FACULTAD DE INGENIERÍA UNAM
DIVISIÓN DE EDUCACIÓN CONTINUA

A LOS ASISTENTES A LOS CURSOS

Las autoridades de la Facultad de Ingeniería, por conducto del Jefe de la División de Educación Continua, otorgan una constancia de asistencia a quienes cumplan con los requisitos establecidos para cada curso.

El control de asistencia se llevará a cabo a través de la persona que le entregó el material didáctico y será registrada por las autoridades de la División, con el fin de entregarle constancia a los alumnos que cumplan como mínimo el 80% de asistencia.

Recomendamos a los asistentes recojan su constancia en la fecha que se les señale al término del evento. La DECFI solo las retendrá por el periodo de un año, pasado este tiempo no se hará responsable de este documento.

Se recomienda a los alumnos participar activamente con sus ideas y experiencias , pues los cursos que ofrece la División están planeados para que los profesores expongan una tesis, pero sobre todo, para que coordinen las opiniones de todos los interesados, constituyendo verdaderos seminarios.

Es muy importante que todos los asistentes llenen y entreguen su solicitud de inscripción al inicio del curso, información que servirá para integrar un directorio de asistentes.

Con el objeto de mejorar los servicios que la División de Educación Continua ofrece, al final del curso deberán requisitar y entregar la evaluación a través de un cuestionario diseñado para emitir juicios anónimos.

Se recomienda llenar dicha evaluación conforme los profesores impartan sus clases, a efecto de llenar en la última sesión las evaluaciones y con esto sean más fehacientes sus apreciaciones.

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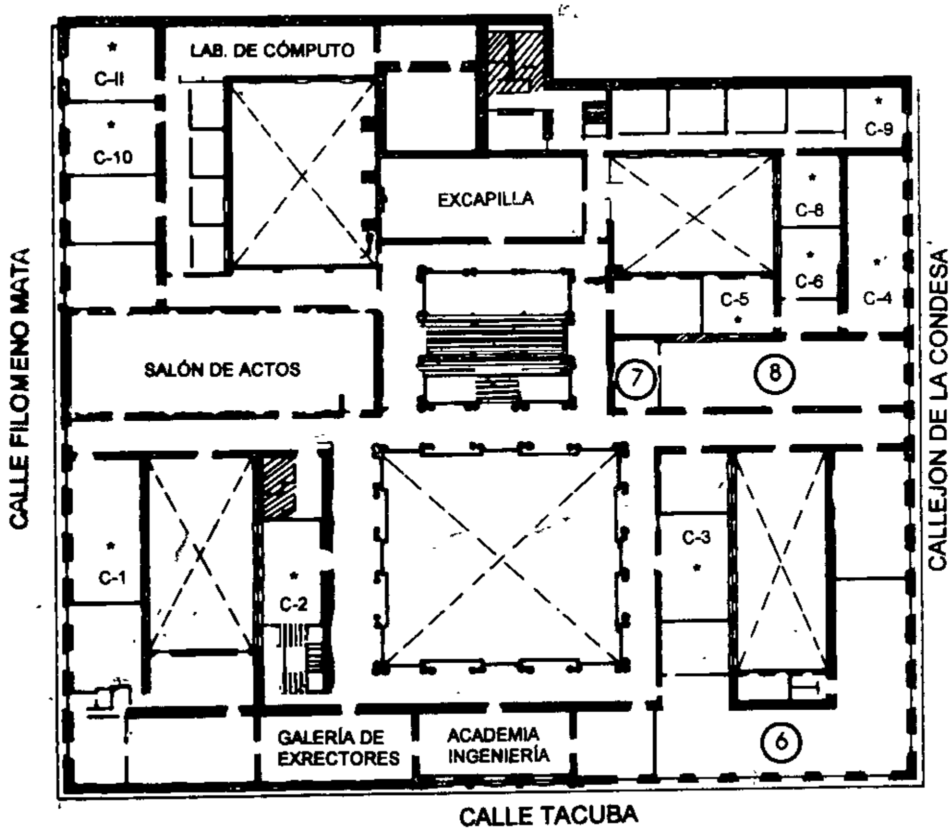

