UNIVERSIDAD NACIONAL DEL CALLAO

FACULTAD DE INGENIERÍA MECÁNICA Y DE ENERGÍA

ESCUELA PROFESIONAL DE INGENIERÍA MECÁNICA

“MEJORA DE LA DISPONIBILIDAD DEL CIRCUITO DE MOLIENDA DE LA CONCENTRADORA 2 DE LA UNIDAD MINERA TOQUEPALA PERTENECIENTE A SOUTHERN PERÚ COPPER CORPORATION”

INFORME DE TRABAJO DE SUFICIENCIA PROFESIONAL PARA OPTAR EL TÍTULO PROFESIONAL DE INGENIERO MECÁNICO

KEVIN JESÚS ESPINOZA OLIVEROS

Callao, 2021

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FACULTAD DE INGENIERÍA MECÁNICA y DE ENERGÍA
TITULACIÓN POR LA MODALIDAD DE TRABAJO DE SUFICIENCIA PROFESIONAL
I CICLO TALLER -2021

(Resolución N°012-2021-C.F-FIME. del 19 de enero de 2021)

ACTA Nº 017 DE EXPOSICIÓN DE INFORME DE TRABAJO DE SUFICIENCIA PROFESIONAL DEL I CICLO TALLER PARA LA OBTENCIÓN DEL TÍTULO PROFESIONAL DE INGENIERO MECÁNICO E INGENIERO EN ENERGIA

LIBRO 001 FOLIO N°046, ACTA Nº 017 DE EXPOSICIÓN DE INFORME DE TRABAJO DE SUFICIENCIA PROFESIONAL DEL I CICLO TALLER PARA LA OBTENCIÓN DEL TÍTULO PROFESIONAL DE INGENIERO MECÁNICO

A los 09 días del mes julio, del año 2021, siendo las **17:10 horas**, se reunieron, en la sala meet: https://meet.google.com/ypx-xsbx-dxh, el **JURADO EVALUADOR DE INFORME FINAL** para la obtención del TÍTULO profesional de **Ingeniero Mecánico de la Facultad de Ingeniería Mecánica y de Energía**, conformado por los siguientes docentes ordinarios de la **Universidad Nacional del Callao**:

- Dr. Palomino Correa, Juan Manuel : Presidente
- Mg. Caldas Basauri, Alfonso Santiago : Secretario
- Mg. Blas Zarzosa Adolfo Orlando : Vocal
- Mg. Collante Huanto, Andrés : Suplente

Se dio inicio al acto de exposición de informe de trabajo para titulación del Bachiller **ESPINOZA OLIVEROS, KEVIN JESÚS,** quien habiendo cumplido con los requisitos para optar el Título Profesional de Ingeniero Mecánico sustenta su informe titulado **"MEJORA DE LA DISPONIBILIDAD DEL CIRCUITO DE MOLIENDA DE LA CONCENTRADORA 2 DE LA UNIDAD MINERA TOQUEPALA PERTENECIENTE A SOUTHERN PERU COPPER CORPORATION",** cumpliendo con la sustentación en acto público, de manera no presencial a través de la Plataforma Virtual, en cumplimiento de la declaración de emergencia adoptada por el Poder Ejecutivo para afrontar la pandemia del Covid- 19, a través del D.S. Nº 044-2020-PCM y lo dispuesto en el DU Nº 026-2020 y en concordancia con la Resolución del Consejo Directivo N°039-2020-SUNEDU-CD y la Resolución Viceministerial Nº 085-2020-MINEDU, que aprueba las "Orientaciones para la continuidad del servicio educativo superior universitario";

Con el quórum reglamentario de ley, se dio inicio a la sustentación de conformidad con lo establecido por el Reglamento de Grados y Títulos vigente. Luego de la exposición, y la absolución de las preguntas formuladas por el Jurado y efectuadas las deliberaciones pertinentes, acordó: Dar por **APROBADO** con la escala de calificación cualitativa **BUENO** y calificación cuantitativa **14 (Catorce)** la presente **EXPOSICIÓN DE INFORME DE TRABAJO DE SUFICIENCIA PROFESIONAL**, conforme a lo dispuesto en el Art. 27 del Reglamento de Grados y Títulos de la UNAC, aprobado por Resolución de Consejo Universitario Nº 245-2018- CU del 30 de Octubre del 2018.

Se dio por cerrada la Sesión a las **17:45 horas** del día 09 de julio del 2021.

Dr. Juan Manuel Palomino Correa
Presidente de Jurado

Mg. Alfonso Santiago Caldas Basauri
Secretario de Jurado

Mg. Adolfo Orlando Blas Zarzosa
Vocal de Jurado

Mg. Andrés Collante Huanto
Suplente de Jurado

UNIVERSIDAD NACIONAL DEL CALLAO
FACULTAD DE INGENIERIA MECANICA Y DE ENERGÍA
Jurado de Exposición

I N F O R M E

Visto, el Trabajo de Suficiencia Profesional, titulado: **"MEJORA DE LA DISPONIBILIDAD DEL CIRCUITO DE MOLIENDA DE LA CONCENTRADORA 2 DE LA UNIDAD MINERA TOQUEPALA PERTENECIENTE A SOUTHERN PERÚ COPPER CORPORATION",** presentado por el señor bachiller en Ingeniería Mecánica, **ESPINOZA OLIVEROS, KEVIN JESÚS.**

A QUIEN CORRESPONDA:

El **Presidente del Jurado** del señor bachiller en Ingeniería Mecánica, **ESPINOZA OLIVEROS, KEVIN JESÚS,** manifiesta que la Exposición del Trabajo de Suficiencia Profesional, se realizó en forma virtual, mediante la sala: meet.google.com/ypx-xsbx-dxh, el día viernes 09 de julio de 2021 a las 17:10 horas, encontrándose observaciones, las mismas que han sido revisadas cuidadosamente por cada uno de los miembros del Jurado, y el interesado ha levantado correctamente.

En tal sentido, en mi calidad de Presidente de Jurado, emito el presente informe favorable para los fines pertinentes.

Bellavista, 15 de setiembre de 2021

JUAN MANUEL PALOMINO CORREA
INGENIERO MECANICO
Reg. del Colegio de Ingenieros N° 20484

....
Dr. Juan Manuel Palomino Correa
Presidente de Jurado de Exposición
Trabajo de Suficiencia Profesional

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I. ASPECTOS GENERALES

1.1. Objetivos

1.1.1. Objetivo general

Aumentar la disponibilidad del circuito de Molienda de la Concentradora 2 de la mina Toquepala ubicada en el distrito de Ilabaya, provincia Jorge Basadre, departamento Tacna, perteneciente a Southern Perú Copper Corporation, instalando una nueva bomba Stand By centrifuga para pulpa de concentrado con capacidad nominal de 7000 m^3/h.

1.1.2. Objetivo especifico

A. Evaluar los Indicadores de Mantenimiento con el objetivo de mejorar la disponibilidad del circuito de Molienda de la Concentradora 2 de la mina Toquepala, ubicada en el distrito de Ilabaya, provincia Jorge Basadre, departamento Tacna, perteneciente a Southern Perú Copper Corporation.

B. Seleccionar una bomba Stand By centrifuga para pulpa de concentrado con capacidad nominal de 7000 m^3/h para el circuito de Molienda de la Concentradora 2 de la mina Toquepala ubicada en el distrito de Ilabaya, provincia Jorge Basadre, departamento Tacna, perteneciente a Southern Perú Copper Corporation.

C. Realizar la evaluación técnica y económica de la viabilidad del proyecto.

1.2. Organización de la empresa o institución

1.2.1. Antecedentes Históricos

Weir Minerals Global

Weir Group plc, es una empresa de ingeniería multinacional británica con sede en Glasgow , Escocia.

La empresa fue fundada en 1871 como una empresa de ingeniería por dos hermanos, George y James Weir , fundando G. & J. Weir Ltd. Los hermanos Weir produjeron numerosos inventos innovadores en equipos de bombeo, principalmente para los astilleros Clyde y el vapor, barcos construidos allí. Estas bombas se hicieron muy conocidas por su uso como bombas de agua de alimentación de calderas y para equipos auxiliares de barcos, como evaporadores.

Bajo WD Weir , la empresa se dedicó a producir municiones y material de guerra en la Primera Guerra Mundial. Además de proyectiles, fabricaron aviones, incluido el caza y bombardero Royal Aircraft Factory FE2 . James George Weir (aviador, hijo de James Galloway Weir), director de la empresa, formó la Cierva Autogiro Company . G & J Weir sería un patrocinador financiero de la empresa durante su existencia. En 1943, proporcionaron los fondos para la construcción del W.9 , un helicóptero experimental, según los requisitos del Ministerio del Aire.

Se cotizó por primera vez en la Bolsa de Valores de Londres en 1946. Las bombas de vapor Weir de doble acción eran prácticamente un equipamiento estándar en los barcos de vapor construidos en Gran Bretaña , y se utilizaban para bombear agua, combustible, aire y mucho más hasta bien entrada la década de 1950. cómo se utiliza en otros barcos en todo el mundo. En diciembre de 1968, Weir Group hizo una oferta para comprar el fabricante de bombas británico rival Worthington-Simpson , siguiendo una oferta de Studebaker-Worthington .

En 1969 Studebaker-Worthington adquirió Worthington-Simpson. Después de algunas negociaciones, Weir's adquirió el 50% de Worthington Simpson. Se creó una nueva empresa conjunta denominada Worthington Weir para gestionar las ventas internacionales de las dos empresas matrices. En 1989, la empresa adquirió Hopkinsons, una empresa con sede en Elland que fabrica válvulas y controles.

SIGLO XXI

En julio de 2005, Weir vendió sus negocios de desalinización y tratamiento de agua (Weir Westgarth, Weir Entropie y Weir Envig) a Veolia Water Systems, parte de la división de agua de Veolia Environnement : Weir Westgarth había sido un pionero de la destilación flash de múltiples etapas proceso utilizado predominantemente para producir agua desalada a partir del agua de mar .

En mayo de 2007, la compañía vendió su Glasgow basados en los negocios Weir Pumps a Jim McColl 's Clyde Blowers plc , con la compañía de la bomba posteriormente ser renombrado a Clyde Pumps Ltd.

En febrero de 2015, la compañía emitió una advertencia de ganancias prediciendo pérdidas significativas; el mercado de esquisto de los Estados Unidos había experimentado una contracción y la empresa tuvo que reducir su fuerza laboral en aproximadamente 650. En octubre de 2020, la compañía vendió su división de petróleo y gas a Caterpillar Inc. por $ 405 millones, habiendo vendido previamente su división de control de flujo en 2019.

Vulco Perú S.A.

También llamada Weir Minerals Perú, fundada en 1986, es una sucursal y división de la empresa británica Weir, que ofrece productos de ingeniería resistentes al desgaste a compañías de los sectores de minería, petróleo y gas, petroquímico e hidráulico. Su oferta incluye bombas, válvulas, caucho, trituradoras, cribas, carretes para mangueras y tubos, centrífugas, pontones y barcazas, hidrociclones, revestimientos Anti-desgaste, revestimientos de molinos, alimentadores, cintas transportadoras, y lavadoras. Vulco Perú también ofrece servicios de ingeniería y productos asociados para mejorar el rendimiento, de mantenimiento preventivo y predictivo de sus equipos, y además otorga facilidades de tipo comercial a sus clientes. Weir Minerals Perú cuenta con centros de servicios, unidades de manufactura y oficinas de ventas en Arequipa y Cajamarca, además de un centro de servicio en Espinar. Su oficina central se halla en Lima.

1.2.2. Filosofía Empresarial

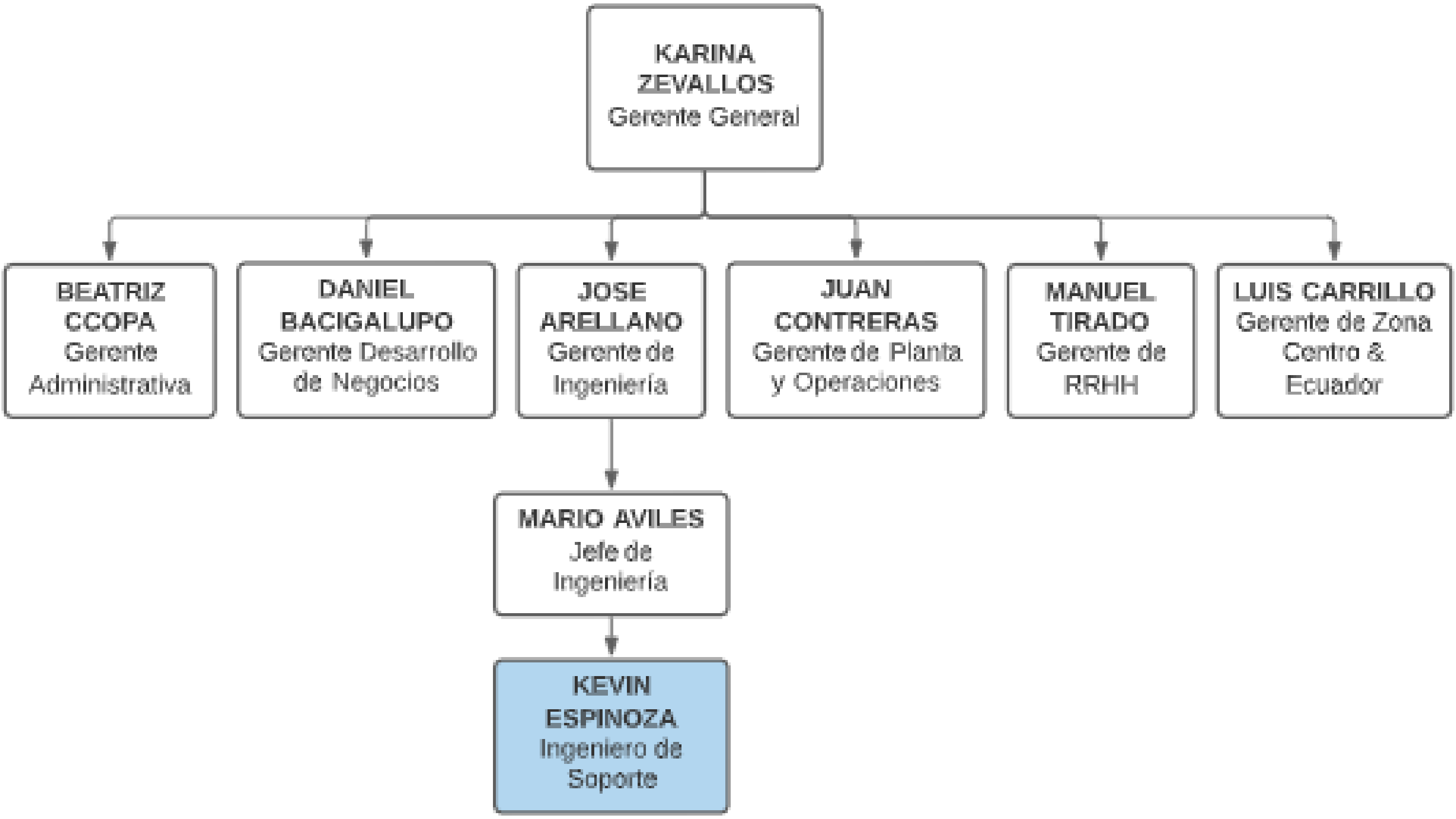
Visión: Ser la empresa de ingeniería más admirada en nuestros mercados.

Misión: Ayudar a nuestros clientes a entregar de manera sostenida y eficiente la energía y los recursos que necesitan en un mundo en crecimiento.

Como lo logramos: Cuidando a nuestros colegas, vecinos y le medio ambiente e inspirándolos a florecer. Trabajando en conjunto para entregar soluciones distintivas que ofrecen una atractiva rentabilidad. Impulsando el desarrollo de nuevas tecnologías y capacidades que lideran el mercado. Entregando excelencia a nuestros clientes, colegas, accionistas y comunidades a través de un fuerte liderazgo y enfoque.

1.3.3. Estructura Organizacional

Figura 1.1



Fuente: Elaboración propia

II. FUNDAMENTACION DE LA EXPERIENCIA PROFESIONAL

2.1. Marco teórico

2.1.1. Bases teóricas

A. Conceptos Para Seleccionamiento De Bombas:

Los sistemas de Bombeo son considerados Sistemas de manejo de energía por lo cual se debe manejar un buen balance para evitar pérdidas innecesarias.

Las bombas son consideradas equipos que añaden energía a los líquidos con la finalidad de transportarlos a determinadas condiciones, el procedimiento para la selección de estos equipos involucra la determinación de ciertos parámetros los cuales se mencionan a continuación:

- Flujos Internos incompresibles (laminar y turbulento)
- NPSH
- Pérdidas de energía en tuberías.
- Potencia Hidráulica
- Altura dinámica total, ADT
- Diámetro Optimo
- Espesor adecuado
- Especificación de material comercial

Para comprender mejor estos términos y, poder realizar un cálculo adecuado, tenemos que conocer ciertos principios y formas de cálculo como se define a continuación.

a) Principio De Bernoulli:

El principio de Bernoulli, también denominado ecuación de Bernoulli o Trinomio de Bernoulli, describe el comportamiento de un fluido moviéndose a lo largo de una corriente.

Este principio puede ser derivado del principio de conservación de la energía, esto indica que, en un flujo constante la suma de todas las formas de energía en un fluido a lo largo de una línea de corriente es la misma, esto requiere a que la suma de la energía cinética, la energía potencial y la energía interna permanezca constante. El principio de Bemoullil se rige a la siguiente ecuación:

Principio de Bernoulli y relación con la perdida presión.

$$p_1 + \frac{\rho V_1^2}{2} + \rho g z_1 = p_2 + \frac{\rho V_2^2}{2} + \rho g z_2$$

O

$$\frac{p_1}{\gamma} + \frac{V_1^2}{2g} + z_1 = \frac{p_2}{\gamma} + \frac{V_2^2}{2g} + z_2$$

Donde:

$P = Presión\ termodinamica$
$\rho = Densidad$
$V = Velocidad$
$g = Gravedad$
$z = Altura$
$\gamma = Peso\ especifico$

b) Flujos Internos Incompresibles:

En la actualidad no existe un análisis general del comportamiento de los fluidos incompresibles; existen diferentes métodos de análisis y comprobación del comportamiento interno de los fluidos desde los análisis y formulaciones matemáticos, luego derivando a las simulaciones mediante software y programación hasta los datos y resultados experimentales.

Uno de los tantos modelos matemáticos es la aplicación del famoso número de Reynolds para la determinación del comportamiento del fluido a determinadas condiciones, siendo así que se puede determinar el movimiento desde ser suave (laminar) hasta ser un flujo agitado (turbulento).

El número de Reynolds se halla con la siguiente ecuación:

$$R_e = \frac{Fuerzas\ Inerciales}{Fuerzas\ Viscosas} = \frac{\rho * D * v}{\mu} = \frac{v * D}{\nu}$$

Donde:

ρ = Densidad del fluido

μ = Viscosidad dinámica del fluido

v = Velocidad del fluido

D = Diámetro interno de la tubería

ν = Viscosidad cinemática del fluido

c) Flujo laminar

Es uno de los dos tipos principales de flujo en fluido. Se llama flujo laminar o corriente laminar, al movimiento de un fluido cuando éste es ordenado, estratificado, suave. En un flujo laminar el fluido se mueve en láminas paralelas sin entremezclarse y cada partícula de fluido sigue una trayectoria suave, llamada línea de corriente. En flujos laminares el mecanismo de transporte lateral es exclusivamente molecular. Se puede presentar en las duchas eléctricas vemos que tienen líneas paralelas.

El flujo laminar es típico de fluidos a velocidades bajas o viscosidades altas, mientras fluidos de viscosidad baja, velocidad alta o grandes caudales suelen ser turbulentos. El número de Reynolds es un parámetro adimensional importante en las ecuaciones que describen en qué condiciones el flujo será laminar o turbulento. En el caso de fluido que se mueve en un tubo de sección circular, el flujo persistente será laminar por debajo de un número de Reynolds crítico de aproximadamente 2300. Para números de Reynolds más altos el flujo turbulento puede sostenerse de forma indefinida. Sin embargo, el número de Reynolds que delimita flujo turbulento y laminar depende de la geometría del sistema y además la transición de flujo laminar a turbulento es en general sensible a mido e imperfecciones en el sistema.

El perfil laminar de velocidades en una tubería tiene forma de una parábola, donde la velocidad máxima se encuentra en el eje del tubo y la velocidad es igual a cero en la pared del tubo. En este caso, la pérdida de energía es proporcional a la velocidad media, mucho menor que en el caso de flujo turbulento.

d) Flujo turbulento

Al aumentar el gradiente de velocidad se incrementa la fricción entre partículas vecinas al fluido, y estas adquieren una energía de rotación apreciable, la viscosidad pierde su efecto, y debido a la rotación las partículas cambian de trayectoria. Al pasar de unas trayectorias a otras, las partículas chocan entre sí y cambian de rumbo en forma errática.

Este flujo se caracteriza porque:

- Las partículas del fluido no se mueven siguiendo trayectorias definidas.
- La acción de la viscosidad es despreciable.
- Las partículas del fluido poseen energía de rotación apreciable, y se mueven en forma errática chocando unas con otras.
- Al entrar las partículas de fluido a capas de diferente velocidad, su momento lineal aumenta o disminuye, y el de las partículas vecina la hacen en forma contraria.

Cuando las fuerzas de inercia del fluido en movimiento son muy bajas, la viscosidad es la fuerza dominante y el flujo es laminar. Cuando predominan las fuerzas de inercia el flujo es turbulento. Reynolds estableció una relación que permite establecer el tipo de flujo que posee un determinado problema. Para números de Reynolds bajos el flujo es laminar, y para valores altos el flujo es turbulento.

- Rangos de valores de número de Reynolds y comportamiento del fluido

$0<Re<1$: movimiento laminar "lento" altamente viscoso

$1 < Re < 100$: laminar, fuerte dependencia del Re

$100 < Re < 10^3$: Laminar, es útil la teoría de capa limite

$10^3< Re < 10^4$: Transición a la turbulencia

$10^4 < Re < 10^6$: Turbulento, moderada dependencia del Re

$10^6 < Re < \infty$: Turbulento, débil dependencia del Re

e) Pérdidas de energía en tuberías:

Los tipos de pérdidas en tuberías se dividen y calculan de la siguiente manera:

- Pérdidas primarias:

  Las pérdidas por fricción o también consideradas pérdidas primarias se determinan mediante la ecuación de Darcy - Weisbach4. Ecuación de Darcy-Weisbach.

$$h_L = \frac{f * L * V^2}{2 * g * \emptyset}$$

h_L= pérdida primaria de energía en metros

f = factor de fricción

L = longitud de la tubería en metros

V = velocidad promedio en la sección transversal del conducto en metros por segundo

g = gravedad en metros por segundo al cuadrado

$\emptyset$ = diámetro de la tubería en metros

El factor de fricción se determina mediante las siguientes ecuaciones analíticas

- ✓ Si el flujo es laminar, con Re < 2300, se determinará mediante la ecuación de Hagen — Poiseuille5.

  Ecuación De Hagen — Poiseuille

$$f = \frac{64}{Re}$$

- ✓ La ecuación de fricción iterativa de Colebrook-White se utiliza para obtener factores de fricción en el régimen de flujo turbulento.

  Ecuación de Colebrook —White

$$f = \left[1,14 - 2\, log\left(\frac{\varepsilon}{D} + \frac{9.35}{Re\sqrt{f}}\right)\right]^{-2}$$

f: Factor de fricción

Re: Número de Reynolds

ε: Rugosidad absoluta

D: Diámetro Hidráulico

- Perdidas Secundarias:

  Son todas aquellas que se originan cuando el fluido pasa a través de instrumentos de medida, cambios de sección, cambios de dirección, etc. Su magnitud se calcula mediante la ecuación de pérdidas secundarias.

  Ecuación de Pérdidas Secundarias

$$h_s = k\frac{v^2}{2g}$$

h_s: Pérdidas secundarias o por accesorios

g: Aceleración de la gravedad (9,81 m/s2)

V: Velocidad del fluido

K: Coeficiente de pérdidas del accesorio, Depende de la geometría y el acabado Superficial interno del accesorio.

Una manera de evaluar con relativa facilidad la magnitud de una perdida secundaria en una red de tuberías es aplicando el concepto de longitud equivalente: es la longitud de una tubería de sección circular que genera la misma caída de presión que un accesorio, asumiendo igual fluido e igual velocidad promedio.

f) Longitud Equivalente

Un método no completamente exacto pero válido a efectos de estimar las pérdidas de carga localizadas consiste en expresarlas en forma de longitud equivalente, es decir, valorar cuántos metros de tubería recta del mismo diámetro producen una pérdida de carga continua que equivale a la pérdida que se produce en el punto singular, su magnitud se calcula mediante la ecuación de longitud equivalente.

Ecuación de Longitud Equivalente

$$L_e = \frac{k \times D}{f}$$

La pérdida de carga total en una tubería de longitud L con 1 accesorios de longitud equivalente Le i cada uno de ellos, será la que produce una longitud total de una tubería del mismo diámetro, su magnitud se calcula mediante la ecuación de longitud equivalente total

Ecuación de Longitud Equivalente Total

$$L_T = L + \sum Le_t$$

g) NPSH

NPSH es un acrónimo de Net Positive Suction Head, también conocido como ANPA (Altura Neta Positiva en la Aspiración) o CNPA (Carga Neta Positiva en Aspiración). Es la caída interna de presión que sufre un fluido cuando este ingresa al interior de una bomba centrífuga. Cuando el fluido ingresa a una bomba centrífuga, lo hace siempre por el centro del rodete impulsor, lugar en donde toma contacto con las paletas de dicho rodete para ser luego impulsado hacia la periferia de la bomba. Pero, al hacer contacto con dicha paletas, el fluido sufre lo que se denomina "Efecto de la Proa de Fuhrmann". Este efecto, establece que el fluido, que ya ha pasado por las pérdidas de fricción y de accesorios del sistema de tuberías, aún continúa perdiendo presión esta vez dentro de la bomba centrífuga, al reacomodarse al contorno de la paleta, en cuya punta el fluido choca contra el extremo, se reacomoda rápidamente, aumenta su velocidad, y por ende disminuye su presión. Otro factor que determina esta caída de presión es el hecho de que el flujo ingresa al centro del rodete de forma axial, y se debe reorientar para seguir el contorno de las paletas.

La NPSH es un parámetro importante en el diseño de un circuito de bombeo: si la presión en el circuito es menor que la presión de vapor del líquido, éste entrará en algo parecido a la ebullición: se vaporiza, produciéndose el fenómeno de cavitación, que puede dificultar o impedir la circulación de líquido, y causar daños en los elementos del circuito. En las instalaciones de bombeo se debe tener en cuenta la NPSH referida a la aspiración de la bomba, distinguiéndose dos tipos de NPSH:

- ✓ NPSH requerida: es la NPSH mínima que se necesita para evitar la cavitación. Depende de las características de la bomba, por lo que es un dato que debe proporcionar el fabricante en sus curvas de operación.

  ***cavítación:** La cavitación ocurre cuando la succión de la bomba se encuentra en unas condiciones de baja presión/alto vacío que hace que el líquido se transforme en vapor a la entrada del rodete. Este vapor es transportado hasta la zona de descarga de la bomba donde el vacío desaparece y el vapor del líquido es nuevamente comprimido debido a la presión de descarga. Se produce en ese momento una violenta implosión sobre la superficie del rodete. Un rodete que ha trabajado bajo condiciones de cavitación de succión presenta grandes cavidades producidas por los trozos de material arrancados por el fenómeno. Esto origina el fallo• prematuro de la bomba.
- ✓ NPSH disponible: depende de las características de la instalación y del líquido a bombear.

$$\boldsymbol{NPSHd = \frac{Pa}{y} \pm H_a - h_F - \frac{Pv}{y}}$$

Donde:

y : es el peso específico del líquido (N/m3).

Pa: es la presión en el nivel de aspiración, en Pa

H_a: es la altura geométrica de aspiración en nn.c.I. (Hase resta si el nivel del líquido está por abajo del ojo del impulsor, se suma si el nivel del líquido está por arriba del impulsor.

hf : es la pérdida de carga en la línea de aspiración, en m.c.I.

Pv: es la presión de vapor del líquido a la temperatura de bombeo, en Pa

+ : Se utilizará este signo, cuando en nivel de aspiración se encuentre por encima del eje de la bomba.

— : Se utilizará este signo, cuando en nivel de aspiración se encuentre por debajo del eje de la bomba.

h) Potencia Hidráulica

La energía hidráulica que una bomba comunica a la sustancia operante se evalúa mediante la ecuación de energía hidráulica.

Ecuación de Energía Hidráulica

$$Pb = p \times g \times Q \times h_b = y \times Q \times h_b$$

En donde:

Pb: es la potencia teórica de la bomba (en Vatios; 1 Hp = 745.7 Vatios)

p: es la densidad del fluido

g: es la aceleración de la gravedad (generalmente se adopta: 9.81 m/s2)

y: es el peso específico del fluido

Q: es el caudal (m3 /s)

hb: es la ganancia de carga en la bomba, o en otros términos, altura dinámica de la bomba (m)

B. Conceptos para análisis Financiero

a. VAN – Valor Neto Actual

El valor actual neto (VAN) es un criterio de inversión que consiste en actualizar los cobros y pagos de un proyecto o inversión para conocer cuánto se va a ganar o perder con esa inversión. También se conoce como valor neto actual (VNA), valor actualizado neto o valor presente neto (VPN).

Para ello trae todos los flujos de caja al momento presente descontándolos a un tipo de interés determinado. El VAN va a expresar una medida de rentabilidad del proyecto en términos absolutos netos, es decir, en nº de unidades monetarias (euros, dólares, pesos, etc).

Fórmula del valor actual neto (VAN)

Se utiliza para la valoración de distintas opciones de inversión. Ya que calculando el VAN de distintas inversiones vamos a conocer con cuál de ellas vamos a obtener una mayor ganancia.

$$\mathrm{VAN} = -\mathrm{I} + \frac{F_1}{(1+i)^1} + \frac{F_2}{(1+i)^2} + \ldots$$

Donde:

Fn: Flujo en el periodo de tiempo establecido

i: Interés tomado como referencia o TIR que logra que el VAN sea 0

I: Inversión Inicial

El VAN sirve para generar dos tipos de decisiones: en primer lugar, ver si las inversiones son efectuables y en segundo lugar, ver qué inversión es mejor que otra en términos absolutos. Los criterios de decisión van a ser los siguientes:

- VAN > 0 : El valor actualizado de los cobro y pagos futuros de la inversión, a la tasa de descuento elegida generará beneficios.
- VAN = 0 : El proyecto de inversión no generará ni beneficios ni pérdidas, siendo su realización, en principio, indiferente.
- VAN < 0 : El proyecto de inversión generará pérdidas, por lo que deberá ser rechazado.

b. TIR – Tasa Interna De Retorno

La tasa interna de retorno (TIR) es la tasa de interés o rentabilidad que ofrece una inversión. Es decir, es el porcentaje de beneficio o pérdida que tendrá una inversión para las cantidades que no se han retirado del proyecto.

Es una medida utilizada en la evaluación de proyectos de inversión que está muy relacionada con el valor actualizado neto (VAN). También se define como el valor de la tasa de descuento que hace que el VAN sea igual a cero, para un proyecto de inversión dado.

La tasa interna de retorno (TIR) nos da una medida relativa de la rentabilidad, es decir, va a venir expresada en tanto por ciento. El principal problema radica en su cálculo, ya que el número de periodos dará el orden de la ecuación a resolver. Para resolver este problema se puede acudir a diversas aproximaciones, utilizar una calculadora financiera o un programa informático.

- ¿Cómo se calcula la TIR?

  También se puede definir basándonos en su cálculo, la TIR es la tasa de descuento que iguala, en el momento inicial, la corriente futura de cobros con la de pagos, generando un VAN igual a cero:

$$\mathrm{TIR} = \sum_{T=0}^{n} \frac{Fn}{(1+\mathrm{i})^{n}} = 0$$

Donde:

Fn: Flujo en el periodo de tiempo establecido

i: Interés tomado como referencia

n: Periodo de tiempo o momento temporal

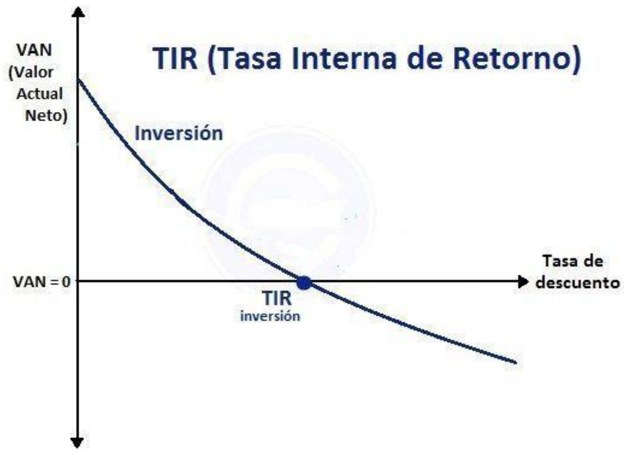
El criterio de selección para determinar si un proyecto es viable, será el siguiente donde “k” es la tasa de descuento de flujos elegida para el cálculo del VAN:

- Si TIR > k, el proyecto de inversión será aceptado. En este caso, la tasa de rendimiento interno que obtenemos es superior a la tasa mínima de rentabilidad exigida a la inversión.
- Si TIR = k, estaríamos en una situación similar a la que se producía cuando el VAN era igual a cero. En esta situación, la inversión podrá llevarse a cabo si mejora la posición competitiva de la empresa y no hay alternativas más favorables.
- Si TIR < k, el proyecto debe rechazarse. No se alcanza la rentabilidad mínima que le pedimos a la inversión.

- Representación gráfica de la TIR

Como hemos comentado anteriormente, la Tasa Interna de Retorno es el punto en el cuál el VAN es cero. Por lo que si dibujamos en un gráfico el VAN de una inversión en el eje de ordenadas y una tasa de descuento (rentabilidad) en el eje de abscisas, la inversión será una curva descendente. El TIR será el punto donde esa inversión cruce el eje de abscisas, que es el lugar donde el VAN es igual a cero:

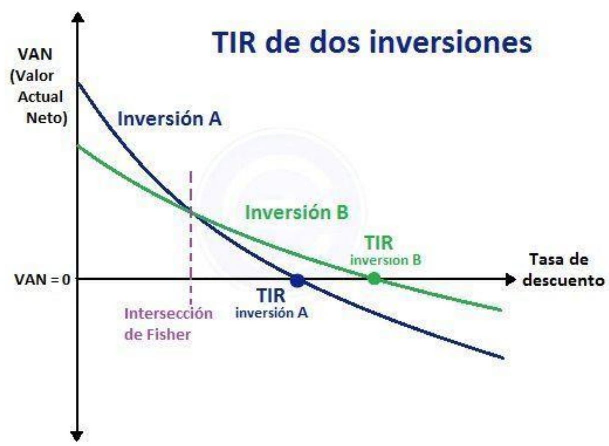
Figura 2.1



Fuente: https://economipedia.com/

Si dibujamos la TIR de dos inversiones podemos ver la diferencia entre el cálculo del VAN y TIR. El punto donde se cruzan se conoce como intersección de Fisher.

Figura 2.2



Fuente: https://economipedia.com/

C. Conceptos para análisis de mantenimiento

a. Disponibilidad:

Es la capacidad de un activo o componente para estar en un estado operativo (arriba) para realizar una función requerida bajo condiciones dadas en un instante dado de tiempo o durante un determinado intervalo de tiempo, asumiendo que los recursos externos necesarios se han proporcionado.

b. Confiabilidad:

Es la capacidad de un activo o componente para realizar una función requerida bajo condiciones dadas para un intervalo de tiempo dado. Es decir, cuando hablamos de confiabilidad el componente trabaja continuamente durante un periodo de tiempo dado, en otras palabras, la función del componente no se interrumpe, el componente se pone en operación (arriba) y se mantiene arriba. Por otra parte, cuando hablamos de disponibilidad el componente es puesto arriba en un instante dado y no importa lo que pase después, la función del componente puede ser interrumpida sin ningún problema.

Veamos ahora las ecuaciones matemáticas que se utilizan en el ámbito operacional para el cálculo de estos dos parámetros, en función de los tiempos de mantenimiento:

La confiabilidad operacional Co

$$Co = MTBF/(MTBF+MTTR)$$

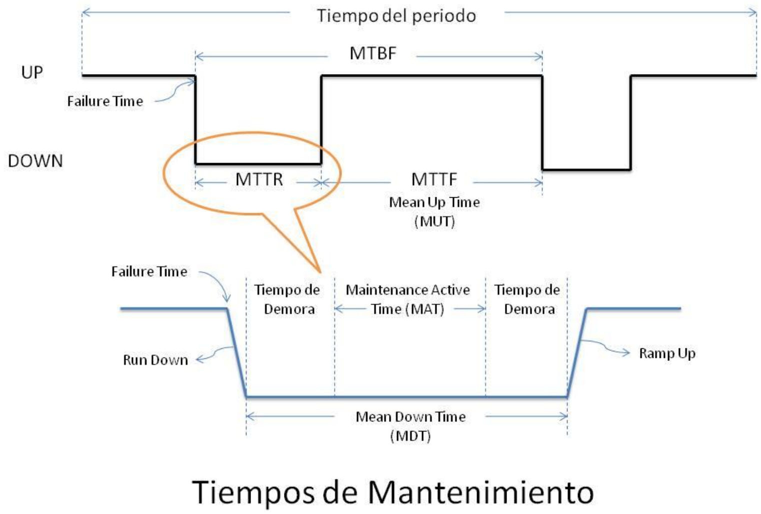
La disponibilidad Operacional Do

$$Do = MUT/(MUT+MTTR)$$

Donde:

- MTBF (Mean Time Between Failures): Es el Tiempo promedio entre Fallas
- MTTR (Mean Time To Repair): Es el Tiempo Promedio para Reparar
- MUT (Mean Up Time): es Tiempo Promedio en Operación (arriba) o Tiempo promedio para fallar (MTTF)

Figura 2.3



Fuente: https://maintenancela.blogspot.com/2011/10/confiabilidad-disponibilidad-y.html.

De las ecuaciones anteriores tenemos que la de Confiabilidad está regida por el tiempo entre fallas (MTBF) el cual involucra la ocurrencia de esta, mientras que la de Disponibilidad tiene que ver con los tiempos de operación (MUT) y los tiempos fuera de servicio (MTTR), estos últimos pueden o no tomar en cuenta a los tiempos dedicados a los mantenimientos preventivo, las actividades de mantenimiento correctivos programados y las reparaciones de fallas de los componentes.

Dicho lo anterior podemos reformular la explicación inicial diciendo que cuando hablamos de confiabilidad nos referimos a los tiempos que involucran la ocurrencia de una falla y cuando hablamos de disponibilidad nos referimos a los tiempos de operación y fuera de servicio de los componentes, incluyendo o no los PM, CM y las fallas.

c. Mantenibilidad:

Mantenibilidad es definida por la ISO/DIS 14224, como la capacidad (o probabilidad si hablamos en términos estadísticos), bajo condiciones dadas, que tiene un activo o componente de ser mantenido o restaurado en un periodo de tiempo dado a un estado donde sea capaz de realizar su función original nuevamente, cuando el mantenimiento ha sido realizado bajo condiciones prescritas, con procedimientos y medios adecuados. Esto quiere decir, que si un componente tiene un 95% de Mantenibilidad en una hora, entonces habrá 95% de probabilidad de que ese componente sea reparado exitosamente en una hora.

La ecuación clásica de la Mantenibilidad es:

$$M(t) = 1 - e^{-(\mu t)}$$

Cuando μ o rata de reparación es constante.

El MTTR (Mean Time To Repair) es el tiempo promedio para reparar de un componente cuando esta falla, es parte del tiempo promedio arriba o en servicio (MDT) y es un indicador directo de la Mantenibilidad.

Podemos definir la rata de reparación (μ) en función del MTTR como

$$\mu = 1/MTTR$$

La rata de reparación es un parámetro el cual permite evaluar la probabilidad que tiene un componente a ser reparado y juega un papel exactamente similar a la rata de falla (λ = 1/MTBF) para el cálculo de la confiabilidad.

Podemos decir entonces que la Mantenibilidad está inversamente relacionada con la duración y el esfuerzo requerido para realizar las actividades de Mantenimiento. Puede ser asociada de manera inversa con el tiempo que se toma en lograr acometer las acciones de mantenimiento en relación con la obtención del comportamiento deseable de un componente.

Existen dos tipos de Mantenibilidad: la intrínseca, que está relacionada al aspecto de diseño de una instalación y que hace una consideración sobre como las características de diseño ayudan al mantenimiento de un componente (accesibilidad y facilidades para el mantenimiento) y la extrínseca, que considera el contexto de dependencia de la gestión de mantenimiento cuando se repara un componente (logística, organización de las tareas, aislamiento, entrega de los equipos etc.), estas dos diferenciaciones deben considerarse al analizar los factores que afectan a la Mantenibilidad.

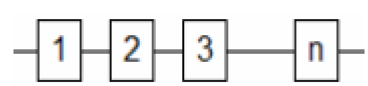
d. Diagrama de Bloques de Confiabilidad

Los diagramas de bloques (DB) son una representación común para calcular la confiabilidad de sistemas (Xu et al., 2009). Esta representación se complementa con técnicas estadísticas para evaluar los parámetros de la mejor función densidad de probabilidad que represente el tiempo entre fallas. En forma resumida el método de DB consiste en obtener el valor de confiabilidad de cada elemento del sistema y posteriormente se encuentra la confiabilidad correspondiente a cada subsistema, al reducir el sistema a un arreglo equivalente conforme a los sistemas en serie y paralelo que lo constituyen.

- Sistema serie

Un sistema serie desde el punto de vista de confiabilidad, es aquel en el cual todos los componentes deben funcionar adecuadamente para que el sistema opere; es decir si uno de los componentes falla, el sistema falla.

Figura 2.4



Fuente: Elaboración Propia

Para un sistema serie, su confiabilidad cuando n elementos son independientes es:

$$\begin{aligned} R_s &= P\left[E_1 \cap E_2 \cap \ldots \cap E_n\right] \\ &= P\left[E_1\right] P\left[E_2\right] \ldots P\left[E_n\right] = R_1 R_2 \ldots R_n \end{aligned}$$

Donde:

$R_1 \ldots R_n$ son las confiabilidades individuales de los componentes 1,…, n

De manera general para n componentes independientes:

$$R_s(t) = \prod_{i=1}^{n} R_i(t) \qquad \text{Confiabilidad del Sistema}$$

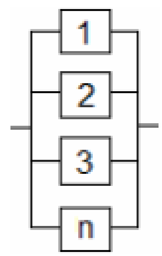
$$Q_s(t) = 1 - \prod_{i=1}^{n} R_i(t) \qquad \text{Probabilidad de Falla del Sistema}$$

Para R_s (t) se puede inferir que la confiabilidad conformada por n componentes individuales conectados en serie, siempre será menor que la confiabilidad del componente menos confiable.

- Sistema paralelo

  Un sistema paralelo desde el punto de vista de confiabilidad es aquel en el cual el funcionamiento adecuado de cualquiera de sus n componentes, implica la operación del sistema.

Figura 2.5



Fuente: Elaboración Propia

Cabe anotar que en esta configuración para que el sistema falle deben fallar todos los n componentes.

$$Q_p = P\left[\overline{E}_1 \cap \overline{E}_2 \cap \ldots \cap \overline{E}_n\right]$$
$$= P\left[\overline{E}_1\right] P\left[\overline{E}_2\right] \ldots P\left[\overline{E}_n\right] = Q_1 Q_2 \ldots Q_n$$

Donde:

Q_n: Son las probabilidades de falla individuales de los componentes n-ésimos, generalmente son funciones del tiempo.

De manera general para un sistema de n componentes no reparables en paralelo se tiene:

$$Q_p(t) = \prod_{i=1}^{n} Q_i(t)$$
$$R_p(t) = 1 - \prod_{i=1}^{n} Q_i(t)$$
$$R_p(t) = 1 - \prod_{i=1}^{n} \left(1 - R_i(t)\right)$$

Para R_s (t) se concluye, que la confiabilidad para un sistema en paralelo con n elementos independientes, crece con el aumento de los subconjuntos bajo estudio.

2.1.2. Aspectos Normativos

A. ASME B16.5: Bridas de Tuberías y accesorios bridados (Edición 2009).

   De la norma en mención se consideró la Tabla II-2-1.1 de la misma, donde se muestra las presiones que soporta los materiales del grupo 1.1 (ASTM a105) a cierta temperatura. De lo cual sirvió para determinar la clase de las bridas y accesorios que se emplearán en el proyecto.

B. API RP Std 14E: Prácticas recomendadas para el diseño e instalación de sistemas de tuberías en plataformas de producción Offshore (5ta Edición-1991).

