tubería es instalada.

3.2.2 Norma API RP 579

Esta norma fue desarrollada por la Materials Properties Council (MPC) en 1990 la cual concentró el desarrollo de tecnología y sus resultados se publicaron en los volúmenes del ASME PVP. La culminación de este programa fue el desarrollo y la publicación de la Práctica Recomendada API 579 para la evaluación por disponibilidad para el servicio.

El API RP 579 es organizado en forma modular basada en el tipo de material dañado o indicación encontrada para facilitar su uso y actualización. Emplea tres niveles de evaluación:

Evaluación de Nivel 1: Puede ser realizada por un ingeniero de planta.

Evaluación de Nivel 2: Requiere por lo menos un ingeniero de planta.

Evaluación de Nivel 3: Debe ser realizado por ingenieros expertos o por un equipo de ingenieros que incluya por lo menos un ingeniero experto.

La aplicación de los niveles superiores de valoración es a menudo limitada por la falta de datos de propiedades de materiales y los datos exactos de operación.

Cada sección del API RP 579 identifica los requerimientos, la aplicabilidad y limitaciones de los procedimientos de evaluación. Los diagramas de flujo, figuras y ejemplos son provistos para simplificar el uso de los procedimientos. Hay también recomendaciones para el monitoreo en servicio y otros métodos paliativos que aplican en situaciones donde la evaluación es difícil. Cada sección entrega recomendaciones en técnicas de análisis de esfuerzos, mediciones NDE, y propiedades de los materiales. Como un paso esencial para la evaluación por disponibilidad para el servicio, la mínima vida remanente (RL) de un componente debe ser evaluado. El valor mínimo de la vida remanente es la base para establecer los intervalos de inspección. En casos donde no es posible evaluar la vida remanente, se deben realizar monitoreos para estar seguro de que algún problema que puede estar desarrollándose y aparecer en el servicio futuro pueda ser detectado y ubicado.

Los equipos de una planta de generación están expuestos a ambientes corrosivos y/o elevadas temperaturas. Bajo estas condiciones, los materiales usados en estos equipos pueden degradarse o envejecer durante su tiempo de servicio. Equipos importantes tales como recipientes a presión, tuberías, y tanques de almacenamiento empiezan a degradarse, el operador de la planta debe decidir si esta puede seguir funcionando segura y confiablemente para evitar daños al personal y al público, daños al medio ambiente, y paradas inesperadas. Los procedimientos para evaluación por disponibilidad para el servicio proveen información para que el operador de planta haga estas decisiones basadas en principios de ingeniería.

La evaluación por disponibilidad para el servicio es un análisis de ingeniería multidisciplinario para determinar si los equipos cumplen los requerimientos para su servicio continuo hasta el fin de su periodo de operación deseado, es decir hasta su próximo overhaul o parada planificada. Existen razones comunes para evaluar la disponibilidad para el servicio de equipos incluyendo el descubrimiento de indicaciones tales como áreas localmente delgadas (LTA) o discontinuidades, defectos que superan los actuales estándares de diseño, y planes para operar bajo las más severas condiciones que originalmente se esperaban. Los principales resultados de la evaluación por disponibilidad para el servicio son (1) una decisión para operar, alterar, reparar, monitorear, o reemplazar el equipo y (2) guía para intervalos de inspección de equipos. La evaluación por disponibilidad para el servicio aplica métodos analíticos para evaluar indicaciones, daños y desgaste del material.

Los métodos analíticos están basados en análisis de esfuerzos, pero ellos también requieren información de la operación del equipo, ensayos no destructivos (NDE), y propiedades del material. El análisis de esfuerzos puede ser realizado usando manuales estándares o fórmulas de normas de diseño o por medio de análisis por elementos finitos (FEA). Con la moderna tecnología de computadoras, el uso de FEA es muy común. La evaluación por disponibilidad para el servicio requiere el conocimiento del historial de las condiciones de operación y una previsión de futuras condiciones de operación. La interacción con el personal de operaciones es requerida para obtener esta información. Los ensayos no destructivos se usan para ubicar, dimensionar, y caracterizar discontinuidades. Las propiedades del material deben incluir información de los mecanismos y comportamiento en el ambiente de servicio, especialmente con los efectos de corrosión y temperatura.

La evaluación por disponibilidad para el servicio es requerida por varias razones. Entre las razones más importantes están los siguientes:

- Mantener la seguridad del personal de la planta y del público.
- Conocimiento del nivel de degradación de las instalaciones.
- Mantener operaciones seguras y confiables.

- Mantener la factibilidad de incrementar la severidad de las operaciones.
- Racionalizar el daño encontrado por más rigurosas inspecciones en servicio que el encontrado por inspecciones realizadas durante la construcción original.

3.3 ENSAYOS NO DESTRUCTIVOS

El objetivo de estos ensayos es caracterizar la condición actual del material de las tuberías sin dañar o destruir algún componente. La examinación es orientada a encontrar discontinuidades o la degradación que puede haberse desarrollado durante la vida operativa del componente o sistema, a la fecha.

Varios métodos NDE pueden ser aplicables a un material en particular entre los cuales se recomiendan los siguientes:

- Inspección Visual
- Partículas Magnéticas
- Ultrasonido
- Radiografía
- Replica Metalográfica
- Análisis de Aleación

3.3.1 Inspección Visual

La examinación visual es probablemente la más antigua y la más ampliamente usada de todas las inspecciones. Se usa para determinar alineamiento de superficies, dimensiones, condición de la superficie, acabado de soldaduras, marcas y evidencia de filtraciones, entre otros.

En muchas instancias la manera de conducir una inspección visual se deja a la discreción del inspector, pero los procedimientos más recientes describen nuevos requerimientos tales como acceso, iluminación, ángulo de visión, uso de equipo directo o remoto, y listas de control en donde se verifican las observaciones requeridas son empleadas actualmente. La examinación visual toma lugar a través del ciclo de fabricación. Para empezar, esto puede consistir de la verificación del material, procedimientos de soldadura, calificación de soldadores, metal de aporte, y alineamiento de la junta, y cumplimiento de fabricación, tales cosas como dimensiones finales, soldadura de la junta, condición de la superficie y limpieza.

3.3.2 Inspección con Partículas Magnéticas

El examen de partículas magnéticas es esencialmente una inspección superficial. Este tipo de examinación esta limitado a materiales que pueden ser magnetizados (materiales paramagnéticos), dado que se basa en las líneas de fuerza dentro de un campo magnético.

La pieza a ser examinada es sometida a una corriente la cual produce líneas de fuerza magnética en la pieza. La superficie es entonces rociada con polvo de hierro. Las partículas del polvo de hierro se alinean con las líneas de fuerza. Cualquier discontinuidad normal a las líneas de fuerza podrá producir un campo alrededor de la discontinuidad y por lo tanto un agrupamiento de polvo de hierro alrededor del defecto. La examinación debe ser repetida a 90° para detectar discontinuidades que pueden ser paralelas a las líneas del campo magnético inicial.

Hay muchas variaciones de examinación por partículas magnéticas dependiendo de la manera en la cual el campo es aplicado y si las partículas son húmedas o secas y fluorescentes o coloreadas.

3.3.3 Inspección Ultrasónica

La inspección ultrasónica es usada en tuberías para la detección de discontinuidades en soldaduras y materiales tanto como para determinar espesor de material.

Una ráfaga corta de energía acústica es transmitida en la pieza a ser examinada y el eco se refleja desde los diferentes bordes de la pieza. Un análisis del tiempo y amplitud de eco provee los resultados de la examinación.

Un reloj en el equipo actúa para iniciar y sincronizar los otros elementos: Acciona un pulso que envía una señal eléctrica de corta duración a un transductor, usualmente a una frecuencia de 2.5 MHz. El transductor convierte la señal eléctrica a una vibración mecánica. La vibración como ultrasonido atraviesa a través de un medio acoplante (como la glicerina B) y a través de la pieza a una velocidad la cual es una función del material. Como el sonido es reflejado desde varios planos, este retorna hacia el transductor o algunas veces a un segundo en donde es convertido a una señal eléctrica la cual es enviada a un amplificador para su despliegue en tubo de rayo catódico. El eje horizontal de la pantalla se relaciona con el tiempo y el eje vertical con la amplitud. La indicación en el extremo izquierdo muestra el tiempo y la amplitud de la señal transmitida desde el transductor. Indicaciones a la derecha mostrarán el tiempo y el grado de reflexión desde varias paredes o discontinuidades internas.

La capacidad de una inspección ultrasónica para detectar discontinuidades depende de la geometría de la pieza y la orientación del defecto. Si el plano de la discontinuidad es perpendicular al haz del sonido, este actuará como una superficie reflejante. Si esta es paralela al haz del sonido, entonces no será una superficie reflejante y de acuerdo a ello no se mostrará en el osciloscopio. Por consiguiente, la técnica de búsqueda debe ser cuidadosamente escogida para asegurar que se pueda cubrir todas las posibles orientaciones del defecto.

El defecto más serio en una junta soldada de una tubería esta orientado en la dirección radial. La técnica más comúnmente usada para detectar tales defectos es la búsqueda con onda de corte. En este procedimiento, el transductor es ubicado a un lado de la soldadura y a un ángulo con respecto a la superficie de la tubería. El ángulo se mantiene con un bloque de plástico que transmite el sonido a través de la tubería y la soldadura. Estando a un ángulo, se reflejará desde las superficies de la tubería hasta que se atenúe. Cualquier superficie que sea perpendicular al haz de sonido, sin embargo, reflejará una parte del sonido y regresará al transductor y se mostrará como una indicación en el osciloscopio. Si el ángulo del haz de sonido y el espesor del material son conocidos, la superficie reflejante podrá ser localizada y evaluada.

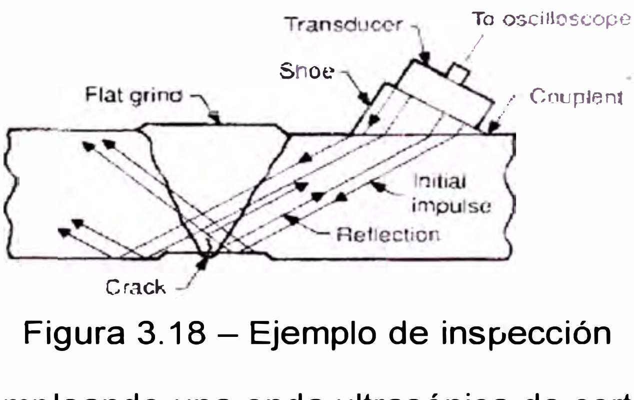


Figura 3.18 – Ejemplo de inspección empleando una onda ultrasónica de corte

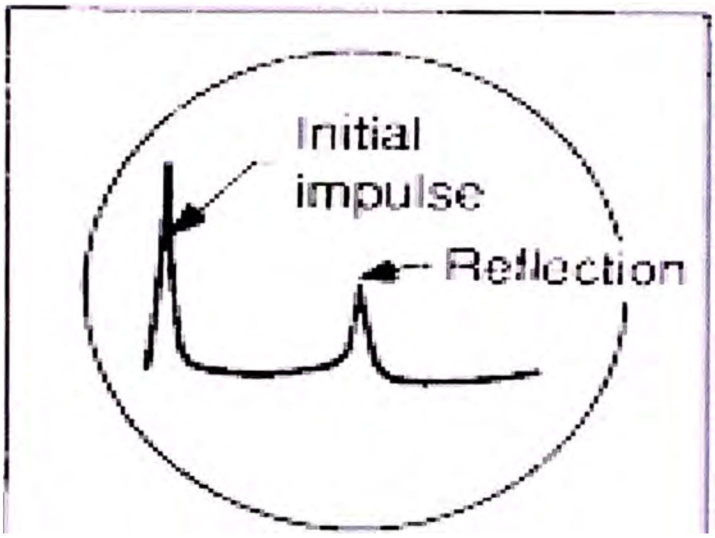


Figura 3.19 – Imagen mostrada en el osciloscopio

Antes de una inspección periódica, el equipo es calibrado contra defectos artificiales para conocer el tamaño y la orientación en un bloque de calibración. El bloque debe ser representativo del material que es inspeccionado (por ejemplo: materiales de similares propiedades acústicas, espesores, forma exterior y acabado superficial).

Una variación de inspección ultrasónica puede ser usada para medir el espesor del material. Si la velocidad del sonido en el material es conocida, el tiempo que la señal toma para atravesar el espesor y retornar puede ser convertido a una medición del espesor.

3.3.4 Radiografía

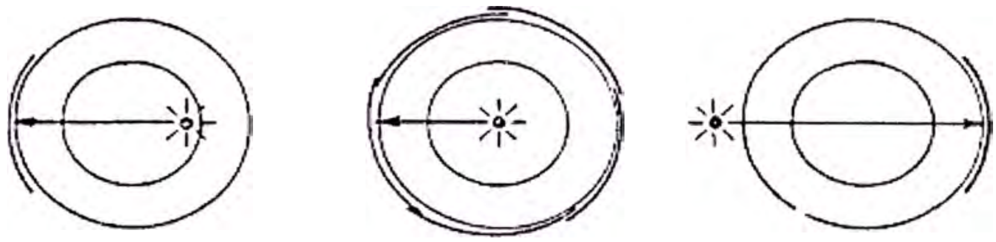
Cuando la necesidad de mayor integridad en la soldadura debe ser demostrada, la examinación más frecuentemente especificada es la radiografía. Desde que la condición interna de la soldadura puede ser evaluada, esta es referida como una examinación volumétrica.

Las fuentes radiográficas usadas para la examinación de tuberías son usualmente Rayos X o Rayos Gamma como isótopos radioactivos. Mientras que los equipos de Rayos X se usan a menudo, tienen a menudo limitaciones debido a que se requieren múltiples exposiciones para una sola junta, y se necesitan equipos especiales, como los aceleradores lineales, para espesores mayores. Aunque los equipos de rayos X producen placas con mejor claridad, no son prácticos en el campo debido a las limitaciones en el espacio y portabilidad. En el campo, los isótopos radioactivos son usados casi exclusivamente debido a su portabilidad y en caso de acceso. Para espesores mayores a 2.5 pulg de acero, el isótopo más comúnmente usado es el Iridio 192, después esta el Cobalto 60 que es usado para espesores por encima de 7 pulg.

Las fuentes radioactivas normalmente usadas en trabajos de tuberías varían en intensidad desde unos pocos curies hasta 100 curies. Cada fuente decae en intensidad de acuerdo a su vida media. Como la intensidad decae, se requieren mayores tiempos de exposición. El iridio 192 tiene una vida media de 75 días, mientras que el cobalto 60 tiene una vida media de 5.3 años.

Las fuentes radioactivas tienen dimensiones definidas y como resultado producen un efecto de sombra en la placa. Esto es referido como la penumbra geométrica, y es directamente proporcional al tamaño de la fuente e inversamente proporcional a la distancia entre la fuente y la película. ASME sección V ha establecido límites para la penumbra geométrica.

Idealmente para tuberías, la fuente es ubicada dentro de la tubería y al centro de la tubería examinada, con la película fuera de la superficie de la soldadura, permitiendo así una exposición panorámica. Donde la penumbra geométrica impide esta práctica, la fuente puede ser ubicada en el interior de la pared opuesta y una parte de la soldadura es impactada y debido a ello varias exposiciones pueden necesitarse. La fuente puede también ser ubicada fuera de la tubería y la exposición hecha a través de dos paredes. De nuevo esto requiere múltiples exposiciones y tiempos de exposición más largos.



Given:
1. 30-in dia. X 7-in wall C/S pipe
2. 10 cu CO_{60} X 0.051 dia. source
3. 50 cu CO_{60} X 0.125 dia. source
4. 100 cu CO_{60} X 0.181 dia. source
5. Max. allowable geometric unsharpness $Ug = \frac{Fd}{D} = 0.07$ in (ASME Sect. V Art. 2 para. T-274)
6. Use Kodak "AA" Film with a 2.5 film density

Min. SFD = 5.1 in for 10 cu AX 0.051 in dia.
12.5 in for 50 cu X 0.125 in dia.
18.1 for 100 cu X 0.181 in dia.

Using the parameter established in 1 above as well as 5 and 6, it would be possible to:

(*a*) Use a 10-cu source as established in 2 above and shoot the pipe using the panoramic technique, sketch B above, with an exposu e time of 3 hrs or

(*b*) Use a 50-cu source as established in 3 above and shoot the pipe using the panoramic technique, sketch B above, with an exposure time of 1 hour 5 min or

(*c*) Use a 100-cu source as established in 4 above and shoot the pipe using the opposite wall technique, sketch A above, with an exposure time of 50 min. As many as six exposures might be required—total 5 hrs.

Figura 3.20 – Efecto del tamaño de la fuente
en la técnica radiográfica

Una radiografía es considerada aceptable si el requerido agujero esencial o tamaño de alambre del indicador de calidad (penetrámetro) de la imagen es visible en la película.

3.3.5 Réplica Metalográfica

Una evaluación de la microestructura puede ser realizada en la superficie del componente a ser evaluado, esta evaluación es llamada Réplica Metalográfica, con la cual se puede obtener una imagen de la microestructura de la superficie. El área a ser examinada es cuidadosamente pulida hasta un acabado de espejo usando tela de paño (pana de pelo corto), lijas cada ves más finas o discos de amolado, y compuestos de pulido (pasta de diamante de 1 micra, alúmina), se realiza un ataque químico con Nital. Luego se aplica una película delgada de plástico a la superficie. Esta película de plástico endurece a medida que va secando en la superficie pulida, mediante esto se queda grabada la microestructura de la superficie en la película de plástico.

Cuando esta técnica es realizada por técnicos experimentados, la resolución de la estructura de la superficie en magnificaciones a 500X o más es casi igual al conseguido en una muestra del mismo metal. La desventaja del método de réplica es que solo la superficie del material puede ser examinada, dejando cualquier daño subsuperficial sin poder ser detectado. Sin embargo, este método es muy útil cuando se aplica a regiones de soldadura, u otras áreas de alto esfuerzo donde se sospecha la degradación del material.

La determinación de la vida remanente realizada por este método no es exacta: la correlación entre el tipo y grado de daño, y el tiempo durante el cual el material estuvo degradándose es solo aproximado. En muchos casos, la inspección continua durante varios años es necesaria para determinar la velocidad de progresión del daño. Usualmente, cuando una red de microfisuras ha sido generada, es tiempo de considerar la reparación o reemplazo del componente evaluado.

3.3.6 Análisis de Aleación

Se recomienda realizar el análisis de aleación en zonas de la tubería de vapor debido a que el efecto de la temperatura y el tiempo de servicio tienen alta influencia sobre los factores que gobiernan las propiedades físicas de los aceros aleados, en el caso de las tuberías de vapor principal de la CT ILO 1 son del material A335 Gr. P11 (1¼Cr-½Mo-Si).

Dada su composición este es un acero resistente al creep, el cual tiene en su composición los elementos Cr, Mo y Si.

El Cr mejora la resistencia a la corrosión y oxidación a elevadas temperaturas.

El Mo incrementa la resistencia mecánica a elevadas temperaturas.

El Si mantiene a estabilidad del Fe a altas temperaturas.

El fenómeno de creep o fluencia en caliente se caracteriza por un alargamiento contínuo y progresivo hasta llegar a la rotura debido a la tensión (o esfuerzo) constantemente aplicada (tensión inferior a la carga de rotura) a una temperatura elevada.

Las más importantes características metalúrgicas del material son la composición química, estructura y tamaño de grano los cuales son afectados por la temperatura de servicio.
Por ello que se recomienda realizar el análisis de aleación para evaluar la composición química del material que conforma las tuberías de vapor principal.

Uno de los equipos más utilizados es el XMET3000TX el cual es un equipo que emite una señal XRF (Fluorescencia de Rayos X) y su funcionamiento es de la siguiente manera:
Las muestras son bombardeadas con rayos X producidas por radioisotopos tales como Fe-55, Co-57, Cd-109 o tubos de rayos X. Cuando una muestra es irradiada, la fuente de rayos X puede seguir esparciendo o absorviendo átomos de la muestra.
Si la energía de la fuente de rayos X es mayor que el límite de energía de absorción de la capa interna del electrón, entonces los electrones de la capa interna del átomo son expulsados, creando espacios vacantes.

La cascada de electrones que atraviesa la capa del núcleo el núcleo llena los espacios vacíos. Los electrones en las capas exteriores tienen altos niveles de energía que los electrones en las capas interiores, y los electrones de las capas externas emiten energía en forma de rayos X el cual atraviesa el núcleo.

El nuevo arreglo de los electrones resulta en la emisión de rayos X con las características del átomo bombardeado. La emisión de rayos X, de esta manera, se denomina Fluorescencia de Rayos X (XRF). Un análisis cualitativo es realizado observando la energía del rayo X característico. Un análisis cuantitativo es realizado a través de la medición de la intensidad del los rayos X de acuerdo a la longitud de onda característica para un metal partícular.

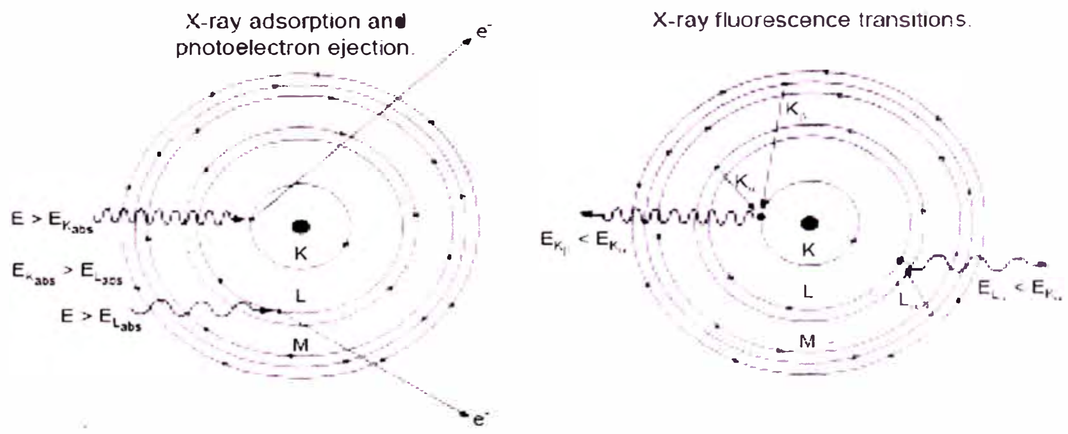


Figura 3.21 – Diagrama esquemático del proceso XRF

Figura 3.22 – Aplicación del Equipo Analizador de Aleación

CAPITULO 4

METODOLOGÍA DE EVALUACIÓN DE LA CONDICIÓN OPERATIVA

La metodología de evaluación de la condición operativa se va desarrollar teniendo en cuenta el siguiente proceso:

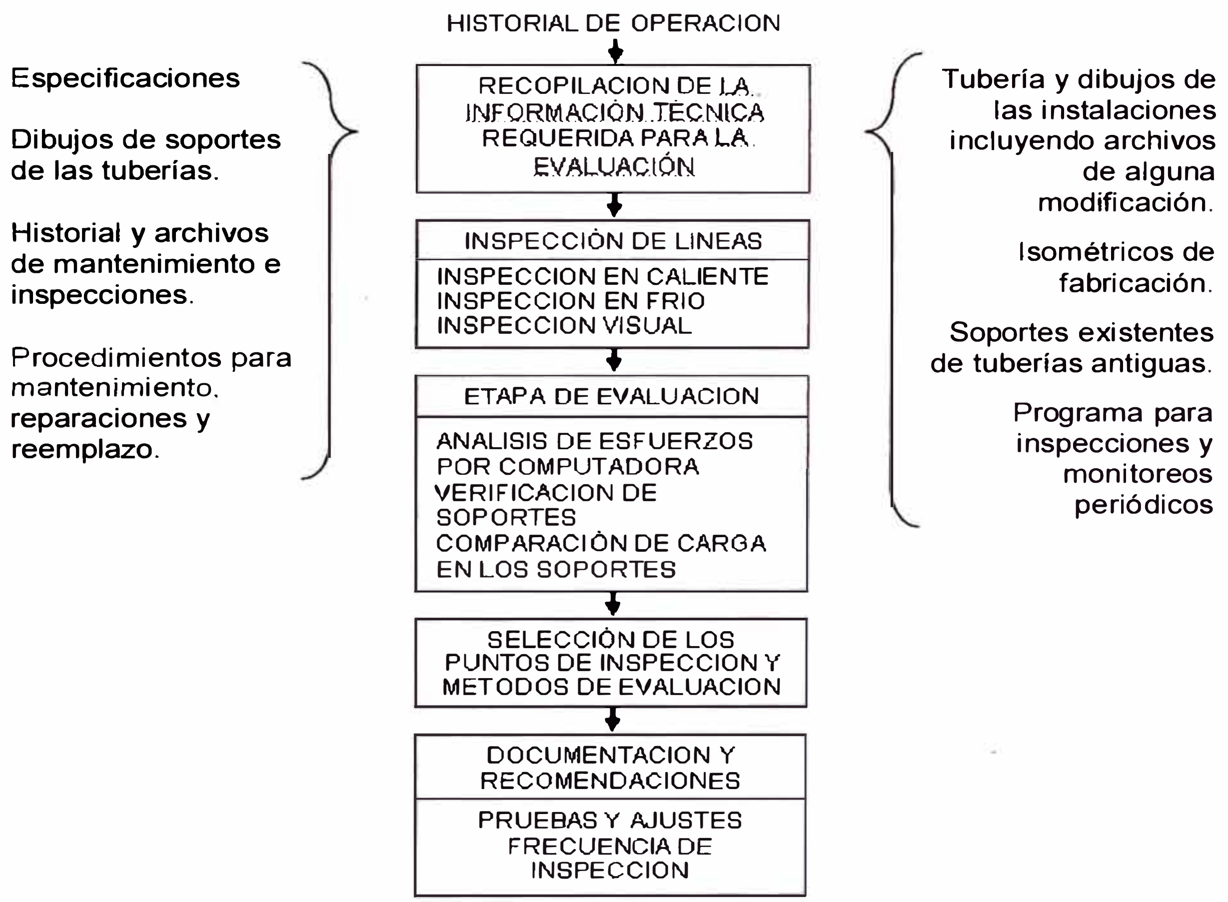


Figura 4.1 – Programa de extensión de vida:

Inspección y evaluación de tubería de vapor principal

4.1 RECOPILACIÓN DE LA INFORMACIÓN TÉCNICA REQUERIDA PARA LA EVALUACIÓN.

Esta etapa en la evaluación de la condición de tubería crítica es para recoger información específica relevante a los sistemas de tuberías. Estos datos incluyen:

- Dibujos de tubería y arreglo general.
- Isométricos de construcción (dibujos de fabricación).
- Especificaciones que cubren el material de la tubería, diámetro externo, espesor de tubería, material, etc.
- Cálculos de esfuerzos y soportes de las tuberías tal como fue diseñado.
- Dibujos de diseño de los soportes de la tubería, incluyendo detalles, ubicación y lista de materiales.
- Archivos de cualquier modificación de la tubería.
- Procedimientos de mantenimiento de las tuberías y sus soportes.
- Datos históricos tales como el historial de rendimiento, datos del modo del ciclo de planta, resultados de los ensayos no destructivos (NDE), etc.
- Cargas permisibles para las conexiones del caldero y turbina.

Estos datos son usados para determinar la cantidad de información requerida para que sea obtenida durante la fase del reconocimiento de la línea y la extensión del análisis puede ser requerida durante la fase de evaluación.

El historial de operación de la planta y los actuales modos de operación para cada sistema debe ser revisado. Las áreas de interés incluyen: El ciclo de operación del sistema, cargas de fluidos transientes que puedan haber ocurrido, modos de operación térmico, y cualquier observación de vibraciones excesivas.

En adición, el historial de mantenimiento es revisado constantemente y debe consultarse con el personal de la planta para identificar los componentes de las tuberías y los soportes que hayan sido reemplazados debido a la falla. El tipo de falla, método de reparación, acción correctiva, y probable causa son revisados para determinar si algún modelo exista que pueda justificar la inclusión del componente en la fase de inspección del programa.

4.2 INSPECCIÓN DE LAS TUBERÍAS DE VAPOR PRINCIPAL.

La fase de reconocimiento consiste en la inspección de las tuberías por ingenieros experimentados en el área de esfuerzos en tuberías y soportes. La información obtenida durante la fase de revisión de documentos es usada para establecer los requerimientos para una exitosa fase de reconocimiento. os datos del cálculo de esfuerzos, dibujos de los soportes de las tuberías, y dibujos de las tuberías son revisados durante el reconocimiento de las líneas.

a) El primer reconocimiento debe ser realizado durante la operación de la planta cuando el sistema de tuberías esta caliente, esta fase de reconocimiento de la tubería establece la condición “en operación” del sistema de tuberías

b) El segundo reconocimiento durante la parada de la planta cuando la tubería esta fría.

El reconocimiento de la línea consiste en una inspección visual del conjunto del sistema de tuberías y estructura soporte. El movimiento de la tubería es registrado revisando los indicadores de desplazamiento de los soportes y el golpe de los amortiguadores así como las marcas realizadas a los componentes de las tuberías con respecto a los datos de los puntos de referencia fijos. Evidencia de daños al aislamiento, interferencia con o desde otras tuberías o equipos, vibración y condición general de los soportes, barras de suspensión, guías, anclajes, accesorios de acero, entre otros son archivados. El reconocimiento es documentado con dibujos de reconocimiento de las tuberías y las hojas de inspección de los soportes, los cuales son usados como datos de la fase de evaluación.

Los problemas típicos identificados durante el reconocimiento de las tuberías incluyen:

- Rajaduras en la soldadura
- Deformación en los cabezales de vapor.
- Evidencia de golpe de vapor y golpe de agua.
- Válvula rotas.
- Vibración excesiva.
- Excesivo pandeo en la tubería.
- Daño a las barras de soporte constante.
- Mal funcionamiento de los aisladores.

4.3 ETAPA DE EVALUACIÓN

La fase de evaluación integra la información obtenida durante el reconocimiento de la línea el cual define la condición "en operación" y el esfuerzo de la revisión de los documentos el cual define la condición "según diseño". Si existen desviaciones significativas de los cálculos "según diseño" o si ningún cálculo esta disponible, entonces los sistemas de tuberías son analizados de acuerdo con el Código ASME/ANSI B31.1 para tuberías de potencia. El análisis resultante es denominado el análisis de esfuerzos "en operación". Las cargas tales como la presión interna, expansión térmica, y el peso del aislamiento de la tubería y sus contenidos están incluidas en el análisis. Las fuerzas de descarga en las válvulas de alivio son determinadas e incluidas apropiadamente. Las cargas tales como el golpe de vapor o el golpe de agua son analizadas si hubiera evidencia de que hayan ocurrido en el pasado. La condición de los soportes registrados durante el reconocimiento de la línea es incluida en el análisis. Ejemplos de esto incluyen: soportes rotos o dañados, interferencias, resortes deformados o soportes rotos.

Los resultados del análisis de esfuerzos de la tubería "en operación" proveen cargas en los equipos, cargas en los soportes, y niveles de esfuerzos a través del sistema de tuberías. Las cargas en los soportes son usadas para evaluar la integridad de los soportes de la tubería existente y los soportes de resorte. Los niveles calculados de esfuerzo son comparados con los niveles permisibles del código ASME B31.1 para tuberías de potencia. En casos donde los niveles de esfuerzo excedan los niveles permisibles de código y/o para mejorar condiciones de problemas crónicos (como la deflexión de tuberías), se determinan modificaciones al actual sistema de soportes de tubería. Los resultados del análisis de esfuerzos son denominados análisis de esfuerzos "en operación modificado".

4.4 SELECCIÓN DE LAS UBICACIONES Y MÉTODOS DE INSPECCIÓN.

En la realización del análisis de esfuerzos se identifican ubicaciones para exámenes no destructivos (NDE). Las ubicaciones típicamente representan áreas de alta concentración de esfuerzos tales como reductores, codos, conexiones de ramales, aletas soporte, o conexiones Y. El objetivo de los NDE es caracterizar la condición actual del material de las tuberías sin dañar o destruir algún componente. La inspección es orientada a encontrar discontinuidades o degradación que puede haberse desarrollado durante la vida operativa del componente o sistema a la fecha.

Varios métodos NDE pueden ser aplicables a un material en particular. El método menos costoso que puede detectar la degradación previsible o rajadura es seleccionado. Las opciones de los ensayos no destructivos incluyen:

- Inspección visual
- Inspección con partículas magnéticas
- Inspección ultrasónica
- Radiografía
- Réplica metalográfica
- Análisis de aleación

Un orden de preferencia para los diferentes métodos puede ser desarrollado basado en los siguientes criterios:

- Ubicación y tipo de defecto a ser detectado (rajaduras, porosidad, escoria, entre otros).
- Tamaño relevante del defecto a ser detectado.
- Composición del material (magnético o no magnético, metálico o no
- metálico).
- Espesor (dimensiones actuales).
- Geometría (simétrica, compleja).
- Tamaño (pequeña o larga).
- Condición del material (tratado en caliente, grueso o de grano fino).
- Método de fabricación (fundido, forjado, soldado, rolado, etc.)
- Condición de la superficie (rugoso, plateado, pulido, pintado, etc.)
- Resultados de las pruebas (archivos permanentes).
- Costo y tiempo para realizar las pruebas.

Determinados los puntos de inspección y las técnicas NDE seleccionadas, la frecuencia de inspección debe ser determinada dependiendo de los siguientes factores:

- Tipo de unidad.
- Historial de falla de la tubería.
- Modo de falla más común.
- Importancia de la unidad a la utilidad del sistema.

Todos estos factores pueden conducir hacia muchas posibilidades para la frecuencia de inspección. Las áreas problema identificadas dentro de cada sistema que no hayan sido corregidos mediante el modelo “en operación modificado” entonces deben ser inspeccionados después de cada evento de parada de la turbina o durante cada parada programada por lo menos una vez por año.

Luego que todas las mejoras sean implementadas, el sistema de tuberías y soportes deben ser inspeccionados entre cada 3 a 5 años o después de cada parada por sobrecarga. Si la inspección inicial del sistema arroja discontinuidades o defectos puede establecerse un compromiso de vida del componente, entonces debe recomendarse un programa de inspección más frecuente o realizar determinaciones de la vida del componente mediante aproximaciones del mecanismo de fractura existente (dependiendo de la naturaleza del defecto). La Tabla N°1 provee recomendaciones genéricas para inspecciones en tuberías de alta potencia.

RECOMENDACIONES GENERALES PARA EVALUACIÓN DE TUBERÍAS DE ALTA POTENCIA		
Tipo	Tubería de Alta Temperatura (>900°F)	Otras Tuberías de Alta Energia
Soldaduras longitudinales	100% V, MT, UT	100% V, MT, UT
Soldaduras circunferenciales	100% V, MT, UT	Ninguno
Sistemas de ramales (soldadas o en longitud)	100% V, MT	100% V, MT
Conexiones miscelaneas soldadas	V, MT	100% V, MT
Conexiones estructurales	100% V, MT	100% V, MT
Replicas	No menos de 2 en zonas de alto esfuerzo	Ninguno
Analizador de aleación	100% de piezas expuestas	Ninguno
Mediciones circunferenciales	2 por cada tamaño de tubería	Ninguno
Espesor de tubería recta	4 lecturas a 90° en cada longitud	4 lecturas a 90° en cada longitud
Espesor de tubería curva	4 lecturas a 90°, en la mitad y descarga	4 lecturas a 90°, en la mitad y descarga
Espesor en la descarga de válvulas, tees, Y'es, laterales	4 lecturas a 90°, en la mitad y 12" despues de la descarga	4 lecturas a 90°, en la mitad y 12" despues de la descarga
Espesor de tees, Y'es y laterales	Lecturas en áreas de impacto para detectar pérdida de material	Lecturas en áreas de impacto para detectar pérdida de material

Tabla N°4.1

4.5 EVALUACIÓN POR DISPONIBILIDAD PARA EL SERVICIO SEGÚN NORMA API RP 579.

Cada sección del API RP 579 tiene un procedimiento de evaluación por disponibilidad para el servicio que consiste de las siguientes partes:

1. General
2. Aplicabilidad y limitaciones de los procedimientos para evaluación por disponibilidad para el servicio.
3. Requerimientos
   3.1. Datos originales de diseño del equipo
   3.2. Historial de operación y mantenimiento

3.3. Datos y mediciones requeridas para la evaluación por disponibilidad para el servicio.

3.4. Recomendaciones para técnicas de inspección y requerimientos dimensionales.

4. Técnicas de evaluación y criterio de aceptación
5. Evaluación de la vida remanente
6. Opciones de reparación
7. Monitoreo en servicio
8. Documentación
9. Referencias
10. Tablas y figuras
11. Problemas ejemplo

El procedimiento del API RP 579 emplea un factor de esfuerzo remanente (RSF), el cual está definido como sigue:

$$RSF = \frac{L_{DC}}{L_{UC}}$$

L_{DC}: Límite del esfuerzo de rotura de un componente dañado

L_{UC}: Límite del esfuerzo de rotura de un componente nuevo.

El procedimiento también incluye recipientes a presión, tuberías y tanques a ser evaluados. Para reevaluar recipientes y tuberías a presión, las siguientes expresiones aplican:

$$MAWP_r = MAWP\left(\frac{RSF}{RSF_a}\right) \text{ for RSF} < RSF_a$$

$$MAWP_r = MAWP \text{ for RSF} \geq RSF_a$$

$MAWP_r$: Presión reducida máxima permisible de trabajo

MAWP: Presión máxima permisible de trabajo

RSF_a: Factor de esfuerzo remanente permisible

El API RP 579 usa aproximación de esfuerzo local para evaluar pérdida de material y discontinuidades por corrosión. La aproximación esta basada en el área efectiva de la sección transversal de la discontinuidad y el esfuerzo de flujo a la tensión del material. Esto ha sido establecido basado en los resultados de extensivos FEA y numerosas pruebas de explosión a toda escala de recipientes a presión y tuberías.

El API RP 579 emplea estudios de fractura mecánica para evaluar discontinuidades tales como las fisuras. Muchas de estas, con un diagrama general de evaluación de la falla (FAD) es usado para computar la proporción de resistencia (K_r) y la proporción de carga (L_r) para un componente con una discontinuidad tal como una fisura. K_r es la proporción del factor de intensidad del esfuerzo elástico lineal (K_I) para la resistencia a la fractura del material (K_{MAT}), mientras L_r es la proporción del esfuerzo de referencia (σ_{ref}) al esfuerzo de fluencia del material (σ_{ys}). Para una discontinuidad y carga dadas, el valor de K_r como una función de L_r es graficado en la FAD. Ninguna falla es predecida para puntos debajo de la línea de la evaluación de la falla, en donde la falla es aproximadamente cercana a ocurrir para puntos en o debajo de la línea de evaluación. Los cálculos son repetidos en otras condiciones para encontrar donde se ubican con respecto a la envolvente o para determinar las condiciones críticas (un punto en la línea de falla) en el cual la falla se predice que esta próxima a ocurrir.

El valor calculado de vida remanente (RL) es un valor que es usado para establecer el intervalo de inspección y por lo tanto el intervalo de operación segura hasta la siguiente inspección.

La presión máxima permisible trabajo (MAWP) puede ser estimada basada en la corrosión permisible, es decir RL puede ser estimada en el espesor de pared.

Para una pérdida general de material, la corrosión efectiva permisible (CA_e) es estimada como sigue:

$$CA_e = t_{perdido} + CR \times tiempo$$

$t_{perdido}$: Cantidad de espesor de material perdido hasta la fecha de la inspección

CR: Velocidad de corrosión futura

tiempo: Tiempo de operación futura.

Alternativamente, RL es estimado según:

$$RL = \frac{(t_{am} - K * t_{min})}{CR}$$

t_{am}: Espesor promedio a la fecha de inspección

t_{min}: Espesor mínimo permisible

K: Factor que es igual a 1 para evaluaciones de nivel 1 y RSF_a para evaluación de nivel 2.

La documentación de los resultados es una parte importante de la evaluación por disponibilidad para el servicio. API RP 579 señala las siguientes recomendaciones para la documentación:

1. Debe ser suficiente repetir la evaluación en una fecha posterior, por ejemplo 5 años después.
2. Debe incluir los datos de diseño original y el historial de mantenimiento y operación.
3. Debe incluir todos los datos de inspección que fueron usados en la evaluación.
4. Debe incluir las hipótesis y resultados analíticos, incluso:
    4.1 Versión, sección y nivel del API RP 579 y otros documentos empleados en el análisis.
    4.2 Diseño y condiciones de operación futuras (temperatura, presión, medio ambiente).
    4.3 Cálculo del espesor mínimo requerido o MAWP.
    4.4 Cálculo de la vida remanente e intervalos de inspección.
    4.5 Mitigación y recomendaciones de monitoreo que son condiciones para operación continua.
5. Todos los documentos deben ser almacenados con los archivos de inspección.

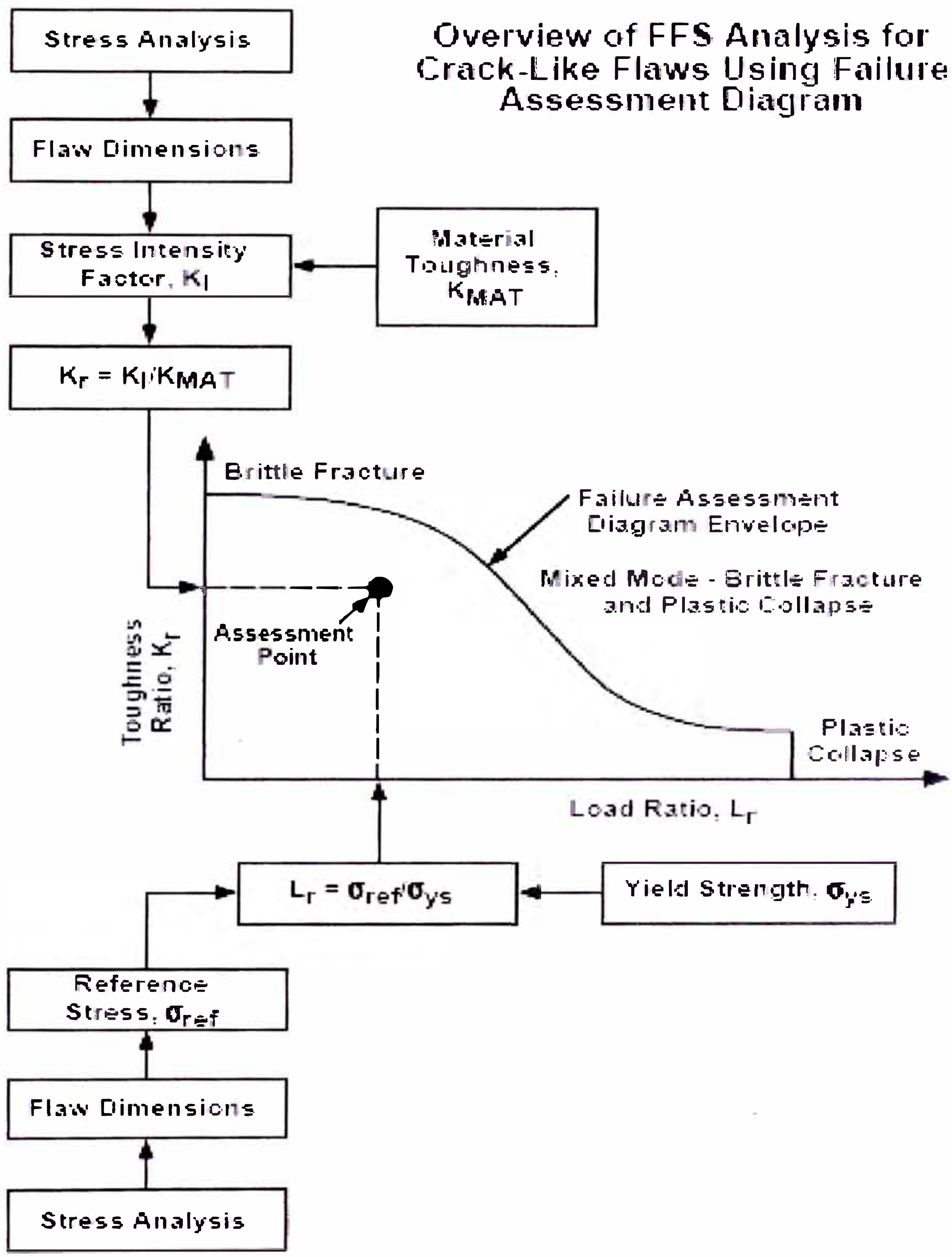


Figura 4.2 – Uso del diagrama general de evaluación de la falla

El procedimiento general de evaluación recomendado por API RP 579 consta de los siguientes pasos:

Paso 1: Identificar el tipo de discontinuidad y el mecanismo de daño del material.

Paso 2: Determinar la aplicabilidad y limitaciones del procedimiento de evaluación.

Paso 3: Definir los requerimientos de información. Esta información requerida para todas las evaluaciones son indicadas en la Sección 2, mientras esta información requerida para un tipo especifico de discontinuidad y mecanismo de daño están dados en la sección que contiene el procedimiento de evaluación para este tipo de discontinuidad y mecanismo de daño.

Paso 4: Aplicar las técnicas de evaluación y el criterio de aceptación.

Paso 5: Evaluar la vida remanente o el tamaño máximo de discontinuidad y establecer un intervalo de inspección basado en lo resultados de la evaluación.

Paso 6: Aplicar los métodos requeridos de remediación para la extensión posible y práctica.

Paso 7: Emplear los procedimientos de monitoreo en servicio cuando la vida remanente y los intervalos de inspección no puedan ser adecuadamente establecidos.

Paso 8: Archivar toda la información usada y decisiones hechas en los pasos 1 al 7, y almacenar la documentación con los archivos de inspección.

Estos 8 pasos deben ser incluidos en cada evaluación por disponibilidad para el servicio de una discontinuidad específica y combinación de componentes.

4.6 REPORTE FINAL

La fase final de la evaluación de sistemas críticos de tuberías de alta potencia es documentar las actividades del programa en un reporte el cual deberá resumir los resultados de la revisión de los documentos, reconocimiento de la línea, y análisis de esfuerzos. Las recomendaciones pueden incluir reparaciones, reemplazo o modificación de componentes de tuberías y soportes, cambios a los procedimientos de mantenimiento, y recomendaciones para realizar más exhaustivos ensayos no destructivos, pruebas metalúrgicas o para incrementar la frecuencia de inspección.

CAPITULO 5

VALORACIÓN ECONÓMICA

Las actividades que se han definido con el fin de realizar la evaluación de la condición de las tuberías de vapor principal de la planta, es decir para todos aquellos sistemas de tuberías cuyo mal funcionamiento puede tener consecuencias sobre la operación, otros equipos y seguridad del personal, requieren una inversión que se describe a continuación.

Se debe decidir si el personal de la planta o personal contratista estará a cargo de la evaluación de la tubería de vapor principal. Hay dos tareas principales: inspeccionar el sistema de vapor principal y realizar el mantenimiento del sistema de vapor. Se puede elegir entre contratar el servicio de inspección de una empresa externa, y tener personal propio que realice dicha tarea. Otra opción es contratar primeramente los servicios de una empresa contratista que realice la primera inspección, registrar toda la información pertinente, y realizar el entrenamiento del personal propio que continúe el programa de evaluaciones.

Las condiciones a favor y en contra de estas tres opciones se describen a continuación:

Empleando personal propio	Empleando una empresa contratista	Empleando una empresa contratista la primera vez y luego realizar estas inspecciones con personal propio.
Condiciones a favor: - El personal propio puede convertirse en experto en la evaluación del sistema de tuberias si se con lo cual podrá resolver problemas complejos, problemas de varios años tales como daños en turbinas, intercambiadores de calor, etc. - La supervisión tiene el control de cómo y cuando son realizadas las inspecciones y el mantenimiento. - Los archivos y reportes son propios y mantenidos en las instalaciones. - El personal propio no puede inclinarse hacia algún fabricante de equipos. **Condiciones en contra:** - Se requiere entrenamiento costoso e inversión de tiempo. - Es dificultoso mantener el mismo personal durante mucho tiempo suficientemente para desarrollar las habilidades necesarias y covertirse en expertos en técnicas de ensayos no destructivos. - Falta de personal disponible debido a otras actividades más importantes. - La supervisión del personal a cargo del programa puede no tener experiencia en inspecciones de este tipo.	**Condiciones a favor:** - Costo fijo para inspecciones y/o mantenimiento - Inspecciones anuales que pueden ser facilmente programadas sin considerar la disponibilidad del personal de planta, sobretiempos, etc. - El personal debe ser experto en su campo. - El contratista debe entregar reportes. - El contratista debe indicar las razones por que los algunos soportes deben ser cambiados e indicar las características que deben tener los nuevos soportes. - Varios expertos desde una variedad de fabricantes y distribuidores pueden ayudar para resolver problemas complejos con el sistema de vapor. - En las instalaciones no se requiere mantener un grupo de expertos en ensayos no destructivos. **Condiciones en contra:** - El nivel de especialización varia ampliamente. - El contratista usualmente tiene el control de los datos y reportes. - Algunos contratistas son distribuidores de varios fabricantes de tuberías y equipos NDE y pueden actuar como representantes para adquirir repuestos.	**Condiciones a favor:** - La asistencia de expertos es fácilmente disponible. - Las instalaciones tienen más control debido al involucramiento de su personal propio. - Las propuestas de los contratistas pueden ser validadas por el personal propio. - El personal propio puede facilitar una facilitar una relación más productiva que pueda formarse entre las instalaciones y el contratista. - La credibilidad de las reparaciones propuestas es verificada mediante la cooperación entre el personal propio y los contratistas. - La inversión en entrenamiento está asegurada. Es improbable que se pueda reemplazar al personal propio y a los contratistas al mismo tiempo. El contratista puede facilmente entrenar a un nuevo personal propio y un personal propio puede facilmente fijar a un nuevo contratista. - Los archivos pueden ser registrados por el personal propio. **Condiciones en contra:** - Se requiere entrenamiento del personal propio.