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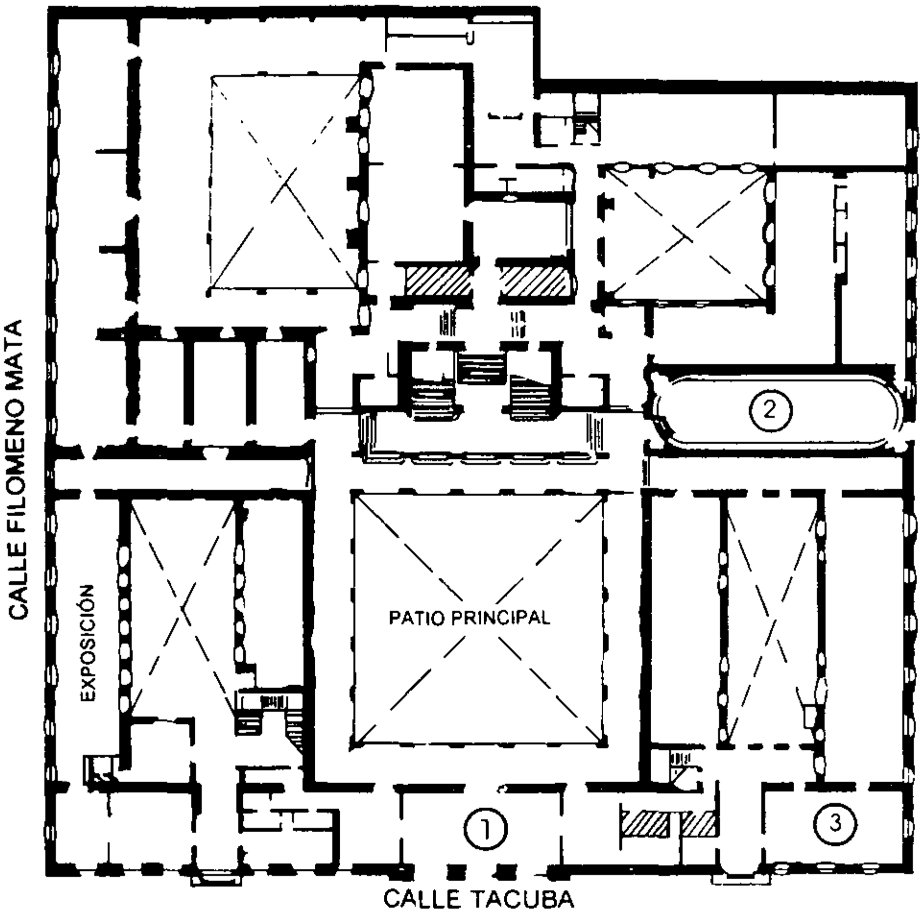

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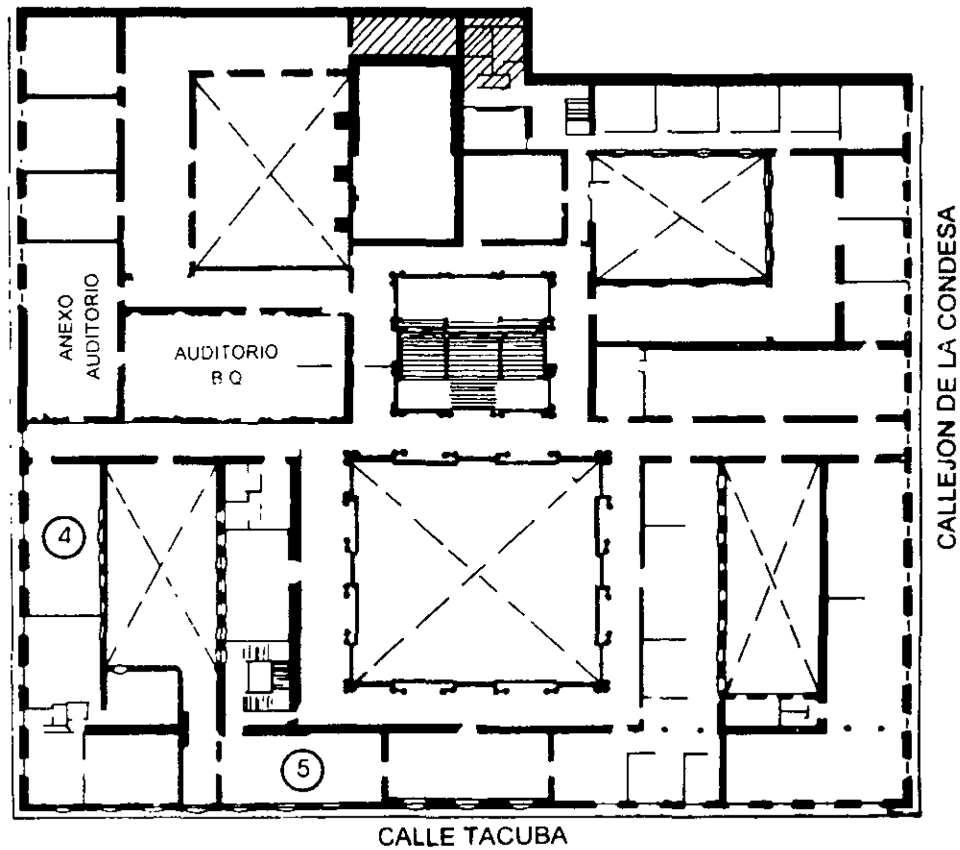


MEZZANINNE

División de Educación Continua, Facultad de Ingeniería, UNAM.

CURSOS ABIERTOS

DISEÑO DE CARCAMOS DE BOMBEO

CA 387

EXPOSITORES: M. EN I. FRANCISCO ORDUÑA BUSTAMANTE
DEL 09 AL 30 DE JUNIO DE 2007
PALACIO DE MINERÍA

CURSO EN LA DIVISIÓN DE EDUCACIÓN CONTÍNUA DE LA FACULTAD DE INGENIERÍA DE LA UNAM.

QUE PRESENTA:

M en I Francisco Orduña Bustamante

TEMA PROPUESTO:

DISEÑO DE CÁRCAMOS DE BOMBEO

DURACIÓN: **20 hrs.** (5 sesiones)

PROGRAMA:

I.- ANTECEDENTES.	(1 hr.)
II.- SELECCIÓN DE EQUIPOS CENTRÍFUGOS DE BOMBEO.	(5 hr.)
III.- DESCRIPCIÓN Y SELECCIÓN DE VÁLVULAS.	(6 hr.)
IV.- DISEÑO GEOMÉTRICO.	(2 hr.)
V.- DISEÑO FUNCIONAL.	(2 hr.)
VI.- EJEMPLOS DE APLICACIÓN.	(4 hr.)