   De la norma en mención se consideró las secciones Z 3 y 4. Las cuales brindaron la orientación para el desarrollo de los cálculos hidráulicos, el diseño de líneas, los criterios de selección de las válvulas y la selección de bridas y accesorios

C. ASME B31.1: Tuberías de Vapor y Sistemas de Potencia (Edición 2001).

   Se uso la norma para aplicar criterios de seleccionamiento de tuberías y accesorios.

D. ANSI/HI 12.1-12.6-2011

   Normativa del Instituto Hidráulico para seleccionar parámetros operativos de bomba.

2.1.3. Simbología teórica

A. Concentradora:

Se denomina Planta Concentradora a una planta de procesamiento de mineral de cobre que tiene como finalidad su procesamiento en varias etapas hasta obtener Concentrado de este metal. Este Concentrado es luego procesado en fundiciones o plantas químicas para obtener cobre en la forma de barras o lingotes.

B. Pulpa:

Las pulpas son mezclas de sólidos y líquidos en forma de suspensiones de tal forma que sus características y comportamiento se pueden estudiar, en lo que a minería se refiere, co- mo un fluido homogéneo. Es un fluido formado por la suspensión de uno o varios minerales en agua

C. Confiabilidad:

La Confiabilidad es la "capacidad de un ítem de desempeñar una función requerida, en condiciones establecidas durante un período de tiempo determinado". Es decir, que habremos logrado la Confiabilidad requerida cuando el "ítem" hace lo que queremos que haga y en el momento que queremos que lo haga

D. Disponibilidad:

Es la capacidad de un activo o componente para estar en un estado (arriba) para realizar una función requerida bajo condiciones dadas en un instante dado de tiempo o durante un determinado intervalo de tiempo, asumiendo que los recursos externos necesarios se han proporcionado.

E. Payback:

El Payback o plazo de recuperación es un criterio para evaluar inversiones que se define como el periodo de tiempo requerido para recuperar el capital inicial de una inversión. Es un método estático para la evaluación de inversiones.

F. VAN:

El valor actual neto (VAN) es un criterio de inversión que consiste en actualizar los cobros y pagos de un proyecto o inversión para conocer cuánto se va a ganar o perder con esa inversión. También se conoce como valor neto actual (VNA), valor actualizado neto o valor presente neto (VPN).

G. TIR:

La tasa interna de retorno (TIR) es la tasa de interés o rentabilidad que ofrece una inversión. Es decir, es el porcentaje de beneficio o pérdida que tendrá una inversión para las cantidades que no se han retirado del proyecto.

2.2. Descripción de las actividades desarrolladas

2.2.1. Etapas de las actividades

Cuadro: 1.1

FASE 1: DETERMINACIÓN DE MEJORA PARA INCREMENTO DE PARÁMETROS DE MANTENIMIENTO	**FASE 2:** SELECCIÓN DE EQUIPO PARA INCREMENTAR PARÁMETROS DE MANTENIMIENTO	**FASE 3:** SELECCIÓN DE BOMBA 700 MCR	**FASE 4:** DESARROLLO DE INGENIERÍA Y CONSTRUCCIÓN
Actividades: 1- Levantamiento de información en campo. 2- Definición de objetivos 3- Análisis de viabilidad del proyecto. 4- Análisis de Factibilidad del proyecto.	Actividades: 1- Selección de Equipo, en este caso una bomba Stand By. 2- Determinación de las características y condiciones operativas. 3- Análisis Económico 4- Análisis de Mantenimiento y Producción. 5- Desarrollo de Ingeniería Conceptual.	Actividades: 1- Selección de bomba 700 MCR. 2- Determinación de parámetros y puntos operativos.	Actividades: 1- Desarrollo de Ingeniería Básica y de Detalle. 2- Inversión económica. 3- Instalación de Bomba

Fuente: Elaboración Propia

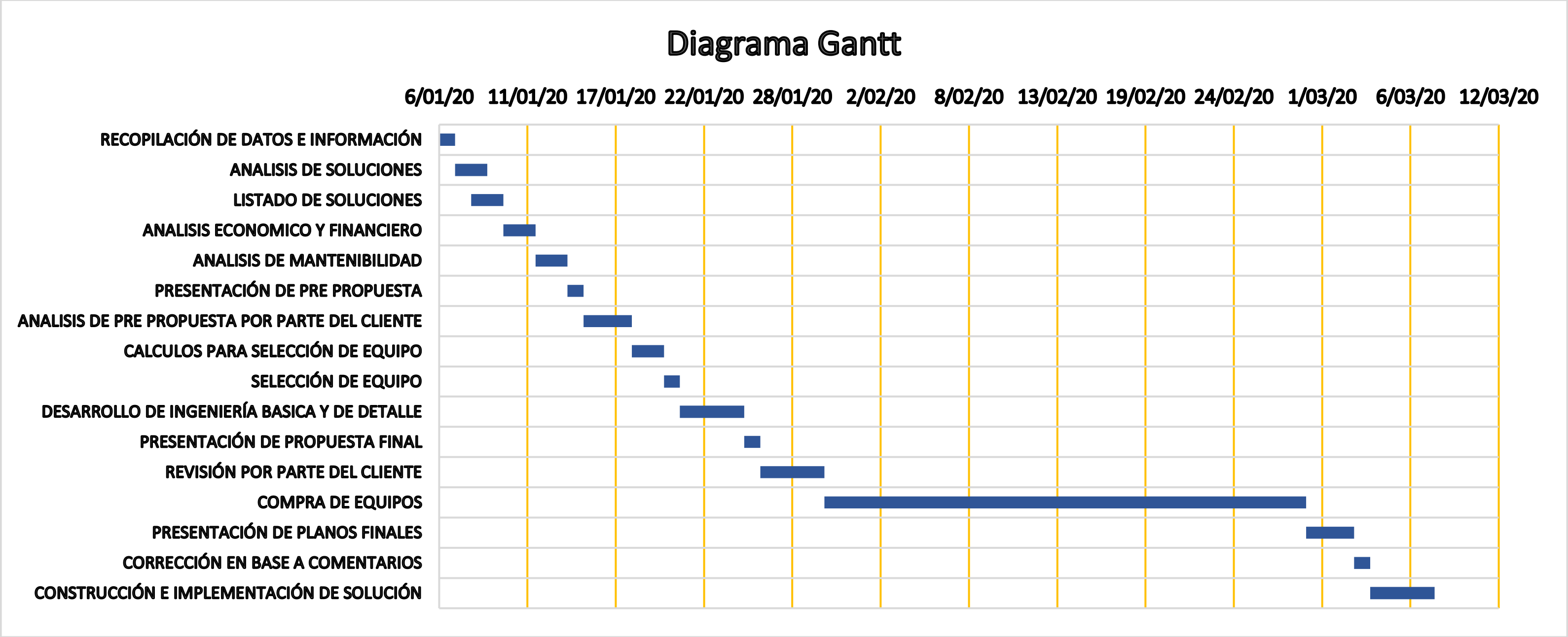
2.2.2. Diagrama de flujo

Cuadro: 1.2

	FECHA NICIO	DURACIÓN EN DÍAS	FECHA FIN
RECOPILACIÓN DE DATOS E INFORMACIÓN	6/01/2020	1	7/01/2020
ANALISIS DE SOLUCIONES	7/01/2020	2	9/01/2020
LISTADO DE SOLUCIONES	8/01/2020	2	10/01/2020
ANALISIS ECONOMICO Y FINANCIERO	10/01/2020	2	12/01/2020
ANALISIS DE MANTENIBILIDAD	12/01/2020	2	14/01/2020
PRESENTACIÓN DE PRE-PROPUESTA	14/01/2020	1	15/01/2020
ANALISIS DE PRE-PROPUESTA POR PARTE DEL CLIENTE	15/01/2020	3	18/01/2020
CALCULOS PARA SELECCIÓN DE EQUIPO	18/01/2020	2	20/01/2020
SELECCIÓN DE EQUIPO	20/01/2020	1	21/01/2020
DESARROLLO DE INGENIERÍA BASICA Y DE DETALLE	21/01/2020	4	25/01/2020
PRESENTACIÓN DE PROPUESTA FINAL	25/01/2020	1	26/01/2020
REVISIÓN POR PARTE DEL CLIENTE	26/01/2020	4	30/01/2020
COMPRA DE EQUIPOS	30/01/2020	30	29/02/2020
PRESENTACIÓN DE PLANOS FINALES	29/02/2020	3	3/03/2020
CORRECCIÓN EN BASE A COMENTARIOS	3/03/2020	1	4/03/2020
CONSTRUCCIÓN E IMPLEMENTACIÓN DE SOLUCIÓN	4/03/2020	4	8/03/2020

2.2.3. Cronograma de actividades

Figura 2.6



Fuente: Elaboración Propia

III. APORTES REALIZADOS

3.1. Cálculos Realizados

A. Cálculos para el seleccionamiento de bomba:

- ✓ Datos de entrada:

Flujo mínimo: 6000.00 m^3 / h

Flujo nominal: 6500.00 m^3 / h

Flujo de diseño: 7500.00 m^3 / h

Propiedades del Fluido:

Densidad del Fluido:	1350	kg/m^3	
Conc. de la pulpa (en peso) C_W:	55.00%		
Conc. de la pulpa (en Volumen) C_V:	39.25%		
Tamaño de partículas (d50):	1000	µm	
Viscosidad cinemática:	3.49	10^{-6} m^2/s	
Viscosidad dinámica:	0.0047	Pa.s	(4.7115 centipoise)
Factor de espuma en succión:	1		
Factor de espuma en descarga:	1		
pH:	10,5 - 11,5		
Temperatura de bombeo:	15	°C	

- ✓ Balance:

BALANCE	Flujo másico (TMH)	Caudal (m^3/h)	Densidad (Kg/m^3)
Agua	3948.8	3948.8	1000.0
Solidos	4826.3	2551.3	1891.7
Pulpa	8775.0	6500.0	1350.0

Condiciones del sitio:

Nivel de succión:	2800.000	m.s.n.m.
Nivel del fluido en tanque:	2807.000	m.s.n.m.
Nivel de descarga:	2812.000	m.s.n.m.

Presión atmosférica (P_{atm}):	7.35	m.c.a.	5.44 m.c.p.
Presión de vapor (p°v):	0.125	m.c.a.	0.09 m.c.p.
Zona sísmica:	Zona 02		
Altura estática (H)	12	m	
Altura de succión (Hs)	7	m	

- ✓ Tabla de selección de tuberías:

Seleccionamos 3 Opciones de Tuberías:

Tabla 3.1

	Material Tubería	Diámetro Nominal	Schedule / SDR	Diámetro Interno DI	Espesor Revest.	Material Revest.	Diámetro Interno D_i'
		(Pulg.)		(Pulg.)	(Pulg.)	(Pulg.)	(Pulg.)
Op. N° 1	ACERO	32	40	30.624	0.375	ACERO	29.874
Op.N° 2	ACERO	34	40	32.624	0.375	ACERO	31.874
Op. N° 3	ACERO	36	40	34.500	0.375	ACERO	33.750

Fuente: Elaboración Propia

- ✓ Cálculo del diámetro de tubería

El cálculo de la velocidad crítica de sedimentación VD, se realiza mediante la siguiente formula:

$$\boxed{V_D = F'_L \left(\frac{d_{50}}{D_i}\right)^{\frac{1}{6}} \sqrt{2g\left(\frac{D_i}{1000}\right)\left(\frac{\rho_S}{\rho_L}-1\right)}}$$

$$\boxed{F'_L = 3.32\left(\frac{Cv}{100}\right)^{0.213}} \qquad = 2.72$$

Donde, F'_L es el factor de corrección.

La velocidad del fluido (V) deberá ser como mínimo 10% mayor a la velocidad de sedimentación (VD) (Se considera 10% como buena práctica).

El cálculo de pérdidas de fricción por unidad de longitud (J), se realiza con la fórmula de Darcy:

$$\boxed{h_f = J.L}$$

Donde:

L: Longitud total de tuberías

F: Factor de fricción

$$\boxed{J = f.\left(\frac{V^2}{2.g.D_i}\right)}$$

El cálculo del factor de fricción (f) se realiza mediante la fórmula de Colebrook:

$$\boxed{\frac{1}{\sqrt{f}} = -2.\log\left(\frac{\varepsilon}{3.7.d} + \frac{2.51}{\mathrm{Re}.\sqrt{f}}\right)}$$

- ✓ Tabla de cálculo de selección de diámetros de tuberías:

Recordamos que la velocidad máxima al interior de una tubería es de 4m/s (recomendación de buena práctica).

Tabla 3.2

	Flujo	Diámetro	V	V_D	Re	f	J	
		Pulg.	m/s	m/s			(m/m)	
Op. N° 1	Mínimo		3.686	2.510	801321	0.0132	0.0121	ok
	Nominal	32	3.993	2.510	868097	0.0131	0.0140	ok
	Máximo		4.607	2.510	1001651	0.0130	0.0186	no cumple
Op. N° 2	Mínimo		3.238	2.565	751040	0.0133	0.0088	ok
	Nominal	34	3.507	2.565	813626	0.0131	0.0102	ok
	Máximo		4.0470	2.5646	938799.7	0.0131	0.0135	no cumple
Op. N° 3	Mínimo		2.888	2.614	709293	0.0133	0.0066	ok
	Nominal	36	3.128	2.614	768401	0.0132	0.0077	ok
	Máximo		3.610	2.614	886617	0.0131	0.0102	ok

Fuente: Elaboración propia

*De los resultados anteriores, concluimos que la tubería recomendada, es la de 36", ya que cumple con los valores máximos admisibles para las velocidades en las tuberías y, la velocidad mínima para evitar sedimentación.

- ✓ Cálculo de la altura dinámica total del sistema

  Cálculo de la pérdida de carga total en la succión de la bomba (Hs)

Tabla: 3.3

Ítem	Flujo (m^3/h)	Velocidad (m/s)	J (m/m)	Cant. de accesorios	Long. de tubería (m)	Pérdida de carga (m)
Tubería	6000.0	3.238	0.0088		4	0.04
Tubería	6500.0	3.507	0.0102		4	0.04
Tubería	7500.0	3.683	0.0112		4	0.04
Salida de Tanque	6000.0	3.24		1		1.10
Salida de Tanque	6500.0	3.51		1		1.17
Salida de Tanque	7500.0	3.68		1		1.17
Válvula Cuchilla	6000.0	3.24		1		1.31
Válvula Cuchilla	6500.0	3.51		1		1.32
Válvula Cuchilla	7500.00	3.68		1		1.32
Curva 3D	6000.0	3.24		1		1.91
Curva 3D	6500.0	3.51		1		1.92
Curva 3D	7500.0	3.68		1		1.92
Tee	6000.0	3.24		0		0.00

Tee	6500.0	3.51		0		0.00
Tee	7500.0	3.68		0		0.00
Reducción	6000.0	3.24		0		0.00
Reducción	6500.0	3.51		0		0.00
Reducción	7500.0	3.68		0		0.00

Fuente: Elaboración propia

		F. Mínimo	F. Nominal	F. Máximo	
Pérdida de carga sin F.S.:		4.346	4.437	4.447	m.c.p.
Pérdida de carga con F.S.	10%	4.781	4.880	4.891	m.c.p.

- ✓ Cálculo de la Pérdida de carga total en la descarga de la bomba (Hd)

Tabla 3.4

Ítem	Flujo (m^3/h)	Velocidad (m/s)	J (m/m)	Cant. de accesorios	Long. de tubería (m)	Pérdida de carga (m)
Tubería	6000.0	3.238	0.0088		36	0.32
Tubería	6500.0	3.507	0.0102		36	0.37
Tubería	6825.0	3.683	0.0112		36	0.40
Válvula Cuchilla	6000.0	3.24		1		10.27
Válvula Cuchilla	6500.0	3.51		1		10.43
Válvula Cuchilla	6825.0	3.68		1		10.43
Válvula Check	6000.0	3.24		0		0.00
Válvula Check	6500.0	3.51		0		0.00
Válvula Check	6825.0	3.68		0		0.00
Curva 5D	6000.0	3.24		5		14.93
Curva 5D	6500.0	3.51		5		14.87

Curva 5D	6825.0	3.68		5		14.87
Reducción	6000.0	3.24		0		0.00
Reducción	6500.0	3.51		0		0.00
Reducción	6825.0	3.68		0		0.00

Fuente: Elaboración Propia

		F. Mínimo	F. Nominal	F. Máximo	
Pérdida de carga sin F.S.:		3.93	4.22	4.33	m.c.p.
Pérdida de carga con F.S.:	10%	4.32	4.64	4.76	m.c.p.

- ✓ El ADT, se calcula mediante la siguiente fórmula:

$$\boxed{ADT = H + h_s + h_d - Hs + P_{salida}}$$

Considerando una presión de salida de 14.5 psi (presión requerida por los ciclones):

	F. Mínimo	F. Nominal	F. Máximo	
ADT	30.7	31.1	31.2	m.c.p.
	41.4	41.9	42.1	m.c.a.
	58.9	59.7	59.9	PSI
	4.1	4.1	4.1	Bar

- ✓ Cálculo de la altura neta positiva de succión disponible de la bomba

$$\boxed{NPSH_d = P_{atm} - p^{\circ}v - hs + Hs}$$

	F. Mínimo	F. Nominal	F. Máximo	
$NPSH_d$	7.57	7.47	7.46	m.c.p.
	10.22	10.09	10.07	m.c.a.

- ✓ Cálculo de las potencias del sistema

$$\boxed{HP_{HIDRAULICO} = \frac{9.81\ .\rho(kg/m^3).Q(m^3/s).ADT\ (m)}{3600\ .(0.746\ )}}$$

	F. Mínimo	F. Nominal	F. Máximo	
HP Hidráulico	907.0	996.0	1154.0	HP

BRAKE HORSE POWER (BHP)

$$\boxed{BHP = \frac{HP_{HIDRAULICO}}{n_{BOMBA}}}$$

Donde: n_{BOMBA}: 60% Mínimo

n_{BOMBA}: 65.0% Nominal

n_{BOMBA}: 70.0% Diseño

	F. Mínimo	F. Nominal	F. Máximo	
BHP	1511.696	1532.2	1648.6	HP

Considerando factores de corrección por modelo de bomba:

H.R. = 0.90

E.R. = 0.87

	F. Mínimo	F. Nominal	F. Máximo	
$P_{CORREGIDA}$	1930.6	1956.9	2105.5	HP

Considerando factor de seguridad sobre potencia hidráulica:

F.S. = 1.15 (recomendación de buena práctica)

	F. Mínimo	F. Nominal	F. Máximo	
$P_{INCLUIDO\ F.S.}$	2220.2	2250.4	2421.3	HP

- ✓ Potencia del motor (hp)

$$\boxed{P_{MOTOR} = \frac{BHP}{n_{MOTOR}}}$$

Donde:

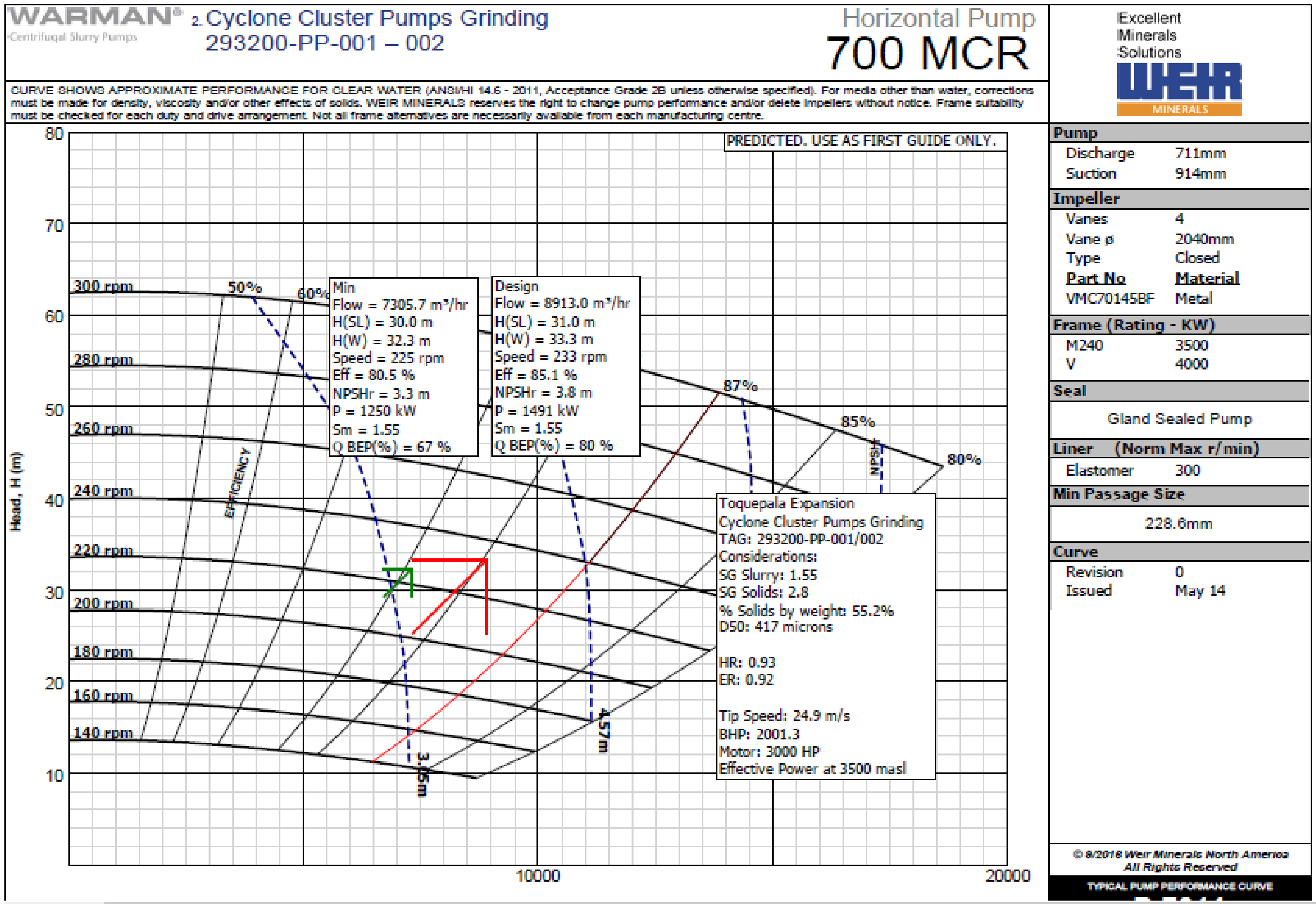
n_{MOTOR}: 92%

	F. Mínimo	F. Nominal	F. Máximo	
P_{MOTOR}	2413.3	2446.1	2631.9	HP

POTENCIA NOMINAL ESTÁNDAR	3000	HP
	2238	kW

La bomba seleccionada para las condiciones anteriores será:

Figura 3.1

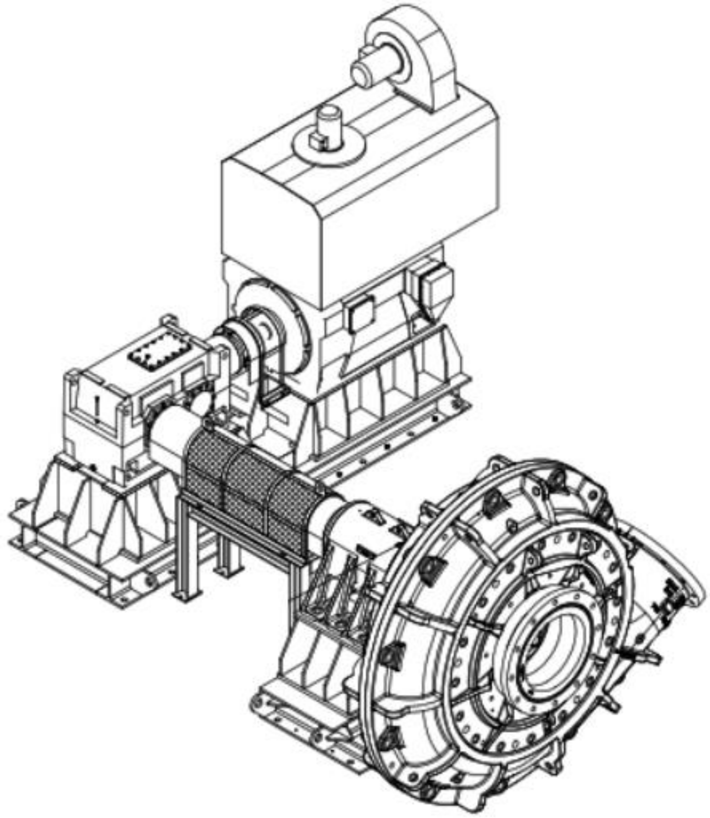


Fuente: Selector Pro – Software created by weir minerals

La cual se alinea a las condiciones operativas actuales y, posee una capacidad mayor para dar holgura a futuras condiciones operativas.

El esquema inicial de la bomba sería el siguiente:

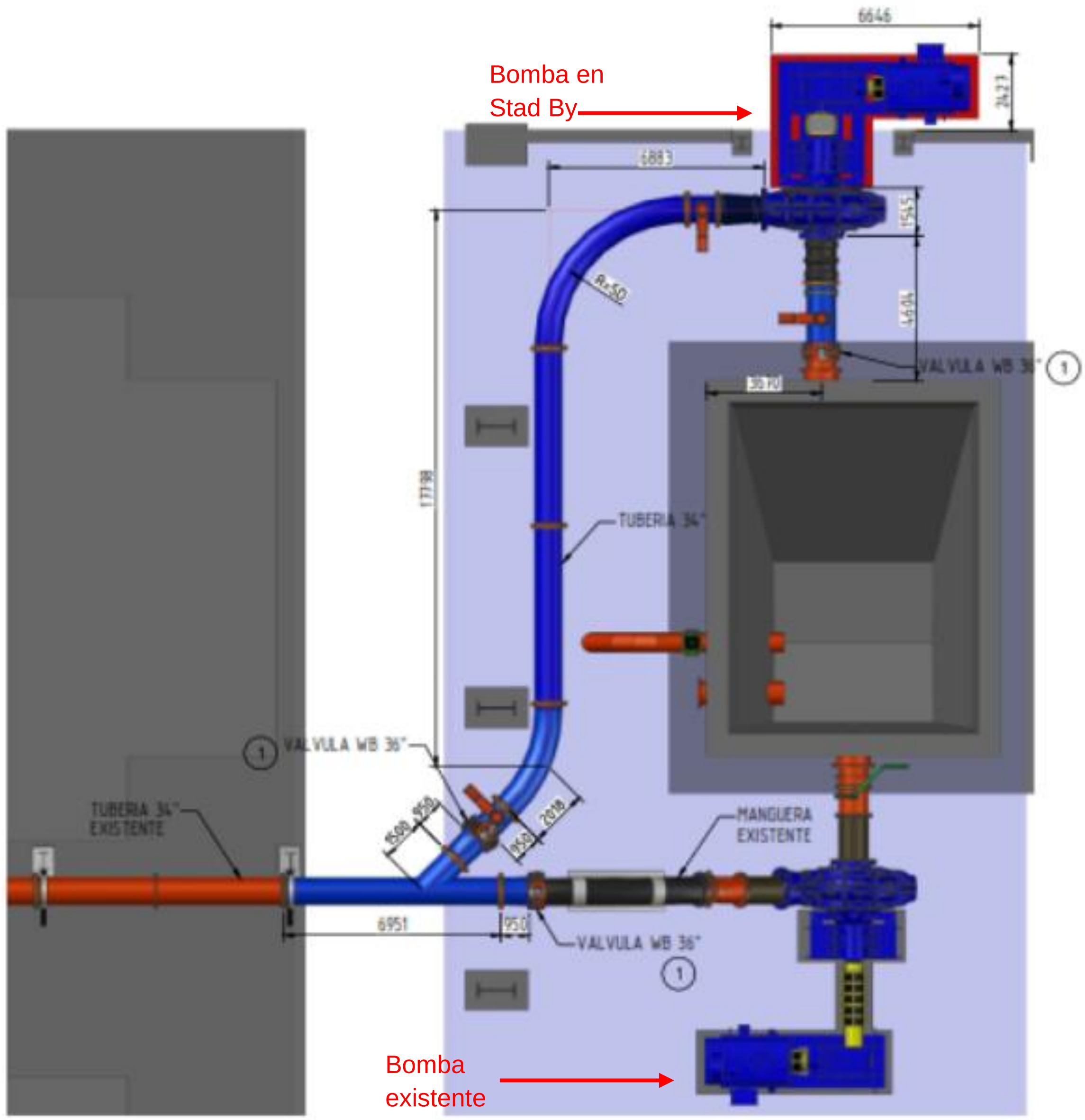
Figura 3.2



Fuente: Plano de Arreglo General – D-VP-1X-1749

El arreglo general tentativo de la bomba 700MCR instalada en Stand By a la bomba ya existente, sería el siguiente:

Figura 3.3



Fuente: Elaboración Propia

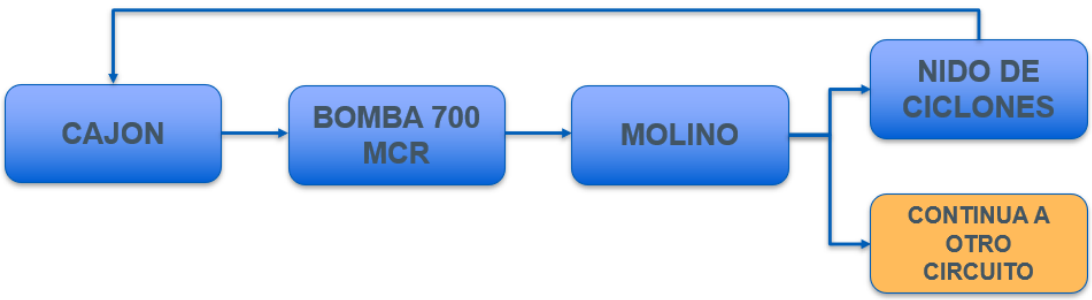
B. Análisis de confiabilidad

Data obtenida del cliente para determinar la confiabilidad de los equipos:

- Confiabilidad Conjunto Bomba – Motor: 0.98
- Confiabilidad Molino: 0.96
- Confiabilidad Nido de Ciclones: 0.97

Esquema de Circuito de Molienda Actual:

Figura: 3.4



Fuente: Elaboración propia

Realizando el análisis por diagrama de bloques:

Tenemos que el circuito se encuentra en serie, por lo tanto, con los valores de confiabilidad obtenidos del cliente tenemos que la confiabilidad del circuito es:

$$R_s(t) = \prod_{i=1}^{n} R_i(t) \qquad \text{Confiabilidad del Sistema}$$

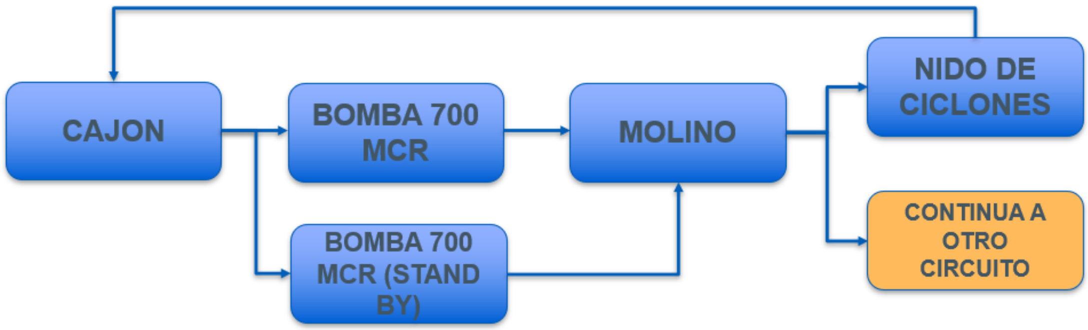
$$R_x(t) = R_{Bomba} * R_{Nido\ ciclones} * R_{Molino}$$

$$R_x(t) = 0.98 * 0.96 * 0.97$$

$$Confiabilidad\ actual\ del\ circuito = 0.912576$$

Colocando la bomba 700MCR en Stand By, tenemos el siguiente esquema:

Figura 3.5



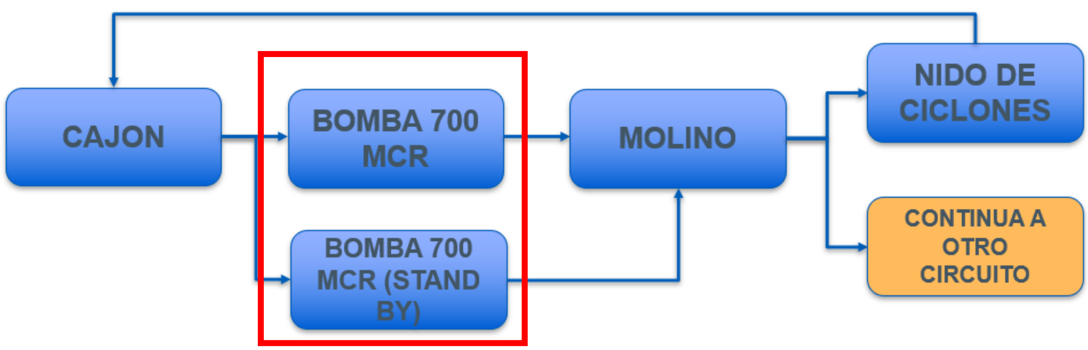
Fuente: Elaboración propia

Analizamos el nuevo circuito por bloques:

Paso 1:

Analizamos el bloque de dos bombas (una en operación y otra en stand by):

Figura 3.6



Fuente: Elaboración propia

$$R_p(t) = 1 - \prod_{i=1}^{n} (1 - R_i(t))$$

$$R_x(t) = 1 - [(1 - R_{Bomba}) * (1 - R_{Bomba})$$

$$R_x(t) = 1 - [(1 - 0.98) * (1 - 0.98)]$$

$$R_x(t) = 0.9996$$

Paso 2:

Analizamos todo el conjunto:

$$R_s(t) = \prod_{i=1}^{n} R_i(t) \qquad \text{Confiabilidad del Sistema}$$

$$R_{x\ nueva}(t) = R_{Ambas\ bombas} * \ R_{Nido\ ciclones} * \ R_{Molino}$$

$$R_{x\ nueva}(t) = 0.9996 * \ 0.96 * \ 0.97$$

$$R_{x\ nueva}(t) = 0.93082752$$

Como resultado Obtenemos que la confiabilidad aumenta en:

$$AUMENTO\ DE\ CONFIABILIDAD = \ R_{x\ nueva}(t) - \ R_x(t)$$

$$AUMENTO\ DE\ CONFIABILIDAD = \ 0.93082752 - \ 0.912576$$

$$AUMENTO\ DE\ CONFIABILIDAD = \ 0.01825152$$

C. Análisis de Disponibilidad

Tenemos los siguientes mantenimientos programados:

Tabla 3.5

ACTIVIDADES DE MANTENIMIENTO PROGRAMADOS		
FRECUENCIA DE MANTENIMIENTO MECANICO WET END	120	Días
TIEMPO DE MANTENIMIENTO MECANICO WET END	12	Horas
FRECUENCIA DE MANTTO MECANICO DISCO SUCCIÓN E IMPULSOR	60	Días
TIEMPO DE MANTENIMIENTO MECANICO DISCO SUCCIÓN E IMPULSOR	8	Horas
FRECUENCIA DE MANTTO ELECTRICO	180.00	Días

TIEMPO DE MANTENIMIENTO ELECTRICO	12.00	Horas

Fuente: Elaboración propia, datos obtenidos de la unidad minera Toquepala

Para calcular la disponibilidad, usamos la siguiente formula:

$$\text{Disponibilidad} = \frac{\text{Horas totales} - \text{Horas parada por mantenimiento}}{\text{Horas totales}}$$

Para calcular las horas totales, sabemos que las horas disponibles que debe de tener el circuito son 24h al día, los días hábiles son 365, considerando 2 paradas de planta programadas de 2 días cada una por año.

Horas totales = 24*365= 8760 horas

Para calcular las horas de parada por mantenimiento, sumamos las horas mostradas en la tabla 1.5.

Horas por parada mantenimiento = 3*12+8*3+12*2+2*24 = 132 horas

Entonces:

$$\text{Disponibilidad actual} = \frac{8760 - 132}{8760} = 0.9849$$

Teniendo la bomba en Stand By, el circuito no pararía por mantenimiento mecánico, por lo tanto, las horas de parada solo se darían en parada de planta programadas.

Las nuevas horas en que parará el circuito por parada de planta programada será:

24*2=48 horas

Entonces:

$$\text{Disponibilidad nueva} = \frac{8760 - 48}{8760} = 0.9945$$

Tenemos un incremento de disponibilidad en: 0.0096

3.2. Evaluación técnica – económico

A. Inversión necesaria

Tabla 3.6

INGENIERIA Y SUMINISTRO	USD.
INGENIERIA WEIR (Mecánica)	0
Bomba 700 MCR	560,000

Motor 3000 HP TOSHIBA	246,986
Sub-Base Bomba	52,800
Reductor	250,229
Spools & auxiliares	161,949
Válvulas	126,309
Soporteria	12,715
Juntas	26,693
Total de Equipos	**1,437,681**
CONSTRUCCIÓN	**USD.**
Construcción Civil	114,500
Montaje mecánico	80,282
Eléctrico/Instrumentación	180,000
Start Up & Comissioning	10,000
Total de Construcción	**384,782**
TOTAL INVERSIÓN	**1,822,463 USD**

Fuente: Elaboración propia

B. ANALISIS VAN

Para calcular el VAN, necesitamos conocer el incremento de ingresos en dolares (USD) por las paradas del circuito evitadas por la Bomba Stand By.

Entonces:

Como datos de proceso tenemos lo siguiente:

Tabla 3.7

DATOS			
Caudal por hora	6000	m3/h	Utilizamos el menor valor del caudal (peor escenario)
Cw:	55.20%		
SGsol:	2.8		
SGslu:	1.55		

Precio cobre aprox:	2.57 Referencia a la fecha enero 2020	USD/lb
Densidad cobre:	8,960	kg/m3
Ley de la mina:	0.65%	
Cv	30.5571%	

Fuente: Elaboración propia, con datos obtenidos de la unidad minera Toquepala

Con estos datos, obtenemos los siguientes valores:

Tabla 3.8

COSTO POR HORA DE LA BOMBA		
Flujo de solidos	1833.428571	m3/h
Flujo de cobre	11.91728571	m3/h
Flujo en peso de cobre	106778.88	kg/h
Peso en libras	48535.85455	Lb/h
Producción por hora en dólares	124,737.15	USD/h
Factor de Costo de producción	0.4101526	
COSTO DE PRODUCCIÓN POR HORA	$51,161.26 USD	
RENTABILIDAD NETA POR HORA	$73,575.88 USD	

Fuente: Elaboración propia

Tenemos los siguientes tipos de mantenimiento programados, los que usaremos para analizar cuanto ganariamos si, en estos mantenimientos no es necesario detener todo el circuito de molienda:

Tabla 3.9

MANTENIMIENTOS PROGRAMADOS		
FRECUENCIA DE MANTTO MECANICO WET END	120	Días
TIEMPO DE MANTENIMIENTO MECANICO WET END	12	Horas
FRECUENCIA DE MANTTO MECANICO DISCO SUCCIÓN E IMPULSOR	60	Días

TIEMPO DE MANTENIMIENTO MECANICO DISCO SUCCIÓN E IMPULSOR	8	Horas
FRECUENCIA DE MANTTO ELECTRICO	180.00	Días
TIEMPO DE MANTENIMIENTO ELECTRICO	12.00	Horas

Fuente: Elaboración propia, datos obtenidos de la unidad minera Toquepala

Con estos valores, de mantenimiento, y conociendo la utilidad neta del circuito por hora, obtenemos los siguientes valores, simplificando y aproximando el aumento de la utilidad por cada tipo de mantenimiento:

Tabla 3.10

TIEMPO DE MANTENIMIENTO POR CAMBIO WET END EN BOMBA		
Frecuencia	120	Días
Tiempo parado	12	Horas
Dinero ganado cada 120 Días	882,910.58 USD	
TIEMPO DE MANTENIMIENTO POR CAMBIO DE DISCO SUCCIÖN E IMPULSOR		
Frecuencia	60	Días
Tiempo parado	8	Horas
Dinero ganado cada 60 Días	588,607.05 USD	

Fuente: Elaboración propia

Ahora, que tenemos los valores incremento de producción y utilidad neta, podemos realizar una corrida de ingresos netos:

MES	INVERSIÓN	1 mes	2 mes	3 mes	4 mes	5 mes	6 mes
Aumento de producción en dinero	$1,822,463	$0.00	$588,607	$0.00	$882,910	$0.00	$588,607

MES	INVERSIÓN	7 mes	8 mes	9 mes	10 mes	11 mes	12 mes

Aumento de producción en dinero	$1,822,463	$0.00	$882,910	$0.00	$588,607	$0.00	$882,910

Con el cuadro anterior, podemos calcular el VAN a 1 año:

Tomando una Tasa de 8%, que es la tasa que Southern Perú Copper Corporation posee, cuando tiene el dinero en el banco:

$$\mathrm{VAN} = -\mathrm{I} + \frac{F_1}{(1+i)^1} + \frac{F_2}{(1+i)^2} + \ldots$$

$$\mathrm{VAN} = -1822463 + \frac{0}{(1+0.08)^1} + \frac{588607.05}{(1+0.08)^2} + \ldots + \frac{882910.58}{(1+0.08)^{12}}$$

$$\mathrm{VAN} = 802{,}324.59\ \mathrm{USD}$$

C. ANALISIS TIR

Para analizar la TIR, igualamos el VAN para 1 año igual a 0:

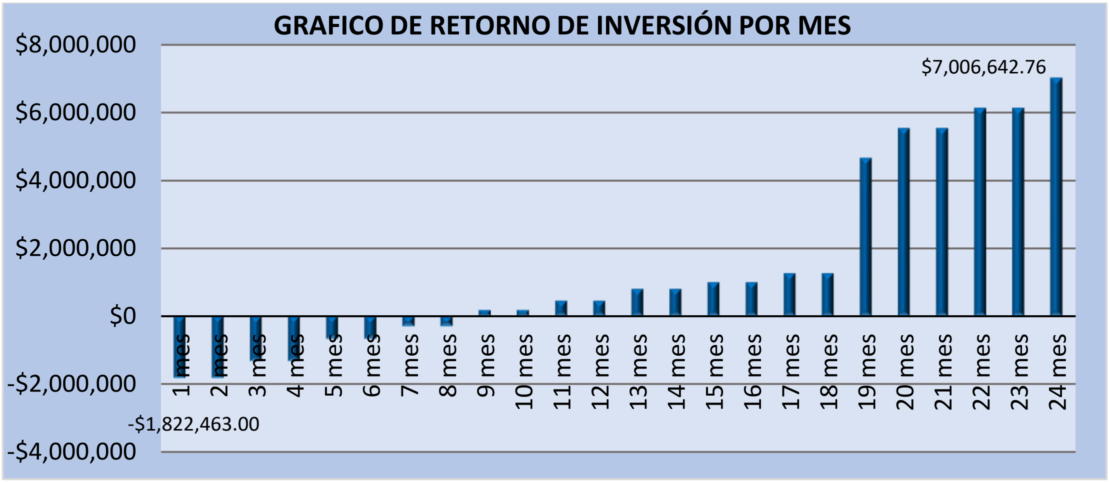
$$\mathrm{TIR} = \sum_{T=0}^{n} \frac{Fn}{(1+\mathrm{i})^n} = 0$$

$$\mathrm{TIR} = \frac{0}{(1+\mathrm{i})^1} + \frac{588607.05}{(1+i)^2} + \ldots.$$

TIR = 13.03%

De esta manera obtenemos que el TIR es igual a 13.03%.

Figura 3.7



Fuente: Elaboración propia

3.3. Análisis de resultados

- ✓ Analizando los valores de confiabilidad y disponibilidad brindados por el cliente, nos damos cuenta que podemos aumentar la confiabilidad y disponibilidad del circuito de molienda colocando una Bomba Stand By, logrando un incremento en estos parámetros ya que la línea no se detendrá por mantenimiento mecánico no programado, porque tenemos un incremento de la disponibilidad del sistema de bombeo del circuito de Molienda en un 0.0096 y la confiabilidad del circuito en 0.01825.
- ✓ Para lograr que el circuito de molienda tenga un aumento de confiabilidad y disponibilidad considerable, hemos seleccionado una bomba marca Warman, modelo 700 MCR la cual cumple con los parámetros operativos que el cliente requiere que son de aproximadamente 7000 m3/h.
- ✓ Colocando una bomba adicional en Stan By con los parámetros operativos requeridos por operaciones buscando aumentar la confiabilidad y disponibilidad del circuito, logramos tener un incremento de ingresos anuales de $ 4,414,551.00, por lo que obtenemos un VAN de 802,324.59 USD y un TIR de 13.03%.

IV. DISCUSION Y CONCLUSIONES

4.1. Discusión

Colocando una bomba Stand By en el circuito de Molienda, mejoramos los parámetros de mantenimiento de disponibilidad y confiabilidad, logrando así un aumento de producción y de ingresos especialmente porque no hay perdidas por lucro cesante.