Tabla N°5.1 – Evaluación de opciones para la contratación de mano de obra

Para la implementación de este programa empleando personal propio se presenta un presupuesto:

Item	Equipo requerido	Costo
1	Equipo de Partículas Magnéticas (MT)	$2,780.00
2	Equipo de Ultrasonido (UT)	$13,831.00
	Subtotal =	$16,611.00

Item	Requerimiento	Costo
1	Entrenamiento en Inspección Visual	$5,000.00
2	Consumibles para MT	$3,000.00
3	Entrenamiento en MT	$5,000.00
4	Cosumibles para UT	$2,500.00
5	Entrenamiento en UT	$5,000.00
	Subtotal =	$20,500.00

Total = $37,111.00

Tabla N°5.2

La evaluación económica sería la siguiente:

El área analizada tiene costo de US$/hora por las paradas no programadas
debido a una falla en el sistema de vapor principal

Costo de $ / hora por parada de la planta
=22MW * $ 8.2/MW-hr * 2 + 66MW * $ 24.6/MW-hr * 2 = $3,608.00 /hr

Las horas que pararía la planta en caso de falla de la tubería de vapor
principal debido a las reparaciones requeridas

Horas de parada 8 hr

Promedio horas parada 8 hr X $3,608.00

Promedio de Costo de parada $28,864.00

EVALUACION ECONOMICA

Análisis Financiero

Nombre del proyecto : ADQUISICIÓN DE EQUIPOS PARA EVALUACION DE LA CONDICIÓN

		FISCAL YEAR			
		2006	2007	2008	2009
		1	2	3	4
1.- INGRESOS / AHORROS DE COSTOS					
		28.864			
TOTAL DE INGRESOS / AHORROS		28.864	0	0	0
3.- TOTAL NETO ANTES IMPUESTOS		28,864	0	0	0
4.- DEPRECIACION		(4.153)	(4.153)	(4.153)	(4.153)
5.- BENEFICIO NETO TOTAL		24.711	(4.153)	(4.153)	(4.153)
6.- IMPUESTOS 30%		7,413	0	0	0
7.- IMPORTE INVERSION PROYECT.	(16,611)				
8.- FLUJO DE CAJA NETO	(16,611)	21,451	0	0	0
FLUJO NETO NOMINAL	(16,611)	21,451	0	0	0
FLUJO NETO REAL @Inflacion 3%	(16,611)	20,826	0	0	0

Indicadores Flujo Nominal	TIR IRR =	29.1%
Indicadores Flujo Real	TIR IRR =	25.4%
Payback periodo anual	AÑOS =	0.80

VAN NPV = $ 2.288

VARIABLES

	Valor	Unidades	
DEPRECIACION (años)	4	Años	(Máximo a utilizar 7 años)
INFLACION ANUAL (%)	3.0%	Porcentaje	
TASA DE DESCUENTO (%)	13.5%	Porcentaje	

Hecho por:
Nombre : LUIS ALCA

Revisado por:
Nombres :

NOTAS: - Los ingresos de dinero deben considerarse positivos y los egresos como negativos

Tabla N°5.3 – Evaluación económica

La cotización de servicios de una empresa contratista para realizar los ensayos requeridos de acuerdo a la metodología propuesta sería la siguiente:

COTIZACION DE SERVICIOS	
CLIENTE:	
CONTACTO:	
REFERENCIA: SERVICIO DE ENSAYOS NO DESTRUCTIVOS EN CT ILO 1 POWER PLANT	
Código / Standard a cumplir: ASME V Artículo 4 / Scope of Main Steam Piping Inspection	
SERVICIO	
Inspección Visual (VT)	
Inspección Partículas Magnéticas (MT)	
Inspección Ultrasonica (UT)	
Supervisores NDT	
RATES / TARIFAS	
Por el servicio de Ensayos No Destructivos en CT ILO 1 POWER PLANT	
Unidad N°1, 2, 3 & 4 todo según especificaciones según detalle:	
1MS - Main Steam, Unit 1	
2MS - Main Steam, Unit 2	
3MS - Main Steam, Unit 3	
4MS - Main Steam, Unit 4	
1HDR2 - U1&U2 Common HDR to Waste BLR	
3HDR4 - U3&U4 Common HDR to Smelter	
PRECIO DEL SERVICIO	**$ 7,750.00**
(Este precio incluye todas las pruebas solicitadas para la evaluación de la condición	
excepto las de análisis de aleación)	
Tiempo Estimado: 15 días con informe incluido	
CONDICIONES:	
1. En estos precios no está incluido el IGV.	
2. Enviar orden de servicio	
3. Sellos ACCP ASNT Level III, incluidos en el servicio.	
4. Forma de pago: A 30 días fecha factura	
5. Será por vuestra cuenta la alimentación de nuestro personal en vuestra planta.	
6. Será por vuestra cuenta la provisión de andamios, personal para armado,	
desarmado y movilización de los mismos o plataforma móvil con el operador.	
7. Se incluye 10 placas RT (cada placa adicional a $50.00)	
Alquiler del equipo para análisis de aleación (Stone & Webster)	**$ 6,000.00**
Total	**$ 13,750.00**

Tabla N°5.4

CONCLUSIONES

- Se recomienda la aplicación de esta metodología debido a que está basada en las prácticas actuales de evaluación de la condición de los sistemas de vapor principal en una Central Térmica con lo cual el operador de la planta tiene una valiosa información para operar sus equipos y evaluar un incremento del nivel de producción de la planta.
- La extensión de la vida de los sistemas de vapor principal en una central termoeléctrica puede realizarse de acuerdo a los requerimientos del código ASME B31.1 y norma API RP 579.
- Esta evaluación requiere una inversión de $16,611.00 para comprar equipos y realizar la evaluación de la condición de las tuberías de vapor principal es factible según a tasa del $TIR_{NOMINAL}$ = 29.1%@16.5, TIR_{REAL} = 25.4%@13.5, VAN = $2,288.00 y para un tiempo de depreciación de cuatro años. Además, se tendrá que realizar una inversión adicional en la contratación de una empresa de servicios, entrenamiento y consumibles los cuales suman en total $50,861.00 como inversión inicial para prevenir una pérdida de $28,864.00 debido a por lo menos 8 horas de parada de planta ocasionada por una falla en el sistema de vapor principal. Por lo tanto, es conveniente realizar la evaluación de la condición operativa de las tuberías de vapor principal empleando una empresa contratista la primera vez y luego realizar esta evaluación con personal propio.
- Es posible establecer una relación entre la condición de los elementos que componen un sistema de vapor principal con la frecuencia de

inspecciones recomendada para realizar la evaluación por disponibilidad para el servicio.

- La evaluación por disponibilidad para el servicio son evaluaciones cuantitativas de ingeniería que son realizadas para demostrar la integridad estructural de un componente en servicio que contiene una discontinuidad o daño. API RP 579 provee lineamientos basados además en la experiencia para conducir estas evaluaciones empleando metodologías específicamente preparadas. Los lineamientos provistos en esta práctica recomendada pueden ser usados para tomar decisiones de reparación y asegurarse de que los componentes a presión que tiene discontinuidades identificadas mediante la inspección puedan continuar bajo operación segura.

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  First Edition, January 2000

- DISEÑO DE TUBERÍAS PARA PLANTAS DE PROCESO

  Autor: Howard F. Rase

  Editorial Blume

  1ra edición española, 1973

- DESIGN OF PIPING SYSTEMS

  The M. W. Kellog Company

  John Wiley & Sons.

  2nd edition, 1956

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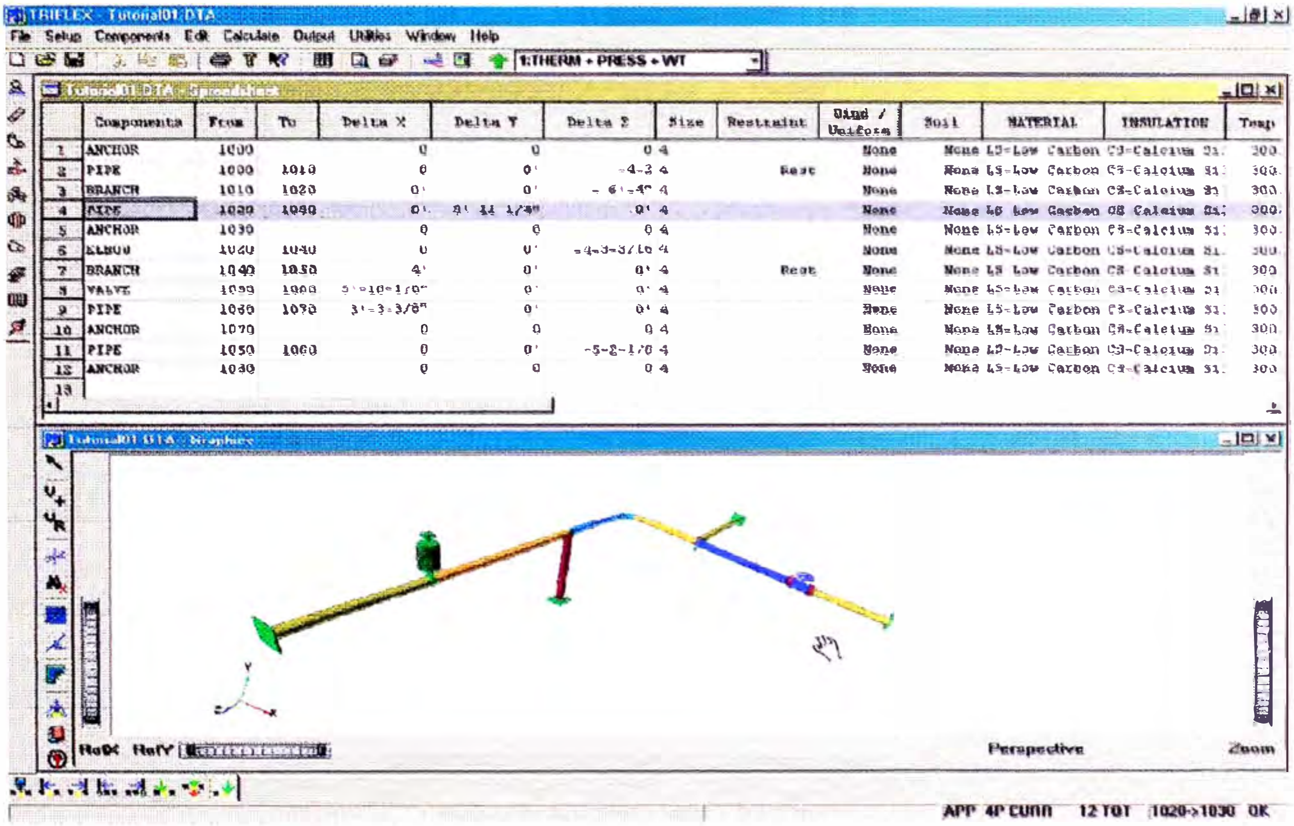
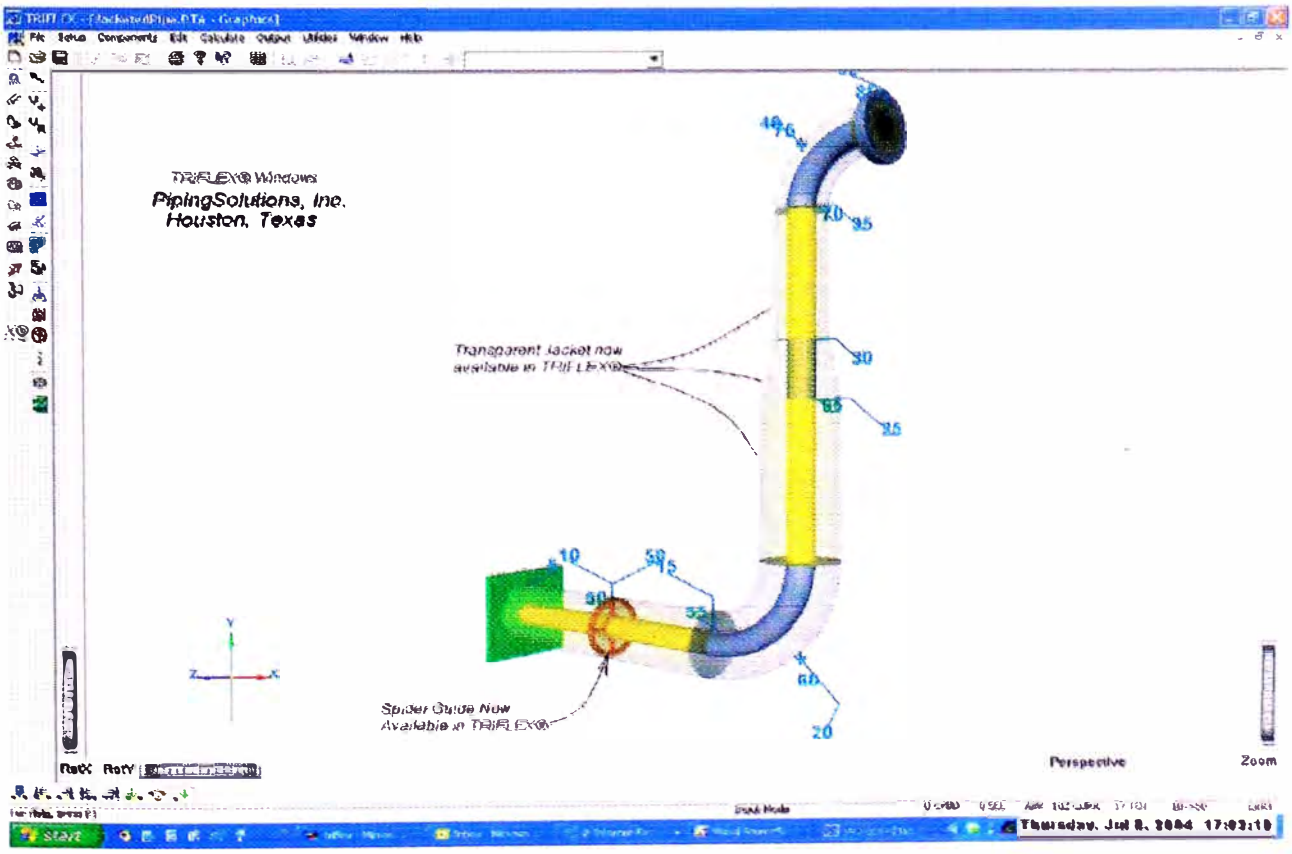
ANALISIS DE ESFUERZOS EN LA TUBERÍA DE VAPOR PRINCIPAL





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Pipe Stress Analysis and Design Program

Version: 08.05.00.07

Edition: Plus-Win

Developed and Maintained by

BENTLEY SYSTEMS, INCORPORATED
1600 Riviera Ave., Suite 300
Walnut Creek, CA 94596

R E S U L T   S U M M A R Y

Maximum sustained stress ratio

Point	:	4145
Stress psi	:	7541
Allowable psi	:	12740
Ratio	:	0.59
Load combination	:	GR + Max P

Maximum displacement stress ratio

Point	:	4170
Stress psi	:	21637
Allowable psi	:	21935
Ratio	:	0.99
Load combination	:	Max Range

* * * The system satisfies ASME B31.1 code requirements * * *
* * * for the selected options * * *