DIRIGIDO A:

-Involucrados en la toma de decisiones en el diseño y operación de cárcamos y estaciones de bombeo.

-Interesados en el ahorro de energía eléctrica.

-Ingenieros involucrados en la selección de equipos de bombeo y válvulas.

PROBLEMÁTICA:

El diseño de cárcamos y estaciones de bombeo para manejo de aguas pluviales y municipales presenta un amplio espectro de problemas por resolver; Partiendo de la selección de los equipos de bombeo, gobernada muchas veces por su disponibilidad en el mercado, su facilidad de instalación o simplemente su haber, En México, el consumo de energía eléctrica empleado para alimentar equipos de bombeo representa el 21.5 % del total consumido en el país; Las condiciones de operación de instalaciones de bombeo pequeñas, lleva muchas veces al uso de válvulas de control, despreciándose en ocasiones su función como disipadoras de energía.

La normatividad aplicable para el diseño geométrico de los cárcamos de las cámaras de succión, queda muchas veces rebasada; Por ejemplo, en la República Mexicana, las normas aplicadas de manera típica a nivel internacional quedan generalmente rebasadas, debido a que las importantes elevaciones topográficas de nuestro país, no corresponden con las de los sitios donde se realizan las pruebas experimentales que dan sustento a las normas.

Aspectos como el contenido de gases disueltos, son muchas veces omitidos en las modelaciones tendientes al diseño, lo que redunda en problemas durante la fase operativa.

OBJETIVOS:

Participar de la conciencia del ahorro de energía eléctrica.

Exponer de manera sucinta aspectos importantes en el diseño geométricos de los cárcamos de bombeo, así como recomendaciones para la selección e instalación de equipos de bombeo y válvulas.

FACULTAD DE INGENIERÍA UNAM
DIVISIÓN DE EDUCACIÓN CONTINUA

División de Educación Continua, Facultad de Ingeniería, UNAM.

CURSOS ABIERTOS

DISEÑO DE CARCAMOS DE BOMBEO

CA 387

TEMA
ANTECEDENTES

EXPOSITORES: M. EN I. FRANCISCO ORDUÑA BUSTAMANTE
DEL 09 AL 30 DE JUNIO DE 2007
PALACIO DE MINERÍA

I.- ANTECEDENTES.

En México, el 21.5 % del total consumo de energía eléctrica se destina para activar motobombas. De la energía destinada al bombeo, el 85 % se destina a máquinas centrífugas, mientras que el 15 % restante se destina a máquinas de otro tipo (desplazamiento positivo). La eficiencia promedio con la que trabajan los equipos de bombeo en México se estima entre el 45 y el 60 %.

El uso de máquinas centrífugas para el manejo de líquidos está ampliamente difundido, de manera que es posible encontrar equipos de este tipo en una amplia gama de potencias y materiales de fabricación, desde acero inoxidable y plástico, hasta vidrio. El sector agrícola requiere de una gran cantidad de agua, por lo que es un importante ámbito para la adecuada selección de equipos de este tipo. Las industria química y metalúrgica utilizan también una gran cantidad de equipos de este tipo. Las instalaciones de mayor responsabilidad se encuentran en el abastecimiento de agua potable, y el enfriamiento de termoeléctricas y nucleoeléctricas; donde el funcionamiento continuo y eficiente de los sistemas de bombeo son de importancia vital.

Las turbomáquinas son artificios muy generosos, y altamente confiables, Así mismo, la responsabilidad de los ingenieros encargados de su selección y correcta instalación.

II.- DESCRIPCIÓN Y SELECCIÓN DE EQUIPOS DE BOMBEO.

La bomba centrífuga está constituida por un rotor dentro de una carcasa. El fluido entra axialmente en la carcasa en la dirección del eje de rotación del rotor; Los álabes del rotor impulsan al líquido para adquirir un movimiento tangencial y radial hacia el exterior del rotor donde es recogido por la carcasa. El fluido aumenta su velocidad y presión cuando transita a lo largo del álabe del rotor. La parte de la carcasa de forma toroidal llamada voluta, decelera el flujo y aumenta aún más la presión.

Normalmente los álabes están curvados hacia atrás, pero existen también diseños de álabes radiales y curvados hacia adelante, con lo cual se cambia ligeramente la presión de salida de la bomba. Los álabes pueden ser abiertos o cerrados, en este último caso los álabes están confinados entre dos discos y el conducto formado entre dos álabes está cerrado propiamente. El difusor puede tener álabes fijos o carecer de ellos.