4.2. Conclusión

4.2.1. Conclusión General

- ✓ Con la instalación de una bomba 700 MCR en Stand By, se logra aumentar la disponibilidad del circuito de Molienda de la Concentradora 2 de la mina Toquepala en un 0.0096.

4.2.2. Conclusiones Especificas

- ✓ Evaluando los Indicadores de Mantenimiento del circuito de Molienda de la Concentradora 2 de la mina Toquepala, vemos que existe una falencia en equipos, lo que ocasiona que exista perdida de disponibilidad en el circuito, por lo que concluimos que la manera de mejorar este indicador, es colocando una bomba Stand By para dicho circuito.
- ✓ Realizando una correcta Ingeniería básica, conceptual y de detalle, concluimos que la bomba que mejor se desempeñara bajo las condiciones operativas mostradas, es una bomba 700 MCR.
- ✓ Concluimos que, teniendo una bomba Stand By, ya no será necesario detener el circuito de molienda por temas de mantenimiento, y por ende, no existirá perdidas por lucro cesante, generándonos una VAN a 1 año de $802{,}324.59$ USD y un TIR de 13.03%. En el tiempo estimado de vida útil de 10 años, donde se prevé que esta bomba opere sin ningún inconveniente, obtendremos una utilidad neta de 44,145,510.00. USD.

V. RECOMENDACIONES

- Se recomienda utilizar repuestos originales, ya que estos han diseñados específicamente para las condiciones operativas solicitadas por la compañía minera Southern Perú Copper Corporation.
- Se recomienda mantener las condiciones operativas dentro del rango indicado en los cálculos y datos de entrada, ya que si superamos el caudal impulsado, superaríamos la velocidad máxima de transporte permitido siendo esta 4m/s.
- Si se desea aumentar el caudal, se recomienda realizar un nuevo calculo para modificar la tubería de descarga y succión.
- Se recomienda realizar el cambio de disco succión e impulsor, cada 60 días de operación o su equivalente en horas cuando opera al 100% de su capacidad.
- Se recomienda realizar un cambio de Wed End cada 120 días de operación o su equivalente en horas cuando opera al 100% de su capacidad.
- Se recomienda usar un impulsor en Alto Cromo A05, ya que este material es el más resistente al desgaste generado por la pulpa bajo molino.

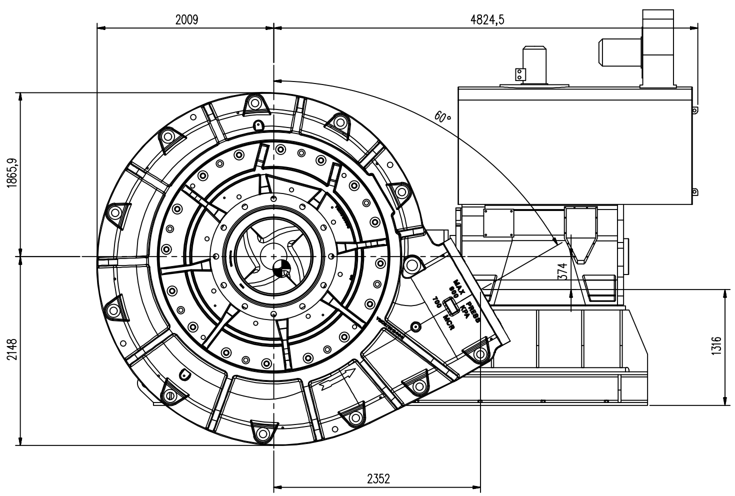
VI. BIBLIOGRAFIA

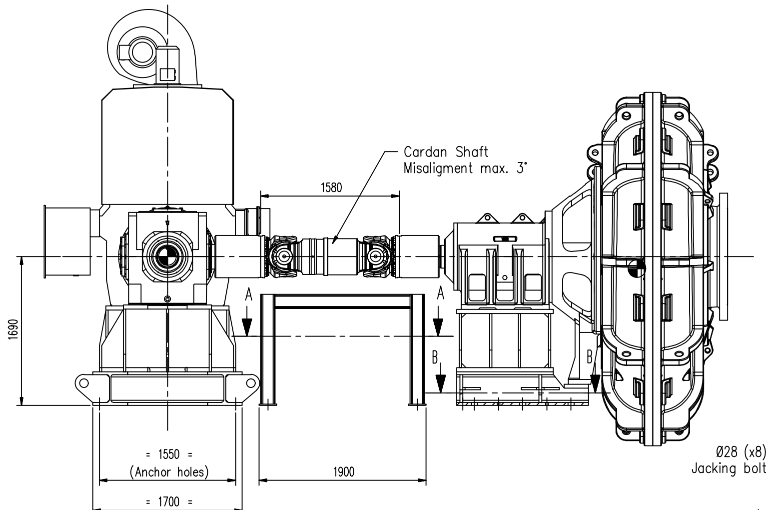
- MANUAL PARA REDACTAR CITAS BIBLIOGRAFICAS Según norma ISO 690 y 690-2 (International Organization for Standardization)
- La norma ISO 690:2010(E)
- The American Society of Mechanical Engineers - ASME (2018). ASME B31.3 Code for Pressure Piping. New York, USA.
- American Petroleum Institute – API (2010). API 610 Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries. Washington, USA.
- ANSI HI 12.1-12.6-2011 Rotodynamic Centrifugal Slurry Pumps
- TQ-IOM-S-WM – Manual de Operación y Mantenimiento Bomba 700 MCR
- MANUEL POLO. Turbomáquinas hidráulicas, México DF. 1983.
- The American Society of Mechanical Engineers - ASME (2016). ASME B31.4 Code for Pipeline Transportation Systems for Liquids and Slurries. New York, USA.

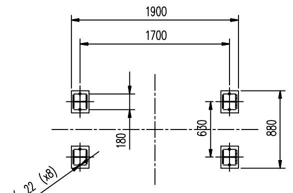
ANEXOS

Anexo 1

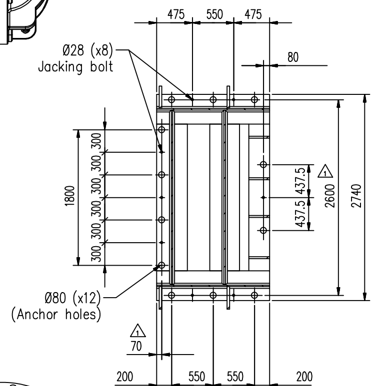
PLANO DE ARREGLO GENERAL



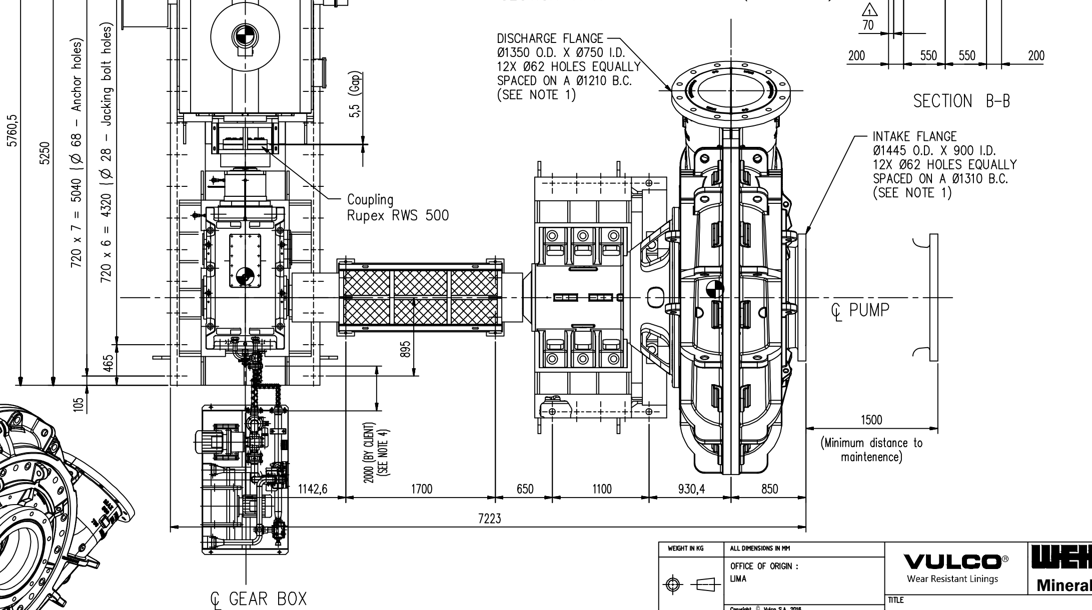




SECTION A-A



SECTION B-B



CERTIFIED PUMP	
CUSTOMER:	SOUTHERN PERU
PROJECT:	COMPLEMENTO AMPLIACION TOQUEPALA A 120,000 TMPD
EQUIPMENT NAME:	SISTEMA DE BOMBEO MOLINO DE BOLAS
PROJECT NUMBER:	PAT-CT-001/M3-PN150141
PO:	150141-P005
QUANTITY:	2
SERIAL NUMBER:	VP-2016-160 / VP-2016-161
PUMP:	700 MCR-M240
BRAND:	WARMAN
TAG:	293200-PP-001, 293200-PP-002
LINER / IMPELLER:	NATURAL RUBBER / HIGH CHROME ALLOY
PUMP SPEED (Nominal / Design):	225/ / 233 RPM
MAX. CASING PRESSURE:	130 psi (900 kPa)
NPSHr (m):	3.8
FLOW CAPACITY (min/max):	7309,7 - 8913 m^2/h
DISCHARGE PRESSURE (kPa):	550 - 580
EFFICIENCY (Nominal/Design):	80,5 - 85,1
MOTOR:	SIEMENS
MODEL MOTOR:	SQUIRREL CAGE
HP:	3000
V/PH/Hz:	4160/3/60
ALTITUDE:	3500 masl
FRAME MOTOR:	1RQ65604JJ30
RPM:	1791
ENCLOSURE:	TEAAC
INSULATION CLASS:	F
SERVICE FACTOR	1,15
MOUNTING	HORIZONTAL
PUMP WEIGHT (kg):	45200
MOTOR WEIGHT (kg):	8800
COUPLING WEIGHT (kg):	217
BASE WEIGHT (kg):	13900
GEAR BOX (kg):	6150
CARDAN SHAFT (kg):	2392
TOTAL WEIGHT (kg):	76660
REQUIREMENT FOR SEAL WATER	
FLOW (m3/h):	4.5 - 9 m3/h
PRESSURE:	650 - 760 kPa
CONNECTION:	1 1/4" NPT
CERTIFIED FOR:	ERLIN ZELADA
SIGNATURE:	
DATE:	NOVEMBER 2016

INTAKE ALLOWABLE FLANGE LOADS			
Axial load	Fy	N	89000
Force in x-z	Fxz	N	97900
Torsional moment	My	Nm	50900
Bending moment	Mxz	Nm	101700
DISCHARGE ALLOWABLE FLANGE LOADS			
Axial load	Fz	N	62300
Force in x-y	Fxy	N	75600
Torsional moment	Mz	Nm	37300
Bending moment	Mxy	Nm	74600

GEAR BOX DATA	
Gearbox:	Siemens
Model:	B2SH18A
Ratio:	7,239 (iN=7.1)
S.F.	2,68
COUPLING DATA	
H.S. Coupling:	Siemens
Model:	Rupex RWS 500
S.F	2.10
CARDAN SHAFT DATA	
Cardan Shaft:	Voith
Model:	RTK2_490.8
B10 Life:	100,000 hrs
Misalignment capacity:	3 degrees
Lenght:	1580-1685 mm
S.F.	5.8

NOTES:

1) Find more details of Suction and Discharge Companion Flanges in drawing D-VP-1X-1749_CF

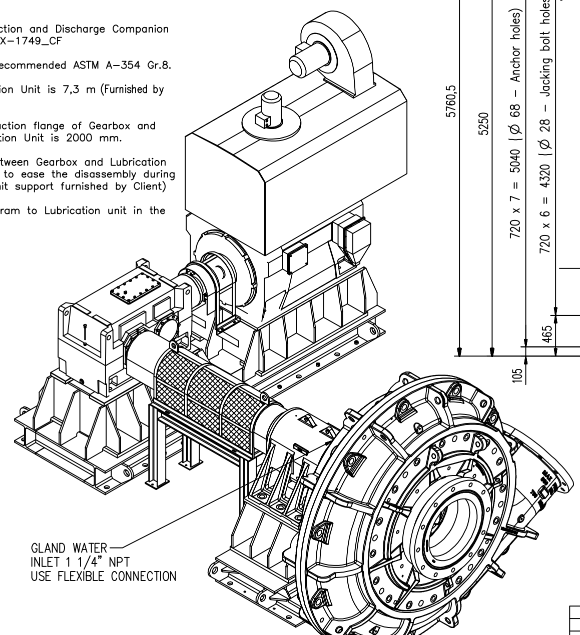
2) Anchor Bolts Material: Recommended ASTM A-354 Gr.8.

3) Hose Length of Lubrication Unit is 7,3 m (Furnished by Vulco)

4) The distance between suction flange of Gearbox and discharge flange of Lubrication Unit is 2000 mm.

5) The height difference between Gearbox and Lubrication Unit is 1140 mm, in order to ease the disassembly during maintenance (Lubrication unit support furnished by Client)

6) Find the anchorage diagram to Lubrication unit in the document TQ-PID-WM.



DOC #: TQ-GA-WM

REV	DESCRIPTION	DATE	REF NO	BY	CHK
0	DRAWING ISSUED TO CERTIFICATE	20/12/2016		RMA	EZL
D	INCLUDE COMMENTS MADE BY CLIENT	30/11/2016		RMA	EZL
C	INCLUDE COMMENTS MADE BY CLIENT	08/11/2016		RMA	EZL
1	ADDED DIMENSION	10/01/2017		RMA	EZL

WEIGHT IN KG
ALL DIMENSIONS IN MM
OFFICE OF ORIGIN : LIMA

SCALE: 1:2
APP: JAR
CHK: EZL
DRN: R. MANAYAY
DATE: 14/09/2016
SHEET 1 OF 1

VULCO® Wear Resistant Linings
WEIR Minerals

TITLE
GENERAL ARRANGEMENT
PUMP 700 MCR/4 ELEMENTS
MOUNTING 60 DEGREES

SHEET SIZE: A3
DRAWING No: D-VP-1X-1749
REVISION: 1

Anexo 2

NORMAS DEL INSITUTO HIDRAULICO

ANSI/HI 12.1–12.6-2011

Approved American National Standard

ANSI

American National Standard for

Rotodynamic (Centrifugal) Slurry Pumps

for Nomenclature, Definitions, Applications, and Operation

ANSI/HI 12.1–12.6-2011

Hydraulic Institute

6 Campus Drive
First Floor North
Parsippany, New Jersey
07054-4406
www.Pumps.org

ANSI/HI 12.1–12.6-2011

American National Standard for

Rotodynamic (Centrifugal) Slurry Pumps

for Nomenclature, Definitions, Applications, and Operation

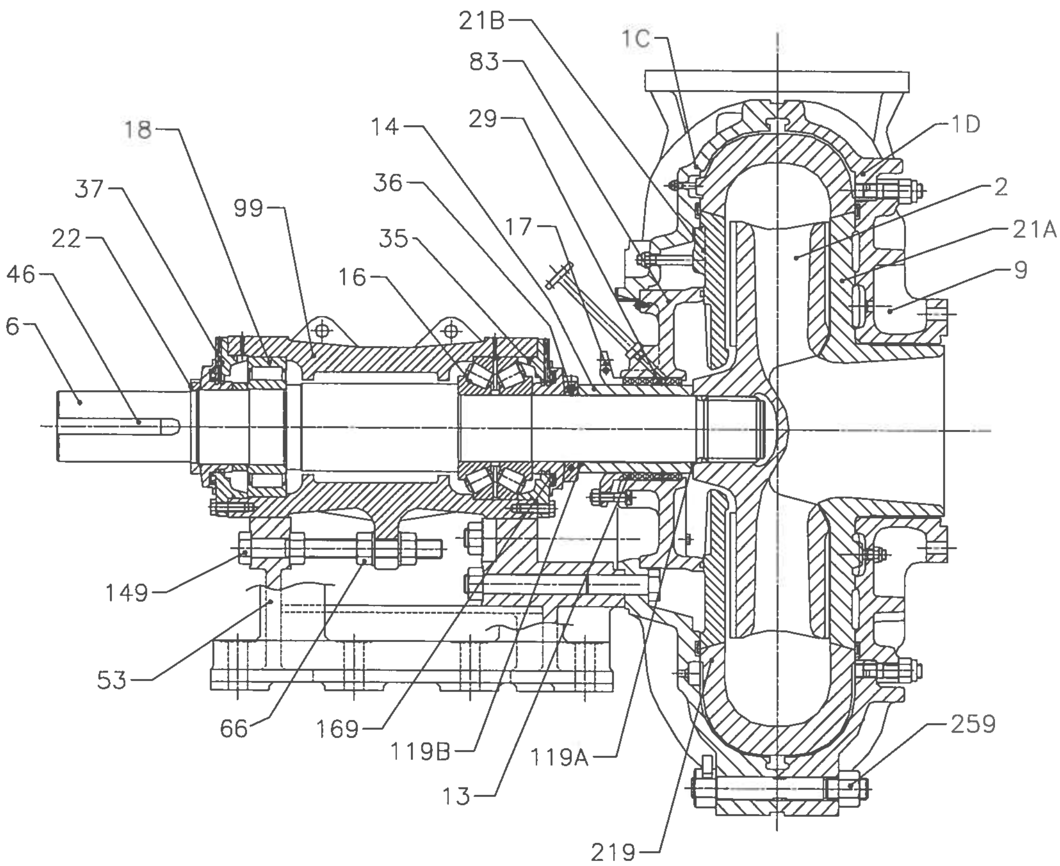
Sponsor

Hydraulic Institute

www.Pumps.org

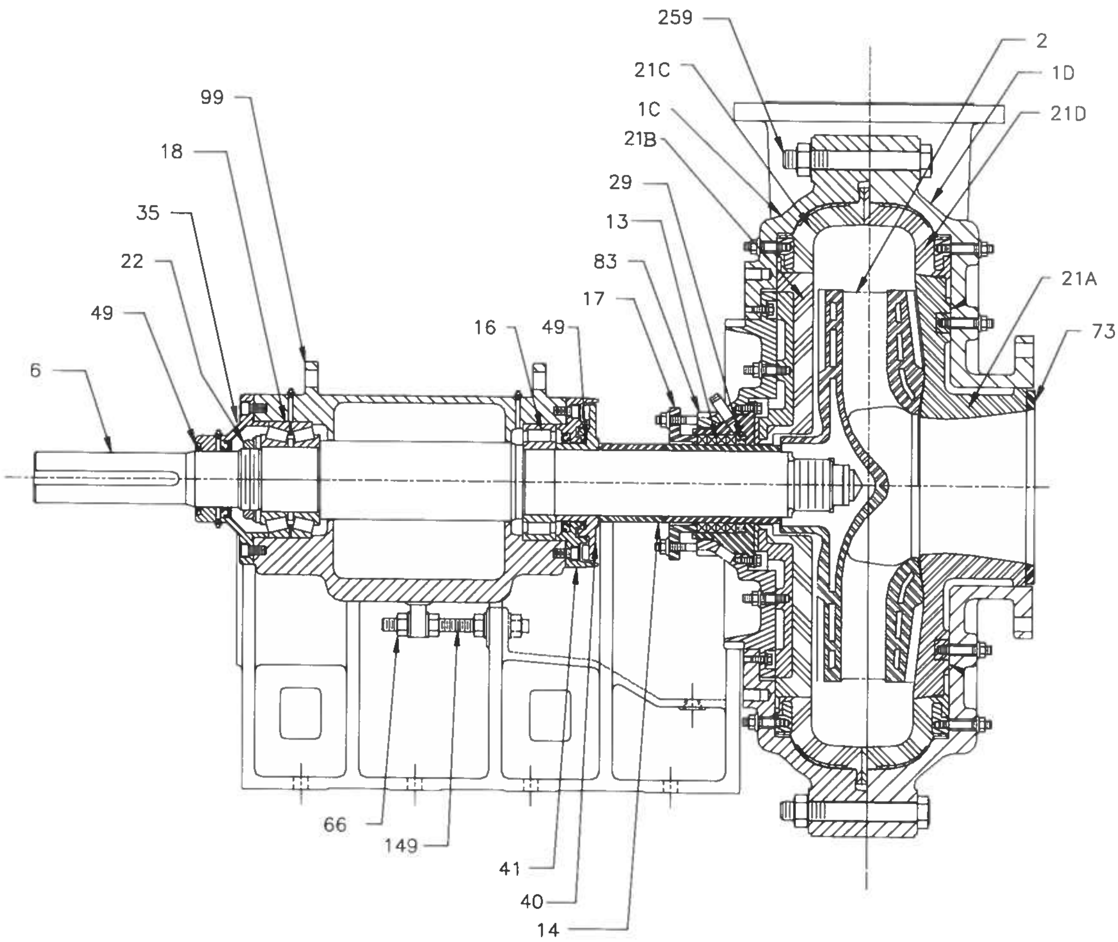
Approved May 19, 2011

American National Standards Institute, Inc.



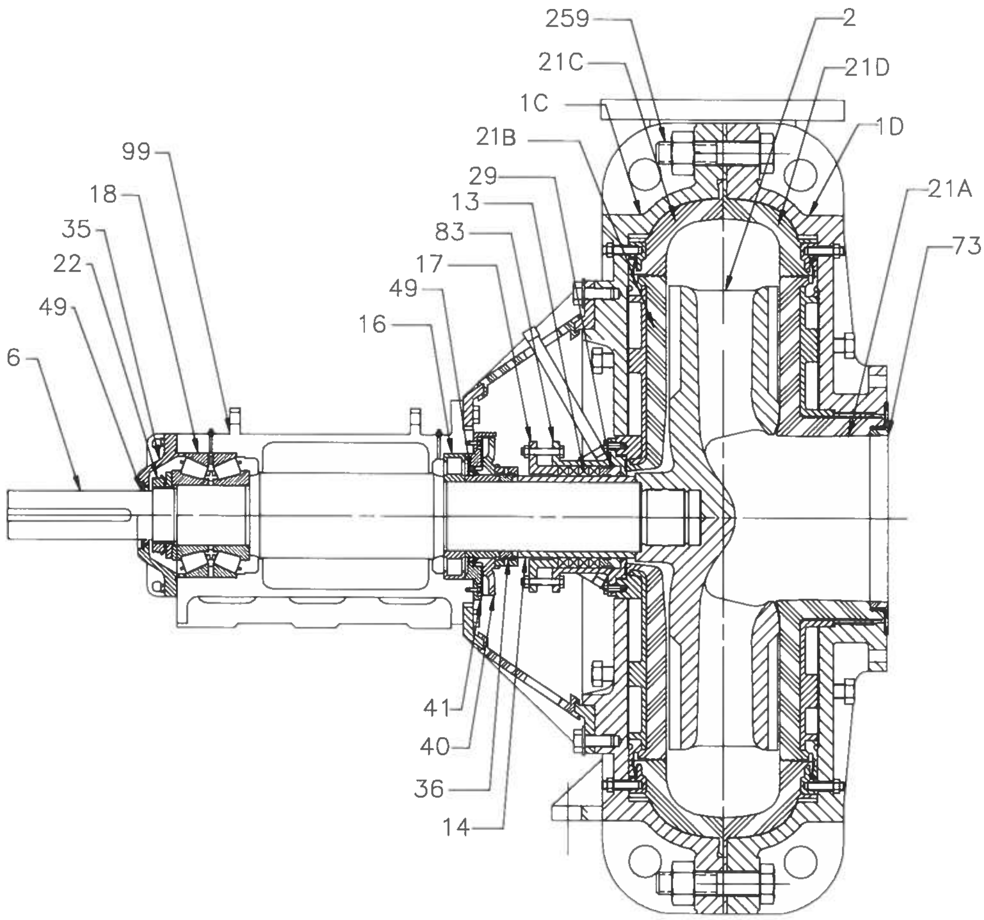
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
9	Cover, suction	13	Packing
14	Sleeve, shaft	16	Bearing, inboard
17	Gland	18	Bearing, outboard
21A	Liner, suction cover	21B	Liner, stuffing box cover
22	Locknut, bearing	29	Ring, lantern
35	Cover, bearing, inboard	36	Collar, release
37	Cover, bearing, outboard	46	Key, coupling
53	Base	66	Nut, shaft adjusting
83	Stuffing box	99	Housing, bearing
119A	O-ring	119B	O-ring
149	Screw, impeller adjusting	169	Seal, bearing housing
219	Liner, casing	259	Bolt, casing

Figure 12.1.2b — Overhung impeller, separately coupled, single-stage, frame-mounted, metal-lined pump (OH0)



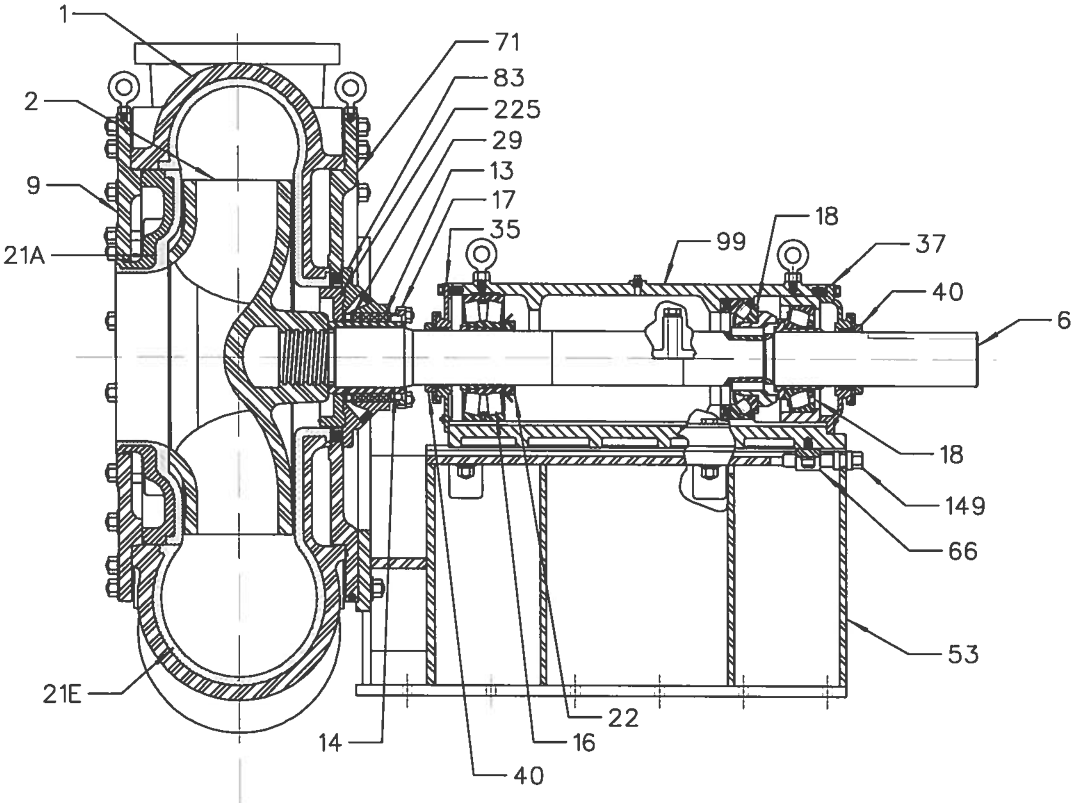
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21B	Liner, stuffing box cover	21C	Liner, gland half
21D	Liner, suction half	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
40	Deflector	41	Cap, bearing, inboard
49	Seal, bearing cover, outboard	66	Nut, shaft adjusting
73	Gasket	83	Stuffing box
99	Housing, bearing	149	Screw, impeller adjusting
259	Bolt, casing		

Figure 12.1.2c — Overhung impeller, separately coupled, single-stage, frame-mounted, elastomer-lined pump (OH0)



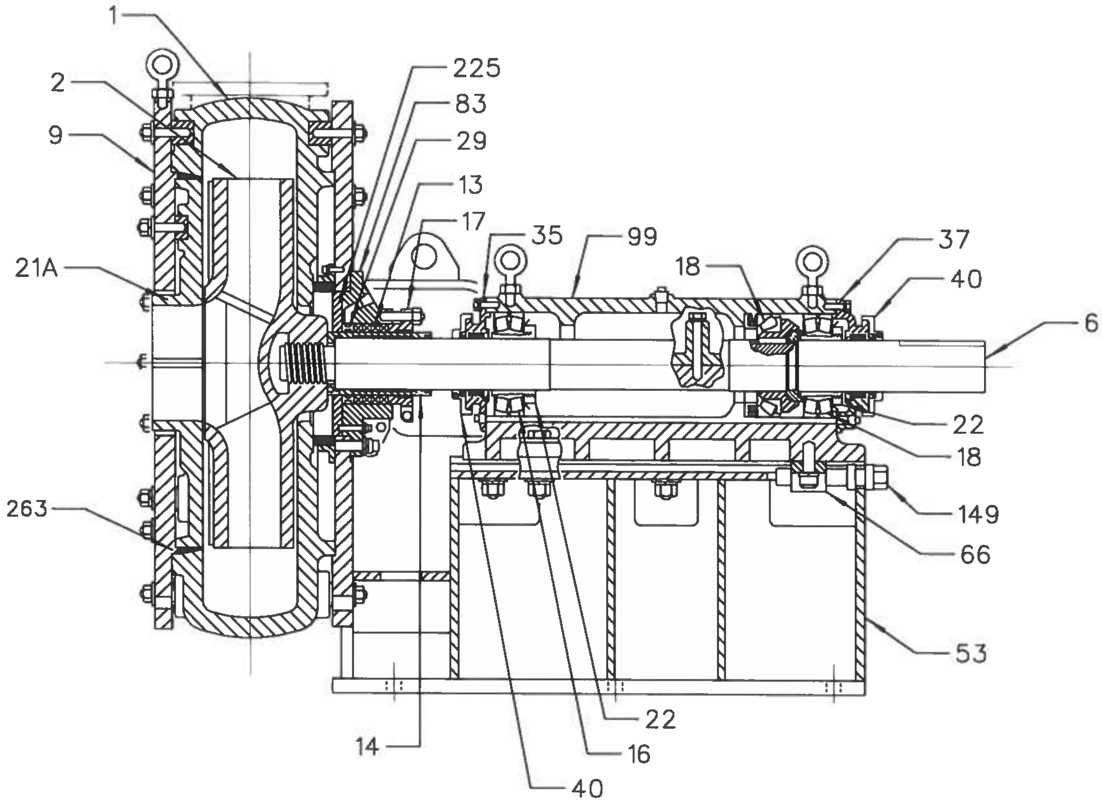
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21B	Liner, stuffing box cover	21C	Liner, gland half
21D	Liner, suction half	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
36	Collar, release	40	Deflector
41	Cap, bearing, inboard	49	Seal, bearing cover, outboard
73	Gasket	83	Stuffing box
99	Housing, bearing	259	Bolt, casing

Figure 12.1.2d — Overhung impeller, separately coupled, single-stage, frame-mounted, elastomer-lined pump, adjustable sideliners (OH0)



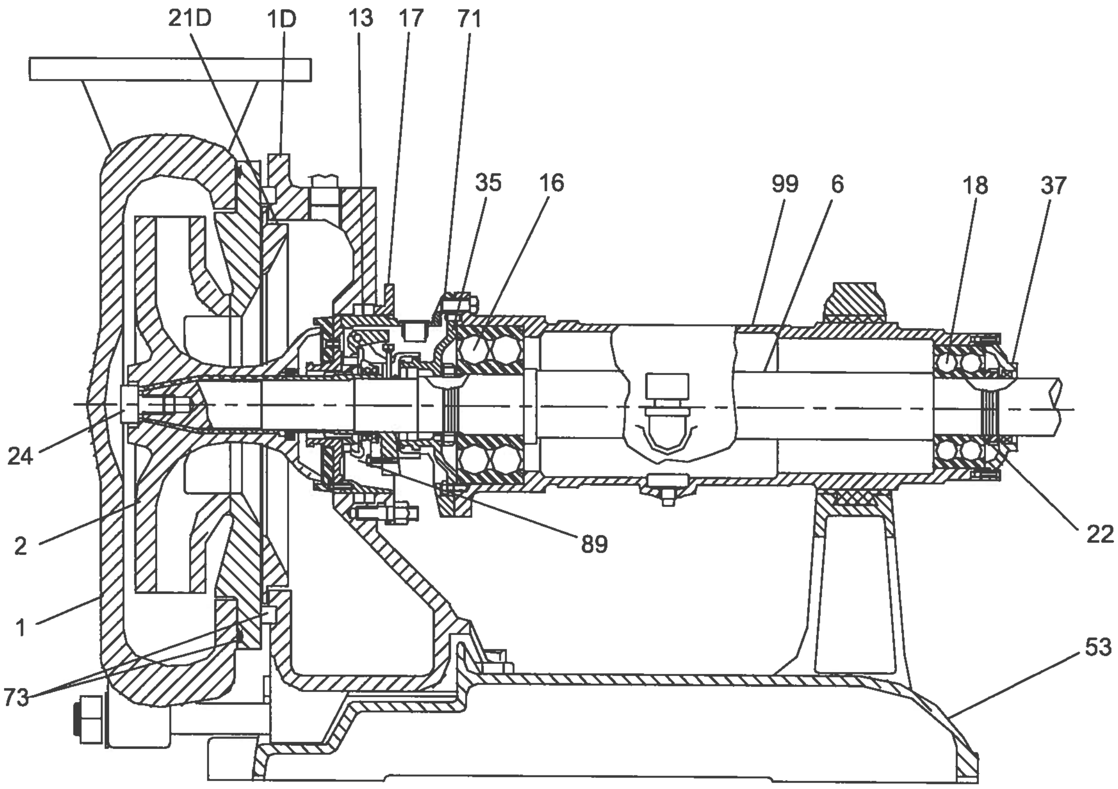
1	Casing	2	Impeller
6	Shaft	9	Cover, suction
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21E	Liner, vulcanized	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
37	Cover, bearing, outboard	40	Deflector
53	Base	66	Nut, shaft adjusting
71	Adapter	83	Stuffing box
99	Housing, bearing	149	Screw, impeller adjusting
225	Plate, wear		

Figure 12.1.2e — Overhung impeller, separately coupled, single-stage, frame-mounted, end suction, vulcanized-elastomer-lined pump (OH0)



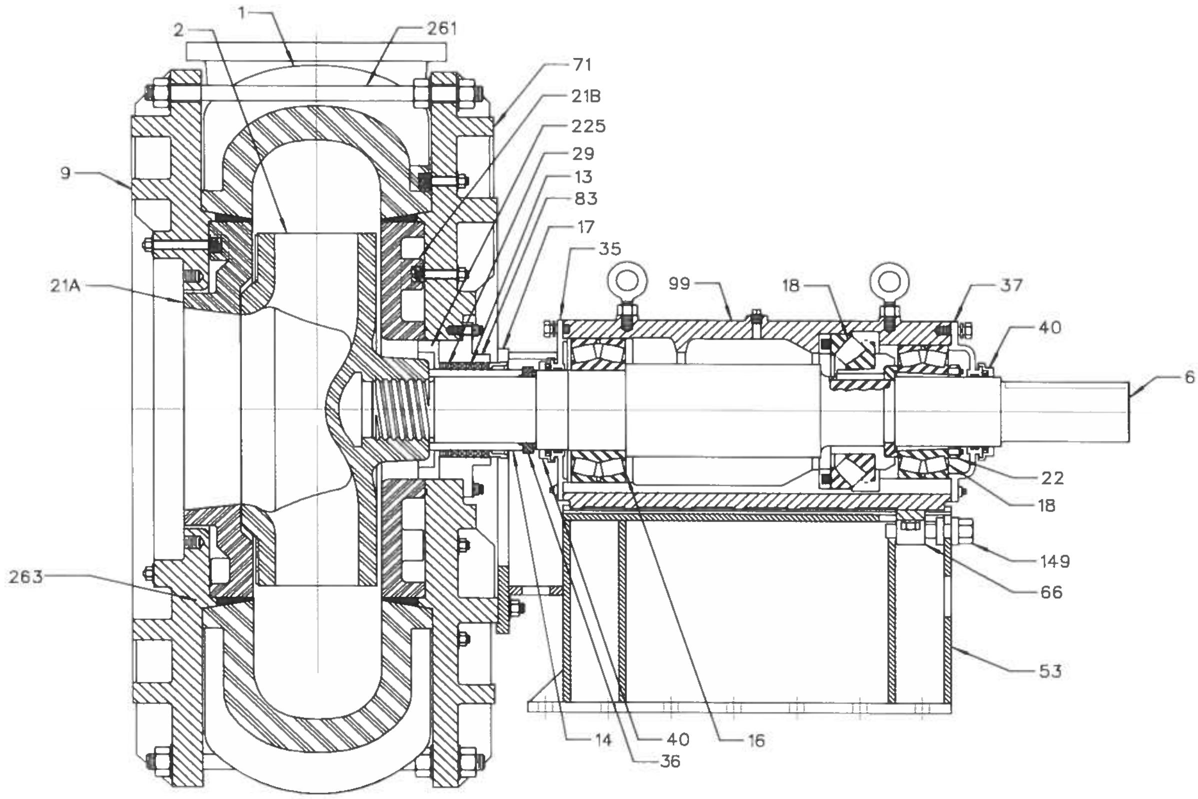
1	Casing	2	Impeller
6	Shaft	9	Cover, suction
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
22	Locknut, bearing	29	Ring, lantern
35	Cover, bearing, inboard	37	Cover, bearing, outboard
40	Deflector	53	Base
66	Nut, shaft adjusting	83	Stuffing box
99	Housing, bearing	149	Screw, impeller adjusting
225	Plate, wear	263	Gasket, snap ring

Figure 12.1.2f — Overhung impeller, separately coupled, single-stage, frame-mounted, end suction, metal, unlined casing pump (OH0)



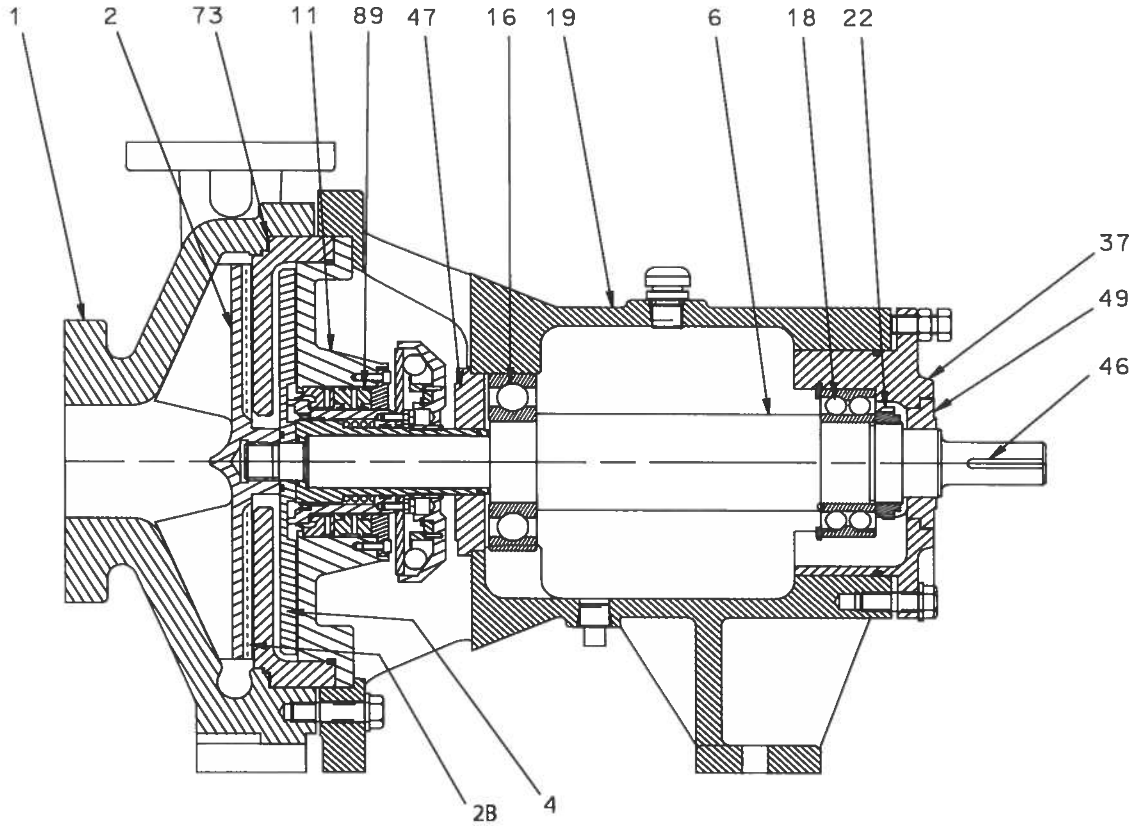
1	Casing	1D	Casing, suction half
2	Impeller	6	Shaft
13	Packing	16	Bearing, inboard
17	Gland	18	Bearing, outboard
21D	Liner, suction half	22	Locknut, bearing
24	Nut, impeller	35	Cover, bearing, inboard
37	Cover, bearing, outboard	53	Base
71	Adapter	73	Gasket
89	Seal	99	Housing, bearing

Figure 12.1.2g — Overhung impeller, separately coupled, single-stage, frame-mounted, side inlet, metal, unlined casing pump (OH0)



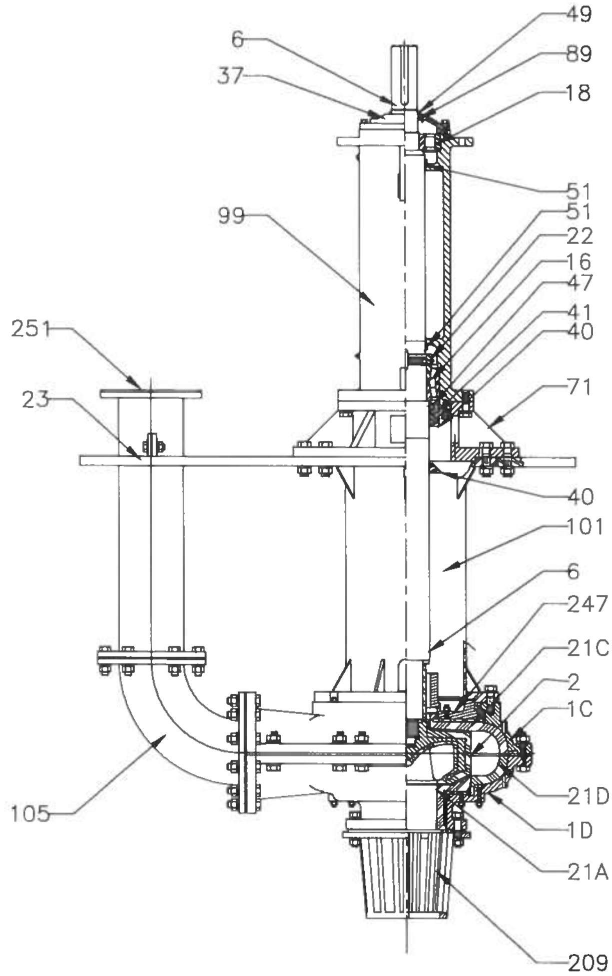
1	Casing	2	Impeller
6	Shaft	9	Cover, suction
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21B	Liner, stuffing box cover	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
36	Collar, release	37	Cover, bearing, outboard
40	Deflector	53	Base
66	Nut, shaft adjusting	71	Adapter
83	Stuffing box	99	Housing, bearing
149	Screw, impeller adjusting	225	Plate, wear
261	Tie bolt	263	Gasket, snap ring

Figure 12.1.2h — Overhung impeller, separately coupled, single-stage, frame-mounted, end suction, metal, tie bolt plate construction pump (OH0)