ATTACHMENT 5

```
ILO PERU - MAIN STEAM - AS-OPERATING R2
02/23/2006 ILO PERU POWER PLANT 1- 850# MAIN STEAM BENTLEY
02:09 PM   AS-OPERATING PIPE STRESS ANALYSIS R2    AutoPIPE+8.51 MODEL  PAGE   1
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 *********  **   **    **    **   **      **         **  **         **
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Pipe Stress Analysis and Design Program

Version: 08.05.01.11

Edition: Plus-Win

Developed and Maintained by

BENTLEY SYSTEMS, INCORPORATED
1600 Riviera Ave., Suite 300
Walnut Creek, CA 94596

ATTACHMENT 5

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--------------------------------------------------------------------------------
ILO PERU - MAIN STEAM - AS-OPERATING R2
02/23/2006 ILO PERU POWER PLANT 1- 850# MAIN STEAM BENTLEY
02:09 PM   AS-OPERATING PIPE STRESS ANALYSIS R2    AutoPIPE+8.51 RESULT PAGE 212
--------------------------------------------------------------------------------

                       R E S U L T   S U M M A R Y
                       ---------------------------


   Maximum sustained stress ratio

                          Point             : 7010
                          Stress    psi     : 8731
                          Allowable psi     : 12740
                          Ratio             : 0.69
                          Load combination  : GR + Max P

   Maximum displacement stress ratio

                          Point             : 6030F
                          Stress    psi     : 21553
                          Allowable psi     : 21935
                          Ratio             : 0.98
                          Load combination  : Amb to T1


   * * * The system satisfies ASME B31.1 code requirements * * *
   * * * for the selected options                          * * *
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ATTACHMENT 6


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        *                                 *******   **  *******   *******
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Pipe Stress Analysis and Design Program

Version: 08.05.01.11

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BENTLEY SYSTEMS, INCORPORATED
1600 Riviera Ave., Suite 300
Walnut Creek, CA 94596

ATTACHMENT 6

R E S U L T   S U M M A R Y

Maximum sustained stress ratio

Point	4145
Stress psi	7539
Allowable psi	12740
Ratio	0.59
Load combination	GR + Max P

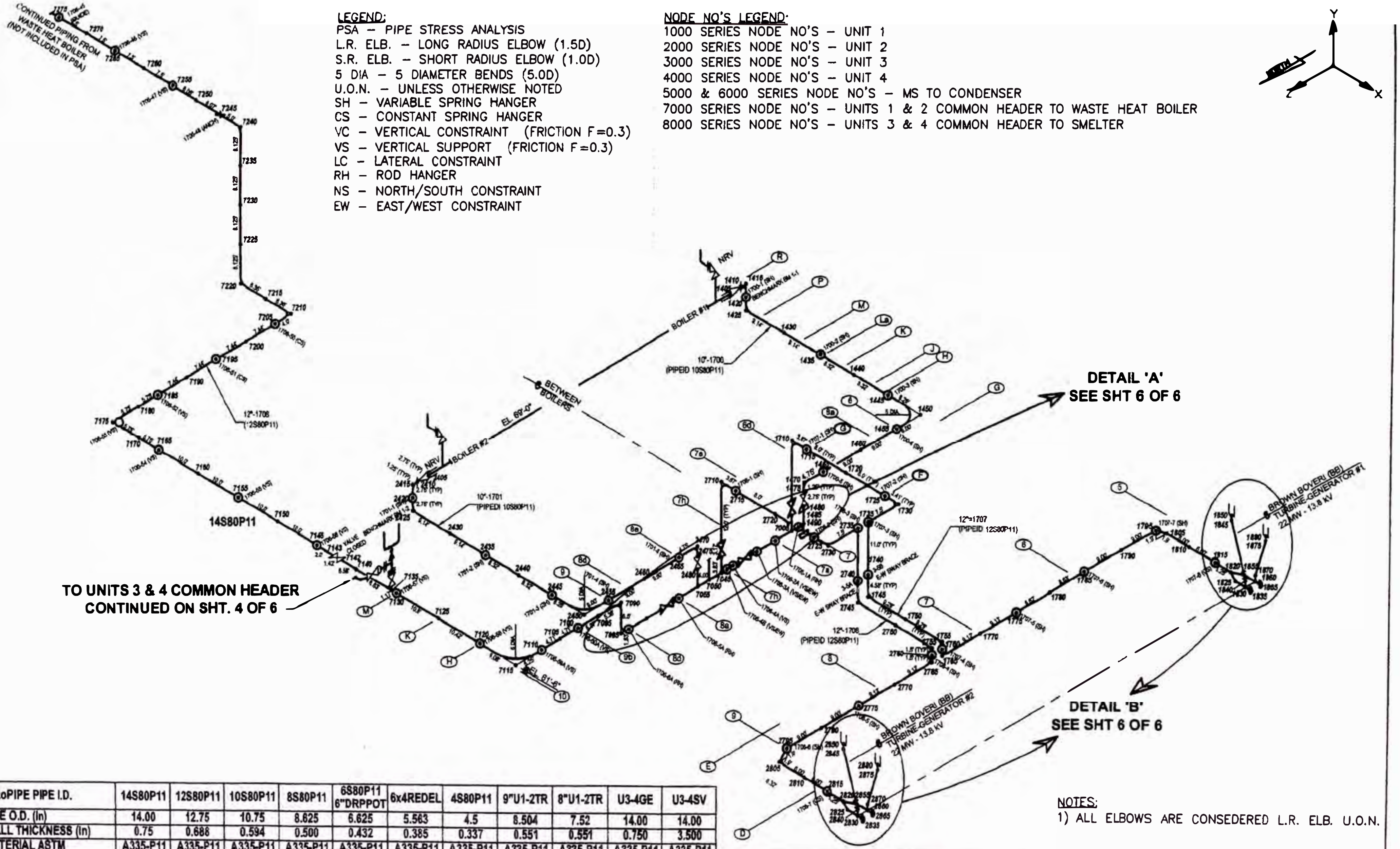
Maximum displacement stress ratio

Point	6030F
Stress psi	21875
Allowable psi	21935
Ratio	1.00
Load combination	Amb to T3

* * * The system satisfies ASME B31.i code requirements
* * * for the selected options * * *

LEGEND:
PSA – PIPE STRESS ANALYSIS
L.R. ELB. – LONG RADIUS ELBOW (1.5D)
S.R. ELB. – SHORT RADIUS ELBOW (1.0D)
5 DIA – 5 DIAMETER BENDS (5.0D)
U.O.N. – UNLESS OTHERWISE NOTED
SH – VARIABLE SPRING HANGER
CS – CONSTANT SPRING HANGER
VC – VERTICAL CONSTRAINT (FRICTION F=0.3)
VS – VERTICAL SUPPORT (FRICTION F=0.3)
LC – LATERAL CONSTRAINT
RH – ROD HANGER
NS – NORTH/SOUTH CONSTRAINT
EW – EAST/WEST CONSTRAINT

NODE NO'S LEGEND:
1000 SERIES NODE NO'S – UNIT 1
2000 SERIES NODE NO'S – UNIT 2
3000 SERIES NODE NO'S – UNIT 3
4000 SERIES NODE NO'S – UNIT 4
5000 & 6000 SERIES NODE NO'S – MS TO CONDENSER
7000 SERIES NODE NO'S – UNITS 1 & 2 COMMON HEADER TO WASTE HEAT BOILER
8000 SERIES NODE NO'S – UNITS 3 & 4 COMMON HEADER TO SMELTER



NOTES:
1) ALL ELBOWS ARE CONSEDERED L.R. ELB. U.O.N.

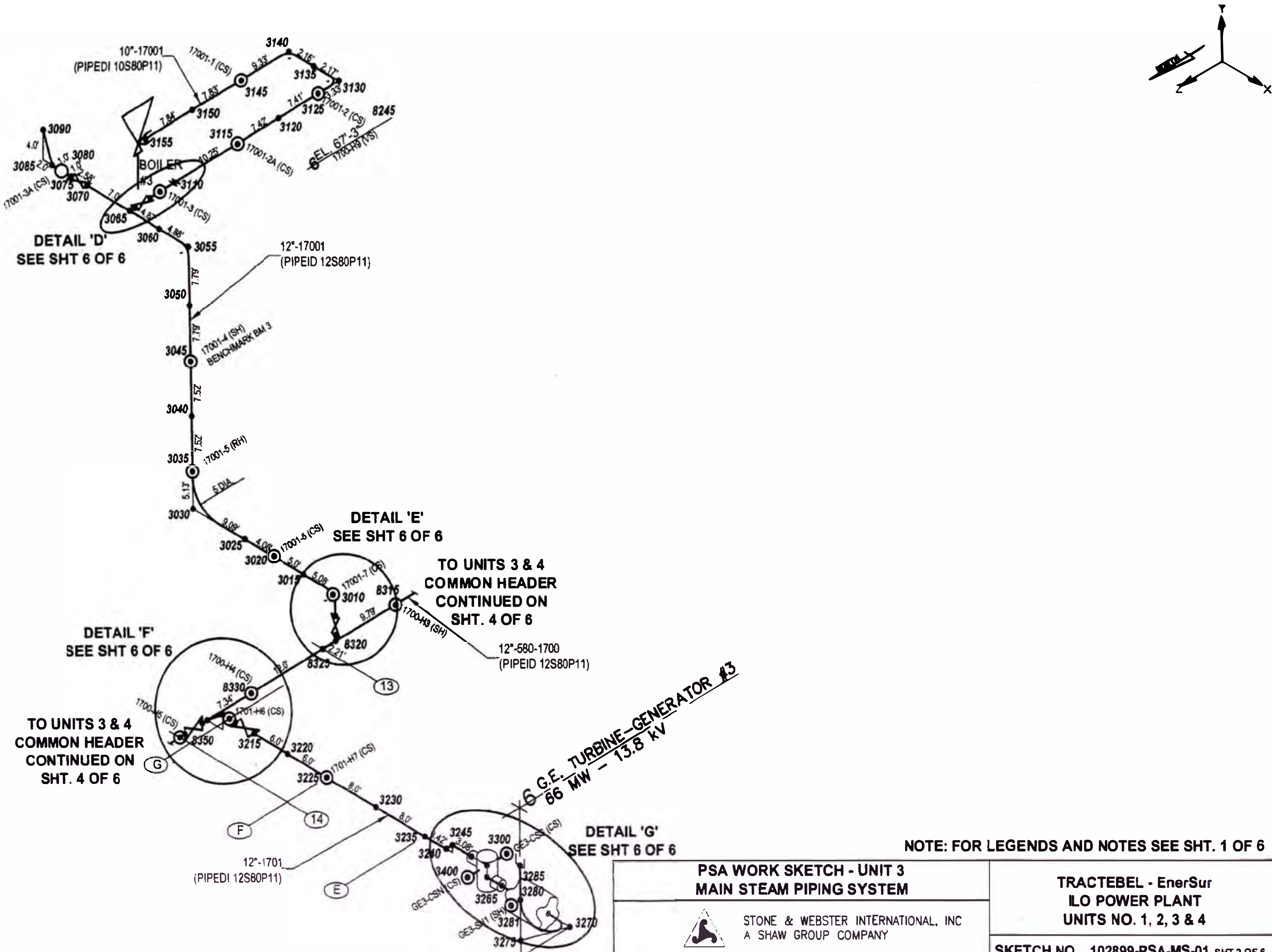
AutoPIPE PIPE I.D.	14S80P11	12S80P11	10S80P11	8S80P11	6S80P11 6"DRPPOT	6x4REDEL	4S80P11	9"U1-2TR	8"U1-2TR	U3-4GE	U3-4SV
PIPE O.D. (In)	14.00	12.75	10.75	8.625	6.625	5.563	4.5	8.504	7.52	14.00	14.00
WALL THICKNESS (In)	0.75	0.688	0.594	0.500	0.432	0.385	0.337	0.551	0.551	0.750	3.500
MATERIAL ASTM	A335-P11	A335-P11	A335-P11	A335-P11	A335-P11	A335-P11	A335-P11	A335-P11	A335-P11	A335-P11	A335-P11
MAX. OPER. TEMP (F°)	910	910	910	910	910	910	910	910	910	910	910
DESIGN PRESS. (psi)	865	865	865	865	865	865	865	865	865	865	865
COEFF. OF EXPAN. (In/ft)	0.0801	0.0801	0.0801	0.0801	0.0801	0.0801	0.0801	0.0801	0.0801	0.0801	0.0801
INSULATION THK. (In)	4.00	4.00	4.00	4.00	4.00	4.00	4.00	3.00	3.00	4.50	4.00
PIPE WEIGHT(lb/ft)	106.00	88.53	64.36	43.34	28.54	21.27	14.97	46.75	40.97	106.00	392.00
INSULATION WGT(lb/ft)	17.28	16.08	14.16	12.12	10.20	9.18	8.16	8.28	7.57	19.98	17.28
TOTAL WEIGHT (lb/ft)	123.28	104.61	78.52	55.46	38.74	30.45	23.13	55.03	48.54	125.98	409.28

PSA WORK SKETCH - UNITS 1 & 2
MAIN STEAM PPG & COMMON HEADER

STONE & WEBSTER INTERNATIONAL, INC
A SHAW GROUP COMPANY

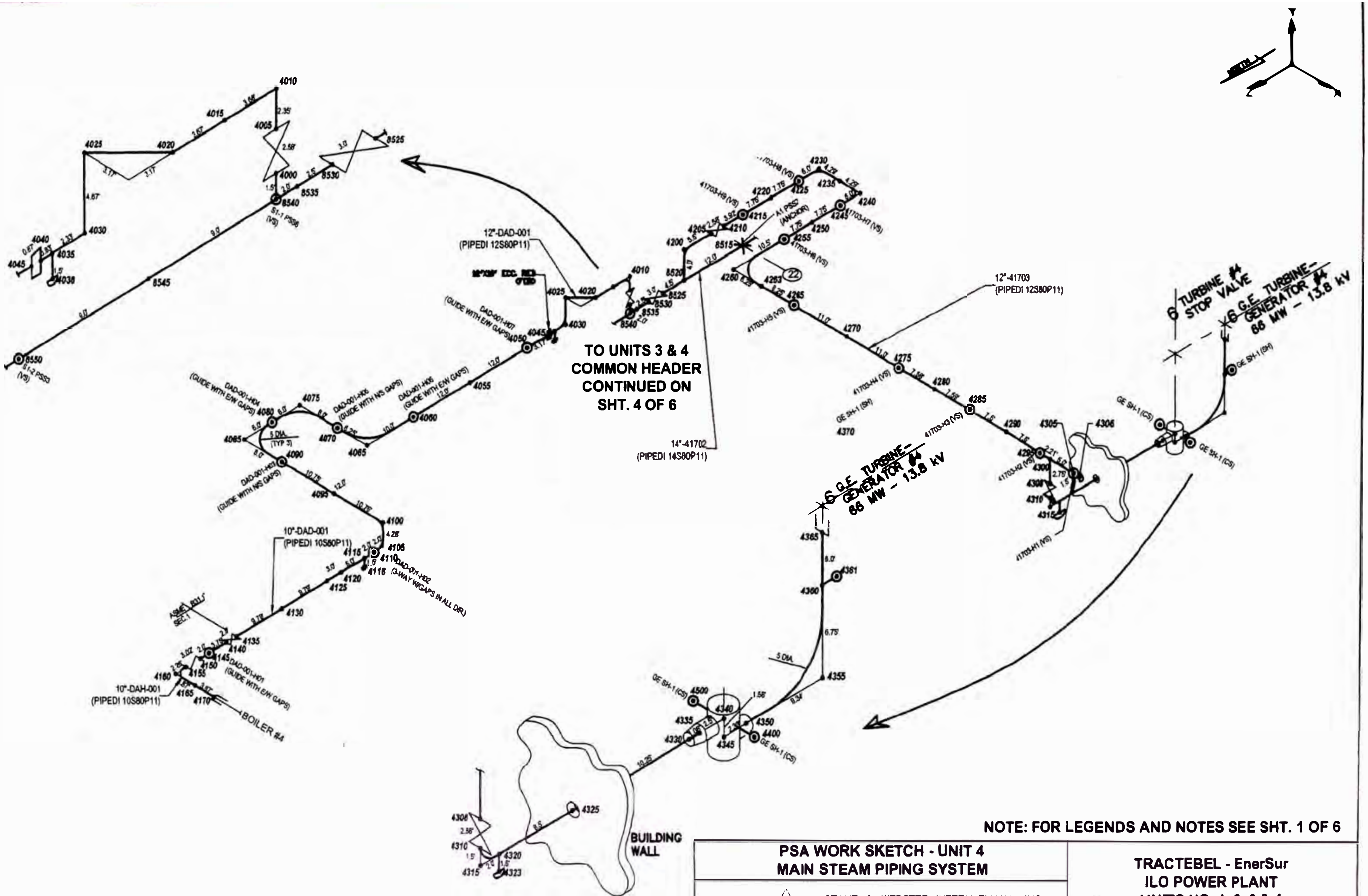
TRACTEBEL - EnerSur
ILO POWER PLANT
UNITS NO. 1, 2, 3 & 4

SKETCH NO. 102899-PSA-MS-01 SHT 1 OF6



NOTE: FOR LEGENDS AND NOTES SEE SHT. 1 OF 6

PSA WORK SKETCH - UNIT 3 MAIN STEAM PIPING SYSTEM	TRACTEBEL - EnerSur ILO POWER PLANT UNITS NO. 1, 2, 3 & 4
STONE & WEBSTER INTERNATIONAL, INC A SHAW GROUP COMPANY	SKETCH NO. 102899-PSA-MS-01 SHT 2 OF 6



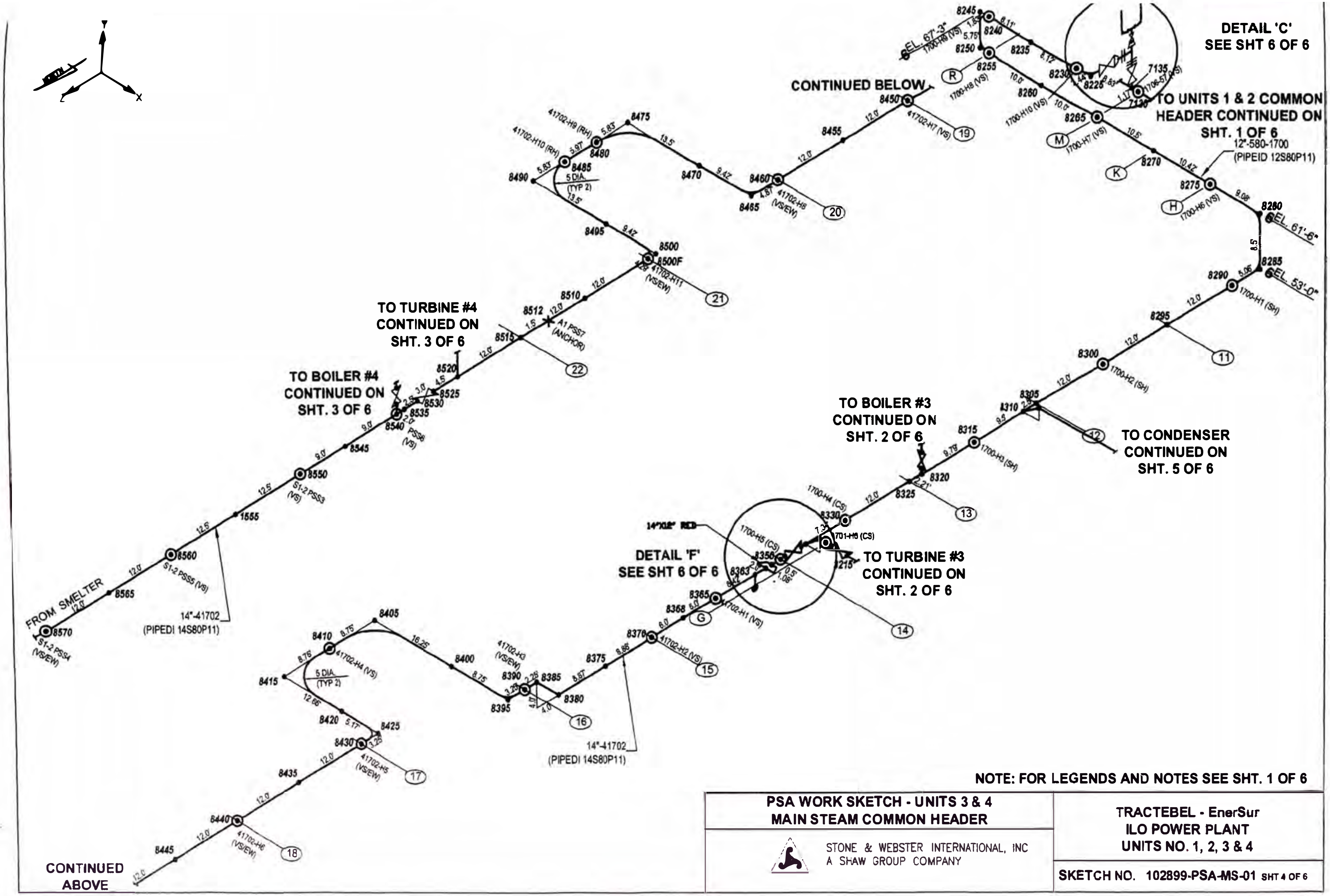
NOTE: FOR LEGENDS AND NOTES SEE SHT. 1 OF 6

PSA WORK SKETCH - UNIT 4
MAIN STEAM PIPING SYSTEM

STONE & WEBSTER INTERNATIONAL, INC
A SHAW GROUP COMPANY

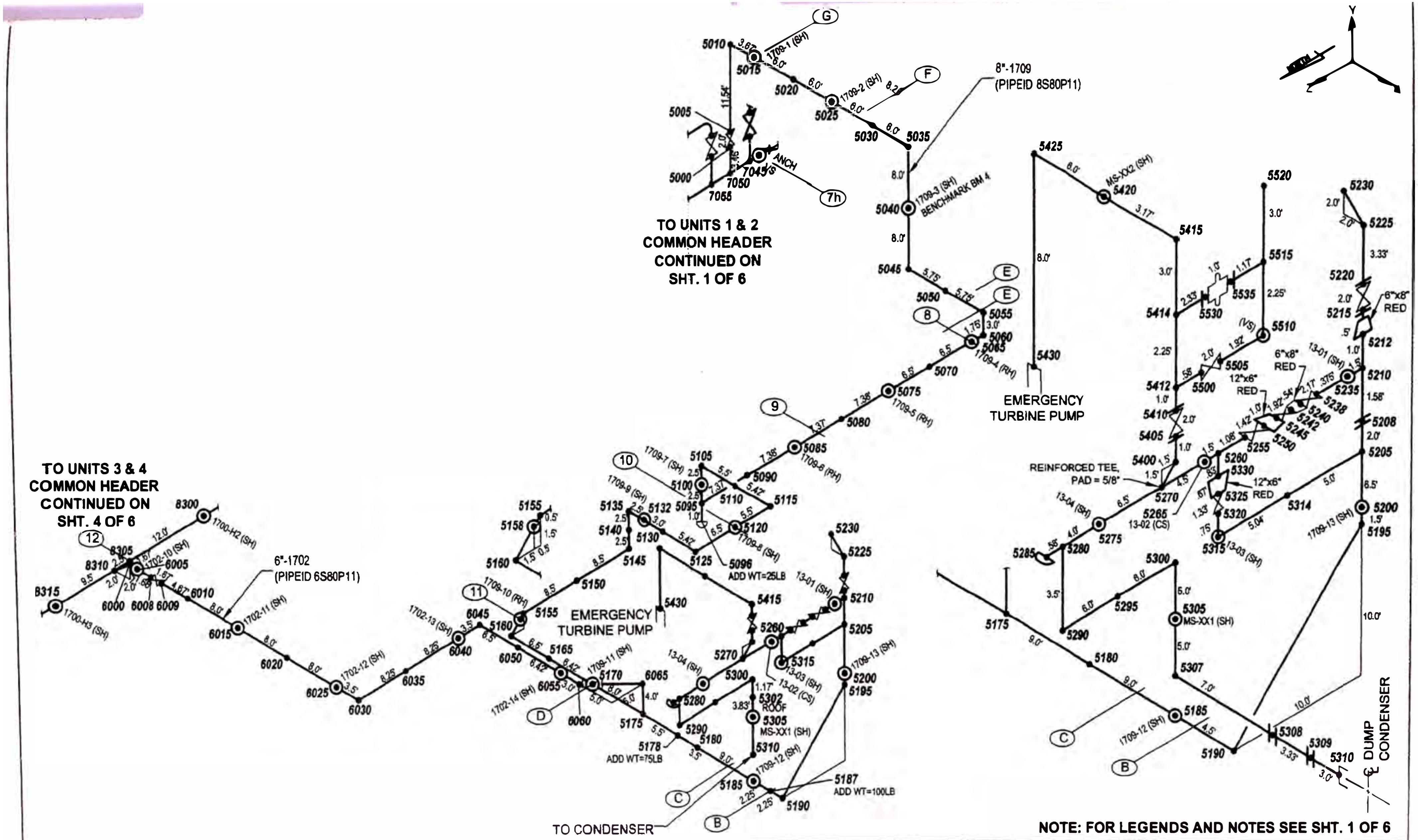
TRACTEBEL - EnerSur
ILO POWER PLANT
UNITS NO. 1, 2, 3 & 4

SKETCH NO. 102899-PSA-MS-01 SHT 3 OF 6



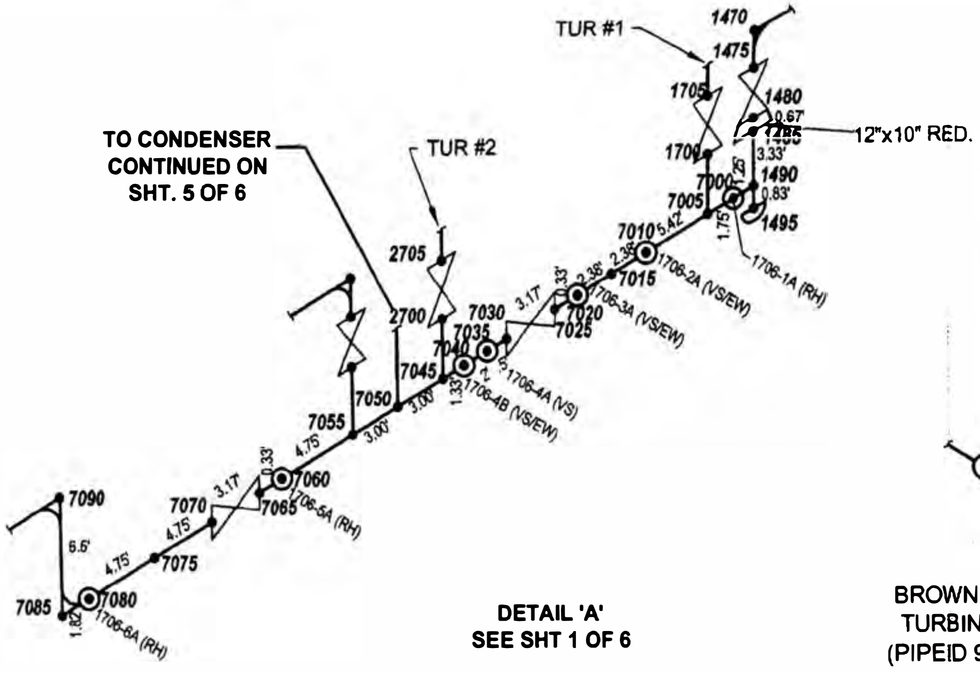
NOTE: FOR LEGENDS AND NOTES SEE SHT. 1 OF 6

PSA WORK SKETCH - UNITS 3 & 4 MAIN STEAM COMMON HEADER	TRACTEBEL - EnerSur ILO POWER PLANT UNITS NO. 1, 2, 3 & 4
STONE & WEBSTER INTERNATIONAL, INC A SHAW GROUP COMPANY	SKETCH NO. 102899-PSA-MS-01 SHT 4 OF 6



NOTE: FOR LEGENDS AND NOTES SEE SHT. 1 OF 6

PSA WORK SKETCH - UNITS 1, 2, 3 & 4 MAIN STEAM PIPING TO CONDENSER	TRACTEBEL - EnerSur ILO POWER PLANT UNITS NO. 1, 2, 3 & 4
STONE & WEBSTER INTERNATIONAL, INC A SHAW GROUP COMPANY	SKETCH NO. 102899-PSA-MS-01 SHT 5 OF 6



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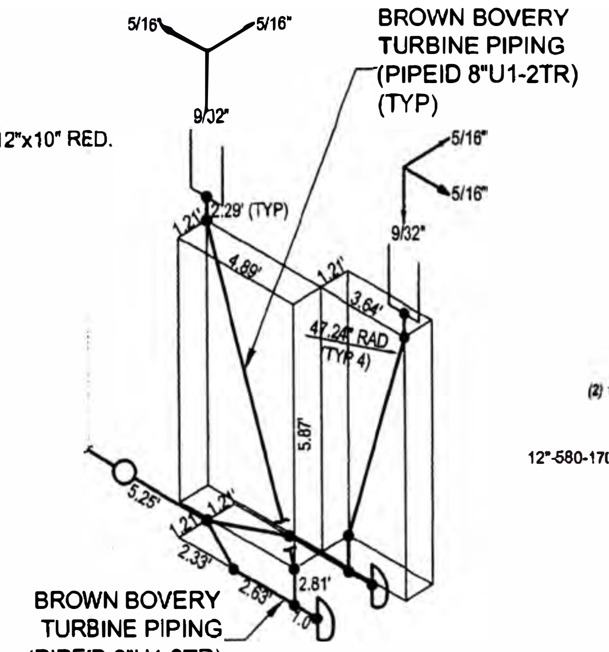

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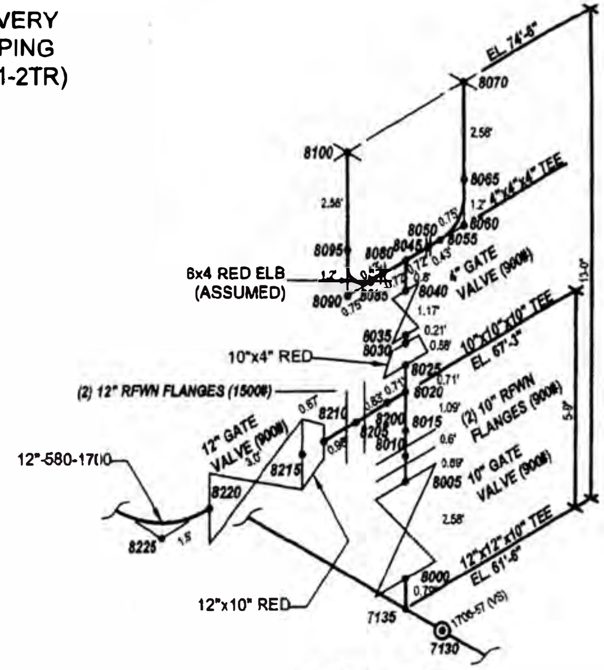

DETAIL 'C'
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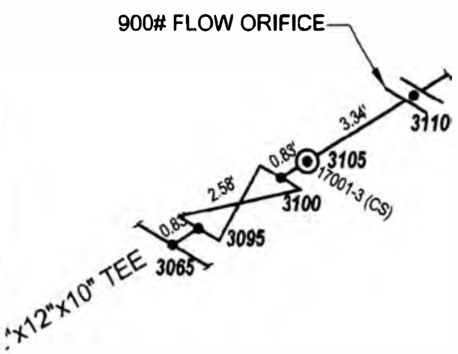

DETAIL 'D'
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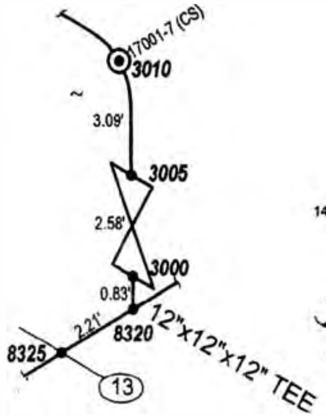

DETAIL 'E'
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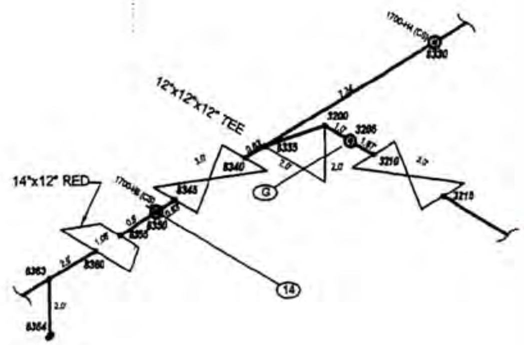
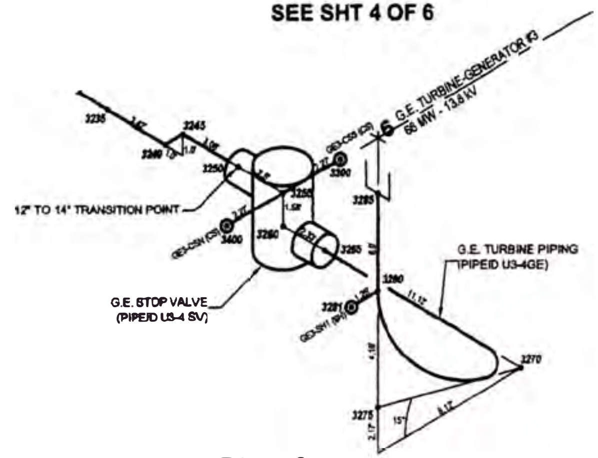


DETAIL 'F'
SEE SHT 2 OF 6



DETAIL 'G'
SEE SHT 2 OF 6

NOTE: FOR LEGENDS AND NOTES SEE SHT. 1 OF 6

PSA WORK SKETCH - UNITS 1, 2, 3 & 4 MAIN STEAM PIPING SYSTEMS - DETAILS	TRACTEBEL - EnerSur ILO POWER PLANT UNITS NO. 1, 2, 3 & 4
STONE & WEBSTER INTERNATIONAL, INC A SHAW GROUP COMPANY	SKETCH NO. 102899-PSA-MS-01 SHT 6 OF 6

NONMANDATORY APPENDIX V RECOMMENDED PRACTICE FOR OPERATION, MAINTENANCE, AND MODIFICATION OF POWER PIPING SYSTEMS[1]

FOREWORD

The B31.1 Power Piping Code prescribes minimum requirements for the construction of power and auxiliary service piping within the scope of para. 100.1. The Code, however, does not provide rules or other requirements for a determination of optimum system function, effective plant operations, or other measures necessary to assure the useful life of piping systems. These concerns are the responsibility of the designer and, after construction turnover, the Operating Company personnel responsible for plant activities.

Past experience has shown that a need exists for the definition of acceptable plant practices for achieving both reliable service and a predictable life in the operation of power piping systems. This nonmandatory Appendix is intended to serve that purpose. For this objective, Appendix V is structured in three parts that recognize and address the following basic concepts.

operation: the design of a piping system is based on specified service requirements and operating limitations. Subsequent operation within these defined limits is assumed and, for some systems, will be important for an acceptable service life.

maintenance: the design of a piping system assumes that reasonable maintenance and plant service will be provided. The lack of this support will, in some cases, introduce an increasing degree of piping system life uncertainty.

modifications: future modifications of a piping system or its operational functions are not assumed in original design unless specified. Modifications must not invalidate the integrity of a piping system design.

The practices in Appendix V are recommended for all plants and systems within the scope of the Power Piping Code, both for new construction and for existing plants in operation. An acceptable implementation of these or equivalent practices will be beneficial for new systems. The application of these practices is recommended for power piping systems in operating plants.

The recommended practices in this Appendix define minimum requirements for establishing a program to accommodate the basic considerations for piping system operation, maintenance, service, modification, and component replacement.

A record-keeping program is prescribed that can serve as a point of reference for analyzing piping system distortions or potential failures. Such a program is intended to identify distortions or failures and assure compatibility between the materials and components of existing piping systems with those portions undergoing repair, replacement, or modification.

DEFINITIONS[2]

Code: ASME Code for Pressure Piping, ASME B31.1 Power Piping.

component: equipment, such as vessel, piping, pump, or valve, which are combined with other components to form a system.

critical piping systems: those piping systems which are part of the feedwater-steam circuit of a steam generating power plant, and all systems which operate under two-phase flow conditions. Critical piping systems include runs of piping and their supports, restraints, and root valves. Hazardous gases and liquids, at all pressure and temperature conditions, are also included herein. The Operating Company may, in its judgment, consider other piping systems as being critical in which case it may consider them as part of this definition.

examination: an element of inspection consisting of investigation of materials, components, supplies, or services to determine conformance to those specified requirements which can be determined by such investigation. Examination is usually nondestructive and includes simple physical manipulation, gaging, and measurement.

failure: that physical condition which renders a system, component, or support inoperable.

maintenance: actions required to assure reliable and continued operation of a power plant, including care, repair, and replacement of installed systems.

modification: a change in piping design or operation and accomplished in accordance with the requirements and limitations of the Code.

[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.

[2] The definitions pertain specifically to this Appendix.

Operating Company: the Owner, user, or agent acting on behalf of the Owner who has the responsibility for performing the operations and maintenance functions on the piping systems within the scope of the Code.

procedure: a document that specifies or describes how an activity is to be performed. It may include methods to be employed, equipment or materials to be used, and sequences of operations.

qualification (personnel): demonstration of the abilities gained through training and/or experience that enable an individual to perform a required function.

renewal: that activity which discards an existing component and replaces it with new or existing spare materials of the same or better qualities as the original component.

repair: to restore the system or component to its designed operating condition as necessary to meet all Code requirements.

specification: a set of requirements to be satisfied by a product, material, or process, indicating, whenever appropriate, the procedure by means of which it may be determined whether the requirements given are satisfied.

V-1.0 GENERAL

V-1.1 Application

V-1.1.1 This Appendix recommends minimum requirements for programs to operate and maintain ASME B31.1 Power Piping systems and also for the repairs to these systems.

V-1.1.2 Local conditions and the location of piping systems (such as indoors, outdoors, in trenches, or buried) will have considerable bearing on the approach to any particular operating and maintenance procedure. Accordingly, the methods and procedures set forth herein serve as a general guide. The Operating Company is responsible for the inspection, testing, operation and maintenance of the piping system and shall have the responsibility for taking prudent action to deal with inherent plant conditions.

V-1.2 Conformance

V-1.2.1 When conformance with time periods for examination recommended in this document is impractical, an extension may be taken if an evaluation demonstrates that no safety hazard is present.

V-1.3 Requirements

V-1.3.1 This Appendix recommends that the following listed items be established and implemented:

(A) complete design and installation records of the "as built" large bore piping systems, including expansion joints, hangers, restraints, and other supporting components. The Operating Company shall define those sizes considered to be large bore pipe.

(B) records of operation and maintenance history.

(C) programs for periodic inspection and monitoring.

(D) procedures for reporting and analyzing failures.

(E) procedures for maintenance, repairs, and replacements.

(F) procedures for abandoning piping systems and for maintaining piping systems in and out-of-service condition.

(G) procedures for assuring that all personnel engaged in direct maintenance of such piping systems as defined in para. V-4.2.1(C) are qualified by training or experience for their tasks or work.

V-2.0 OPERATING AND MAINTENANCE PROGRAM

V-2.1

Each Operating Company shall develop an operating and maintenance program comprising a series of written procedures, keeping in mind that it is not possible to prescribe a single set of detailed operating and maintenance procedures applicable to all piping systems. The operating and maintenance procedures shall include: personnel qualifications as defined by the Operating Company, material history and records, and supplementary plans to be implemented in case of piping system failures.

V-2.2

Each plant should maintain and file the following documentation that exists for each unit:

(A) current piping drawings

(B) construction isometrics (or other drawings) that identify weld locations

(C) pipeline specifications covering material, outside diameter, and wall thickness

(D) flow diagrams

(E) support drawings

(F) support setting charts

(G) records of any piping system modifications

(H) material certification records

(I) records of operating events that exceed design criteria of the piping or supports

(J) valve data

(K) allowable reactions at piping connections to equipment

(L) welding procedures and records

V-3.0 REQUIREMENTS OF THE OPERATING, MAINTENANCE, AND MODIFICATION PROCEDURES

V-3.1

The Operating Company shall have procedures for the following:

(A) to perform normal operating and maintenance work. These procedures shall include sufficiently detailed instructions for employees engaged in operating and maintaining the piping systems.

(B) to prescribe action required in the event of a piping system failure or malfunction that may jeopardize personnel safety, safe operation, or system shutdown. Procedures shall consider

(B.1) requirements defined for piping system operations and maintenance and should include failure conditions under which shutdown may be required. Procedures should include both the action required and the consequence of the action on related systems or subsystems.

(B.2) the designation of personnel responsible for the implementation of required action, and minimum requirements for the instruction, training, and qualification of these personnel.

(C) to periodically inspect and review changes in conditions affecting the safety of the piping system. These procedures shall provide for a system of reporting to a designated responsible person in order that corrective measures may be taken.

(D) to assure that modifications are designed and implemented by qualified personnel and in accordance with the provisions of the Code.

(E) to analyze failures to determine the cause and develop corrective action to minimize the probability of recurrence.

(F) to intentionally abandon unneeded piping systems, or portions thereof, and to maintain those which are out of service for extended periods of time as defined by the Operating Company.

(G) to ensure that instruction books and manuals are consulted in performing maintenance operations.

(H) to log, file, maintain, and update instruction books.

(I) to log operating and maintenance records.

(J) to periodically review and revise procedures as dictated by experience and changes in conditions.

V-4.0 PIPING AND PIPE SUPPORT MAINTENANCE PROGRAM AND PERSONNEL REQUIREMENTS

V-4.1 Maintenance Program

V-4.1.1 The maintenance program shall include the following listed features:

(A) a purpose for the program

(B) the frequency for performing all elements of maintenance and inspection listed in the program

(C) generic requirements as related to initial hanger positions at time of unit startup, changes and adjustments in hanger positions at periodic inspections, and review of manufacturer's instruction and maintenance manuals applicable to components included in the program

(D) updating and modification as may be desirable by reason of Code revisions and technological advances or other considerations

(E) steps to keep maintenance and inspection personnel aware of program revisions

V-4.2 Personnel

V-4.2.1 To the extent necessary for conformance with the maintenance program of the Operating Company, only qualified and trained personnel shall be utilized for the following:

(A) observation, measurement, and recording the position of piping systems and hanger readings

(B) adjustment of hangers and all other components of support and restraint systems

(C) repair and periodic maintenance routines including, but not limited to

(C.1) routine piping assembly, including welding of integral attachments

(C.2) mechanical repair of valves, traps, and similar types of piping specialty components, including packings

(C.3) removal and replacement of piping insulation

(C.4) lubrication of applicable piping and hanger components, such as valves and constant supports, maintenance of fluid levels in hydraulic restraints; and stroking of hydraulic and mechanical dynamic restraints (snubbers)

(C.5) routine surveillance for changing conditions including changes in position of piping and settings of piping hangers and shock suppressors (snubbers)

V-4.2.2 Review of records and failure reports and decisions concerning corrective actions or repairs should be carried out by or under the direction of a qualified piping engineer.

V-4.2.3 Welding and Heat Treatment Personnel

(A) Welders shall be qualified to approved welding procedures. Qualification of weld procedures and the qualification performance of the welder shall be in accord with the requirements of para. 127.5.

(B) Trained personnel shall perform preheat and postheat treatment operations as described in the requirements of para. 131.

V-4.2.4 Examination, Inspection, and Testing Personnel. Trained personnel shall perform nondestructive examinations (NDE), including visual inspections and leak tests (LT), all in accord with the requirements of para. 136.

V-5.0 MATERIAL HISTORY

V-5.1 Records

V-5.1.1 Records shall be maintained to the extent necessary to permit a meaningful failure analysis or reconstruction of a prior condition should the need arise. These records may be limited to those systems identified as critical as defined herein.

V-5.1.2 The records listed below are recommended for inclusion in the materials history and, where possible, be traceable to specific components in a piping system.

(A) procurement documents including specifications

(B) original service date and operating conditions

(C) list of materials, both original and replacement, with system location and material specification

(D) physical and mechanical properties from material test reports (if available), including, the following as applicable:

(D.1) Manufacturer's Material Test Reports or Certificate of Conformance

(D.2) chemical analysis

(D.3) impact tests

(D.4) special processing, i.e., heat treatment, mechanical working, etc.

(E) wall thicknesses where available from construction or maintenance records, including design minimum wall requirements

(F) record of alterations or repairs

(G) nondestructive examination reports (including radiographs, if available)

(H) special coatings, linings, or other designs for corrosion or erosion resistance

(I) failure reports

V-5.2 Failure Reports

V-5.2.1 The Operating Company shall be responsible for investigating all material failures in critical piping systems. The cause for failure shall be established. A report of the results of this investigation shall be included in the material history file and shall, as a minimum, contain the following information:

(A) summary of design requirements

(B) record of operating and test experience of failed components

(C) any previous history of the component

(D) any special conditions (corrosion, extraordinary loads, thermal excursions, etc.) which may have contributed to failure

(E) conclusions as to cause

(F) recommendations for corrective actions to minimize recurrence

(G) corrective actions that were taken, including verification of satisfactory implementation

(H) corrective action details and recommendations, if any, for similar action in other piping systems

V-5.3 Restoration After Failure

V-5.3.1 Defective component(s) shall be repaired or replaced with comparable or upgraded materials permissible by this Code after evaluation of the failure and taking into account conclusions as to cause. Even when materials are replaced by same or upgraded items, a formal failure report should follow as in para. V-5.2.

V-5.3.2 Care shall be exercised when replacing system components to ensure no parts of the system are overstressed. The stresses in the repaired system shall be equal to or less than the original stresses unless analysis permits increased stresses. During the replacement of the component, the piping system should be temporarily supported or restrained on both sides of the component to be removed so as to maintain its as-found cold position until the component(s) is (are) installed. If the desired piping position cannot be maintained, an analysis shall be made to determine the reason for the problem. A new stress analysis may be necessary. Care shall be exercised when working on a system that has been subjected to self-springing or cold pull.

V-5.3.3 Weld preparations and fit-up of the weld joints shall meet the requirements of Chapter V.

V-5.3.4 Welding procedures and preheat/postheat treatments of the weld joints shall meet the minimum requirements of Chapter V.

V-5.4 Weld Records

V-5.4.1 Records shall be maintained for all welds in critical piping systems. These records shall include, but not be limited to, the following:

(A) original installation records, where available

(B) repair and modification welds including excavation location and depth

(C) welding procedures and qualification tests

(D) nondestructive examination reports

(E) heat treatment performed

V-5.5 Inspection Program for Materials With Adverse History

V-5.5.1 Materials that have been reported to the industry to exhibit an adverse performance under certain conditions shall be given special attention by the Operating Company through a program of planned examination and testing. This program shall include the development of procedures for repair or replacement of the material when the Operating Company determines that such action is necessary.

V-5.5.2 Methods of surveillance and analysis shall be determined by the Operating Company.

V-5.5.3 The frequency of the material inspection shall also consider the expected service life of the component.

V-5.6 Nondestructive Examination

V-5.6.1 Nondestructive examinations used to investigate any suspect materials or problem areas shall be in accordance with Chapter VI.

V-6.0 PIPING POSITION HISTORY

V-6.1 General

V-6.1.1 Movements of critical piping systems from their design locations shall be used to assess piping integrity. The Operating Company shall have a program requiring such movements be taken on a periodic basis along with a procedure precluding the unnecessary removal of insulation when measurements are taken; refer to para. V-6.3. Piping system movement records shall be maintained. The Operating Company shall evaluate the effects of position changes on the safety of the piping systems and shall take appropriate corrective action.

V-6.1.2 Although the Code recognizes that high temperature piping systems seldom return to their exact original positions after each heat cycle due to relaxation, critical piping systems as defined herein, must be maintained within the bounds of engineering evaluated limitations.

V-6.2 Visual Survey

V-6.2.1 The critical piping systems shall be observed visually, as frequently as deemed necessary, and any unusual conditions shall be brought to the attention of personnel as prescribed in procedures of para. V-3.1. Observations shall include determination of interferences with or from other piping or equipment, vibrations, and general condition of the supports, hangers, guides, anchors, supplementary steel, and attachments, etc..

V-6.3 Piping Position Markers

V-6.3.1 For the purpose of easily making periodic position determinations, it is recommended that permanent markings on critical piping systems be made by providing markings or pointers attached to piping components. The position of these markings or pointers should be noted and recorded with respect to stationary datum reference points.

V-6.3.2 Placement of pointers shall be such that personnel safety hazards are not created.

V-6.4 Hangers and Supports on Critical Piping Systems and Other Selected Systems

V-6.4.1 Hanger position scale readings of variable and constant support hangers shall be determined periodically. It is recommended that readings be obtained while the piping is in its fully hot position, and if practical, when the system is reasonably cool or cold sometime during the year as permitted by plant operation. Pipe temperature at time of reading hangers shall be recorded.

V-6.4.2 Variable and constant support hangers, vibration snubbers, shock suppressors, dampeners, slide supports and rigid rod hangers shall be maintained in accordance with the limits specified by the manufacturers and designers. Maintenance of these items shall include, but not necessarily be limited to, cleaning, lubrication and corrosion protection. All dynamic restraints (snubbers) should be stroked periodically.

V-6.5 Records on Critical Piping Systems and Other Selected Systems

V-6.5.1 Pipe location readings and travel scale readings of variable and constant support hangers shall be recorded on permanent log sheets in such a manner that will be simple to interpret. See Fig. V-6.5 for suggested hanger record data sheet. Records of settings of all hangers shall be made before the original commercial operation of the plant. Log sheets should be accompanied by a pipe support location plan or piping system drawing with hanger mark numbers clearly identified. In addition, records are to be maintained showing movements of or in expansion joints including records of hot and cold or operating and shutdown positions, particularly those not equipped with control rods or gimbals.

V-6.6 Recommendations

V-6.6.1 After complete examination of the records (para. V-6.5), recommendations for corrective actions needed shall be made by a piping engineer or a qualified responsible individual or organization. Repairs and/or modifications are to be carried out by qualified maintenance personnel for all of the following items:

(A) excessively corroded hangers and other support components

(B) broken springs or any hardware item which is part of the complete hanger or support assembly

(C) excessive piping vibration; valve operator shaking or movements

(D) piping interferences

(E) excessive piping deflection which may require the installation of spring units having a greater travel range

(F) pipe sagging which may require hanger adjustment or the reanalysis and redesign of the support system

(G) hanger unit riding at either the top or the bottom of the available travel

(H) need for adjustment of hanger load carrying capacity

(I) need for adjustments of hanger rods or turnbuckle for compensation of creep or relaxation of the piping

(J) loose or broken anchors

(K) inadequate clearances at guides

(L) inadequate safety valve vent clearances at outlet of safety valves

(M) any failed or deformed hanger, guide, U-bolt, anchor, snubber, or shock absorber, slide support, dampener, or supporting steel

Company ______________________ System ______________________ Prepared by ______________________

Plant Name ______________________ Unit No. ______________________ Sheet __________ Of __________

Date ______________________

				Constant Support Hanger Data		Rigid & Variable Support Data		Design Elev. Cold [Note (3)], ft.-in.	Hanger Scale Reading [Note (4)]							
									Date:		Date:		Date:		Date:	
						Design Load, lb										
Hanger Mk. No.	Hanger Size [Note (1)]	Hanger Type [Note (2)]	Dsgn. Mvt.	Design Load, lb	Design Total Travel	H	C	(4)	Cold	Read by	Hot	Read by	Cold	Read by	Hot	Read by

NOTES:

(1) Hanger size to be taken from hanger fabrication drawing.

(2) For constant support and variable support types indicate by CS or VS. For rigid, anchor, guide, sliding, or other type support, indicate by letter R, A, G, or S, respectively.

(3) Elevation of center line of pipe after cold springing and final hanger settings with line cold.

(4) "0" indicated the highest scale position with "5" being the midpoint, and "10" being the lowest scale position.

Fig. V-6.5 Hanger Record Sheet

(N) unacceptable movements in expansion joints

(O) low fluid levels in hydraulic pipe restraints

V-7.0 PIPING CORROSION

V-7.1 General

V-7.1.1 This section pertains to the requirements for inspection of critical piping systems that may be subject to internal or external corrosion-erosion, such as buried pipe, piping in a corrosive atmosphere, or piping having corrosive or erosive contents. Requirements for inspection of piping systems in order to detect wall thinning of piping and piping components due to erosion/corrosion, or flow-assisted corrosion, is also included. Erosion/corrosion of carbon steel piping may occur at locations where high fluid velocity exists adjacent to the metal surface, either due to high velocity or the presence of some flow discontinuity (elbow, reducer, expander, tee, control valve, etc.) causing high levels of local turbulence. The erosion/corrosion process may be associated with wet steam or high purity, low oxygen content water systems. Damage may occur under both single and two-phase flow conditions. Piping systems that may be damaged by erosion/corrosion include, but are not limited to, feedwater, condensate, heater drains, and wet steam extraction lines. Maintenance of corrosion control equipment and devices is also part of this section. Measures in addition to those listed herein may be required.

V-7.1.2 Where corrosion is cited in this section, it is to be construed to include any mechanism of corrosion and/or erosion. Recommended methods for monitoring and detection, acceptance standards, and repair/replacement procedures for piping components subjected to various erosion/corrosion mechanisms, including flow-assisted corrosion, are provided in nonmandatory Appendix IV.

V-7.1.3 Guidance for the evaluation and monitoring of carbon steel piping susceptible to erosion/corrosion (flow assisted corrosion) is provided in Appendix IV, para. IV-5.0.

V-7.2 Procedures

V-7.2.1 The Operating Company shall establish procedures to cover the requirements of this paragraph.

V-7.2.2 Procedures shall be carried out by or under the direction of persons qualified by training or experience in corrosion control and evaluation of piping systems for corrosion damage.

V-7.2.3 Procedures for corrosion control shall include, but not be limited to the following:

(A) maintenance painting to resist external ambient conditions

(B) coating and/or wrapping for external protection of buried or submerged systems

(C) lining to resist internal corrosion from system fluid when applicable

(D) determining the amount of corrosion or erosion of the piping system internals caused by the flowing fluid

(E) determining the amount of external corrosion caused by ambient conditions, such as atmosphere, buried in soil, installed in tunnels or covered trenches, and submerged underwater

(F) preparing records which shall include all known leakage information, type of repair made, location of cathodically protected pipe, and the locations of cathodic protection facilities including anodes

(G) examining records from previous inspection and performing additional inspections where needed for historical records

V-7.3 Records

V-7.3.1 Tests, surveys, and inspection records to indicate the adequacy of corrosion control shall be maintained for the service life of the piping system. This should include records of measured wall thickness and rates of corrosion.

V-7.3.2 Inspection and maintenance records of cathodic protection systems shall be maintained for the service life of the protected piping.

V-7.3.3 Observations of the evidence of corrosion found during maintenance or revision to a piping system shall be recorded.

V-7.4 Examination of Records

V-7.4.1 Records shall be examined and evaluated by trained personnel.

V-7.4.2 Where inspections or leakage history indicate that active corrosion is taking place to the extent that a safety hazard is likely to result, applicable portions of the system shall be replaced with corrosion resistant materials or with materials which are protected from corrosion, or other suitable modifications shall be made.

V-7.5 Frequency of Examination

V-7.5.1 Within 3 years after original installation, each piping system shall be examined for evidence of corrosion in accordance with the requirements established by the Operating Company's procedures. Piping in severe service or environmental conditions should be inspected initially within a time frame commensurate with the severity of the service or environment. Corrective measures shall be taken if corrosion is above the amount allowed for in the original design.

V-7.5.2 Continued examination shall be made at intervals based upon the results of the initial inspection, but not to exceed 5 years, with corrective measures being taken each time that active corrosion is found.

V-7.5.3 Examination for evidence of internal corrosion shall be made by one of the following:

(A) drilled hole with subsequent plugging

(B) ultrasonic test for wall thickness determination

(C) removal of representative pipe section at flange connections or couplings

(D) removal of short section of pipe

(E) radiography for evidence of wall thinning

(F) borescope or videoprobe examination

(G) a method equivalent to those above

V-7.5.4 Examinations for evidence of external corrosion shall be made after removal of covering, insulation or soil on a representative short section of the piping system taking into consideration varying soil conditions.

V-8.0 PIPING ADDITION TO EXISTING PLANTS

V-8.1 Piping Classification

V-8.1.1 Piping and piping components which are replaced, modified, or added to existing piping systems are to conform to the edition and addenda of the Code used for design and construction of the original systems, or to later Code editions or addenda as determined by the Operating Company. Any additional piping systems installed in existing plants shall be considered as new piping and shall conform to the latest issue of the Code.

V-8.2 Duplicate Components

V-8.2.1 Duplicates of original components and materials are permitted for permanent replacements, provided the renewal is a result of reasonable wear and not the result of the improper application of the material, such as temperature and corrosive environment.

V-8.3 Replacement Piping and Piping Components

V-8.3.1 Where replacement components differ from the original components with respect to weight, dimensions, layout, or material, the design of the affected piping system shall be rechecked for the following design considerations.

(A) Hangers and supports shall be adequate for additional or altered distribution of weight. They shall accommodate the flexibility characteristics of the altered piping system.

(B) Changes in stresses imposed on both existing and replacement components of the piping shall be evaluated and compensation shall be made to prevent overstress in any part of the entire altered piping system.

V-9.0 SAFETY, SAFETY RELIEF, AND RELIEF VALVES

V-9.1 General

V-9.1.1 This section is applicable to safety, safety relief, and relief valve installations (see Appendix II for definitions of these terms.) Except as otherwise noted, all reference to "safety" valve(s) shall be considered to include all three types. Safety valves shall be maintained in good working condition. Also, discharge pipes and their supports shall be inspected routinely and maintained properly. Any evidence of blowback at the drip pan of open safety valve vent systems should be noted and its cause determined and corrected.

V-9.2 Testing and Adjustment

V-9.2.1 Testing of safety valves for pressure setting shall be in accordance with written procedures which incorporate the requirements of regulatory agencies and manufacturer's instructions. Testing should be performed just prior to a planned outage so that any required repair or maintenance, except spring and blowdown ring adjustments, can be performed during the outage, thereby assuring tight valves upon return to service.

V-9.2.2 The setting or adjustment of safety valves shall be done by personnel trained in the operation and maintenance of such valves. Safety valves shall be tested after any change in setting of the spring or blowdown ring. Appropriate seals should be used to assure that there is no unauthorized tampering with valve settings. Repairs to safety valves and disassembly, reassembly, and/or adjustments affecting the pressure relief valve function, which are considered a repair, should be performed by an authorized repair organization.[3]

V-9.3 Operation

V-9.3.1 The precautions stated in the manufacturer's operating manual or instruction books shall be followed when operating safety valves. In general, these precautions will include the following:

(A) Hand lifting is permitted. Assistance, as required, may be accomplished by the use of small wires or chains.

(B) Striking or hammering the valve body shall not be permitted. Only the hand-test lever shall be used.

(C) Attempts to stop leakage through the valve seat shall not be made by compressing the spring.

V-10.0 DYNAMIC LOADING

V-10.1 Water Hammer

V-10.1.1 Water hammer includes any water or other liquid transient event such as pressure surge or impact loading resulting from sudden or momentary change in flow or flow direction.

[3] Examples of organizations that may be authorized by the owner, or by the local jurisdiction, to perform repairs on safety valves include but are not limited to the original valve manufacturer or a repair organization that holds a National Board of Boiler and Pressure Vessel Inspectors (NB-23) VR stamp.

V-10.1.2 Should significant water hammer develop during plant operation, the cause should be determined and corrective action taken. Water hammer could be the result of an incorrectly sloped pipe intended for steam condensate drainage. Water hammer in piping systems may cause damage to hangers, valves, instrumentation, expansion joints, piping and equipment integral with the piping. The Operating Company should develop procedures to deter water hammer and to determine when corrective action is necessary.

V-10.1.3 Water hammer problems resulting from accumulated condensate in a steam line cannot be solved simply by adding restraints. Corrective action may include changing line slopes, adding drain pots, adding warm-up lines around valves, checking for leaking desuperheaters, faulty electrical controls on automatic drains, etc.

V-10.1.4 Water hammer due to column separation in feedwater or booster pump suction piping results when the deaerator pegging pressure is not maintained. This type of water hammer can be particularly severe and requires prompt attention to control and reduce it.

V-10.1.5 As a priority, corrective action should address the cause of water hammer first. If such corrective action is ineffective in reducing the effects of water hammer to acceptable levels, installation of restraints may be necessary to limit piping displacements and/or damage from fatigue.

V-10.2 Steam Hammer

V-10.2.1 Dynamic loads due to rapid changes in flow conditions and fluid state in a steam piping system are generally called steam hammer loads. Piping response to these momentary unbalanced loads can be significant in high pressure steam systems, such as main steam, hot and cold reheat steam, bypass and auxiliary steam systems that are subject to rapid interruption or establishment of full steam flow.

V-10.2.2 The Operating Company should develop procedures to determine any adverse effects of steam hammer, such as excessive pipe movement, damage to hangers and restraints, and high pipe stress and reactions at pipe connections to equipment. Where such movements, stresses, and reactions exceed safe limits or allowable loadings, a program of remedial action should be implemented.

V-11.0 HIGH-TEMPERATURE CREEP

V-11.1 General

V-11.1.1 Catastrophic failure, including rupture, can occur due to excessive creep strains resulting from operation of the piping above design pressure, or temperatures, or both, for extended periods of time. The expected life of a piping system operating in the creep range can be reduced significantly through prolonged exposure to pressure or temperature, particularly temperature, above design values. Paragraph 102.2.4 provides criteria for occasional short operating periods at higher than design pressure or temperature.

V-11.1.2 This section provides the minimum requirements for evaluating critical piping systems in order to detect creep damage and to assist in predicting the remaining life under expected operating conditions. The remaining useful life may be estimated by determining the extent of creep damage sustained by the pipe.

V-11.2 Procedures

V-11.2.1 The Operating Company shall establish procedures to cover the requirements of this paragraph.

V-11.2.2 The procedures shall be carried out by or under the direction of persons qualified by training or experience in metallurgical evaluation of high temperature creep effects in power plant piping.

V-11.2.3 An evaluation program to determine the extent of creep damage and estimate remaining life of high temperature piping shall be carried out in three phases, as follows:

(A) review of material specifications, design stress levels, and operating history.

(B) indirect measurements to determine extent of creep damage. These would include diametral measurements to detect creep swelling. In addition, dye penetrant, magnetic particle, ultrasonic, and radiographic methods may be used to detect internal and surface cracks.

(C) examination of the microstructure to determine the degree of material degradation. This can be performed by replication techniques or by metallography using specimens obtained by boat-sampling or trepanning.

V-11.3 Records

V-11.3.1 Records of creep damage evaluation survey findings shall be maintained for the service life of high temperature piping systems operating in the creep range.

V-11.4 Examination of Records

V-11.4.1 Records of creep damage surveys and test reports shall be examined by personnel qualified by training and experience to evaluate and interpret NDE and metallographical studies.

V-11.4.2 Where surveys and inspections of critical piping systems indicate that high temperature creep damage has progressed to an unacceptable level, affected portions of the piping system shall be replaced.

V-11.5 Frequency of Examination

V-11.5.1 Periodically, all critical piping systems operating within the creep range shall be examined for evidence of high temperature creep damage. Particular attention shall be given to welds.

V-11.5.2 The examination shall be repeated at periodic intervals which shall be established on the basis of earlier survey findings, operating history, and severity of service.

V-12.0 RERATING PIPING SYSTEMS

V-12.1 Conditions

V-12.1.1 An existing piping system may be rerated for use at a higher pressure and/or temperature if all of the following conditions are met:

(A) A design analysis shall be performed to demonstrate that the piping system meets the requirements of the Code at the new design conditions.

(B) The condition of the piping system and support/restraint scheme shall be determined by field inspections and the examination of maintenance records, manufacturer's certifications, and/or other available information to ensure conformance with the Code requirements for the new design conditions.

(C) Necessary repairs, replacements, or alterations to the piping system are made to conform with the requirements prescribed in (A) and (B) above.

(D) The system has been leak tested to a pressure equal to or greater than that required by the Code for a new piping system at the new design conditions.

(E) The rate of pressure and temperature increase to the higher maximum allowable operating conditions shall be gradual so as to allow sufficient time for periodic observations of the piping system movements and leak tightness.

(F) Records of investigations, work performed, and pressure tests conducted in rerating the piping systems shall be preserved for the service life of the piping systems.

(G) All safety valves, relief valves, and other pressure relieving devices must be examined, and recertified for the new pressure/temperature design conditions. Capacity of relieving equipment shall be investigated if the design pressure and/or temperature are changed in rerating a piping system.

SECTION 2 – Fitness-For-Service Engineering Assessment Procedure

(Jan, 2000)

2.1 General

2.1.1 This document contains Fitness-For-Service (FFS) assessment procedures that can be used to evaluate pressurized components containing flaws or damage. If the results of a fitness-for-service assessment indicate that the equipment is suitable for the current operating conditions, the equipment can continue to be operated at these conditions provided suitable monitoring/inspection programs are established. If the results of the fitness-for-service assessment indicate that the equipment is not suitable for the current operating conditions, calculation methods are provided to rerate the component. For pressurized components (e.g. pressure vessels and piping) these calculation methods can be used to find a reduced Maximum Allowable Working Pressure $(MAWP)$ and/or coincident temperature. For tank components (shell courses) the calculation methods can be used to determine a reduced Maximum Fill Height (MFH).

2.1.2 The Fitness-For-Service assessment procedures in this document are organized by flaw type and/or damage mechanism. A list of flaw types and damage mechanisms and the corresponding section which provides the FFS assessment methodology is shown in Table 2.1. In some cases, it may be necessary to use the assessment procedures from multiple sections if the primary type of damage is not evident. For example, the metal loss in a component may be associated with general corrosion, local corrosion and pitting. If multiple damage mechanisms are present, a degradation class, e.g., corrosion/erosion, can be identified to assist in the evaluation. An ov rview of degradation classes in this document is shown in Figure 2.1. As indicated in this figure, several flaw types and damage mechanisms may need to be evaluated to determine the Fitness-For-Service of a component. Each section referenced within a degradation class includes guidance on how to perform an asse sment when multiple damage mechanisms are present.

2.1.3 The general Fitness-For-Service assessment procedure used in this Recommended Practice (RP) for all flaw types is provided in this section. An overview of the procedure is provided in the following eight steps. The remaining sections in this RP utilize this assessment methodology for a specific flaw type or damage mechanism and provide specific details covering Steps 2 through 8 of this procedure.

2.1.3.1 *Step 1 – Flaw and Damage Mechanism Identification:* The first step in a Fitness-For-Service assessment is to identify the flaw type and cause of damage (see paragraph 2.1.2). The original design and fabrication practices, the material of construction, and the service history and environmental conditions can be used to ascertain the likely cause of the damage. An overview of damage mechanisms which can assist in identifying likely causes of damage is provided in Appendix G. Once the flaw type is identified, the appropriate section of this document can be selected for the assessment (see Table 2.1 and Figure 2.1).

2.1.3.2 *Step 2 – Applicability and Limitations of the FFS Assessment Procedures:* The applicability and limitations of the assessment procedure are described in each section, and a decision on whether to proceed with an assessment can be made.

2.1.3.3 *Step 3 – Data Requirements:* The data required for a FFS assessment depend on the flaw type or damage mechanism being evaluated. Data requirements may include original equipment design data, information pertaining to maintenance and operational history, expected future service, and data specific to the FFS assessment such as flaw size, state of stress in the component at the location of the flaw, and material properties. Data requirements common to all FFS assessment procedures are covered in this section. Data requirements specific to a damage mechanism or flaw type are covered in the section containing the corresponding assessment procedures.

2.1.3.4 *Step 4 – Assessment Techniques and Acceptance Criteria:* Assessment techniques and acceptance criteria are provided in each section. If multiple damage mechanisms are present, more than one section may have to be used for the evaluation.

2.1.3.5 *Step 5 – Remaining Life Evaluation*: An estimate of the remaining life or limiting flaw size should be made for the purpose of establishing an inspection interval. The remaining life is established using the *FFS* assessment procedures with an estimate of future damage. The remaining life can be used in conjunction with an inspection code to establish an inspection interval.

2.1.3.6 *Step 6 – Remediation*: Remediation methods are provided in each section based on the damage mechanism or flaw type. In some cases, remediation techniques may be used to control future damage associated with flaw growth and/or material degradation.

2.1.3.7 *Step 7 – In-Service Monitoring*: Methods for in-service monitoring are provided in each section based on the damage mechanism or flaw type. In-service monitoring may be used for those cases where a remaining life and inspection interval cannot adequately be established because of the complexities associated with the service environment.

2.1.3.8 *Step 8 – Documentation*: Documentation should include a record of all information and decisions made in each of the previous steps to qualify the component for continued operation. Documentation requirements common to all *FFS* assessment procedures are covered in this section. Documentation requirements specific to a damage mechanism or flaw type are covered in the section containing the corresponding assessment procedures.

2.2 Applicability And Limitations Of The FFS Assessment Procedures

2.2.1 The *FFS* assessment procedures in this document were developed to assess components with a flaw resulting from single or multiple damage mechanisms. In the context of this document, a component is defined as any pressurized part that is designed using a nationally recognized code or standard (see paragraph 2.2.2). Equipment is defined to be an assemblage of components. Therefore, the pressurized equipment covered in this document includes all pressure boundary components of pressure vessels, piping, and tank shell courses of storage tanks Fitness-for-service procedures for fixed and floating roof structures, and bottom plates of tanks are covered in Section 2 of API 653.

2.2.2 The *FFS* assessment procedures in this document were developed assuming that the component was designed and fabricated to a recognized code or standard (see Section 1, paragraphs 1.2.2 nd 1.2.3).

2.2.3 For equipment components that are discovered to not have been designed, or constructed to the original design criteria, the principles in this document may be used to evaluat the in-service damage and as-built condition relative to the intended design. *FFS* assessments of this type shall be performed by an Engineer (see Section 1, paragraph 1.4.3) knowledgeable and experienced in the design requirements of the applicable code.

2.2.4 Each section in this document where *FFS* Assessment procedures are described include a segment which states the applicability and limitations of the procedures. The limitations and applicability of an analysis procedure are stated relative to the Level of Assessment (see paragraph 2.4).

2.3 Data Requirements

2.3.1 Original Equipment Design Data

2.3.1.1 The following original equipment design data should be assembled to perform a *FFS* assessment. The extent of the data required depends on the damage mechanism and assessment level A dat sheet is included in Table 2.2 to record the required information that is common to all *FFS* assessments. In addition, a separate data sheet is included with each section of this document to record information specific to the flaw type, damage mechanism, and assessment procedure.

a. Data for pressure vessels may include some or all of the following:

1. An ASME Manufacturer's Data Report or, if the vessel is not Code stamped, other equivalent documentation or specifications.

2. Vessel fabrication drawings showing sufficient details to permit calculation of the $MAWP$ of the component containing the flaw. If a rerate to a different condition of pressure and/or temperature is desired (i.e. increase or decrease in conditions), this information should be available for all affected components. Detailed sketches with data necessary to perform $MAWP$ calculations may be used if the original fabrication drawings are not available.

3. The original or updated design calculations for the load cases in Table A.1 of Appendix A, as applicable, and anchor bolt calculations.

4. The inspection records for the component at the time of fabrication.

5. User Design Specification if the vessel is designed to the ASME Code, Section VIII, Division 2.

6. Material test reports.

7. Pressure-relieving device information including pressure relief valve and/or rupture disk setting and capacity information.

8. A record of the original hydrotest including the test pressure and metal temperature at the time of the test or, if unavailable, the water or ambient temperature.

b. Data for piping components may include some or all of the following:

1. Piping Line Lists or other documentation showing process design conditions, and a description of the piping class including material specification, pipe wall thickness and pressure-temperature rating.

2. Piping isometric drawings to the extent necessary to perform a FFS assessment. The piping isometric drawings should include sufficient details to permit a piping flexibility calculation if this analysis is deemed necessary by the Engineer in order to determine the $MAWP$ (maximum safe or maximum allowable operating pressure) of all piping components. Detailed sketches with data necessary to perform $MAWP$ calculations may be used if the original piping isometric drawings are not available.

3. The original or updated design calculations for the load cases in Table A.1 of Appendix A, as applicable.

4. The inspection records for the component at the time of fabrication.

5. Material test reports.

6. A record of the original hydrotest including the test pressure and metal temperature at the time of the test, or if unavailable, the water or ambient temperature

c. Data for tanks may include some or all of the following:

1. The original API data sheet.

2. Fabrication drawings showing sufficient details to permit calculation of the maximum fill height (*MFH*) for atmospheric storage tanks and the *MAWP* for low pressure storage tanks. Detailed data with sketches where necessary may be used if the original fabrication drawings are not available.

3. The original or updated design calculations for the load cases in Table A.1 of Appendix A, as applicable, and anchor bolt calculations.

4. The inspection records for the component at the time of fabrication.

5. Material test reports.

6. A record of the last hydrotest performed including the test pressure and metal temperature at the time of the test or, if unavailable, the water or ambient temperature.

2.3.1.2 If some of these data are not available, physical measurements or field inspection of the component should be made to provide the information necessary to perform the assessment.

2.3.2 Maintenance And Operational History

2.3.2.1 A progressive record including, but not limited to, the following should be available for the equipment being evaluated. The extent of the data required depends on the damage mechanism and assessment level.

a. The actual operating envelope consisting of pressure and temperature, including upset conditions should be obtained. If the actual operating conditions envelope is not available, an approximation of one should be developed based upon available operational data and consultation with operating personnel. An operating histogram may be required consisting of pressure and temperature data recorded simultaneously for some types of *FFS* assessments (e.g., Section 10.0 for components operating in the creep regime).

b. Documentation of any significant changes in service conditions including pressure, temperature, fluid content and corrosion rate Both past and future service conditions should be reviewed and documented.

c. The date of installation and a summary of all alterations and repairs including required calculations, material changes, drawings and repair procedures. The calculations should include the required wall thicknesses and *MAWP* (*MFH* for atmospheric storage tanks) including definition and allowances for supplemental loads such as static liquid head, wind, and earthquake loads.

d. Records of all hydrotests performed as part of the repair including the test pressure and metal temperature at the time of the tests or, if unavailable, the water or ambient temperature at the time of the test if known.

e. Results of prior in-service examinations including wall thickness measurements and other NDE results that may assist in determining the structural integrity of the component and in establishing a corrosion rate.

f. Records of all internal repairs, weld build-up and overlay, and modifications of internals.

g. Records of "out-of-plumb" readings for vertical vessels.

h. Foundation settlement records if the corrosion being evaluated is located in the bottom shell course of the tank.

2.3.2.2 If some of these data are not available, physical measurements should be made to provide the information necessary to perform the assessment.

2.3.3 Required Data/Measurements For A FFS Assessment

2.3.3.1 Each section in this document which contains *FFS* assessment procedures includes specific requirements for data measurements and flaw characterization based on the damage mechanism being evaluated. Examples of flaw characterization include thickness profiles for local corrosion/erosion, pitting depth, and dimensions of crack-like flaws. The extent of information and data required for a *FFS* assessment is dependent on the assessment level and damage mechanism being evaluated.

2.3.3.2 The Future Corrosion Allowance (*FCA*) should be established for the intended future operating period. The *FCA* should be based on past inspection information or corrosion rate data relative to the component material in a similar environment. Corrosion rate data may be obtained from API Publication 581 or other sources (see paragraph A.2.7 of Appendix A). The *FCA* is calculated by multiplying the anticipated corrosion rate by the future service period considering inspection interval requirements of the applicable inspection code. The *FFS* assessment procedures in this document include provisions to ensure that the *FCA* is available for the future intended operating period.

2.3.4 Recommendations For Inspection Technique And Sizing Requirements

Recommendations for Non Destructive Examination *(NDE)* procedures with regard to detection and sizing of a particular damage mechanism and/or flaw type are provided in each section.

2.4 Assessment Techniques And Acceptance Criteria

2.4.1 Three Levels of assessment are provided in each Section of this document which cover *FFS* assessment procedures. A logic diagram is included in each Section to illustrate how these assessment levels are interrelated. In general, each assessment level provides a balance between conservatism, the amount of information required for the evaluation, the skill of the personnel performing the assessment, and the complexity of analysis being performed. Level 1 is the most conservative, but is easiest to use. Practitioners usually proceed sequentially from a Level 1 to a Level 3 analysis (unless otherwise directed by the assessment techniques) if the current assessment level does not provide an acceptable result, or a clear course of action cannot be determined. A general overview of each assessment level and its intended use are described below.

2.4.1.1 *Level 1* – The assessment procedures included in this level are intended to provide conservative screening criteria that can be utilized with a minimum amount of inspection or component information. Level 1 assessments may be performed by either plant inspection or engineering personnel (see Section 1, paragraphs 1.4.2 and 1.4.3).

2.4.1.2 *Level 2* – The assessment procedures included in this level are intended to provide a more detailed evaluation that produces results that are more precise than those from a Level 1 assessment. In a Level 2 Assessment, inspection information similar to that required for a Level 1 assessment are needed; however, more detailed calculations are used in the evaluation. Level 2 assessments would typically be conducted by plant engineers, or engineering specialists experienced and knowledgeable in performing *FFS* assessments.

2.4.1.3 *Level 3* – The assessment procedures included in this level are intended to provide the most detailed evaluation which produces results that are more precise than those from a Level 2 assessment. In a Level 3 Assessment the most detailed inspection and component information is typically required, and the recommended analysis is based on numerical techniques such as the finite element method.

A Level 3 analysis is primarily intended for use by engineering specialists experienced and knowledgeable in performing FFS assessments.

2.4.2 Each of the FFS assessment methodologies presented in this document utilize one or more of the following acceptance criteria:

2.4.2.1 *Allowable Stress* – This acceptance criteria is based upon calculation of stresses resulting from different loading conditions, classification and superposition of stress results, and comparison of the calculated stresses in an assigned category or class to an allowable stress value. An overview and aspects of these acceptance criteria are included in Appendix B. The allowable stress value is typically established as a fraction of yield, tensile or rupture stress at room and the service temperature, and this fraction can be associated with a design margin. This acceptance criteria method is currently utilized in most new construction design codes. In FFS applications, this method has proven to have limited applicability because of the difficulty in establishing suitable stress classifications for components containing flaws. As an alternative, assessment methods based on elastic-plastic analysis can be used (see Appendix B, paragraph B.6.4). Elastic-plastic analysis methods were used to develop the Remaining Strength Factor (see paragraph 2.4.2.2).

2.4.2.2 *Remaining Strength Factor* – Structural evaluation procedures using linear elastic stress analysis with stress classification and allowable stress acceptance criteria provide only a rough approximation of the loads which a component can withstand without failure. A better estimate of the safe load carrying capacity of a component can be provided by using nonlinear stress analysis to: develop limit and plastic collapse loads, evaluate the deformation characteristics of the component (e.g. deformation or strain limits associated with component operability), and assess fatigue and/or creep damage including ratcheting.

a. In this document, the concept of a remaining strength factor is utilized to define the acceptability of a component for continued service. The Remaining Strength Factor (RSF) is defined as:

$$RSF = \frac{L_{DC}}{L_{UC}} \tag{2.1}$$

where

L_{DC} = Limit or plastic collapse load of the damaged component (component with flaws), and

L_{UC} = Limit or plastic collapse load of the undamaged component.

b. With this definition of the RSF, acceptance criteria can be established using traditional code formulas, elastic stress analysis, limit load theory, or elastic-plastic analysis. For example, to evaluate local thin areas (see Section 5), the FFS assessment procedures provide a means to compute a RSF. If the calculated RSF is greater than the allowable RSF (see below) the damaged component can be placed back into service. If the calculated RSF is less than the allowable value, the component can be repaired, rerated or some form of remediation can be applied to reduce the severity of the operating environment. The rerated pressure can be calculated from the RSF as follows:

$$MAWP_r = MAWP\left(\frac{RSF}{RSF_a}\right) \quad for \quad RSF < RSF_a \tag{2.2}$$

$$MAWP_r = MAWP \quad for \quad RSF \geq RSF_a \tag{2.3}$$

where

$MAWP_r$ = Reduced permissible maximum allowable working pressure of the damaged component,

$MAWP$ = Maximum allowable working pressure of the undamaged component (see paragraph A.2.1 of Appendix A),

RSF = Remaining strength factor computed based on the flaw and damage mechanism in the component, and

RSF_a = Allowable remaining strength factor (see paragraph 2.4.2.2.d).

c. For tankage, the RSF acceptance criteria is:

$$MFH_r = MFH\left(\frac{RSF}{RSF_a}\right) \quad for \quad RSF < RSF_a \tag{2.4}$$

$$MFH_r = MFH \quad for \quad RSF \geq RSF_a \tag{2.5}$$

where RSF and RSF_a are defined in paragraph 2.4.2.2.b and,

MFH_r = Reduced permissible maximum fill height of the damaged tank course, and

MFH = Maximum fill height of the undamaged component (see paragraph A.2.1 of Appendix A).

d. The recommended value for the allowable Remaining Strength Factor, RSF_a, is 0.90 for equipment in process services. This value has been shown to be conservative (see Appendix H). This value may be reduced based upon the type of loading (e.g. normal operating loads, occasional loads, short-time upset conditions) and/or the consequence of failure. For example, a lower factor could be utilized for low pressure piping containing a flaw which conveys cooling water, or for a shell section containing a flaw subject to normal operating pressure and design wind loads.

2.4.2.3 *Failure Assessment Diagram* – The Failure Assessment Diagram *(FAD)* is used for the evaluation of crack-like flaws in components.

a. The FAD approach was adopted because it provides a convenient, technically based method to provide a measure for the acceptability of a component with a crack-like flaw when the failure mechanism is measured by two distinct criteria: unstable fracture and limit load. Unstable fracture usually controls failure for small flaws in components fabricated from a brittle material and plastic collapse typically controls failure for large flaws if the component is fabricated from a material with high toughness. In a FFS analysis of crack-like flaws, the results from a stress analysis, stress intensity factor and limit load solutions, the material strength, and fracture toughness are combined to calculate a toughness ratio, K_r, and load ratio, L_r. These two quantities represent the coordinates of a point which is plotted on a two-dimensional FAD to determine acceptability. If the assessment point is on or below this curve, then an acceptable margin below the postulated failure curve on the FAD (the failure curve represents the upper bound on component acceptability), the component is suitable for continued operation. A schematic which illustrates the procedure for evaluating a crack-like flaw using the Failure Assessment Diagram is shown in Figure 2.2.

b. In the assessment of crack-like flaws, partial safety factors are utilized along with the FAD acceptance criteria to account for variability of the input parameters in a deterministic fashion. Three separate partial safety factors are utilized: a factor for applied loading; a factor for material toughness; and a factor for flaw dimensions. The partial safety factors are applied to the stresses resulting from a stipulated loading condition, the fracture toughness and the flaw size parameters prior to the FAD analysis. The partial safety factors recommended for use with Section 9 of this document (see Table 9.2) were developed based upon the results of a series of probabilistic analyses of components with crack-like flaws. Other values for these factors may be used based on a risk assessment where the potential failure modes and type of loading (e.g., normal operating loads, occasional loads, short-time upset conditions) are considered.

c. The in-service margin for a component with a crack-like flaw provides a measure of how close the component is to the limiting condition in the FAD. The in-service margin is defined by how far the assessment point, which represents a single operating condition, is within the failure envelope of the FAD. This point is determined based on the results from stress and fracture mechanics analyses after applying the three partial safety factors discussed above. The in-service margin is defined to be greater than or equal to one when the point resides underneath or on the FAD failure curve. The recommended minimum allowable value for the in-service margin is set at 1.0.

2.4.3 The FFS assessment procedures provided in this document are deterministic in that all information required for an analysis (independent variables) are assumed to be known. However, in many instances all of the important independent variables are not known with a high degree of accuracy. In such cases, conservative estimates of the independent variables are made to ensure an acceptable safety margin, and this approach can lead to overly conservative results. The following types of analyses can be used to provide insight into the dependency of the analysis results with variations in the input parameters. The deterministic FFS assessment procedures in this Practice can be used with any of these analyses.

2.4.3.1 *Sensitivity Analysis* – The purpose of such an analysis is to determine if a change in any of the independent (input) variables has a strong influence on the computed safety factors. The sensitivity analysis should consider the effects of different assumptions with regard to loading conditions, material properties and flaw sizes. For example, there may be uncertainties in the service loading conditions; the extrapolation of materials data to service conditions; and the type, size, and shape of the flaw. Confidence is gained in an assessment when it is possible to demonstrate that small changes in input parameters do not dramatically change the assessment results; and when realistic variations in the input parameters, on an individual or combined basis, still lead to the demonstration of an acceptable safety margin. If a strong dependence on an input variable is found, it may be possible to improve the degree of accuracy used to establish the value of that variable.

2.4.3.2 *Probabilistic Analysis* – The dependence of the safety margin on the uncertainty of the independent variables can be evaluated using this type of analysis. All or a limited number of the independent variables are characterized as random variables with a distribution of values. Using Monte Carlo simulation, first order reliability methods or other analytical techniques, the failure probability is estimated. These methods can be used to combine a deterministic FFS assessment model with the distributions prescribed for the independent variable to calculate failure probabilities. Once a probability of failure has been determined, an acceptable level must be established based on multiple factors such as jurisdictional regulations and the consequence of failure.

2.4.3.3 *Partial Safety Factors* – Individual safety factors that are applied to the independent variables in the assessment procedure. The partial safety factors are probabilistically calibrated to reflect the effect that each of the independent variables has on the probability of failure. Partial safety factors are developed using probabilistic analysis techniques considering a deterministic model, distributions of the main independent variables of the model, and a target reliability or probability of failure. The advantage of this approach is that uncertainty can be introduced in an assessment by separately combining the partial safety factors with the independent variables in a deterministic analysis model; the format of the analysis is similar to that used by many design codes. Partial safety factors are only utilized in the assessment of crack-like flaws (see Section 9 and paragraph 2.4.2.3 b)

2.5 Remaining Life Assessment

2.5.1 Once it has been established that the component containing the flaw is acceptable at the current time, the user should determine a remaining life for the component. The remaining life in this document is used to establish appropriate inspection interval and/or in-service monitoring plan, or the need for remediation. The remaining life is not intended to provide a precise estimate of the actual time to failure. Therefore, the remaining life can be estimated based on the quality of available information, assessment level, and appropriate assumptions to provide an adequate safety factor for operation until the next scheduled inspection.

2.5.2 Each FFS assessment section in this document provides guidance on calculating a remaining life. In general, the remaining life can be calculated using the assessment procedures in each section with the introduction of a parameter that represents a measure of the time dependency of the damage taking place. The remaining life is then established by solving for the time to reach a specified operating condition such as the $MAWP$ (MFH) or a reduced operating condition $MAWP_r$ (MFH_r) (see paragraph 2.4.2.2.b).

2.5.3 Remaining life estimates will fall into one of the following three general categories.

2.5.3.1 *The Remaining Life Can be Calculated With Reasonable Certainty* – An example is general uniform corrosion, where a future corrosion allowance can be calculated and the remaining life is the future corrosion allowance divided by the assumed corrosion rate from previous thickness data, corrosion design curves, or experience in similar services. Another example may be long term creep damage, where a future damage rate can be estimated. An appropriate inspection interval can be established at a certain fraction of the remaining life. The estimate of remaining life should be conservative to account for uncertainties in material properties, stress assumptions, and variability in future damage rate.

2.5.3.2 *The Remaining Life Cannot be Established With Reasonable Certainty* – Examples may be a stress corrosion cracking mechanism where there is no reliable crack growth rate data available or hydrogen blistering where a future damage rate can not be estimated. In this case remediation methods should be employed, such as application of a lining or coating to isolate the environment, drilling of blisters, or monitoring. Inspection would then be limited to assuring remediation method acceptability, such as lining or coating integrity.

2.5.3.3 *There is Little or No Remaining Life* – In this case remediation, such as repair of the damaged component, application of a lining or coating to isolate the environment, and/or frequent monitoring is necessary for future operation.

2.6 Remediation

2.6.1 As mentioned in the previous paragraph, under some circumstances remediation is called for. Examples include: where a flaw is not acceptable in its current condition; the estimated remaining life is minimal or difficult to estimate; or the state-of-the-art analysis/knowledge is insufficient to provide an adequate assessment. Appropriate remediation methods are covered within each *FFS* assessment section.

2.6.2 Only general guidelines are provided in this document; each situation will require a customized approach to remediation. Periodic checks should be made to ensure that the remediation steps have prevented additional damage from occurring, and are in a condition that they can be expected to continue to provide protection in the future. The user may need to refer to other documents for detailed remediation procedures; for example, weld repair guidelines can be found in applicable repair codes, such as API 510, API 570, API 653 and NBIC 23.

2.7 In-Service Monitoring

Under some circumstances, the future damage rate/progression cannot be estimated easily or the estimated remaining life is short. In-service monitoring is one method whereby future damage or conditions leading to future damage can be assessed, or confidence in the remaining life estimate can be increased. Monitoring methods typically utilized include: corrosion probes to determine a corrosion rate; hydrogen probes to assess hydrogen activity; various ultrasonic examination methods and acoustic emission testing to measure metal loss or cracking activity; and measurement of key process variables and contaminants. Appropriate in-service monitoring methods are covered within each FFS assessment section.

2.8 Documentation

2.8.1 A Fitness-For-Service analysis should be sufficiently documented such that the analysis can be repeated at a later date. Documentation requirements specific to a particular assessment are described in the corresponding section covering the FFS assessment procedure The following items should be included in the documentation.

2.8.1.1 The equipment design data, and maintenance and past operational history to the extent available should be documented for all equipment subject to a FFS assessment.

2.8.1.2 Inspection data including all readings utilized in the FFS assessment.

2.8.1.3 Assumptions and analysis results including:

- Section, edition, and analysis level of this document and any other supporting documents used to analyze the flaw or damage.
- Future operating and design conditions including pressure, temperature and abnormal operating conditions.
- Calculations for the minimum required thickness and/or $MAWP$.
- Calculations for remaining life and the time for the next inspection.
- Any mitigation/monitoring recommendations that are a condition for continued service

2.8.2 All calculations and documentation used to determine the fitness-for-service of a pressurized component should be kept with the inspection records for the component or piece of equipment in the owner-user inspection department. This documentation will be a part of the records required for mechanical integrity compliance.

2.9 References

2.9.1 Ainsworth, R.A., Ruggles, M.B., and Takahashi, Y., "Flaw Assessment Procedure for High-Temperature Reactor Components," *Journal of Pressure Vessel Technology*, Vol. 114, American Society of Mechanical Engineers, New York, May, 1992, pp. 166-170.

2.9.2 API, Base Resource Document on Risk-Based Inspection, API Publication 581, American Petroleum Institute, Washington D.C., 1996.

2.9.3 Buchheim, G.M., Osage, D.A., Prager, M., Warke, W.R., "Fitness-For-Service and Inspection for the Petrochemical Industry," ASME PVP-Vol. 261, American Society of Mechanical Engineers, New York, 1993, pp. 245-256.

2.9.4 Buchheim, G.M., Osage, D.A., Warke, W.R., Prager, M., "Update for Fitness-For-Service and Inspection for the Petrochemical Industry," ASME PVP-Vol. 288, American Society of Mechanical Engineers, New York, 1994, pp. 253-260.

2.9.5 Kim, D.S., Reynolds, J.T., "Fitness-For-Service Analysis in Turnaround Decision Making," ASME PVP-Vol. 261, American Society of Mechanical Engineers, New York, 1993, pp. 283-294.

2.9.6 Osage, D.A. and Prager, M., "Status and Unresolved Technical Issues of Fitness-For-Service Assessment Procedures for the Petroleum Industry," ASME PVP-Vol. 359, American Society of Mechanical Engineers, New York, 1997, pp. 117-128.

2.9.7 Yin, H., Bagnoli, D.L., "Case Histories Using Fitness-For-Service Methods," ASME PVP-Vol. 288, American Society of Mechanical Engineers, New York, 1994, pp. 315-328.

2.10 Tables And Figures

Table 2.1
Overview of Flaw and Damage Assessment Procedures

Flaw or Damage Mechanism	Section	Overview
Brittle Fracture	3	Assessment procedures are provided for evaluating the resistance to brittle fracture of existing carbon and low alloy steel pressure vessels, piping, and storage tanks. Criteria are provided to evaluate normal operating, start-up, upset, and shut-down conditions.
General Metal Loss	4	Assessment procedures are provided to evaluate general corrosion. Thickness data used for the assessment can be either point thickness readings or detailed thickness profiles. A methodology is provided to utilize the assessment procedures of Section 5 when the thickness data indicates that the metal loss can be treated as localized.
Local Metal Loss	5	Assessment techniques are provided to evaluate single and networks of Local Thin Areas and groove-like flaws in pressurized components. Detailed thickness profiles are required for the assessment. The assessment procedures can also be utilized to evaluate blisters as provided for in Section 7.
Pitting Corrosion	6	Assessment procedures are provided to evaluate widely scattered pitting, localized pitting, pitting which occurs within a region of local metal loss, and a region of localized metal loss located within a region of widely scattered pitting. The assessment procedures can also be utilized to evaluate a network of closely spaced blisters as provided for in Section 7.
Blisters and Laminations	7	Assessment procedures are provided to evaluate isolated and networks of blisters and laminations. The assessment guidelines include provisions for blisters located at weld joints and structural discontinuities such as shell transitions, stiffening rings, and nozzles.
Weld Misalignment and Shell Distortions	8	Assessment procedures are provided to evaluate stresses resulting from geometric discontinuities in shell type structures including weld misalignment and shell distortions (e.g. out-of-roundness, bulges, and dents).
Crack-Like Flaws	9	Assessment procedures are provided to evaluate crack-like flaws. Solutions for stress intensity factors and reference stress (limit load) are included in Appendices C and D, respectively. Methods to evaluate residual stress as required by the assessment procedure are described in Appendix E. Material properties required for the assessment are provided in Appendix F. Recommendations for evaluating crack growth including environmental concerns are also covered.
High Temperature Operation and Creep	10	Assessment procedures are provided to determine the remaining life of a component operating in the creep regime. Material properties required for the assessment are provided in Appendix F. Recommendations for evaluating crack growth including environmental concerns are also covered.
Fire Damage	11	Assessment procedures are provided to evaluate equipment subject to fire damage. A methodology is provided to rank and screen components for evaluation based on the heat exposure experienced during the fire. The assessment procedures of the other sections of this publication are utilized to evaluate component damage.

Table 2.2
Overview of Data Required for Flaw and Damage Assessment

The following data are required for most types of Fitness-For-Service assessments and it is recommended that this completed table accompany the data table completed for the specific damage type which are located in the respective section.

Equipment Identification: ______________________________
Equipment Type: _____ Pressure Vessel _____ Storage Tank _____ Piping Component
Component Type & Location:
Design Code: _____ ASME Section VIII Div. 1 _____ ASME Section VIII Div. 2 ASME B31.3
API 650 API 620
other: ______________________________
Material of Construction (e.g. ASTM Specification): ______________________________
MAWP: ______________________________
MFH: ______________________________
Minimum Required Wall Thickness: ______________________________
Temperature: ______________________________
Cyclic Operation: ______________________________

Type of Damage
Metal Loss – General: ______________________________
Metal Loss – Local: ______________________________
Metal Loss – Pitting: ______________________________
Blisters: ______________________________
Misalignment: ______________________________
Dent: ______________________________
Bulge: ______________________________
Crack-Like Flaw: ______________________________
Creep Damage: ______________________________
Fire Damage: ______________________________

Location of Damage (provide a sketch)
Internal/External: ______________________________
Near weld: ______________________________
Orientation: ______________________________

Environment
Internal: ______________________________

External: ______________________________

Repair and Inspection History

Operations History

Future Anticipated Operations

Figure 2.1
FFS Assessment Procedures For Various Degradation Classes

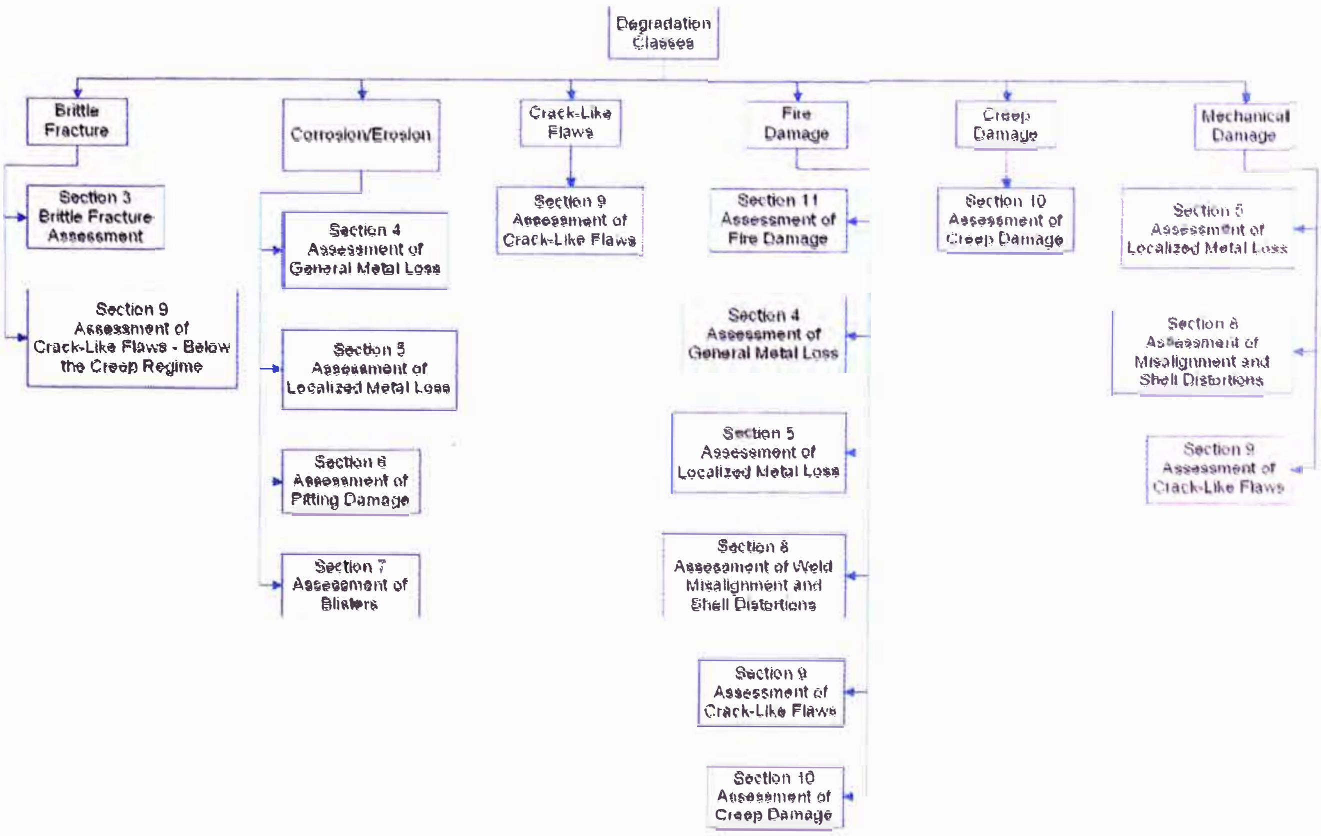
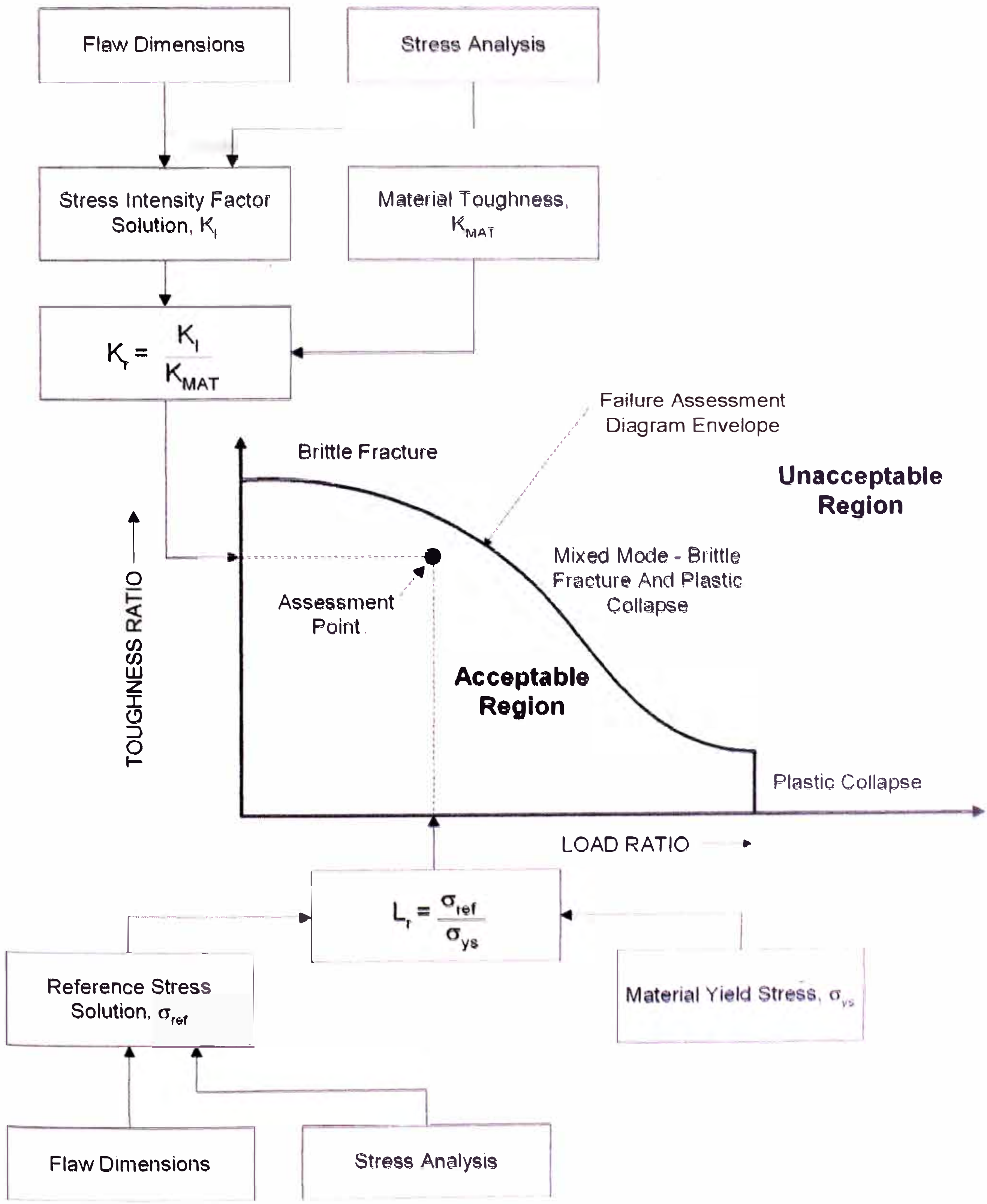


Figure 2.2
Overview Of An FFS Analysis For Crack-Like Flaws Using The Failure Assessment Diagram



2.11 Example Problems

Example problems are included for each Section of this document which contains FFS assessment procedures. The example problems are provided to illustrate the application of the rules and evaluation procedures for a Level 1 and/or Level 2 Assessment. Example problems are provided in both metric and English units.

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The American Society of
Mechanical Engineers

THE ART OF CHECKING PIPE STRESS COMPUTER PROGRAMS

Liang-Chuan Peng
Peng Engineering
Houston, Texas

...RACT

...With the computer getting more and more soph... ...ed, the chance of getting a bug in a program ...misapplication in an analysis also becomes ...and more likely. Analysts need some rules of ... to quickly spot problem areas and to make a ...check if necessary. This paper outlines some ...general rules used in checking boundary con... ..., unbalanced forces, and irregularities. It ...ses specific examples to demonstrate the ...ng of some elementary functions. Special dis... ...ns are given on advanced features such as ...t friction, thermal bowing, and expansion ... elements.

...DUCTION

...With the new requirements given on the design ...odern plant piping, the only practical tool for ...sign analysis is the computer. The computer ...m designed for pipe stress analysis gets more ...re sophisticated every day. Some programs ...one through several generations of develop... ...mploying completely different background of ...nel. The new generation normally will not ...he good work done by their predecessors. ..., they make layers of shells around the exis... ...rk. The completed program becomes very ...nized. Therefore, it is safe to say that a ... pipe stress computer program is bound to ...ome inconsistencies.

...ipe stress analysts are normally too timid in ...ging a well established computer program. ...r, if we recognize that to err is computer ...m, we may be able to more objectively ensure ...lity of our analysis. It is important to real... ...t everything has its so called norm. In other ... if something looks unrealistic then it probably is unreal. Therefore, it is important to be able to look at the output and point out the irregularities that might exist. That is the art. From time to time we have seen some experienced engineers who are able to judge whether a system is satisfactory just by looking at the model. The computer analysis is just a confirming check. However, they are the exceptional rather than the normal.

The inconsistent results in an analysis comes either from the bug in the program or from the mis-application of the program. Nowadays, people like to boast that you don't even need to read the manual to use their computer program. The so called user friendly is probably what they intended to say, but somehow the impression they give is not. You type in some data, then you get some results. It sounds easy, but is scary. To ensure a good analysis the analyst has to have at least a clear picture of what the program functions are. He or she should also be able to spot the inconsistencies when they occur.

PROGRAM VERIFICATION

A program is systematically verified before being released for production. The verification involves almost every step of the program's operation and function. The results of the verification are documented in the verification reports. This is the function of the program developer and should not be a burden to the users.

Verification by the user is occasionally required by the inhou e QA procedure, or to simply satisfy the curiosity of the user or the boss. To an analyst, to be able to personally verify a couple of analyses will definitely increase his or her confidence in the program. The most common approach of the verification is to check against known results. The book by Kellogg Company [1] contains quite a few hand calculation results which can be ch ck d against the ansion stress calculation. A more formal calcu- on intended to be a benchmark was published by ΛE [2] in 1972. Unfortunately, this benchmark blem contains some misprints, which have never n corrected, and also the unusual non-circular ss section elements. Because of these difficulties, problem has created a huge frustration in the ng industry. Everywhere, engineers are trying nake a comparison in vain. Later in 1980 U.S. C published a set of representative piping bench- k problems [3]. This set of problems was taken n real systems laidout in nuclear power plants. s mainly used to check the earthquake analysis g the response spectra method.

The benchmark problems check only the general aviors of the program. The general behavior of iven program differs very little from the original k box on which most of the programs are based. refore, very little deviation shall be expected n these tests. The most important items to be cerned with are the ones particular to individual grams. These items need to be checked very rately.

IATION

In comparing the test results with published or chmark results, the relative deviation is used. term error is not used because the difference ht be caused by the error of the published or so ed known results. Even the so called exact solu- might have some seemingly insignificant terms red. However, if the deviation is small then e is a good chance that both the testing program the benchmark are correct. This is more so n then testing program uses an entirely different tion technique than that used by the benchmark.

In evaluating the deviation, some common sense to be applied to avoid unnecessary arguments. given quantity, R, whose exact solution is shwon igure 1 (a). Its corresponding result, R', from test program may be shifted to as shown in re 1 (b). Then by some methods of evaluation,

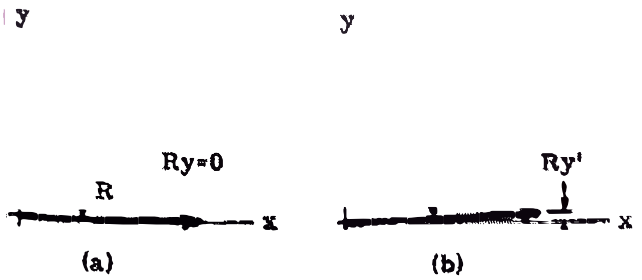


Figure 1, Standard Deviation

ay be concluded that there is no comparison at Because the deviation is essentially infinite on onent Ry. But we all know that the real differ- between the two solutions is very small. This be easily proved because if we rotate the axes 5 degrees, the deviation will almost disappear completely. The point is that a number is meaning- less if its quantity is entirely dependent of the sel- ection of the coordinate axes. Therefore, it is im- portant to have the deviation properly defined as follows :

$$dev(Ry) = (Ry' - Ry) / Ry \quad \text{(Meaningless)}$$

$$dev(Ry) = (Ry' - Ry) / R \quad \text{(Local)}$$

$$dev(Ry) = (Ry' - Ry) / Ro \quad \text{(Global)}$$

Where R is the resultant quantity at the point of interest, and Ro is the maximum resultant quantity in the entire system analyzed. The global deviation is introduced, because at a given point the resu tant quantity itself may be insignificant. Whether it is significant or not, the tool to measure is the global comparison. The evaluation of the local deviation requires some personal judgement, but the global deviation should be limited to about 10 percent.

BOUNDARY CONDITIONS

The first step in quick checking an analysis is to make sure that the results match the boundary conditions of the systems. This can be done easily with the help of a good output arrangement. Most computer programs have a separate report for the anchor and support forces and moments as shown in Table 1. For this particular one [4] the friction and the pipe displacements are also given. This makes the checking of the boundary condition very easy.

By using reports such as Table 1, the boundary conditions can be checked directly by looking at the pipe displacements. At an anchor point the pipe displacement should be the same as the input d spl- acement, and at the limit stop location the pipe displacement shall be equal to or smaller than the gap specified. However, it should be noted that the support displacement specified in the input is for the support structure. The actual pipe displacement at that point may or may not be the same as the sup- port depending on the rigidity of the support. If the support is rigid then the pipe and the structure will have the same displacement. But if the support is flexible then the pipe displacement and the support movement are different as shown in Figure 2.

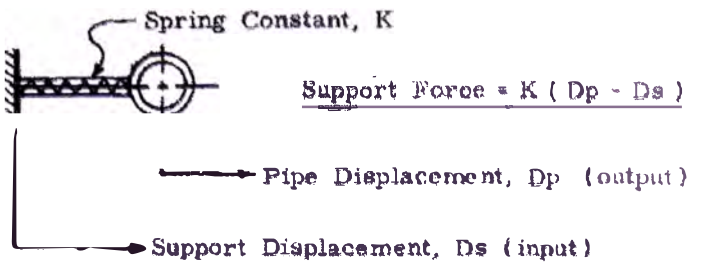


Figure 2, Support and Pipe Displacements

88/12/12. PENG ENGINEERING - SIMFLEX-II (REL-4.0) (ANSI-B31.3) PAGE 11

SAMPLE PROBLEM, 1989 ASME/JSME PVP CONFERENCE
FRICTION FACTOR 0.4 ON ALL VERTICAL SUPPORTS

CASE 1 TH + WT RESULTS LOAD= THM,WGT,BNG,FOR,UFM,CSP,PRES
****

*** ANCHOR AND SUPPORT FORCES - INCLUDING FRICTION (ACTING ON SUPPORT) ***

		SUPPORT FORCE AND MOMENT						FRICTION				DEFLECTION			
SUPT	DATA	*** FORCES (N) ***			--- MOMENTS (N-M) ---			-- FORCES (N) --			T(N-M)	(MM)			NOTES
TYPE	PT	FX	FY	FZ	MX	MY	MZ	FFX	FFY	FFZ	FMT	DX	DY	DZ	
VESS	5	-4552	2898	2491	-1951	-739	-763	0	0	0	0	-.0	-3.6	.0	
	LOCAL	2898	-4552	-2491	-739	-1951	764								
		RADIAL	MERIDN	TANGNTL	TORSION	CIRCUMF	LONGTDL								
SPRG	20	0	-6845	0	0	0	0	0	0	0	0	2.9	-6.1	-1.0	
LSX	65	0	0	0	0	0	0	0	0	0	0	3.0	-.0	34.4	INACTIVE
LSX	65	1401	0	0	0	0	0	0	0	0	0	3.0	-.0	34.4	
NLY	65	0	-10154	0	0	0	0	353	0	4047	0	3.0	-.0	34.4	
STX	70	1389	0	0	0	0	0	0	0	0	0	.0	-.0	16.4	
NLY	70	0	-5041	0	0	0	0	0	0	2017	0	.0	-.0	16.4	
LSX	90	-7646	0	0	0	0	0	0	0	0	0	-6.0	-.0	-1.7	
LSX	90	0	0	0	0	0	0	0	0	0	0	-6.0	-.0	-1.7	INACTIVE
NLY	90	0	-450	0	0	0	0	-173	0	-47	0	-6.0	-.0	-1.7	
LSZ	100	0	0	-7647	0	0	0	0	0	0	0	-7.7	1.8	-6.0	
SPRG	105	0	-2976	0	0	0	0	0	0	0	0	.1	6.8	4.8	
VESS	120	7043	-2298	-2871	-3087	-5748	-7360	0	0	0	0	.0	2.0	8.0	
	LOCAL	2299	-2871	-7043	5748	-7369	3088								
		RADIAL	MERIDN	TANGNTL	TORSION	CIRCUMF	LONGTDL								
SPRG	150	0	-1535	0	0	0	0	0	0	0	0	-3.2	29.5	40.4	
SPRG	160	0	-6599	0	0	0	0	0	0	0	0	11.4	17.1	.0	
ITZ	160	0	0	2009	0	0	0	0	0	0	0	11.4	17.1	.0	
ANCH	180	2188	-371	4	-1267	3501	-9188	0	0	0	0	-2.0	1.5	.0	
NET FORCES		-177	-33371	-6014				180	0	6017	0				

For systems which include the support friction, the force and the direction of the friction can be readily checked against the normal support force ... the pipe movement.

...EM EQUILIBRIUM

The pipe stress analysis result, regardless of ... method used to get it, shall still conform the ... of equilibrium. The summation of the forces and moments applied at a given point shall be zero, and ... summation of the forces and moments applied to ... entire system shall also be zero.

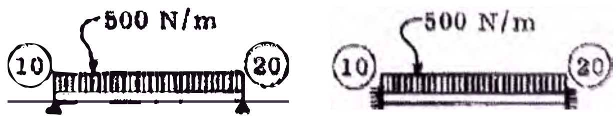
Needless to say that to check the equilibrium ... every point in a system manually is not practi... But if there is any doubt about a given point, it can be checked manually. In well designed ...ams, there is a scheme to automatically check ...ecord the equilibrium of all the points in the ...m. The analyst should always look for messages ... if any significant unbalanced force has been ...ed. A significant unbalanced force always sig... a problem in the analysis.

The total system equilibrium can be checked by the support load table given in Table 1. In this the total system forces are summarized at the ...n. In a system without any external forces ...ed explicitly, the vertical force should be the ... as the total weight load. The horizontal force ... be equal and in the opposite to the horizontal ...n force. This is very fundamental, but can be ...d by even the expert. For instance, in Problem ... of the ASME 1972 verification book [2], one of ...lutions presented has an apparent error in the ...action support force. This can be checked by ...w of equilibrium but the writer preferred to ...t explained as the difference of the programs in the comparison.

...ENTARY FUNCTIONS

It is true that the pipe stress computer prog...s designed to handle an assembly of pipes. ...er, it should still be able to calculate some simple situations. A pipe stress program con... of mainly two types of elements, straight pipe ...urved pipe. If the program is to function pro... then these two basic elements have to function ...ly. Therefore, if we can check out the basic ...n of these two elements, we will have more ...nce in the program.

The straight pipe element is just a beam. Its ...n can be checked against the beam formula ...e learned from text books. However, there ... few differences that need to be mentioned. One ...m is that some program approaches are not ...ar as the text book. Take the two uniformly ... beams as shown in Figure 3 for example, if ...n them through the computer you may find ...ere is no stress at all in one, or even both, cases. In (a), because the program evaluates ...e stresses at node points 10 and 20, and the ...es at these two points happen to be zero. ... is no reason that a program can not be pro-



(a) Simple Beam (b) Fixed Beam

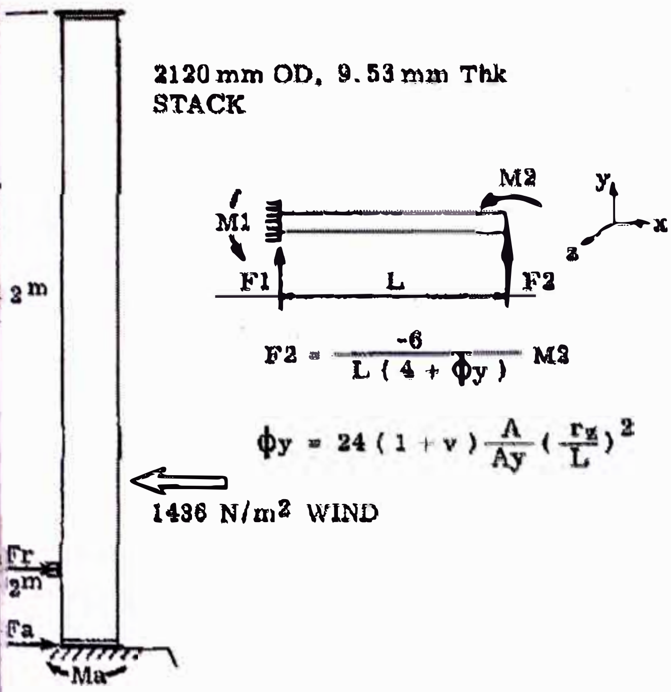
Figure 3, Beam Paradox

grammed to find the maximum stress of the entire beam element, but this is not done normally. The reason is that a complete pipe stress analysis can involves several load cases. If all the maximum stresses at each element are to be combined together regardless of their location, then the calculation can become overly conservative. Also the simple beam condition does not really exist in a piping system. If required, an additional point at the midspan can be entered. The case (b), on the other hand is somewhat more troublesome. In some finite element programs the uniform load is divided into nodal loads which are applied at the node points. In the fixed beam case, the uniform load is divided into two concentrated loads which are applied at the ends. This will produce the proper reaction force, but no reaction moment nor beam stress. So, meprograms of this type are still widely used in the piping industry. Analysts should make themselves aware of the problem involved.

Another item that needs to be mentioned is shear deformation. The shear deformation is not normally included in the beam formula we use, but it is included in most pipe stress programs. There is not much difference if the length of the beam is at least several times the cross sectional dimension. However, if the beam length is short the difference can be very great. Figure 4 shows a stack guided at a very short distance from the base to resist the wind. The problem is reduced to a fixed-supported beam applied with an end moment. As can be seen from the results tabulated, the shear deformation term has a very significant effect on the ...other and support loading if the guide is rigid.

For the curved pipe element, the formula given by J. E. Brock [5] can be used for cross check, if the cumbersom calculation can be managed. An alternative way is to divide the bend into multiple sections to see if the results agree with those of the undivided bend.

The curved pipe element involves fle...ibility and stress intensification factors. These factors are further influenced by the presense of flanged ends and internal pressure. In checking the int...nsification factor it is necessary to find out the program option in implementing the pressure effect. It is also desirable to understand the implications of the application. The pressure will tend to make the system more stiff, thus resulting in higher support loads against the thermal expansion. On the other hand it also tends to make the cross section more difficult to ovalize, thus reducing the stress intensification. The



culation hod	Support Fr (N)	Anchor Fa (N)	Anchor Ma (N-m)
h Shear Deform. ld Restraint	77,200	-53,700	63,400
hout Shear Deform id Restraint	182,300	-158,800	-64,700
h Shear Deform. 00 N/mm Rest.	1,800	21,600	155,300
hout Shear Deform 00 N/mm Rest.	1,500	22,000	155,700

Figure 4, Effect of Shear Deformation Term

blem is that when the pressure is removed the perature will likely to stay at near operating for e time. At this time the pipe moment is not uced but the pressure is not there to help prevent lization. Therefore, the logical application is to into account the increased stiffness, but not the reased stress intensification.

CIAL FEATURES

Each pipe stress computer program has its special features. These features are normally available in benchmark problems. Their functions l to be checked by special schemes. Since it is not sible to cover all the features, this discussion concentrate on three popular items. They are port friction, bellow elements, and thermal bow-

Support Friction

The support friction has a very significant ef- on the analysis results in certain cases. The areas most sensitive to the friction are rotating equipment piping, long offsite piping, and transmission pipe lines. For instance, at a large rotating equipment, the friction due to a single support can often determine if the piping load exceeds the allowables or not.

There are different ways of implementing the friction effect in the program, but they are not all equal. Some methods require more computer time but are more inherently stable. Others are quick but prone to be unsatble. A detailed discussion on this subject is given in a separate topic [6]. In this paper the discussion is limited to the quick check of the results.

The validity of friction application depends on the type of the system analyzed. If the system is relatively rigid then the analysis tends to be correct regardless of which method is used. On the other hand, if the system is relatively flexible then the correct analysis can only be achieved with certain methods. This is because in a flexible system the friction not only affects pipe force, it also has the potential of changing the direction of the movement. To check the friction feature, it needs to check its application on a flexible system. With a support load report similar to Table 1 the function of the friction can be checked easily by the following steps:

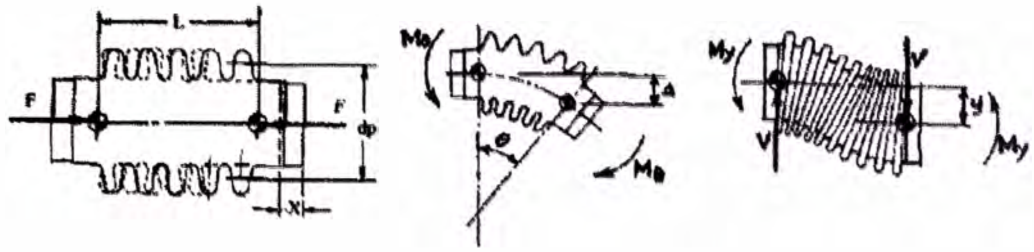
(1). If the piping is moving, then the resultant friction force should be equal to the normal support force multiplied by the friction factor. The direction, acting on the support, should be the same as the pipe movement. It reverses when acting on pipe.

(2). If the piping is stopped by the friction and not moving then the friction force should be equal or smaller than the full friction force calculated in (1).

(3). Most importantly, the above friction force is applied to the system. This can be checked by balancing the nodal forces at the support location. With a support load report similar to Table 1, the application of the friction can be checked by comparing the total friction force against the total system force. They should be the same if no other external force is applied to the system.

(b). Bellow Element

Bellow expansion joints can be simulated by the conventional zero length flexible connectors. However, to be able to represent the versatility of the bellow arrangements, the use of bellow elements is preferred. With the bellow element, the program can easily simulate all the common bellow expansion joints such as single bellow, tied bellows, universal joints, and pressure balanced universal joints. The program will correctly apply the flexibility of the joint in all the translational and rotational directions. It also applies the proper pressure thrust force at the end of the bellow. The more advanced program can also combine all three dimensional motions to calculate the equivalent maximum axial displacement



(1) Axial Motion (2) Rotation (3) Lateral Motion

Figure 5. Elementary Function of Bellows

convolution. This is the vital information used he manufacturers to check the acceptability of r bellows.

Implementation of the bellow element involves e tricky maneuvers, but to check is simple. The ansion Joint Manufacturers Association (EJMA) a set of formulas [7] that can be used readily checking the function of the bellow element. se formulas are copied below for easy reference.

$$ex = x / N$$

$$e\theta = \theta \cdot dp / (2N)$$

$$ey = 3\ dp \cdot y / (N \cdot L)$$

$$F = fw \cdot ex = (fw/N) \cdot x = Ka \cdot x$$

$$M\theta = fw \cdot dp \cdot e\theta / 4 = [(fw/N) \cdot dp^2 / 8] \cdot \theta$$

$$V = fw \cdot dp \cdot ey / (2L) = 1.5\ (fw/N)(dp/L)^2 \cdot y$$

$$My = fw \cdot dp \cdot ey / 4 = [0.75\ (fw/N) \cdot dp^2 / L] \cdot y$$

re,

- ex = Axial displacement per convol. due to x
- eθ = Axial displacement per convol. due to θ
- ey = Axial displacement per convol. due to y
- x = Differential axial displ. across bellow
- θ = Differential rotation across bellow
- y = Differential lateral displ. across bellow
- N = Number of convolution
- dp = Pitch diameter of the convolution
- L = Effective length of the bellow element
- F = Axial force required to move x
- fw = Axial spring rate per convolution
- Mθ = Moment required to bend θ
- V = Lateral force required to move y
- My = Moment created by y- movement
- Ka = Axial spring rate of the bellow element

From the above formulas it is clear if, for nce, the axial spring rate, pitch diameter, and ellow length are given, then the spring constants l the other directions can be determined. The alent axial displacement per convolution can be found without needing additional data.

In checking the bellow function, a few items to be further explained. As can be seen from ormula, the lateral spring rate is inversely ortional to the square of the bellow length. The deformation expected during operation can have nificant effect on the lateral spring rate. The analyst should try to input the shortest possible length in the analysis. It should also be noted that by laterally moving the bellow not only creates lateral force, but also the bending moment. In bending the bellow, the EJMA formula signifies that a lateral movement as well as a rotation is being created.

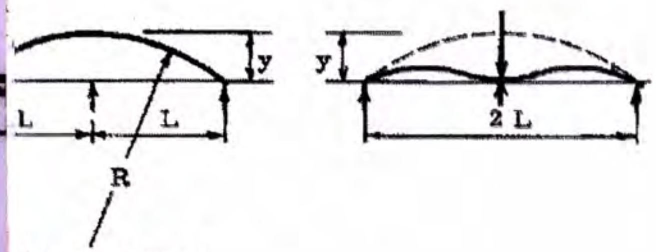
The checking can be done easily by fixing one end of the bellow element and apply the loading or displacement at the other end. This eliminates the trouble of finding the differential displacements of both ends. However, the real function of the bellow can only be evaluated by the checking of the differential movements. Once the elemental function is checked, its application to the piping assembly is not much different from the other elements.

(c). Thermal Bowing

Piping is normally assumed to have a uniform temperature across its cross section. However, due to stratified flow or some other reason, the temperature can vary greatly between the top and the bottom of the pipe. This situation can occur during the startup of large steam lines [8] or cryogenic lines [9]. It can also occur at a petrochemical transfer line hwen it is being quenched or when it has coke forming at the bottom of the pipe. When the temperature around the cross section is not uniform the pipe will form an arc shape. This bowing phenomenon may or may not create damaging stress in the pipe itself depending on the shape of the temperature distribution. If the distribution is linear, then no internal stress is created. If the distribution is not linear then large internal stress may be created. In either case, the bowing has the potential of creating huge displacements and rotations in the pipe. This huge movement can tear off connections if enough flexibility is not provided.

The bowing feature can be checked with two simple steps. Figure 6 shows a two span simply supported pipe. If no clamp or hold-down is installed in the mid-span, the pipe at mid-span will move up due to bowing. The amount of the move-up can be checked against the formula derived using the linear temperature distribution [9]. That is if the difference in expansion rate between the top and the bottom of the pipe is e mm/mm, the pipe diameter is d mm, then the radius of the curvature is $R = d/e$ mm. If the span length is L mm, then the expected move-up displacement $y = R - \sqrt{R^2 - L^2}$ mm.

After the bowing movement is checked, the (b) can be used to check the combined effect. In this case, the mid-span is rigidly held down. The hold down load can be checked by using the simple beam formula applied with a concentrated force at the mid-span. The hold down load should equal the concentrated load with which a mid-span displacement of y mm is created. Of course the weight and other loads should not be included in making this check.



(a) Free Bowing (b) Hold Down Bowing

Figure 7, Bowing Function

STRESS REPORT

A well laid-out stress report can facilitate the checking of the analysis. The stress report should contain, in one continuous printout, all the input, interpretation of the input, generated system information, load case results, and stress and load compliance tables. Other graphical or tabular forms of presentations which are not integral parts of the report should be clearly identified for its association with the stress report.

The most important item to be checked on a stress report is the truthfulness of the mathematical model. This generally refers to the correctness of the input data, but includes also the correct interpretation of the program requirements. A good input echo and a good isometric picture generated directly from the input data can be very helpful.

Input Echo

With the popularity of the menu input approach, input data made by an analyst is converted into another way of expression almost immediately. The so-called input echo printed out by some programs may not even be recognised by the analyst who have prepared the input in the first place. It is, needless to say, a nightmare to the checker. The input echo is the most important document of the report should be readable not only to the machine, but also with the analyst and checker. The best form is the one which preserves all the styles and letters the analyst has entered. Figure 7 shows the echo of the input which is done with the popular piping package.

Faithful Isometric

A good isometric is an invaluable tool for quick checking the mathematical model. The isometric need not be pretty but has to be faithful. It should show all the bends, valves, flanges, and other components. The restraints should be shown in the correct location and also in the correct direction. The nodal numbers should all be identified properly. Figure 8 shows the typical isometric drawing which is printed directly and automatically from the input data.

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5. Brock, J. E., "Expansion and Flexibility", Chapter 4, Piping Handbook, 5th edition, edited by King and Crocker, McGraw-Hill, 1967.

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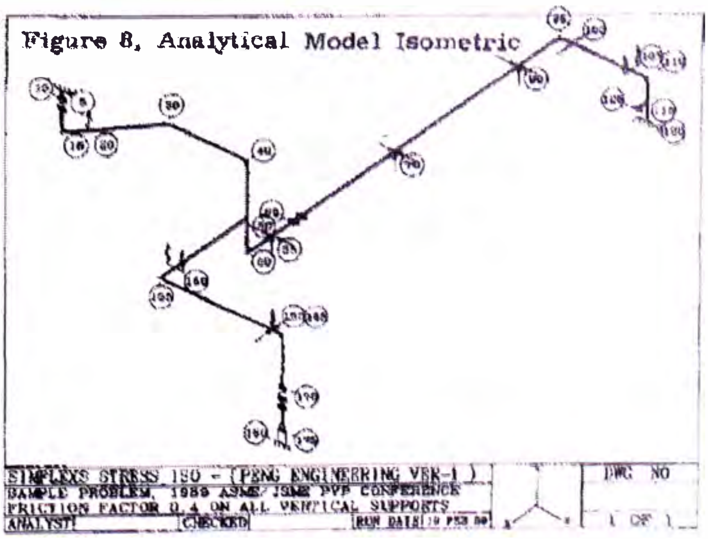
8. Machacek, S., and Zelenka, T., "Design and Operation of Large Diameter (1.67m) steamline," ASME/CSME PVP Conf, Montreal, June 1978.

9. Flieder, W. G., Loria, J. C. and Smith, W. J. "Bowing of Cryogenic Pipelines," Trans ASME, J. Appl Mech, Sep. 1961, pp 409-416.

Figure 7, Input Data Echo