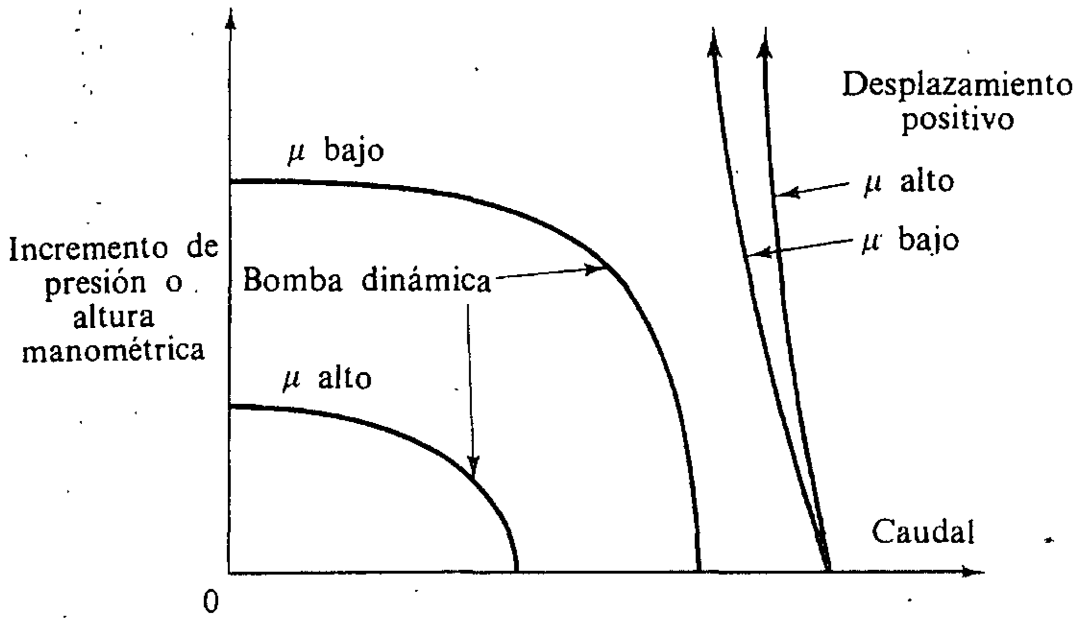


figura 1. Comaración del comportamiento de bombas del tipo centrífugas, respecto a equipos de desplazamiento positivo.

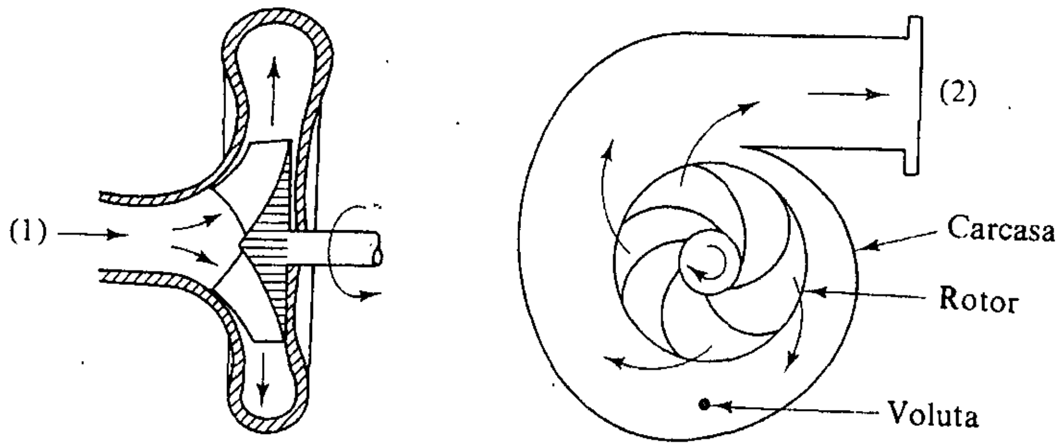


figura 2. Partes constitutivas básicas de una bomba centrífuga.

Las curvas características se trazan generalmente para velocidades de giro N constantes. El caudal Q se toma como variable independiente básica. Como variable dependiente se toma la altura manométrica H, la potencia al freno P_f y el rendimiento η. La altura manométrica (Carga Dinámica Total), permanece prácticamente constante con caudales bajos y después desciende ligeramente para alcanzar el punto de máxima eficiencia y cae súbitamente para alcanzar $H = 0$.

Un punto que a menudo no se tiene en cuenta, es que las curvas características que se obtienen para una máquina, son solo aplicables estrictamente a un fluido de una cierta densidad y viscosidad, generalmente agua. Si la bomba se usara para bombear por ejemplo, mercurio, la potencia al freno debería ser unas trece veces más alta, mientras que Q y η apenas cambiarían, incluso el valor de H sería igual, con la salvedad de que se estaría interpretando como metros de columna de mercurio y no metros de columna de agua. Si la misma máquina se utilizara para bombear aceite SAE 30, todos los valores (potencia al freno, Q, η, H) cambiarían debido a la gran diferencia de viscosidades (número de Reynolds).

Otra curva que se obtiene para las máquinas hidráulicas, es la Carga de succión Positiva Neta CSPN en castellano o NPSH por sus siglas en inglés, que es la presión que debe presentarse en el ojo del impulsor para evitar que la bomba cavite. La CSPN se define como:

$$CSPN_r = \frac{p_i}{\gamma} + \frac{v_i^2}{2g} - \frac{p_v}{\gamma}$$

Donde p_i y v_i son respectivamente la presión y la velocidad a la entrada de la bomba, y p_v es la presión de vaporización del líquido.

En una instalación particular, la máquina se instala a una altura z_i sobre el nivel estático del agua en la succión, así mismo, en el trayecto del conducto de succión, puede calcularse el valor de la carga disipada en el conducto h_{fi} en función de v_i, de este modo, la instalación tendrá un valor de $CSPN$ disponible, que se denominará $CSPN_d$, y se estimará como:

$$CSPN_d = \frac{p_a}{\gamma} - z_i - h_{fi} - \frac{p_v}{\gamma}$$

donde p_a es el valor de la presión atmosférica en el sitio donde se hará la instalación.