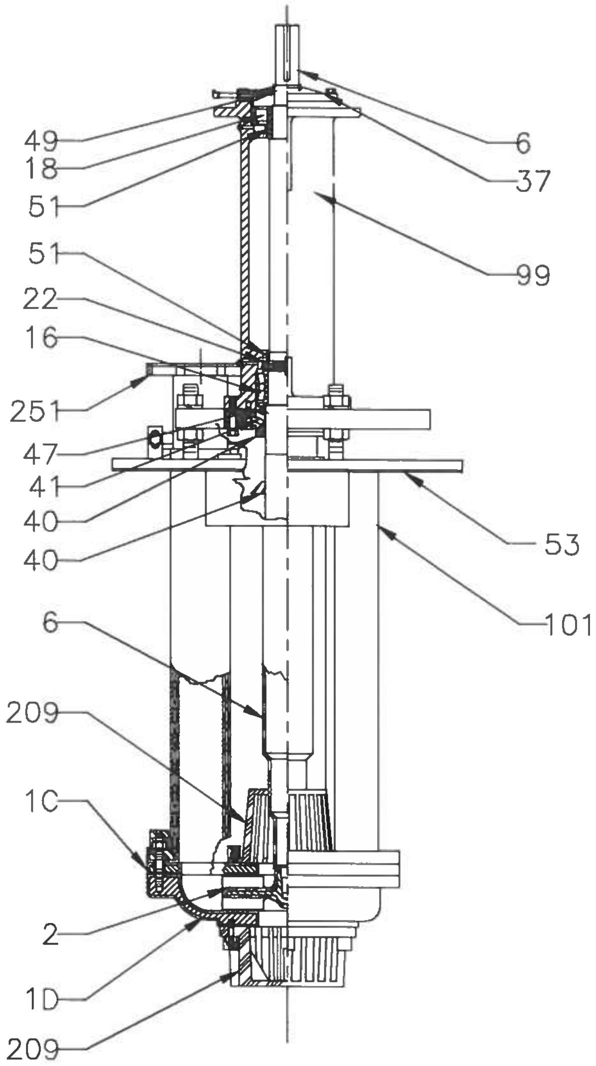
1	Casing	2	Impeller
2B	Expelling vanes	4	Expeller
6	Shaft	11	Cover, stuffing box or seal chamber
16	Bearing, inboard	18	Bearing, outboard
19	Frame	22	Locknut, bearing
37	Cover, bearing, outboard	46	Key, coupling
47	Seal, bearing cover, inboard	49	Seal, bearing cover, outboard
73	Gasket	89	Seal

Figure 12.1.2i — Overhung, open impeller, separately coupled, single-stage, frame-mounted, metal, ASME B73.1 type pump (OH1)



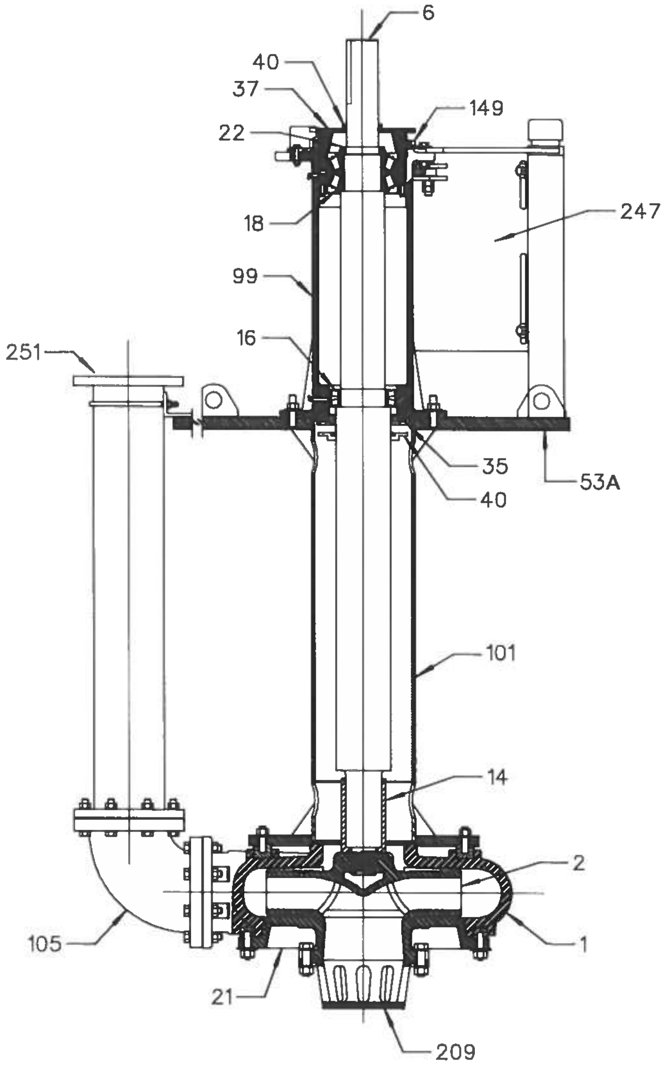
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
16	Bearing, inboard	18	Bearing, outboard
21A	Liner, suction cover	21C	Liner, gland half
21D	Liner, suction half	22	Locknut, bearing
23	Baseplate	37	Cover, bearing, outboard
40	Deflector	41	Cap, bearing, inboard
47	Seal, bearing cover, inboard	49	Seal, bearing cover, outboard
51	Retainer, grease	71	Adapter
89	Seal	99	Housing, bearing
101	Pipe, column	105	Elbow, discharge
209	Strainer	247	Adaptor, casing
251	Flange, discharge		

Figure 12.1.2j — Overhung impeller, separately coupled, single-stage, wet pit cantilever, elastomer-lined, single suction pump (VS5)



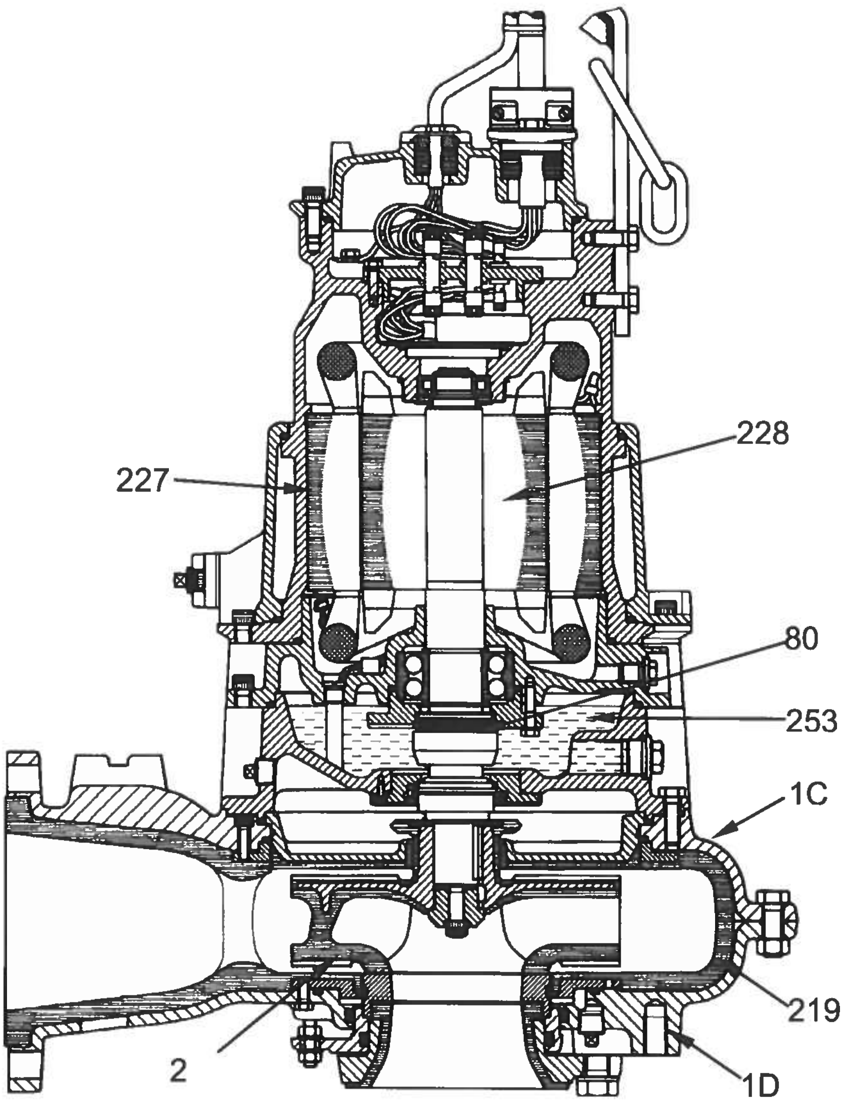
1C	Casing, upper	1D	Casing, lower
2	Impeller	6	Shaft
16	Bearing, inboard	18	Bearing, outboard
22	Locknut, bearing	37	Cover, bearing, outboard
40	Deflector	41	Cap, bearing, inboard
47	Seal, bearing cover, inboard	49	Seal, bearing cover, outboard
51	Retainer, grease	53	Base
99	Housing, bearing	101	Pipe, column
209	Strainer	251	Flange, discharge

Figure 12.1.2k — Overhung impeller, separately coupled, single-stage, wet pit cantilever, elastomer, vulcanized-lined, double suction pump (VS5)



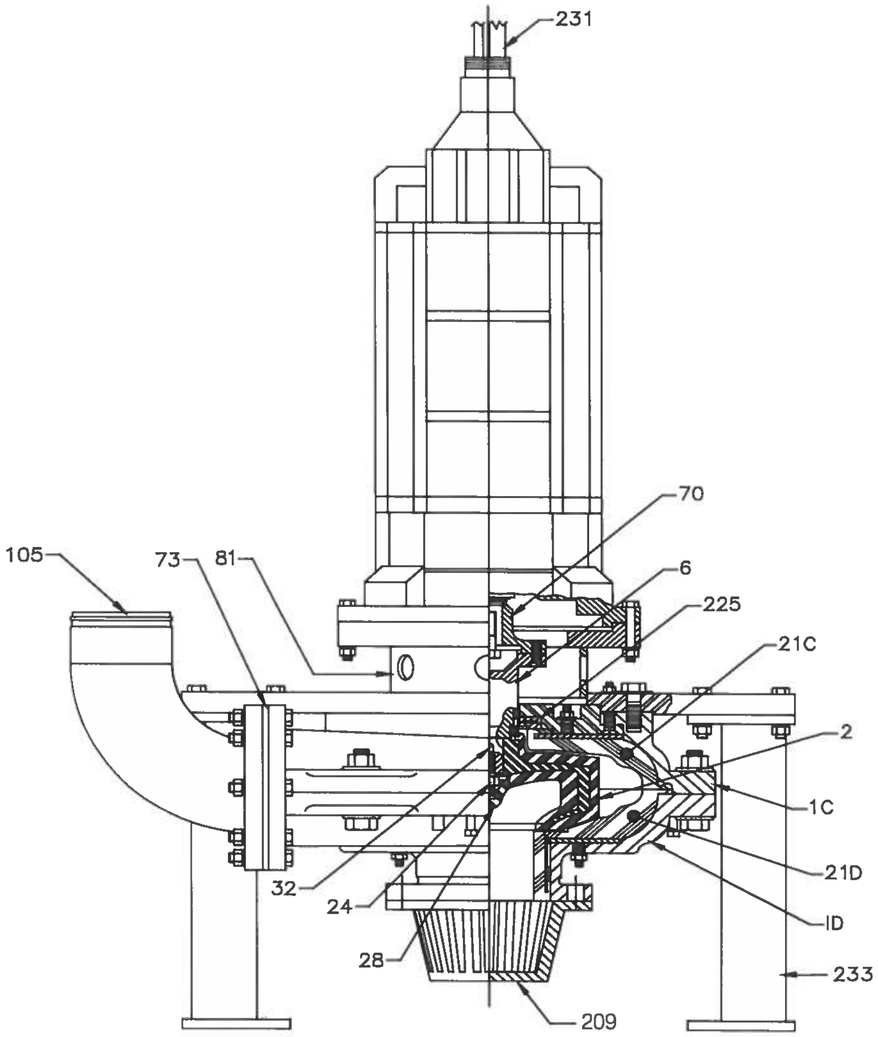
1	Casing	2	Impeller
6	Shaft	14	Sleeve, shaft
16	Bearing, inboard	18	Bearing, outboard
21	Liner, frame	22	Locknut, bearing
35	Cover, bearing, inboard	37	Cover, bearing, outboard
40	Deflector	53A	Plate, floor mounting
99	Housing, bearing	101	Pipe, column
105	Elbow, discharge	149	Screw, impeller adjusting
209	Strainer	247	Adaptor, casing
251	Flange, discharge		

Figure 12.1.2l — Overhung impeller, separately coupled, single-stage, wet pit cantilever, unlined, metal, single suction pump (VS5)



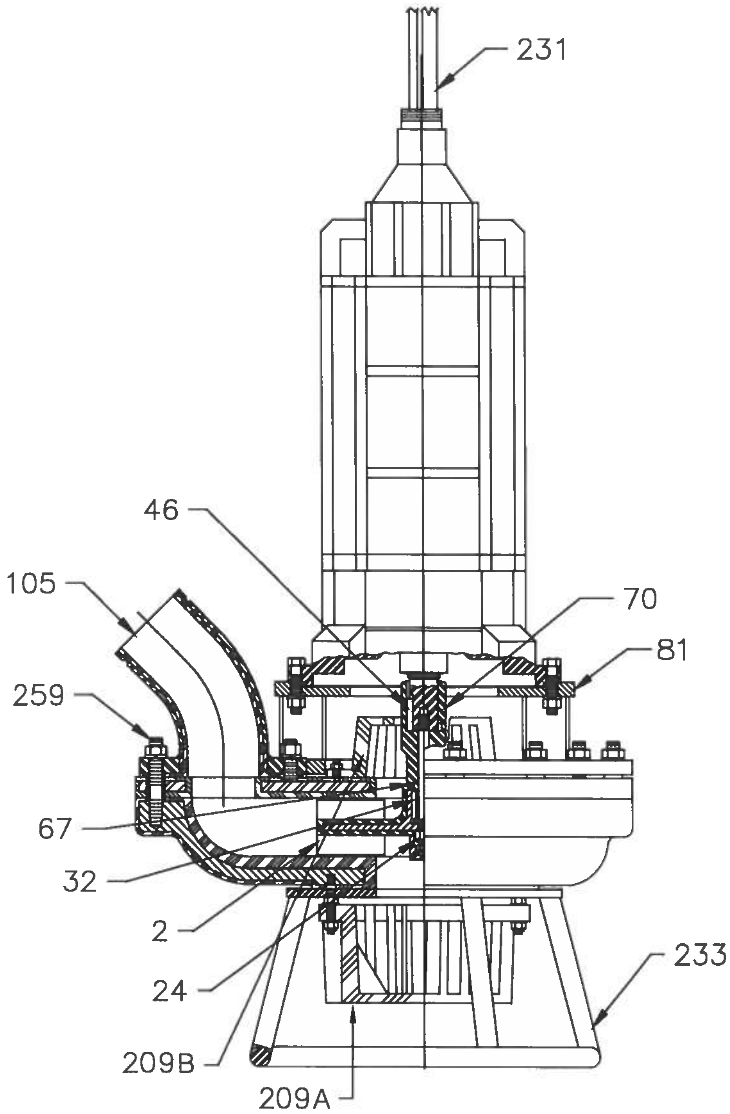
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	80	Seal, mechanical, rotating element
219	Liner, casing	227	Motor, stator
228	Motor, rotor	253	Chamber, barrier liquid, submersible

Figure 12.1.2m — Overhung impeller, close-coupled, single-stage, submersible, elastomer-coated, single suction pump (OH8B)



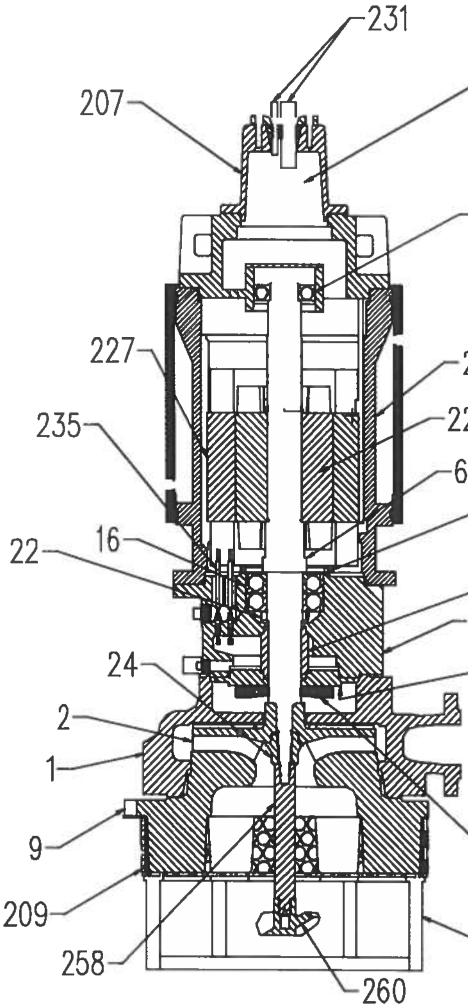
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
21C	Liner, gland half	21D	Liner, suction half
24	Nut, impeller	28	Gasket, impeller screw
32	Key, impeller	70	Coupling, shaft
73	Gasket	81	Pedestal, driver
105	Elbow, discharge	209	Strainer
225	Plate, wear	231	Cable, electric power supply or control
233	Stand, pump		

Figure 12.1.2n — Overhung impeller, close-coupled, single-stage, submersible, elastomer-lined, single suction pump (OH8B)



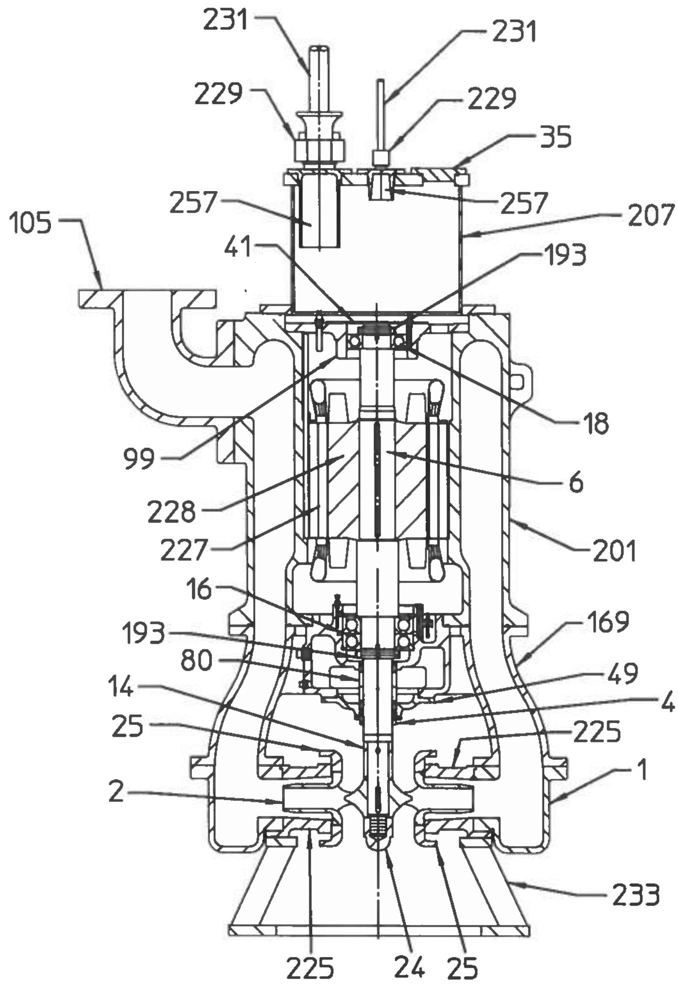
2	Impeller	24	Nut, impeller
32	Key, impeller	46	Key, coupling
67	Shim	70	Coupling, shaft
81	Pedestal, driver	105	Elbow, discharge
209A	Strainer, lower	209B	Strainer, upper
231	Cable, electric power supply or control	233	Stand, pump
259	Bolt, casing		

Figure 12.1.2o — Overhung impeller, close-coupled, single-stage, submersible, elastomer-lined, double suction pump (OH8B)



1	Casing	2	Impeller
6	Shaft	9	Cover, suction
16	Bearing, inboard	18	Bearing, outboard
22	Locknut, bearing	24	Nut, impeller
40	Deflector	49	Seal, bearing cover, outboard
65	Seal, mechanical, stationary element	169	Seal, bearing housing
193	Retainer, bearing	201	Housing, stator
207	Cover, motor end	209	Strainer
227	Motor, stator	228	Motor, rotor
231	Cable, electric power supply or control	233	Stand, pump
235	Probe, moisture detection	257	Seal, cable, epoxy
258	Extender, shaft	260	Agitator, mechanical

Figure 12.1.2p — Overhung impeller, close-coupled, single-stage, end suction, metal, submersible pump with agitator (OH8B)



1	Casing	2	Impeller
6	Shaft	14	Sleeve, shaft
16	Bearing, inboard	18	Bearing, outboard
24	Nut, impeller	25	Ring, suction cover
35	Cover, bearing, inboard	40	Deflector
41	Cap, bearing, inboard	49	Seal, bearing cover, outboard
80	Seal, mechanical, rotating element	99	Housing, bearing
105	Elbow, discharge	169	Seal, bearing housing
193	Retainer, bearing	201	Housing, stator
207	Cover, motor end	225	Plate, wear
227	Motor, stator	228	Motor, rotor
229	Clamp, cable	231	Cable, electric power supply or control
241	Jacket, submersible motor	257	Seal, cable, epoxy

Figure 12.1.2q — Overhung impeller, close-coupled, single-stage, submersible, metal, double suction pump (OH8B)

Table 12.1.12a — Slurry pump nomenclature – alphabetical listing

Part name	Item No.	Abbreviation	Definition
Adapter	71	Adpt	A machined part used to permit assembly of two other parts or as a spacer.
Adapter, casing	247	Adpt csg	Part used to mount the pump casing to drive structure.
Agitator, mechanical	260	Agtr mech	A device attached to the pump shaft that fluidizes settled solids.
Base	53	Base	A pedestal to support a pump.
Baseplate	23	Base pl	Member on which the pump and driver are mounted.
Bearing, inboard	16	Brg inbd	Bearing farthest from the coupling of an end suction pump.
Bearing, outboard	18	Brg outbd	Bearing nearest to the coupling of an end suction pump.
Bolt, casing	259	Blt csng	A threaded bar used to fasten casing halves together, lined pump.
Bolt, tie	261	Blt tie	Threaded bar connecting hub and suction plates.
Bushing, bearing	39	Bush brg	Removable portion of a sleeve bearing, in contact with the journal.
Bushing, pressure-reducing	117	Bush press red	Replaceable piece used to reduce liquid pressure at the stuffing box by throttling the flow.
Bushing, stuffing box	63	Bush stfg box	Replaceable sleeve or ring in the end of the stuffing box opposite the gland.
Bushing, throttle, auxiliary	171	Bush throt aux	Stationary ring or sleeve in the gland of a mechanical seal subassembly to restrict leakage in the event of seal failure.
Cable, electric control	231	Cbl elec ctrl	Conductor for motor instrumentation.
Cable, electric power supply	231	Cbl elec pwr	Conductor for motor power supply.
Cap, bearing, inboard	41	Cap brg inbd	Removable portion of the inboard bearing housing.
Cap, bearing, outboard	43	Cap brg outbd	Removable portion of the outboard bearing housing.
Casing	1	Csg	Portion of the pump that includes the impeller chamber and volute.
Casing, gland half	1C	Csg gld half	The gland half of a radially split casing.
Casing, suction half	1D	Csg suc half	The suction half of a radially split casing.
Chamber, barrier liquid, submersible	253	Chmbr bar liq subm	Volume located between two seals that, when filled with liquid, acts as a barrier between the liquid being pumped and the motor cavity, submersible pump.
Clamp, cable	229	Clmp, cbl	A device to fix the position of a cable, submersible pump.

Table 12.1.12a — Slurry pump nomenclature – alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Collar, release	36	Clr rel	Split ring device to ease removal of the impeller.
Collar, shaft	68	Clr sft	A ring used to establish a shoulder on a shaft.
Column, discharge		Col disch	*See* pipe, discharge.
Coupling half, driver	42	Cplg half drvr	The coupling half mounted on driver shaft.
Coupling half, pump	44	Cplg half pump	The coupling half mounted on pump shaft.
Coupling, oil pump	120	Cplg oil pump	The coupling for the oil pump.
Coupling, shaft	70	Cplg sft	Mechanism used to transmit power from the driver shaft to the driven shaft.
Cover, bearing end	123	Cov brg end	Enclosing plate for the end on the bearing housing.
Cover, bearing, inboard	35	Cov brg inbd	Enclosing plate for the impeller end of the bearing housing of end suction pumps.
Cover, bearing, outboard	37	Cov brg outbd	Enclosing plate for the coupling end of the bearing housing of end suction pumps.
Cover, motor end	207	Cov mot end	Removable piece that encloses end(s) of a motor stator housing.
Cover, oil bearing cap	45	Cov oil brg cap	A lid or plate over an oil filler hole or inspection hole in a bearing cap.
Cover, stuffing box or seal chamber	11	Cov stfg box	A removable piece of an end suction pump casing used to enclose the outboard side of the impeller and includes a stuffing box.
Cover, suction	9	Cov suct	A removable piece of an end suction pump casing used to enclose the suction side of the impeller. The suction nozzle may be integral.
Deflector	40	Defl	A flange or collar around a shaft and rotating with it to inhibit passage of liquid, grease, oil, or heat along the shaft.
Elbow, discharge	105	Ell disch	An elbow in wet pit cantilever or submersible pump by which the liquid leaves the pump.
Elbow, suction	57	Ell suct	A curved water passage attached to the pump inlet.
Expeller	4	Explr	A secondary impeller fitted with vanes used to reduce or balance pressure at the stuffing box of a slurry pump.
Expelling vanes	2B	Exp vane	Vanes on the front, back, or both shrouds of a slurry pump impeller used to limit recirculation, to reduce the concentration of solids between the impeller and casing sides, and to reduce the pressure at the stuffing box.

Table 12.1.12a — Slurry pump nomenclature – alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Extender, shaft	258	Extdr sft	A part that extends the pump drive shaft outboard of the impeller such that a mechanical agitator can be driven from the pump shaft.
Flange, discharge	251	Flg disch	A pipe connection at the pump liquid outlet.
Frame	19	Fr	A member of an end suction pump to which are assembled the liquid end and rotating element.
Gasket	73	Gskt	Resilient material used to seal joints between parts to prevent leakage.
Gasket, impeller screw	28	Gskt imp scr	Resilient material used to seal the joint between the hub of the impeller and the impeller screw.
Gasket, shaft sleeve	38	Gskt sft slv	Resilient material used to provide a seal between the shaft sleeve and the impeller.
Gasket, snap ring	263	Gskt snp ring	Trapezoidal section shaped resilient material used to provide a seal between liner and casing.
Gauge, sight, oil	143	Ga sight oil	Device for visual determination of oil level.
Gland	17	Gld	A follower that compresses packing in a stuffing box or retains a stationary element of a mechanical seal.
Gland, stuffing box, auxiliary	133	Gld stfg box aux	A follower for compression of packing in an auxiliary stuffing box.
Guard, coupling	131	Gld cplg	A protective shield over a shaft coupling.
Housing, bearing	99	Hsg brg	A body in which a bearing or bearing set is mounted.
Housing, bearing, inboard	31	Hsg brg inbd	*See* bearing (inboard) and bearing housing.
Housing, bearing, outboard	33	Hsg brg outbd	*See* bearing (outboard) and bearing housing.
Housing, seal	237	Hsg seal	A body in which the shaft seals are mounted.
Housing, stator	201	Hst sttr	A body in which a stator core assembly is mounted.
Impeller	2	Imp	The bladed member of the rotating assembly of a pump that imparts the principal energy to the liquid pumped.
Jacket, submersible motor	241	Jkt, sub mtr	A chamber located in close proximity to the submersible pump motor windings, in which coolant is available for maintaining acceptable motor temperature.
Journal, thrust bearing	74	Jnl thr brg	Removable cylindrical piece mounted on the shaft that turns in the bearing. It may have an integral thrust collar.
Key, bearing journal	76	Key brg jnl	A parallel-sided piece used for preventing the bearing journal from rotating relative to the shaft.

Table 12.1.12a — Slurry pump nomenclature – alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Key, coupling	46	Key cplg	A parallel-sided piece used to prevent the shaft from turning in a coupling half.
Key, impeller	32	Key imp	A parallel-sided piece used to prevent the impeller from rotating relative to the shaft.
Liner, casing	219	Lnr csg	A replaceable metal or elastomer insert that provides a renewable waterway in the casing of a slurry pump.
Liner, frame	21	Lnr fr	A part within the frame carrying one or more of the bearings.
Liner, gland half	21C	Lnr gld half	A part within the casing, gland half.
Liner, stuffing-box cover	21B	Lnr stfg box cov	A part within the stuffing-box cover.
Liner, suction cover	21A	Lnr suct cov	A part within the suction cover.
Liner, suction half	21D	Lnr suct half	A part within the casing, suction half.
Liner, vulcanized	21E	Lnr vul	An elastomer liner within the casing.
Locknut, bearing	22	Lknut brg	Fastening that positions an antifriction bearing on the shaft.
Locknut, coupling	50	Lknut cplg	A fastener holding a coupling half in position on a tapered shaft.
Lockwasher	69	Lkwash	A device to prevent loosening of a nut.
Motor, rotor	228	Mtr rotr	The rotating part of an electric motor, submersible pump.
Motor, stator	227	Mtr statr	The stationary part of an electric motor, submersible pump.
Nut, impeller	24	Nut imp	A threaded piece used to fasten the impeller on the shaft.
Nut, shaft adjusting	66	Nut sft adj	A threaded piece for altering the axial position of the rotating assembly.
Nut, shaft sleeve	20	Nut sft slv	A threaded piece used to locate the shaft sleeve on the shaft.
O-ring	119	Ring O	A radial or axial elastomer seal.
Packing	13	Pkg	A pliable lubricated material used to provide a seal around the portion of the shaft located in the stuffing box.
Pipe, column	101	Pipe col	A vertical pipe by which the pumping element is suspended.
Pipe, discharge	103	Disch pipe	Pipe used to provide a convenient discharge connection at or above the floor mounting plate of vertical sump pumps.

Table 12.1.12a — Slurry pump nomenclature – alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Plate, floor mounting	53A	Mtg pl	Plate used to suspend a vertical sump pump over the sump that it draws from.
Plate, side	61	Pl side	A replaceable piece in the casing or cover of a pump to maintain a close clearance along the impeller face.
Plate, wear	225	Wp pl	A removable, axial clearance part used to protect the casing, stuffing box, or suction cover from wear.
Probe, moisture detection	235	Prob moist detct	Conductivity probe to allow the detection of water leakage past the primary seals, submersible pump.
Pump, oil	121	Pump oil	A device for supplying pressurized lubricating oil.
Retainer, bearing	193	Ret brg	A device used to support the shaft bearing.
Retainer, grease	51	Ret grs	A contact seal or cover to keep grease in place.
Ring, lantern	29	Ring ltrn	An annular piece used in the stuffing box to establish a path for lubricating or flushing liquid around the shaft sleeve.
Ring, oil	60	Ring oil	A rotating ring used to carry oil from the reservoir to the bearings.
Ring, suction cover	25	Ring suct cov	A stationary ring to protect the suction cover at the running fit with the impeller ring or impeller.
Screw, impeller	26	Scr imp	A special screw to fasten the impeller to the shaft.
Screw, impeller, adjusting	149	Scr imp adj	A special screw to adjust the axial movement of shaft/ impeller or sideliner to control front seal face clearance.
Seal	89	Seal	A device to prevent the flow of a liquid or gas into or from a cavity.
Seal, bearing cover, inboard	47	Seal brg cov inbd	A labyrinth seal, bearing isolator, or lip seal for the bearing cover (inboard).
Seal, bearing cover, outboard	49	Seal brg cov outbd	A labyrinth seal, bearing isolator, or lip seal for the bearing cover (outboard).
Seal, bearing housing	169	Seal brg hsg	A contact seal for a bearing housing on the stuffing-box end having a smooth, flat seal face lined against the rotating element.
Seal, cable, epoxy	257	Seal cbl epxy	Where an insulating resin is used to seal the electric supply cable entry to the motor housing, submersible pump.
Seal, cable jacket	255	Seal cbl jkt	A resilient component that stops the passage of liquid between the jacket of an electric cable and a component enclosure, submersible pump.
Seal, mechanical, rotating element	80	Seal mech rot elem	A subassembly consisting of multiple parts mounted to the pump shaft within the stuffing box and having a smooth, flat sealing face.

Table 12.1.12a — Slurry pump nomenclature – alphabetical listing (*continued*)

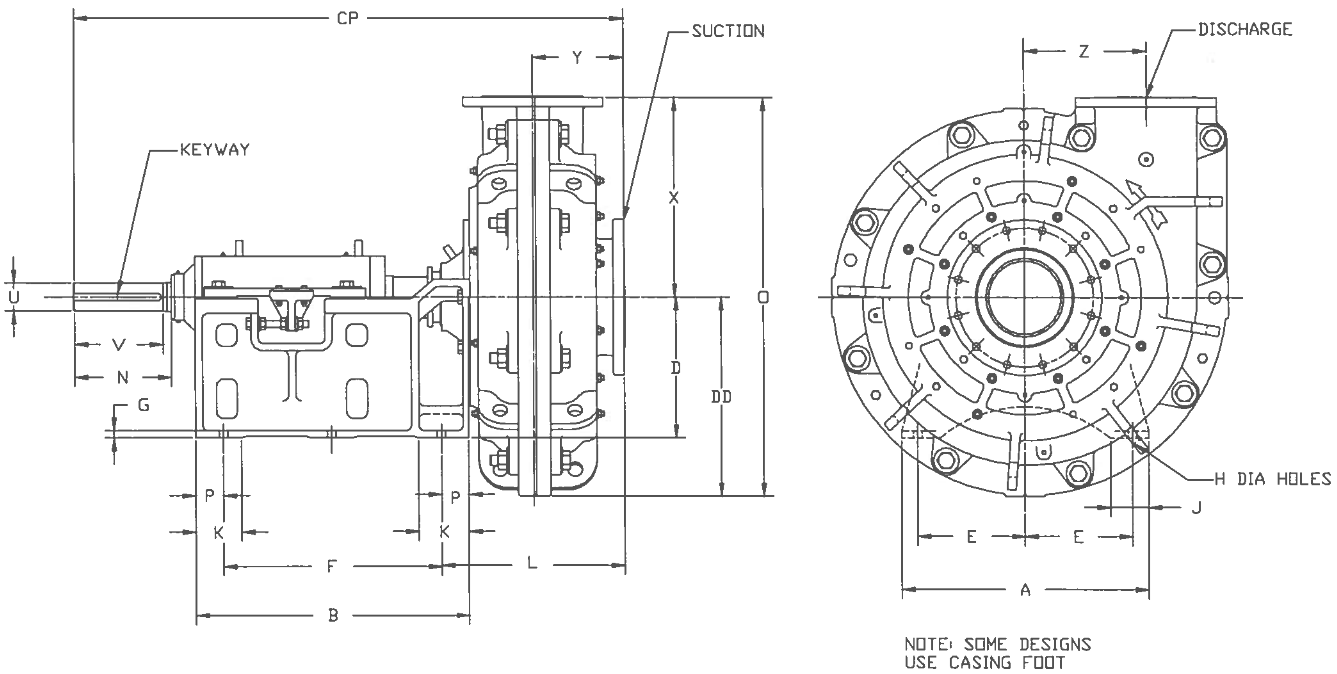
Part name	Item No.	Abbreviation	Definition
Seal, mechanical, stationary element	65	Seal mech sta elem	A subassembly consisting of one or more parts mounted in or on a stuffing box and having a smooth, flat sealing face.
Shaft	6	Sft	The cylindrical member on which the impeller is mounted and through which power is transmitted to the impeller.
Shield, oil retaining	107	Shld oil retg	A device to prevent oil leaking from the bearing housing.
Shim	67	Shim	A piece of material placed between two members to adjust their relative position or fill in a gap.
Sleeve, impeller hub	34	Slv imp hub	A replaceable, cylindrical wear part mounted on the extended pump impeller hub.
Sleeve, shaft	14	Slv sft	A cylindrical piece fitted over the shaft to protect the shaft through the stuffing box and that may also serve to locate the impeller on the shaft.
Spacer, bearing	78	Spcr brg	Sleeve that fits over a shaft to locate antifriction bearings.
Spacer, coupling	88	Spcr cplg	A cylindrical piece used to provide axial space for the removal of the rotating assembly without removing the driver.
Stand, pump	233	Stnd pmp	A part that will support the pump weight.
Strainer	209	Str	Device used to prevent oversized objects from entering pump.
Strainer, lower	209A	Str, lwr	A device used to prevent large objects from entering the pump, located above the impeller centerline.
Strainer, upper	209B	Str, upr	A device used to prevent large objects from entering the pump, located below the impeller centerline.
Stuffing box	83	Stfg box	A portion of the casing through which packing and a gland or a mechanical seal is placed to prevent leakage.
Stuffing box, auxiliary	75	Stfg box aux	A recessed portion of the gland and cover of a mechanical seal subassembly designed to accommodate one or more rings of packing.
Suction extension		Suct ext	*See* tailpipe.
Support, discharge pipe	249	Supt disch pipe	A device to support the discharge pipe.
Tailpipe		Tailpipe	A length of pipe used to extend the suction inlet of vertical slurry pumps.
Thrower (oil or grease)	62	Thwr (oil or grs)	A disk rotating with the pump shaft to carry the lubricant from the reservoir to the bearing.

Table 12.1.12b — Slurry pump nomenclature - numerical listing

1	Casing	67	Shim
1C	Casing, gland half	68	Collar, shaft
1D	Casing, suction half	69	Lockwasher
2	Impeller	70	Coupling, shaft
2B	Expelling vanes	71	Adapter
4	Expeller	73	Gasket
6	Shaft	74	Journal, thrust bearing
9	Cover, suction	75	Stuffing box, auxiliary
11	Cover, stuffing box or seal chamber	76	Key, bearing journal
13	Packing	78	Spacer, bearing
14	Sleeve, shaft	80	Seal, mechanical, rotating element
16	Bearing, inboard	83	Stuffing box
17	Gland	88	Spacer, coupling
18	Bearing, outboard	89	Seal
19	Frame	99	Housing, bearing
20	Nut, shaft sleeve	101	Pipe, column
21	Liner, frame	103	Pipe, discharge
21A	Liner, suction cover	105	Elbow, discharge
21B	Liner, stuffing-box cover	107	Shield, oil retaining
21C	Liner, gland half	117	Bushing, pressure reducing
21D	Liner, suction half	119	O-ring
21E	Liner, vulcanized	120	Coupling, oil pump
22	Locknut, bearing	121	Pump, oil
23	Baseplate	123	Cover, bearing end
24	Nut, impeller	131	Guard, coupling
25	Ring, suction cover	133	Gland, stuffing box, auxiliary
26	Screw, impeller	143	Gauge, sight, oil
28	Gasket, impeller screw	149	Screw, impeller, adjusting
29	Ring, lantern	169	Seal, bearing housing
31	Housing, bearing, inboard	171	Bushing, throttle, auxiliary
32	Key, impeller	193	Retainer, bearing
33	Housing, bearing, outboard	201	Housing, stator
34	Sleeve, impeller hub	207	Cover, motor end
35	Cover, bearing, inboard	209	Strainer
36	Collar, release	209A	Strainer, lower
37	Cover, bearing, outboard	209B	Strainer, upper
38	Gasket, shaft sleeve	219	Liner, casing
39	Bushing, bearing	225	Plate, wear
40	Deflector	227	Motor, stator
41	Cap, bearing, inboard	228	Motor, rotor
42	Coupling half, driver	229	Clamp, cable
43	Cap, bearing outboard	231	Cable, electric control
44	Coupling half, pump	231	Cable, electric power supply
45	Cover, oil bearing cap	233	Stand, pump
46	Key, coupling	235	Probe, moisture detection
47	Seal, bearing cover, inboard	237	Housing, seal
49	Seal, bearing cover, outboard	241	Jacket, submersible motor
50	Locknut, coupling	247	Adapter, casing
51	Retainer, grease	249	Support, discharge pipe
53	Base	251	Flange, discharge
53A	Plate, floor mounting	253	Chamber, barrier liquid, submersible
57	Elbow, suction	255	Seal, cable jacket
60	Ring, oil	257	Seal, cable, epoxy
61	Plate, side	258	Extender, shaft
62	Thrower (oil or grease)	259	Bolt, casing
63	Bushing, stuffing box	260	Agitator, mechanical
65	Seal, mechanical, stationary element	261	Bolt, tie
66	Nut, shaft adjusting	263	Gasket, snap ring

12.1.13 Letter dimensional designations

The letter designations used in Figures 12.1.13a through 12.1.13c provide a common means of identifying various pump dimensions and serve as a common language that will be mutually understandable to the purchaser, manufacturer, and anyone writing specifications for pumps and pumping equipment.



A Width of base support

B Length of base support

CP Length of pump

D Vertical height from bottom of base support to centerline of pump

DD Distance from pump centerline to bottom casing

E Distance from centerline pump to centerline hold-down bolts

F Distance from centerline to centerline of outer-most hold-down bolts

G Thickness of pads on support or height of baseplate

H Diameter of hold-down bolt holes

J Width of pads for hold-down bolts

K Length of support pads for hold-down bolts

L Horizontal distance from suction nozzle face to centerline nearest hold-down bolt holes

N Distance from end of bearing housing to end of shaft

O Vertical distance from bottom of casing to discharge nozzle face

P Length from edge of support or baseplate to centerline of bolt holes

U Diameter of straight shaft – coupling end

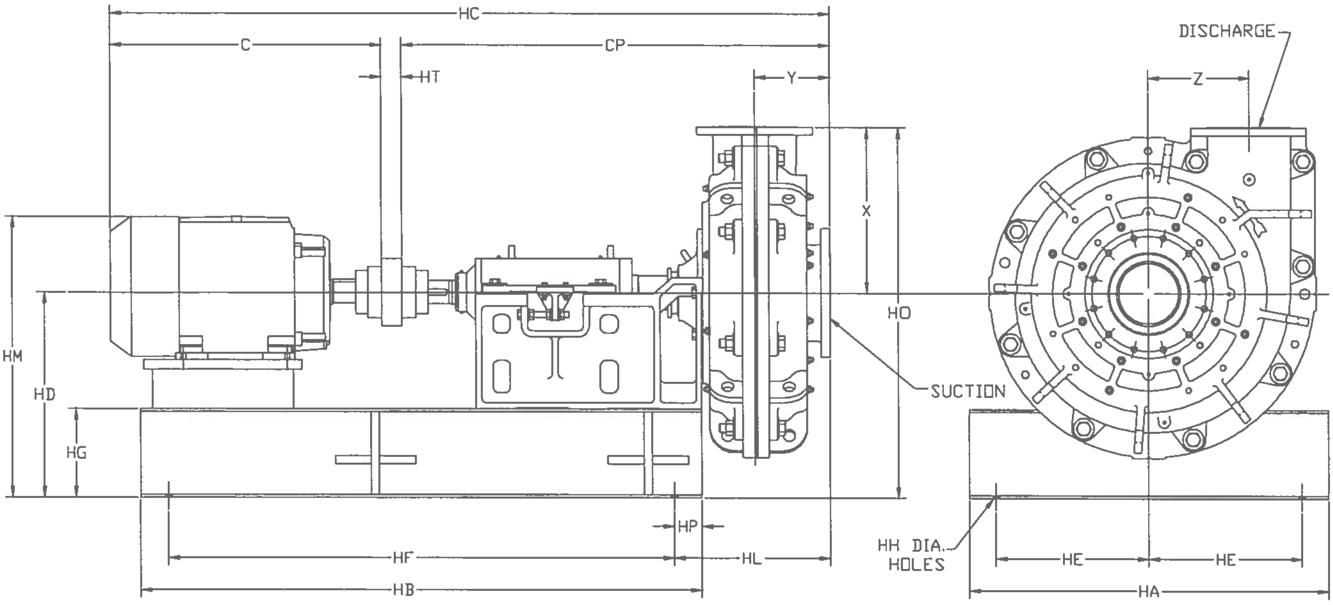
V Length of shaft available for coupling or pulley

X Distance from discharge face to centerline of pump

Y Horizontal distance from centerline discharge nozzle to suction nozzle face

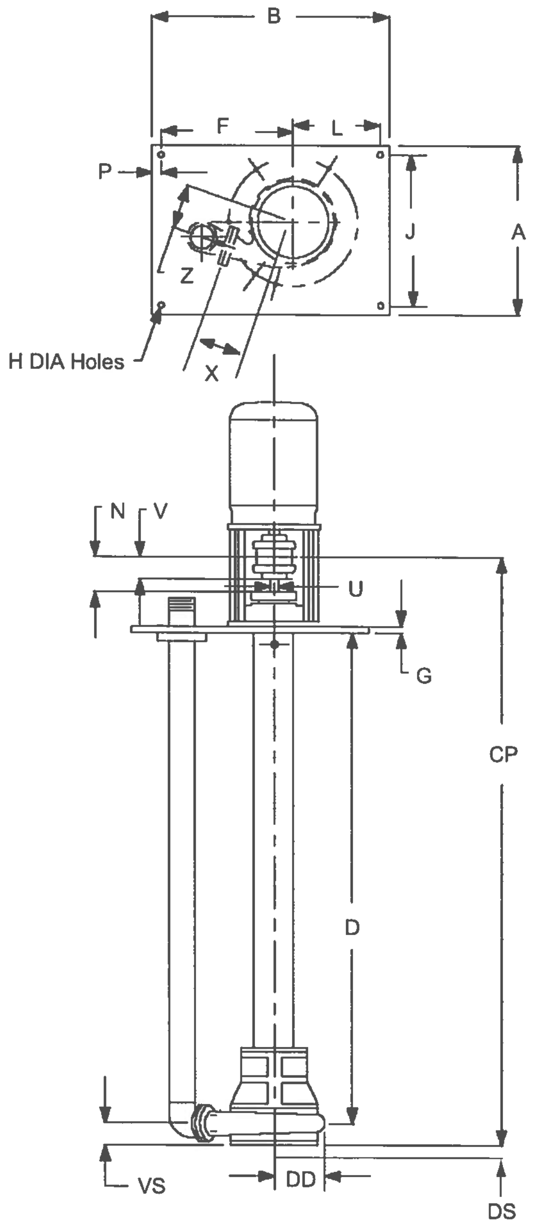
Z Centerline discharge nozzle to centerline of pump

Figure 12.1.13a — Horizontal pump dimensions



C	Length of driver	HH	Diameter of hold-down bolt holes
CP	Length of pump	HL	Horizontal distance from suction nozzle face to centerline nearest hold-down bolt holes
HA	Width of base support	HM	Height of unit from bottom of base to top of driver
HB	Length of base support	HO	Vertical distance from bottom of support to discharge nozzle face
HC	Overall length of combined pump and driver when on base	HP	Length from edge of support, or baseplate, to centerline of bolt holes
HD	Vertical height from bottom of base support to centerline of pump	HT	Horizontal distance from between pump and driving shaft
HE	Distance from centerline pump to centerline hold-down bolts	X	Distance from discharge face to centerline of pump
HF	Distance from centerline to centerline of hold-down bolt holes	Y	Horizontal distance from centerline discharge nozzle to suction nozzle face
HG	Thickness of pads on support or heights of baseplate	Z	Centerline discharge nozzle to centerline of pump

Figure 12.1.13b — Direct drive pump and motor assembly dimensions



A Width of base support

B Length of base support

CP Length of pump

D Vertical height from bottom of base support to centerline of pump

DD Distance from centerline pump to casing

DS Minimum distance from suction to floor

F Distance from centerline pump to centerline of furthest hold-down bolts

G Thickness of pads on support or height of baseplate

H Diameter of hold-down bolt holes

J Distance between hold-down bolts on the short side

L Horizontal distance from suction nozzle face to centerline nearest hold-down bolt holes

N Distance from end of bearing housing to end of shaft

P Length from edge of support or baseplate to centerline of bolt holes

U Diameter of straight shaft-coupling end

X Distance from discharge face to centerline of pump

VS Vertical distance from centerline discharge nozzle to suction nozzle face

Z Centerline discharge nozzle to centerline of pump

Figure 12.1.13c — Vertical pump dimension

12.2 Definitions

This section defines terms used in slurry pump applications. Principal symbols, terms, and units are described in Table 12.2a and subscripts in Table 12.2b.