```
50, EL=34.44, TEE
60, EL=33.0, BR
65, Z=-1.22, LSX(-3, 3), NLY(,,0.4,-1)
68, Z=-1, WNF, +FSTV, +WNF
70, Z=-5, STX, NLY(,,0.4,-1)
90, Z=-6, LSX(-6, 0), NLY(,,0.4,-1)
95, Z=-2.0, BR
100, X=0.61, LSZ-6
105, X=2.6, SPRING=2
```

Figure 8, Analytical Model Isometric



Thermal insulation and pipe stress

An often-overlooked function of insulation in piping designs is to mitigate weather effects

.. C. Peng, Peng Engineering, Houston, Texas; and
T. L. Peng, The M. W. Kellogg Co., Houston, Texas

Thermal insulation is mainly used to reduce heat loss and noise level. It is also used to prevent burn injuries. However, a lesser-known yet important function of thermal insulation is to reduce pipe stress.

In a petrochemical/refinery complex, piping systems are constantly subjected to the abuses of weather and environmental changes. Occasional rain showers, for instance, can generate very high pipe stresses. Repeated occurrences can eventually lead to pipe failure. Some typical situations when a seemingly innocent rain shower may damage a pipe are discussed. Some field problems can actually be solved with a simple application of insulation. Unfortunately, the engineer who relies only on a computer to design and analyze piping will miss out on this kind of common sense.

A case history. Fig. 1 shows a piping system used to transfer a hot gas mixture from the primary reformer to the secondary reformer in an ammonia fertilizer plant. The gas mixture was operating at approximately 1,500°F and 500 psi. The main portion of the piping was constructed with thick, internal refractory-insulation to reduce the pipe metal temperature to approximately 200°F. This section of the piping is called the cold-wall portion, in contrast to the externally insulated hot-wall portion, whose metal temperature is close to the fluid temperature of 1,500°F. Carbon steel is used for cold-wall piping, and alloy steel is used for hot-wall piping.

This plant was built in the early 1970s. Most of the piping was designed using a cold-wall approach due to economic benefits of using common carbon steel material and the desire to reduce thermal expansion. Hot-wall construction was used only at the piping segments connecting to the reformers. Four transition joints connected cold-wall pipe to the hot-wall pipe.

After operating without problems for the first 10

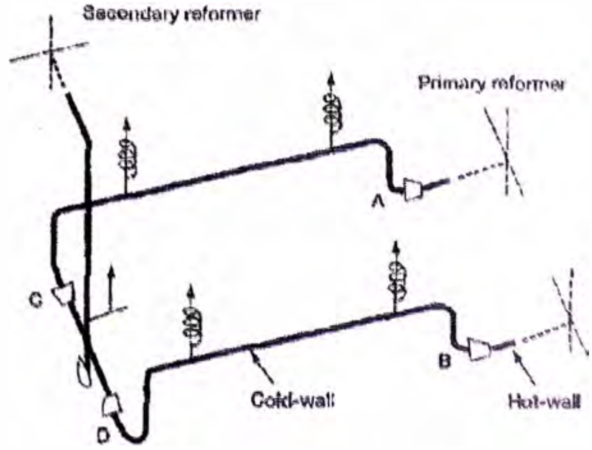


Fig. 1. Ammonia plant reformer piping.

years, maintenance was performed on the piping to repair the refractory and to replace the hot-wall special-alloy pipe. Strangely, after this revamp, the miter elbows at points A and B near the cold-wall/hot-wall junctions developed leaking cracks about every four months. The revamp contractor was called in to investigate the problem.

As expected, their first step was to input the system into a computer for a stress analysis. However, all the computer indicated was that everything was in good shape. So the contractor modified some springs based on the computer analysis. Ironically, the original design was probably done without the help of a sophisticated computer program. After spending thousands of dollars replacing the spring hangers, the system still faithfully failed about every four months. You can bet it was very frustrating for the plant engineers.

After this exercise, the plant engineers decided to get help from a large contractor. For unknown reasons, the original contractor was not called. A large contractor naturally has a greater depth of engineers. The contractor first performed a series of heat-transfer calculations to check the cold-wall section's metal wall temperature. They understood that a good analysis needs good data. With the newly calculated metal wall temperature, they made a refined stress analysis of the system. Again, the computer said everything was in good order. Nothing could be done, or needed to be done.

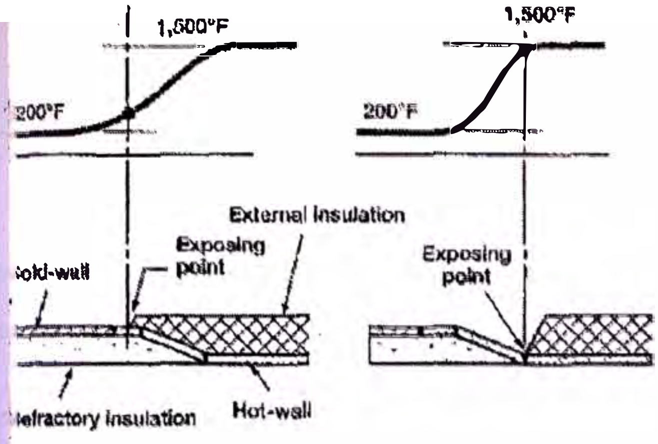


Fig. 2. Cold-wall to hot-wall junctions in the a) correct arrangement and b) wrong arrangement.

...vever, out of professional conscience, or maybe to ...ify the fee for their service, they recommended that the spring hangers be replaced with constant-effort ...ngs. Result: the same frequent failure except the ...em changed into a pile of twisted spaghetti piping. ... constant-effort spring hangers had a difficult time ...ling the system together.

...he problem was later solved by a small modifica... in the insulation arrangement. It all started with ...sual look at photos and some casual discussions ...h the plant engineers. Subsequently, all the origi... spring hangers were reinstalled. The fancy con...t-effort spring hangers were removed and destined ...ll the warehouse.

...d-wall/hot-wall junctions. Joints connecting the ...-wall pipe and hot-wall pipe require special arrange...nts in both pipe material and insulation. Just like a ... contractor who knows how to treat a shingle-to-... junction, an experienced piping contractor knows ...tly how the cold-wall to hot-wall junction must be ...structed. A small mistake in the detail by the roofer ...nally results in constant roof leakage.

...ig. 2 shows two arrangements of cold-to-hot junc...s. The two do not really appear different to the inex...enced eye; however, the consequence is the differ... between failure and safety. Fig. 2a shows the ...ect arrangement, and Fig. 2b shows the often-used ...ng arrangement. The original system in Fig. 1 was ...structed with the arrangement shown in Fig. 2a, ... the drawing somehow had indicated Fig. 2b. Due ...e revamp contractor's inexperience, the junctions ... constructed as Fig. 2b, causing constant cracks ...e pipe.

...he pictures are not obvious, but once the tempera... profiles are constructed using some common sense, ... become very clear. In case 2a, the pipe wall tem...ture decreases gradually from 1,500°F to the design ...-wall temperature of 200°F. Ensure that the wall ...perature at the dissimilar weld location is below ...°F to avoid high thermal stress due to different ...ansion rates between carbon steel and high-alloy ...l.

...onversely, the temperature in case 2b drops much ...r than in 2a. The focal point is at the location called ...xposing point. This is the highest temperature point

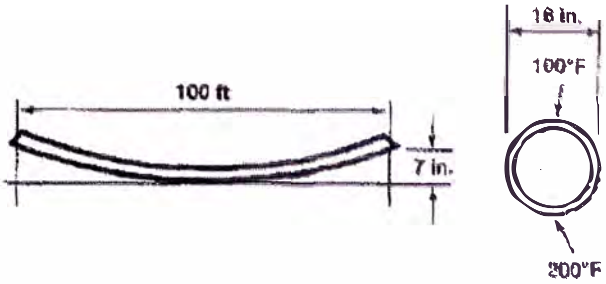


Fig. 3. Thermal bowing effect.

exposed to open air that can be quenched by a rain shower. In 2a, the temperature at the exposing point is about 400°F, whereas in 2b, the temperature at the exposing point is close to 1,500°F. It is clear that a much higher thermal stress will be generated in case 2b.

Thermal stress. The thermal stress caused by a temperature gradient or discontinuity normally does not produce any gross distortion. Therefore, it is often overlooked. However, if we appreciate how high a stress can be generated, we would pay more attention to it. The magnitude of the thermal stress can be roughly estimated by:

$$S = E\,a\,T \tag{1}$$

where S = thermal stress, psi
a = expansion rate, in./in./°F
E = modulus of elasticity, psi
T = temperature difference, °F

For carbon steel pipe, a 500°F discontinuity will mean a 105,000-psi thermal stress. The stress will be even greater for a stainless steel pipe due to its higher expansion rate. This kind of stress greatly exceeds the safe reference value of twice the yield strength and should not be ignored. It is difficult to estimate exactly how much of a temperature gradient can be generated by a rain shower, but any local area with a metal temperature of 500°F or higher should be protected.

Showers. In a petrochemical/refinery complex, some locations are especially susceptible to rain shower damage:

High-temperature flange connections. The ASME B31.3 piping code contains a clause that stipulates that the design temperature of uninsulated flanges, including those on fittings and valves, can use 90% of the fluid temperature (par. 301.3.2). Thus, some high-temperature flanges are purposely designed without insulation covering. Although the situation has changed since the 1970s energy crisis, many uninsulated high-temperature flanges are still in the field. These can easily develop leaks and cracks after repeated rain showers. It should be cautioned, however, that insulating these flanges now might make them unacceptable to the code due to a design temperature increase.

Expansion joints. It could be for ease of inspection or just due to logistical problems in construction, but many expansion joints are not insulated. Once it is built that way, the plant engineers will not change it, even if it develops cracks. They are very often incensed at know

t the simple application of insulation will solve blem. They thought that it was something they uffer through due to technological impossibilities event these joints from being properly constructed.
1-temperature valves on the cold-wall sec- ue to practicality, valves used in the cold-wall of hot-fluid applications often use hot-wall (meaning without internal refractory insula- hese valves are often left uninsulated for the easons as high-temperature flange connections. ation is not desirable, then at least some type shield should be provided.

al bowing. For thermal stresses, we are con- mainly with the high-temperature areas. These areas that can create high enough thermal s to cause cracks. However, in some systems, ough the temperature is not high, another ther- ect may create a different kind of problem. This esser-known bowing effect.
xample, assume we have a 16-in. gas line that sulated and operates at 200°F. During a sum- ower, the pipe's top may suddenly quench to while the bottom maintains 200°F. This 100°F on the top produces a shrinkage of 0.00065 pipe surface. This shrinkage will bend the pipe arc with a radius of curvature equal to R = 065 = 24,615 in. This bowing effect (Fig. 3) can lly lift the ends of a 100-ft long pipe up 7 in. h the actual lift will be greatly reduced by the eight, its significance cannot be ignored.

Damage caused by thermal bowing is often very ghostly. It normally happens without anybody actually seeing it. In the above example, when the shower starts, the ends move up and possibly tear off some supports or small connections. However, when the rain stops or when the temperatures even out, the pipe returns innocently to its initial position. It leaves the damage without giving any clue of the cause. ■

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̄oward more consistent ipe stress analysis

Presented are some guidelines in applying stress intensification factors to piping weight loading and small branch connections. It is hoped this information will alleviate the controversy and lead to standardization

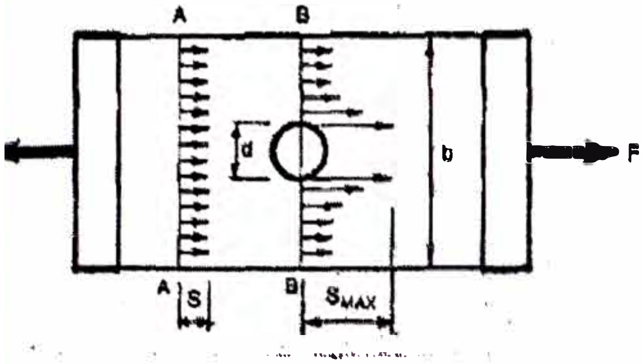
. Peng, Consulting Engineer, Houston

ɪ PIPING DESIGN HISTORY, 1955 is a monumental year. hat year the stress range concept was formally recog-d by the Code for Pressure Piping[1] as the basis for uating thermal expansion stress. Although the code been expanded and clarified over the years, there still unsettled arguments regarding application of the ı in certain areas. Two areas where inconsistencies still : are stress intensification factors for weight and r steady loadings and stress intensification factors small branch connections. These areas will explored g with suggestions for applying the code.

tress intensification factor for weight and other ıdy loadings. The stress intensification factors given he code are intended for flexibility analyses. No ific intensification factor for weight and occasional ings is mentioned in the Chemical Plant and Petro-ı Refinery Piping Code.[2] Due to this tacit position he code, piping designers are divided in actual prac-Some designers will apply the code stress intensifica-factors to all categories of loads, while many other gners tend to ignore the stress intensification factors pletely in steady load analysis. One component ac-able to one designer can be rejected by others due ifferent opinions in the interpretation.

• **Small branch connections.** The stress intensification factors given in the code for branch connections are derived from full size branch connections. These factors, although applicable to small branch connections, can become excessively conservative for small connections on big pipes. Because of the apparent overconservatism, designers often ignore stress intensification factors at small branch connections. However, practices are never consistent. For instance, it is easy to see that the stress intensification due to a ¾-inch connection can be ignored in the analysis of a 20-inch header, but for a 3-inch connection, the factors to apply will differ among designers.

Stress intensification factors given in the code for branch connections can be too conservative for small connections on large pipes



F: 1—Stretching a bar with a small hole

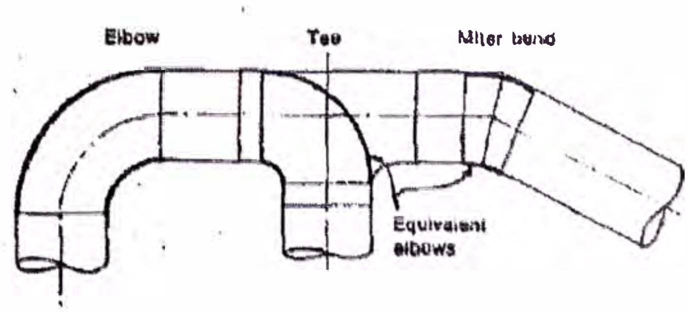


Fig. 2—Equivalent elbows

nese two examples are related to the application of s intensification factors. Apply or not to apply very means several times difference in the allowable loads. se are determined solely by designers' personal preferences and inclinations. A more consistent approach s to be developed and adopted.

STRESS INTENSIFICATION

hen a structural member is stretched, stress in the uniform section can normally be calculated by simformulas, but the stress in a locally notched or stif- d discontinuous section is either very complicated or ossible to calculate. For practical design purpose, s at the discontinuous section is estimated by applying ress intensification factor over the stress calculated he main uniform section. This stress intensification be derived theoretically or determined by test.

a structural discontinuity, stress intensification can uite different for different types of loading. Fig. 1 s a long rectangular bar with a small hole in the lle of the section. At Section A-A outside the influence nd fixtures and the hole, the stress is uniformly distributed at a magnitude of $S = F/(bt)$. But at Section due to discontinuity in strain flow, the stress is unly distributed. A maximum stress, S_{max}, of about three s the uniform stress occurs at the edges of the hole. stress decreases very rapidly at points away from the of the hole. Theoretically, the hole has created a intensification factor of three, but its significance fferent for different materials.

or a brittle material such as glass, the hole will degrade bar to one-third its original strength because it fails on as the maximum stress reaches failure stress. Piping rials, on the other hand, are normally very ductile, a considerable amount of yielding takes place before member fails. With ductile materials the stress inification needs to be interpreted in two different gories, namely steady and cyclic.

ady loading. Under steady loading the highly local- stress concentration will be redistributed to the adnt area once the local stress reaches the yield point. ntually the load will spread evenly to the whole cross- on before the bar fails. The important stress is the redistributed stress prior to the failure. Since the redistributed stress is essentially the average stress, the stress intensification factor for steady loading is

$$i_s = \frac{F/(b-d)}{F/(b)} = \frac{b}{(b-d)} \qquad (1)$$

which is entirely due to reduction of the cross-sectional area.

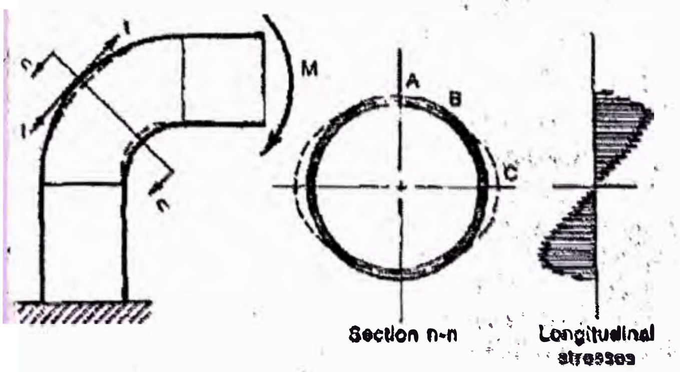
Cyclic loading. Under cyclic loading the member fails due to fatigue. Since the primary measure of fatigue failure is the local strain range per cycle, redistribution of stress due to plastic flow is not very important. Therefore the stress intensification factor for cyclic loading is

$$i_c = S_{max}/S \qquad (2)$$

which is the measure of the maximum local strain. S_{max} is the maximum equivalent elastic stress rather than the actual stress.

Elbow stress intensification factor. In piping stress analysis, the elbow stress intensification factor is particularly important not only because the elbow constitutes a major portion of the system but also because it is the basis for deriving the stress intensification factor for other component shapes. For instance, Markl[8] successfully used elbow analogy to correlate his fatigue test results on tees and miter bends. Using the equivalent elbows as shown in Fig. 2 and making adjustments for actual crotch radius and thickness, a set of stress intensification factors was constructed using a single flexibility characteristic parameter, h. A detailed discussion on elbow characteristics is beneficial in understanding the general trend of all components.

An elbow behaves very differently from a straight pipe in resisting bending moments. When a straight pipe is bent, its cross-section remains circular and stress increases linearly with distance from the neutral axis. However, when an elbow is bent as shown in Fig. 3, the cross-section deforms to an oval shape. This ovalization is due to less rigidity at extreme fibers in the tangential (-) direction, and less energy being needed for the elbow to assume an oval shape than to maintain a circular cross-section. Top and bottom portions of the pipe wall simply buckle in to escape from carrying their proper share



g. 3—Stress deformation of an elbow

f the load. The bending moment is resisted essentially y the shaded effective section. The maximum stress oint is shifted from Point A to the effective extreme oint B. As the cross-section ovals, a local bending stress also produced around the circumference. The maximum rcumferential stress occurs at Point C where the radius f curvature is the smallest.

Mathematically the maximum longitudinal stress and cumferential stress can be calculated by using the llowing stress intensification factors:[4]

$$\left.\begin{array}{l}\beta_i = 0.84/h^{2/3}\\ \gamma_i = 1.80/h^{2/3}\end{array}\right\}\quad \text{in-plane bending} \qquad (3)$$

$$\left.\begin{array}{l}\beta_o = 1.08/h^{2/3}\\ \gamma_o = 1.50/h^{2/3}\end{array}\right\}\quad \text{out-plane bending} \qquad (4)$$

The experimentally measured distributions of the ongitudinal and circumferential stresses of a 30-inch ipe elbow subject to in-plane bending[5] are shown in ig. 4. Maximum circumferential stress is normally greater han the maximum longitudinal stress. However, the ature of the two stresses is quite different. The longiudinal stress is a membrane stress working directly against e moment, while the circumferential stress is a skin ending stress resulting from local deformation.

Code stress intensification factors. The stress innsification factors given in the code[2] are intended for ermal expansion and other displacement loads. The ature of thermal expansion load is different from that f weight and other sustained loads. Thermal expansion self-limiting. It is a strain controlled loading such that nce the strain reaches a point large enough to comensate for the expansion, growth stops regardless of e actual stress developed in the system. It can not nornally cause any structural damage in one single applicaon, but can cause fatigue failure through repeated exansion and contraction cycles. Therefore, for evaluating ermal expansion, the stress intensification factor is determined by the ratio of the stress causing failure over a iven number of cycles in a straight pipe to the stress ausing failure at a component subject to an equal numer of stress cycles. Code stress intensification factors are yclic or fatigue stress intensification factors in which he local peak stress is governing.

Theoretically these intensifications are equal to the maximum stress intensification existing in any region and direction within a component. In an elbow, for instance, the circumferential stress intensification factors $1.80/h^{2/3}$ and $1.50/h^{2/3}$ for in-plane and out-of-plane bendings, respectively, should be used. However, intensive fatigue tests on various components[3] have shown that by using unity as the fatigue life of girth welded or clamped pipe, the effective stress intensification factors of elbows in bending fatigue were about half the theoretical value. By dividing the theoretical factor by two, the code stress intensification factor for elbows is as follows:

In-plane stress intensification factor

$$i_i = 0.90/h^{2/3} \qquad (5)$$

Out-of-plane stress intensification factor

$$i_o = 0.75/h^{2/3} \qquad (6)$$

The stress intensification factors for other components are derived by using elbow analogy correlated with test results.

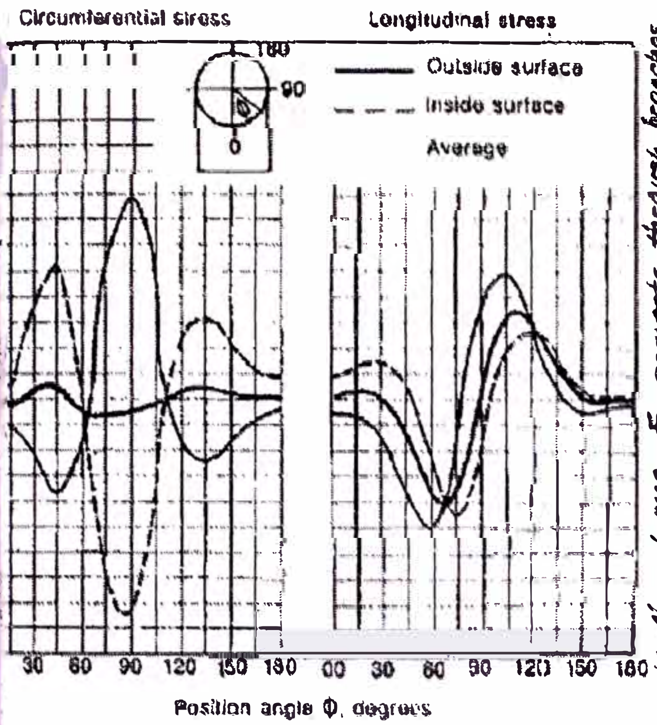
STRESS INTENSIFICATION FACTORS FOR WEIGHT AND OCCASIONAL LOADS

No stress intensification factor is explicitly stated in the Chemical Plant and Petroleum Refinery Piping Code for weight and occasional loads. Weight and wind are sustained loadings. They are not self-limiting, and always require a static equilibrium between the stress developed in the component and the load applied. Once yield point or collapse load is reached, the component will fail regardless of the amount of deformation that has occurred. Therefore, the stress to be considered in weight and other sustained loadings should possess the following characteristics:

► The stress is in a direction directly against the loading. Only the stresses acting against the load are load-carrying stresses.

► The stress is the average stress across the wall thickness. The average stress is actually the remaining stress available for external equilibrium after the internal mutual cancellation.

Theory and experiment indicate the same code stress intensification factors intended for flexibility analysis should be used in weight, occasional and other sustained load analyses



ig.—Variation of stress around the circumference of an with a 30-inch OD, 0.515-inch wall and a 45-inch bend

This discussion is mainly for moments through run. For moments through branches the actual SIF can be bigger for small branches. d/D = 0.5 is about the worst for through branch load.

n these two criteria and referring to Fig. 4, it can ıcluded that in an elbow the stress intensification ight and other sustained loads should be approxi- equal to the longitudinal stress intensification. The intensification in the circumferential direction is portant here, because it is not in the loading direc- nd has very small average stress.

e the stress-raising factor in a girth weld will not antly affect the load-carrying capacity, the theore- stress intensification factors shown in Equations 5 can be used directly even in reference to girth pipe. By comparing Equations 3, 4, 5 and 6 it r that the code stress intensification factors can also d for weight and other sustained loadings without much accuracy.

se are purely mathematical deductions which need ubstantiated by experiments. The stress intensifica- easure of a component subject to a sustained load- its collapsing strength. Bolt and Greenstreet[6] have substantial tests in determining the collapse loads ows. Schroeder,[7] on the other hand, has done the for branch connections. Some of their test results mmarized in Table 1. The elbow collapse moments in the table are taken at the center of elbow ther than at the loading end of elbow edge as in ture cited 6. The code stress intensification factors e tested special tees are calculated by assuming a reinforcing pad having a diagonal dimension the as the throat dimension of the crotch radius.

m Table 1, again it is apparent that the stress ification factors as represented by M_p/M_c are very to the code stress intensification factors intended for loadings. The somewhat larger factor experienced e stainless steel elbow appears to be caused by the inherent round-house stress-strain curve of the material which flattens the load-deflection curve at an earlier stage.

STRESS INTENSIFICATION FACTORS FOR SMALL BRANCH CONNECTIONS

The code stress intensification factors for tees and branch connections were derived from full-sized branch connections. In applications where the branch size is much smaller than the run size, application of these factors can be grossly too conservative. Although Code Case No. 53, which was subsequently incorporated in the code, provided some relief to the branch itself, it did nothing to relieve the moment load transferred through the run pipe. Therefore, when it comes to the run moment, the current practice is to completely ignore the very small branches which are defined rather arbitrarily by individual designers.

Basically, the present code requires that a uniform stress intensification factor be used for moments acting both through the branch and through the run. For a reduced outlet the section modulus used in determining branch stress can use so-called effective branch wall thickness instead of the actual thickness. The effective branch thickness, T_e, is the lesser of run thickness, T, and the product of out-of-plane stress intensification and branch thickness i_oT_b. In other words, the stress intensification based on branch section modulus can be reduced by a factor of T_b/T for moments acting through branch. There is still no relaxation given to the moments carried straight through the run pipe.

Empirically, the stress intensification factor for an out-of-plane bending moment applied to the branch pipe can be expressed as[8]

$$i_b = A(r/T)^{2/3}(r_b/r)^{1/2}(T_b/T) \tag{9}$$

Except for the $(r_b/r)^{1/2}$ term, Equation 9 is an exact expression of code requirements. The term $A(r/T)^{2/3}$ is the code stress intensification factor with $A = 0.335$ for a welding tee and so forth, and (T_b/T) is the effective thickness factor stipulated in Code Case No. 53. Since the (T_b/T) factor has been included in the code definition of effective branch wall thickness, it can be removed from the equation. By rearranging the equation, we have

$$i_b = i_o(r_b/r)^{1/2} \tag{10}$$

where i_o is the code stress intensification factor. Without the effective thicknes factor, Equation 10 can also be used for moments acting through the straight runs. This equation can serve as a gradual transition from full-sized outlets to small connections.

CONCLUSIONS

Currently there is no explicit statement in the Chemical Plant and Petroleum Piping Code requiring the application of a stress intensification factor in weight and other sustained load analyses. Application of these factors is therefore determined by the design specification prepared

TABLE 1—Collapse moments on elbows and tees

piece	Material	Yield stress (ksi)	Moment direction	Collapse moments (in.-kip) Test piece M	Straight pipe M_n	M_n/M	Code stress intensif. i
6 Sch. 40 LR elbow...............	ASTM A-106B	50.0	In-plane open	235	564	2.4	2.27
6 Sch. 40 LR elbow...............	ASTM A-106B	50.0	In-plane close	208	564	2.71	2.27
6 Sch. 40 LR elbow...............	ASTM A-106B	50.0	Out-plane	234	564	2.41	1.89
6 Sch. 80 LR elbow...............	ASTM A-106B	37.8	In-plane open	435	627	1.44	1.64
6 Sch. 80 LR elbow...............	ASTM A-106B	37.8	In-plane close	358	627	1.75	1.64
6 Sch. 80 LR elbow...............	ASTM A-106B	37.8	Out-plane	409	627	1.53	1.37
6 Sch. 40 SR elbow...............	ASTM A-106B	39.6	In-plane open	184	447	2.43	2.96
6 Sch. 40 SR elbow...............	ASTM A-106B	39.6	In-plane close	175	447	2.55	2.96
6 Sch. 40 SR elbow...............	ASTM A-106B	39.6	Out-plane	196	447	2.28	2.43
6 Sch. 40 LR elbow...............	ASTM A-312	37.7	In-plane close	117	426	3.64	2.27
3 in. OD 0.14-in. t tee...............	AISI 1020C	25.0	In-plane	27	42.5	1.57	1.72
3 n. OD 0.14-in. t tee...............	AISI 1020C	31.4	Out-plane	28	54	1.93	1.96

the owner or its agent. However, there are widely ided opinions regarding the magnitude of the factors be used. From the discussions presented in this article, ppears that both theory and experiment have indicated same code stress intensification factors intended for ibility analysis should also be used in weight, oc- onal and other sustained load analyses.

or branch connections, the code stress intensification ors were basically obtained from tests on full-sized et connections. In small-sized outlet connections, the e has provided some relief for moments acting through nches but no relief is given for moments acting through straight runs. Although common practice is to ignore ss intensifications at very small branches, a guideline eeded for making the decision. With the unresolved ation that exists, a designer's rather arbitrary decision can artifically make a component several times weaker or stronger. This inconsistency can be greatly mitigated by multiplying the code stress intensification factor with a gradual size reduction factor $(r_b/r)^{1/2}$. This factor has been adopted in the Power Piping Code[9] for certain branch connections.

About the author

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NOMENCLATURE

$\beta = S_{lmax}/S$, longitudinal stress intensification factor
$\gamma = S_{cmax}/S$, circumferential stress intensification factor
S_{lmax} = Maximum longitudinal stress, psi
S_{cmax} = Maximum circumferential stress, psi
$S = M/Z$, equivalent bending stress developed in a straight pipe of identical cross-section, psi
M = Bending moment, in.-lb.
Z = Section modulus of the pipe section, in.^{3}
$h = TR/r^2$, the flexibility characteristic
R = Bend radius, in.
T = Wall thickness of the pipe, in.
r = Mean radius of the pipe cross-section, in.
i_b = Stress intensification factor for branch connection
r_b = Mean radius of branch pipe, in.
T_b = Thickness of branch pipe, in.
A = Empirical correlation constant.

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Quick Check on Piping Flexibility

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BSTRACT

ne major requirement in piping design is to provide adequate flexibility for absorbing the thermal expansion of the pipe. However, ue to lack of quick method of checking, pipings are often laid-out o be either too stiff or too flexible. In either case, valuable ime and material are wasted. This paper presents some of the quick ethods for checking piping flexibility. These methods include isual, hand calculation, and micro computer approaches. They are ll quick and easy for designers to use in planning their layouts. nce the designers have taken care of the flexibility problem, the terative procedure between the stress engineers and the designers ecome simpler. The project schedule can also be improved.

IPING FLEXIBILITY

s the pipe temperature changes from the installation condition to he operating condition, it expands or contracts. In the general erm, both expansion and contraction are called thermal expansion. hen a pipe expands it has the potential of generating enormous orce and stress in the system. However, if the piping is flexible nough, the expansion can be absorbed without creating undue force r stress. Providing the proper flexibility is one of the major asks in the design of piping system.

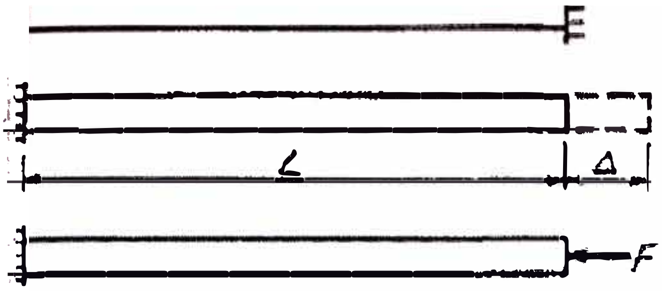
iping is used to convey a certain amonut of fluid from one oint to another. It is obvious that the shorter the pipe is used he lesser the capital expenditure is required. The long pipe may lso generate excessive pressure drop making it unsuitable for the roper operation. However, the direct shortest layout generally is ot acceptable for absorbing the thermal expansion.

igure 1 shows what will happen when a straight pipe is directly onnected from one point to another. First, consider that only one nd is connected and the other end is loose. The loose end will xpands an amount equal to

$$\Delta = e\ L$$

owever, since the other end is not loose, this expansion is to be bsorbed by the piping. This is equivalent to squeezing the pipe o move the end back an Δ distance. This amount of squezzing reates a stress of the magnitude

$$S = E\ (\Delta / L) = E\ e$$



Where,

Δ = thermal expansion, in
e = expansion rate, in/in
L = pipe length, in
S = axial stress, psi
E = modulus of elasticity, psi
A = pipe cross section area, in2
F = axial force, lbs

Figure 1

T force required to squeeze this amount is

$$F = A S = A E e$$

T e a 6-inch standard wall carbon steel pipe for instance, an i rease of temperature from 70F ambient to 300F operating creates a axial stress of 42300 psi and an axial force of 236000 lbs in t pipe. These are excessive even though the temperature is only 3 F. It is clear that the straight line direct layout is not eptable to most of the piping. Flexibility has to be provided.

'ANSION LOOP

P ing flexibility are provided in many different ways. The turns offsets needed for running the pipe from one point to another P vides some flexibility by themself. This inherent flexibility or may not be sufficient depending on the individual cases. A itional flexibility can be provided by adding expansion loops o expansion joints. In the straight line example discussed above, t stress can be reduced by a loop installed as shown in Figure 2 o by an expansion joint as shown in Figure 3.

T idea in Figure 2 is to provide ie pipe perpendicular to the d ection of expansion. In this when the pipe expands it bends loop leg first before transmi- t ng any load to the anchor. The l ger the loop leg the lesser the f ce will be created. The force c ated is inversely proportional the cube of the loop length and stress generated is roughly ersely proportional to the square of the loop length. The loop ietimes can take considerably more space and piping than what is ilable, or economically justifiable. This is especially true large high temperature low ssure pipings. In this case the ter method is to use expansion nt. Expansion joints are more histicated than the pipe loops ch are just extra lengths of same piping. For this and

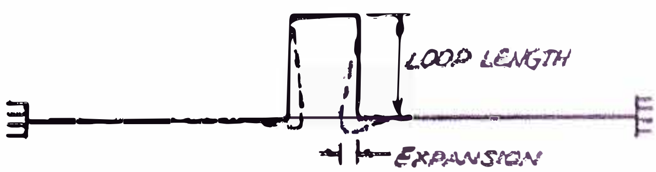


Figure 2

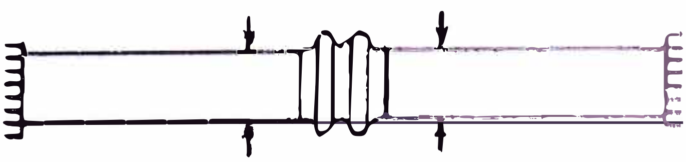


Figure 3

ther reasons, engineers tend to favor piping loops over expansion oints. However, expansion joints can be used effectively in many pplications when they are properly designed. One of the major equirements in the design of expansion joint system is to install ufficient restraints for maintaining the stability. This article eals mainly the loop approach.

HE CRITICAL PATH

n designing a plant, the piping is generally routed or laid-out y the piping designers then checked by the stress engineers as hown in figure 4.

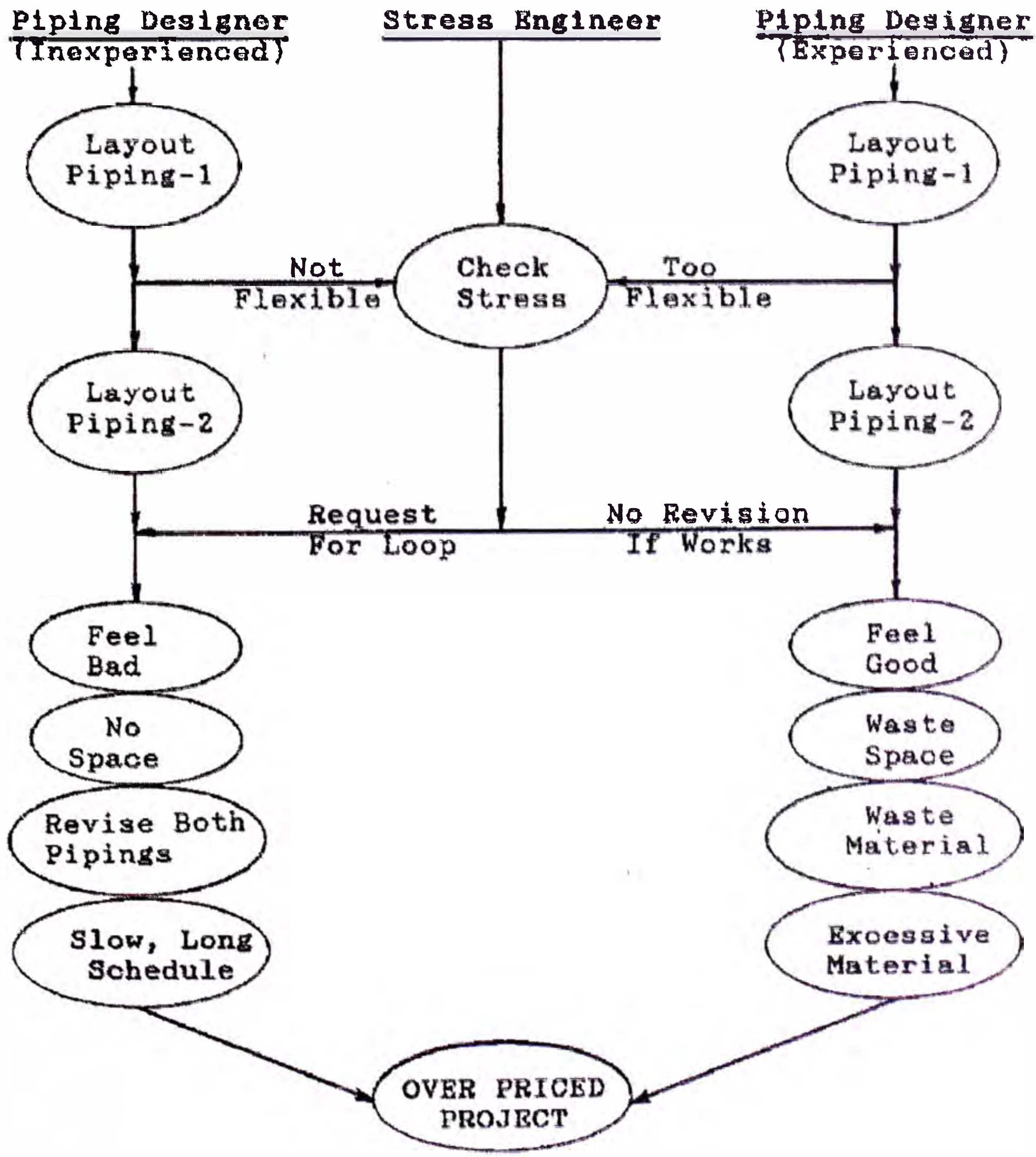


Figure 4

There is a marked difference in the layout done by the experienced and the inexperienced designers. The experienced designers know the importance of the flexibility. However, they tend to provide too much flexibility in contrast to the inexperienced ones who tend to provide little flexibility. In either case, the result is an over priced project.

The layout done by an inexperienced designer is normally too stiff because the designer does not know how or too timid to add loops or offsets. If a piping system is too stiff, the stress engineer will almost certain to find it out. The stress engineer will send the design, with recommended loops, back to the designer for revision. At this time, the designer have made some more layouts in the same area making the revision very difficulty. On the other hand, a layout done by an experienced designer often contains the loops which are excessive or not needed. The excessive loops are normally maintained without revision, becuase it is a common practice not to change something which works. The experienced one might have saved the manhour needed for the revision. The cost of the excessive loops can be prohibitive.

The cost of the project can be reduced substantially if the right amount of flexibility is built in the piping at the initial layout stage. This requires some quick methods which can be used by the designers to check the piping flexibility.

VISUAL CHECK

The visual check is the first important examination on anything we do. If the design looks strange, then most likely something is wrong with it. By now we at least know that we can not run a piping straight from one point to another. This also applies to the situation when there are two or more line stops installed at a straight header as shown in Figure 5. The line stop or axial stop acts directly against the expansion of the pipe. When two axial stops installed on the same straight leg, the thermal expansion of the pipe located between the stops has no place to relieve.

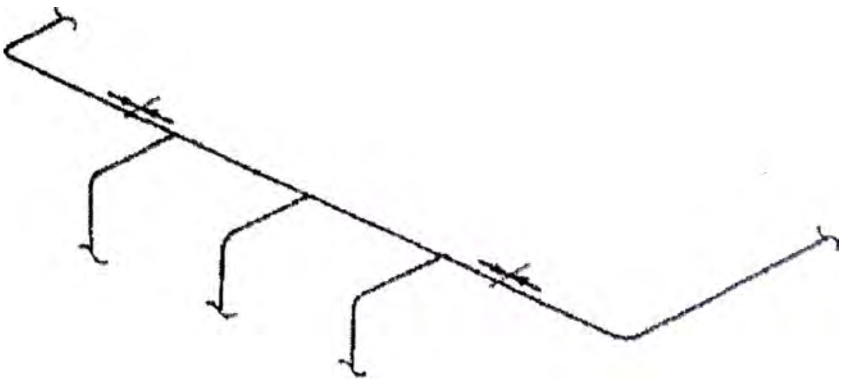


Figure 5

The visual check of the piping flexibility is to look for the pipe legs located in the direction perpendicular to the line connecting two anchor or other restraint points. The length of the leg is he direct measure of the flexibility. Therefore, the key is to ocate the availability of the perpendicular leg and to determine f the length of the leg is sufficient. The required leg length an be estimated by the rule of thumb equation (1) derived by the uided cantilever approach, for steel pipes.

$$\ell = 5.5 \sqrt{D\Delta} \qquad (1)$$

where,

ℓ = leg length required, ft
D = pipe outside diameter, in
Δ = expansion to be absorbed, in

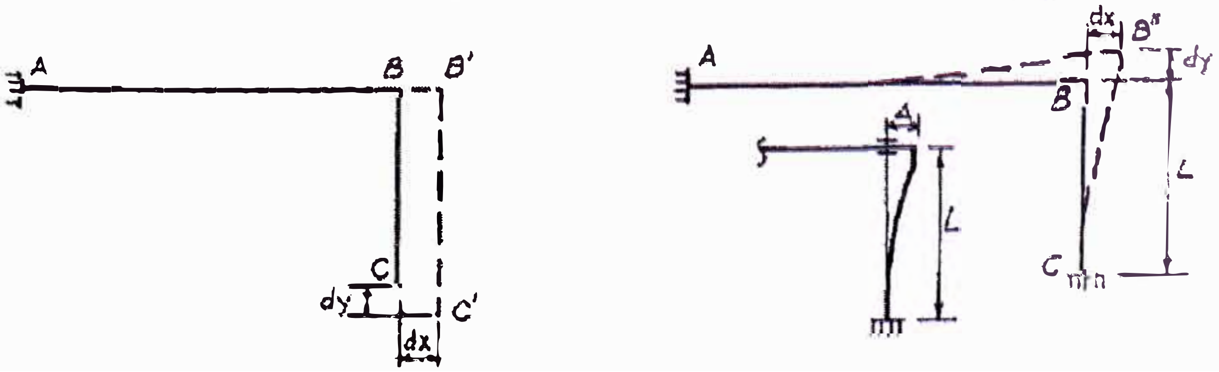
o use Equation (1) efficiently the expansion rate of the pipe has o be remembered. Table 1 shows the expansion rates of carbon and tainless steel pipes at several operating temperatures. The rate t other temperature can be estimated by proportion. By combining quation 1 and Table 1, the designer can estimate the leg length equired without needing a pencil. For instance, an 80 feet long -inch carbon steel pipe operating at 600F expands about 4 inches hich requires a 30 feet leg to absorb it. It should be noted that n expansion loop is considered as two legs with each leg absorbs ne half of the total expansion.

able 1

		Expansion Rate, in/100 ft pipe				
	Temp,F	70	300	500	800	1000
arbon Steel		0	1.82	3.62	6.7	8.9
tainless Steel		0	2.61	5.01	8.8	11.5

AND CALCULATION

here are several simplified calculations can be performed quickly th hand. The most popular one is the so called guided cantilever pproach. The method can be explained using the L-bend given in gure 6 as an example. When the system is not constrained the



(a) Free Expansion (b) Constrained Expansion

Figure 6

ints B and C will move to B' and C' respectively due to thermal xpansion. The end point C moves dx and dy respectively in X- and directions, but no internal force or stress will be generated. However, in the actual case the ends of the piping are always nstrained as shown in Figure 6(b). This is equivalent in moving e free expanded end C' back to the original point C forcing the int B to move to B". The dx is the expansion from leg AB, and from leg CB. The deformation of each leg can be assumed to llow the guided cantilever shape. This is conservative because e end rotation is ignored. The force and stress of each leg can w be estimated by the guided cantilever formula. The leg AB is guided cantilever subject to dy displacement and leg CB a guided ntilever subject to dx displacement respectively.

om the basic beam theory, the moment and displacement relation a guided cantilever is

$$M = \frac{6\ E\ I}{L^2}\Delta, \qquad F = \frac{2\ M}{L} \tag{2}$$

r thin wall pipes, Equation (2) can be further reduced. By using $\pi r^3 t$ and $S=M/(\pi r^2 t)$, the above equation becomes

$$S = \frac{6\ E\ r}{L^2}\Delta = \frac{E\ D\ \Delta}{48\ \ell^2} \tag{3}$$

ere, S = thermal expansion stress, psi
E = modulus of elasticity, psi
r = mean radius of the pipe, in
Δ = total expansion to be absorbed, in
L = length of the leg perpendicular to , in
ℓ = length in feet unit, ft
D = outside diameter of the pipe, in

uation (3) is a convenient formula for the quick estimation of e expansion stress. By pre-setting E=29.0x10 psi and S=20000 i, Equation (3) becomes Equation (1) used in finding the leg ngth required for steel pipes.

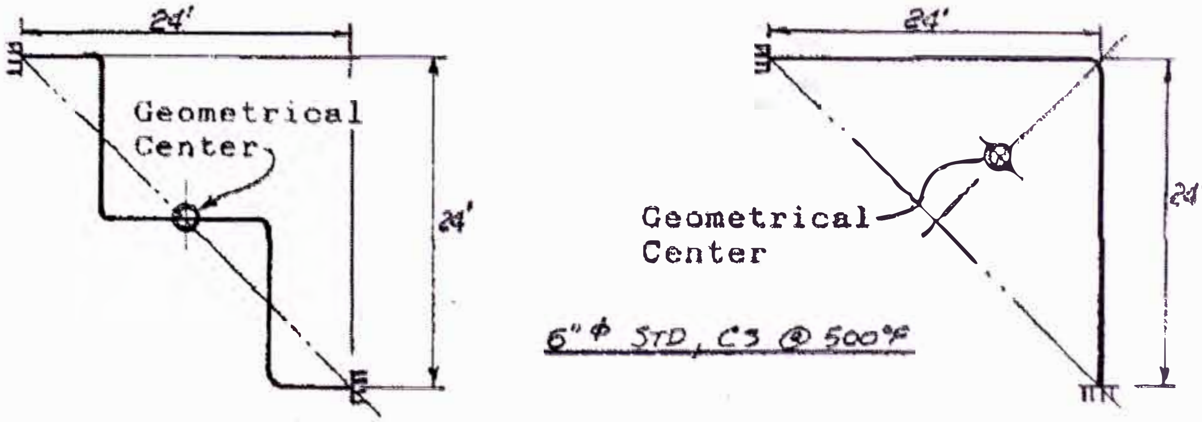
e other formula can be used for the quick check is the one given ANSI B31 Piping Codes. The Code uses Equation (4) as a measure adequate flexibility, subjects to other requirements of the de.

$$\frac{D\ y}{(\ L-U\ )^2} < 0.03 \tag{4}$$

ere, D = outside diameter of the pipe, in
y = resultant of total displacement to be absorbed, in
L = developed length of piping between anchors, ft
U = straight line distance between anchors, ft

uation (4) is actually equivalent to Equation (1), if (L-U) is nsidered as the perpendicular leg length.

quation (4) has to be used with great care, because the same extra ength of pipe can have very diferent effects depending on the ways he pipe is laid-out. Normally more flexibility will be achieved if he pipe is placed farther away from the elastical or geometrical enter. For instance with the same extra length of piping, when it s laid-out as shown in Figure 7 (a) it has much higher flexibility han when it is laid-out as in Figure 7 (b). Designers often have he misconception about the amount of flexibility can be provided y the zig-zag arrangement. Due to the extra elbows placed in the ayout, one tends to think that additional flexibility should have een created. Unfortunately, the additional flexibility from the lbows is not enough to compensate the loss of flexibility due to he placement of pipe toward the geometrical center.



(a) Stress = 13764 psi (b) Stress = 8226 psi

Figure 7

ICRO COMPUTER APPROACH

irrently most large engineering companies use CAD system to do the iping design. It is possible that one day the system will be able tell you if you need any extra flexibility, as soon as you place e line on the screen. However, before that time comes, we still ave to survive the current situation to be able to see the good ing coming. Nevertheless, the technology of the micro computer is advanced enough for us to perform accurate flexibility analyses ight beside the drafting board.

e micro computer programs are normally so user friendly that it akes only a couple of hours to master their usage. With respect the flexibility check, a piping designer can do almost as good job as a stress engineer can. What it is needed is to enter the ipe and geometrical information to the program which will almost stantly give you the forces and stresses expected in the system. rom that information, the designer can then decide if additional oops or offsets are required.

e use of the micro computer differs substantially depending on e individual program setup. Each program has its preferred ethod of entering the data and generating the output. Appendix A ows the sample operating procedure using PENG.QFLEX program to alyze the simple system given in Figure 8.

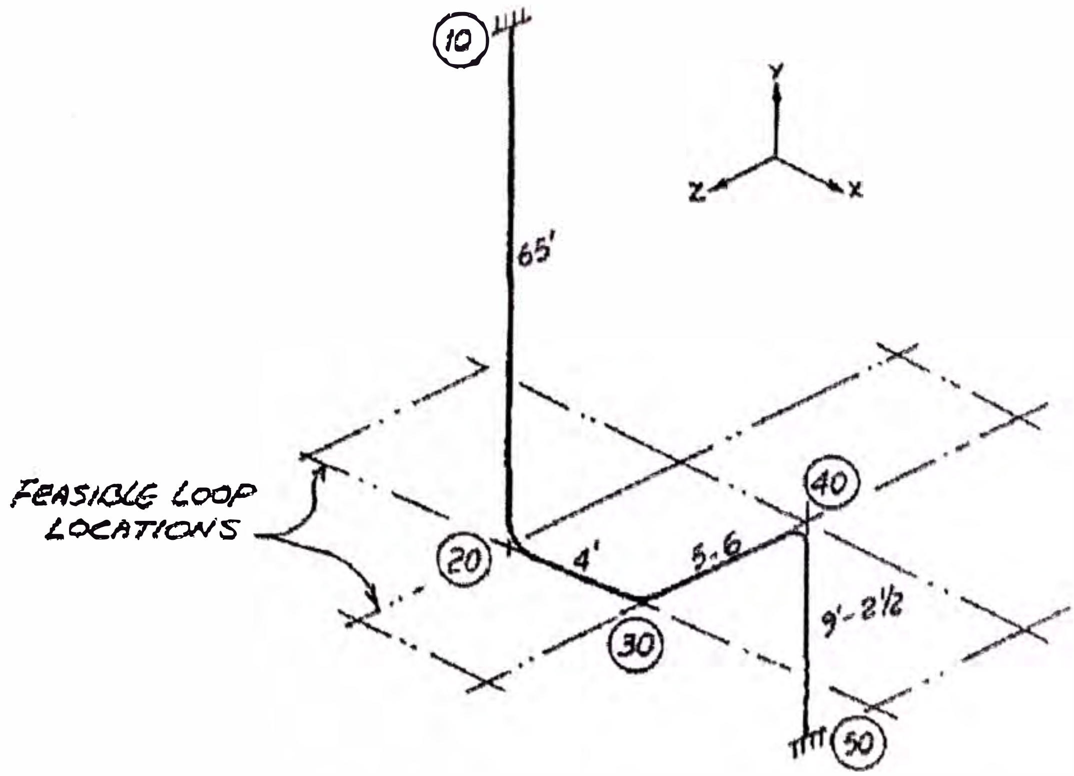


Figure 8

ce it is determined that an expansion loop is required, the loop
n be placed at one of the feasible locations before the area is
ngested by other layouts. This also saves the iterative process
tween the piping designers and the stress engineers.

NCLUSION

e traditional piping design procedure depends heavily on the
ress engineer to check piping flexibility. With the availability
quick methods in checking the flexibility, the designer can now
yout the pipe to provide the proper flexibility at the very
ginning. This substantially reduces the number of iterations
quired between the piping designer and the stress engineer. The
st of the plant can be reduced by the shorter schedule and less
npower required.

:QUIPMENT RELIABILITY IMPROVEMENT
HROUGH REDUCED PIPE STRESS

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The load and stress imposed from a connecting piping system can reatly affect the reliability of an equipment. These loads, either from xpansion of a pipe or from other sources, can cause shaft iisalignment, as well as shell deformation, interfering with the ternal moving parts. Therefore, it is important to design the piping ystem to impose as little stress as possible on the equipment. leally, it is preferred to have no piping stress imposed on an quipment, but that it is impossible. The practical practice is for the quipment manufacturer to specify a reasonable allowable piping load nd for the piping designer to design the piping system to suit the llowables. The allowable piping loads given these days are generally etermined solely by the equipment manufacturers without any articipation from the piping engineering community. The values so etermined are usually to low to be practical.

The low allowable pipe load given by the manufacturer results in weaker machine for enduring the day to day operating environment. It lso complicates the layout of the piping system in meeting the llowable. Unusual configurations and restraining systems are often sed to make the calculated piping load satisfy the given allowable. lowever, all these efforts are very often just exercises of computer echnology. The main reliability problem has not been solved. A better esigned equipment with some common sense piping arrangement is the asis for improving reliability.

LLOWABLE LOAD

Process equipment, especially the rotating equipment, generally ave a very low allowable piping load. Piping engineers often think the ianufacturers give low allowables to protect their own interests. his notion is not necessarily true, because many equipment indeed annot take too much a load. The problem is that a weak link exists iat is often overlooked in the design of an equipment. Figure 1 shows typical pump installation which can be divided into three main parts: ie pump body, the foundation, and the pedestal/base plate. Without ie input or threat from the piping or equipment engineers, the routine

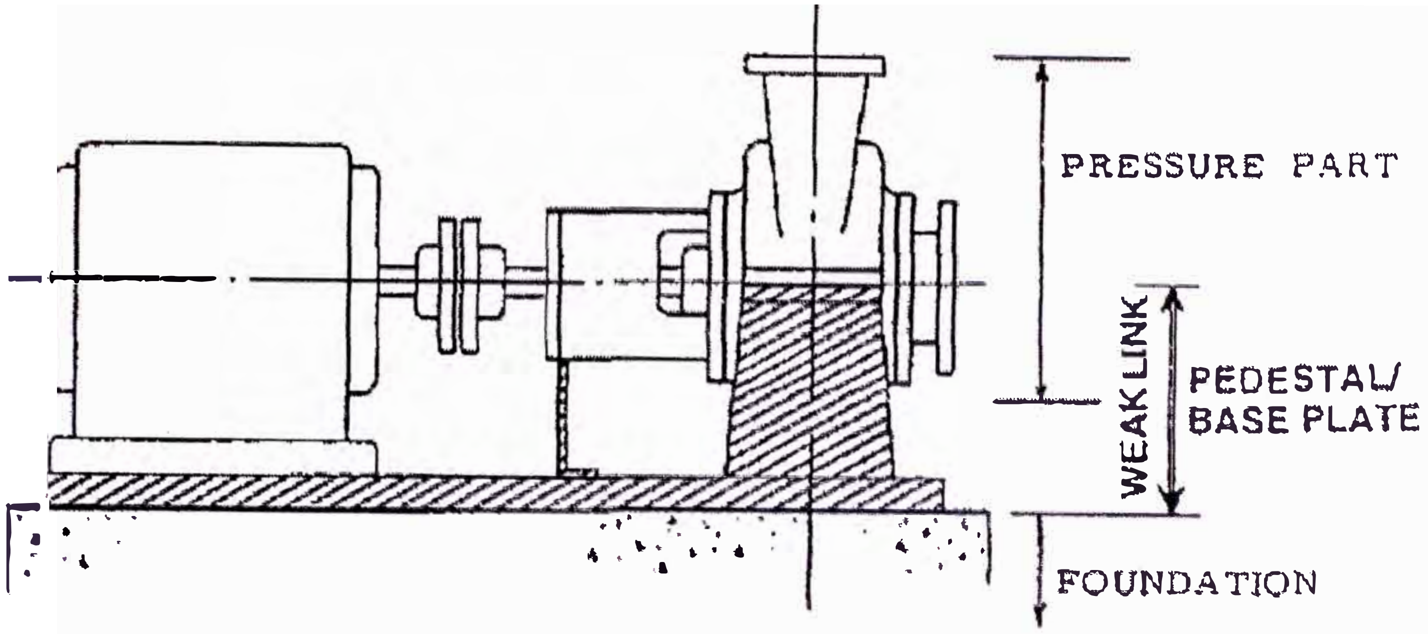