REGLAS DE SEMEJANZA.

$$\frac{Q_2}{Q_1}=\frac{N_2}{N_1}\left(\frac{D_2}{D_1}\right)^3$$

$$\frac{H_2}{H_1}=\left(\frac{N_2}{N_1}\right)^2\left(\frac{D_2}{D_1}\right)^2$$

$$\frac{P_2}{P_1}=\frac{\rho_2}{\rho_1}\left(\frac{N_2}{N_1}\right)^3\left(\frac{D_2}{D_1}\right)^5$$

Las reglas de semejanza pueden utilizarse para estimar el efecto de cambio de fluido, velocidad o tamaño de cualquier turbomáquina dinámica, bomba o turbina, dentro de una familia geométricamente semejante. De acuerdo a lo expresado podría esperarse que para dos máquinas semejantes $\eta_2=\eta_1$, pero esto no ocurre así. Las máquinas más grandes son más eficientes debido a que los efectos de la viscosidad se van reduciendo con el tamaño de la máquina, una fórmula empírica para estimar el cambio en el rendimiento debido al tamaño es:

$$\frac{1-\eta_2}{1-\eta_1}\approx\left(\frac{D_1}{D_2}\right)^{1/4}$$

Esta fórmula, primero desarrollada para turbinas, es muy usada tanto en bombas como en turbinas cuando se carece de datos de rendimiento.

LA VELOCIDAD ESPECÍFICA.

En muchas aplicaciones de las bombas, se conoce la Carga Dinámica Total H, y el caudal Q para el sistema en particular más un rango de velocidades de giro del motor eléctrico N. El proyectista selecciona el mejor tamaño y forma (centrífuga, helicocentrífuga, axial) de la bomba. Para ayudar a esta selección necesiatamos un parámetro adimensional que relacione la velocidad, caudal y Carga Dinámica Total. Esto se consigue eliminando el diámetro en los parámetros adimensionales obteniéndose una relación conocida como velocidad específica, que de manera estricta debería corresponder a un parámetro adimensional, pero que se maneja comúnmente en unidades inglesas:

Forma estricta:

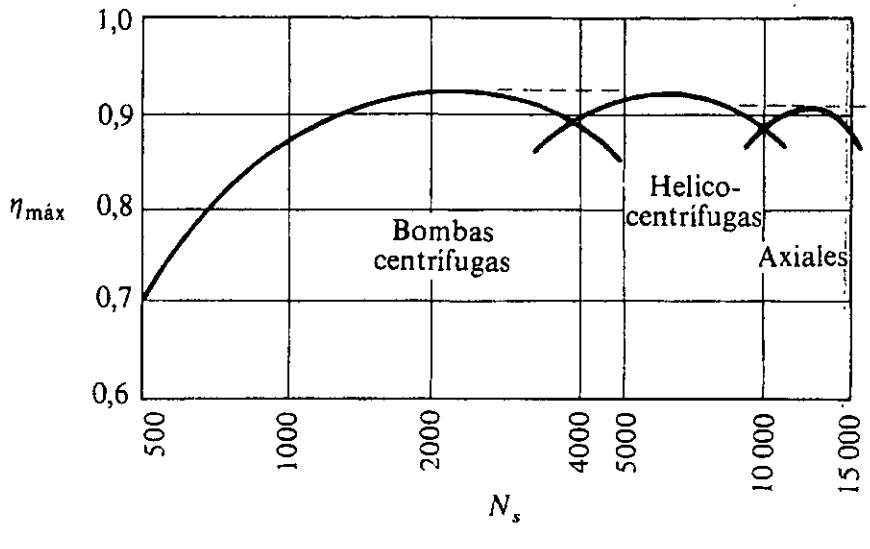
$$N_s=\frac{N\sqrt{Q}}{(gH)^{3/4}}$$

Forma común:

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$

$$N_s = \frac{[rev/\min][gal/\min]^{1/2}}{[ft]^{3/4}}$$

La velocidad específica es útil para la selección del equipo más eficiente.



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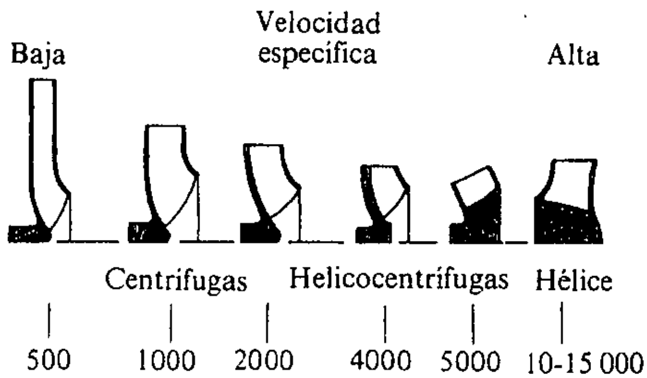

figura 3. Selección del tipo de bomba dinámica, en función de su velocidad específica.

VELOCIDAD ESPECÍFICA DE SUCCIÓN.