Table 12.2a — Principal symbols

Symbol	Term	To convert from US customary units (USCS)	Abbr	Into metric units	Abbr	Multiply by conversion factor
A	Area	square inch	in^2	square millimeter	mm^2	645.2
C_v	Concentration by volume	percentage	%	percentage	%	1
C_w	Concentration by weight	percentage	%	percentage	%	1
Δ (delta)	Difference	dimensionless[a]	—	dimensionless	—	—
D	Diameter	inch	in	millimeter	mm	25.4
η (eta)	Efficiency	percent	%	percent	%	1
γ (gamma)	Specific weight	Pound force per cubic foot	lbf/ft^3	Newton per cubic meter	N/m^3	157.1
g	Gravitational acceleration	foot/second squared	ft/s^2	meter/second squared	m/s^2	0.3048
h	Head (general term)	foot	ft	meter	m	0.3048
H	Pump total head	foot	ft	meter	m	0.3048
l	Static lift	foot	ft	meter	m	0.3048
ID	Inside diameter of pipe	inch	in	millimeter	mm	25.4
L	Length	inch	in	millimeter	mm	25.4
μ (mu)	Coefficient of friction	dimensionless	—	dimensionless	—	—
n	Speed	revolution/minute	rpm	revolution/minute	rpm	1
NPSHA	Net positive suction head available	foot	ft	meter	m	0.3048
NPSHR	Net positive suction head required	foot	ft	meter	m	0.3048
n_s (N_s)	Specific speed n_s (N_s) = $nQ^{0.5}/H^{0.75}$	Index number	—	Index number	—	0.0194[b]
ν (nu)	Kinematic viscosity	foot squared/second	ft^2/s	meter squared/second	m^2/s	0.092903
ϕ (phi)	Velocity in vibration	inch/second	in/s	millimeter/second	mm/s	25.4
π	pi = 3.1416	dimensionless	—	dimensionless	—	1
p	Pressure	pound/square inch	psi	kilopascal	kPa	6.895
P	Power	horsepower	hp	kilowatt	kW	0.7457
Q	Rate of flow (capacity)	US gallon/minute	US gpm	cubic meter/second	m^3/s	0.0000631
Q	Rate of flow (capacity)	US gallon/minute	US gpm	cubic meter/hour	m^3/h	0.2271

Table 12.2a — Principal symbols (*continued*)

Symbol	Term	To convert from US customary units (USCS)	Abbr	Into metric units	Abbr	Multiply by conversion factor
RT	Radial thrust	pound (force)	lbf	newton	N	4.448
ρ (rho)	Density	pound mass/cubic foot	lbm/ft^3	kilogram/cubic meter	kg/m^3	16.02
s	Specific gravity (general term)					
S_m	Specific gravity (relative density) of slurry (mixture)	dimensionless	—	dimensionless	—	1
S_s	Specific gravity (relative density) of solid particles					
S_w	Specific gravity (relative density) of water, or carrier liquid					
T	Temperature	degree Fahrenheit	°F	degree Celsius	°C	(°F–32) × (5/9)
τ (tau)	Torque	pound-foot	lb•ft	newton-meter	N•m	1.357
U	Residual unbalance	ounce-inch	oz-in	gram-centimeter	g-cm	72
v	Velocity	foot/second	ft/s	meter/second	m/s	0.3048
w	Width	inch	in	millimeter	mm	25.4
X	Exponent	none	none	none	none	1
Z	Elevation gauge distance above or below datum	foot	ft	meter	m	0.3048

[a] Δ is a dimensionless symbol used to indicate a difference. This term takes on the units of the measured or calculated quantity associated with the difference.

[b] Where US cutomary units are ft, US gpm, and rpm, then the corresponding metric units are m, m^3/s, and rpm.

Table 12.2b — Subscripts

Subscript	Term
a	Absolute
atm	Atmospheric
b	Barometric
d	Discharge

Subscript	Term
min	Minimum
mot	Motor
OA	Overall unit
ot	Operating temperature

Subscript	Term
sol	Solid
stp	Condition at which the particles stop moving along the bottom of the pipe
t	Theoretical
v	Velocity (and volume in C_v)

Table 12.2b — Subscripts (*continued*)

Subscript	Term	Subscript	Term	Subscript	Term
g	Gauge	s	Suction (and solid particles in Ss)	vp	Vapor pressure
m	Slurry (mixture)	sg	Specific gravity	w	Water (and weight in C_w)
max	Maximum	smax	Slurry concentration corresponding to highest deposit velocity		

12.2.1 Rate of flow (*Q*)

The rate of flow of a pump is the total volume throughput per unit of time at suction conditions. It assumes no entrained gases at the stated operating conditions. The terms *flow rate* and *capacity* are also used. Preferred units are cubic meters/hour (m^3/h) and US gallons/minute (US gpm).

12.2.2 Speed (*n*)

The number of revolutions of the pump or driver shaft in a given unit of time. Speed is expressed as revolutions per minute (rpm).

12.2.3 Head (*h*)

Head is a measure of the energy content of the liquid expressed in meters (feet) of the liquid column.

12.2.3.1 Gauge head (h_g)

The energy of the liquid due to its pressure as determined by a pressure gauge or other pressure-measuring device. Negative pressure or vacuum readings can also be expressed in millimeters of mercury (mm Hg) and inches of mercury (in. Hg).

12.2.3.2 Velocity head (h_v)

The kinetic energy of the liquid at a given cross section. Velocity head is expressed by the following equation:

$$h_v = \frac{v^2}{2g}$$

Where v is obtained by dividing the flow by the cross-section area of the pipe at the point of gauge connection; v = velocity, m/s (ft/s); and g = gravitational acceleration, m/s^2 (ft/s^2).

(Metric)

$$v = \frac{278 \times Q}{A} \text{ m/s} \quad Q = m^3/h \quad A = mm^2$$

(US customary units)

$$v = \frac{0.3205 \times Q}{A} \text{ ft/s} \quad Q = \text{US gpm} \quad A = \text{in}^2$$

12.2.3.3 Elevation head (*Z*)

The potential energy of the liquid due to its elevation relative the pump's datum expressed in meters (feet) of liquid column.

12.2.3.4 Datum

The pump's datum is a horizontal plane that serves as the reference for head measurements. For horizontal pumps this datum is considered by convention to be the centerline of the impeller. The datum for single suction vertical pumps is the eye of the impeller and for double suction pumps is the plane through the upper eye, as shown in Figure 12.2.3.4.

Irrespective of pump mounting, the pump's datum is maintained at the eye of the first-stage impeller.

12.2.3.5 Total suction head (h_s), open suction

For open suction wet pit installations, the impeller is submerged in a pit. The total suction head (h_s) at the datum is the submergence (Z_w).

If the average velocity head of the flow in the pit is small enough to be neglected, then:

$$h_s = Z_w$$

Where:

Z_w = Vertical distance in meters or feet from free water surface to the pump datum.

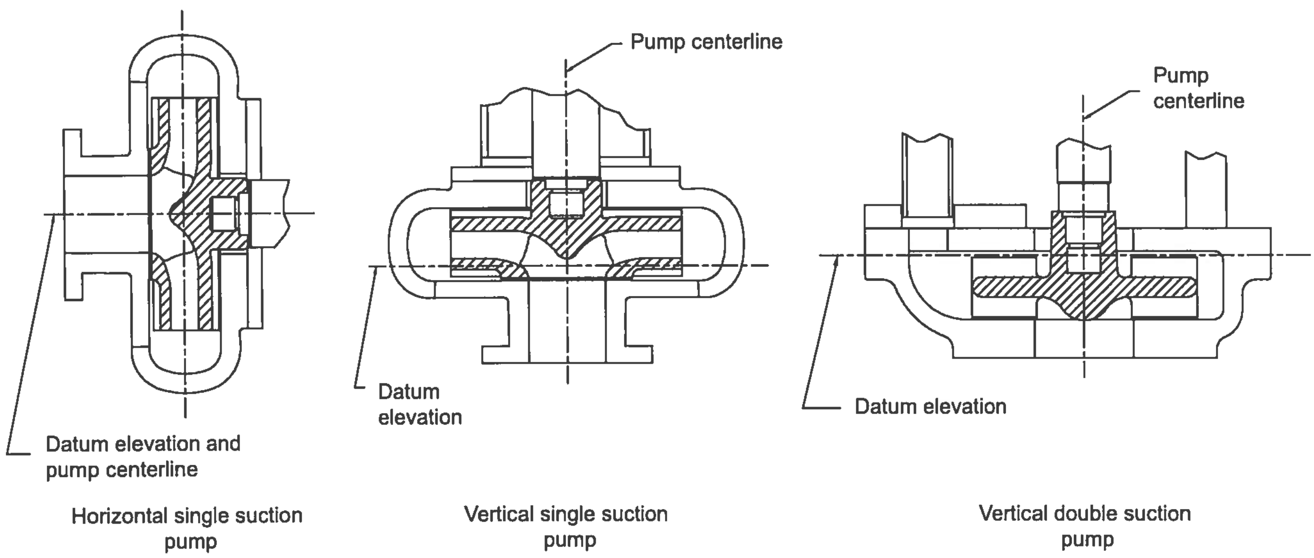


Figure 12.2.3.4 — Datum elevations for various slurry pump designs

12.2.3.6 Total suction head (h_s)

Total suction head (h_s), relative to the eye of the first-stage impeller, is the algebraic sum of the suction gauge head (h_{gs}) plus the velocity head (h_{vs}) at point of gauge attachment, plus the elevation head (Z_s) from the suction gauge centerline (or manometer zero) to the pump datum:

$$h_s = h_{gs} + h_{vs} + Z_s$$

The suction head (h_s) is positive when the suction gauge reading is above atmospheric pressure and negative when the reading is below atmospheric pressure, by the amount exceeding the sum of the elevation head and the velocity head.

12.2.3.7 Total discharge head (h_d)

Total discharge head (h_d) is the sum of the discharge gauge head (h_{gd}) plus the velocity head (h_{vd}) at the gauge attachment point, plus the elevation head (Z_d) from the discharge gauge centerline to the pump datum, where Z_d is positive if the gauge is above the pump datum.

$$h_d = h_{gd} + h_{vd} + Z_d$$

12.2.3.8 Pump total head (H)

Pump total head (H) is the difference between the total discharge head (h_d) and the total suction head (h_s). This is the head normally specified for pumping applications because the complete characteristics of a system determine the total head required.

$$H = h_d - h_s$$

12.2.3.9 Atmospheric head (h_{atm})

Local atmospheric pressure expressed in meters or feet of liquid being pumped column.

12.2.3.10 Friction head (h_f)

Friction head is the hydraulic energy required to overcome frictional resistance of a piping system to liquid flow, expressed in meters or feet of liquid column.

12.2.4 Condition points

12.2.4.1 Rated condition point

Rated condition point applies to the capacity, head, net positive suction head, and speed of the pump, as specified by the order.

12.2.4.2 Specified condition point

Specified condition point is the same as rated condition point.

12.2.4.3 Normal condition point

Applies to the point on the rating curve at which the pump will normally operate. It may be the same as the rated condition point.

12.2.4.4 Best efficiency point (BEP)

The flow rate and head at which the pump efficiency is the maximum at a given speed and impeller diameter.

12.2.4.5 Shutoff

The condition of zero flow where no liquid is flowing through the pump, but the pump is primed and running.

12.2.4.6 Allowable operating range

The flow range of a slurry pump at the specified speeds, as limited by cavitation, vibration, noise, heating, shaft deflection, and fatigue, and taking into account wear considerations (see Section 12.3.4). This range shall be defined by the manufacturer.

12.2.5 Suction conditions

12.2.5.1 Submerged suction

A submerged suction exists when the centerline of the pump inlet is below the level of the liquid in the supply tank.

12.2.5.2 Net positive suction head available (NPSHA)

Net positive suction head available is the absolute suction head of liquid, determined at the first-stage impeller datum, minus the absolute vapor pressure of the liquid at inlet process conditions, expressed in meters or feet of mixture:

$$\text{NPSHA} = h_{sa} - h_{vp}$$

Where:

$$\text{Absolute suction head} = h_{sa} = h_{atm} + h_s$$

$$\text{or NPSHA} = h_{atm} + h_s - h_{vp}$$

12.2.5.3 Net positive suction head required (NPSHR, NPSH3)

NPSH required (NPSHR) by a pump is the value of NPSH available (NPSHA) needed to operate the pump at a specific rate of flow. This value is normally recommended by a pump vendor.

NPSH3 is the value of NPSHR that results in a 3% loss of head (first-stage head in a multistage pump) determined by the vendor by testing with water.

12.2.5.4 Maximum suction pressure

This is the highest suction pressure to which the pump will be subjected during operation.

12.2.6 Power

12.2.6.1 Electric motor input power (P_{mot})

The electrical input power to the motor, normally defined by reading voltage and current and expressed by calculating kilowatts.

12.2.6.2 Pump input (shaft) power (P_p)

The power delivered to the pump shaft from the electric motor or other driver. It is also called *brake horsepower* and is expressed in kilowatts or horsepower.

12.2.6.3 Pump output (useful) power (P_u)

The power imparted to the liquid by the pump.

(Metric)

$$P_u = \frac{Q \times H \times s}{367.1} \text{ kW}$$

(US customary units)

$$P_u = \frac{Q \times H \times s}{3960} \text{ bhp}$$

12.2.6.4 Pump efficiency (η_p)

This is the ratio of the energy imparted to the liquid by the pump (P_u) to the energy delivered to the pump shaft (P_p), expressed in percent.

$$\eta_p = \frac{P_u}{P_p}$$

12.2.6.5 Overall efficiency (η_{OA})

This is the ratio of the energy imparted to the liquid (P_u) by the pump, to the energy supplied to the motor (P_{mot}), or the ratio of the water horsepower to the power input to the primary driver, expressed in percent.

$$\eta_{OA} = \frac{P_u}{P_{mot}}$$

12.2.7 Pump pressures

12.2.7.1 Working pressure (p_d)

The maximum discharge pressure in the pump when it is operated at rated speed and suction pressure for the given application.

12.2.7.2 Maximum allowable casing working pressure

The highest pressure, at a specified pumping temperature, for which the pump casing is designed. This maximum pressure shall be equal to or greater than the maximum discharge pressure.

12.2.7.3 Test pressure

The hydrostatic pressure applied to demonstrate that the pump, when subjected to hydrostatic pressures, will not leak or fail structurally as defined in ANSI/HI 14.6 *Rotodynamic Pumps - Hydraulic Performance Acceptance Tests.*

12.2.8 Mechanical seal terms

12.2.8.1 Pusher seal

Seal design where the secondary seal in the axially flexible assembly slides on the pump shaft or cartridge seal sleeve to compensate for wear and misalignment. The most common pusher seal secondary seal is an elastomeric O-ring.

12.2.8.2 Nonpusher seal

Seal design where the secondary seal in the axially flexible assembly is not required to slide in contact with the pump shaft or cartridge seal sleeve to compensate for wear and misalignment. Elastomeric bellows and metal bellows seals are examples of nonpusher type seals.

12.2.8.3 Dual seal

Seal design using two or more axially flexible assemblies. The inboard seal of a dual seal arrangement seals the product, and the outboard seals a buffer/barrier fluid.

12.2.8.4 Dual pressurized seal

Dual seal arrangement that has a secondary fluid in the outer seal cavity, termed a *barrier fluid*, at a pressure greater than the product pressure in the pump seal chamber. Dual pressurized seals were previously called *double seals*.

12.2.8.5 Dual unpressurized seal

Dual seal arrangement that has a secondary fluid in the outer seal cavity, termed a *buffer fluid*, at a pressure lower than the product pressure in the pump seal chamber. Dual unpressurized seals were previously called *tandem seals*.

12.2.8.6 Buffer fluid

An externally supplied fluid at a pressure lower than the pump seal chamber to lubricate the outer seal in a dual seal arrangement. The buffer fluid creates a buffer between the product pumped and atmosphere.

12.2.8.7 Barrier fluid

An externally supplied fluid used in the outer seal cavity of a dual seal arrangement at a pressure greater than that in the pump seal chamber, creating a barrier between the product pumped and atmosphere to eliminate product leakage to atmosphere.

12.2.8.8 External flush fluid

A fluid from an external source, not pumped fluid, which is introduced into the stuffing box or seal chamber to cool and lubricate the seal faces. Sometimes termed an *external injection* and designated by ANSI Plan No. 7332.

12.2.8.9 Secondary seal

A device, such as an O-ring, elastomeric, or metal bellows, that prevents leakage around the primary sealing faces of a mechanical seal. The term *secondary seal* also refers to static seals, such as O-rings or gaskets, used in ancillary components to prevent leakage from a high-pressure area to a low-pressure area.

12.2.8.10 Single seal

A seal arrangement with only one mechanical seal per stuffing box or seal chamber.

12.2.9 Slurry terminology

12.2.9.1 Slurry

A mixture consisting of solid particles (solids) dispersed in a liquid.

12.2.9.2 Apparent viscosity

The viscosity of a non-Newtonian slurry at a particular rate of shear, expressed in terms applicable to Newtonian fluids.

12.2.9.3 Minimum carrying velocity

The velocity of the specific slurry in a particular conduit, above which the solids remain in suspension, and below which solid–liquid separation occurs.

12.2.9.4 Mean effective particle diameter, or average particle size (d50)

The single particle size used to represent certain behavior of a mixture of various sizes of particles in slurry. This particle size is where 50% by weight passes through a designated size screen. The d50 size is normally specified in micrometers (μm) but may also be in other units, such as the Tyler Mesh as shown in Figure 12.2.9.4. This d50 designation is used by some engineers to calculate system requirements and pump performance. Figure 12.2.9.4 shows how it may be used to classify slurries and provides different d50 size equivalents.

12.2.9.5 Solids d85 size

The particle size where 85% by weight passes through a designated size screen. The d85 size is normally expressed in micrometers (μm), but may also be in other units, such as the as the Tyler Mesh, as shown in Figure 12.2.9.4.

12.2.9.6 Maximum particle size

The maximum particle size expected in the slurry, under normal conditions, that has to pass though the pump.

12.2.9.7 Friction characteristic

A term used to describe the resistance to flow that is exhibited by solid–liquid mixtures moving at various rates of flow in pipes or conduits.

12.2.9.8 Heterogeneous mixture

A mixture of solids and a liquid in which the solids are not distributed uniformly and tend to be more concentrated in the bottom of the pipe.

12.2.9.9 Homogeneous mixture

A mixture of solids and a liquid in which the solids are distributed uniformly.

12.2.9.10 Homogeneous flow (fully suspended solids)

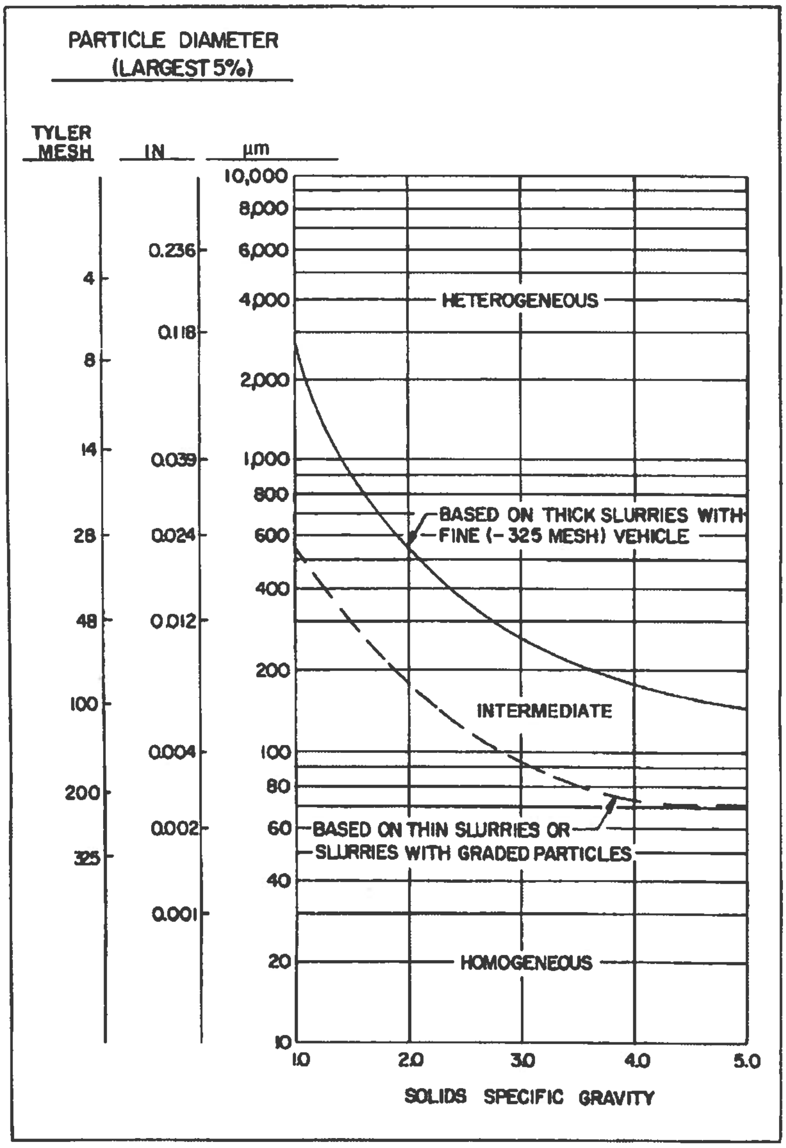
A type of slurry flow in which the solids are thoroughly mixed in the flowing stream and a negligible amount of the solids are sliding along the conduit wall.

12.2.9.11 Nonhomogeneous flow (partially suspended solids)

A type of slurry flow in which the solids are stratified, with a portion of the solids sliding along the conduit wall. Sometimes called *heterogeneous flow* or *flow with partially suspended solids.*

12.2.9.12 Nonsettling slurry

A slurry in which the solids do not settle to the bottom of the containment vessel or conduit but remain in suspension, without agitation, for long periods of time.



SLURRY FLOW REGIME (HETEROGENEOUS, HOMOGENEOUS) IS A FUNCTION OF SOLIDS SIZE AND SPECIFIC GRAVITY.

Figure 12.2.9.4 — Schematic classification of slurries in industrial pipeline applications

12.2.9.13 Concentration of solids by volume (C_v)

The actual volume of the solid material in a given volume of slurry, divided by the given volume of slurry, multiplied by 100, and expressed in percent.

12.2.9.14 Concentration of solids by mass or weight (C_w)

The mass (or weight) of dry solids in a given volume of slurry, divided by the total mass (or weight) of that volume of slurry, multiplied by 100, and expressed in percent.

12.2.9.15 Saltation

A condition that exists in a moving stream of slurry when solids settle in the bottom of the stream in random agglomerations that build up and wash away with irregular frequency.

12.2.9.16 Settling slurry

A slurry in that the solids move to the bottom of the containment vessel or conduit at a discernible rate but that remain in suspension if the slurry is agitated constantly.

12.2.9.17 Settling velocity

The rate at which the solids in slurry fall to the bottom of a container of liquid that is not in motion. (Not to be confused with the deposit velocity of slurry.)

12.2.9.18 Deposit velocity (V_{stp})

Deposit velocity, or velocity at the limit of stationary deposition, is the velocity at which particles form a stationary bed in a moving slurry mixture at a given concentration.

12.2.9.19 Maximum value of deposit velocity (V_{smax})

The deposit velocity, at the slurry concentration, that results in the highest value of V_{stp}.

12.2.9.20 Specific gravity of solids (S_s)

The relative density of solids to that of water. The specific gravity of various materials is shown in Appendix D.2.

12.2.9.21 Specific gravity of slurry (S_m)

The relative density of the slurry (mixture) to that of water at the same temperature.

12.2.9.22 Water head (H_w)

Head developed when pumping water, expressed in meters or feet of water column.

12.2.9.23 Slurry head (H_m)

Head developed when pumping slurry, expressed in meters or feet of slurry column.

12.2.9.24 Head ratio (H_r)

Head ratio is the ratio of the head developed in meters or feet of slurry compared with the head developed on water.

$$H_r = \frac{H_m}{H_w}$$

12.2.9.25 Efficiency ratio (η_r)

Ratio of the efficiency realized while pumping a given slurry mixture divided by the efficiency achieved while pumping water, $\eta_r = \eta_m/\eta_w$.

12.2.9.26 Head reduction factor (R_h)

Decimal expression of the value of 1 minus the head ratio (H_r), as $R_h = 1 - H_r$.

12.2.9.27 Efficiency reduction factor (R_η)

Equal to the value of 1 less the efficiency ratio η_r, expressed as $R_\eta = 1 - \eta_r$.

12.2.9.28 Water efficiency (η_w)

Efficiency achieved when pumping clear water, expressed in percent.

12.2.9.29 Water power (P_w)

Power required to pump water, expressed in kilowatts or horsepower.

12.2.9.30 Slurry efficiency (η_m)

Efficiency realized when pumping a given slurry, expressed in percent.

12.2.9.31 Slurry power (P_m)

Power required to pump a slurry mixture, expressed in kilowatts or horsepower (analogous to P_w).

12.2.9.32 Specific gravity correction factor (C_{sg})

Correction applied to head reduction factor (R_h) to correct for slurries with solids of specific gravity other than 2.65 ($S_s \neq 2.65$).

$$C_{sg} = \left(\frac{S_s - 1}{1.65}\right)^{0.65}$$

12.2.9.33 Fine-particle correction factor (C_{fp})

Correction applied to head reduction factor (R_h) to correct for slurries containing particles of less than 75 μm.

$$C_{fp} = (1 - \text{fractional content of particles by weight} < 75\ \mu\text{m})^2$$

12.2.9.34 Concentration correction factor (C_{cv})

Correction applied to head reduction factor (R_h) to correct for slurries with concentration factor (C_v) other than 15%.

$$C_{cv} = \frac{C_v\%}{15}$$

12.2.9.35 Stationary bed

A nonmoving bed of stationary solids particles, on the bottom of a flowing pipeline, with carrier liquid passing over the top of the bed.

12.2.9.36 Laminar region

The region of flow and mean velocity inside a pipe, where the internal viscous fluid forces predominate over the inertial fluid forces.

12.2.9.37 Turbulent region

The region of flow and mean velocity inside a pipe, where the inertial fluid forces predominate over the viscous fluid forces.

12.2.9.38 Transition region

The region of flow and mean velocity inside a pipe, where the inertial and viscous fluid forces are approximately equal.

12.2.9.39 Slurry service classes

Class 1: light, Class 2: medium, Class 3: heavy, Class 4: very heavy. These are detailed in Section 12.3.4.2 Pump wear and Section 12.3.5 Hydraulic design and application considerations.

12.2.9.40 Impeller seal face

A part of the impeller, typically near the suction eye, designed to create a close clearance with a stationary part of the pump, such as the suction liner, to limit internal recirculation.

12.2.9.41 Slurry abrasivity

The tendency of a particular moving slurry to produce abrasive and erosive wear.

12.2.9.42 Abrasive wear

Wear due to hard particles or hard, small surface protrusions forced against and moving along a material surface.

12.2.9.43 Abrasion–corrosion

A synergistic process involving both abrasive wear and corrosion. Each of these processes is affected by the simultaneous action of the other, which may accelerate the overall wear rate.

12.2.9.44 Corrosion

Loss of material created by chemical or electrochemical reaction within the pump environment.

12.2.9.45 Erosion

Progressive loss of material from a material surface due to mechanical interaction between that surface and a fluid, a multicomponent fluid, or impinging liquid or material particles.

12.2.9.46 Erosion–corrosion

A loss of material due to both erosion and corrosion, in which each of these processes is affected by the simultaneous action of the other, and, in many cases, is thereby accelerated.

12.2.9.47 Miller number

A measure of slurry abrasivity as related to the instantaneous rate of mass loss of a standard metal wear block at a specific time on the cumulative abrasion–corrosion time curve as defined in *ASTM Standard G75-01*.

12.2.9.48 SAR number

A measure of the relative abrasion response of any material in any slurry, as related to the instantaneous rate of mass loss of a specimen at a specific time on the cumulative abrasion–corrosion time curve, converted to volume or thickness loss rate as defined in *ASTM Standard G75-01*.

12.2.9.49 Specific energy (E_{sp})

The erosive energy of particles in J/m^3 (lb•ft/in^3) required to remove a unit volume of the target wear material.

12.2.9.50 Wear coefficient (W_c)

The volume of the target wear material removed for a given unit particle energy in m^3/J (in^3/lb•ft).

12.3 Design and application

12.3.1 Scope

The purpose of this section is to outline the minimum design requirements for slurry pumps and to provide guidelines for their application.

Slurry pumps are similar to other rotodynamic pumps. Refer to ANSI/HI 1.1-1.2 *Rotodynamic (Centrifugal) Pumps for Nomenclature and Definitions* for general definitions, nomenclature, design application, installation, operation, maintenance, and testing requirements.

Slurry pump design and application differences determine the wear performance and the ability to effectively pump solids. This section concentrates on design and application considerations unique to slurry pumps, and presumes that background knowledge exists relative to rotodynamic pumps in general.

Equipment data sheets, suitable for use by suppliers and users, are referenced in Appendix A and Appendix B.2, reference 25. These sheets should be used to specify the user's duty and equipment needs.

12.3.2 Slurry services

12.3.2.1 Slurry applications

Pumps are used to move mixtures of liquids and solids in many industries. Typical uses for slurry pumps include cleaning, processing, drilling, and transport.

In cleaning applications such as flue gas scrubbers, large volumes of water are used to capture chemicals and some solids in an exhaust stream. These slurries are generally of fairly low concentration and the challenges focus on material selection and shaft sealing.

Fertilizer production processing applications involve solids as a contaminant that must be removed from the final product. These mixtures can contain highly corrosive carriers and higher concentrations of solids complicating material selection and resulting in additional problems, such as settling and plugging pipes.

Drilling applications involve injection of slurries into wells and the removal of earth and rock cuttings generated at the drill face. These slurries are typically not corrosive, but proper material must be selected for erosion resistance.

The purpose of transport service is to move the maximum volume of solids economically. These services include dredging, the movement of ore, rock, oil sands, or matrix into a mine, mill, separation or wash plant; the transport of the product and waste through the plant; and final disposal of the tails and fine waste. In general, pumping slurries can be an economical method to transport solid particles in the volumetric concentration range of 15–40%.

The most suitable transport concentration depends on the characteristics of the slurry and the application in which it is used. Due to the large size and high concentrations of solids, settling, pipe plugging, increased pipe friction, and component wear become major concerns.

Figure 12.3.2.1 provides a convenient tool to correlate transport rate and pipe velocity to assist the designer in sizing the pump/piping system to optimum cost-effectiveness. Here, a horizontal marker line from the transport rate should be drawn to the solids specific gravity S_s (normally 2.65) graph line, then vertically to the concentration by volume graph line. Then the marker line should be drawn horizontally to the appropriate pipe diameter graph line and then vertically, to read the mean pipeline velocity, or horizontally, to get the mean flow rate.

12.3.2.2 Characteristics of slurries

Usually slurry concentrations are discussed in volumetric terms. This alleviates the variables particular to a given slurry. However, within a given industry or field, slurry concentrations are often discussed in terms of the concentration by weight or the specific gravity of the mixture. The relationship between these different measurements is shown in Figure 12.3.2.2.

12.3.2.3 Slurry types

Depending mostly on the size of the particles, slurries tend to be classified as settling or nonsettling. Nonsettling slurries act in a homogeneous manner and, in most cases, exhibit non-Newtonian characteristics at high concentration.

Settling slurries, depending mostly on the size of the particles, can form a stationary bed and flow in a stratified heterogeneous manner. Depending on the specific gravity and the size of solids, there are also slurries that may be either heterogeneous or homogeneous, forming an intermediate type depending on the actual concentration and the presence of any fine-sized clays. Slurries composed of mostly large-sized particles may even move as a sliding bed. Figure 12.2.9.4 gives a guide to the slurry types and flow mechanism.

12.3.2.4 Settling slurries

For every settling slurry, there is a deposit velocity (V_{stp}) at which solids will drop out of suspension and form a bed on the bottom of the pipe. A pumping system (pump and piping) must be sized and operated so that the velocity in the pipe exceeds V_{stp} or the pipe will plug. Therefore, a system must be designed for the lowest acceptable value of settling velocity.

V_{stp} is dependent on pipe size, particle size, concentration, and specific gravity of the solids. If pipe size, particle size, and specific gravity are assumed constant, V_{stp} varies with concentration. It is lowest at high concentrations and increases as concentration decreases. The point where V_{stp} reaches a maximum is defined as the maximum

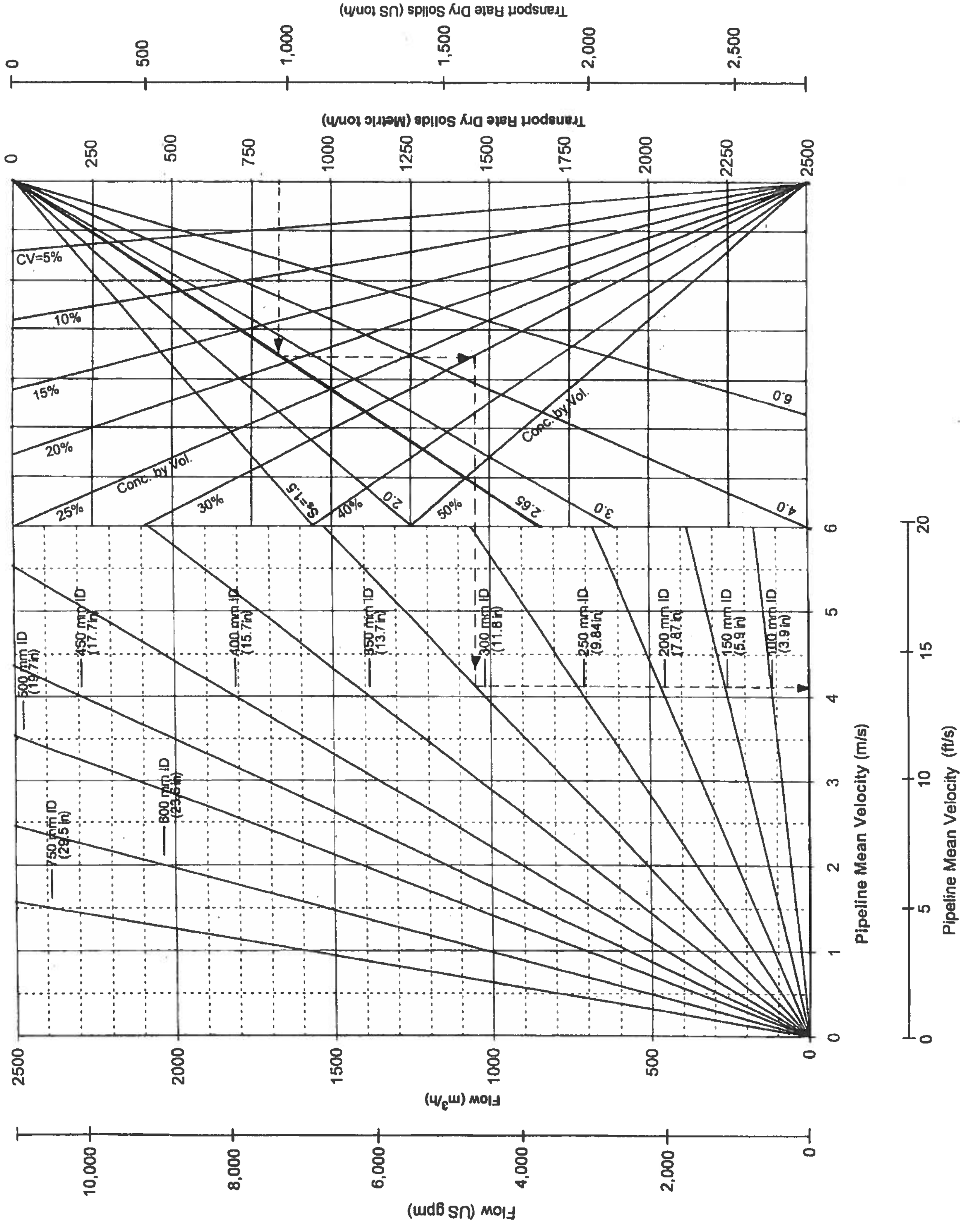


Figure 12.3.2.1 — Solids transport rate

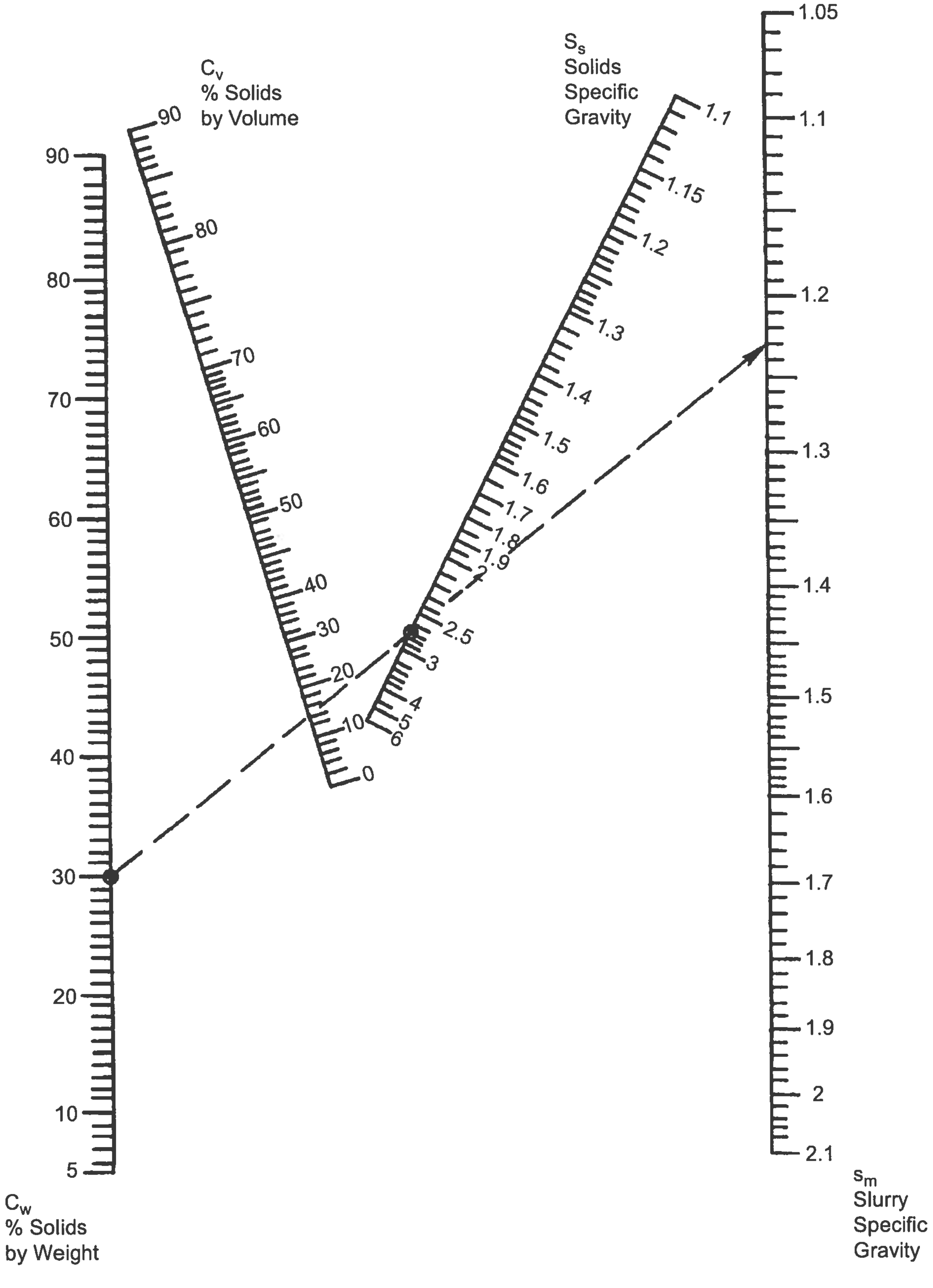


Figure 12.3.2.2 — Nomograph for the relationship of concentration to specific gravity in aqueous slurries

value of deposit velocity or maximum velocity at limit of stationary deposition (V_{smax}). The nomograph in Figure 12.3.2.4 can be used to determine V_{smax}. Because concentration cannot usually be controlled, the pumping system should preferably be designed so that the velocity in the piping always exceeds V_{smax} by at least a 10% margin.

To obtain a value for V_{smax}, draw a straight line on the left-hand half of Figure 12.3.2.4 from the appropriate pipe diameter value over the solids particle diameter line. Where that line intersects, the middle axis gives the value of V_{smax}. In the cases of solids specific gravities other than 2.65, a second line from the center axis V_{smax} value to the right across the relative density value in mind gives a corrected value at the point where that line intersects the far right-hand axis.

Figure 12.3.2.4 can also be used to evaluate the sensitivity of the system to changes in particle size. For example, V_{smax} in a 0.3-m (11.8-in) pipe is virtually unaffected by a variation of particle diameter between 0.4 and 1.0 mm (0.016 and 0.039 in), however, a reduction in particle diameter to 0.15 mm (0.006 in) or an increase to 15 mm (0.59 in) reduces V_{smax} by more than 40%.

If the particle size is not closely controlled, the worst-case particle size (that giving the highest V_{smax}), should be used to determine V_{smax} for design. This will typically be a particle size of 0.4 to 0.6 mm (0.016 to 0.024 in) depending on pipe size. This yields conservatively high values of V_{smax}, especially in large pipe diameters.

12.3.2.5 Effect of slurry on performance

The performance of a centrifugal pump on slurries will differ from the performance on water, which is the basis for most published curves. Head (*H*) and rate of flow (*Q*) will normally decrease as solids size and concentration increases. Power (*P*) will increase and starting torque may also be affected. This "solids effect" is shown schematically in Figure 12.3.2.5 along with the head and efficiency derating terms used.

Effects of solids on a slurry pump cavitation performance are dependent on the slurry type and the pump design and can be highly variable. The value of net positive suction head required in order not to exceed 3% head drop, NPSH3, will increase, in most circumstances.

For settling slurries of low to medium concentration, a modest increase in NPSH3 can be expected. For a particular application, this increase can be conservatively estimated by dividing the value of NPSH3 on water by the head derating factor discussed below.