Figure 1, The Weak Link

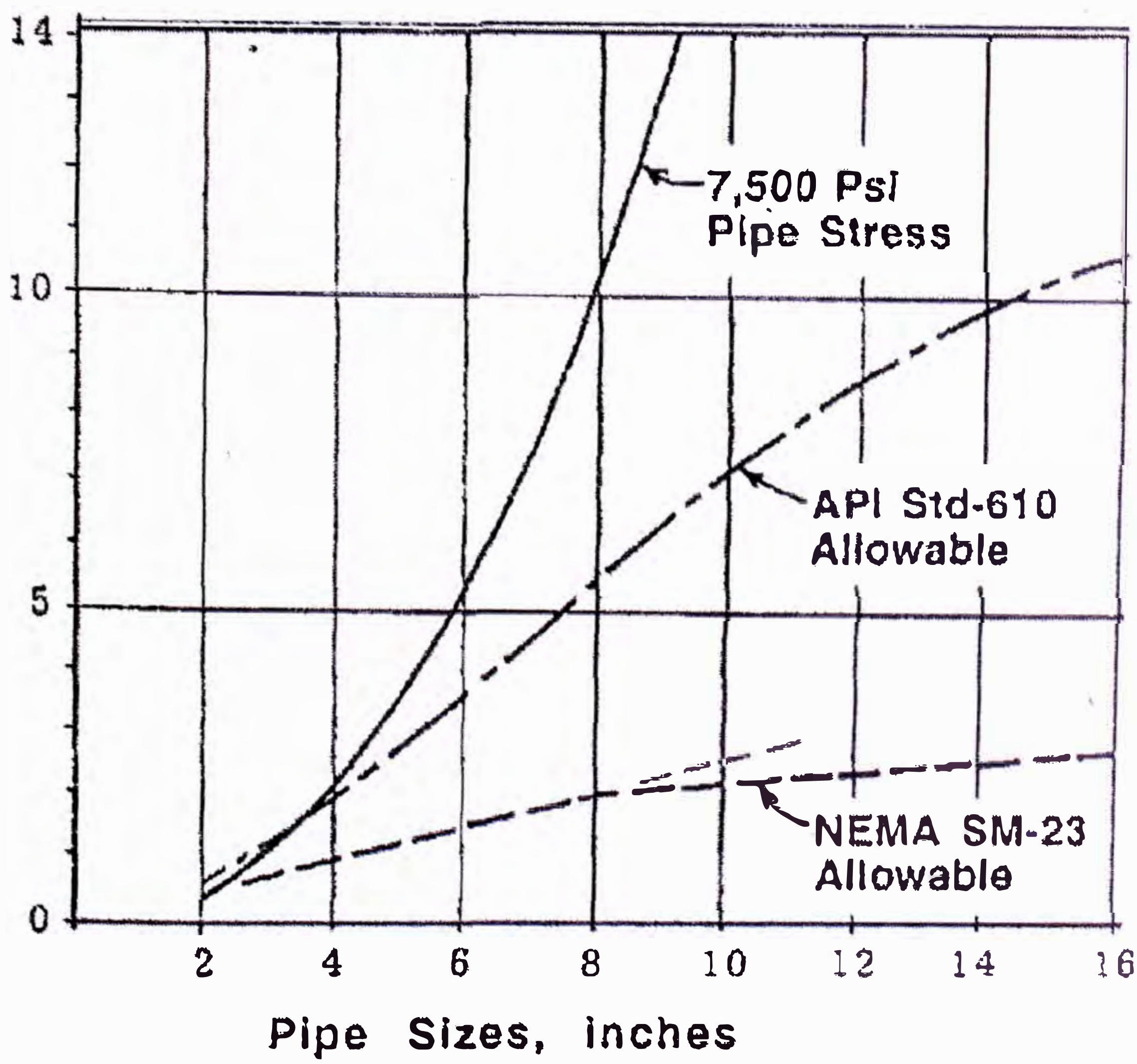


Figure 2, Allowable Piping Loads

design of the pump assembly can have different significance on different parts of the pump. The pump body is designed to be as strong, if not stronger, than the piping so that the body can resist the same internal design pressure as the piping. The foundation, normally designed with the combined pump/motor assembly weight, is also massive and stiff due to the limitations of the soil bearing capacity. However, the pedestal/baseplate is a different story. Without considering the taking of any piping load, the pedestal/baseplate is generally designed only by the pump weight. This design basis creates a very weak pedestal/baseplate which can take very little load from the piping, hence the famous story of the vendor who claimed his equipment cannot take any piping load. Nowadays, most vendors have more sense than to claim such a thing, but the allowable piping load is still not large enough to be desirable. The weak link, of course, is the pedestal/baseplate assembly.

By understanding the situation, the problem can actually be ...cied very easily. Improvement has already been seen in pump applications. Pump application engineers who long realized the low allowable piping load problem customarily specified double (2X) or triple (3X) base plates to increase the allowable piping load by two or three times, respectively. Surprisingly, to most engineers, the cost of 2 or 3X pump was only marginally more than that of a regular pump. Actually, it should not have been the least bit surprising, since all a vendor has to do to make it 2X or 3X is to provide a couple of braces or stiffeners. Recognizing the popular demand for the 2X or 3X baseplate, the API formally adopted it to its pump standards. Since the sixth edition of the API Std-610 [1], the allowable has been increased to a level that makes the 2X and 3X specification no longer necessary. In other words, the strength of the whole pump assembly has become ...ird uniform that no additional allowable can be squeezed out without adding a substantial cost. Unfortunately, at present this philosophy has not been shared by other manufacturers. For example, the 1956 NEMA [2] ...rt the allowable load is probably the most unreasonable of its kind. The API Std-617 [3] centrifugal compressor and the ASME/ANSI B73.1 [4] ...un are not far behind. The API Std-617 uses 1.85 times the NEMA allowable, and the ANSI B73.1 vendors often use 1.30 times the NEMA values for the allowables. Figure 2 shows the comparison of the pipe strength, the allowable API Std-610 piping load, and the NEMA allowable piping load. The pipe strength curve is based on a 7500 psi bending stress. It should be noted that the allowable pipe stress against thermal expansion can be as much as three times higher than 50... psi.

Looking at Figure 2, it is clear that the piping load that can be applied to an equipment is much smaller than the strength of the pipe itself. Therefore, in designing the piping connected to an equipment, the equipment allowable load is the controlling factor. For low allowable items, such as a large size steam turbine, an extensive expansion loop, and a restraining system is generally required. This is a fact and should be understood by all parties concerned.

Because of the elaborate design of the piping system attached to a sensitive equipment, engineers may sometimes get too trapped in the computer maze and overlook engineering fundamentals. Typical examples that can cause unreliable operation are discussed in the following.

EXCESSIVE FLEXIBILITY

Adequate piping flexibility at an equipment is required to reduce the piping load to the acceptable value. However, a good design should consider the realistic flexibility from the support structure and the proper use of the protective restraints. Without the properly located restraints, a piping system , no matter how flexible it is, has difficulty meeting the allowable load imposed by the equipment. Figure 3 shows a pump piping system which was designed without any restraints installed. This is a common mistake made by inexperienced engineers who think that a restraint can only increase the stiffness, thus increasing the load. It is true that a restraint will tend to decrease the flexibility of the system as a whole and will increase the maximum stress and force in the system. However, a properly designed restraint can shift the stress from the portion of piping near the equipment to a portion further away from the equipment.

Although extensive loops are used in the piping given in the figure, the piping load still may not meet the equipment allowable due to the lack of a restraining system. The excessive flexibility makes the system prone to vibration, because it is easily excited by small, disturbing fluid forces. In addition, the piping loops enhances the internal fluid disturbance by creating cavities and other flow discontinuities due to excessive pressure drops. A system similar to the shown in Figure 3 experienced very severe vibrations in one petrochemical plant. The operational engineer had to put a large cross beam to anchor all the loops in the field to suppress the vibration to a manageable level. This shows that the function of the original loops were lost by the anchoring system. The piping still experiences larger than normal vibrations due to flow disturbance caused by the loop

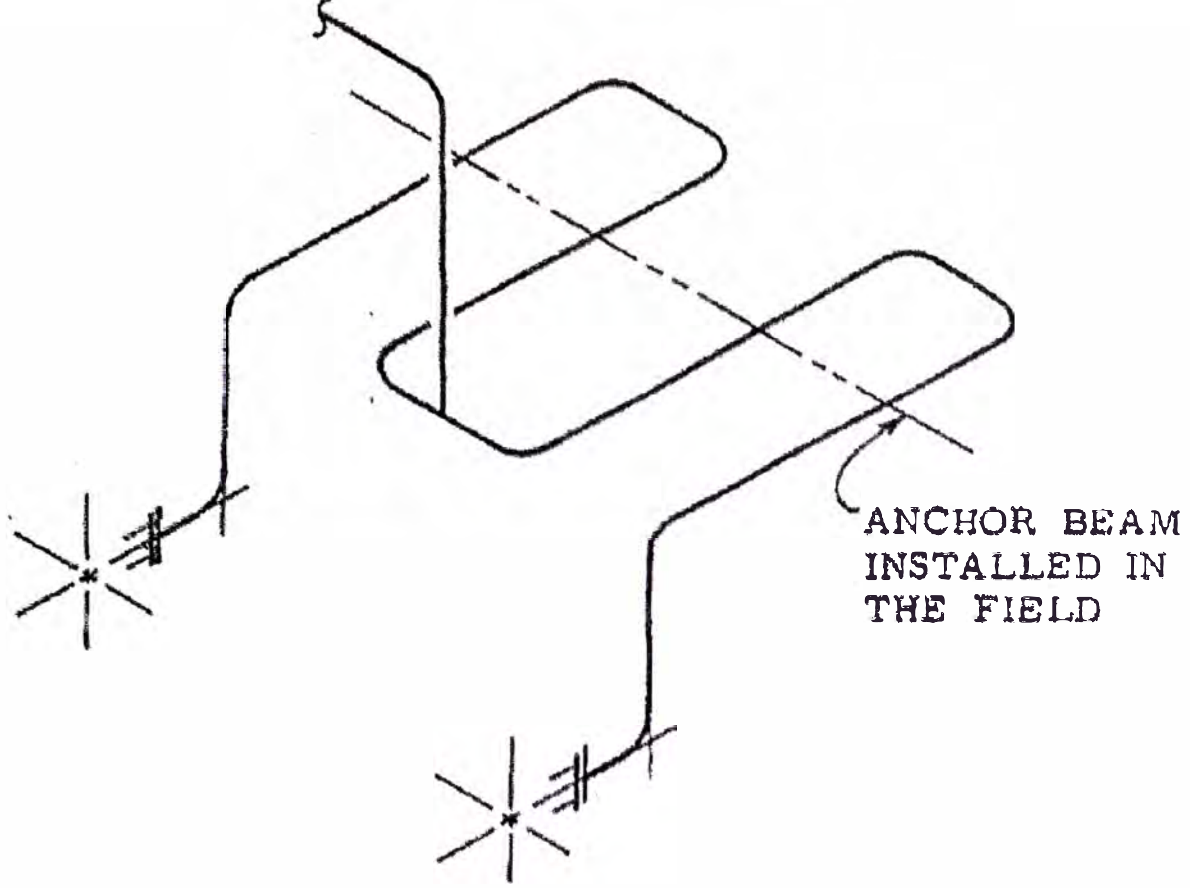


Figure 3, Too Much Flexibility

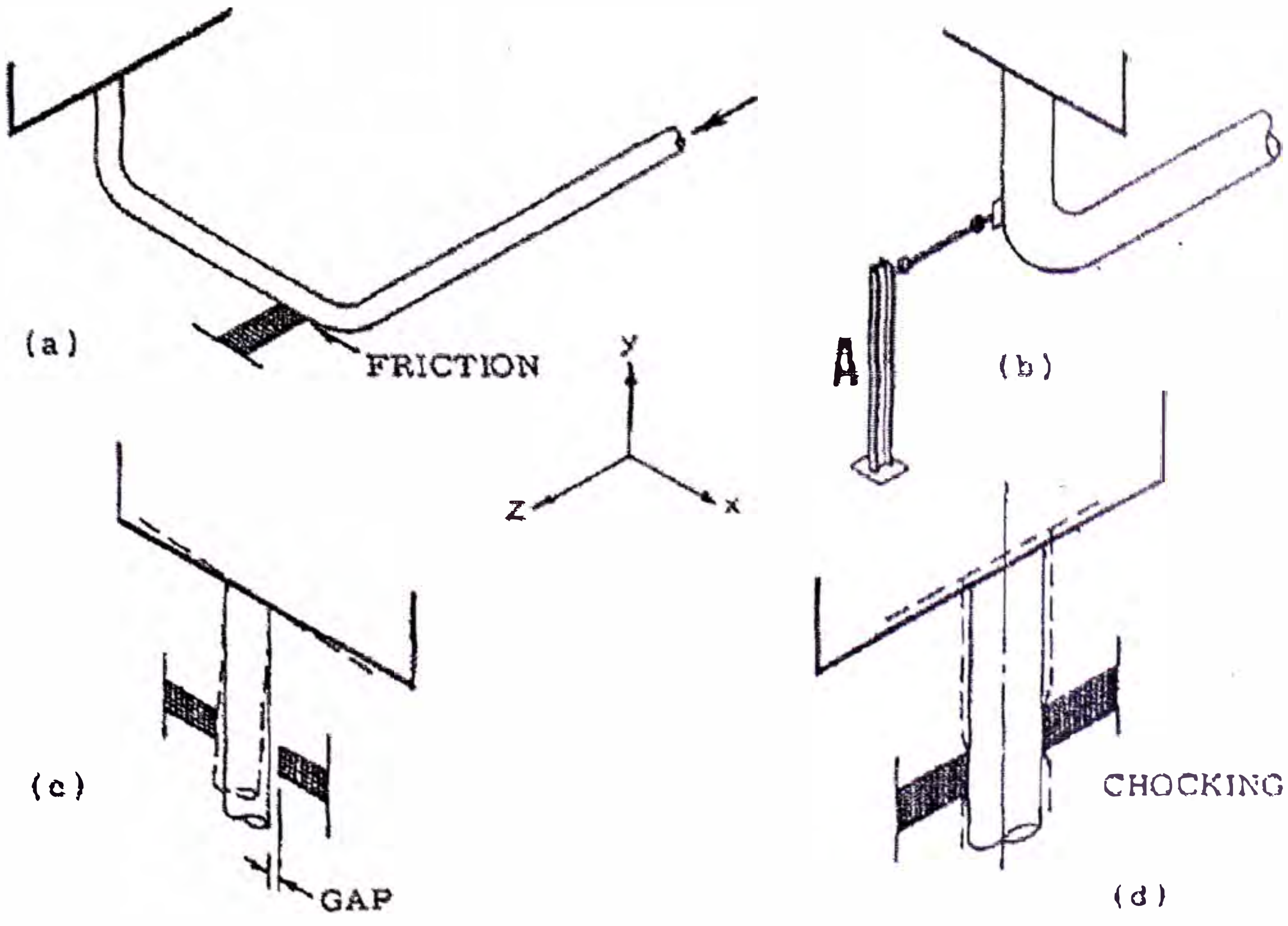


Figure 4, Problems with Theoretical Restraints whin is structurally fixed, but hydraulically still open to many dire tional changes.

THEORETICAL RESTRAINTS

A properly designed piping system generally has some restraints o ontrol the movements and to protect the sensitive equipment. However, there are also restraints which are placed in desperation by piping engineers trying to meet the allowable load of the equipment. These so-called computer restraints give a very good computer analysis result on paper, but are often very ineffective and sometimes we harmful. Figure 4 shows some typical situations which work on the computer, but do not work on a real piping system. These pitfalls are caused by the differences between the real system and the computer model. Some important discrepancies are described in the following.

FRICTION is important in the design of the restraint system near n equipment. Figure 4 (a) shows a typical stop placed against a long -direction line to protect the equipment. In the design calculations, if the friction is ignored, the calculated reaction at the equipment is lite very small. However, in reality, the friction at the stop surface will prevent the pipe from expanding to the positive X-direction. This iction effect can cause a high X-direction reaction to the equipment. calculation including the friction will predict this problem forehand. A proper type of restraint, such as a low friction plate or a tru would then be used.

An INEFFECTIVE SUPPORT MEMBER is another problem often encountered in the protective restraints. Figure 4 (b) shows a popular arrangement to protect the equipment. The engineers direct instinct is to always put the fix at the problem location For instance, if the computer shows that the Z-direction reaction is too high, the natural x to place a Z-direction stop near the nozzle connection. This may e all right on the computer, but in reality it is very ineffective. For support to be effective, the support member A has to be at le st one order of magnitude higher than the stiffness of the pipe which is en stiff in this case due to the support's relatively short distance r the nozzle.

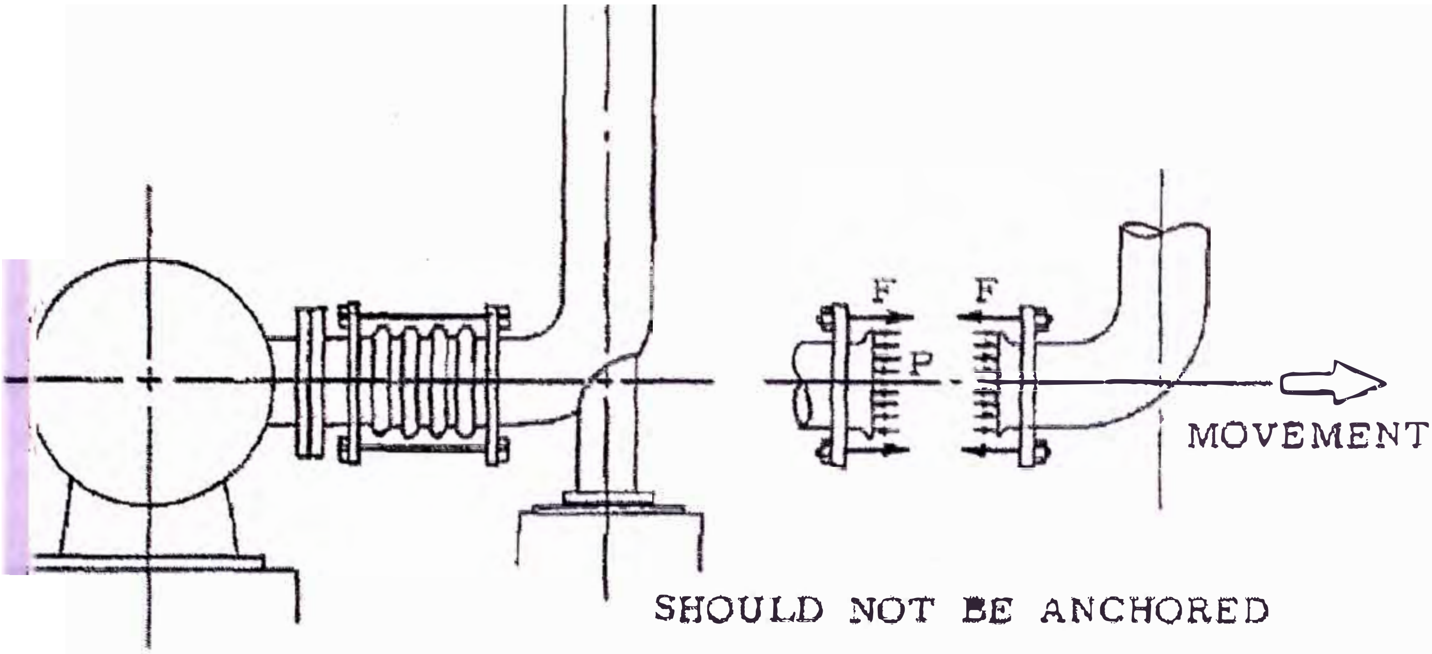
A GAP is generally required in the actual installation of a stop. Therefore, if a stop is placed too close to the nozzle connection, its tiveness is questionable due to the inherent gap. As shown in e 4 (c), because of the gap, the pipe has to be bent or moved a nce equal to the gap before the stop becomes active. Due to the ness of the stop to the equipment, this is almost the same as be ding the equipment that much before the pipe reaches the stop. This is ot acceptable, because the equipment generally can only tolerate a much smaller deformation than the construction gap of the stop.

CHOKING is another problem relating to the gap at the stop. Some engineers are aware of the consequences of the gap at the stop metioned above and try to solve it by specifying that no gap be allowed at the stop. This gives the appearance of solving the problem, but another problem is actually waiting to occur. As shown in Figure 4 (d), when the gap is not provided, the pipe will be choked by the stop as soon as the pipe temperature starts to rise. We all know to pay attention to the longitudinal or axial expansion of the pipe, but we often forget that the pipe expands radially as well. When the temperature rises to a point when the radial expansion is completely choked by the support, the pipe can no longer slide along the stop surface. The axial expansion will then move upward, pushing the whole e ipment up.

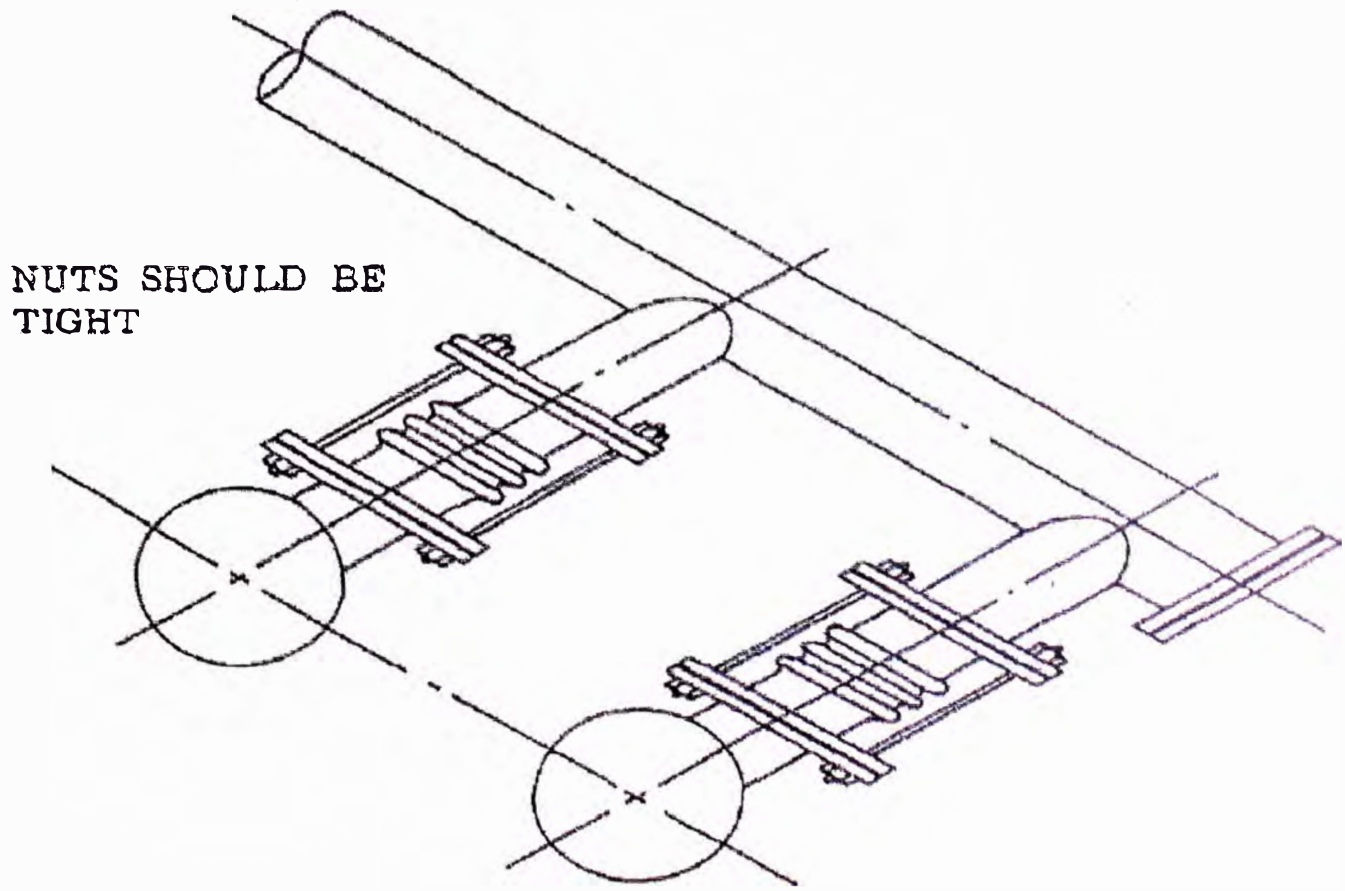
ANSION JOINT

An alternative solution to meet the allowable pipe loading to an equipment is the use of bellow expansion joints. Regardless of the constant objection from plant engineers, the bellow expansion joint is ver popular in the exhaust system of a steam turbine drive which has an extremely low allowable pipe load for pipes 8" and above. The bellow joints are also often used for fitting the large multi-unit assemblies as shown in Figure 5 (b). Although a properly installed and maintained bellow expansion joint should have the same reliability as other components, such as flanges and valves. In real applications, it s ften found to be very undesirable due to the difficulty in maitenance. For instance, when covered with insulation, the e ansion joint looks just like a pile of blanketed scraps. Nobody knows exactly what is going on inside the mixed layers of covering. Du to blindness anxiety, many installers have resorted to an isulated arrangement. This not only creates an occupatio al safety cern, but it can also cause cracks due to thermal shock from the ronment and/or weather changes.

One important factor often overlooked by engineers in the n llation of a bellow expansion joint is the pressure thrust force n le the pipe. The bellow is flexible axially. Therefore, the bellow is able to transmit or absorb the axial internal pressure end force. pressure end force has to be resisted either by the anchor at the pment or by the tie-rod straddling the bellow. With the exception of ery low pressure applicators, such as the pipe connected to a



(a)



(b)

Figure 5, Tie-Rods on Expansion Joints

s·rage tank, most equipment are not strong enough to resist the p·ssure end force equal to the pressure times the bellow cross section a·a. The pressure thrust force has to be taken by the tie-rod. S·nehow this idea is not obv'ous to many engineers, resulting in some o·erational problems. Figure 5 shows two actual problems. Figure 5 (a shows one of many steam turbine exhaust pipings installed at a p·rochemical plant. The expansion joint layout scheme appears to be s·und, but the construction was not done properly. The actual ir·tallation had a sliding base elbow anchored with four bolts. This p·blem often escapes the eyes of even experienced engineers. When th base elbow is anchored, the tie-rod loses its function as soon as th pipe starts to expand. In this case, the pipe expands from the a·chor toward the bellow joint, making the tie-rod loose and i·effective. The large pressure thrus' force pushes the turbine, c·using shaft misalignment and severe vibrations. Figure 5 (b) is a si·ilar situation. The bellow expansion joints were used solely for fi·ng up the connections. The tie-rods were supposed to be locked. H·wever, before the start-up operation, one engineer had loosened the ti·rod nuts, apparently thinking the tie-rods defeat the purpose of the e·pansion joint. The start-up was very shaky and had to be quickly h·ted. It took quite awhile before anyone discovered that the problem w·s caused by the loose tie-rods. When the nuts are loose, the p·ssure end force simply pushes the pump way out of alignment.

O·HER PRACTICAL CONSIDERATIONS

As discussed above, the reduction of pipe stress is not at all st·aight forward. Especially when dealing with the low allowable of ·ne equipment, the technique becomes tricky and very often only w·ks on paper. Other practical approaches may be explored to further i·rove overall reliability. One very important resource often ignored in this country is the experience found in operating plants. We often ·: a good, simple working layout changed to a complicated, shaky la·out only because a computer liked it that way. Undoubtedly, ·nputers are important tools, but they are only as good as the in·rmation we give them. Since there are so many things, like fri·tion, anchor flexibility, etc., that cannot be given accurately, c·nputer results need to be interpreted carefully. It is time to realize th·t if something works well in a plant day in and day out, it should be ·isidered good, regardless of whether or not the computer predicted it to be good. The process of evolution is very important in designing a ·od, reliable plant.

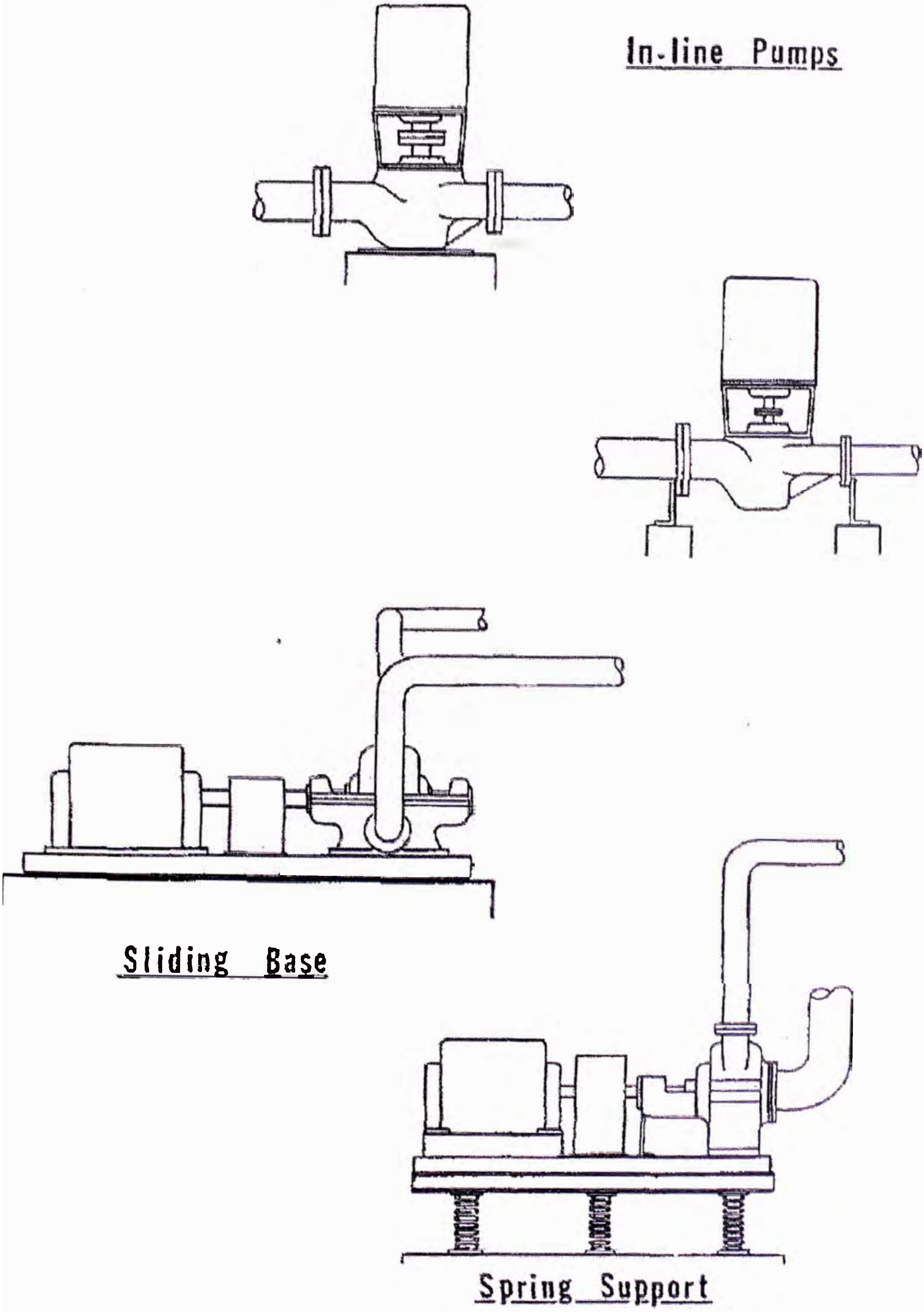


Figure 6, Alternative Machine Assemblies

Other ideas, such as the use of sliding supports, spring supports, a more compact in-line arrangements as shown in Figure 6, can also be seriously considered. It is understood that engineers do not feel too confident on the movable assembly, but it is important to distinguish the difference between the movement of the whole assembly and the movement of only the pump or turbine. When the whole assembly moves, the shaft alignment can still be maintained if the distortion of the equipment is not excessive. That is, if the piping load is still within the allowable. It should be noted, however, that these movable assemblies are just potential alternatives. One should not be oversold the idea and blindly use it in a plant. To make the sliding base or the spring support scheme workable, an extra strong baseplate is required. Then again, if we have that strong of a baseplate in the first place, the allowable piping load would have increased substantially.

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Hazardous situations created by improper piping analyses

Some computer programs do not handle hot condition calculations correctly

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Piping flexibility and stress analysis is required in design of most piping systems before the piping is installed in an industrial plant. It is intended to ensure plant safety and, thus, protect the interests of the owner and the general public. Due to availability of powerful software packages, the analysis has become simple and routine. However, improper piping analysis may actually create a hazardous situation rather than ensure plant safety.

Improper piping analyses stem from many areas. Misunderstanding of the software approach, input errors and wrong boundary conditions from CAD data are some common ones. These common mistakes are easy to detect and check if proper isometrics and input data echoes are provided. However, some areas of improper analysis are not as obvious and may even be misconstrued.

One area of improper analysis, which is openly mishandled by some software packages, warrants special attention. This improper analysis involves the so-called piping resting or single-acting support. The resting support allows the pipe to move up without restriction, but will support the pipe preventing it from moving downward. It is obvious that when the pipe moves up and off the contact point, it is no longer being supported. This very fundamental fact somehow is not recognized by some popular software packages and their users. They treat the pipe as being supported in the hot condition simply because it is supported during the cold condition.

To ensure structural integrity of the pipe, at least two stress categories normally have to be evaluated. One is sustained stress that is controlled by the pressure and weight. The other is expansion or displacement stress that is determined by thermal expansion of the pipe and movement of the connecting equipment. The sustained stress maintains the same magnitude through plant operating life. Its allowable is determined by the pipe's yield, rupture and creep strength at operating temperature.

On the other hand, the displacement stress relaxes once the material yields or the temperature reaches a certain point. Its allowable is determined by the strain range that is conveniently measured by the strength at cold condition, adjusted slightly by the strength at operating temperature. Treating a pipe that lifts off its support as being supported can greatly underestimate the sustained stress. It may not create a noticeable problem in the beginning, but the seemingly satisfactory piping system may actually have a safe operating life of only a small fraction of what is intended. This situation can better be explained by a typical example.

Piping system example. A typical piping system is commonly placed directly on the support structure as shown in Fig. 1. This is the

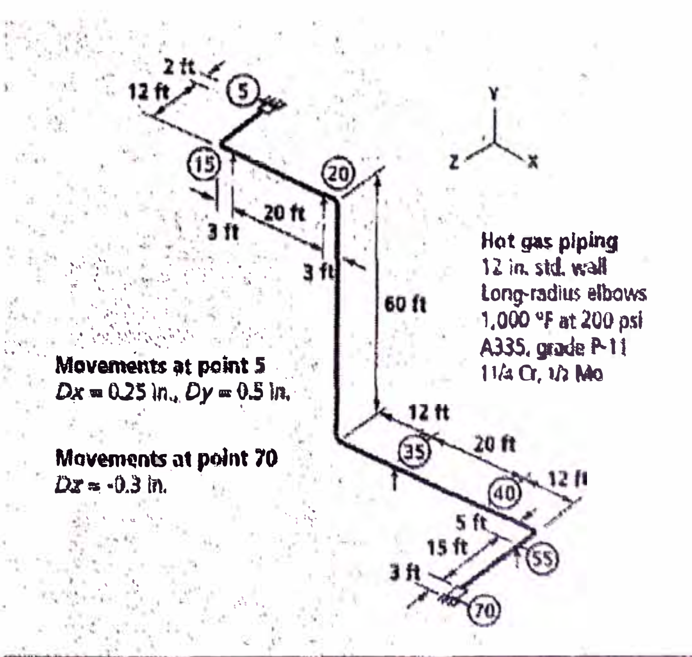


FIG. 1. A typical piping system is commonly placed directly on the support structure.

most economical and direct approach of installing a piping system. Depending on operating temperature and pipe size, modifications are often required to ensure piping safety and operating life. In this particular example, because of the 12-in. size and 1,000°F temperature, an experienced engineer will install one or more spring supports to ensure that the sustained stress due to weight and pressure is not excessive at operating temperature.

However, the situation may appear to be different to a computer-oriented engineer, because the system as it is laid out will meet piping code requirements when run through some popular computer programs. According to these computer programs, no spring support is required. This, on paper, saves substantial money for the owner. But in reality, it puts the plant in great danger. To understand this situation, finding out how a computer program handles resting supports is necessary.

Computer programs have become an essential part of piping stress analysis. As the technology progresses, these computer programs also get more powerful and sophisticated. The capability of handling resting supports and other single-acting restraints has been available for over two decades. To analyze the resting support, a good

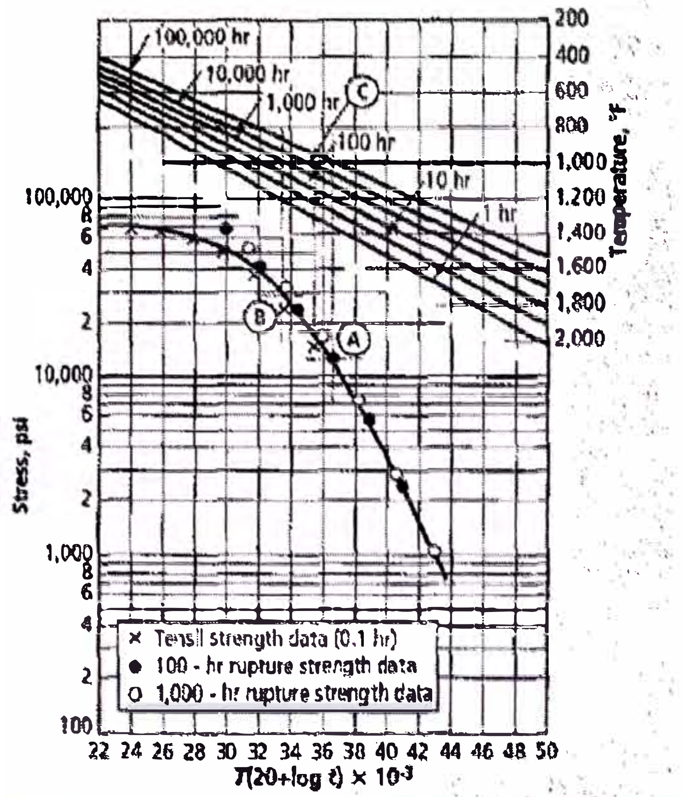


FIG. 2. Larson-Miller graphs can be used to estimate piping life at high temperatures.

mputer program will first check the normal operating condition to if the pipe will lift off a support when operating. If the pipe will off from a support, then that particular support will be unable to pport weight during operating conditions. That support will be nored in calculating sustained stress. Therefore, with the Fig. 1 ample, the program will show that supports at points 15 and 20 are t active. Without these supports, the piping is greatly over-stressed der weight and pressure. This in turn signals the engineer to add me spring supports.

This kind of simple logic, however, is not adopted by all comter programs. Some popular computer programs instead use a npler, but tricky, approach by calculating the sustained stress at ld condition assuming all supports are active. They then calculate e thermal expansion stress by allowing the pipe to move up on all ting supports. This erroneous approach completely ignores the stained stress at operating conditions, when the plant life is counted. e piping may be all right by their account, but this simplified proach of calculating sustained stress has already greatly shorted the piping life.

To estimate piping life at high temperature, Larson-Miller graphs n be used. Fig. 2 shows the Larson-Miller graph for 1¼ Cr, ½ Mo el. The graph shows that at 1,000°F, for an 11,700 psi stress (point the rupture time is 100,000 hr. However, for a stress of 28,500 psi oint B), rupture time is only 1,000 hr. According to ASME piping de, allowable sustained stress is taken as 67% of the average creep pture stress at 100,000 hr. In other words, if the piping is designed thin the allowable stress, it should last 100,000 hr at full temperure with a safety factor of 1.5 applied to the stress value. As given ove, for 1¼ Cr, ½ Mo steel at 1,000°F, average rupture stress at 0,000 hr is 11,700 psi. Therefore, the allowable is 11,700 × 0.67 7,800 psi. This allowable stress value is revised periodically based on the latest available data. The current allowable is 6,300 psi.

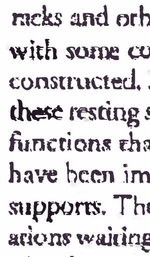
With the piping system in Fig. 1, the correct maximum sustained stress, considering the inactive supports removed, is 18,000 psi. This greatly exceeds the code allowable. However, maximum sustained stress calculated assuming all supports are active, as performed by some popular computer programs, is only 5,800 psi, which is within the piping code allowable. The difference between these two results is very significant. The piping may have been checked to be in compliance with the code by some computer programs, when in reality it is highly overstressed.

With an 18,000 psi actual sustained stress, the pipe will rupture in about 20,000 hr (Fig. 2, point C) without any safety factor applied. If a safety factor of 1.5 is applied to the stress, then piping safe operating life is only about 10 thousand hr. This is much less than the 100,000 hr intended by the code. It is apparent here that the piping analyzed by assuming all the resting supports are active, as in the cold condition, can create a real hazard to the plant.

Evaluating supports correctly. When piping lifts off from the support during operation, its sustained weight stress should be calculated considering the support as inactive. This common sense is not recognized by many engineers. It is even more puzzling that this situation is not handled correctly by some popular computer programs. Some engineers have argued that the pipe will eventually settle to the support either by yielding or by creeping. This may or may not be true, depending on the amount of uplift and the piping configuration.

Even if the pipe eventually settles to the support, it still raises three major issues: The piping is still not in compliance with the piping code, which requires that the sustained stress be within the allowable all the time, not just some of the time; before the pipe is eventually supported, it may already have been sufficiently damaged; and when the pipe settles to the support at operating temperature, usually huge stresses and loads will be generated during the cool down. This is because the support will prevent the pipe from moving downward as required by the shrinkage due to cool down.

The most common and economical approach in dealing with the countless piping in a process plant is to rest the piping on pipe racks and other support structures. Using this approach, combined with some commonsense engineering, many safe plants have been constructed. However, since the use of powerful software packages, these resting supports have been misinterpreted to have some magic functions that do not exist. Validated by the computer, engineers have been improperly designing many piping systems with resting supports. They may have already created numerous hazardous situations waiting to compromise the plant safety. Therefore, it is imperative that owners and operators of these plants evaluate these support situations thoroughly to ensure that the investment and safety of the general public are protected. **HP**



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Boiler Fitness Survey for Condition Assessment of Industrial Boilers

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BR-1635

Abstract

In today's markets, reducing costs and optimizing the production of installed capacity is essential. The electric utilities have depended upon life extension of existing boilers since the early 1980s to meet generation needs. Similarly, general industry faced with competition and cost control are looking to extend the lives and in many instances upgrade the capacities of existing boilers. The Babcock & Wilcox (B&W) boiler fitness survey has proven to be an effective program for analyzing the condition of aging boilers. The extent of the fitness survey is geared to the level of cost and effort that will provide the plant owner with the information needed to make decisions for future operation of the boiler(s); as such, the survey can be tailored to the needs of the customer. The following describes the method of developing the scope of the boiler fitness survey as well as the key components, damage mechanisms, nondestructive examination methods, and techniques for determining boiler remaining useful life.

Introduction

For many years condition assessment and life extension programs have been common in the electric utilities. Much of the emphasis in the utility generating stations has been on components with expected finite lives where degradation and failure are associated with creep and creep-fatigue. Components such as steam outlet headers, main steam and reheat steam piping, and steam turbines are all subject to eventual material failure from operating at high temperatures and stresses. As a general class, industrial boilers typically operate at much lower temperatures and pressures. As a result, the life of these boilers is not necessarily defined by a finite material life. In fact there are numerous examples of very old boilers (>50 years operating life) which can still be operated. Many times these older boilers are retired for reasons other than reliability or safety.

Today, the need to establish formalized programs for assessing the condition of boilers in industrial plants is becoming increasingly important. Industrial installations are under the same pressures to reduce costs as are the electric utilities. Additionally, boiler upgrades and capacity increases are potential lower cost options for plants anticipating expansions. Babcock & Wilcox's boiler fitness survey is a comprehensive program for assessing the integrity and fitness of aging boilers. Boiler fitness surveys combine the knowledge of B&W's field service engineering with the specialized inspection technologies and experience that have been developed by B&W for boiler condition assessment. Presented in the following paper is the program and methods recommended by Babcock & Wilcox for the assessment of industrial boilers.

Industrial versus Utility

The condition assessment program for industrial boilers is not unlike the approach used for the electric utilities, except for the primary mechanisms of failure and the inspections methods that are used. Industrial boilers come in all shapes and sizes and can be found in a variety of industries and institutions. For purposes of condition assessment classification, industrial boilers are those units designed for outlet steam temperatures of 900 F (482 C) or less with design pressures that typically do not exceed 1500 psig (104 bar). By contrast, the modern era utility boiler is designed for outlet steam temperatures of 1000 F to 1050 F (538 to 566 C), with design pressures ranging from 1800 to 3850 psig (124 to 265 bar). Utility class boiler designs can be found in non utility plants: however, the discussion presented below will focus on the industrial class boiler designs.

Components that are more likely to have adverse effects on boiler performance as they deteriorate with age include:

- Air heaters - recuperative (tubular), regenerative (Ljungstrom), steam coil
- Fans - induced draft, forced draft, primary air (pulverized coal firing)
- Burners
- Fuel preparation equipment (especially coal firing, i.e. pulverizers)
- Boiler settings such as casing and BRIL (brickwork, refractory, insulation and lagging)
- Structural supports

Nondestructive Examinations (NDE)

NDE can be an important part of the boiler fitness survey. NDE is done to obtain sufficient data to allow the engineer to assess and make decisions regarding the integrity of the component. The choice of NDE methods will depend upon location and type of potential damage as well as the limitations caused by the arrangement and geometry of the component itself. NDE methods that have been used by B&W on industrial boilers are visual examination (VE), magnetic particle testing (MT) and wet fluorescent magnetic particle testing (WFMT), liquid dye penetrant testing (PT), ultrasonic testing (UT), remote field eddy current testing (RFEC), electromagnetic acoustic transducer based testing (EMATs), metallographic replication (MR), and acoustics or acoustic emissions (AE) testing. Radiography, an important NDE method for field testing of welds, is primarily used to test welds following boiler erection or repair and is not normally used for fitness surveys.

NDE methods have advantages and disadvantages and it is important to select the correct method for each component. The extent that an NDE method can be used depends on the access to the component as well as surface preparation. Described below are the types of problems found in the various components as well as the recommended NDE methods. In general, visual examination, the most basic of NDE, is done for all components. It is common to photo-document the inspection to provide a permanent record in the report. Internal inspections are frequently done by video probe and recorded on tape.

Drums: The steam drum is the single most expensive component in the boiler. Consequently, any assessment program must address the steam drum as well as any other drums in the convection passes of the boiler. In general, problems in the drums are associated with corrosion. In some instances, where drums have rolled tubes, rolling may produce excessive stresses that can lead to damage in the ligament areas. Problems in the drums normally lead to indications that are seen on the surfaces – either ID or OD. *Assessment: Inspection* and testing focuses on detecting surface indications. The preferred NDE method is WFMT. Because WFMT uses fluorescent particles which are examined under ultraviolet light it is more sensitive than dry powder type MT and it is faster than PT methods. WFMT should include the major welds, selected attachment welds and at least some of the ligaments. If locations of corrosion are found then ultrasonic thickness testing (UTT) may be performed to assess thinning due to metal loss. In rare instances metallographic replication may be performed. Replication is done by polishing the surface of the drum to a mirror finish, etching the polished surface with a nital acid, and then lifting an image of the metal surface by applying a softened acetate tape (the replica). The procedure, analogous to finger printing, allows the metal grain structure to be examined under a microscope.

Headers: Boilers designed for temperatures above 900 F (482 C) can have superheater outlet headers that are subject to creep – the plastic deformation (strain) of the header from long term exposure to temperature and stress. For high temperature headers, tests can include metallographic replication and ultrasonic angle beam shear wave inspections of higher stress weld locations, as well as B&W's bore hole Hone & Glow® test of ligaments to detect damage associated with creep and creep-fatigue. However, industrial boilers are more typically designed for temperatures less that 900 F (482 C) such that failure is not normally related to creep. Lower temperature headers are subject to corrosion or possible erosion. Additionally, cycles of thermal expansion and mechanical loading may lead to fatigue damage. *Assessment:* NDE should include testing of the welds by MT or WFMT. In addition, it is advisable to perform internal inspection with a video probe to assess waterside cleanliness, to note any buildup of deposits or maintenance debris that could obstruct flow, and to determine if corrosion is a problem. Inspected headers should include some of the water circuit headers as well as superheater headers. If a location of corrosion is seen then UTT to quantify remaining wall thickness is advisable.

Piping – Main Steam: For lower temperature systems the piping is subject to the same damage as noted above for the boiler headers. In addition the piping supports may experience deterioration and become damaged from excessive or cyclical system loads. *Assessment*: The NDE method of choice for testing of external weld surfaces is WFMT. MT and PT are sometimes used if lighting or pipe geometry make WFMT impractical. Non drainable sections such as sagging horizontal runs are subject to internal corrosion and pitting. These areas should be examined by internal video probe and or UTT measurements. Volumetric inspection, i.e. ultrasonic shear wave, of selected piping welds may be included in the NDE; however, concerns for weld integrity associated with the growth of subsurface cracks is a problem associated with creep of high temperature piping and is not a concern on most industrial installations.

Feedwater Piping: A piping system often overlooked is feedwater piping. Depending upon the operating parameters of the feedwater system, the flow rates, and the piping geometry, the pipe may be prone to corrosion or flow assisted corrosion (FAC). This is also referred to as erosion-corrosion. If susceptible, the pipe may experience material loss from internal surfaces near bends, pumps, injection points and flow transitions. Ingress of air into the system can lead to corrosion and pitting. Out-of-service corrosion can occur if the boiler is idle for long periods. *Assessment*: Internal visual inspection with a video probe is recommended if access allows. NDE can include MT, PT or WFMT at selected welds. UTT should be done in any locations where FAC is suspected to ensure there is not significant piping wall loss.

Deaerators: Overlooked for many years in condition assessment and maintenance inspection programs, deaerators have been known to fail catastrophically in both industrial and utility plants. The damage mechanism is corrosion of shell welds which occurs on the ID surfaces. *Assessment*: Deaerators' welds should have a thorough visual inspection. All internal welds and selected external attachment welds should be tested by WFMT.

Attemperators: The spray flow attemperator, a device for controlling superheater outlet steam temperature, is normally located in the piping system between the primary (1st stage) superheater outlet and the secondary (2nd stage) superheater inlet. The attemperator is subject to failures associated with thermal fatigue cracking of its components and welds. Since it is in a closed loop of the boiler, failures may go undetected until pieces of the attemperator lead to other damage, such as superheater tube failures. In addition to the B&W attemperator, condensing-type attemperators (commonly called "sweetwater condensers"), have experienced failures associated with fatigue. These steam temperature control systems should also be part of the boiler fitness survey testing. *Assessment:* For the B&W spray attemperator, inspection is recommended by removal of the spray head assembly. The spray head is inspected visually and tested nondestructively by MT/PT methods. Following removal of the spray head from the body of the attemperator, the attemperator thermal liner can be internally inspected with a video probe. Sweetwater condensers are subject to damage primarily from water hammer leading to cracks in the condenser shell. All welds on the internal surface of the condenser shell as well as shell connections and auxiliary piping should be inspected by MT, PT methods. If internal access to the condenser shell is not possible then ultrasonic angle beam shear wave testing can be done to detect shell cracking.

Tubing: By far the greatest number of forced outages in all types of boilers are caused by tube failures. Failure mechanisms vary greatly from the long term to the short term. Superheater tubes operating at sufficient temperature can fail long term (over many years) due to normal life expenditure. For these tubes with predicted finite life, B&W offers the NOTIS® test and remaining life analysis. However, most tubes in the industrial boiler do not have a finite life due to their temperature of operation under normal conditions. Tubes are more likely to fail because of abnormal deterioration such as: water/steam-side deposition retarding heat transfer, flow obstructions, tube corrosion (ID and/or OD), fatigue and tube erosion. *Assessment:* Tubing is one of the components where visual examination is of great importance because many tube damage mechanisms lead to visual signs such as distortion, discoloration, swelling or surface damage. The primary NDE method for obtaining data used in tube assessment is contact UTT for tube thickness measurements. Contact UTT is done on accessible tube surfaces by placing the UT transducer onto the tube using a couplant, a gel or fluid which transmits the UT sound into the tube. A new measurement technique developed by B&W under an EPRI-sponsored project is the FST-GAGE™ [3] which utilizes EMAT, ElectroMagnetic Acoustic Transducer technology. EMAT utilizes electromagnetic induction to produce the ultra frequency pulse in the tube; this eliminates the need for a couplant. Contact UTT and the FST-GAGE have very accurate measurement capability which gives a measurement within plus or minus 0.005 in. (0.127 mm). The FST GAGE's UT accuracy without need of couplant makes it excellent for scanning tubes where isolated damage is a concern. Variations on standard contact UTT have been developed due to access limitations. Examples are IRIS-based techniques (internal rotating inspection system) in which the UT signal is reflected from a high RPM rotating mirror to scan tubes from the ID - especially in the area adjacent to drums; and B&W's immersion UT where a multiple transducer probe is inserted into boiler bank tubes from the steam drum to provide measurements at four orthogonal points. IRIS and immersion UT require tubes to be flooded with water. Remote field eddy current (RFEC) probes have also been developed for internal inspection of tubes. RFEC has the advantage of not requiring a tube to be flooded or have a column of water but it has the disadvantage of not providing the measurement accuracy of UT. Tube inspection systems based on laser profilometry have been developed that provide for inspection and mapping of tube surface topography. These systems can be advantageous in the assessment of pitting.

Remaining Life

Various methods have been developed for assessing the remaining useful life (RUL) of key boiler components. For thin wall high temperature components, such as superheater tubing, failures are associated with creep rupture. Analysis of RUL is done using well established life fraction theories and creep-rupture material data bases.[4] For heavy section components such as headers and piping that operate at high temperature, failures are characterized by the initiation and growth of cracks under the influence of creep and/or fatigue. Analysis methods have also been developed which consider time dependant fracture mechanics (TDFM) and that allow quantification of component life based on crack growth.[5] These creep related life prediction methods are not normally needed for the industrial boilers because of their lower operating temperatures.

For the industrial boiler the most common tool for RUL assessment is analysis of corrosion and erosion rates and comparison of actual component wall thicknesses versus American Society of Mechanical Engineers (ASME) Boiler Code calculated minimums. Since tube life and tube failure tend to be the major cause of forced outages in aging boilers, RUL of low temperature tubes is a large part of industrial tube RUL assessment. In 1985, B&W developed a guideline that boiler owners could use for setting a flag or bench mark thickness in assessing tube wall measurement data. The guideline, released as B&W Plant Service Bulletin PSB-26, Tube Thickness Evaluation Repair or Replacement Guideline, was intended to help boiler owners by giving them guidance, while not being overly conservative. For many boilers, tube failures most often have economic consequences such that the owner/operator will operate existing components as long as it is economically justified compared to the cost of replacement. For these units, many years of component life may exist in the margin between ASME Code minimum and the absolute minimum where failures occur. For water-cooled tubes, B&W guidelines established 70% specified wall thickness for the flag point below which the tube should be replaced. This guideline assumes that specified wall is approximately ASME minimum – a valid assumption unless the original design included tube wall thickness with a corrosion allowance over and above ASME minimum. The guidelines are set so that tube replacement is planned before tube stresses reach the yield strength of the tube. The PSB flag point guideline only applies to boilers where tube leaks occur internal to the boiler settings as a result of fire-side wall loss from erosion and corrosion, and tube failures do not pose an unusual safety risk. Failures that occur external to the boiler setting from mechanisms such as acid attack or corrosion fatigue must be treated with greater urgency and caution if the tube leak can cause injury to personnel. In addition, special circumstances dictate necessarily conservative guidelines. For example, process recovery boilers in the pulp and paper industry can experience an explosive smeltwater reaction if tube leaks occur in the water wall tubes. For these boilers, tube replacement is recommended when the tube wall drops below ASME minimum thickness.

Summary

Extending the life of industrial boilers which operate at lower temperatures and pressures is a viable option to support plant steam production needs. The boiler fitness survey has proven to be an effective program for determining the current condition and remaining useful life of aging industrial boilers. Extensive visual examination by an experienced B&W Field Service Engineer complemented by NDE allows comprehensive assessment and provides the owner with the information needed to make long range decisions regarding the subject boiler.

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RISK BASED INSPECTION (RBI) OF STEAM SYSTEMS

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ABSTRACT
This paper describes the implementation of a risk-based inspection program for process and utility steam lines in a large chemical process facility. The paper addresses first the development of an RBI matrix, the likelihood attributes, the consequence scores, and the overall risk in terms of personnel safety and costs. Systems are plotted on the RBI matrix to develop inspection priorities. The RBI ranking is followed by inspection planning, acceptance criteria, and wall thickness inspection techniques, including UT, pulsed eddy current and digital radiography.

INTRODUCTION
Process and utility steam systems are an important part of chemical process plant operation. The integrity of some steam systems is essential to safe and reliable operation; and these systems must be inspected and maintained regularly. The challenge of every operation is to implement a necessary and sufficient inspection strategy. This is achieved by risk ranking steam systems across the plant, focusing inspection resources on high risk systems. There are several good techniques for risk-based inspections (RBI). The RBI method selected and presented here is generally based on the principles of the American Petroleum Institute's API Recommended Practice 580 Risk-Based Inspection[1] adapted to reflect the specifics of steam systems in process plants. The steam systems convey saturated steam from 15 psi up to to 350 psi.

FIVE STEPS
The RBI process consists of five steps, as outlined in Figure 1. This paper addresses the first two blocks of Figure 1: Risk ranking and Inspection Planning: How we decided which systems to inspect and why, which inspection techniques to apply, and which acceptance criteria to use to make run-or-repair decisions.

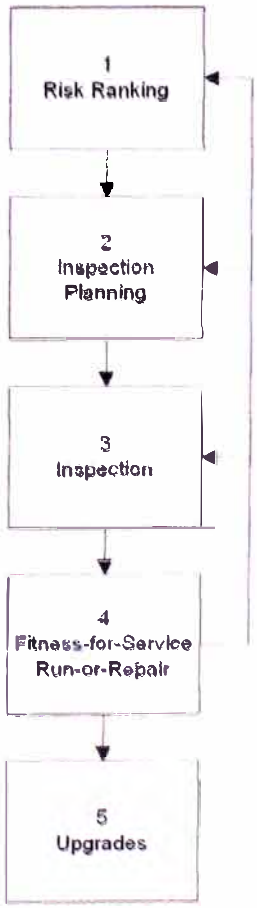

Figure 1 - Five Steps of RBI Process

more evident than 1 to 5 or A to E used in the API matrix.

- The API matrix was modified to extend the "Low Risk" region to encompass all "Very Low" consequence events, as shown in Figure 2.

A joint management and engineering team was convened to develop the lines of inquiries which would define the likelihood and consequence scores, and populate the risk matrix.

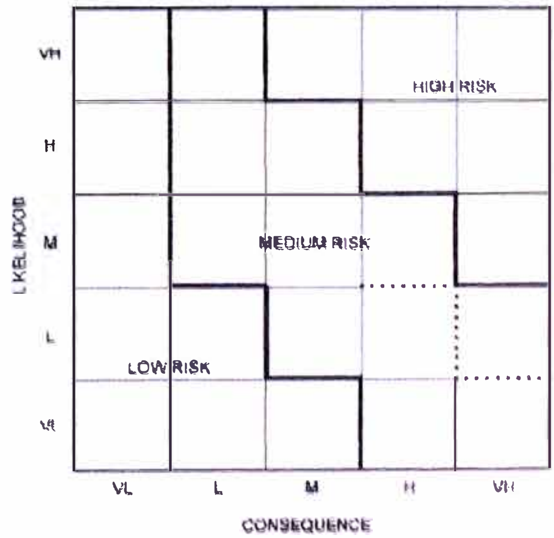

Figure 2 - Risk Matrix

LIKELIHOOD OF FAILURE

Steam systems are assigned a likelihood of failure score, from VL (very low) to VH (very high), following lines of inquiry related to failure history, corrosion potential, process upsets, and age. Scores are assigned individually to each attribute and then averaged to provide the total likelihood of failure. The attributes and corresponding scores are:

(1) Prior failure
- None = VL
- In similar systems in industry = L
- In similar systems on-site = M
- In the particular system being assessed = VH

(2) Degradation mechanisms
- None = VL
- Possible = M
- Known = VH

(3) Novelty of Process
- None = VL
- Some = M
- New process = VH

(4) Abnormal loads (in our case, steam hammer)

CODES & STANDARDS

The design and construction code for facility steam piping systems is ASME B31.1[2] for main steam headers and transmission to the facilities and ASME B31.3[3] for process steam within the process areas. Non-mandatory Appendix V of ASME B31.1 refers to "continued examination" to be conducted "at intervals based upon the results of the initial inspection, but not to exceed 5 years", and the practice in ASME B31.1 fossil power plants is to indeed inspect periodically high risk systems such as main steam and hot reheat. ASME B31.3 does not address periodic inspections or maintenance.

API 570[4] is a piping inspection code widely used in the refining and petrochemical industries to inspect flammable and toxic systems, but it does exclude "Water (including fire protection systems), steam, steam-condensate, boiler feed water, and Category D fluid services, as defined in ASME B31.3". The inspection interval in API 570 is based on risk: the likelihood of failure based on corrosion rate, and the consequence of failure based on the system "class", from class 1 (most critical) to class 3 (less critical).

The National Board Inspection Code ANSI-NB-23[5], is a code commonly imposed by State or local jurisdictions for maintaining the safe operation of boilers and certain pressure vessels and piping systems. Section RB addresses "Inservice Inspection of Pressure-Retaining Items" including inspection of piping systems. The NBIC recognizes that "frequency of test and inspection of ... piping service is greatly dependent on the nature of the contents and operation of the system and only general recommendations can be given". It does recommend an annual inspection of steam piping systems (NB-23 Section RB-8410).

API recommended practice RP 580[1] and publication 581[6] provide guidance for the development of risk-based inspection (RBI) programs. While these documents are intended for petroleum and petrochemical applications, the concepts of RBI are readily applicable to other process systems, and are currently being adapted by the ASME Post-Construction Committee in developing an inspection planning standard.

RISK RANKING – THE RISK MATRIX

Risk ranking is achieved through a 5x5 matrix of likelihood and consequence, the 5x5 format is adopted from API 580 Risk-Based Inspection and API 581 Risk-Based Inspection Base Resource Document. With two modifications:

- The five categories are ranked VL (very low), L (low), M (medium), H (high) and VH (very high), which is

None = VL
Low possibility = L
Possible = M
Anticipated to occur = VH

(5) System age
Less than 5 years = VL
5 to 15 years = M
15 to 30 years = H
Over 30 years = VH

Likelihood points were selected and assigned for each attribute, within the following ranges:
VL = 0 to 20 points
L = 21 to 40 points
M = 41 to 60 points
H = 61 to 80 points
VH = 81 to 100 points

LEAK OR BREAK

It became evident, early in the ranking process that we would have to differentiate between the likelihood of a leak as in Figure 3 (not uncommon in steam systems), and a break (rupture) as in Figure 4 (rather uncommon in steam systems).

Figure 3 – Pinhole Caused a Leak

For our facilities, steam line leaks and a ruptures were assigned the likelihood scores in Table 1.

Table 1 – Site-Specific Results of Likelihood Scores

Likelihood Attributes	Leak	Rupture
(1) Prior failure	100	21
(2) Degradation mechanisms	100	41
(3) Novelty of Process	0	0
(4) Abnormal loads	50	21
(5) System age	100	81
Average Score	70 (High)	33 (Low)

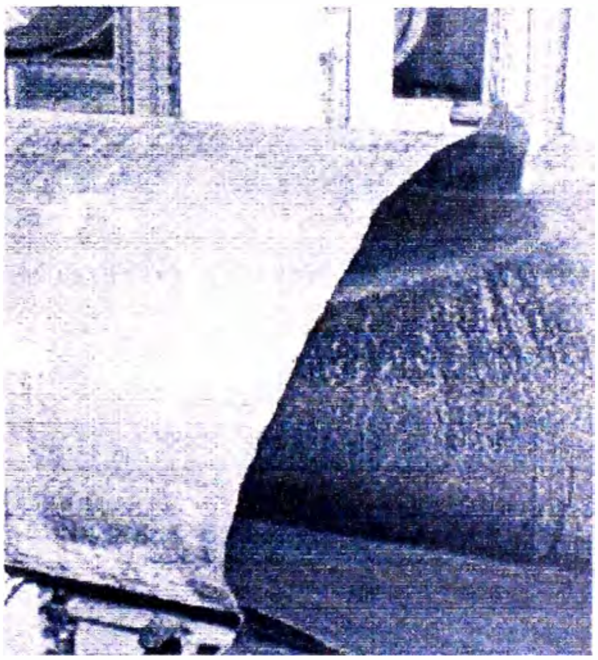

Figure 4 – Pipe Rupture

The score on abnormal loads (steam bubble collapse water hammer) reflects the plant experience, where large water hammers caused leakage but not rupture. In one such event, the steam hammer sheared off overhead supports, Figure 5, and buckled an expansion joint, Figure 6, but neither the carbon steel line nor the joint ruptured.

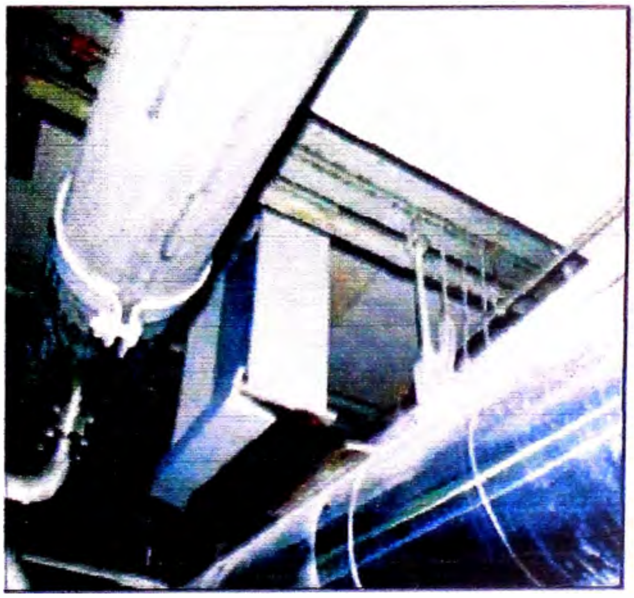

Figure 5 - Rupture of Overhead Steam Line Guide

Figure 6 - Buckled Steam Line Expansion Joint

CONSEQUENCE OF FAILURE

Steam systems are assigned a consequence of failure rank from VL to VH, following lines of inquiry related to public and worker health and safety, environmental damage, production impact, and recovery costs. Unlike likelihood, the consequence score is not an average; instead, it is assigned the worst ranking of all consequences. The consequence attributes are:

(1) Public health and safety
- None = VL
- Reportable = H
- Dangerous = VH

(2) Worker health and safety
- None = VL
- Potential lost time = L
- Probable lost time = M
- Potential fatality = H
- Probable fatality = VH

(3) Environmental impact
- None = VL
- Alert (degradation of critical controls) = L
- Site emergency = M
- General emergency (beyond site boundary) = H

(4) Operation impact
- Less than 1 month = VL
- 1 to 3 months = L
- 3 to 4 months = M
- 4 to 6 months = H
- Over 6 months = VH

(5) Recovery costs (cleanup and repairs)
- Less than $ 0.5 M = VL
- $ 0.5 to 1 M = L
- $ 1 to 2 M = M
- $ 2 to 5 M = H
- Over $ 5 M = VH

THE DANGERS OF STEAM

The assessment of consequence depended on whether the steam line leak or rupture was in a confined space, an enclosed space or outdoors. A confined space is a permitted entry space, consistent with OSHA 29 CFR 1920-146, unless it is amply ventilated. An enclosed space, for the purpose of the steam line evaluations, is a space such as an office, conference room, lunch room, etc., unless it is amply ventilated. The reason a steam leak is critical in a confined or enclosed space has to do with its burn potential to skin, throat and lung. Burn from steam is expected to occur under two conditions:

- If the ambient temperature reaches $T > 120°F$, and
- Moisture is exceeds 12% steam by volume

For a ¼" hole leak, these conditions are achieved within 1 minute, in an un-vented room of the volume listed in Table 2.

Table 2 - One-Minute Fatal Volumes

P (psi)	25	100	200	300
V (ft^3)	70	200	410	550

On the basis of the above facts, and the team's experience, the following conditions were identified for inclusion in the risk-ranking process. Conditions other than these would be of low risk:

Condition 1, OSHA confined space, entry permit required.
Condition 2, Enclosed space, with little ventilation (office, stairwell, etc.).
Condition 3, Outdoor, $P > 150$ psi, 2" and larger pipe, within 3' of walkways and roads.
Condition 4, Outdoor, $P \leq 150$ psi, pipe smaller than 2", within 3' of walkways and roads.
Condition 5, Steam piping within 3' of elevated walkway, ladder, without cage or rail.

The consequence rank was established on the basis of these five conditions, as summarized in Table 3.

Table 3 - Site-Specific Consequence Ranking

Condition	Rupture	Leak
1. Confined space	R1 = VH	L1 = H
2. Enclosed space	R2 = H	L2 = VL
3. Outdoor $P > 150$, etc.	R3 = VH	L3 = VL
4. Outdoor $P \leq 150$, etc.	R4 = M	L4 = VL

5. Within 3ft of elevated	R5 = VH	L5 = M

RISK RANKING

The Likelihood and consequence having been assigned, steam systems can now be plotted on the 5x5 risk matrix, Figure 7. The nomenclature for Figure 7 is described in table 3.

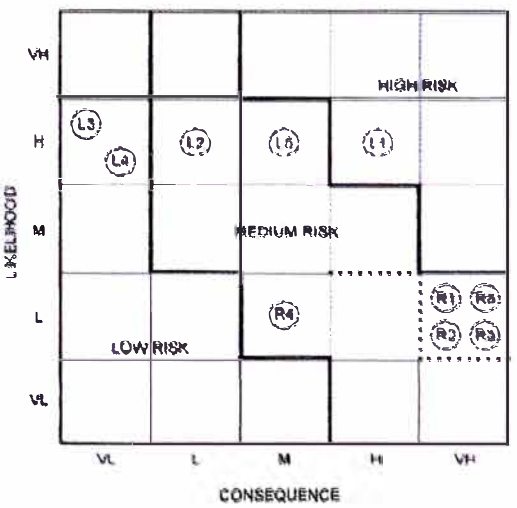

Figure 7 Risk Based Ranking of Steam Systems

RISK INSPECTION PRIORITIES

Only one category of steam systems (L1) ended up in the high risk category. These lines were prioritized for mandatory inspection.
Steam systems in categories R1, R2, R3 and R5, were also identified for mandatory inspections.
The inspection of lower risk categories (L2 to L5, R4) was left to the discretion of each operating unit.

CHECK

The results of the above risk ranking are quite logical:

- The RBI matrix justifiably differentiates between leaks and breaks, both on likelihood and consequence.
- The matrix highlights the particular risk associated with enclosed and confined spaces, which is well in line with industry experience related to steam fatalities.
- The RBI process was also an opportunity to better quantify the rapid heat-up of an enclosed space, quickly leading to a fatal environment.

EXCEPTIONS

Because operational experience is paramount in establishing risk of failure, each operating facility was provided the opportunity to review and approve the risk ranking or submit justification for exceptions, or dissention from the RBI ranking, to a Management Coordinating team. This did not occur, as each operating facility judged the RBI ranking to be applicable to their operation.

As an alternative to the inspection of steam in confined spaces, R1/L1, administrative controls may be put in place to prevent confined space entry where steam lines are in service. That is, R1/L1 locations could be eliminated from inspections if there are adequate controls and protections, or steam is isolated prior to entry into confined spaces.

INSPECTION PLANNING

Inspection planning consists of the following activities:

- Selection of inspection locations
- Selection of inspection technique
- Selection of acceptance criteria and method for fitness-for-service assessment

INSPECTION LOCATIONS GUIDE

Having identified the priority inspection system attributes, each steam engineer was requested to select a minimum of 10 inspection locations in each facility, in the high risk categories R1/L1, R2, R3 and R5. This resulted in close to 100 inspection locations.

The degradation mechanism for wall loss in carbon steel steam lines, in saturated steam service, is erosion corrosion (flow accelerated corrosion FAC). Guidance for selection of inspection points included:

- Areas of prior repairs and known corrosion
- Branch connections, at sides and bottom of pipe
- Outer arc (extrados) of elbows
- Up to 10-diameters downstream of elbows and orifice plates
- Dead legs
- At signs of damage to lagging with breach of insulation
- At signs of wetness or leakage
- At low points in vertical legs or sagging spans were condensate could accumulate
- Where water and steam mix (popping sounds from bubble collapse)
- Changes in cross section (reducer)

INSPECTION TECHNIQUES

The choice of inspection techniques has to address two challenges:

- Because the inspection will take place in winter time, the inspections will take place with steam lines in service.
- Many steam lines date back to the 1950's and have asbestos insulation.

In light of these difficulties, three inspection techniques have been evaluated: standard straight beam ultrasonic testing (pulse echo), pulsed eddy current, Figure 8, and digital radiography, Figure 9. Because the lines had to be inspected while in-service, avoiding removal of insulation, pulsed eddy current and digital radiography were the preferred techniques. Pulsed eddy current inspections had been contracted two years earlier, with good results. This time, the inspection technique selected was digital radiography.

Figure 9 is a digital radiography of a 1" schedule 80 steam pipe, it shows longitudinal striations as the flow enters the reducer section, and marked wall thinning in the weld itself.

Figure 8 - Pulsed Eddy Current Through-Insulation

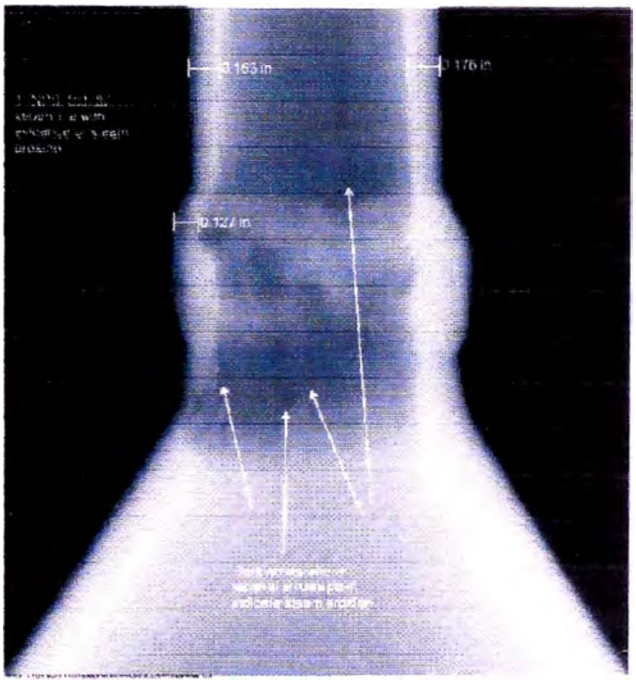

Figure 9 - Digital Radiography of Reducer

INSPECTION ACCEPTANCE CRITERIA

Prior to inspection, it is essential to define acceptance criteria. The design basis for the site piping systems is ASME B31.1 for steam production and distribution, and ASME B31.3 for steam use in facilities. A three level acceptance criterion was defined in terms of minimum wall thickness:

Green, steam line acceptable for continued service, re-inspect in 5 years, if

$$t_{mm} > t_{min} + 50 \text{ mils, and}$$
$$t_{mm} > 20\% \, t_{nom} + 50 \text{ mils}$$

The Green criterion above is based on (a) compliance with the ASME B31 required minimum wall, (b) providing a future corrosion allowance of 10 mpy x 5 years, till the next inspection in 5 years, and (c) assuring that there is no excessive wall loss, not to exceed 80% of the wall. The last criterion (c) is consistent with B31G[7] for oil and gas pipelines, and is similar in concept to API 579[8] fitness-for-service rules for general wall thinning.

Yellow, detailed evaluation required within 60 days, if

$$t_{mm} > t_{min}, \text{ and}$$
$$t_{mm} > 20\% \, t_{nom}$$

The yellow criterion above is based on (a) compliance with the ASME B31 required minimum wall, without a future corrosion allowance, and (b) assuring that there is no excessive wall loss, not to exceed 80% of the wall. This condition cannot be left as-is for 5 more years of service, it needs to be evaluated and resolved within 60 days.

Red, immediate assessment, safeguarding and shutdown if necessary, if

t_{mm} below yellow

t_{mm} = minimum measured wall thickness
t_{min} = minimum wall thickness required by ASME B31 code
t_{nom} = nominal wall thickness, in

For more detailed assessments, in particular for the evaluation of yellow and green readings, the rules of API 579, may be applied to evaluate the fitness-for-service and remaining life of the corroded steam line.

CONCLUSION

A risk-based inspection (RBI) program was developed and is being implemented to prevent steam accidents that could jeopardize safety or production. Steam systems were ranked on the basis of likelihood and consequence of failure. Inspection techniques and acceptance criteria were developed and will be applied to inspect utility and process steam lines, site-wide.

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THE ART OF DESIGNING PIPING SUPPORT SYSTEMS

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ABSTRACT

The standard approach to pipe support design is to follow well known, accepted, practices; the art of designing support systems is to go beyond these common practices.

This paper uses specific examples to demonstrate that the use of some common practices can lead to real problems in some situations. Support type based on vertical thermal displacement, spring loads set to balance the weight at hot condition, anchors at expansion joint installation, springs sized to minimize the vertical load at equipments are among the specific items discussed. The potential of utilizing friction forces to replace the expensive snubbers is also presented.

INTRODUCTION

A basic design is normally created by following common rules formulated from the past experiences of the industry as a whole. The common rules are essential for the day to day design practice. However, the rules passed from generation to generation are only those which are broad enough and simple enough to warrant a space in the company standards or technical books. These rules are valid for most of the situations, but invalid for certain cases. The exceptions are often so inconspicuous that they can be overlooked even by experienced engineers. To deal with these is an art which requires exceptions to the rules.

An art is undoubtly abstract. Therefore, instead of presenting principles, this article will use some specific examples to demonstrate the ideas. For instance, what can go wrong by (1) selecting the support types based on vertical thermal expansion displacement, (2) by making the equipment nozzle take no direct weight load, (3) by setting the spring to balance the weight at hot condition or (4) by installing anchor liberally at expansion joint installations.

With the advance of the computer technique, almost all the calculations today are done by computers. However, the old maxim "garbage in garbage out" is still true. In order to provide better data for the computer, a couple of practices have been evolved. First, all the support members and attachments are designed to be super stiff. Secondly, friction forces are greatly reduced by using teflon sliding plates or ball jointed struts. With these painstaking arrangements, analysts can now boast of the validity of their analyses. However, few have realized that the brute force approach has thrown away two of the major ingredients that have helped preserve the structural integrity of the design. These two ingredients are flexibility and friction, and they should be once again put to work to our advantage.

SUPPORT TYPES

The types of supports are normally selected based on the vertical thermal displacements expected at the support locations. Rigid supports are used at places where the expected thermal displacement is very small, variable springs are used for medium displacements, and constant effort supports are used when displacements are great. The practice is very logical, but problems arise occasionally. Oddly enough, the use of springs and constant supports create more problems than the use of rigid supports. Although it is true that a rigid support should not be used even at a place having a small expected thermal displacement, the mis-application of rigid supports will be detected s oon as an analysis is performed. On the other hand, the analysis on a spring or constant effort supported system cannot readily tell the mis-application of th springs.

Figure 1 shows one situation that might end up with a problem. A free thermal expansion analysis shows a vertical displacement of 11 inches (280 mm) at the middle of the span due to arching effect. The displacements at other support points are all greater than 3 inches (76 mm). By using the free thermal displacement as a guidance, constant effort supp rts will be used for the entire system as shown in Figure

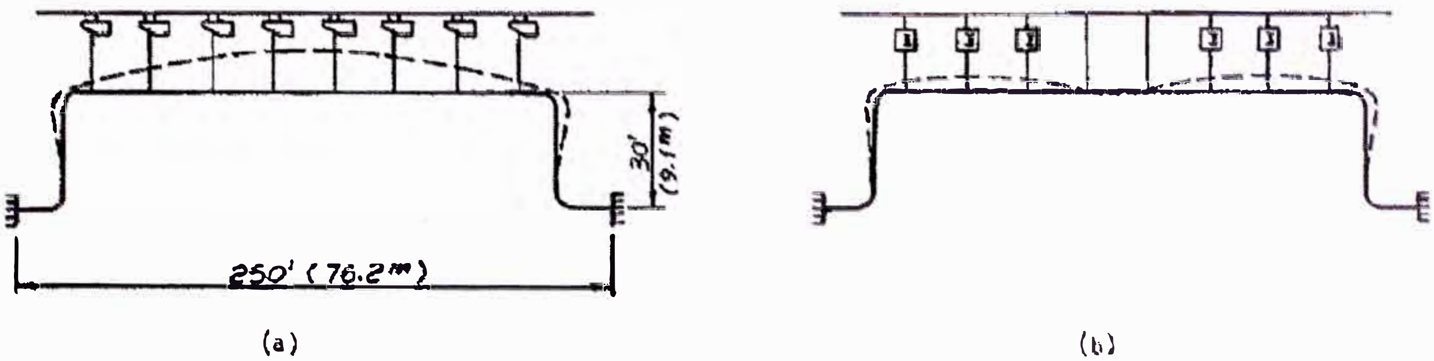


Fig. 1 Free Thermal Displacement

a). Some snubbers might also be added if the tem is to be designed for earthquake. Aside from hardware cost, the arrangement appears to have no blems. The computer analysis shows perfect results, the installation will have no problem either. The blem nevertheless occurs when the system is ready operation and the travel stops are removed. It ldn't have come at a worse time, but that is the ure of most the problems.

s system may collapse if the actual pipe, insula- n, and attachment weight is considerably heavier n the theoretical or assumed weight used in the igh. The system may weigh as much as 15 percent e than the design capacity of the supports and this make the field adjustments almost impossible. One argue that the weight should have been estimated e conservatively, but the point is that the system igned is uanble to absorb the uncertainty due to ufacturing tolerance. The system can also be under- ght making the field adjustment equally impossible. n with a properly adjusted system, because of the ction associated with the linkages a lot of banging be expected during the start-up and shut-down. movement tends to be stuck for a while then an ermittent sudden release.

ure 1 (b) shows a better design by placing rigid ports at the middle spans where the free thermal placements are the greatest. This system is much e capable for absorbing the weigh variation dis- arded by the computer. It also costs a lot s than the one shonw in 1 (a).

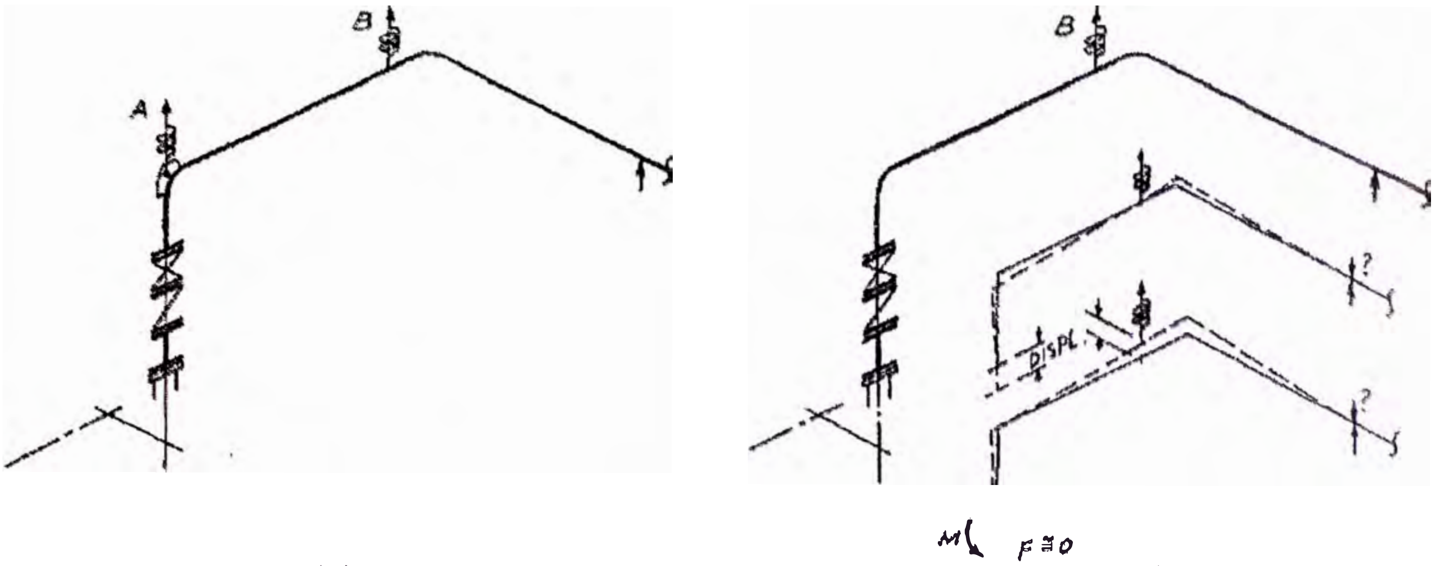
BALANCE

a high temperature system, in order to minimize the ep, the spring is set in such a way that the spring ce and the system weight will balance out each other er the hot operating condition. It is important t the sustained stress be reduced to minimum at ep temperature. For a low temperature piping re little creep is expected, it is still a good a to do the same so the unbalanced load is mini- ed under the operating condition. Hot balance such a good practice that is considered as one of the basic principles of piping engineering.

By adopting the hot balance approach, the springs have to be locked in place during installation. The locks or stops are removed when the system is ready for operation. The problem is that in many cases severe twisting and jerking occur when the stops are removed. This is somewhat expected because the spring forces at cold condition are different from those required for balancing the weight. This calculated preloading is alright if everything is as ideal as calculated. As mentioned previously the pipe weight, insulation weight, clamp weight and so forth can vary considerably from the theorectical values. Therefore the actual loading applied to the system can be quite different from the one calculated.

The deviations of the weight and the analytical model are so difficult to predict that the hot balance approach intended to minimize th hot load is in fact applying unpredictable loads on both cold and hot conditions. Alignment problems have frequently occurred on large rotating equipment. The theoretical minimum hot load is actually only a paper promise. It hits the target some of the times and is off the target at other times. This kind of incertainty is simply too much of a risk to be taken on an expensive delicate machine which is often the heart of the en- tire plant. Therefore, a more reliable approach is needed. Contrary to the common practice, the reliable approach is the cold balance approach.

In the cold balance approach, the fit up of the piping to a compressor or turbine is normally done with springs unlocked. The construction engineer will then try to adjust the spring load to bring the pipe con- nection to the equipment nozzle with a minimum help of outside force. In this way, it is sure that the piping load at cold condition is almost zero, although some load is expected under the operating condition. However, this hot load caused by spring force variation is highly predictable. The cold balance has become more and more popular lately. Designers who fail to understand the situation will make the field adjustment very difficult and will also create unnecessary argu- ment with the construction engineers. The system



(a) (b)

Fig. 2 Springs Sized By Releasing the Anchor

designed intentionally to balance the weight at cold condition will make the field adjustment much easier. With this understanding, the designer can simply instruct the field to set the spring at calculated hot load instead of the shifted cold load.

ZERO WEIGHT LOAD ON NOZZLE

It is a common practice to adequately support a piping system that no weight is imposed on rotating equipment. This can be done fairly easily by placing proper supports at proper locations. The only problem associated with this practice is the blind dependence on the computers.

Some piping stress computer programs used in the industry today have an automatic spring selection capability. They also have the option of releasing the vertical translational constraint at certain anchors during the spring selection process. This option will force the springs to carry all the weight leaving very little direct weight load on an equipment nozzle. This option is useful if it is applied correctly. For instance, many designers do not recognize that the scheme reduces only the direct weight load but not necessarily the weight moment. To have the scheme do the job right, springs have to be located at suitable locations. Otherwise, the springs selected by this anchor release option can make the system worse than the ones selected without the anchor release option.

Figure 2 shows a typical pump discharge piping. In 2 (a), since there is a spring directly over the valve assembly, the spring selection process with the anchor release option will force the springs to carry the entire weight leaving very little load to the pump nozzle. If the anchor release option is not used, then most likely spring A and pump nozzle will each carry about one half of the assembly weight. This of course happens only when the springs are selected by computer program

The situation will be different if the spring A is not available as shown in Figure 2 (b). In this ase, if the spring is selected with the anchor release option, the spring B will be forced to pick up the entire weight including the whole valve assembly. This will leave very little vertical force on the nozzle but will create a huge bending moment on the nozzle. Unfortunately, this huge moment may escape the attention of the analyst in some cases. Some computer programs, in an attempt to speed up the process, give the weight load case results taken from the ones obtained with anchor released. By doing so, the high moment at the nozzle will not show up in the output report. The only clue to this problem is the significant vertical displacement shown at the anchor point. This vertical displacement is not very obvious and is often overlooked or ignored. A properly designed computer program will apply the spring force selected and the anchor fixed to recalculate the weight result. With this type of proper analysis, the high moment will appear at the nozzle together with an upward displacement at the spring location. With systems as shown in Figure 2 (b), it will be more favorable to select the spring with the anchor fixed. In this way the anchor will absorb some vertical weight force but not the huge bending moment.

EXPANSION JOINT ANCHOR

One of the most important requirements in designing bellow expansion joint system is to install sufficient anchors for resisting pressure end forces. Figure 3 (a) shows the potential pipe movement when no proper anchor is installed. Figure 3 (b) represents the system stabilized by the anchor. The anchor normally needs to be designed to absorb only the vectorial sum of the two end forces. However, if the sy tem is expected to experience flow surges, the unequality of the two end forces at any time instant also needs to be considered. It is also possible that a valve is located at one side of the anchor as shown in Figure 3 (c). In this case the anchor has to be designed also for the condition when the valve is closed. If this valve shut-off condition is not designed for, the anchor can fail due to inadequate design, especially when the bend angle is small.

re are also cases when anchors should not be used. ure 4 shows a tied expansion joint which is used absorb the lateral differential expansion. By coming with Figure 3 arrangement, it is tempting to an anchor at the base support to resist the bel- end force. This anchor appears so natural that blems are often overlooked even by an experienced cker. The problem of the anchor can be explained m the start up sequence. When the pipe is heated both B and C ends expand into the bellow leaving ck at the tie-rods. As soon as the tie-rods get se, the pressure end force pushing the turbine is balanced. This pressure force normally is sufficnt to push the turbine off alignment causing severe rational problems. In a correct installation, this hor is not used. The pressure will push the base port outward ensuring a balancing force on the ties to cancel the pressure end force acting on the bine.

EE HINGE SYSTEM

metry and balance are normally considered two major nciples in a good design. However, there are asions when symmetry can also mean handicap. The ee hinge systme frequently used in solving plane ansion problem is one of the examples.

Figure 5 shows a three hinge system to be installed in large diameter piping connected between two major pieces of equipment. Figure 5 (a) is the perfect symmetric layout favored by many designers, including experienced ones. The only problem with this layout is that the three hinges are lined up in a perfect straight line. For the hinge 2 to be active it has to move when the system is heated up. However, this is almost impossible due to the perfect symmetry. For instance, if a line is drawn between hinges 1 and 3 to divide the space into two half spaces I and II, it is clear that any given point x in half space I there is a corresponding symmetric point xx in half space II. In other words, if the hinge 2 can move to x, it can certainly move to xx too. Since it cannot be at two different locations at the same time, the hinge will be simply stuck without moving anywhere. This is an example of pure symmetric case. In reality certain unsymmetrical effect will be built-in in the system to allow the hinge to move.

Figure 5 (b) shows the movement of hinge 2 which is located slightly off the symmetric line due to construction deviation. The hinge 2 in this case will move toward the half space II, but the magnitude of the movement can be unexpectedly high. For instance, with the dimension and the temperature shown, the calculated

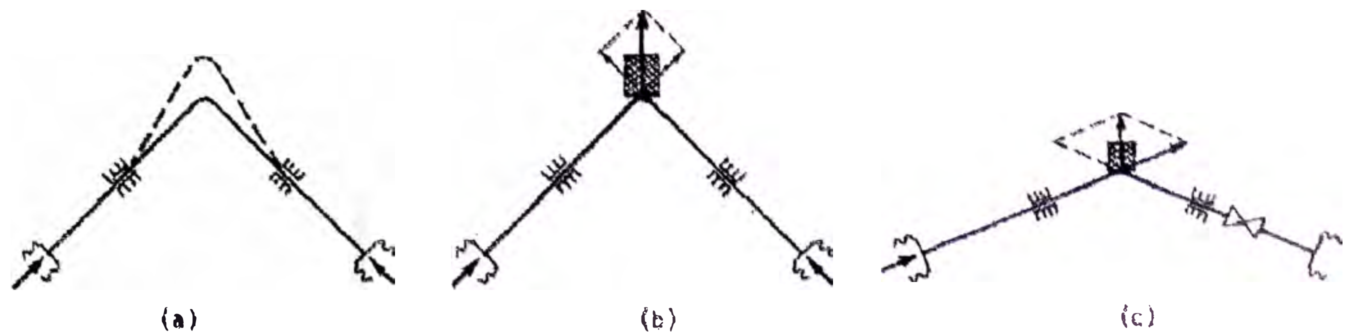


Fig. 3 Adequate Anchor Is Essential

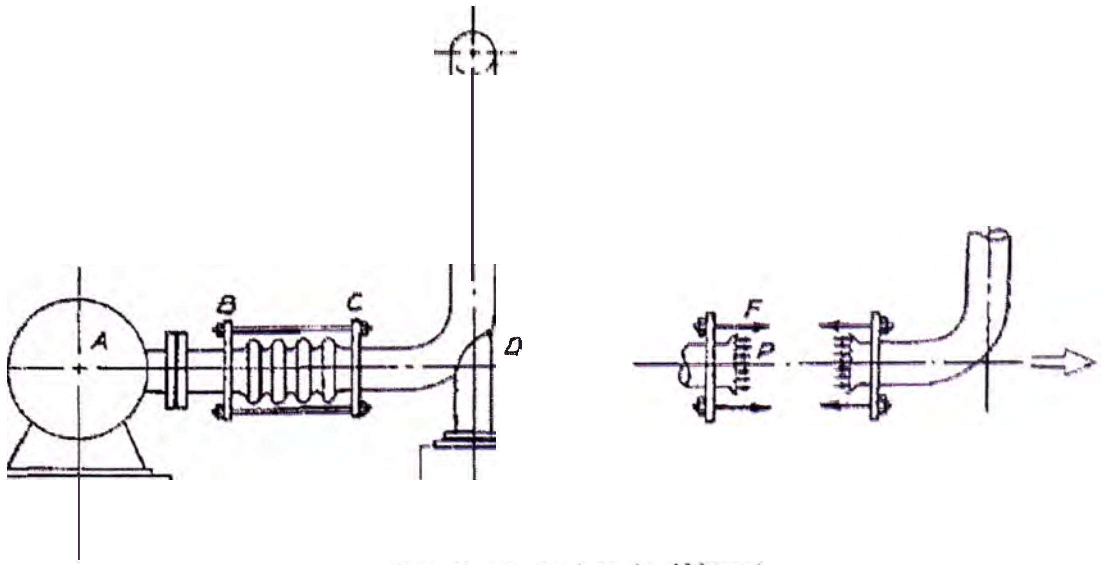


Fig. 4 No Anchor is Allowed

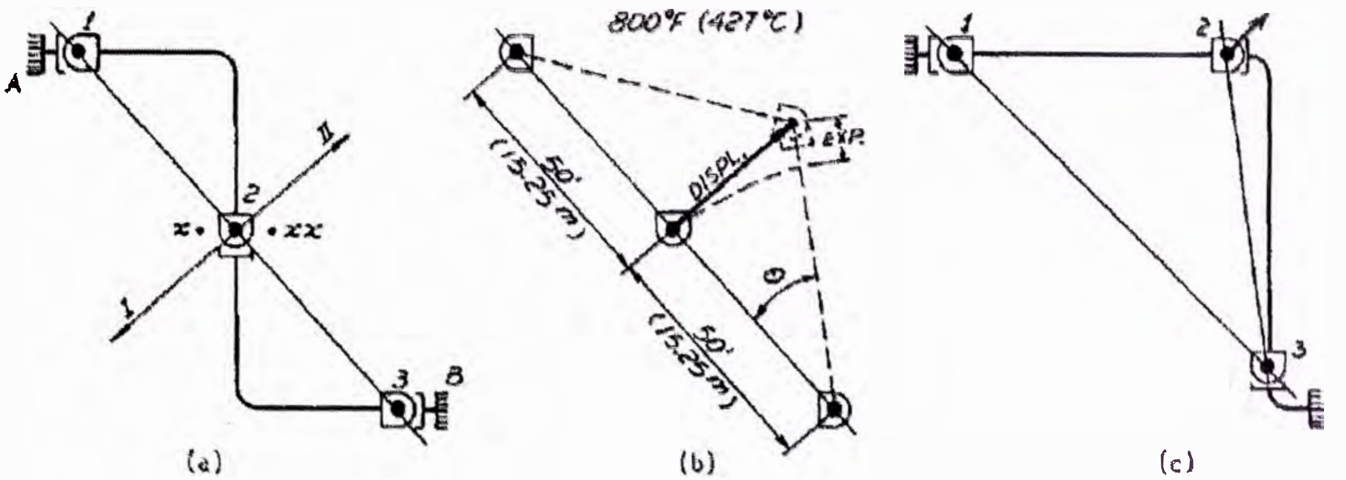


Fig. 5 Symmetry Can Also Mean Problems

vement of hinge 2 is 65 inches (1651 mm), and the gle of hinge rotation is 12 degrees. This movement too much to be accommodated by support system. The iction on the support will also have very great fect on the equipment loading.

e system can be greatly improved by locating the nge 2 away from the line connecting the end hinges and 3 as shown in Figure 5 (c). With this alter- tive layout, the expected hinge 2 movement is re- ced to about 5 inches (127 mm) with a hinge rota- on of only about one degree. This order of magni- de is well within the normal support system capacity.

ICTION RESTRAINT

systems which need to be designed for shock load- gs the common approach is to install snubbers, ther hydraulic or mechanical, at points where rigid straints are not permitted due to thermal expansion quirement. It works fine except there are also fficulties. These snubbers are not only expensive t also require constant maintenance. The snubber so has a built-in play that allows the restraint int to move a certain amount before being stopped. is slack makes the snubber a poor restraint for all amplitude, steady state vibrations. A fric- onal restraint may be more suitable for some cases.

gure 6 shows a horizontal loop system whose vertical tion can be easily restrained with rigid supports. e horizontal motion, however, is somewhat complicat- . Each leg of the loop needs an intermediate hori- ntal restraint to resist the earthquake load. How- er, because of the thermal expansion, a rigid hori- ntal restraint will create too much thermal expan- on stress. In this case the straight forward ap- oach is to install a snubber. The question here is there is an alternative approach. The main reason e loop needs the horizontal restraints is because e unrestrained system will shake in the neighborhood the peak response spectra. Once the horizental straints are installed, the natural frequency will ift upward to a more favorable spectra range.

Therefore, it is interesting to note that the horizon- tal restraint force required is only about 500 pounds (2224N). This magnitude of force is normally tolerable to an 8 inch (219 mm outside diameter) pipe. By using frictional sway braces adjusted at 500 pounds force, the braces will act as rigid stops during earthquake event, while putting limited restraint force against thermal expansion movement. Table 1 shows the stresses for different support scheme used. The table is constructed by assuming that the snubbers impose no resistance to thermal expansion. In reality because of the tight seal requirement, the resistance imposed by a snubber can be significant.

Table 1, Pipe Stress Generated by Different Support Schemes

Restraint	Pipe Stress(psi), 1 psi=6.789 KPA	
	Thermal Expansion	Earthquake
Without Restraint	8950	8030
Rigid Restraints	38390	1380
Snubbers	8950	1380
Frictional brace	14400	1380

The above discussion demonstrates the use of frictional brace to stop a dynamic motion. The frictional restraints can also be used to absorb the dynamic motion. For energy absorption, the support point has to be allowed to move a small amount. The small movement coupled with a friction force can effectively absorb the vibration energy thus increases the damping of the system.

CONCLUSION

Piping support systems are generally designed by two major rules. The support locations are determined by the guidance of the maximum allowable spans, and the support types are selected based on the expected vertil thermal displacements. There are also rules and ıctices adopted to facilitate the design and to ›id common errors. However, as demonstrated in the ›ve discussions, there are always exception to each the rules. These exceptions, if not handled pro-ly, can cause difficulties in installation and ›ate problems during operation.

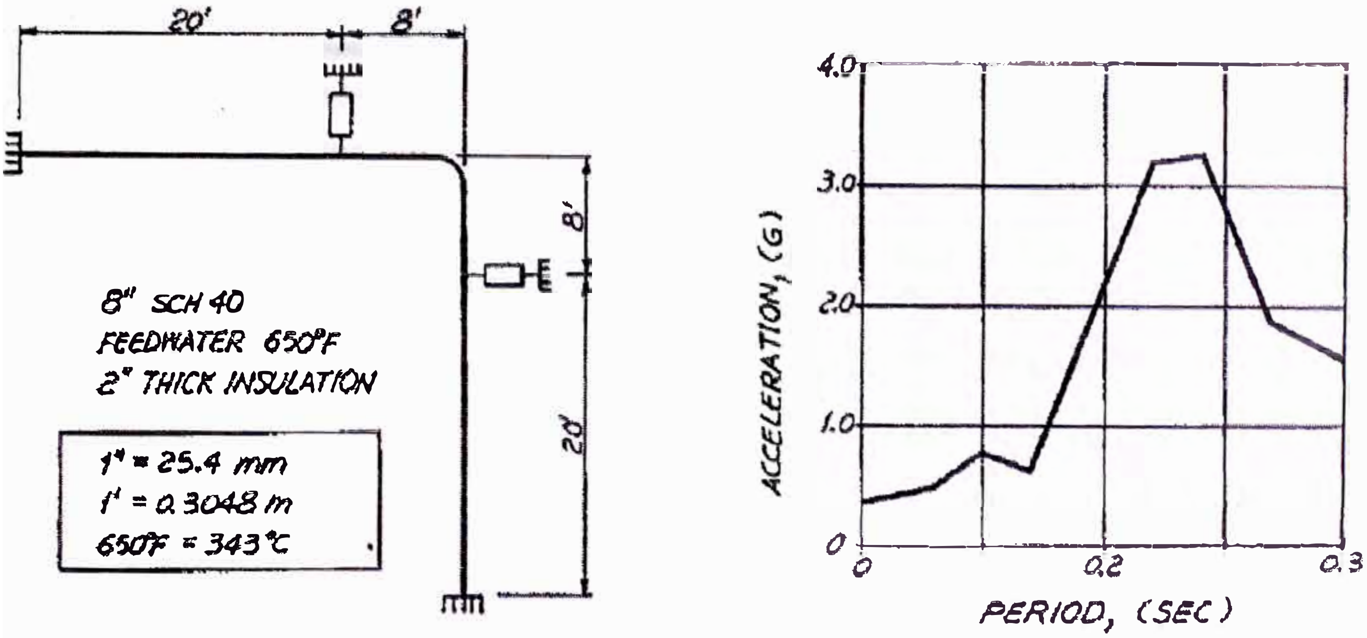


Fig. 6 Friction Restraints at Work

COLD SPRING OF RESTRAINED PIPING SYSTEM

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ABSTRACT

Power piping is often installed with cold spring to control the initial hot reaction and to protect the connected equipment. However, cold springing of a restrained or a branched system is a very sophisticated procedure which can lead to an unpredictable result if it is not done properly. This paper discusses the procedure and the problem associated with the cold spring process. It also presents the method adopted by computer programs in analyzing the cold spring affect and in preparing the cold spring data.

INTRODUCTION

Cold spring, prespring, and cold pull are all referring to the process which stresses the piping at the installed or cold condition in order to reduce the stress at the operating or hot condition. The process involves laying out the piping somewhat shorter than the installing space. This creates a gap at the final weld location when the system is erected. The system is then pulled or pushed according to a predetermined procedure to close the gap and to finish the final weld. The gap is sized depending on the cold spring factor desired. A 100 percent cold sprung system will have the gap size equal to the amount of the system expansion minus the differential anchor movements. By the same token, in a 50 percent cold sprung system the gap size is set to one half of the system expansion minus the differential anchor movements. A 100 percent cold sprung system, if installed properly, will have the expansion stress reduced to zero when the system reaches the operating temperature. It will be free of any thermal expansion stress under the hot operating condition.

Cold spring is often applied to a piping system to, 1) reduce the hot stress to mitigate the reep damage, 2) reduce the initial hot reaction force on connecting equipment, and 3) control the movement space. However, at the creep range the stress will be relaxated to the relaxation limit even if the pipe is not cold sprung. The general belief is that the additional creep damage caused by the initial thermal expansion stress is insignificant if the total expansion stress range is checked within the allowable limit. The real gain of the cold spring has become the reduction of the hot reaction. The control of the movement space is secondary.

The general philosophy and the Code rules of the cold spring have been discussed by many writers [1, 2, 3, 4]. The detailed cold spring procedure has also been discussed fully [5, 6, 7, 8] in one of the special sessions given in the 1981 ASME Pressure Vessel and Piping Conference. This paper will focus on the discussion of the systems with intermediate restraints. Since the intermediate restraints are used for different purposes, each different system may have to have a different approach in cold springing.

LOCALIZED COLD SPRING

Cold spring is used mostly to reduce the h t reaction. The Code [9] stipulates that the hot reaction can be calculated by Equation (1).

$$R_h = \left(1 - \frac{2}{3} C \right) \frac{E_h}{E_c} R \qquad (1)$$

Where, R_h = maximum reaction estimated to occur in the hot condition

C = cold spring factor varying from zero for no cold spring to 1.00 for 100% cold spring.

E_h = modulus of elasticity in the hot condition.

E_c = modulus of elasticity in the cold condition.

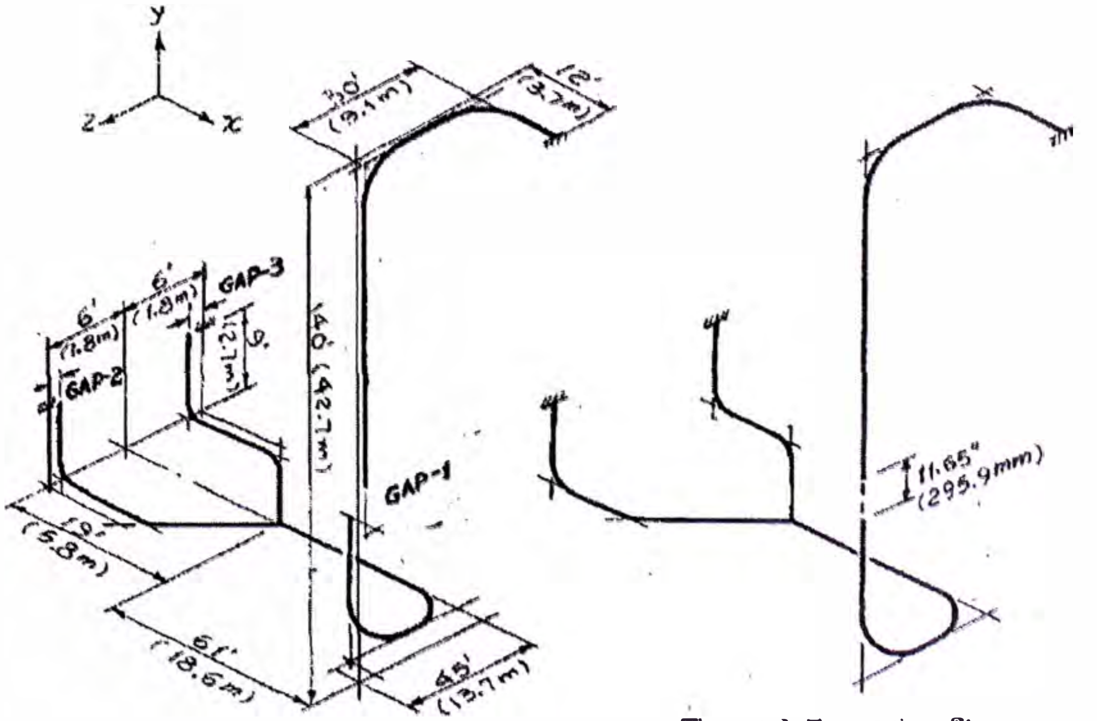
R = maximum reaction calculated for full expansion range based on cold modulus of elasticity and without considering the cold spring.

Equation (1) is applicable only when the entire ,tem is cold sprung uniformly in all the directions. achieve this uniformity, every leg of the piping to be fabricated with cut short. Each branch of system has to be erected with a cold spring gap shown in Figure 1-(a). Although in most turbine nections the GAP-2 and GAP-3 are not required, theory each branch has to have a cold spring gap the system can be sprung uniformly.

As demonstrated in Figure 1-(a), in order to ieve the uniform cold spring the entire system is ject to the so called cold spring engineering. All drawings have to be dual dimensioned with design or spatial coordinates along with the cut rts or erection coordinates. The expected cold ing movements also have to be specified in the same drawing or on another set of drawings. These all add up to the complexity of the design and the cost of the plant. To simplify the procedure, a less elegant approach of springing only one certain direction may be adopted. This unidirectional local cold spring often achieves the same effectiveness as the uniform cold spring, but at a much reduced cost.

In the system shown in Figure 1-(a), the spatial dimensions between the two fixed end points are : X=92'(28m), Y=131'(40m), and Z=15' (4.6m). From the basic beam theory it can be easily shown that at a given displacement or expansion, the displacement stress produced in the pipe is roughly inversely proportional to the square of the length. Furthermore, the displacement imposed on a given leg is proportional to the length of the lateral leg. By combining the above, it is clear that the stress produced by each leg is proportional to the cube of the length of the leg. In the piping system shown, although the Y-leg is only about 42 percent longer than the X-leg, the stress created by the Y-leg is about three times of the stress created by the X-leg. Therefore by 100 percent cold springing only the Y-direction as shown in Figure 1-(b), the expected hot stress will be reduced to about one third of the design stress range.

PIPE : 24" Sch-120 (610mm O.D., 46mm Thick)
ASTM A213, T11 (1-1/4 Cr - 1/2 Mo)
1000° F (538° C)



Thermal Expansion Stress:
No Cold Spring (hot): 9715
Local Cold Spring (hot): 2016
Local Cold Spring (cold): 9236

(a) Uniform Cold Spring (b) Local Cold Spring

Figure 1, Cold Spring Gaps

actual calculation shows that the stress will be ...ed to about one fifth of the design stress range. ... is approaching the expected reduction from the ...rm cold spring in view of the fact that the uni... cold spring is much more difficult to accom-

...y comparing both Figures 1-(a) and 1-(b), it is ...ent that the dual dimensional drawing is not ...d for the unidirectional local cold spring. The ...ary layout drawing can be used with little modi...on. The only additional dimension required is ...ap location and the gap size. It should be noted the gap location shown may not be the best loca... available. It is only used to demonstrate the ...licity of the method.

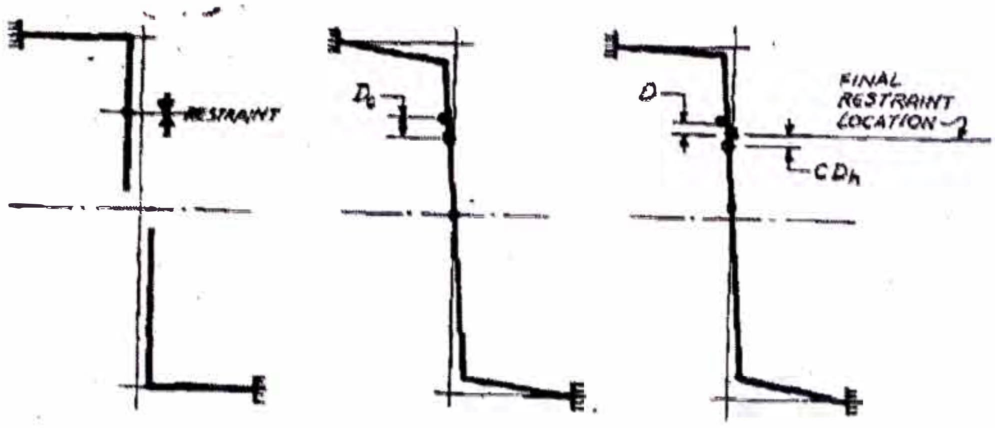
...D SPRING PRACTICE

...old springing is beneficial in reducing the hot ...ion and in assisting a system to reach the re... stage quicker. When a system reaches the ...ed stage quicker, it mitigates the creep damage ...d by the hot stress during the initial operation ...d. Though with all these benefits, the practice ...ld spring varies from industry to industry. In ...ossil power industry, due to the turbine manu...rers' insistence and the industry's long time ...tion, almost all the main steam and reheat ...n pipings are cold sprung. This is mainly be... these lines are always operating at the creep ... and the cold spring has been recognized to be ... to reduce the creep damage. Also because the ...spring is a rather expensive procedure, once ...ecision is made to spring it is generally set to ...ve the maximum benefit from it. That is if the ...is to be cold sprung, it is almost always done ...100 percent. Other lines are very seldom cold ...ng due to the low operating temperature. Cold ...g can be used at low temperature lines in re...g the hot reaction, but the benefit is not as ... as in high temperature lines. The hot allowable ...ion and the cold allowable reaction normally ... very little. This makes the shifting of the re...n from the hot condition to the cold condition ...ery attractive. In the nuclear power industry ...iping is not cold sprung again because of its ...r low operating temperature.
...ngineers in the petrochemical industry are nor... not very keen in cold springing, although many ...ms in the process plants operate in the creep .... This is partly due to practicality and partly ...o the opinions of their pioneers [1, 2, 3]. The ...g in a process plant is generally more complex ... the piping in a power plant. Also the engineer...nd construction schedule is generally very .... This combination makes the cold spring very ...actical. However, that does not mean that the ...ess industry does not want to take the advantage ...ld springing. It is just a matter of the cost ...tiveness. The cost of the additional engineering ...he extended construction schedule outweighs the ...ed reduction in creep damage at the initial ope...g period. Cold spring is occasionally used to reduce the equipment load in a process plant, but that is only limited to localized springing. The cold spring factor in this case seldom exceeds 50 percent. It should also be noted that normally special approval is required to perform a cold spring in a process piping. There are a lot of places where cold spring is not allowed. These are at the areas where cold spring is most logically needed. For instance, one of the most difficult piping to design is the compressor piping which has to meet the very low allowable nozzle load imposed by the manufacturer. If cold spring can be applied judiciously, the load would be easier to meet. But cold springing on those pipings is generally not allowed. One of the reasons which prohibits the cold spring is the low operating temperature involved in those pipings. The theoretical cold spring gap and the springing movement involved are all very small. Since it is very difficult to measure and control these small displacements up in the air in the field, the effectiveness of the cold spring is unpredictable. It might even produce a load which is damagingly high to the equipment. In any case if a cold spring is desired, then a special procedure has to be invoked to ensure the intended result.

ANALYTICAL PROCEDURE

With the advantage of the computer technology nowaday almost all the pipe stress calculations are done by digital computers. Therefore, it is important to see how a computer program is implemented to analyze the cold spring effect. In a modern pipe stress computer program the cold spring is analyzed by the combination of the cold spring gap element and the support displacement. Of course, if the cold spring is uniform, the system can be analyzed by adjusting the expansion rate to match the cold spring factor desired. However, by adjusting only the expansion rate will not be enough to produce the data needed for cold springing a system with intermediate restraints.

The cold spring gap element is used to pull both ends of the gap together. If the gap element is used at the operating temperature, the analysis simulates the hot condition. On the other hand if the gap element is used with the ambient temperature, then the analysis simulates the theoretical cold springing process giving the pipe forces, displacements, and stresses of the system after the pull. However, if the system has a rigid intermediate restraint as shown in Figure 2, the analysis is somewhat more complicated. Since the computer program cannot automatically compensate for the restraint movement adjusted during the cold springing, the analysis actually considers the restraint as fixed before the cold springing. This of course does not represent the actual case when the restraint is continuously adjusted during the springing. Therefore, a valid analysis will need the input of the restraint movement in addition to the cold spring gap. The movement of the restraint can be calculated as in Equation (2).



(a) Erected Position (b) Gap Closed (c) Restraint Installed

Figure 2, Restraints Installed after Cold Spring

$$D = D_c + CD_h \quad (2)$$

here, D = restraint displacement adjusted during the cold springing.
D_c = cold spring displacement without including the restraint.
C = cold spring factor
D_h = hot displacement without considering the gap and the restraint.

The displacements given in Equation (2) are all reference to the erected position just before the ringing. Equation (2) can be obtained by performing o analyses. One for D_c and the other for D_h. It n also be achieved by a single combined analysis ing an expansion rate equals to the actual expanon rate modified with the cold spring factor, comned with the cold spring gap. The final stress alysis is done with three different computer runs. e Code stress compliance check is performed thout considering the cold spring. The Code does t give the cold spring any credit in reducing the : stress. The hot reaction is calculated by using 3 of the design cold spring gap plus 2/3 of the straint displacement as calculated in Equation (2). e Code only gives 2/3 of the credit in calculating : hot reaction. Finally, the cold reaction is callated by using the ambient temperature plus the ll cold spring gap plus the full restraint displaceent. The cold reaction caused by the self springing e to the stress relaxation also needs to be checked ing the formula given in the Code [9].