Si usamos la $CSPN$ en lugar de H en la expresión de la velocidad específica, hallaremos la velocidad específica de succión. Donde el valor necesario es el de la $CSPN_d$, es decir la carga de succión positiva neta disponible en el sistema. Se sabe que la cavitación puede existir cuando $N_{ss} \geq 8100$. En ausencia de datos, puede utilizarse este parámetro para saber si existe riesgo de cavitación en la instalación.

CAVITACIÓN.

Existen tres requerimientos fundamentales para que ocurra cavitación. La primera, la existencia de pequeñas partículas gaseosas que generen la posibilidad de que el líquido pueda volatilizarse y formar cavidades llenas de vapor; La segunda, una caída de presión aún breve en el tiempo en el seno del fluido que provoque vaporización del fluido; Tercera, la presión ambiental del fluido alrededor de la cavidad debe ser mayor que la presión de vapor del líquido, provocando el colapso súbito de las burbujas.

La cavitación no es un fenómeno exclusivo para los impulsores de las bombas, ocurre también de manera común en la cara posterior de válvulas cuyo diferencial de presión aguas arriba aguas abajo, es demasiado grande. Puede llegar a presentarse en codos y derivaciones de tuberías donde la velocidad del flujo es excesivamente grande.

La cavitación es un serio problema de operación en bombas y válvulas; Es un gran problema financiero ya que implica importantes costos en las reparaciones; Muchas bombas operan en condiciones de cavitación; Un propósito fundamental de la correcta selección de bombas y válvulas, así como su correcta instalación es reducir al mínimo los efectos de la cavitación.

Un factor fundamental que contribuye a los daños erosivos de la cavitación, es la tolerancia de las normas de utilizar equipos de bombeo solo ligeramente arriba se su valor de $CSPN_r$

La cavitación, tal como ocurre en impulsores, válvulas o placas de orificio, genera ruido, fluctuaciones en la presión, vibraciones y eventualmente, llega a reducir la eficiencia del dispositivo. La relación entre $CSPN_r$, erosión y generación de ruido debidos a la cavitación, ha sido investigado por Deeprose, King, McNulty, Pearshall y Grist. La referencia de Deeprose muestra que el máximo ruido y erosión en bombas ocurre desde antes que la eficiencia de la máquina comienza a decrecer. La referencia de Grist sugiere que, en promedio, el valor de la $CSPN_d$ debe ser alrededor de tres veces el valor de la $CSPN_r$ del fabricante, para evitar daños por cavitación, esta última recomendación es también sustentada en el trabajo de Deeprose, y se menciona que el Instituto de Ingeniería de la UNAM sugiere considerar $CSPN_d > 2.5CSPN_R$. Los mayores daños por cavitación, ruido y vibraciones ocurren antes de que los dispositivos muestren detrimento en sus funciones hidráulicas.

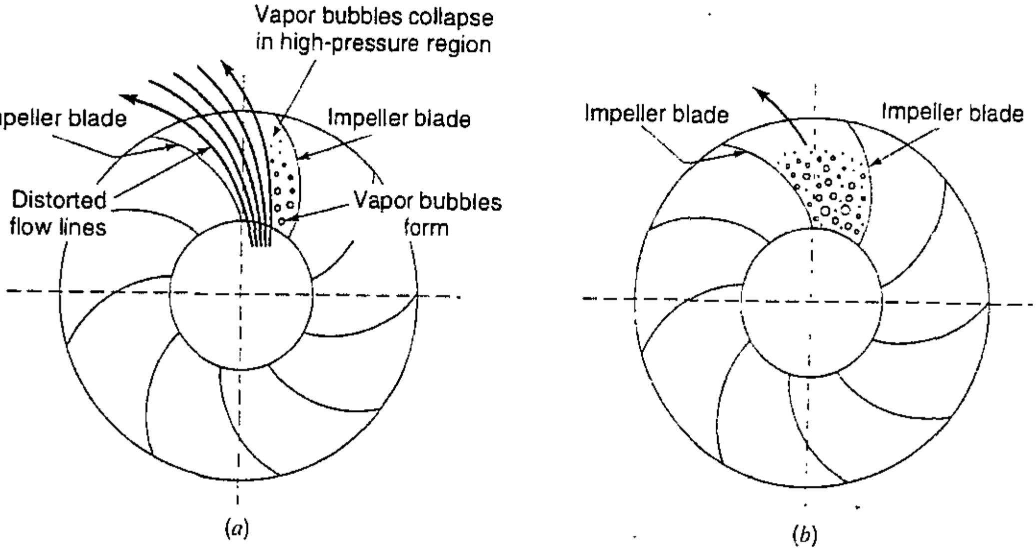

figura 4. Ilustración del problema de cavitación en los impulsores de una bomba centrífuga.

III.- VÁLVULAS.

Las válvulas son importantes en los sistemas de bombeo y líneas de conducción. Se utilizan para regular presión y caudal, protegen a la tubería por sobrepresiones estacionarias o derivadas de efectos transitorios, igualmente protegen a la bomba y sirven para disminuir los efectos de los fenómenos transitorios, evitan el retorno de caudal en la línea de conducción remueven el aire atrapado en el conducto y desarrollas otras varias funciones. Si las válvulas no se seleccionan de manera adecuada pueden convertirse en causa de problemas por ejemplo: Cierre súbito de una válvula, el uso de una válvula de no retorno inadecuada o el llenado demasiado rápido de una línea de conducción. Si una válvula está sujeta a cavitación, sufrirá un rápido desgaste, tendrá fallas en los sellos y requerirá reemplazo.