For viscous and nonsettling slurries or slurries with entrained air, the effect on pump cavitation performance can be significantly greater. The pump manufacturer should be consulted for guidance regarding slurry effects on NPSHR.

Different approaches can be used for predicting the centrifugal pump performance change from water to slurry, depending on the slurry type.

When the solids–fluid mixture as shown in Figure 12.2.9.4 is considered homogeneous, the pump performance viscosity correction methods discussed in Section 12.3.2.6 can be applied. For heterogeneous settling slurries, a method using the solids size and pump impeller diameter is outlined in Section 12.3.2.7. Pumping of frothy slurries is discussed separately in Section 12.3.3.

These are empirical methods based on the best test data available from sources throughout the world. There are many factors for a particular pump geometry and flow conditions that are not taken into account. However, the methods provide for dependable approximations when limited data on the application are available.

Pump users should consult with pump manufacturers for more accurate predictions of performance for a particular pump and particular slurry.

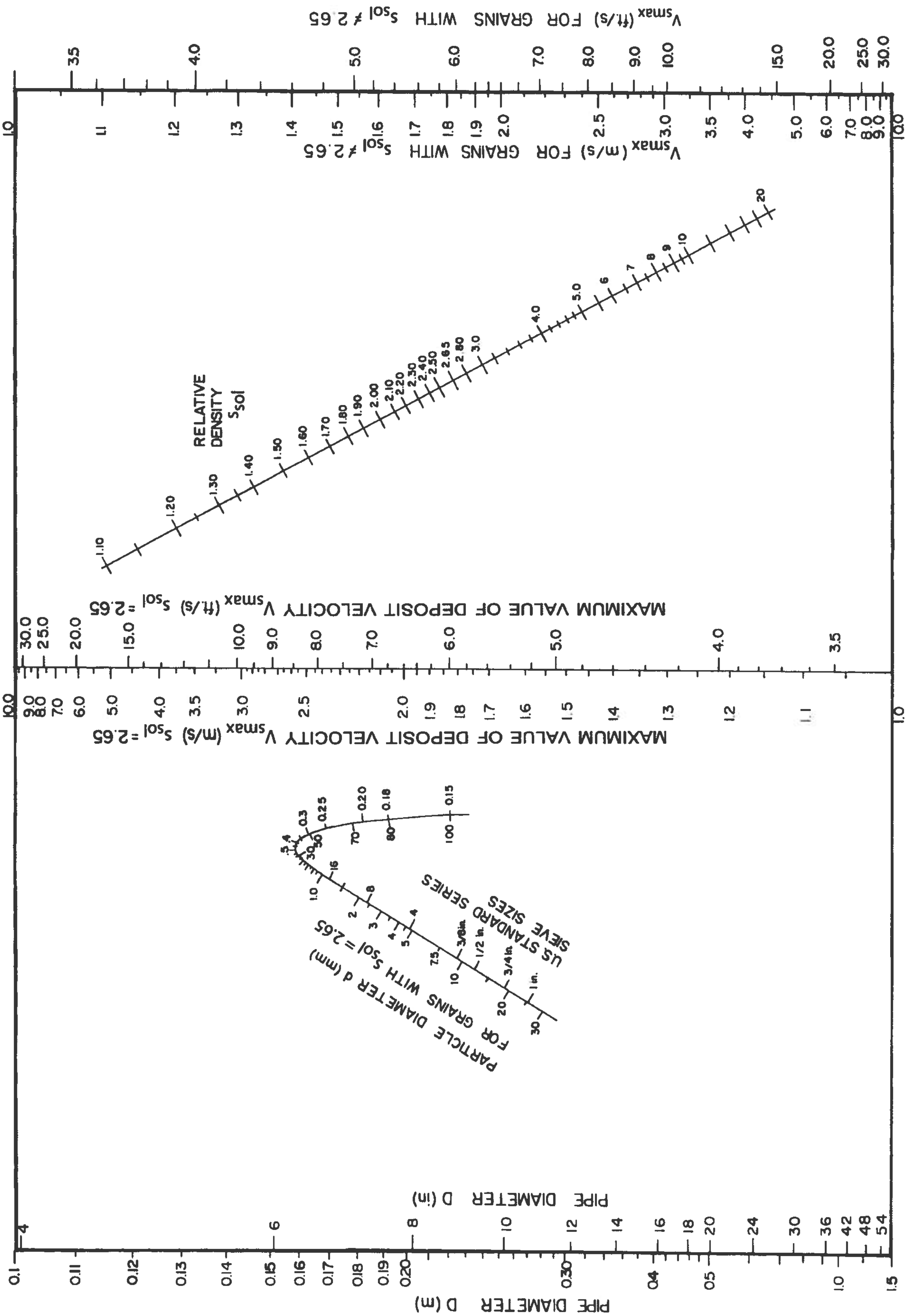
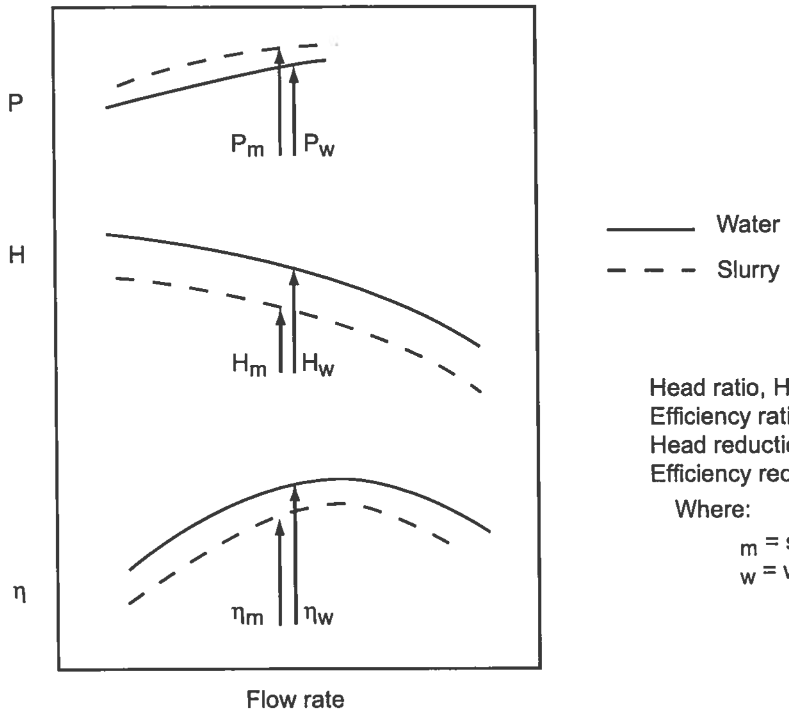


Figure 12.3.2.4 — Nomograph for maximum velocity at limit of stationary deposition of solids



Head ratio, $H_r = H_m/H_w$
Efficiency ratio, $\eta_r = \eta_m/\eta_w$
Head reduction factor, $R_h = 1 – H_r$
Efficiency reduction factor, $R_\eta = 1 – \eta_r$
Where:
m = slurry
w = water

Figure 12.3.2.5 — Effect of settling slurry on pump characteristics (schematic)

12.3.2.6 Performance derating based on viscosity

Newtonian behavior is characterized by a fluid that is free flowing and has constant viscosity regardless of shear rate. The rheological plot of a Newtonian fluid, correspondingly, shows zero yield stress and a linear (straight-line) plot of shear stress versus shear rate. Low concentration, fine-particle slurries often exhibit this behavior.

When the solids–fluid mixture as shown on Figure 12.2.9.4 is considered homogeneous and exhibits Newtonian behavior, the ANSI/HI method for pump performance viscosity correction can be applied as outlined in ANSI/HI 9.6.7 *Effects of Liquid Viscosity on Rotodynamic Pump Performance.* With typical industrial slurries, viscosity is usually 50 to 100 times that of water (at 20 °C [68 °F]), resulting in reductions of 5 to 10% in head and efficiency, depending on the flow rate and head.

Theoretical methods based on loss analysis may provide more accurate predictions of the effects of fluid viscosity on pump performance when the geometry of a particular pump is known in more detail.

At higher concentrations, homogeneous, nonsettling slurries often are not free flowing and correspondingly exhibit a yield stress. The presence of this yield stress results in non-Newtonian behavior with nonconstant apparent viscosity, which varies with shear rate and/or flow. This varying viscosity and the uncertainty associated with determining an appropriate shear rate make the use of ANSI/HI 9.6.7 cumbersome, less accurate, and likely to underestimate the performance derating at low-volume flow. Neglecting a possible severe performance derating at low relative rate of flow could result in a pump performance curve that intersects the system curve at multiple operating points and results in unstable flow. The pump manufacturer should be consulted for guidance regarding non-Newtonian effects on pump performance.

12.3.2.7 Performance derating based on solids size and content

Where the slurry is heterogeneous, Figure 12.3.2.7 can be used to determine the head and efficiency reductions from the original water performance for different sizes of slurry pumps of low to medium specific speeds for a slurry mixture concentration by volume of 15% and with negligible portions of less than 75-µm fines.

For solids of S_{sol} other than 2.65, for concentrations other than 15% by volume, and with significant amounts of fines present, the values of R_h shall be modified by multiplying them by the correction factors C_{sg}, C_{fp}, and C_{cv} noted below, applied concurrently.

Specific gravity correction factor (C_{sg}), where $C_{sg} = [(S_s - 1)/1.65]^{0.65}$ or using Table 12.3.2.7.

Table 12.3.2.7 — Specific gravity correction factor

S_s	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0	3.2	3.4	3.6	3.8	4.0
C_s	0.25	0.40	0.52	0.62	0.72	0.81	0.90	0.98	1.06	1.13	1.21	1.28	1.34	1.41	1.47

Fine-particle correction factor (C_{fp}), where C_{fp} = (1 – fractional content of particles by weight < 75 μm)^2.

Concentration correction factor (C_{cv}), where C_{cv} = (C_v %/15).

For power, it is assumed that the efficiency reduction factor follows the head reduction factor ($R_h = R_\eta$) so that power consumption increases directly with the slurry specific gravity ($P_m = s_m \times P_w$). This assumption is usually conservative on large, heavy-duty slurry pumps, but is adequate to safely size motors. With small pumps and slurries of well over 20% volumetric concentration, the power may be up to 1.5 times larger than the power on water, dependent on the individual properties of the solids.

A pump with a 914-mm (36-in) diameter impeller that produces 61 m (200 ft) of head at 80% efficiency while pumping water, when pumping a 2.65 solids specific gravity slurry of average particle size of 1 mm (0.040 in) at a concentration by volume of 15% will, according to Figure 12.3.2.7, have an R_h of 8% and an H_r of 0.92. The head

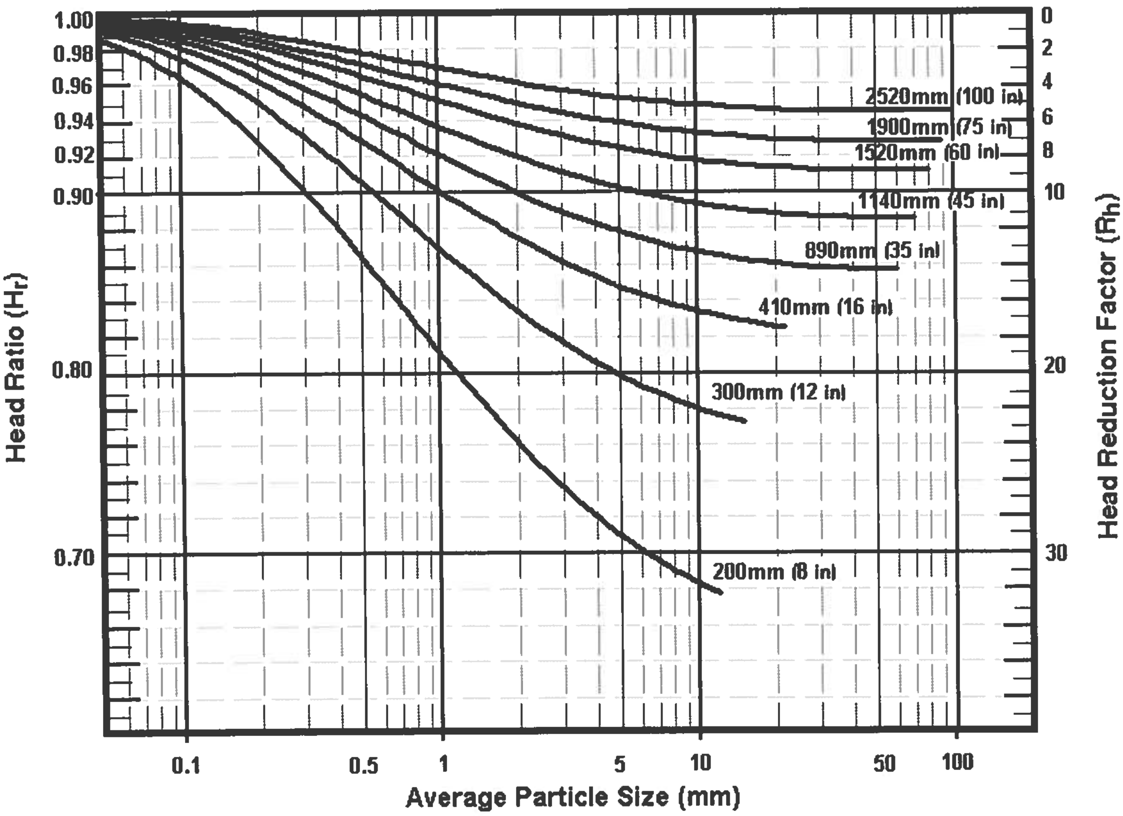


Figure 12.3.2.7 — Effect of average particle size and impeller diameter on H_r and R_h (For solids concentration by volume, C_v = 15% with solids S_s = 2.65 and a negligible amount of fine particles. Impeller diameters are given in millimeters and inches.)

produced while pumping slurry will therefore be 61 m × 0.92 = 56.1 m or 184 ft of mixture. R_η = R_h = 8%, so η_r = 0.92 and the efficiency produced while pumping slurry will therefore be 80% × 0.92 = 73.6%.

If the concentration by volume was 30%, then the R_h value would double to 16% and the head produced would now be 61 × 0.84 = 51.2 m or 168 ft of mixture.

If 20% by weight of the above slurry is less than 75 µm, then the R_h value above would become $R_h = 16 \times (1 - 0.20)^2 = 10.2$ and the head produced would be 61 × 0.898 = 54.8 m or 179 ft of mixture.

If the above slurry (at C_v = 30 and 20% fines) had a solids specific gravity of 3.0, then the R_h value would be 10.2 × 1.13 = 11.5 and the head produced would be 61 × 0.885 = 54 m or 177 ft of mixture.

It should be noted that slurry mixture characteristics may vary significantly depending on the size of the solids, their shape, solids specific gravity, concentration, and carrier liquid, which cannot always be predicted by generalized formulations such as those above. Whenever possible, actual field comparisons should be made and the more sophisticated methods of the different manufacturers, noted in Appendix B, Source Material and References, used.

12.3.3 Froth pumping

Froth is an aerated liquid medium (slurry) that occurs naturally or is created on purpose. Natural occurrence may be due to the nature of the ore processed in the mineral industries, creating a general nuisance in many cases. Froth is created for the purpose of separating minerals floating the product from the waste or vice versa. Froth is created by the aeration of the slurry through air injection during agitation with the addition of polymers to increase the surface tension, creating bubbles to which the product or waste adheres. This allows for the separation and collection of the sought-after mineral for further refining.

The transfer of froths with centrifugal slurry pumps is a special purpose application commonly encountered in the launders of flotation circuits. The very large proportion of air in the froth being handled upsets the normal relationships that are used to predict pumping performance and requires a unique approach in selecting and applying pumps for this service.

It is well known that the presence of air at the suction inlet will decrease the head, flow, and efficiency of a pump and, with increasing amounts of air, the losses will increase. The NPSH required increases with increasing air content and, at a certain critical level, the pump loses prime and stops pumping. Studies done with air injected into the suction of the pump have confirmed this, but the data cannot be applied directly to froth pumping.

Depending on the process, type of slurry, or frothing agents used, a certain amount of air or gas will separate from the froth and can lead to problems with pump performance. The change in performance due to the presence of this air or gas could be quantified based on various factors, such as pump geometry, specific speed, and suction pressure. However, it is practically impossible to determine with reasonable accuracy what amount of free air or gas will separate from the froth at the impeller inlet. This problem requires a special approach in selecting a pump to successfully handle the froth application.

The usual approach is to oversize the pump for the application by use of a "froth factor." The froth factor is a multiplier that increases the process design capacity to allow for the increased passing volume caused by the gas in the froth. The factored volume usually causes the pump to be at least one pipe size larger than would normally be selected.

Oversizing the pump increases its ability to handle multiphase mixture containing air or gas due to the following reasons:

- The larger inlet diameter provides for more physical space to cope with an accumulation of air without restricting the fluid flow into the pump.
- The increased impeller eye diameter requires suction pipe of larger diameter. This leads to reduction in the suction line velocity, thus reducing line losses, and therefore increases the available NPSH of the system.

- The impeller of larger diameter will run at a lower speed of rotation, which will result in reduction of the NPSH required by the pump.

It is important to understand that it is the volume of air at the pump inlet that affects the pump performance. Air volumetric concentration directly depends on suction pressure. For this reason, the froth pumps should always operate with positive suction pressure, and the higher the pressure, the better the pump performance that can be expected.

Although applying a larger pump may be considered a cost implication, this approach has proven to work well in many processes, especially with vertical pumps, applications of which normally fall into the low system head range, with heads seldom over 20 m (65.6 ft). Double suction vertical pumps with semiopen impellers are often applied in this duty. This type of pump is effective because it has over twice the inlet eye area of a conventional pump design and the arrangement allows some air to vent out the top suction inlet.

For practical reasons, such as space or head generation, some installations are better suited to horizontal pumps. When standard horizontal pumps are used, large froth factors are applied, resulting in pumps typically one or two sizes larger than would normally be used for nonaerated duties. Open and semiopen impellers are preferred to closed impellers because of the larger vane passageways and their theoretical ability to handle a larger air bubble before air binding.

Special-purpose horizontal froth pumps are available with increased inlets, flow inducers, and specially profiled vanes to increase air-handling capability.

Where higher heads are required, multiple pumps can be used in series. With increased suction pressure, subsequent stages are expected to experience reduced air volume and improved performance. Testing of conventional horizontal pumps handling air-entrained liquids, however, has still shown significant performance derating at elevated suction pressure, even with relatively low air volume.

Pumps for mineral froth service can be either elastomer-lined or constructed of hard metal to provide abrasion and corrosion resistance. Pumps for bitumen froth duties are typically hard metal, although liners of synthetic elastomers, such as neoprene (see Section 12.3.7.4), are a viable choice, depending on maximum particle size.

Froth collection systems should permit as much gas as possible to escape from the froth before it enters the pump, and some applications will require water sprays to "knock down" the froth to prevent air locking the pump.

The froth factor is normally specified by the pump buyer and is based on previous plant experience. The factors are usually in the range of 1.5 to 4, but can be as high as 8. Many factors influence the size of the froth factor and these may include the viscosity of the liquid, the size of grind of the mineral, and the chemistry used in the process. The type of pump selected will also have an effect on the froth factor used, and the pump manufacturer should be consulted for sizing recommendations. Some typical vertical pump froth factors for common processes are given in Table 12.3.3. These are only approximate values; the most reliable factors will come from the users.

Table 12.3.3 — Approximate froth factors

Application	Pump Froth Factor
Copper rougher concentrates	1.5
Copper cleaner concentrates	3.0
Molybdenum rougher concentrates	2.0
Molybdenum concentrates	3.0
Potash	2.0
Iron concentrates	4.0 to 6.0
Coal	6.0

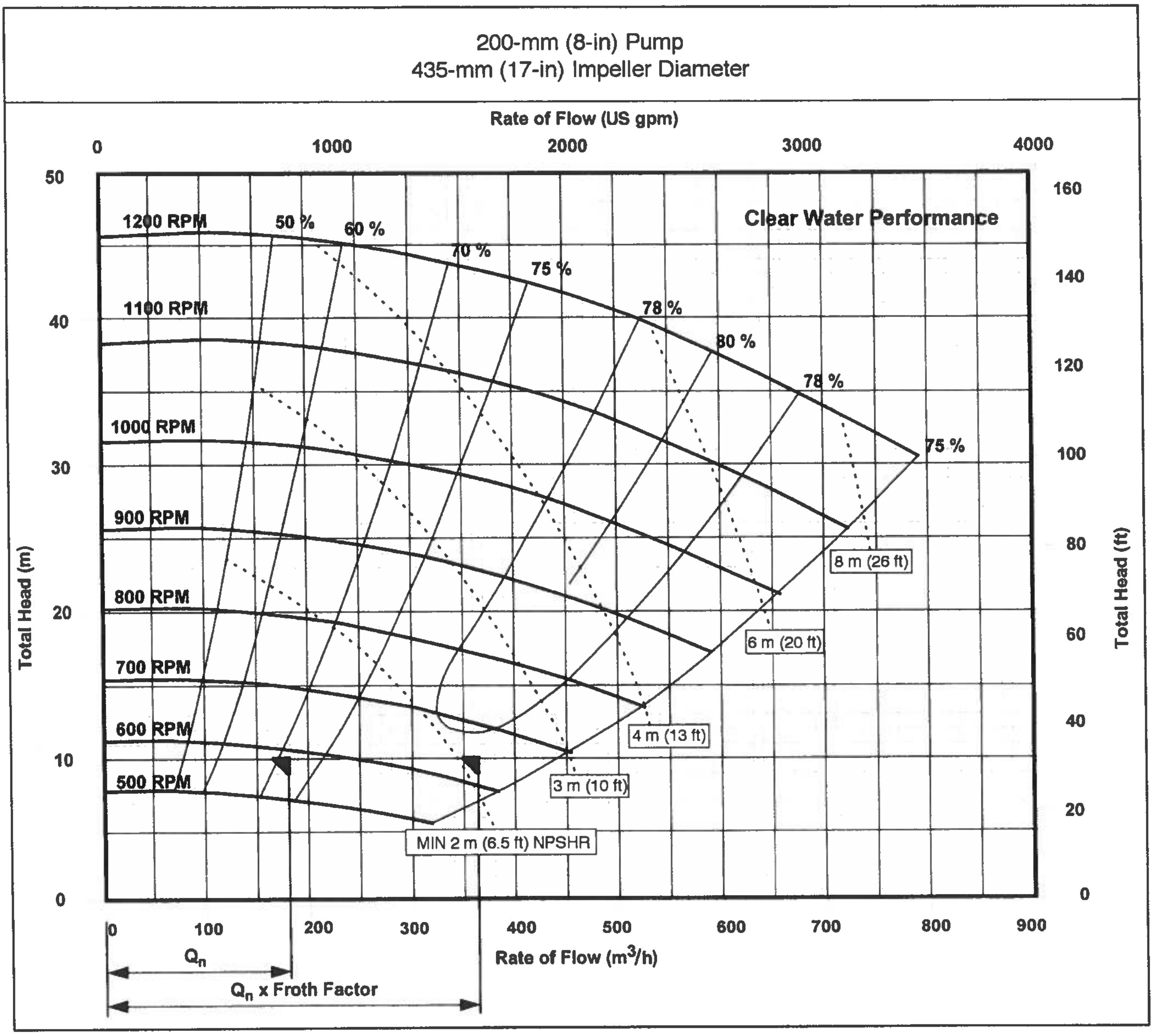


Figure 12.3.3 — Application of froth factor to pump selection

Figure 12.3.3 shows one traditional method in which a froth factor is used to increase the process design capacity and select a conventional pump for froth service. Example: molybdenum rougher concentrates with a design capacity of 182 m^3/h (800 US gpm) at a required head of 10 m (33 ft). A 150-mm (6-in) discharge size pump would normally be selected for this flow, but this example has a froth factor of 2.0. The factored capacity will be 364 m^3/h (1600 US gpm). This flow rate is better suited to a 200-mm (8-in) pump. The original condition point has been plotted on the curve along with the factored condition point. In this method the required head does not change and is simply transferred to the new flow rate. The pump speed will increase from 580 rpm to 640 rpm, as shown in the figure.

The pump drive must be selected to have sufficient power to handle the factored capacity (including froth) at the pipeline (zero froth) specific gravity. This is necessary because there will be times when the system is operated without froth. These conditions will exist at system start-up, after power outages, or when system shut-downs allow the froth to collapse or deaerate over time. It is important to plan for these upset conditions for a successful installation.

1) Obtain the desired operating condition point and the froth factor from the user.

2) Multiply the process design capacity by the froth factor and select an appropriately sized pump for that flow rate.

3) If more than one size impeller is available for the selected pump model, select the largest impeller available to keep the operating rpm at the lowest possible value.

4) Select the pump operating speed at the factored flow rate and the original required head.

5) Size the motor to have enough power to operate at the factored flow rate at the highest expected specific gravity.

Other manufacturer/pump design-specific methods exist that use both air content and system curve review to determine pump selection, performance derating, speed, and expected power draw.

The sump design will affect the performance of the pump. Where froth or air entrainment is a minor annoyance, the sump size should be at least equivalent to one minute of flow (capacity) of the pump in question at its design flow to ensure the sump is not too active.

Additional methods used to enhance the pump performance when handling difficult froths include:

- Misting, via nozzles, to provide a gentle rain-like spray to pierce the froth bubbles
- A partly submerged feed box within the sump to control the entrance of air and slurry and allow for venting in the remainder of the sump
- Inclined plates to separate or baffle the air-entrained slurry feed into the sump from the pump suction and provide additional surface area, a torturous path, additional travel time within the sump, and a dead zone under the baffle, all to enhance liberation of air before reaching the pump suction
- A vent on the suction line to allow air/froth to escape
- A tall sump to maximize suction static height and corresponding suction pressure and reduce air volume at the pump inlet through the relationship $p_1v_1 = p_2v_2$
- The use of eccentric reducers at the sump and pump suction, with the reducer at the sump having the belly side up to vent back into the sump, and the reducer at the pump suction having the belly side down, so as to not trap air in the inlet pipe

12.3.4 Wear in centrifugal slurry pumps

12.3.4.1 Wear considerations

Slurry pumps are usually designed for specific applications. When this involves transporting large solids and/or high concentrations, component wear will be a major factor and must be considered in the selection of the pump and the configuration of the pump installation.

The major slurry erosive mechanisms inside a pump are sliding abrasion and particle impact. Sliding abrasion typically involves a bed of particles bearing against a surface and moving tangentially to it. Impact wear occurs where particles strike the wearing surface at an angle.

Abrasive wear varies with the number of particles or volume concentration of the solids, the velocity of the eroding particles to a power of 2.5 to 3, the abrasivity of the eroding solids, and the wear resistance of the surface being impacted.

The specific energy E_{sp} in J/m^3 (lb•ft/in^3) is defined as the erosive energy of particles required to remove a unit volume of the target wear material. The reciprocal of E_{sp} is referred to as the *wear coefficient* (W_c). The larger the value of E_{sp}, the lower the expected wear rate for identical slurry flow conditions. A specific energy may be empirically determined for either sliding wear or impact wear.

In computing a sliding wear rate, the friction power of the slurry layer adjacent to the wear surface is estimated from the solid–liquid flow field. The friction power is simply the product of wall shear stress and the velocity tangential to wear surface. A computational fluid dynamics calculation may be essential for determining these in pump components. The friction power (at any location on the wetted surface) divided by the sliding wear specific energy E_{sp} yields the local wear rate at that position. Note that the friction power intrinsically includes the effect of local concentration and particle size.

The wall shear stress may be approximated in terms of the local volumetric concentration of particles, material density of the particles, and the solids tangential velocity. In such a case, the friction power is computed as the product of the local concentration, particle density, the third power of the tangential velocity of the solids, and a multiplicative factor. The multiplicative factor is, for convenience, absorbed into the definition of a modified specific energy E_{sp} (or its reciprocal, W_c). In many practical computations, this simpler approach is found to yield quite reliable results.

In the case of impact wear, the specific energy E_i is a function of the angle of impact α. For cast-iron alloys, normal impact at right angles to the surface gives the lowest E_i (highest wear rate). A number of impact wear specimens with different angles of the wear test wedge pieces are tested to characterize E_i as a function of α. Similar to the friction power, the impact power carried by the particles is proportional to the particle density, concentration, and approximately the cube of the particle velocity. Dividing the impact power by E_i (for the specific angle of impact), the impact wear rate is computed.

Theoretically, for a given slurry/wear surface combination, E_{sp} and E_i (α) are expected to be constant. Experiments indicate that particle size plays an important role in determining E_{sp} and E_i (α).

Figure 12.3.4.1a shows examples for different sand particle sizes against different resisting materials for sliding abrasion in a neutral pH medium. The sliding wear coefficient (W_c) values shown in Figure 12.3.4.1a are used on the modified definition (including the approximation to the wall shear stress and the attendant multiplicative factor) as outlined in the foregoing. The values shown in Figure 12.3.4.1a will vary with different types of solids, hardness, solids specific gravity, and sharpness of particles. The abrasivity of a particular slurry will be used as a measure of this difference.

ASTM Standard G75-01 presents details of characterizing the abrasivity of slurries using the Miller test. The Miller number helps rank the abrasivity of the slurries in terms of the wear of a standard reference material. The higher the Miller number, the greater is the wear on the standard Miller test specimen, and hence the greater is the slurry abrasivity. The Miller test apparatus consists of a reciprocating arm with the wear specimen attached to it. The wear specimen thus reciprocates in the slurry of known concentration. The test is run in three two-hour stages with different orientations of the wear specimens. In the typical test, two wear samples reciprocating in separate slurry trays (made of plastic) are used to ensure reliable test data. The cumulative mass loss on each wear sample is recorded after each two-hour duration of testing. The data points are fitted to a power-law curve to obtain the erosion rate.

In practice, slurries of 50% concentration (by weight) are used. It is found that slurries of higher concentration yield essentially the same Miller number. Apart from slurry concentration, the hardness, size, and shape of particles are important. In addition, the corrosivity of the slurry can significantly affect the Miller test.

The effect of corrosion can be isolated by neutralizing the slurry carrier, rerunning the test, and comparing the results. If the Miller number drops significantly, the corrosion effect is dominant. The pump materials may then be appropriately chosen to minimize corrosion.

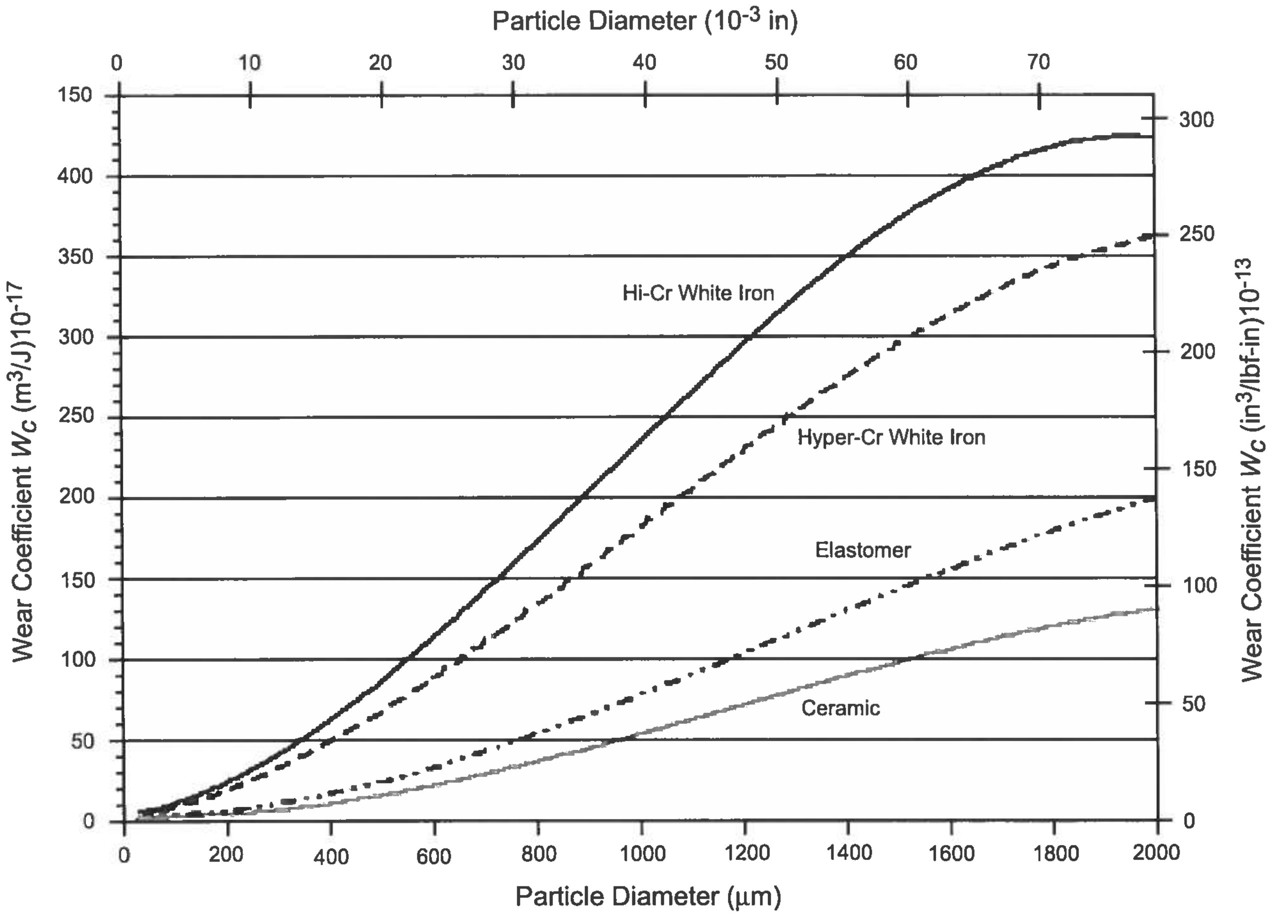


Figure 12.3.4.1a — Wear coefficient W_c for different resisting materials in a neutral pH media for different average-sized abrading particles

Slurries with a Miller number of 50 or lower can usually be pumped with minor erosive damage to the pumping system. Such slurries may be classified as "light duty."

For impact wear, the erosion rate varies for different materials and solids impact angles. Figure 12.3.4.1b shows how ductile (elastomer) and brittle (white iron) materials respond.

It is possible to model the slurry particle velocities within the main components and, using the wear coefficients noted, calculate the wear. Modeled values for wear are for specific geometries and conditions. The uncertainties associated with the modeling are such that it is not suitable for use in a standard.

The foregoing is provided as an overview of the factors associated with slurry pump internal component wear and an insight into the relative material and other effects. For slurry pump selection, a wear service class method along with various limiting velocities (described later) is used in this standard, in conjunction with field experience with specific slurries.

Slurries may be corrosive in chemical composition, which creates an "erosion–corrosion" wear condition. This can be much more aggressive than either erosion or corrosion, so standard wear predictions become highly uncertain. Proper application and material selection (refer to Section 12.3.7) are needed to maximize life. This can involve trial-and-error evaluation due to the large variety of slurry solids and fluid chemicals being pumped.

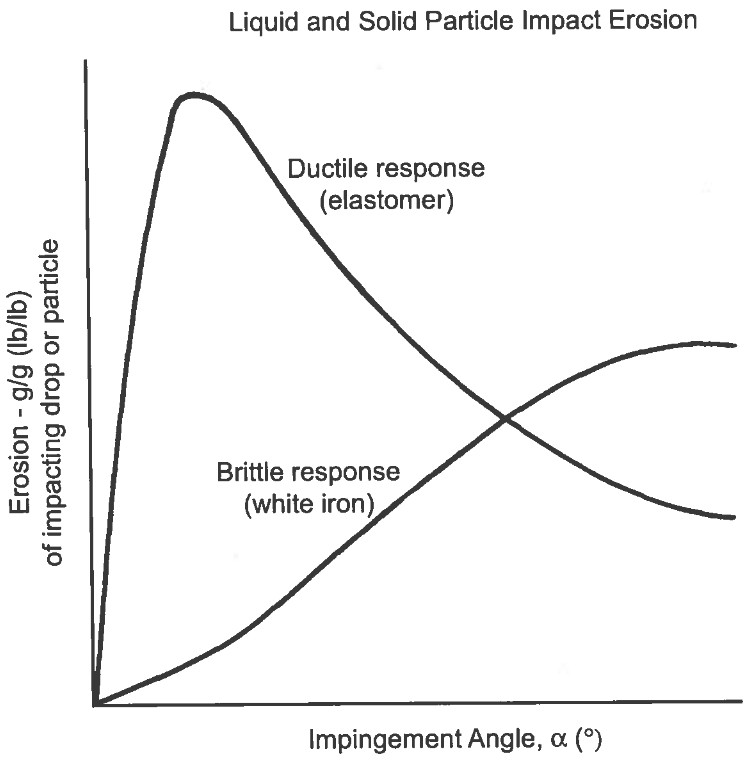


Figure 12.3.4.1b — Erosion response for different impingement angles and materials

12.3.4.2 Pump wear

Pump wear depends on the pump design, the abrasivity of the slurry, the specifics of the application or duty conditions, the way in which the pump is applied or selected for the duty, and the actual conditions of service.

Wear inside the pump varies significantly depending on the velocity, concentration, and impact angle of the particles. It is normally most severe in the impeller seal face area of the suction liner, followed by the vane inlet and exit. The casing wear amount and location also varies with the shape of the collector, and as a percentage of the actual operating conditions compared to the best efficiency point flow.

Many slurry pump wear parts may last for years with only routine maintenance. Services such as transportation of high concentrations and very abrasive or large solids can sometimes reduce part life to several months. Larger pumps with thicker sections, more wear material, and slower operating speeds can improve life in all applications, although the significant associated product cost increase may not be warranted in each particular case. There are analytical and numerical models for making qualitative predictions of wear. Their limitations and the variability of slurry service are such that wetted component life prediction is still only good for estimation and should not be used for guarantees. These estimates are normally based on the specified operating condition of the pump and may vary greatly if the pump is operated at significantly different conditions. Using such an analysis, a life-cycle cost (LCC) evaluation of the capital, power, wear, and other costs associated with the pump operation can be used to estimate the best balance between different pump designs. Such analysis is largely theoretical, however, as wear can be unpredictable in actual service.

Ranking the slurry into light (class 1), medium (class 2), heavy (class 3), and very heavy (class 4) services, as shown in Figure 12.3.4.2a, provides a practical tool for pump selection and, in conjunction with Table 12.3.5a, a means of recommending limiting pump operating heads.

The boundary lines between the service class areas in the chart approximate limits of constant wear modified for practical considerations and experience.

Capital and operating cost considerations are such that different (higher specific speed) designs may be employed for the lighter service classes.

Slurry service ranking shown in Figure 12.3.4.2a is based on aqueous slurries of silica-based solids pumping (S_S = 2.65). It can also be used to provide guidance for mineral slurries if an equivalent specific gravity for the mineral slurry is used to determine the service class. For slurry solids other than silica sand, the ASTM 075-95 Miller number may be utilized to account for the different solids abrasivity in the use of Figure 12.3.4.2a. Here Figure 12.3.4.2b of Miller number versus modified abrasivity (A_{mod}) may be employed to obtain corrected values of S_m and d50 using the relations:

$$S_m \text{ corrected } = S_m \text{ actual} \times 0.4\, A_{mod} + 0.6$$

$$\text{d50 corrected } = \text{ d50 actual} \times (A_{mod})^{0.4}$$

The chart is used by drawing a horizontal line for the concentration by volume of solids (or specific gravity when the solids specific gravity = 2.65), and a vertical line for the average size of the solids. Where the lines intersect determines the class of service. The equivalent specific gravity of the mineral slurry is computed by multiplying the actual slurry specific gravity by the ratio of the Miller number for the mineral slurry to the Miller number of the equivalent silica slurry. This simplified approach can be adjusted based on field experience.

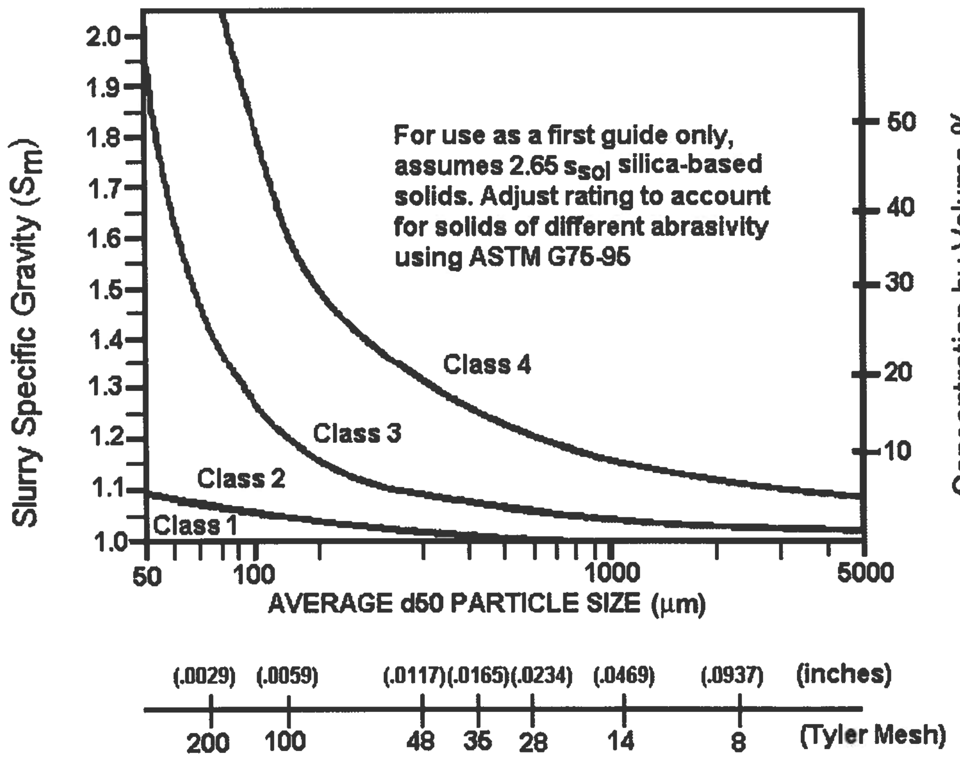


Figure 12.3.4.2a — Service class chart for slurry pump erosive wear

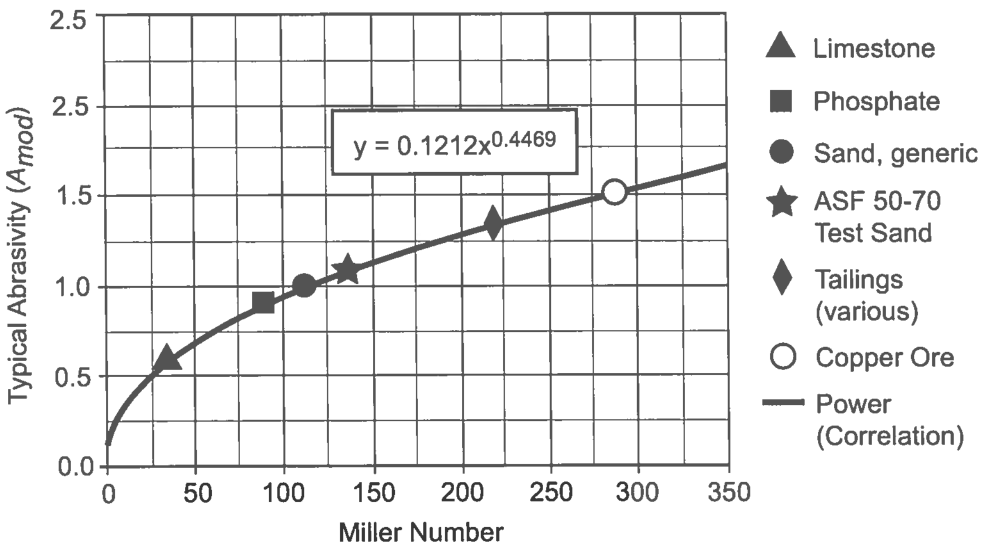


Figure 12.3.4.2b — Miller number versus material abrasivity

12.3.5 Hydraulic design and application considerations

Slurries can be very abrasive and contain large particles, making wear life and the ability to pass these solids major considerations in the design and application of slurry pumps. Compromises in design must be made to provide large hydraulic passages, and obtain satisfactory wear life in these erosive conditions. As a result, slurry pumps tend to be larger, have flatter head–capacity curves, and require more power than their clear-liquid counterparts.