ESTRAINT INSTALLATION

Restraints are used for: 1) supporting the weight, resisting occasional loads such as earthquake and eam hammer, and 3) controlling the pipe force to otect sensitive equipment. In each application a ecific way of cold spring procedure is normally called for.

In a cold sprung system, the weight is normally supported by spring and constant supports to reduce the expansion stress. However, at places where the expected thermal displacement or the expected restraint force is small, the rigid supports are used to reduce the plant cost and also to help stabilize the system. Rigid supports also serve to resist the occasional load. Since rigid supports are normally those of adjustable type, they are used to support the weight as usual during the cold springing. The support points are adjusted constantly throughout the cold springing process. The final location are set to the values as calculated in Equation (2).

In addition to supporting the pipe weight, restraints are often required in resisting occasional loads. Restraints which are designed solely for the occasional load can be installed after the cold springing is completed. For instance, in a uniform 100 percent cold sprung system the restraints can be installed when the system is hot at the operating temperature. Since the stresses and the restraint forces are zero at hot condition, the restraints can be installed very easily without any springing. The force and stress at cold condition can be calculated either by using a negative expansion rate (contraction rate) or by simply reversing the quantities calculated with the operating temperature without cold spring gap. In either case the restraints have to be included with no displacements.

For systems which are designed for non-uniform cold spring or for less than 100 percent cold spring the restraints are normally installed right after the cold springing at the cold condition. The restraints are then adjusted to move an amount which is given in Equation (2). The support adjustment displacement given by Equation (2) is to be done in reference to the erected position before springing. If the reference is to be taken based on the cold sprung position before any restraint is installed, then the adjustment displacement CD_h should be used. It ɔuld be noted, however, that once the system is ıled by any one of the restraints the entire system sition will be shifted from the original cold sprung sition. The reference points will be changed due to is shifting. It appears that it is easier to use the ected position as the reference. Because the erect- position is either the same as or in parallel to the sign spatial position.

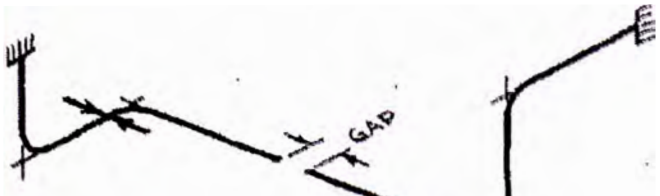
Another category of restraints are those used in the protection of the sensitive equipment. Most pipe ress engineers know that the most difficult part of be stress analysis is to meet the allowable loads ven by the vendors of the rotating and other types sensitive equipment. To meet the allowable, an genious layout coupled with strategically located straints are generally called for as shown in gure 3. Occasionally a localized cold spring is so applied to split the expansion force into, cold d hot, two parts. In this case, the restraints have be installed before the cold springing. Otherwise, e cold load due to cold spring would be too much most of the systems. This is due to the fact that thout the restraints the load acting on the machine n be as high as one order of magnitude over the ad of the restraint protected system. The cold ring factor used under this type of application is rmally set at 50 percent. The maximum value ed is 75 percent. With 75 percent cold spring, the t and cold load can be calculated based on the code le as in Equation (3).

$$R_h = [1.0 - 0.75\,(2/3)]\,R = 0.5\,R$$

$$R_c = 0.75\,R$$

Since most of the vendors will allow 50 percent ɔre load when the machine is idle, the above hot d cold loads have the same equal significance. The ld spring in effect reduces the piping load by 50 rcent in relation to the equipment allowable.

The calculation in this situation is very straight ward. The cold load and stress are calculated by cluding both the gap and the restraint, at ambient pe temperature. The theoretical hot load and stress e calculated again by including both the gap and restraint, but at the design pipe temperature. In is application, since the support is stationary ring the cold springing, no support displacement eds to be included. As the Code only allows 2/3 the cold spring credit, the gap should be reduced 2/3 before being used in the hot load calculation.



gure 3, Restraints Installed before Cold Spring

CONCLUSIONS

Although the Code [9] does not allow any cold spring credit in evaluating the thermal expansion stress, cold spring of a piping system has a definite benefit in mitigating the creep damage. Cold spring is a sophisticated process requiring well established design and installation procedure which can increase the plant cost and delay the construction schedule. Engineers should also be aware the fact that different industries have different cold spring practices. If a given industry is not prepared to do the cold spring for economic or other reasons, then it should be avoided.

With the analytical tools available today, the cold spring effect can be analyzed as easily on nonuniform cold spring as on uniform cold spring. In view of the complexity associated with the uniform cold spring, now may be the time to start thinking about the non-uniform springing. A localized unidirectional cold spring, which is much simpler to apply, an be used to achieve the same effectiveness as the uniform cold spring in many cases.

Intermediate restraints are used in piping system to: 1) support the weight, 2) resist occasional load, and 3) protect equipment. Each application has its own cold spring procedure. The weight support is normally installed before the cold springing with the support element constantly adjusted throughout the springing process. The restraints designed solely for resisting occasional load are normally installed after the cold spring is completed. If the system is 100 percent uniformly cold sprung, then these restraints can be installed when the system is hot. This avoids the springing of the restraint element. If the system is not 100 percent uniform sprung, then restraints are installed at cold condition with a displacement applied to achieve the required springing. For those restraints used in protecting sensitive equipment, they are installed before the cold springing, and are fixed in place from the very beginning. Because of these different procedures, the analysis method is also different for each application.

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TREATMENT OF SUPPORT FRICTION IN PIPE STRESS ANALYSIS

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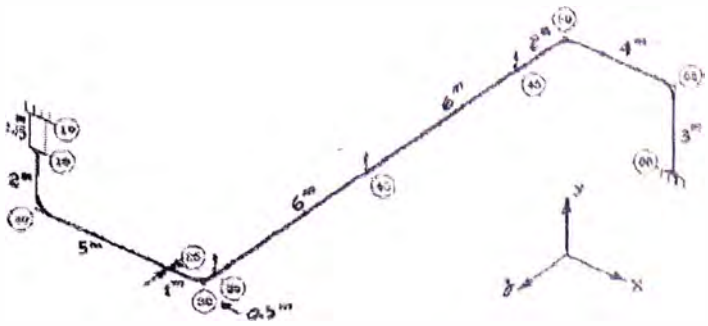
TRACT

The friction force at the pipe support has a ificant effect on the behavior of a piping system. like an analysis without including the restraint t, an analysis without including the support ion may be meaningless in some cases. The tment of support friction in the pipe stress ysis is not yet well defined in practice. This r will try to outline the procedure to be used in analysis. The paper first presents a typical lem to show the significance of the support fric- It then discusses some techniques used in the sion of the friction in a computer program. iled discussion is given in the arrangement of analysis to comply with the piping code require- ts of separating the sustained stress from the -limiting stress. Special treatment of wind and hquake loads are also discussed. The paper deals with the static aspect of the analysis.

RODUCTION

Support friction in a piping system can prevent pipe from free expansion thus creating a higher ss in the pipe and a higher load on the connecting pment. However, in certain instances the friction help stabilize the system and reduce damage. 1 in dealing with pure thermal expansion, the ion can serve as guides thus preventing a large from being transmitted to the rotating equipment. refore, there is no rule of thumb as to whether nonconservative to ignore the friction. In gene- when dealing with the dynamic load, the friction s to reduce the magnitude of both the pipe stress the equipment load. In this case, the omission e friction is conservative. However, there is no ral rule governing the static load. In this case, effect of the friction need to be investigated to ulate as closely as possible the real situation.

The effect of the friction is more important in some areas. In the analysis of the long transmission pipeline [1], it is entirely the balancing of the friction force against the potential expansion force. Without including the friction the analysis would have been meaningless. Another area of importance is the piping connected to the rotating equipment. The rotating equipment is notorious for its low allowable piping loads. Sometimes the friction at one support can completely change the acceptability of the piping system. Take the system shown in Figure 1 for instance. The restarint at 25 is installed to protect the compressor at 10. The effect of the friction at 25 is demonstrated by comparing the analysis results of the case with friction against the case without the friction. It is clear that the friction at restraint 25 is significant. By applying API STD-617 [7] criteria, only the load calculated with the restraint but without the friction is acceptable. The API criteria is evaluated separately and is not included in this paper.



Pipe Data : 323.9 mm O.D. (12" nominal), 9.5 mm tk (Std), 150°C, E=192360 MPa, exp rate = 1.53 mm/m, wt=75.5 kg/m friction factor at 25 = 0.4

Figure 1, Effect of Friction on Compressor Piping

1, Pipe Load at Discharge Nozzle Flange 15

;ion	Forces (N)			Moments (N-m)		
	Fx	Fy	Fz	Mx	My	Mz
ıint	-1145	-3727	3227	-4519	-10152	-1829
ıint nt- ıon	-1485	-3913	-289	1143	-17	-2988
aint ıon	-4735	-3397	-339	591	206	-7308

By including the friction in the analysis, the ner will appreciate the requirement of using low on type sliding plates or struts. Some might that low friction sliding plates should have been in the first place. The truth is that the friction ;y often needed for the smooth operation of the ;ne. It stabilizes the piping and dampens out the ıial vibration. Furthermore, the popular low on sliding plate adds a considerable problem in operation and maintenance of the plant.

LINEAR RESTRAINTS

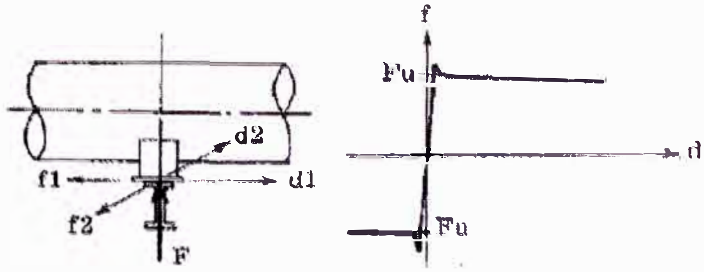
In a finite element computer program the fric- s handled by the friction element. However, to the input more efficient and the interaction direct, the support element and the friction nt are often combined into one three dimen- l interface element [2]. In piping it is called by eneral term, Non-Linear Restraint [3].

The non-linear restraint defines the restraint tion which is perpendicular to the sliding surface. ı non-linear restraint to be able to include the on, it has to have the capability to perform the ons as shown in Figure 2, and as described in llowing :

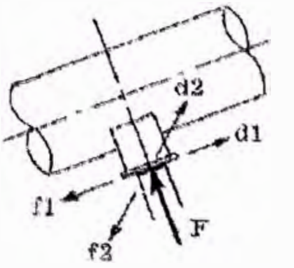
(1). Create a friction vector in the sliding sur- Normally this is defined by two local mutually ndicular vectors which are perpendicular to the aint direction. In a Y-direction restraint, the on vector is to be determined by the X- and ctors.

(2). If the potential friction force is sufficient p the pipe from moving along the restraint ce, the pipe will be stopped. The resultant on force created is less than the potential fric- orce. It is equal to the force required to elas- y stop the pipe from moving.

(3). If the potential friction force is not large h to stop the pipe from moving along the res- surface, the pipe will move. The resultant on force created is the product of the normal aint force and the coefficient of friction. The on applied to the pipe is directly opposite to ipe movement.



$f = \sqrt{f1^2 + f2^2} < Fu$, Stopped



$f = \sqrt{f1^2 + f2^2} = Fu$, Moving

$\frac{f1}{d1} = \frac{f2}{d2}$

f = Friction force
F = Normal restraint force
u = Coefficient of friction
d = Pipe displacement

Figure 2. Friction Restraints

The potential friction force is the product of the restraint normal force and the coefficient of friction. In the calculation, a small displacement is assumed to be required to develop the full potential friction. This displacement is taken as so small that its existence will not affect the result of the analysis.

Theoretically both static and sliding friction coefficients have to be used. However, even if the translational pipe motion is being stopped, the rotation and jerking of the pipe make it difficult to maintain the static coefficient. In practice only the sliding or dynamic coefficient is used.

In addition to the friction handling capability, a non-linear restraint can also handle gap, initial load, and plasticity of the restraint element.

PIPING MOVEMENTS

When a piping system expands, its movement at a given support is not likely to be in a straight line. Therefore, the friction at the support does not maintain the same direction throughout the whole expansion process. The situation is even more pronounced in a system which is restrained by limit stops. The pipe can start out in one direction, then make a sharp change after reaching the stop. In this case, the direction of the friction would also have to be adjusted constantly throughout the expansion process. This type of analysis can be done with a series of analyses at incremental steps. At each step the expansion is increased by a certain increment with the friction force balanced at the end of each step. The force and moment at each step are recorded and enveloped to ensure that the most severe result is obtained.

Although it is preferable to perform the incremental analysis to ensure that no extreme load is overlooked, the current practice is to make a one step analysis. The pipe at the support location is assumed to move in a straight line from the initial position to the final operating position. In this way the friction is applied based on the final displacement. All the intermediate displacements are ignored, because their existence is temporary in nature. However, sound engineering judgement should be exercised to see if a more elaborated analysis is justified.

COMPUTER IMPLEMENTATION

The concept of the friction element is clear, but the computer program implementation can be different from one software package to another. For the sake of explaining the implementation detail, a general discussion on the static problem solution procedure is in order. The static pipe stress problem is solved by first assembling the equilibrium equation (1).

$$[K]\ X = F \qquad (1)$$

Where, [K] = Stiffness matrix of the piping system
X = The unknown nodal displacement vector
F = The known nodal load vector

The load vector F includes weight, thermal initial load, pressure, external force, and so forth. The unknown displacement can be solved by the inversion of [K], or most likely by the decomposition of [K] as in Equation (2).

$$[K] \quad [L]\ [D]\ [L]^T \qquad (2)$$

Where, [D] = A diagonal matrix
$[L]$ and $[L]^T$ are unit triangular matrices being each the transpose of the other.

The equilibrium equation is then solved by letting

$$[L]^T\ X = Y \qquad (3)$$

$$\text{or} \quad [L]\ [D]\ Y = F \qquad (4)$$

Where Y is an intermediate solution vector which is being used as a bridge of the solution. A forward substitution is performed on Equation (4) to solve Y. This Y is then used in Equation (3) to solve the displacement X by back substitution. The decomposition step in Equation (2) takes a lot more computer time than the substitution steps in Equations (3) and (4). This makes the avoidance of the decomposition step highly desirable.

The computer program implementation of the support friction can be categorized into three groups being discussed in the following. They all use the interative approach, but each group has its strong and weak points. Some emphasize the saving of the computer time, while others are more concerned about the convergence and stability. In the long run, the idea originally intended to save computer time might end up using more computer time due to unexpected slowness in convergence. A scheme has very little practical value if it does not converge, or if it is not stable. The following are the detailed discussion on the chracteristics of each scheme.

(1). Direct Substitution of Friction Force

The most simple method is the direct substitution of the forces expected from the friction. The analysis starts out with no friction to find out the potential movement of the piping. The friction forces corresponding to these movements are then included in the load vector, F, for a new analysis. The procedure continues iteratively until the convergence is reached when no significant change occurs between two consecutive analyses.

This method is straight forward. It requires no additional decomposition of the stiffness matrix at each analysis. Therefore, it appears to have the potential of saving some computer time. The method works fine in some rather rigid systems where the friction does not affect the direction of the movement, but works very poorly for most practical piping systems which always have considerable flexibility to absorb thermal expansion. This scheme does not have the capability to stop the pipe when the stopping force required is less than the potential friction force. Instead it keeps applying the same full friction force to the system resulting in a back and forth oscillatory but no ending iterations.

This method also quickly becomes divergent when applied to the system which has considerable movements in the flexible direction. In this flexible direction, if the friction force is applied against the movement, a very large displacement will be created opposite to the original movement. The situation reverses during the next iteration with the displacements randomly getting bigger and bigger in each subsequent iteration.

(2). Fixed Stiffness Method

In this method, each frictional restraint is assigned two factitious orthogonal restraints laying on the plane perpendicular to the main supporting restraint as shown in Figure 3. The spring rate of these two restraints are taken to be the same as the initial slope given in Figure 2.

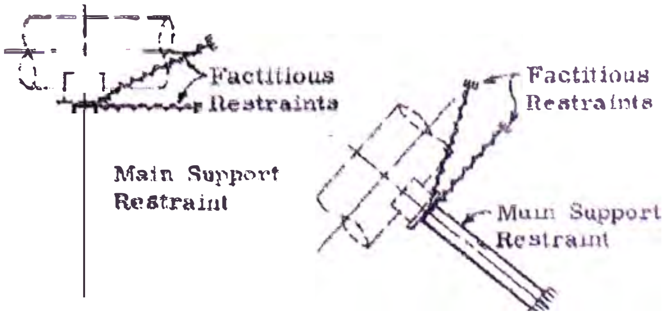


Figure 3, Factitious Restraints

When the pipe moves along the support plane, resistance forces will be generated on these two

:titious restraints. These forces are the simulation the friction effect. When the resultant force from ise two restraints is less than the potential friction ce, the pipe is being stopped by the friction. The ces generated are the actual friction forces acting the corresponding directions. If the resultant force ceeds the potential friction force, then the pipe is ving. In this case, since the friction force gener- d is greater than the potential friction force, an justment needs to be made. The method applies a verse external force back to the system to counter ance the excessive resistance force generated. The a is to make the combined effort, from the resis- ce of the restraints and the reverse external force, uivalent to the friction expected. The iteration con- ues until the combined effect at each restraint tches the friction expected.

Each iteration in this fixed stiffness method nges only the load vector, F. No redecomposition the stiffness matrix is required. Yet it has two titious stiff springs which serve to stabilize system and to readily stop the pipe from moving en it is called for. This method has the advantage the first method in preserving the decomposed ffness matrix, but has less tendency in getting into domain of divergence. It is quite popular in the ite element analysis [4]. However, the method does ve some undesirable behaviors. Again these unde- able behaviors are more pronounced in flexible stems. First, since the two factitious restraints fairly stiff in most cases, it can take a large mber of iterations to have the pipe moved to the al destination. The most disturbing part, however, when the movement reverses at a certain point in next iteration, the reverse external force derived m the previous iteration will tend to reinforce the versal. This can often lead to an instable analysis.

Variable Stiffness Method

The first two methods discussed are all based the idea of preserving the most computer time ensive martix decomposition process. However, in ing stress analysis, the use of limit stops, single- ing restraints, and other non-linear features have come common place. To account for these nonlinear tures, the revision and redecomposition of the ffness matrix has become a necessity through each iteration. Based on this premise the saving of the decomposed stiffness matrix has become less important of a factor in pipe stress analysis.

Like the fixed stiffness method, the variable stiffness method also assigns two factitious restraints at each restraint location to simulate the friction. Only the stiffness or the spring rate of these factitious restraints are not fixed. Depending on the developers, some schemes start out with some stiffness, while others start out with zero resistance. The stiffness of these factitious restraints at each iteration is estimated from the previous iteration. The stiffness matrix is then revised for these updated spring rates and for the activity changes of other non-linear restraints. It is then redecomposed for the new iteration. A more detailed discussion on this method can be found in Reference [5].

The variable stiffness method normally converges to the required accuracy much quicker than the other two methods discussed. This makes the total computing effort required by this method not much different from those of the other methods, although the matrix decomposition is performed at every iteration. The convergence in this method can also get quite slow if there are multiple locations where the pipe is being stopped by the friction in the system. The solution, however, is always stable.

EXAMPLE

All the above three methods have been in use by different pipe stress computer programs. No matter which method is adopted, there are refinements that need to be made in the programming. These include the methods of increasing the convergence rate and the schemes for avoiding an instable solution. These refinements are proprietary to the program developer and are not obvious to the users. Their effectiveness can only be measured by their actual performance. The example system shown in Figure 4 with the partial results tabulated in Table 2 can serve as a benchmark. This is a flexible off-site system with considerable movement in the flexible direction of the system. It is a typical system whose solution can easily become instable when using some of the less sophisticated iteration schemes.

Table 3 compares the analysis results o the case with friction against the case without f iction.

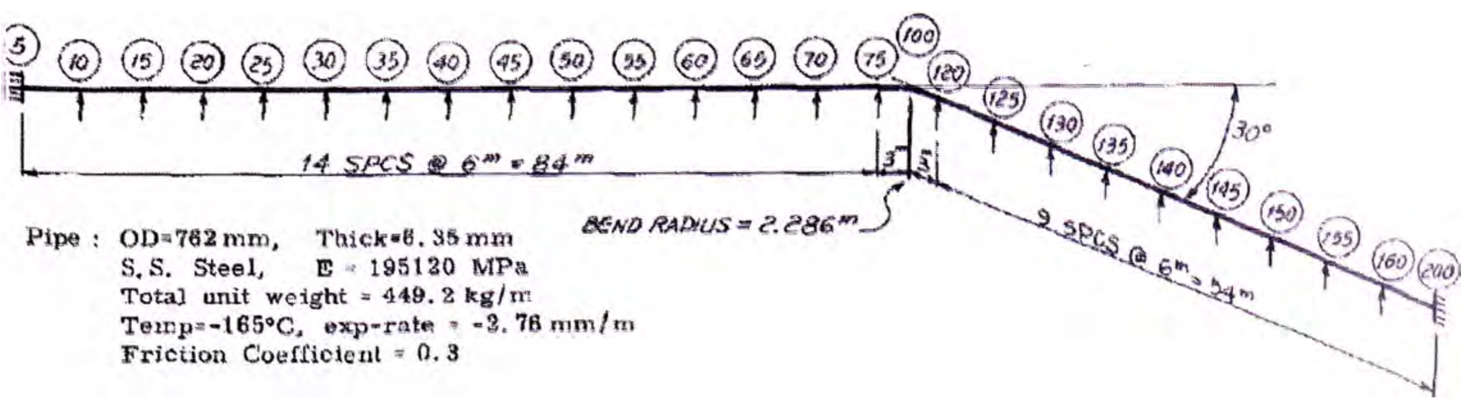


Figure 4, Example Off-site Piping System e 2, Partial Results of the Example Analysis

CASE 1 TN + MT RESULTS   LOAD= THR,WGT,BWB,FOR,BFR,CSP,PRES
****

*** ANCHOR AND SUPPORT FORCES - INCLUDING FRICTION (ACTING ON SUPPORT) ***

SUPT TYPE	DATA PT	FX	FY	FZ	MX	MY	MZ	FFX	FFY	FFZ	FMT	DX	DY	DZ	NOTES
		SUPPORT FORCE AND MOMENT: FORCES (N)			MOMENTS (N-M)			FRICTION: FORCES (N)			T(N-M)	DEFLECTION (MM)			
ANCH	5	222557	-13214	198	-304	9984	-13213	0	0	0	0	.0	-.0	.0	
NLY	10	0	-26429	0	0	0	0	-7914	0	-436	0	-16.1	-.0	-.9	
NLY	15	0	-26429	0	0	0	0	-7876	0	-885	0	-32.2	-.0	-3.6	
NLY	20	0	-26429	0	0	0	0	-7823	0	-1275	0	-48.3	-.0	-7.9	
NLY	25	0	-26429	0	0	0	0	-7785	0	-1503	0	-64.5	-.0	-12.5	
NLY	30	0	-26429	0	0	0	0	-7892	0	-1434	0	-80.6	-.0	-14.8	
NLY	35	0	-26429	0	0	0	0	-7982	0	-895	0	-96.8	-.0	-11.0	
NLY	40	0	-26429	0	0	0	0	-7916	0	310	0	-113.0	-.0	4.4	
NLY	45	0	-26430	0	0	0	0	-7578	0	2209	0	-129.2	-.0	37.7	
NLY	50	0	-26425	0	0	0	0	-6592	0	4285	0	-145.5	-.0	94.5	
NLY	55	0	-26446	0	0	0	0	-5279	0	5820	0	-161.7	-.0	178.5	
NLY	60	0	-26359	0	0	0	0	-4119	0	6686	0	-177.9	-.0	288.7	
NLY	65	0	-26721	0	0	0	0	-3359	0	7240	0	-194.2	-.0	418.5	
NLY	70	0	-25215	0	0	0	0	-2678	0	7057	0	-210.5	-.0	554.3	
NLY	75	0	-27359	0	0	0	0	-2610	0	7776	0	-226.7	-.0	675.2	
NLY	120	0	-27337	0	0	0	0	-2534	0	7795	0	-217.0	-.0	669.5	
NLY	125	0	-25218	0	0	0	0	-2178	0	7230	0	-163.8	-.0	543.6	
NLY	130	0	-26720	0	0	0	0	-1973	0	7734	0	-103.9	-.0	407.3	
NLY	135	0	-26358	0	0	0	0	-1238	0	7728	0	-49.3	-.0	288.1	
NLY	140	0	-26445	0	0	0	0	-317	0	7830	0	-7.1	-.0	174.5	
NLY	145	0	-26424	0	0	0	0	1502	0	7656	0	19.0	-.0	96.8	
NLY	150	0	-26429	0	0	0	0	4114	0	6662	0	29.0	-.0	47.0	
NLY	155	0	-26428	0	0	0	0	6258	0	4825	0	25.9	-.0	20.0	
NLY	160	0	-26429	0	0	0	0	7048	0	3635	0	14.4	-.0	7.4	
ANCH	200	-145898	-13214	-96251	-7008	-28484	11211	0	0	0	0	-.0	-.0	-.0	
NET FORCES		76669	-634174	-96053				-76651	0	96059	0				

It is obvious that without the friction the axial anchor load would have been very small. On the other hand, the friction tends to prevent the pipe from moving into the flexible lateral direction. This, in essence, serves as the guide preventing the anchor from getting the moment created by the lateral movement of the pipe. Because of the restriction on the lateral movement, the system becomes more stiff. Thus resulting in higher bending moment at the bend.

e 3, Comparison of the Example Analysis

	Anchor Load at 5		Bend Moment at 100A
	Fx (N)	My (N-m)	My (N-m)
ion	222600	10001	328480
out ion	16746	123942	124462

ERENCES (cont.)
. Building Code Requirements for Minimum Design s, ASNI A58.1, American Nat. Std. Inst., N.Y.C.
. API Standard 617, Centrifugal Compressors for ral Refinery Services, American Petroleum Inst. ington, D. C.

CODE COMPLIANCE ANALYSIS

In order to comply with the Piping Code requirements, the analysis has to separate the sustained stress from the self-limiting stress. The friction in a piping system is generally caused by the weight being pushed by thermal expansion. Weight is sustained loading but thermal expansion is self-limiting. The friction on the other hand is passive which by itself does not have the damaging potential. For the convenience of the analysis, the friction can be treated as self-limiting same as in the case of thermal expansion. To satisfy the Code requirement of separating the stresses, separate load cases for weight, thermal and occasional loads have to be performed. But without the weight the thermal expansion will hardly have any friction resistance. That is, if the straight weight or expansion is applied to the corresponding load cases, the friction force will completely disappear from the picture. Therefore, special arrangements have to be made so the friction effect can be accounted for properly.

One method to include the friction yet still be able to separate the sustained stress from the self-limiting stress is to apply the weight loading under the normal operating condition as the initial support load. With this method, the weight load under the normal operating condition is first determined at each support. This load is then used as the support initial load for the analysis of the thermal and occasional load cases. That is, if the normal operating weight load is 3000 N, and the thermal expansion load is 1000 N, then the support friction is included based on the support normal force of 3000 N. The single-acting restraint activity status will also be checked based on the premise that the support initial force is there.

The normal operating weight load is the balanced weight load under the operating condition. It is the weight load calculated by removing the inactive restraints at where the pipe is pushed off the support by the thermal expansion.

OCCASIONAL LOAD ANALYSIS

The occasional load, by piping code criteria, is to be combined with the sustained load. In theory it can be directly added to the weight load for the analysis. However, because the sustained load has its own separate requirements and the occasional load is always considered as dual directional, the weight and occasional loads are normally analyzed with separate load cases in practice.

In the occasional load analysis, the initial weight load may also be included in the calculation of the friction effect. Although some may argue that the weight initial load is already included in thermal expansion analysis and should not be included again in the occasional load analysis. But the friction due to weight is still there to resist the occasional load motion whether the pipe has gone through the expansion process or not. Nevertheless, since the friction tends to help the system in resisting the occasional load, its inclusion to the occasional load analysis requires some justification. The practice varies due to the different beliefs in the availability of the initial weight load during the occasional load condition.

The earthquake and wind load analyses are normally done with the equivalent static method given by ANSI A58.1 [6]. In this method the piping is applied with the horizontal load appropriate for the location of the piping. The vertical load is not addressed in tacit recognition of the adequacy of the normal support structure in resisting the load. The vertical load may or may not be adequately supported by the normal support structure, but it does have a significant effect on the initial weight load. During the earthquake, because of the upward acceleration existing at a certain instant, the pipe may be lifted up fully or partially from the support leaving very little weight load on the support. Therefore, the inclusion of the initial load in estimating the friction will be nonconservative. The same, in a lesser degree, is also true during the hurricane condition. Based on the above consideration, some companies require that the initial weight load, if not the friction, cannot be included in earthquake and wind load analyses. Others have allowed the initial weight load in the wind load case, but not in the earthquake load case. In any case the magnitude of the occasional loads and their method of analysis shall be clearly defined in the Design Specification.

CONCLUSION

The support friction can have a significant effect on the pipe load at the connecting equipment. It can also increase the thermal expansion stress by several folds in some cases. There are different methods which can be used to implement a computer program in handling the support friction. Some are more effective in certain cases, and some may not give a stable result under certain conditions. Owing to the inherent flexibility in a piping system, the simple direct substitution of the friction force scheme does not work well in the analysis of piping systems.

In order to comply with the ASME B&PV and ANSI B31 Piping Code requirements of separating the sustained stress from the self-limiting stress, separate load cases are performed for weight, thermal expansion, and occasional loads. The weight support load is included in the thermal expansion analysis for calculating the friction effect. However, the weight support load may or may not be included in the occasional load analysis depending on the individual design specification. While it is generally more conservative in including the friction in thermal expansion analysis, a separate expansion analysis without including the friction is also recommended to check the loading condition after the system has gone through a long period of operation with the friction effect shaken off. The inclusion of the initial weight in the occasional load analysis requires some serious consideration. The initial weight should not be included if the pipe is likely to be lifted off the support either fully or partially during the occurance of the event.

REFERENCES

1. Peng, L. C., "Stress Analysis Methos for Underground Pipe Lines," Pipe Line Industry, Apr & May, 1978
2. ANSYS User's Manual, STIF52, Swanson Analysis Systems, Inc., Houston, Pennsylvania.
3. SIMFLEX-II, Peng Engineering, Houston, Texas.
4. Shah, V. N. and Gilmore, C. B., "Dynamic Analysis of a Structure with Coulomb Friction," ASME paper 82-PVP-18, July 1982.
5. Sobieszczanski, J., "Inclusion of Support Friction Into a Computerized Solution of a Self-Compensating PipeLine," ASME paper 71-WA/PVP-1, Nov, 1971.

ADEMINSAC

ANDRRO DE ENERGIA Y MANTENIMIENTO INDUSTRIAL S.A.C

ANALISIS VIBRACIONAL, ALINEAMIENTO CON RAYOS LASER BALANCEO DINAMICO ENSAYOS NO DESTRUCTIVOS ULTRA SONIDO PARTICULAS MAGNETICAS RADIOGRAFIA HOLIDAY DETECTOR TECNICAS ANTICORROSION TINTES PENETRANTES ANALISIS DE COMBUSTION SOLDADURA EN FRIO RECTIFICADO IN-SITU REPARACIONES FABRICACIONES CAPACITACION Y VENTA DE EQUIPOS

Lima, 19 de Mayo del 2006
AD/CI – 051 – 06

Sres :

Att.:
Dpto. de Mantenimiento Predictivo

REF.: COTIZACIÓN POR EQUIPOS END PARA PARTICULAS MAGNETICAS (MT) Y ULTRASONIDO (UT).

Estimado Ingeniero Alca:

Por medio de la presente, sírvanse encontrar nuestra propuesta económica por los equipos que a continuación se describe.

ITEM	DESCRIPCION	COSTO
01	SISTEMA PORTATIL YUGO MAGNETICO MAGNAFLUX Y8 BATTERY POWERED YOKE	US $ 1720.00
02	LAMPARA DE LUZ NEGRA MAGNAFLUX ZB100F (produce mejor longitud de onda de luz ultravioleta) Hand Held, 100 Watt, 230V/60Hz/ 1 WITH A CART	US $ 1060.00
03	01-1716-72 #1 Gray MAGNAFLUX NON-FLUORESCENT DRY POWDER 12 - 1 lb. Containers	US $ 345.00
04	01-1780-72 #8A Red MAGNAFLUX NON-FLUORESCENT DRY POWDER 12 - 1 lb. Containers	US $ 346.00
05	01-1725-38 14A Aqua-Glo Wet Method Fluorescent Fluorescent Magnetic Particle Inspection Materials (Premix of 14A MPI Powder + Water Based Medium) Case of 12 Aerosol Cans &265 gr	US $ 280.00

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AHORRO DE ENERGIA Y MANTENIMIENTO INDUSTRIAL S.A.C.

ANALISIS VIBRACIONAL, ALINEAMIENTO CON RAYOS LÁSER BALANCEO DINAMICO ENSAYOS NO DESTRUCTIVOS ULTRA SONIDO PARTICULAS MAGNETICAS RADIOGRAFIA HOLIDAY DETECTOR TECNICAS ANTICORROSION TINTES PENETRANTES ANALISIS DE COMBUSTION SOLDADURA EN FRIO RECTIFICADO IN-SITU REPARACIONES FABRICACIONES CAPACITACION Y VENTA DE EQUIPOS

CONDICIONES GENERALES

- Emitir Orden de Compra a favor de *ADEMINSAC.*

  Dirección: Calle Cerro Azul N° 479 Urb. San Ignacio de Monterrrico – Santiago de Surco. Lima.

  Telf.: ++511-2750082 ó 2751588 / Fax 2751589 Cel. ++511-97518809

  Persona Contacto: Iván Díaz \ Field Service Manager
- La forma de pago será Factura a los 30 días de la entrega de los Equipos.
- Tiempo de Entrega: Antes del 15 de Junio, Con orden de compra dentro de las 12 horas de recibida la presente.
- Condiciones de entrega: Entregas parciales en Almacenes BHP Tintaya Lima.
- **Los Precios no Incluyen el 19% del I.G.V. y han sido calculados como "Precio Paquete".**
- La Garantía contra defectos de Fabricación será de 1 Año.

En esperas de sus gratas ordenes, quedamos de Usted.

Atentamente,

Ing. Alberto Reyna O.
Senior Vibration Analyst
NDT – Level III – VA – TA
ASNT Level III – UT, PT, MT
CWI - AWS N° 121052
CIP N° 34856
Gerente

William Iván Díaz D.
Field Service Manager
NDT – Level II – PT, MT, UT, VA
ASNT & ASME member

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ANDINO DE ENERGIA Y MANTENIMIENTO INDUSTRIAL S.A.C.

ANALISIS VIBRACIONAL, ALINEAMIENTO CON RAYOS LASER BALANCEO DINAMICO ENSAYOS NO DESTRUCTIVOS ULTRA SONIDO PARTICULAS MAGNETICAS RADIOGRAFIA HOLIDAY DETECTOR TECNICAS ANTICORROSION TINTES PENETRANTES ANALISIS DE COMBUSTION SOLDADURA EN FRIO RECTIFICADO IN-SITU REPARACIONES FABRICACIONES CAPACITACION Y VENTA DE EQUIPOS

YUGOS PORTATILES ACCIONADOS POR BATERIA

Y-8 Battery Powered Yoke

Yoke:	*Y-8 Battery Powered Yoke ONLY* (Part #620145)* **Must choose the appropriate voltage for the battery charger.*
Battery Charger:	*115v Charger (Part #620142)* *230v Charger (Part #620709)*
Current Draw:	*4 amps @ 6 volts DC (Yoke Only)*
Current Draw:	*0.2 amps @ 115v or 230v (Charger Only)*
Demag:	*No*
Yoke Weight:	*7 ½ lb.*
Leg Capacity:	*0" – 12" (0-30 cm) across poles*
Cord:	*12 ft. coiled*

BATTERY SPECIFICATIONS

Rechargeable Battery: 6 volt, 12-amp hour

Battery Weight: 5 ½ lb.

Under normal duty cycle, battery lasts 8 hours between charges.

PORTABLE BATTERY POWERED MINI CONTOUR PROBE®

The B-310 BDC Mini Contour Probe from Parker Research is a very portable, battery-operated instrument for Magnetic Particle Inspection, and is especially convenient where power is limited or where operator safety is a concern.

The B-310 BDC Magnetic Inspection Yoke is designed to perform Magnetic Particle inspections quickly and reliably, producing highly defined defect indications. The unit will comply with the 40 to 50 pound weight-lift test.

The exclusive strain-relief feature allows the power cord to enter from the rear or top of the unit permitting greater access to confined work areas. The overall length of the unit is only 7.25".

As with all Parker Contour Probes, the new B-310 BDC has fully adjustable legs permitting the DC field to be applied to the precise area of inspection.

The rugged body assembly is injection molded of the same durable material used in heavy duty power tools. It is shaped to fit the hand comfortably to reduce operator fatigue.

The model B-310 BDC may be ordered with a convenient 12V, foam-padded battery pack with belt loops and shoulder strap. Batteries are available in either 7.2 or 14.4 amp hour size and the pack includes a battery charger. The batteries are good for 150 to 1500 recharges depending on the amount you discharge the battery and storage conditions.

The B-310 BDC comes with a 10 foot cord and can be plugged into any 12V automotive cigarette lighter socket.

LAMPARAS DE LUZ NEGRA

BLACK LIGHTS

ZB-100F

Portable Black Light System produces longwave ultraviolet light at 365 nanometers, providing optimum fluorescence in MAGNAGLO® and ZYGLO® testing materials.

- *100 watt black light with transformer, cords, GFCI and plug*
- *Built-in fan keeps light cool to the touch.*
- *Running temperature <85°F.*
- *Typical output 5,000 µW/cm².*
- *#[illegible] portable unit includes carrying base and stand for storage of cord.*
- *#[illegible] Hand-held unit similar to #[illegible], except no carrying base provided.*

Benefits

One significant feature that separates the ZB-100F from other black lights – it has a built-in, quiet-operating fan located at the rear of the lamp housing.

The advantage of this design feature is obvious – dramatically reduced running temperature. In practical terms, the fan-cooled ZB-100F means more comfort and safety for operators plus longer bulb life.

Because black lights are often hand-held during fluorescent inspections, the heat they create is an important consideration. The typical running temperature of the ZB-100F is less than 85 degrees Fahrenheit, so it does not significantly raise the ambient temperature around the operator, nor is it a safety concern.

The ZB-100F also shares all of the features of the ZB-100, our most popular NDT black light.

- Polycarbonate hood helps keep operators' hands away from hot glass during use
- Cord-through-handle design reduces operator fatigue, improves balance and provides greater safety
- Fiberglass reinforced handle always stays cool
- Tough aluminum housing withstands rugged use
- Recessed lens design protects lens and bulb from damage

Technical Specifications for ZB-100F

Power Supply:	[illegible] (ZB-100F[illegible])
	[illegible] (ZB-100F[illegible])
	[illegible] (ZB-100F[illegible])
Current Draw:	1.0 amps
Typical Visible Light:	[illegible] 15" (38 cm)
Bulb:	100 watts
Running Temperature:	[illegible]
Weight - Hand Lamp:	3 lbs (1.4 kg)
Weight - Hand Lamp with Transformer:	12.5 lbs (5.7 kg) Control Unit
Lamp to Transformer:	10 ft. (3.? m) Cable [illegible]
Main Cable Supply:	12 ft. (3.7 m) with Ground Fault Protection

- *UV intensity meets industry and military standards @ 15" (38 cm)*

part # 400004

ZB-100F and ZB-100 Hand-Held Black Lights

- 100 watt, hand-held model includes transformer, cords, GFCI and plug.
- 115v or 230v
- Typical intensity @15" (38 cm): 4000 Microwatts/cm²
- Polycarbonate head protects operator from burns
- Fan cooled model reduces running temperature <85°F. Keeps light cool to the touch

Spare Medium Base Bulb Part # [illegible]

ZB-100F Hand-Held, Fan-Cooled Black Light

Part #:	Voltage:
600004	115v/60 hz/1ph with camera/stand
600005	115v/60 hz/1ph
600048	Camera/stand for ZB-100F
621312	230v/50/1 with camera/stand
621311	220v/50/1 with camera/stand
621100	Carrying case for ZB-100F

ZB-100 Black Light

Part #:	Voltage:
600000	115v/60 hz/1ph
600001	230v/60 hz/1ph
600002	220v/50 hz/1ph
600003	115v/50 hz/1ph
[illegible]	Carrying Case for ZB-100

ZB-100F MB – MAG Shot Black Light

We have taken our ZB-100F Fan-Cooled black light and added the convenience of energizing your Power Packs or Wet Benches REMOTELY, when checking for cracks in extra long parts.

Our ZB-100F MB has a remote button integrated into the handle of the black light to eliminate the need to continuously return to the MH Unit to initiate a mag shot.

Spare Medium Base Bulb Part # [illegible]

Part #: 600085

León & Russo Ingenieros S.A.
Av. Caminos del Inca 1851 Oficina 301 Surco Lima 33
Tel : 51 1 275 0844 Fax: 51 1 275 0877
info@ndt-innovations.com

Cot-06-1580

Surco 19 de Mayo de 2006

Estimados señores:

Por medio de la presente nos es grato hacerles llegar la cotización de Pedido Directo de nuestra representada **PANAMETRICS-NDT (bussines of R/D TECH INSTRUMENTS)**, por los siguientes equipos.

<u>DETECTOR DE FALLAS POR ULTRASONIDO:</u>

MODEL EPOCH 4 PLUS
ADVANCED DIGITAL MICROPROCESSOR-BASED
ULTRASONIC FLAW DETECTOR
Includes:

EP4-MCA (Mini Charger Adapter)
EP4-BAT (Nickel Metal Hydride Rechargeable Battery)
EP4-CAL-NIST (NIST Calibration Certificate)
EP4-MAN (Instruction Manual)
EP4-TC (Transport Case)
EP4-PS (Stainless Steel Pipe Stand)
EP4-HS (Hand Strap)
Internal Alphanumeric Data logger with Editing Capability
Narrowband Filters
Turn able Square Wave Pulser
RF Display Mode
Microsecond readout for Time of Flight measurements
Dual Gate with Echo-to-Echo Measurements
RS-232 and High Speed Parallel outputs
Analog output
VGA output
DAC and TVG Curve

Precio FOB: **US$ 9,750.00**

ACCESORIOS SUGERIDOS PARA EMPEZAR A UTILIZAR LA TECNICA DE ULTRASONIDO:

1	EP4/RPC	Rubber Protective Case	US$ 278.00

PATRONES DE CALIBRACION:

1	TB7541-1	Test Block IIW	US$ 534.00
1	F129	Hard Case Wood for IIW	US$ 84.00
1	2212-E	1018 Carbon Steel Block	US$ 219.00
1	2214-E	1018 Carbon Steel Block	US$ 237.00

CABLES PARA TRANSDUCTORES:

2	BCB-74-6	BNC to BNC Cable	US$ 53.00
2	BCM-74-6	BNC to Microdot Cable	US$ 53.00

TRANSDUCTORES NORMALES:

1	A106S-RM	2.25 Mhz. 13mm Contact Tranducer	US$ 297.00
1	A109S-RM	5 Mhz. 13mm Accuscan S Transd.	US$ 305.00
1	A104S-RB	2.25Mhz. 25mm Accuscan Transd.	US$ 306.00

TRANSDUCTORES Y ZAPATAS ANGULARES:

1	ABWM-5T 30°	Accupath 30°	US$ 53.00
1	ABWM-5T 45°	Accupath 45°	US$ 53.00
1	ABWM-5T 60°	Accupath 60°	US$ 53.00
1	ABWM-5T 70°	Accupath 70°	US$ 53.00
1	ABWML-5T 90°	Accupath 90°	US$ 80.00
1	A540S-RM	2.25 Mhz. 13mm Angle Bean Transducer	US$ 291.00
1	ABWM-7T 30°	Accupath 30°	US$ 53.00
1	ABWM-7T 45°	Accupath 45°	US$ 53.00
1	ABWM-7T 60°	Accupath 60°	US$ 53.00
1	ABWM-7T 70°	Accupath 70°	US$ 53.00
1	ABWML-7T 90°	Accupath 90°	US$ 80.00
1	A551S-SM	5Mhz. 10mm Angle Bean transducer	US$ 291.00

TRANSDUCTOR Y ZAPATAS AWS:

1	ABWS-8-45	AWS Snail Wedge 45°	US$ 53.00
1	ABWS-8-60	AWS Snail Wedge 60°	US$ 53.00
1	ABWS-8-70	AWS Snail Wedge 70°	US$ 53.00
1	A432S-SB	2.25 Mhz. 0.75x0.75” Angle Transducer	US$284.00

Total: US$ 13,831.00

<u>CONDICIONES</u>:

PRECIOS:	FOB Waltham MA. USA, Incoterms 2000
FORMA DE PAGO:	30 días luego de recibido el equipo en los almacenes de su embarcador en Houston USA.
PLAZO DE ENTREGA:	4 semanas luego de confirmada la orden de compra.
GARANTIA:	Todos los equipos están garantizados por un año desde la fecha de compra (baterías no incluidas).

Sin otro particular por el momento, quedamos a la espera de sus gratas órdenes.

Atentamente,

Gonzalo Arrieta M.Q.
Director Gerente

Stone & Webster, Inc.
100 Technology Center Drive
Stoughton, MA 02072-4705
617-589-5111
FAX: 617-589-2088

Shaw® Stone & Webster, Inc.

Juan Cano Cordova
EnerSur
Ilo Power Plant
Ilo, Peru

January 09, 2006

Re: NDE Services, Alloy Analyzer

Dear Mr. Cordova

Stone & Webster will ship an X-MET3000TX alloy analyzer to the Ilo Power Station this week for work that will be performed in accordance with our contract. The alloy analyzer is provided on a rental agreement for the sum of $6500.

Stone & Webster's Francisco Morales our NDE expert is full qualified to operate the machine. Please note that the instrument uses the latest x-ray technology and has no isotope. Page 2 of the attached brochure describes that no licensing is required and there are no restrictions in travel.

If you have any questions or concerns, please contact me 617-589-7736.

Sincerely,

Peter E. Lannon

Peter Lannon
Project Manager


X-MET3000TX
Lightweight, hand-held, XRF analyzer
X-MET3000TX
scrap sorting and metals analysis
quality control and material verification
measurement of heavy metals in soils
OXFORD
INSTRUMENTS

SISTEMA INTERCONECTADO NACIONAL