Por sus funciones en la línea de conducción, las válvulas se clasifican en las siguientes categorías:

1) Control
2) Reguladoras de presión
3) No retorno
4) Control de aire

Esta clasificación no es estricta, debido a que una válvula puede llevar a cabo más de una de las funciones mencionadas, sin embargo, debe verificarse que su eficiencia y posibilidades de cavitación no obren en menoscabo del buen funcionamiento de la instalación. Una buena referencia para la correcta selección de válvulas es el libro de Tullis.

1) Las válvulas de control se utilizan principalmente para permitir o impedir el flujo por alguna línea, en el caso de una red de tuberías pueden utilizarse para aislar una parte de la red que deba sujetarse a trabajos de mantenimiento o reparación; Sirven también, como su nombre lo indica, para controlar el caudal que transita en un conducto. En este tipo de válvulas se cuenta con las válvulas de compuerta, que se recomiendan para ser usadas completamente abiertas o cerradas, ya que sus características como disipadoras de energía no son muy buenas, para un servicio semejante pero con mayor economía, se encuentran la válvulas de mariposa que, con las debidas precauciones pueden utilizarse para regular caudal. Otro tipo de válvulas de este tipo cuyo costo es mayor a las dos ya mencionadas son las de bola o esfera.
2) Para regular presión, se recomienda el uso de válvulas tipo globo. Este tipo de válvulas puede usarse para flujo en ambos sentidos. Tienen un patrón de flujo bastante complicado, por lo que no se recomienda su uso para trabajar completamente abiertas. Su desempeño ante los efectos de cavitación puede mejorarse bastante haciendo correr el sello en el interior de una camisa perforada. Los orificios disipan la energía y reducen la

cavitación; Utilizando múltiples camisas concéntricas logrando disipar la energía en varias etapas, la cavitación puede llevarse al mínimo, debe tenerse en cuenta que la camisa perforada afecta bastante el desempeño de la válvula.

3) Las válvulas de no retorno se utilizan principalmente para evitar el retorno de caudal hacia los equipos de bombeo, lo cual podría llegar a generar severos daños en las componentes mecánicas y en los motores y sistema eléctrico. Para muchos estilos, la orientación de la instalación es importante, dado que el peso actúa en el cierre de la válvula. Estas válvulas deben instalarse en tramos rectos de tubería, ya que la influencia en el flujo de codos y otras piezas especiales, puede generar oscilaciones en la clapeta y reducir su capacidad de cierre en el mediano o largo plazos. Existen una gran cantidad de diseños de válvulas de no retorno en el mercado, de modo que el proyectista tiene la responsabilidad de buscar la más adecuada al caso particular. Entre las menos recomendables se cuentan las "check tipo bola" ya que disipan demasiada energía y su condición de cierre puede ser muy súbita; La válvula tipo columpio es de las más comunes, sin embargo, la clapeta tiene una masa muy grande por lo que la repetición de ciclos de cierre puede dañarla; Además puede presentar condiciones de cierre perniciosos para el caso en que se busque minimizar los efectos de los transitorios de presión. Para equipos de bombeo donde se han instalado cámaras de aire cerca de la bomba, ocurren retornos de caudal hacia la bomba. Válvulas de no retorno convencionales (ej. Tipo columpio), no se recomiendan. Para instalaciones importantes, es necesario contar con un análisis detallado de los efectos transitorios en el conducto, lamentablemente, no se cuenta con información experimental abundante respecto a las condiciones de cierre de los diferentes tipos de válvulas de no retorno.
4) Control de aire. Para el correcto funcionamiento de una línea de conducción, debe permitirse la salida lenta del aire atrapado, ya que de lo contrario, pueden presentarse problemas graves. En el caso contrario, cuando el conducto está siendo vaciado, debe permitirse la entrada de aire. Generalmente estas funciones son llevadas a cabo de manera automática por válvulas de admisión, y válvulas de expulsión de aire.

IV.- DISEÑO GEOMÉTRICO Y FUNCIONAL.

A menos que se conozca a detalle una geometría funcional con la que puedan manejarse criterios de semejanza para una nueva propuesta geométrica de una estructura para instalar la succión de los equipos de bombeo, se hace necesaria la construcción de un modelo físico a escala. Aquellas cubetas para manejar caudales mayores a 3.0 m^3/s o con restricciones geométricas severas en la conducción de llegada, SIEMPRE deberán probarse en un modelo físico a escala. Lo anterior debido a que las recomendaciones de diseño publicadas, por ejemplo, por el BHRA, el HIS, o Dicmas, están referidas al diseño geométrico de estructuras pequeñas o medianas.

La prerotación en la succión de los equipos de bombeo, es la principal responsable de los problemas de vorticidad y entrada de aire, que consecuentemente reduce la capacidad neta de bombeo de líquido y genera vibración y ruido en las bombas. El más leve de los inconvenientes producidos por la prerotación, es la reducción de eficiencia de la máquina hidráulica derivada de la desviación del ángulo de ataque de los álabes debidos al patrón de flujo condicionado.

La velocidad promedio en la campana de succión no debe superar 1.1 m/s para evitar problemas de cavitación y vibración excesiva, del mismo modo, la velocidad promedio en el mismo conducto debe ser siempre menor que 1.5 m/s.