The pump designer maximizes part life by avoiding sharp edges and adding more material to key wear locations. Blunt edges do not wear as fast as sharp ones and more material can be worn from a thicker section before it becomes unusable. The presence of large solids will require wider-than-normal impellers to pass the solids and fewer and thicker vanes to withstand impact loads. This type of design usually compromises optimum hydraulic geometry to some degree, depending on the size of the pump and the intended service. Thus efficiencies of slurry pumps are lower than that of clear-liquid pumps, but wear life is greatly increased. Extra material is needed to combat wear as the size and severity of service increase, so large class 4 pumps will have a casing thickness up to four times greater than would be used for water pumps. Dimensional compromises to accommodate large solids are usually more pronounced on smaller pumps. As a result, small slurry pumps on class 1 services with solids no larger than 50 µm (0.002 in) will approximate the size and performance of water pumps, but small pumps handling large solids and large pumps in severe services will be larger and heavier and performance will be significantly affected. The pump supplier must be advised of the properties of the slurry, including the maximum size solids, so the proper design can be furnished.

Wear must also be controlled by proper pump application. Wear is related to the relative velocity between the pumped liquid and the pump parts. Liquid velocities must be reduced for more severe services to obtain satisfactory life. The different suction liner, impeller, and casing components and their wetted surfaces are exposed to different velocities, slurry concentrations, and impingement angles, making it difficult to provide limits that cover all cases and all components.

Table 12.3.5a provides recommended service limitations for different service classes that, when coupled with proper design and material selection, has resulted in acceptable wear life.

The heads referred to in Table 12.3.2.5a are actual heads after being corrected with the methods outlined in Section 12.3.2.5.

Table 12.3.5a — Recommended service limitations for acceptable wear

	Service class			
	1	2	3	4
Head per stage: m (ft)	105 345	73 240	55 180	40 130
Impeller peripheral speed: All-metal pump m/s (ft/min)	46 9000	38 7500	33 6500	28 5500
Rubber-lined pump	Head generated by impellers made of natural rubber is generally limited to 40 m (130 ft) per stage, which corresponds to peripheral speed of 28 m/s (5500 ft/s). Synthetic elastomers may allow higher limits.			

While the best service life (shut-down for some wear parts change out) of a pump in class 4 service can be limited to four months or less, the expectation for the lower number classes (at the same head) can be approximately quadrupled for each change to a lower class number. At the higher allowable heads in Table 12.3.5a, the wear lives, however, are at least doubled.

Capital and operating cost considerations are such that different (higher specific speed) designs may be employed for the lighter service classes.

In Class 4 service, high rotational speed with large pump suction diameters over 750 mm (30 in) will lead to high wear in the impeller front sealing area, while life of the suction liner can be three months or less. Where downtime or other considerations require longer life, the impeller peripheral speeds should be lowered, noting that a 20% drop in pump rotational speed will roughly double the wear life.

If the service conditions are intermittent, or if experience warrants, then lower class and/or higher impeller peripheral speeds and heads can be used.

Even the largest, most robust pump running at slow speeds will still have increased wear and gouging. Where the pump is operated relative to its best efficiency design point is extremely important. Acceptable gouge-free wear depends on the radial shape of the discharge casing, its width, and the actual percentage of BEP flow rates experienced during operation.

Here a concentric casing is one where the radial distance above the impeller remains constant, while the near volute radial distance increases (approximately) linearly from a cutwater in an angular manner around the periphery of the impeller. The semivolute lies between the two. These are the most commonly accepted types.

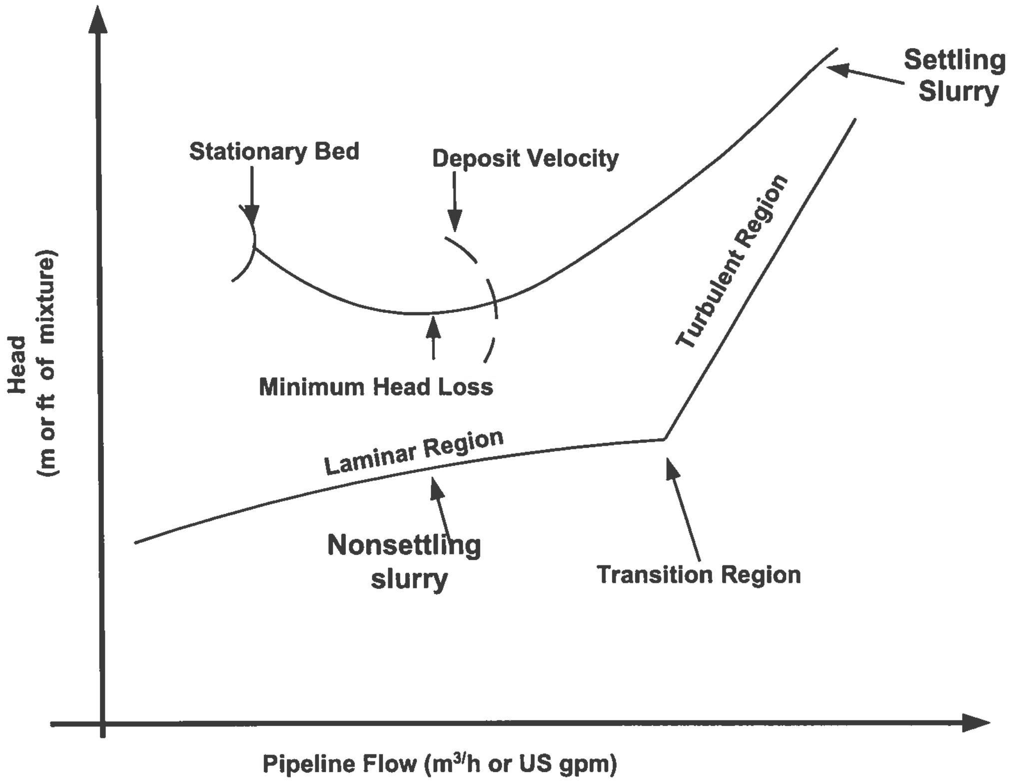


Figure 12.3.6 — Typical constant concentration slurry pipeline friction loss characteristics

While it should be noted that the width and the actual shape of the casing cross sections can modify the wear and its location around the casing, Table 12.3.5b shows the generally acceptable range of flow rates for the different designs.

Table 12.3.5b — Acceptable range of relative flow rates for slurry pump casing optimum wear

Operating Limits	Casing Type	Service Class			
		1	2	3	4
Recommended percent range of BEP flow rate	Annular	20 – 120%	30 – 110%	40 – 100%	50 – 90%
	Semivolute	30 – 130%	40 – 120%	50 – 110%	60 – 100%
	Near volute	50 – 140%	60 – 130%	70 – 120%	80 – 110%

12.3.6 System design

The characteristics of slurries require special considerations in the design of pumping systems. In general, flow velocity must be kept within an optimal range. Higher velocities result in high energy requirements and wear. Lower velocities cause instability and plugging of pipes. It is especially important in a slurry pipeline to keep velocities low and at the same time above a certain minimum. Figure 12.3.6 gives a generalized depiction of slurry effects in piping systems.

System head and velocity requirements for nonsettling slurries can be treated very similarly to clear viscous liquids. Systems should be designed to operate near the transition point (refer to Figure 12.3.6) to obtain energy-efficient and stable operation.

System head and velocity requirements for settling slurries can be determined by similar means, but the deposit velocity (Figure 12.3.2.4) must also be taken into account to avoid plugging pipes. For safe operation, the flow velocity should be the larger of the minimum head loss velocity (Figure 12.3.6) or 110% of the V_{smax}.

Wherever possible, energy usage and operating cost must be minimized within the constraints imposed by the slurries themselves and the associated wear. Here it will be noted that the total cost measurement and any efficiency comparison will be to the transport solids, not water, and the cost should include energy, capital, rotating assembly maintenance, wear parts replacement cost, and any associated downtime cost related to stoppages to replace wear parts.

Information is available in references to calculate the friction loss of slurry flows. Different manufacturer's methods and experience are also valuable sources of knowledge. They must be used with care as the friction loss can be significant and vary drastically for different slurries, so it is necessary to know the characteristics of the specific slurry being pumped. Actual experience with the specific slurry being pumped is the best source of information. If the slurry concentration by volume is over 5% and the piping system length is more than a few hundred meters, published literature may not be adequate. In such cases, tests on the actual slurry should be considered if experience is insufficient.

There will always be some inaccuracy in calculations due to the variations in slurry, piping geometry, etc. that will occur. These small variations can have a significant effect on system head requirements; therefore, provisions should be designed into the system to adjust the pump output to match the actual system needs. This can be accomplished by changing the pump speed using a variable-frequency drive, sheave changes, or by changing the impeller diameter. In either case, excess driver power must be available to accommodate the new pump requirements.

12.3.7 Wetted materials of construction

12.3.7.1 General usage selections

A large variety of metals and elastomers are used for slurry pumps because of the diverse range of applications. Slurries can be erosive, corrosive, or erosive/corrosive. Proper material selection depends on the properties of the mixture to be pumped and the pump design. Figure 12.3.4.2, introduced earlier, may be used to rank purely erosive wear. Table 12.3.7.1 is a selection guide for various materials commonly used in these services along with their appropriate erosive wear service class. Appendix D.1 lists the various international standards for materials typically used in slurry pump construction.

Table 12.3.7.1 — General suitability of wetted materials

Wetted Material	Abrasive characteristics of pumpage	Applicable wear service class	Corrosive characteristics of pumpage
Gray cast iron	Very mild, fine particles	1	Noncorrosive
Ductile iron	Moderate	2	Noncorrosive
White irons	Severe	4	Mildly corrosive
Martensitic stainless steel	Moderate	3	Mildly corrosive
Austenitic stainless steel	Mild	1	Corrosive
Duplex stainless steel	Moderate	2	Corrosive

Table 12.3.7.1 — General suitability of wetted materials (*continued*)

Wetted Material	Abrasive characteristics of pumpage	Applicable wear service class	Corrosive characteristics of pumpage
Super-duplex stainless steel	Moderate	2	Highly corrosive
Elastomers	Severe, fine particles	3[a]	Mildly corrosive

[a] Primarily for impellers. Elastomeric liners are used in class 4 service dependent on application specifics.

Metals resist erosion through a combination of proper hardness and toughness. Toughness is defined as the ability of a material to absorb energy and even deform plastically before fracturing. Hardness provides resistance to sliding wear. Toughness diminishes crack formation and propagation encountered in impacting wear situations, providing resistance to impact fracture. Harder materials are better choices for sliding wear services. A very hard, brittle material that fractures easily may not perform as well as a softer metal that resists brittle fracture. Erosion resistance should not be judged only on the hardness of the material.

Material selection is further complicated when corrosive carrier liquids are involved. Materials that are highly resistant to erosion are usually not highly resistant to aggressive corrosion. Material selection is a compromise between erosion and corrosion resistance properties to achieve optimum wear life for any specific installation.

Metals resist corrosion by forming a passivated surface layer that protects against further corrosion. Effectiveness is determined by how tough the passive layer is and how fast it forms. In slurry services, the passive layer is continually being worn away and reformed so corrosive attack is accelerated.

Elastomers resist erosion through resilience and tear resistance. These are soft and the solid particles rebound without damaging the elastomer by abrasion or fracture. Large or sharp particles may tear the elastomer, so material selection must be carefully matched to the slurry.

Elastomers do not depend on a passivated layer to resist corrosion. The basic chemical resistance is a function of proper material selection, and is not significantly changed by exposure to erosive environments. Slurry pumps usually have thicker liners than other elastomer-lined pumps. Experience has shown that when lining thickness is increased, wear life increases by a factor of approximately 2:1, within the limits of a practical liner thickness.

Elastomers can be easily bonded to metals to combine the strength and rigidity of the metal with the elasticity of the elastomer. They can also be bonded to materials such as fiberglass-reinforced plastic (FRP) or thermosetting phenolic/nylon cloth to stiffen liners to prevent collapsing during process disruptions, such as cavitation or surges. They can be bonded to ceramics to take advantage of the best of both materials. Process and environmental temperatures must be considered, as some elastomers do not perform well above 80 °C (180 °F).

Erosive and erosive/corrosive wear may occur under different mechanisms. Because of the complex nature, the wear results may vary substantially from case to case. Experience with similar applications is always the best guide to selecting materials. If there is inadequate experience, wear testing can be performed to help evaluate the level and characteristics of wear factors. Some typical wet wear tests include the ASTM G75-01 Miller procedure, slurry-jet wear testing, and Coriolis wear testing. Corrosion, erosion, and corrosion/erosion testing may be required to analyze erosive/corrosive applications where experience is not available.

12.3.7.2 Irons

Gray cast iron is relatively soft and brittle, so it has limited application in slurry services. It is normally only used on very low concentrations of fine soft particles in a noncorrosive carrier, or for clear-water service.

Ductile iron is significantly tougher than gray or other irons due to its microstructure of nodular graphite in an iron matrix. Elongation can be up to 18%. This makes it suitable for pumping large particles where impact can be a problem. It is still only moderately abrasion-resistant and is limited to moderate concentrations in noncorrosive services.

Hard irons are used for the most erosive slurries and those with weak acid or caustic carriers. These are classified as chromium-nickel (NiHard), chromium-molybdenum, and high-chromium white irons. These irons can normally be used for slurries with pH values between 3.5 and 10.0, depending on the chloride level. At zero chlorides, for example, a pH of 4.5 is satisfactory, whereas at 20,000 ppm chlorides, the pH is limited to 6.5.

These irons gain erosion and corrosion resistance primarily by the addition of chromium, which promotes the formation of chromium carbides in a softer matrix of ferrite, martensite, or austenite. These chromium carbides are three to four times harder than the matrix and provide excellent wear resistance, while the softer matrix maintains strength and some ductility. Higher chromium levels usually increase the corrosion resistance, making them more suitable for solutions with pH farther from a neutral.

Machining hard irons is relatively difficult and grinding will be required in some cases to obtain desired dimensions. One of the advantages of a hard chrome iron is that it can be annealed, machined, and rehardened.

12.3.7.3 Stainless steels

Stainless steels must be used for more severe corrosive applications. Their erosion resistance is much less than the hard irons, but is offset by increased corrosion resistance. Martensitic, ferritic, austenitic, and duplex grades can be used, depending on the application. Corrosion-resistant properties are achieved by large additions of chromium, nickel, molybdenum, and copper.

Martensitic stainless steels (400 series ASTM A487 CA15) are used for mildly corrosive applications. Martensite is quite hard, but is not highly corrosion-resistant. These are the most wear-resistant stainless steels. They are suitable for moderately abrasive slurries, but are limited to relatively mild corrosive services.

Austenitic stainless steels and other high-nickel alloys (300 series ASTM A744 CN7MCu) are used for highly corrosive applications. Austenite granules are very tough and corrosion-resistant, but the matrix is quite soft. These are the softest steels commonly used in slurry services. They are corrosion-resistant, but are limited to very light slurry applications.

Duplex stainless steels (ASTM A890 CD4MCu) are two-phase alloys, which contain both ferrite and austenite in the microstructure. This provides better corrosion resistance than high-alloyed martensitic steels, and better strength and erosion resistance than austenitic stainless steels. These steels are used for light slurries with aggressive carrier liquids.

Super-duplex stainless steels (ASTM A890 CD3MWCuN), in addition to main properties of duplex steels, provide further improved resistance to acids, acid-chlorides, caustic solutions, and other tough environments in the chemical/petrochemical, pulp, and paper industries, often replacing 300 series stainless steel, austenitic steels, nickel-based alloys, and more common duplex stainless-steel grades.

12.3.7.4 Elastomers

The most commonly used polymer is natural rubber (NR). The common form is "pure gum" natural rubber, which is usually defined by having a specific gravity of < 1.0, a hardness of approximately 40 Shore A, and very high resilience. This high resilience gives maximum erosion resistance, providing the slurry particles are not too large (< 6 mm [0.24 in] for impellers) or too sharp, causing excessive cutting and tearing. Natural rubber components are chemically resistant to most slurries, which are mildly acidic or basic, and at temperatures less than about 80 °C (180 °F).

Natural rubber can be compounded with fillers such as carbon black of extremely fine carbon particles and/or silica to increase hardness and stiffness. This increases resistance of rubber to cutting and tearing and allows it to handle larger particles (< 13 mm [0.50 in]) and higher tip speeds. Resilience is decreased, so erosion resistance may decrease.

The following synthetic rubber elastomers are used, mainly for improved chemical and/or heat resistance.

Polychloroprene (CR) is commonly known as *neoprene*. It is used for increased heat resistance (up to approximately 100 °C [212 °F]), resistance to certain hot acids, and moderate oil resistance. It is not as erosion-resistant as natural rubber, but is better than most others.

Polyurethane is often used for fine-particle slurries (less than 210 μm or 0.008 in), as it is much harder than other elastomers with the same resilience, so increased impeller tip speeds are possible. Oil and solvent resistance is good. Care must be taken to select the proper type of urethane to prevent problems with hydrolysis in hot (80 °C [180 °F] maximum) service.

Butyl (IIR), chlorobutyl (CIIR), or bromobutyl (BIIR) are sometimes used for hot acid service but erosion resistance is generally poor. Ethylene-propylene-diene-monomer (EPDM) has much the same chemical resistances as the butyl's, generally better heat resistance, and considerably better erosion resistance. It has often replaced butyl in hot acid service.

Nitrile (NBR) is used where maximum resistance to nonpolar oils (total petroleum hydrocarbon oils that do not have a charge at the end of the molecule) and solvents is required. Erosion resistance is only fair. Carboxylated nitrile (XNBR) has better erosion resistance. Hydrogenated nitrile (HNBR) has better erosion resistance along with much better heat resistance. It is also resistant to hot water (up to approximately 175 °C [347 °F]). It is extremely expensive, so use is usually restricted to small parts.

12.3.8 General arrangement details

12.3.8.1 Impellers

Both semiopen and closed impellers are used in slurry services. The control of leakage back into suction is usually accomplished with a combination of clearing or expelling vanes on the impeller and close axial clearances. Because these axial clearances increase with wear, pumps should be arranged to allow simple clearance adjustments to maintain performance. Close radial clearances wear quickly when solids are present and cannot be conveniently corrected with external adjustment, and should only be used on low concentrations of fine slurries. An axial clearance arrangement between the impeller inlet diameter and liner is common for providing leakage control for high-wear services.

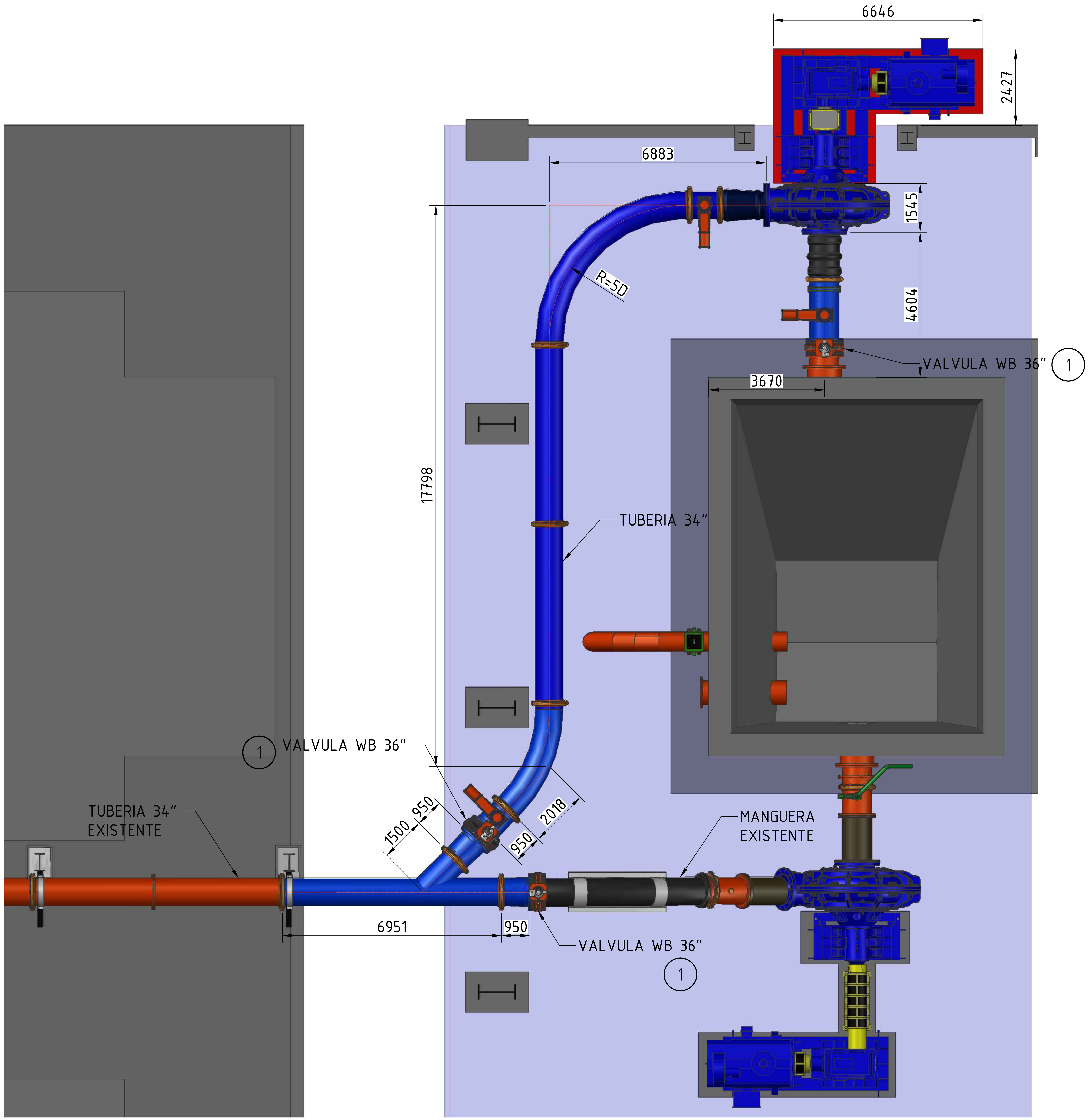
Impeller attachment methods vary by manufacturer and service requirements. Various bolted designs and threaded designs are used successfully. When pumping highly abrasive slurries, the impeller attachment should be protected from wear to optimize service life. An internally threaded impeller is typically used in high-wear services to provide this protection.

Balancing requirements for slurry pump impellers are different than those applied to impellers for clear liquids. An impeller balanced for clear-liquid service is expected to remain substantially in balance for most of its operating life. As a slurry pump impeller wears in service, it will naturally begin to change its balance due to the erosion of metal along the wear surfaces. Consequently, the bearings and shafts in a slurry pump must be designed for a large amount of unbalance in the impeller. In general, slurry pump impellers will be balanced to a lesser standard (higher residual unbalance) than a clear-liquid impeller. The levels of residual unbalance allowed are determined by the manufacturer and are based on a number of operational and design factors. As a rule of thumb, slurry pump impeller balance requirements will fall between balance quality grade G40 on the high (large amount of residual unbalance) side and grade G6.3 on the low (small amount of residual unbalance) side as defined in ISO1940/1 *Mechanical vibration - Balance quality requirements for rotors in constant (rigid) state.*

Slurry pump impellers are typically disk shaped and can be balanced in a single plane. Low specific-speed impellers are usually statically balanced on balance rails or a roller shaft arrangement. For greater sensitivity and

Anexo 3

ESQUEMAS DE INSTALACIÓN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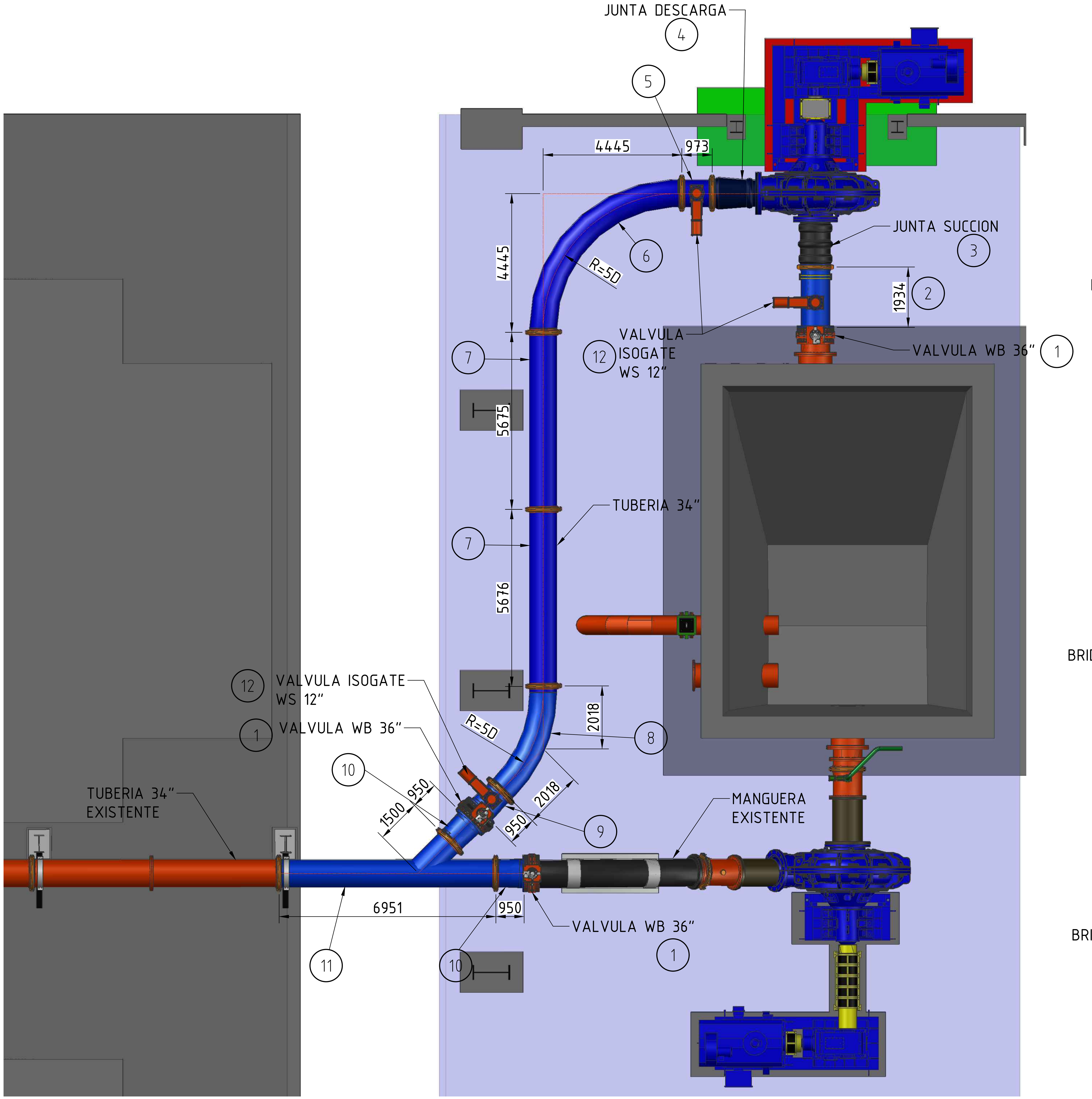
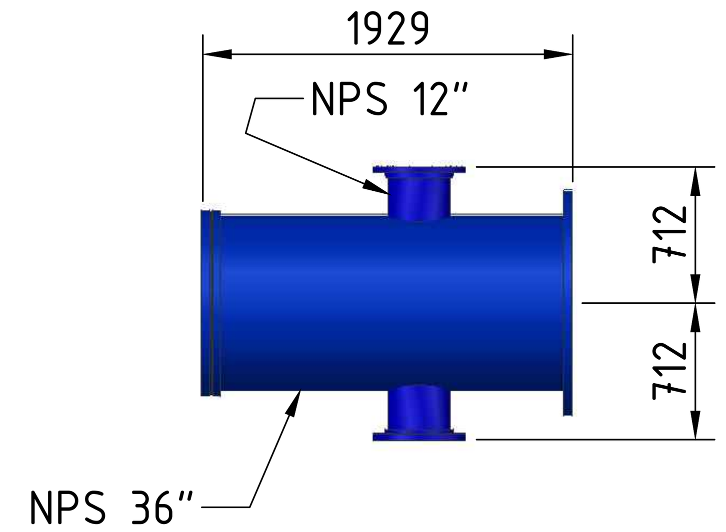
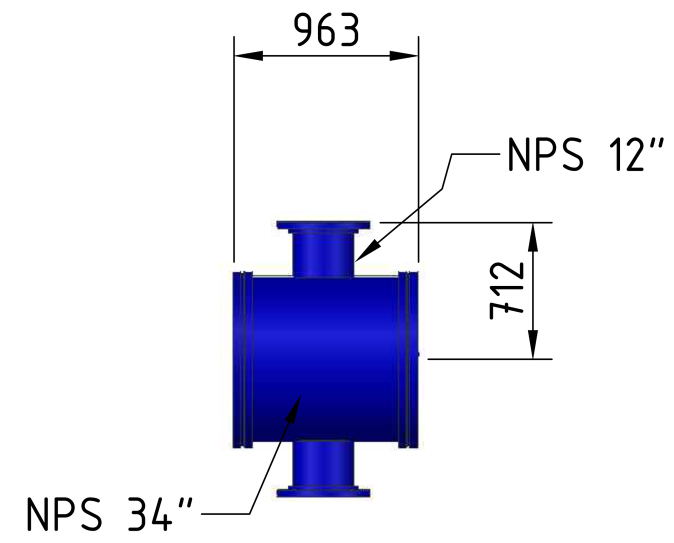




ITEM 2



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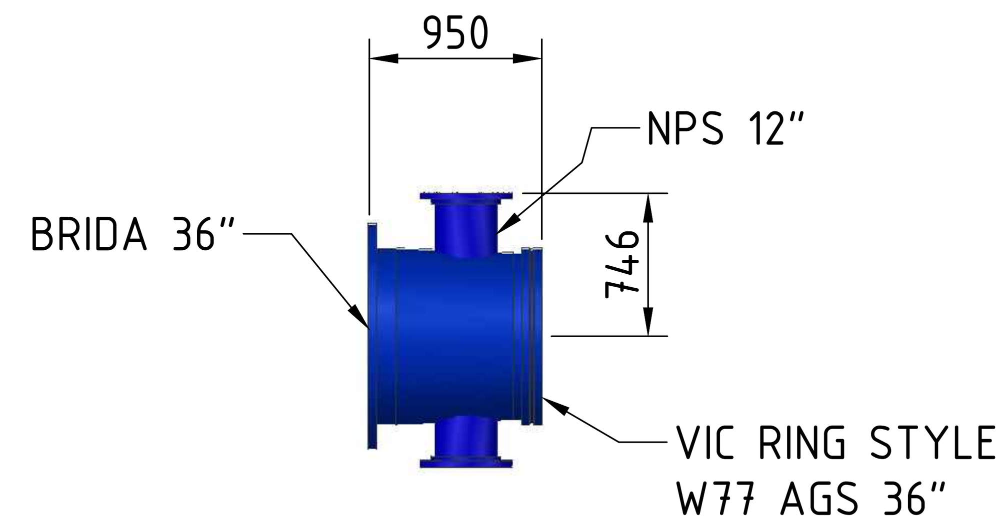
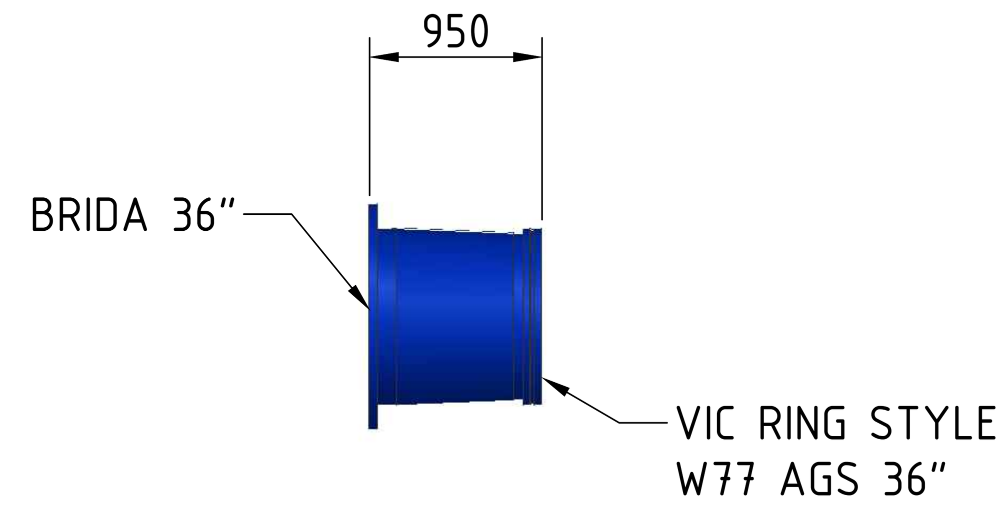


ITEM 9
RED. Conc. 36"x34"



ITEM 10
RED. Conc. 36"x34"

Anexo 4

TABLAS

MINERALS

Material Specification Index Q.A.P.22C

Issue: 29 Apr 2014
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Code	Material Name	Revision	
500	48 Shore A Natural Rubber	new	Sep 3,2002
505	46 Shore A Grey Natural Rubber	new	Sep 3,2002
510	42 Shore A Tan Natural Rubber	new	Sep 3,2002
545	60 Shore A Black Polychloroprene	new	Sep 3,2002
A03	Ni - Hard 1	5	Sep 21,2006
A04	Ultrachrome ® 24% Cr	7	Aug 29,2011
A05	Ultrachrome ® 27%Cr	16	Sep 6,2011
A05C	Ultrachrome ® 27%Cr	14c	Jun 12,2012
A05S	Ultrachrome ® 27%Cr (subcontractor)	14	Jun 12,2012
A05X	Ultrachrome ® 28%Cr	5	Jun 12,2012
A06	Ni - Hard 4	4	Sep 20,2006
A07	15Cr - 3Mo	8	Feb 11,2011
A08	Ultrachrome ® 27% Cr for thick castings	7	Jan 11,2013
A09	20Cr - 2Mo	3	Dec 9,2008
A10	Ultrachrome ® 27% Cr (As-Cast)	4	Jun 12,2012
A11	Ultrachrome ® 27%Cr (A05 tempered, UTS)	4	Jun 12,2012
A12	Hyperchrome ® 30% Cr	9	Aug 5,2011
A13	Ultrachrome®27%Cr thick castings, tempered	New	May 6,2013
A14	Ultrachrome ® Tough 27% Cr	3	Feb 11,2011
A17	20 % Cr-Mo (As Cast)	0	Sep 3,2009
A18	20 % Cr-Mo (Hardened)	0	Sep 3,2009
A20	Pearlitic Wear-Resistant Steel Type L2C	0	Oct 14,2013
A21	Pearlitic Manganese Steel Type L1B	2	Dec 9,2008
A22	Pearlitic Wear-Resisting Steel Type L2B (Mod)	2	Oct 6,2000
A23	Cast Chrome Moly Steel (ECCRMO)	0	Jul 20,2005
A24	Wear-Resisting Alloy Steel (85CM)	3	Jul 20,2005
A25	Ni - Cr - Mo Steel (Cast Steel)	4	May 19,2011
A26	INCO 646 (Modified)	1	Apr 30,1990
A27	Cast Steel (WCB)	0	Jun 17,2009
A28	Hyperchrome ® 35% Cr (As Cast)	1	Sep 21,2006
A29	Grade 90-60 Cast Steel	1	Jun 11,2013
A31	Type H1A Manganese Steel	3	Dec 9,2008
A35	Hardox 500 Wear Plate	0	Jul 29,2013
A36	Hardox 600 Wear Plate	0	Jul 29,2013
A37	BRINAR 450	0	Aug 14,2013
A46	Abrasion Resistant Steel - Hardox 450	0	Jan 30,2013
A47	Ultrachrome® 28% Cr, Low C, H/T	1	May 15,2008
A48	Ultrachrome® 28%Cr, Low C (As Cast)	2	Jul 2,2010
A49	Ultrachrome® 28% Cr, Low C, H/T	14	Jul 2,2010
A491	Ultrachrome ® 28% Cr, Low C (H/T)	2	Nov 27,2006

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Code	Material Name	Revision	
A51	Ultrachrome ® 36% Cr, Low C	8	Jul 2,2010
A52	Ultrachrome ® 36% Cr, Low C H/T	1	Jul 2,2010
A53	Ultrachrome ® 38% Cr, Low C	2	Jul 2,2010
A55	Machinable Ultrachrome 38%	3	Sep 20,2006
A61	Hyperchrome ® 30% Cr (Heat Treated)	7	Feb 23,2011
A62	Hyperchrome®3.8%C (Heat Treated)	0	Mar 5,2013
A63	Hyperchrome ® 35% Cr (Heat Treated)	1	Sep 22,2009
B01	Building Material (Generic)	New	Feb 9, 1999
C01	Ni-Resist 15.6.2	1	Apr 30,1990
C02	Ni-Resist (copper free)	2	Apr 30,1990
C03	S.G. Ni-Resist 20.2	3	May 27,2011
C10	CA-40 Casting Stainless Steel	New	Jul 20,2010
C11	Type 420C Stainless Steel	1	Mar 9,1993
C14	Duplex Stainless Steel	1	Jul 3,1991
C15	17-4PH (H925) Stainless Steel	0	Sep 17,2009
C16	17-4PH (H1100) Stainless Steel	0	Sep 17,2009
C17	17-4PH (H900) Stainless Steel	2	Sep 16,2009
C18	17-7PH (TH1050) Stainless Steel	0	Jan 18,2013
C19	Cast Duplex Stainless Steel (2507 type)	1	Jun 3,2008
C20	Type H3A Stainless Steel	1	Apr 30,1990
C21	Type 420C Stainless Steel (Cast Alloy)	8	Feb 1,2013
C22	Type CF-8 Stainless Steel (Cast Alloy)	1	Apr 30,1990
C23	Type CF-8M (316) Stainless Steel (Cast Alloy)	3	Mar 6,2006
C24	Type H5C Stainless Steel	1	Apr 30,1990
C25	Alloy 20	3	Aug 7,1991
C26	Type CD-4MCuN Duplex Stainless Steel	5	Aug 10,2012
C27	' 825 ' Alloy	2	Apr 30,1990
C28	450 Stainless Steel	1	Dec 7,2001
C29	Wrought Austenitic Stainless Steel 2RK65	1	Apr 24,1996
C30	27 Cr 31 Ni Stainless Steel	2	Apr 30,1990
C31	Wrought BHP Alloy Steel, Grade 3Cr12	New	Oct 21,1991
C33	Duplex Stainless Steel with Nitrogen	new	Sep 4,2009
C34	Type CD3MN Duplex Stainless Steel	0	Mar 19,2010
C38	Type 420C Stainless Steel	New	
C41	316L Stainless Steel	2	Sep 6,1991
C42	Wrought 431 Stainless Steel	2	Sep 6,1991
C43	317L Stainless Steel	2	Sep 6,1991
C44	Type 440C Stainless Steel	1	Nov 30,2010
C46	LDX 2101 Duplex Stainless Steel	0	Sep 9,2011
C50	Uranus 50 Duplex Stainless Steel	0	Oct 22,2009

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Code	Material Name	Revision	
C51	Type 410 Stainless Steel (T)	0	Oct 27,2008
C52	Cobalide 2	1	Apr 30,1990
C54	Stainless Steel, CK3MCuN, (Cast Alloy)	0	Jun 16,2009
C55	Ferralium ® 255	2	Apr 30,1990
C60	Wrought Duplex Stainless Steel SAF 2205	2	Apr 23,1996
C61	Wrought Duplex Stainless Steel SAF 2507	1	Dec 7,2006
C62	Cast Zeron 100 Super Duplex Stainless Steel	New	Dec 12,2006
C63	Wrought Zeron 100 Super Duplex Stainless Steel	New	Dec 13,2006
C64	Type 654 SMO Stainless Steel	0	Feb 12,2010
C65	Wrought Duplex Stainless Steel	0	Dec 7,2009
C69	AISI 329 Duplex Stainless Steel	0	Mar 11,2014
C70	Nickel Coated C22	0	Sep 18, 2008
C71	Type 420C Stainless Steel	0	Jun 11,2009
C72	Type 304 Stainless Steel	0	Jun 12,2009
C73	Type 316 Stainless Steel	1	Jan 11,2013
C74	Type 304L Stainless Steel	0	Jun 15,2009
C75	Type 302 Stainless Steel	0	Sep 3,2009
C80	Sanicro 28	0	Oct 21,2009
C83	B8M Class 2 Stainless Steel* (AISI 316)	0	Jan 21,2013
C85	Hard Chrome Plated C73	1	Nov 8, 2013
C92	Alloy 926 Stainless Steel	0	Oct 14,2009
D19	Ductile S.G. Iron, Grade 350-22	3	Feb 27,2008
D20	Ductile S.G. Iron, Grade 400-15	4	Aug 2,2012
D21	Ductile S.G. Iron, Grade 500-7	3	Feb 14,2007
D22	Ductile S.G. Iron, Grade 600-3	3	Nov 7,2006
D23	Ductile S.G. Iron, Grade 700-2	2	Feb 27,2008
D24	Ductile S.G. Iron, Grade 800-2	3	Feb 27,2008
D81	Zinc Plated D21	2	Aug 16, 1991
D82	Fluoropolymer Coated D21	0	Apr 7, 2008
D83	Nickel Coated D21	0	Sep 16, 2008
E01	BHP-300PLUS Structural Steel	New	Oct 27,1998
E02	Plain Carbon Structural Steel	4	Jul 25,2005
E03	Wrought 1030 Carbon Steel	3	Aug 6,2008
E04	Grade 350 Structural Steel	1	Nov 7,2011
E05	Wrought K1045 Carbon Steel	2	Apr 19,1993
E06	Wrought K1046 Carbon Steel	New	Mar 11,1990
E07	Wrought 1050 Carbon Steel	New	Mar 11,1990
E08	Wrought Free - Cutting Steel	New	Nov 1,1990
E09	Wrought K1042 Carbon Steel	0	May 10,2005
E10	Medium to High Tensile Fasteners Grade 8.8	2	Aug 1,2006

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Code	Material Name	Revision	
E11	Forged Engineering Steel	New	Apr 17,1997
E12	High Tensile Fasteners Grade 12.9	SW	Aug 1,2006
E13	Medium to High Tensile Fasteners Grade 10.9	New	Feb 12,2008
E16	Wrought SAE 1026 Carbon Steel	0	Aug 26,2009
E19	Carbon Steel (En9), Normalised	0	Jul 18,2013
E20	Cold Work Tool Steel (ARNE^)	1	Jul 12,2011
E21	Mid To High Tensile Alloy Steel (4140,En19A)	3	Jan 11,2013
E22	Mid to High Tensile Alloy Steel (En25, X9931)	3	Jul 18,2006
E23	High Tensile Alloy Steel (En 26, 4340)	5	Mar 22,2007
E24	Bisplate® 400	1	Mar 16,1999
E25	Vanadium Microalloyed Steel (MW 450U)	1	Jan 14,2011
E26	High Strength Low Alloy Steel, Grade 450	New	Oct 25,2002
E28	Molybond Coated E03	New	Sep 15, 1994
E30	Cast Steel (LCB) - For Low Temperature	0	Aug 26,2011
E31	AS 2074/L1B Low Alloy Cast Steel	0	Jun 7,2010
E32	ASG 22 Grade C-3 Steel	1	Apr 30,1990
E33	Steel Plates for Pressure Equipment	New	Jun 18,2004
E34	Seamless Carbon Steel Pipe	New	Jun 18,2004
E35	Molybond Coated E10	1	Aug 1, 2006
E36	Mould Steel P20	0	Apr 8,2008
E37	High Grade Case Hardening Steel AISI 3310	0	Apr 8,2008
E38	Seamless Steel Pipe - For Low Temperature	0	Dec 13,2011
E39	Pressure Vessel Steel Plate	0	Jan 18,2013
E40	AISI 8640 Alloy Steel	0	Oct 22,2009
E41	SS400 Structural Steel (AR)	0	Nov 1,2010
E43	4340 Alloy Steel Forging	0	May 2,2011
E44	817M40T Steel	0	Mar 1,2012
E45	Graphalloy Type 453	0	Sep 28,2010
E46	Molybond Coated E21	3	Apr 15, 1997
E47	Molybond Coated E22	3	Apr 15, 1997
E48	Molybond Coated E23	New	Apr 15, 1997
E51	L7 Alloy Steel* (AISI 4140/4142 or 4145)	0	Jan 21,2013
E52	Quenched & Tempered Alloy Steel*	0	Jan 21,2013
E57	RIE4090/4606 Coated E22	0	Oct 28, 2009
E58	RIE4090/4606 Coated E23	0	Oct 28, 2009
E61	Zinc Plated E02	0	Sep 26, 2013
E62	Zinc Plated E02	1	Oct 31, 1991
E63	Zinc Plated E03	1	Oct 31, 1991
E65	Zinc Plated E05	2	Aug 7, 1991
E66	Galvanised E02	2	Aug 7, 1991

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Code	Material Name	Revision	
E67	Fasteners Grade 10.9 Zinc Plated	New	Feb 12,2008
E68	Zinc Plated E08	0	Jun 7, 2007
E69	Med Tensile Fasteners Grade 8.8 Zn plated	New	Feb 24,2009
E70	Copper Coated E02	0	Sep 14, 2012
E77	DACROMET Coated E22	0	May 10, 2012
E78	DACROMET Coated E23	0	May 10, 2012
G01	Grey Iron Grade 200	5	Jul 7,2009
G02	Cast Iron Grade 250	4	Jul 7,2009
G03	Cast Iron Grade 300	3	Jul 7,2009
G04	Cast Iron Grade 350	3	Jul 7,2009
G05	Cast Iron Grade T400	1	Apr 30,1990
G61	Zinc Plated G01	2	Aug 16, 1991
G71	Galvanised G01	0	May 6, 2010
H02	Type HH Stainless Steel	2	Jul 20,2005
H03	Type HK40 Stainless Steel	3	May 18,2011
H05	Type HU Stainless Steel	1	Apr 30,1990
J02	Tungsten Carbide (V21) Coated C02	1	Apr 1, 1993
J04	Material Composite V54/C21	0	May 13, 2010
J05	Cromium Oxide Plasma Coated C11	2	Apr 1, 1993
J06	Chrome Oxide Coated C21	New	May 7, 1997
J10	Chromium Oxide Plasma Coated C25	2	Apr 1, 1993
J11	Chromium Oxide Plasma Coated C23	2	Apr 1, 1993
J14	Tungsten Carbide Coated E02	1	Sep 13, 1993
J19	Material Composite V19/E25	0	Jan 14, 2011
J21	Tungsten Carbide V21 coated C21	3	Apr 19, 2000
J22	Chromium Oxide Coated N22	1	Apr 5, 1993
J24	Tungsten Carbide V21 Coated C23	3	Jun 16, 1995
J25	Tungsten Carbide V21 Coated C11	New	Apr 5, 1993
J26	Chrome Oxide Y03 Coated C26	1	Apr 5, 1993
J27	WC/Chromium/Nickel Coated C26	New	Apr 30, 1999
J31	Material Composite V36/C38	2	Jun 5, 2003
J33	Material Composite V36 / C23	New	Mar 4, 2002
J34	Material Composite V36/E02	0	Nov 27, 2012
J35	Material Composite V36/E21	0	Nov 27, 2012
J36	Material Composite V36 Coated C26/C60	2	Nov 2, 2009
J38	Material Composite V38/C28	0	Feb 3, 2009
J40	Material Composite V36/A08	0	Oct 12, 2010
J41	Tungsten Carbide V21 coated D21	0	Oct 27, 2010
J42	Material Composite V36/A05	0	Nov 1, 2010
K01	Wrought Copper	New	Nov 10,2009

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Code	Material Name	Revision	
K06	Cupro Nickel 10%	New	Aug 2,1991
K11	Cartridge Brass	1	Sep 13,1990
K21	Wrought Aluminium Bronze, 9%	new	Mar 23,2001
K24	Type 907 Phosphor Bronze	2	Feb 3,2009
K25	Type 88.10.2 Gunmetal	1	Apr 30,1990
K31	Type 85.5.5.5. Leaded Gunmetal	1	Apr 30,1990
L02	Type LM4 Aluminium	1	Apr 30,1990
L06	Type 356 Aluminium	0	Mar 10,2010
L12	Type LM25 Aluminium	2	Mar 11,2014
L13	Aluminium Alloy 5083 - H321	1	Nov 14,2003
L61	Aluminium Alloy 6061-T6	New	Nov 4,2003
L63	Aluminium Alloy 6063-T5	0	Sep 10,2009
M01	Wrought Titanium Grade 2	2	Aug 7,1991
N02	63 Ni 30 Cu Alloy (Monel Metal)	2	Apr 30,1990
N04	Nickel Alloy CW-12MW (Hastelloy C)	5	May 23,2003
N05	55Ni-28Mo-5Fe Alloy (Hastelloy B)	3	Oct 14,1992
N22	58Ni-22Cr-13Mo-3W-Fe Alloy (Hast C22)	1	Jun 17,1991
N23	Wrought Hastelloy®C22	1	Aug 30,1993
N24	Wrought Hastelloy® C276	1	Nov 24,2003
P01	Cast Epoxy Resin (Electrical)	2	Jul 21, 1997
P02	Epoxy Fiberglass	New	May 13, 1998
P03	Filled PTFE	2	Jul 21, 1997
P04	Epoxy Resin	0	Jun 16, 2011
P05	PTFE	New	May 13, 1998
P06	Teflon Molybdenum	0	Mar 23, 2007
P07	Polyester Fibreglass SMC 25%	3	May 13, 1998
P08	Polyester Fibreglass DMC (Grey)	1	Jun 4, 2008
P09	Polyester Fibreglass DMC (Yellow)	11	Feb 25, 2013
P10	Mylar Polyester Film	New	Jul 6, 2000
P11	Polyester Fibreglass SMC 60%	4	Aug 19, 1999
P13	Vinyl	0	Sep 20, 2011
P19	MDPE	New	Feb 14, 2000
P20	Polyester Fibreglass (belt guard Somersby)	New	Jul 6, 2000
P21	Polyvinylester Fibreglass - liner backing	New	Jul 6, 2000
P22	Phenolic Cotton - Fufnol Bear	New	May 30, 2000
P23	Carbon Filled Teflon (CFT)	New	Sep 16, 2013
P25	Aramid / Glass Tape	New	Nov 2, 2001
P27	Nylon	New	May 13, 1998
P30	PVC	New	May 13, 1998
P31	SiC Polymer Casting VE 220	1	Jun 7, 2012

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Code	Material Name	Revision	
P32	PVC impregnated Nylon fabric	0	Oct 21, 2010
P33	Phenolic Cloth	1	Aug 10, 2009
P34	Chemical Resistant DMC	1	May 13, 1998
P37	Vinyl Ester Fibreglass	2	Jul 21, 1997
P37	Vinyl Ester Fibreglass	3	May 13, 1998
P41	Bakelite	2	Jul 21, 1997
P41	Bakelite	3	May 13, 1998
P50	Polyphenylene Sulphide (RYTON)	3	Jul 21, 1997
P60	UHMW Polyethylene	3	May 13, 1998
P61	UHMW HT Polyethylene	0	May 13, 1998
P62	High Density Polyethylene	2	Jul 21, 1997
P64	Polycarbonate	0	Jun 16, 2011
P66	Polypropylene	0	Jun 15, 2011
P70	Acrylonitrile Butadiene Styrene (ABS)	0	May 27, 2009
Q05	Standard Gland Packing	1	May 31, 1990
Q06	Aluflon Packing PTFE	New	Feb 27, 2002
Q07	Braided Teflon Fiber	New	Mar 13, 2003
Q10	PTFE Braided Yarn	New	May 13, 1998
Q21	Heavy Duty Gland Packing	0	Oct 7, 2009
Q22	Heavy Duty Gland Packing	1	May 31, 1990
Q23	Aramid/PTFE UC gland seal packing	1	May 15, 1995
Q24	Graphite Impregnated PTFE Packing	New	Feb 8, 1999
Q25	Polyimide and PTFE Packing	New	Feb 8, 1999
Q26	Flex 900 PTFE Gland Packing	1	Mar 11, 2010
Q27	Packing - Pilotsil 8422	New	May 14, 2004
Q28	Packing - Pilotpack 5020	New	May 14, 2004
Q29	Graphite and PTFE yarn Liongraf Packing	New	Jul 22, 2004
Q30	Fluograf Packing	0	Apr 8, 2008
Q32	Somersby Packing	New	
Q33	Somersby Packing	New	Mar 3, 2000
Q34	"A" Engineering Felt	New	Jul 18, 2003
Q42	Aramid Strand	0	Jul 24, 2009
Q43	PTFE White	0	Jul 24, 2009
Q44	Graphite	0	Jul 24, 2009
Q45	Reinforced Graphite Yarn	0	Jul 24, 2009
Q46	Acrylic Fiber Yarn	0	Jul 24, 2009
Q47	PTFE PACKING WITH RUBBER CORE	New	Apr 27, 2011
Q48	Graphite Strands with Carbon Fibre Yarn	0	Nov 2, 2011
R00	Natural Rubber (External)	4	Dec 4,2007
R01	Natural Rubber (External)	New	May 13,1998

MINERALS

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Code	Material Name	Revision	
R03	Natural Rubber SMR/BR (dummy spec see M Lum)	1	Oct 26,2009
R04	Linagard FG Food Grade Rubber Sheet	0	Feb 10,2011
R08	Standard Impeller Rubber	6	May 13,1998
R09	Food Grade Elastomer	5	May 13,1998
R10	Hard Natural Rubber (Strainers)	6	Dec 22,2003
R11	Hard Natural Rubber	2	Dec 22,2003
R12	Natural Rubber 70°A	1	Oct 20,1990
R24	Anti Thermal Breakdown Rubber	7	Sep 22,1997
R26	Standard Liner Rubber	8	Sep 22,1997
R28	70 Duro Natural Rubber/Chloroprene blend	New	May 2,2005
R33	Natural Rubber Pump Liner	8	Nov 5,2003
R35	Linatex Premium Rubber Sheet - 38 Shore A	1	Jun 8,2011
R37	Linatex VS Rubber Sheet - 40 Shore A	1	Feb 3,2012
R38	50 Duro Natural Rubber	4	Nov 19,2003
R45	60 Shore A FGD Liner Rubber	5	Sep 6,2012
R49	Natural Rubber Hard	4	Sep 22,1997
R55	Mill Discharge Liner, 50 Shore A	3	May 10,2011
R57	Linard 50	0	Jul 3,2013
R60	RASA 60 Duro Rubber	2	Aug 9,1990
R63	65 Shore A Natural Rubber / Polybutadiene	2	Nov 14,2013
R64	153741 Blk	0	
R65	153763 Blk	0	
R66	60 Duro Natural Rubber	5	May 17,2010
R68	Linard 60 Rubber Sheet - 60 Shore A	0	Feb 10,2011
R69	Linard HD60 Rubber Sheet - 60 Shore A	0	Feb 10,2011
R70	Knife gate valve liners 65NR	1	Feb 10,2000
R71	WS Valve Natural Rubber 70A	1	Mar 30,2009
R72	KNIFE GATE VALVE NR1	0	
R74	Linard HD70 Rubber Sheet	1	Aug 28,2012
R75	Linard HDS Rubber Sheet	0	Feb 10,2011
R80	50 Duro Natural Rubber (Envirotech 500 - Somersby)	New	Jul 6,2000
R81	Natural Rubber-Envirotech Somersby 510 (Off White)	New	Jul 6,2000
R82	Hard Natural Rubber Galigher Impeller (510M1)	New	Jul 6,2000
R83	Hard Natural Rubber Screens 510M2 Weir 01-75-01	2	Feb 5,2002
R84	Natural Rubber Screen Somersby 526	New	Jul 6,2000
R85	Natural Rubber (Handlay) Somersby 510H Off White	New	Jul 6,2000
R87	44 Duro Natural Rubber	New	Apr 21,2005
R88	55 Duro Natural Rubber	New	Apr 21,2005
R89	70 Duro Natural Rubber	New	Apr 21,2005
R91	60 Duro Natural Rubber	New	Apr 21,2005

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Code	Material Name	Revision	
R92	45 Duro Natural Rubber	New	Apr 21,2005
R99	Natural Rubber (External)	2	May 13,1998
S00	EPDM Rubber (External)	1	May 13,1998
S01	EPDM Seal Rubber	12	Dec 5,2011
S02	EPDM Liner Rubber	8	Dec 5,2011
S03	High Temperature EPDM	1	Nov 18,1999
S04	Lubricated EPDM Rubber	New	Mar 22,2004
S05	EPDM 70 Shore A	0	Apr 1,2009
S06	EPDMl Rubber Sheet	0	Nov 24,2010
S08	Food Grade Nitrile Rubber	0	Sep 20,2011
S09	White Buna Nitrile Rubber	0	Sep 14,2011
S10	Nitrile Rubber (External)	1	Aug 21,1990
S11	65 Duro Nitrile Rubber	1	Jul 10,2009
S12	Nitrile Rubber NBR	6	Sep 12,1997
S13	Nitrile Rubber (Ornel)	6	Sep 12,1997
S14	Nitrile Rubber (70Duro)	4	May 13,1998
S15	Linatrile	0	Dec 16,2009
S16	Hydrogenated Nitrile Rubber HNBR	New	Mar 22,2004
S17	HNBR 70 Shore A	0	Apr 1,2009
S18	Nitrile Seal Rubber	3	Nov 18,1999
S19	Linagard NBR Nitrile Rubber Sheet	0	Feb 10,2011
S20	Butyl Rubber (External)	2	May 13,1998
S21	Butyl Rubber IIR	0	Apr 14,2008
S21B	Halobutyl Rubber	5	May 13,1998
S22	Butyl Rubber (Strainers)	3	May 13,1998
S23	65 Duro Bromobutyl Rubber	New	Apr 21,2005
S24	Linagard BB Bromo Butyl Rubber Sheet	0	Feb 10,2011
S30	Chlorosulphonated polyethylene (external)	1	Aug 21,1990
S31	Chlorosulphonated Polyethylene	11	Nov 12,2013
S33	Delta Valve Sleeve - Hypalon Linatex	0	Jan 24,2011
S34	70 Duro Hypalon Rubber	New	Apr 21,2005
S35	CSM Rubber (Hypalon)	0	Sep 20,2011
S40	Polychloroprene (External)	2	May 13,1998
S41	60 Duro Chloroprene Rubber	New	Apr 21,2005
S42	Polychloroprene	0	Sep 5,2011
S43	Polychloroprene (60 Duro)	5	Nov 20,1990
S44	Polychloroprene (70 Duro)	8	Jun 13,2012
S45	High Temperature Hydrocarbon Resist.	5	May 13,1998
S46	65 Duro Chloroprene Rubber	New	Apr 21,2005
S47	55 Duro Chloroprene	1	Aug 15,2012

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Code	Material Name	Revision	
S48	65 Duro Chloroprene Rubber	New	Apr 21,2005
S49	65 Duro Chloroprene Rubber	New	Apr 21,2005
S50	Fluoroelastomer (External)	2	Sep 28,2011
S51	Fluoroelastomer (Viton)	New	May 13,1998
S52	Fluoroelastomer	6	Sep 9,1999
S53	70 Duro Viton Rubber	New	Apr 21,2005
S54	Linatex Viton	0	Apr 14,2011
S60	SILICON RUBBER	New	May 15,2000
S61	70 Duro NR/PBD/CR blend	New	May 2,2005
S62	70 Duro NR/PBD/CR blend	New	May 2,2005
S63	Silicone	New	May 20,2009
S70	Bromobutyl / Neoprene Rubber 70 A	0	Jan 28,2010
S71	Neoprene Rubber - Chile Compound 668	0	Sep 3,2013
S72	Polychloroprene - VN72	0	Oct 16,2013
S90	Polychloroprene Envirotech 545 Somersby	2	Feb 15,2002
S91	Polychloroprene Envirotech 535 Handlay	1	Jul 6,2000
S92	Hypalon - Somersby 556	New	Jul 6,2000
S93	Styrene Butadiene Rubber SBR	New	Jul 6,2000
S94	Polychloroprene 70 Shore A	0	Apr 1,2009
S95	EPDM 567 Rubber	New	Mar 22,2004
S96	65 Duro Styrene Butadiene Rubber	New	Apr 21,2005
S98	Linagard OSR Polychloroprene Rubber Sheet	0	Nov 16,2010
S99	Polychloroprene	0	Sep 20,2001
T01	Bearing Composite	New	May 31, 1990
T23	Fibre Glass Jointing	New	May 31, 1990
T31	Red Vulcanised Fibre	New	May 13, 1998
T41	Bearing Composite	New	May 31, 1990
T50	Compressible Vegetable Fibre Gasket	New	Jul 6, 2000
T51	Gasket - synthetic fibre non asbestos joining	New	Jul 7, 2000
T52	Compressed Non-Asbestos Gasketing	New	Jul 7, 2000
T53	Composite Elastomeric Bearing Thordon SXL	New	Mar 1, 2000
T54	Composite Bearing Vesconite	New	Mar 1, 2000
T55	Aramid Fibre Blended Nitrile Gasket	New	Mar 1, 2000
T56	Glass Fibre Blended Nitrile Gasket	New	Mar 1, 2000
T57	Carbon Fibre Blended Nitrile Gasket	New	Feb 8, 2012
T60	Leather	New	Mar 19, 2004
T61	Synthetic Leather	New	Mar 19, 2004
U01	Wear Resistant Polyurethane	7	Jan 21,2002
U02	Wear Resistant Polyurethane	6	May 13,1998
U03	Wear Resistant Polyurethane	7	May 13,1998

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Code	Material Name	Revision	
U07	Wear Resistant Polyurethane	3	May 13,1998
U15	83 Duro Polyurethane	New	Apr 21,2005
U27	60A Wear Resistant Polyurethane	2	Feb 13,2001
U31	MDI Ether Polyurethane - Linatex 101	0	Sep 2,2013
U32	MDI Ester Polyurethane - Linatex 183	0	Sep 2,2013
U33	Polyester Polyurethane - Linatex 885	0	Sep 2,2013
U34	PTMEG Polyurethane - Linatex 890	0	Sep 2,2013
U35	PTMEG Polyurethane - Linatex 895	New0	Sep 10,2013
U36	PTMEG Polyurethane - Linatex 121	0	Oct 16,2013
U37	PTMEG Polyurethane - Linatex 131	0	Oct 16,2013
U38	Polyurethane (MDI - Polyether)	1	Aug 26,2011
U39	57 Shore D Polyurethane	1	Oct 10,2007
U49	Polyurethane Spray	0	Aug 18,2009
U80	Polyurethane Foam	New	Mar 12,1998
U85	Polyurethane	New	Jan 31,2012
V19	WC / NiSiB Coating Material	0	Jan 14, 2011
V21	Tungsten Carbide/Cobalt Powder	2	Apr 20, 1995
V22	Tungsten Carbide/Cobalt/Chromium Powder	New	Apr 20, 1995
V23	Tungsten Carbide/Nickel/Chromium Powder	New	Apr 20, 1995
V24	Molybond 1PX1 Coating	1	Aug 1, 2006
V25	RIE4090/4606 Coating	0	Oct 28, 2009
V26	DACROMET Coating	0	May 10, 2012
V36	WC Based Coating Powder	3	Jun 5, 2003
V38	Spherical WC Coating Powder	0	Feb 3, 2009
V54	Grey Alumina Coating	0	May 11, 2010
V91	Chromium Carbide Weld Deposit (Grade 2355)	0	Jun 11, 2008
W01	Material Composite C21/D21/R55	0	May 29, 2008
W02	Material Composite E02/R35	0	Sep 5, 2011
W03	Material Composite S42/E02	0	Oct 4, 2011
W04	Material Composite S42/E05	0	Oct 4, 2011
W05	Material Composite S42/D21	0	Oct 4, 2011
W06	Material Composite S12/D21	0	Oct 13, 2011
W07	Material Composite C21/E02/R55	0	Oct 18, 2011
W08	Material Composite Y12 / U01	0	Nov 17, 2011
W09	Material Composite Y12 / E02 / U01	0	Nov 17, 2011
W10	Material Composite C23/U38	0	Dec 8, 2011
W11	Material Composite C73/U38	0	Dec 8, 2011
W12	Material Composite Y14 / U85	0	Jan 31, 2012
W13	Material Composite R55/C19	0	May 9, 2012
W14	Material Composite C73/R35	0	May 8, 2013

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Code	Material Name	Revision	
W15	Material Composite C73/R68	0	May 8, 2013
W16	Material Composite Y02/U03	0	Jul 12, 2013
W17	Material Composite C73/S12	0	Sep 27, 2013
W18	Material Composite A61/U38	0	Oct 30, 2013
W19	Material Composite C22/R55	0	Nov 13, 2013
W20	Material Composite C26/R55	0	Nov 13, 2013
W21	Material Composite E02/R37	0	Dec 10, 2013
W26	Material Composite L02/S20	0	Mar 10, 2014
W27	Material Composite L02/S40	0	Mar 10, 2014
W28	Material Composite L12/S20	0	Mar 11, 2014
W29	Material Composite L12/S40	0	Mar 11, 2014
W91	Chromium Carbide Hardfacing Plate V91/E02	0	Jun 11, 2008
Y02	Alumina 92%	1	May 4, 2012
Y03	Chromium Oxide	1	Jun 14, 1993
Y04	Alumina 85%	0	Oct 14, 2010
Y05	Alumina 96%	0	Sep 5, 2008
Y06	Sintered Silicon Carbide	1	Sep 11, 1997
Y07	Alumina 99%	New	Jun 1, 1990
Y08	Silicon Nitride Bonded Silicon Carbide	1	Sep 12, 1997
Y09	Alumina 94%	0	Sep 8, 2008
Y11	Fine Grained SiN / SiC	New	Sep 22, 1998
Y12	Double Sintered Nitride Bonded Silicon Carbide	0	Oct 20, 2011
Y14	Fine Grained Reaction Bonded SiC	2	Jul 19, 2011
Y15	Tungsten Carbide (94 %)	0	Oct 25, 2013
Y30	Poxilac 983-07/57 Elastic Epoxy Putty	New	Jul 29, 2004
Z00	Material Composite E02/K24	2	Aug 7, 1991
Z01	Material Composite C23/K24	1	Aug 7, 1991
Z02	Material Composite E02/C23	2	Aug 7, 1991
Z03	Material Composite E02/C22	2	Aug 7, 1991
Z04	Material Composite E02/C61	New	Sep 6, 2004
Z05	Materials Composite A05/R38	New	
Z06	Material Composite A08/R38	New	Jun 23, 2006
Z07	Material Composite Y07/A12	2	Jun 4, 1993
Z08	Materials Composite Y06/E02	New	Apr 12, 2005
Z09	Materials Composite Y07/E02	0	Jul 30, 2008
Z10	Material Composite Y11/U03	New	Feb 15, 2001
Z11	Material Composite Y11 / U01	New	Sep 23, 1998
Z12	Material Composite Y11/A12	New	Sep 23, 1998
Z13	Material Composite Y11/A05	New	Sep 23, 1998
Z14	Material Composite Y14 / U80	New	Mar 5, 1998

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Code	Material Name	Revision	
Z145		New	Sep 11, 1997
Z15	Material Composite Y08 / U01	3	Dec 24, 1996
Z151		New	Sep 12, 1997
Z16	Material Composite Y08/A05	2	Dec 24, 1996
Z17	Material Composite Y07/A05	2	Dec 24, 1996
Z18	Material Composite Y07 / U02	New	Dec 7, 2005
Z19	Material Composite Y14 / U02	New	Dec 7, 2005
Z20	Material Composite E02/S30	1	May 9, 2011
Z21	Material Composite E02/S40	2	Aug 7, 1991
Z22	Material Composite E02/S20	2	Aug 7, 1991
Z23	Material Composite E02/R00	4	Jun 4, 2010
Z24	Material Composite K24/R26	New	Jul 12, 1990
Z25	Material Composite E02/S01	0	Jul 31, 2007
Z26	Material Composite E02/S02	0	Jul 31, 2007
Z27	Material Composite C43/R08	New	Jul 12, 1990
Z28	Material Composite C23/R08	New	Jul 12, 1990
Z29	Material Composite C23/S42	New	Jul 12, 1990
Z30	Material Composite C23/S21	New	Jul 12, 1990
Z31	Material Composite C23/S31	New	Jul 12, 1990
Z32	Material Composite C23/S51	New	Jul 12, 1990
Z33	Material Composite C23/R49	New	Jul 12, 1990
Z34	Material Composite D81/Y03	1	Jun 19, 1995
Z35	Material Composite C23/U01	New	Jul 12, 1990
Z36	Material Composite C22/R00	New	Aug 2, 1991
Z37	Materials Composite Y07/R33	New	Mar 30, 2005
Z38	Material Composite C21/R38	New	Sep 21, 2005
Z39	Material Composite Y08 / C23	0	May 14, 2009
Z40	Material Composite Z02/R26	New	Jun 19, 1998
Z41	Material Composite C27/R49	New	Jul 12, 1990
Z42	Materials Composite P133/S42	New	Oct 13, 2005
Z43	Material Composite E21/R00	New	Jul 6, 2000
Z44	Material Composite E21/R85	New	Jul 6, 2000
Z45	Material Composite E02/C23	New	Jul 6, 2000
Z46	Material Composite R49/S14	New	Aug 2, 2000
Z47	Material Composite E02/D21	0	Apr 22, 2008
Z48	Material Composite R55/Z47	0	Apr 22, 2008
Z49	Material Composite M01/R49	New	Jul 12, 1990
Z50	Material Composite E38/R55	0	Apr 3, 2012
Z51	Material Composite M01/S51	New	Jul 12, 1990
Z52	Material Composite M01/S52	New	Aug 1, 1994

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Code	Material Name	Revision	
Z53	Material Composite Y14 / U38	1	Jan 11, 2012
Z54	Material Comosite Y14 / R55	New	Dec 7, 2005
Z55	Material Composite A08/R55	0	Sep 22, 2006
Z56	Material Composite A05/R55	0	Sep 28, 2006
Z57	Material Composite Y07 / R55	New	Dec 7, 2005
Z58	Material Composite Y08 / R55	New	Dec 7, 2005
Z59	Material Composite A07 / R55	New	Mar 20, 2009
Z60	Material Composite E02/T01	1	Jul 12, 1990
Z61	Material Composite E02/T01	New	Aug 7, 1991
Z62	Material Composite D21/R55	0	Mar 13, 2008
Z63	Material Composite E02/R55	0	Mar 13, 2008
Z64	Material Composite G01/R55	0	Mar 13, 2008
Z65	Material Composite C21/R55	0	Dec 3, 2008
Z67	Material Composite Y07 / Y30	New	Dec 7, 2005
Z68	Material Composite Y08 / Y30	New	Dec 7, 2005
Z69	Material Composite Y14 / Y30	New	Dec 7, 2005
Z70	Material Composite E02/U38	New	Nov 20, 2009
Z71	Material Composite D21/U01	New	Jul 12, 1990
Z72	Material Composite E02/D21	1	Jun 4, 2010
Z73	Material Composite D21/R01	New	Mar 16, 2009
Z74	Material Composite L12/R01	New	Mar 16, 2009
Z75	Material Composite E05/R00	New	Jul 24, 2008
Z76	Material Composite Y05/E63	0	Sep 9, 2008
Z77	Material Composite Y07/E63	0	Sep 9, 2008
Z78	Material Composite Y08/E63	0	Sep 9, 2008
Z79	Materials Composite Y02/E02	0	Jun 21, 2013
Z80	Material Composite Z45/R00	New	Jul 6, 2000
Z81	Material Composite Z02/S12	New	Jul 6, 2000
Z82	Material Composite P21/R80	New	Feb 27, 2000
Z83	Materials Composite P05/S51	0	Jul 17, 2009
Z84	Materials Composite P60/S05	0	Jul 17, 2009
Z85	Material Composite C41/S15	New	Dec 16, 2009
Z86	Material Composite A61/R55	0	Oct 2, 2009
Z87	Material Composite C23/R55	0	Jan 20, 2010
Z88	Material Composite E02/D20	0	Feb 17, 2010
Z89	Material Composite R55/D20	0	Jul 22, 2010
Z90	Material Composite A05/S42	0	Dec 20, 2010
Z91	Material Composite C41/S42	0	Sep 1, 2008
Z92	Material Composite P05 / C73	0	Jul 14, 2009
Z93	Materials Composite P21/R33	New	May 14, 2002

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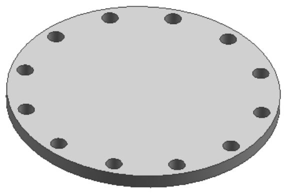
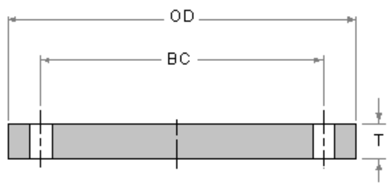
Z94	Materials Composite P21/R45	0	Dec 13, 2006
Z95	Materials Composite P21/R55	0	Dec 13, 2006
Z96	Material Composite P33 / R55	0	Jul 12, 2007
Z97	Material Composite C23/S40	0	Aug 10, 2010
Z98	Material Composite A215/R55	0	Oct 12, 2010
Z99	Material Composite J40/R55	0	Oct 12, 2010

DNBrida

Nota:
Class B: 86 psi
Class D: 175-150 psi
Class E: 275 psi
Class F: 300 psi

Brida AWWA C207 Ciega Class B (86 psi)





El ratio de presión a temperatura atmosférica es de 86 psi.

Dimensiones en pulgadas y (milímetros)

Tamaño Nominal Tubería	Ø exterior (OD)	Nº de taladros	Ø tornillo	Ø Círculo tal (BC)	Espesor mín. (T)	Peso (lbs)
4 (100)	9 (228.6)	8	0.625 (15.9)	7.5 (190.5)	0.625 (15.88)	10.8 (4.9)
5 (125)	10 (254)	8	0.75 (19.1)	8.5 (215.9)	0.625 (15.88)	13.2 (6)
6 (150)	11 (279.4)	8	0.75 (19.1)	9.5 (241.3)	0.688 (17.48)	17.9 (8.1)
8 (200)	13.5 (342.9)	8	0.75 (19.1)	11.75 (298.5)	0.688 (17.48)	27.3 (12.4)
10 (250)	16 (406.4)	12	0.875 (22.2)	14.25 (362)	0.688 (17.48)	37.9 (17.2)
12 (300)	19 (482.6)	12	0.875 (22.2)	17 (431.8)	0.719 (18.26)	56.4 (25.6)
14 (350)	21 (533.4)	12	1 (25.4)	18.75 (476.3)	0.791 (20.09)	75.8 (34.4)
16 (400)	23.5 (596.9)	16	1 (25.4)	21.25 (539.8)	0.892 (22.66)	107.4 (48.7)
18 (450)	25 (635)	16	1.125 (28.6)	22.75 (577.9)	0.95 (24.13)	129.2 (58.6)
20 (500)	27.5 (698.5)	20	1.125 (28.6)	25 (635)	1.04 (26.42)	171.3 (77.7)
22 (550)	29.5 (749.3)	20	1.25 (31.8)	27.25 (692.2)	1.132 (28.75)	214.1 (97.1)
24 (600)	32 (812.8)	20	1.25 (31.8)	29.5 (749.3)	1.216 (30.89)	272.1 (123.4)
26 (650)	34.25 (870)	24	1.25 (31.8)	31.75 (806.5)	1.307 (33.2)	334.7 (151.8)
28 (700)	36.5 (927.1)	28	1.25 (31.8)	34 (863.6)	1.398 (35.51)	406.3 (184.3)
30 (750)	38.75 (984.3)	28	1.5 (38.1)	36 (914.4)	1.477 (37.52)	481.9 (218.6)
32 (800)	41.75 (1060.5)	28	1.5 (38.1)	38.5 (977.9)	1.581 (40.16)	600.8 (272.5)
34 (850)	43.75 (1111.3)	32	1.5 (38.1)	40.5 (1028.7)	1.661 (42.19)	693.1 (314.4)
36 (900)	46 (1168.4)	32	1.5 (38.1)	42.75 (1085.9)	1.751 (44.48)	809.3 (367.1)
38 (950)	48.75 (1238.3)	32	1.5 (38.1)	45.25 (1149.4)	1.853 (47.07)	965 (437.7)

40 (1000)	50.75 (1289.1)	36	1.5 (38.1)	47.25 (1200.2)	1.933 (49.1)	1091.1 (494.9)
42 (1050)	53 (1346.2)	36	1.5 (38.1)	49.5 (1257.3)	2.023 (51.38)	1245.4 (564.9)
44 (1100)	55.25 (1403.4)	40	1.5 (38.1)	51.75 (1314.5)	2.114 (53.7)	1414.9 (641.8)
46 (1150)	57.25 (1454.2)	40	1.5 (38.1)	53.75 (1365.3)	2.194 (55.73)	1579.4 (716.4)
48 (1200)	59.5 (1511.3)	44	1.5 (38.1)	56 (1422.4)	2.285 (58.04)	1774.3 (804.8)
50 (1250)	61.75 (1568.5)	44	1.5 (38.1)	58.25 (1479.6)	2.377 (60.38)	1991.4 (903.3)
52 (1300)	64 (1625.6)	44	1.5 (38.1)	60.5 (1536.7)	2.468 (62.69)	2224.2 (1008.9)
54 (1350)	66.25 (1682.8)	44	1.75 (44.5)	62.75 (1593.9)	2.559 (65)	2460.6 (1116.1)
60 (1500)	73 (1854.2)	52	1.75 (44.5)	69.25 (1759)	2.82 (71.63)	3293.9 (1494.1)
66 (1650)	80 (2032)	52	1.75 (44.5)	76 (1930.4)	3.092 (78.54)	4350.2 (1973.2)
72 (1800)	86.5 (2197.1)	60	1.75 (44.5)	82.5 (2095.5)	3.353 (85.17)	5516.4 (2502.2)
78 (1950)	93 (2362.2)	64	2 (50.8)	89 (2260.6)	()	()
84 (2100)	99.75 (2533.7)	64	2 (50.8)	95.5 (2425.7)	()	()
90 (2250)	106.5 (2705.1)	68	2.25 (57.2)	102 (2590.8)	()	()
96 (2400)	113.25 (2876.6)	68	2.25 (57.2)	108.5 (2755.9)	()	()
102 (2550)	120 (3048)	72	2.5 (63.5)	114.5 (2908.3)	()	()
108 (2700)	126.75 (3219.5)	72	2.5 (63.5)	120.75 (3067.1)	()	()
114 (2850)	133.5 (3390.9)	76	2.75 (69.9)	126.75 (3219.5)	()	()
120 (3000)	140.25 (3562.4)	76	2.75 (69.9)	132.75 (3371.9)	()	()
126 (3150)	147 (3733.8)	80	3 (76.2)	139.25 (3537)	()	()
132 (3300)	153.75 (3905.3)	80	3 (76.2)	145.75 (3702.1)	()	()
144 (3600)	167.25 (4248.2)	84	3.25 (82.6)	158.25 (4019.6)	()	()

Notas:

1. Todas las bridas son de cara plana.

2. Conversión métrica:
Tamaño Nominal de la tubería [in] x 25 = [mm]
Dimensiones [in] x 25,4 = [mm]
Presión [psi] x 6,895 = [kPa]

 c/ Miquel i Badia 12-14 4-4 · 08024 Barcelona, España · telf. +34 690288839 info@dnbrida.com

DNBrida

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Brida Norma

· ANSI/ASME
· AWWA C207
· DIN
· EN-1092-1
· BS-3293
· EEMUA 145
· JIS B2220

Bridas AWWA C207

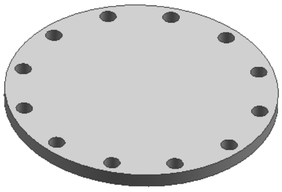
· Brida Ring AWWA C207 Class B
· Brida Ring AWWA C207 Class D
· Brida Ring AWWA C207 Class E
· Brida Ring AWWA C207 Class F

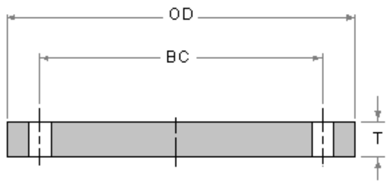
· Brida Ciega AWWA C207 Class B
· Brida Ciega AWWA C207 Class D
· Brida Ciega AWWA C207 Class E
· Brida Ciega AWWA C207 Class F

· Brida Hub AWWA C207 Class D
· Brida Hub AWWA C207 Class E

Nota:
Class B: 86 psi
Class D: 175-150 psi
Class E: 275 psi
Class F: 300 psi

Brida AWWA C207 Ciega Class D (175-150 psi)





Máximo ratio de presión a temperatura atmosférica:

• Tamaño de 4 a 12 in (100 a 300 mm): 175 Psi (1,207 KPa)
• Tamaño mayor de 12 in (300 mm): 150 Psi (1,034 KPa)

Dimensiones en pulgadas y (milímetros)

Tamaño Nominal Tubería	Ø exterior (OD)	Nº de taladros	Ø tornillo	Ø Círculo tal (BC)	Espesor mín. (T)	Peso lbs (kg)
4 (100)	9 (228.6)	8	0.625 (15.9)	7.5 (190.5)	0.625 (15.88)	10.8 (4.9)
5 (125)	10 (254)	8	0.75 (19.1)	8.5 (215.9)	0.65 (16.51)	13.9 (6.3)
6 (150)	11 (279.4)	8	0.75 (19.1)	9.5 (241.3)	0.693 (17.6)	18.1 (8.2)
8 (200)	13.5 (342.9)	8	0.75 (19.1)	11.75 (298.5)	0.812 (20.62)	32.2 (14.6)
10 (250)	16 (406.4)	12	0.875 (22.2)	14.25 (362)	0.953 (24.21)	52.9 (24)
12 (300)	19 (482.6)	12	0.875 (22.2)	17 (431.8)	1.117 (28.37)	88.2 (40)
14 (350)	21 (533.4)	12	1 (25.4)	18.75 (476.3)	1.133 (28.78)	108.7 (49.3)
16 (400)	23.5 (596.9)	16	1 (25.4)	21.25 (539.8)	1.265 (32.13)	152.1 (69)
18 (450)	25 (635)	16	1.125 (28.6)	22.75 (577.9)	1.331 (33.81)	180.6 (81.9)
20 (500)	27.5 (698.5)	20	1.125 (28.6)	25 (635)	1.448 (36.78)	237.7 (107.8)
22 (550)	29.5 (749.3)	20	1.25 (31.8)	27.25 (692.2)	1.568 (39.83)	295.6 (134.1)
24 (600)	32 (812.8)	20	1.25 (31.8)	29.5 (749.3)	1.661 (42.19)	370.2 (167.9)
26 (650)	34.25 (870)	24	1.25 (31.8)	31.75 (806.5)	1.786 (45.36)	455.7 (206.7)
28 (700)	36.5 (927.1)	28	1.25 (31.8)	34 (863.6)	1.906 (48.41)	552.7 (250.7)
30 (750)	38.75 (984.3)	28	1.5 (38.1)	36 (914.4)	2.008 (51)	652.3 (295.9)
32 (800)	41.75 (1060.5)	28	1.5 (38.1)	38.5 (977.9)	2.15 (54.61)	813.7 (369.1)
34 (850)	43.75 (1111.3)	32	1.5 (38.1)	40.5 (1028.7)	2.252 (57.2)	936.1 (424.6)
36 (900)	46 (1168.4)	32	1.5 (38.1)	42.75 (1085.9)	2.37 (60.2)	1091.1 (494.9)

38 (950)	48.75 (1238.3)	32	1.5 (38.1)	45.25 (1149.4)	2.506 (63.65)	1300.5 (589.9)
40 (1000)	50.75 (1289.1)	36	1.5 (38.1)	47.25 (1200.2)	2.609 (66.27)	1467.6 (665.7)
42 (1050)	53 (1346.2)	36	1.5 (38.1)	49.5 (1257.3)	2.729 (69.32)	1676 (760.2)
44 (1100)	55.25 (1403.4)	40	1.5 (38.1)	51.75 (1314.5)	2.849 (72.36)	1901.9 (862.7)
46 (1150)	57.25 (1454.2)	40	1.5 (38.1)	53.75 (1365.3)	2.952 (74.98)	2115.1 (959.4)
48 (1200)	59.5 (1511.3)	44	1.5 (38.1)	56 (1422.4)	3.072 (78.03)	2378.3 (1078.8)
50 (1250)	61.75 (1568.5)	44	1.75 (44.5)	58.25 (1479.6)	3.196 (81.18)	2654.4 (1204)
52 (1300)	64 (1625.6)	44	1.75 (44.5)	60.5 (1536.7)	3.315 (84.2)	2964.1 (1344.5)
54 (1350)	66.25 (1682.8)	44	1.75 (44.5)	62.75 (1593.9)	3.435 (87.25)	3294.4 (1494.3)
60 (1500)	73 (1854.2)	52	1.75 (44.5)	69.25 (1759)	3.779 (95.99)	4405.7 (1998.4)
66 (1650)	80 (2032)	52	1.75 (44.5)	76 (1930.4)	4.136 (105.05)	5806.8 (2633.9)
72 (1800)	86.5 (2197.1)	60	1.75 (44.5)	82.5 (2095.5)	4.48 (113.79)	7358.4 (3337.7)
78 (1950)	93 (2362.2)	64	2 (50.8)	89 (2260.6)	()	()
84 (2100)	99.75 (2533.7)	64	2 (50.8)	95.5 (2425.7)	()	()
90 (2250)	106.5 (2705.1)	68	2.25 (57.2)	102 (2590.8)	()	()
96 (2400)	113.25 (2876.6)	68	2.25 (57.2)	108.5 (2755.9)	()	()
102 (2550)	120 (3048)	72	2.5 (63.5)	114.5 (2908.3)	()	()
108 (2700)	126.75 (3219.5)	72	2.5 (63.5)	120.75 (3067.1)	()	()
114 (2850)	133.5 (3390.9)	76	2.75 (69.9)	126.75 (3219.5)	()	()
120 (3000)	140.25 (3562.4)	76	2.75 (69.9)	132.75 (3371.9)	()	()
126 (3150)	147 (3733.8)	80	3 (76.2)	139.25 (3537)	()	()
132 (3300)	153.75 (3905.3)	80	3 (76.2)	145.75 (3702.1)	()	()
144 (3600)	167.25 (4248.2)	84	3.25 (82.6)	158.25 (4019.6)	()	()

Notas:

1. Todas las bridas son de cara plana.

2. Conversión métrica:
Tamaño Nominal de la tubería [in] x 25 = [mm]
Dimensiones [in] x 25,4 = [mm]
Presión [psi] x 6,895 = [kPa]

FABRICACION DE HERRAMIENTAS ESPECIALES DE BOMBAS								
No	**FASES A INSPECCIONAR**	**CARACTERISTICA A INSPECCIONAR**	**METODO INSPECCION**	**DOCUMENTO DE REFERENCIA**	**CRITERIO DE ACEPTACION**	**REGISTRO APLICABLE**	**RESPONSABLE**	
							FABRICANTE	**VULCO**
1	Revisión de documentación de fabricación.	Requerimientos fabricación.	Documentario	- Especificaciones - Planos de fabricación aprobados por Ingeniería Vulco.	N.A.	N.A	DR	DR
2	Materiales	- Certificados de materiales (acero, material de aporte, pernería, etc).	- Visual - Dimensional - Instrumental	- Especificaciones - Planos	-Especificaciones del material.	N.A.	WP	DR
3	Soldadura	- WPS/PQR - WPQ (soldador) - Procedimiento de reparación.	- Visual - Dimensional - Instrumental	- Planos. - AWS D1.1	Según AWS D1.1	Registrar documentos de sustento.	WP	DR
4	Inspección y ensayos no destructivos.	- Visual soldadura 100%. - 100% de cordones MT ó PT u otro por indicación de Ingeniería.	- Visual - Instrumental - NDT	-Planos. -Procedimiento de NDT. -AWS D1.1	Según AWS D1.1	Registros de ensayos no destructivos	HP	WP

FABRICACION DE HERRAMIENTAS ESPECIALES DE BOMBAS								
No	**FASES A INSPECCIONAR**	**CARACTERISTICA A INSPECCIONAR**	**METODO INSPECCION**	**DOCUMENTO DE REFERENCIA**	**CRITERIO DE ACEPTACION**	**REGISTRO APLICABLE**	**RESPONSABLE**	
							FABRICANTE	**VULCO**
5	Control dimensional	- Dimensional.	- Visual - Instrumental	-Plano de fabricación.	-Dimensiones según plano.	-Registro de control dimensional.	WP	DR
6	Prueba de ensamble/funcionamiento y verificación de número de serie.	- Dimensional. - Funcional	- Visual - Instrumental	-Plano de fabricación.	-Ensamble de acuerdo al plano.	N.A.	HP	WP
7	Preparación superficial y pintura.	- Grado de preparación de superficie. - Espesor de película seca.	- Visual - Instrumental -Document.	- De acuerdo a especificaciones de Vulco.	-De acuerdo a especificaciones de Vulco.	-Según procedimientos del fabricante.	HP	WP

FABRICACION DE HERRAMIENTAS ESPECIALES DE BOMBAS								
No	**FASES A INSPECCIONAR**	**CARACTERISTICA A INSPECCIONAR**	**METODO INSPECCION**	**DOCUMENTO DE REFERENCIA**	**CRITERIO DE ACEPTACION**	**REGISTRO APLICABLE**	**RESPONSABLE**	
							FABRICANTE	**VULCO**
8	Liberación.	- Condiciones generales del producto. - Identificación trazable.	- Visual - Instrumental	- Plano	-Inspección física y documentaria aprobada por el área QC.	Dossier de calidad. Acta de Liberación del producto.	HP	HP

DR: Document Review WP: Witness Point N.A: Non Appliance HP: Hold Point

FABRICANTE: Vulco Perú y/o proveedor.

Preparado por:	Revisado por:	Aprobado por:
Mario Cubillas **Jefe de Laboratorio**	**Andrés Quintana** **Ingeniero de Diseño**	**José Luis Arellano** **Gerente de Ingeniería**