La prerotación y vorticidad están asociadas generalmente a la falta de sumergencia de la succión. Siempre debe procurarse que la voluta de la bomba permanezca por debajo del nivel mínimo estático de agua en la cámara de succión y, de manera más conservadora, aún los sellos mecánicos de la máquina deber permanecer bajo el nivel de agua en el cárcamo. La sumergencia no debe ser nunca menor a 1.5 veces el diámetro de la campana de succión. La acumulación de arenas en el fondo de la cámara de succión puede incrementar la velocidad del líquido al ingresar al conducto de succión, con lo que la sumergencia requerida tendería a incrementarse. Esquemas con 1.2-1.5 m/s en el conducto de succión deben respetar una sumergencia mínima de 0.9-1.1 m respectivamente.

A propósito de la sumergencia en el conducto de succión, debe tenerse en cuenta que la rugosidad de la conducción a presión puede sobreestimarse voluntariamente para prever las condiciones de operación del sistema de bombeo al cabo de una o dos decenas de años de operación, principalmente en zonas costeras donde pueden alojarse organismos adherentes o el contenido de sales disueltas posibiliten la incrustación de las tuberías metálicas; Sobreestimar el coeficiente de disipación en la tubería de presión en el diseño implica que en los primeros años de operación del sistema, la máquina trabajará con un caudal superior al de diseño, generando en primer lugar un decremento en la eficiencia operativa de la bomba, y respecto a la geometría de la cámara de succión, una tendencia a abatir el nivel estático en la succión con los problemas consecuentes sobre la sumergencia de la campana de succión, prerotación, vorticidad, etc.

En el manual de diseño de la Comisión Nacional del Agua se describen las partes constitutivas de un cárcamo de bombeo:

Canal o Tubo de Llegada: Cuando el agua se capta de una fuente superficial como el mar, un lago o una corriente superficial y el caudal de bombeo es considerable, es conveniente diseñar un canal de llamada que conduzca al líquido hasta el cárcamo. La llegada puede hacerse con un canal o una tubería.

Transición de Llegada: Ya en la proximidad del cárcamo, el canal de llegada deberá adecuarse gradualmente en forma y dimensiones a la sección de entrada al cárcamo. En ocasiones, esta transición no existe.

Zona de Control y Cribado: Normalmente se necesita disponer de dispositivos de control, como compuertas, para aislar al cárcamo de la fuente en casos de limpieza, reparación o mantenimiento además, debe cribarse el agua para evitar el paso de sólidos flotantes, malezas animales acuáticos.

Pantalla: En ocasiones, se coloca un muro vertical (pantalla o mampara) en la entrada de la zona de control y cribado, de manera que penetre hasta una profundidad inferior al nivel del espejo del agua. Su función principal es retener los objetos flotantes y los aceites que pudieran llegar hasta allí.

Rejillas Primarias: Estos elementos fijos constituyen la primera línea de protección contra la entrada de sólidos acarreados por el agua. Consisten en una rejilla gruesa ya que su función es retener objetos de tamaño grande.

Desarenador y Bomba de Lodos: El desarenador o trampa de arena es una cavidad en el fondo, al final de la zona de transición o al inicio de la zona de desbaste, que detiene a los sedimentos arrastrados para evitar sullegada a las bombas. La bomba de lodos extrae del desarenador los sólidos atrapados para su transporte a lugares de tiro convenientes.

Rejillas Secundarias: Estos elementos, con aberturas menores que las rejillas primarias retienen sólidos de tamaño pequeño (del orden de 1 cm). Debe asegurarse que el claro entre barras sea menor que el paso de esfera de las bombas. Normalmente cuentan con limpiadores mecánicos accionados de manera automática,

Cámara de Bombeo: En la zona inmediata a la bomba, se encauza el flujo hacia ella. En esta zona deben cuidarse los aspectos asentados en las recomendaciones de diseño. La mayor parte de los vórtices, tanto superficiales como sumergidos, se presentan allí. La cámara puede ser aislada por medio de una compuerta.

Lineamientos generales de las obras auxiliares:

En virtud de que un cárcamo tiene dos partes auxiliares, es necesario indicar los lineamientos generales de estas obras.

a) Velocidad de llegada. La Asociación Británica de Investigaciones Hidromecánicas recomienda que la velocidad en el canal o tubo de llegada sea como máximo 1.2 m/s. Esta velocidad es a la entrada del cárcamo, NO a la llegada de la campana de succión, la cual debe ser como máximo 0.30 m/s.
b) Transición. En el caso de que la llegada sea con un tubo de diámetro menor (que produzca una velocidad mayor a 1.2 m/s), debe hacerse una transición para reducir la velocidad. La transición no debe tener ángulos mayores de 20° de acuerdo con la Asociación Británica de Investigaciones Hidromecánicas.

La longitud de transición puede obtenerse utilizando la siguiente expresión:

$$L_T = \frac{\left(\frac{V_1}{V_2}b_1 - \frac{V_2}{V_1}b_2\right)}{2\tan\theta}$$

L_T : Longitud total de la zona de transición.
V_1 y V_2: Velocidad máxima a la entrada y a la salida de la zona de transición (m/s)
b_1 y b_2: Ancho a la entrada y salida de la zona de transición.
θ : Ángulo máximo.

c) Rejillas: Se pueden colocar rejillas en dos zonas: antes de que entre el agua al canal de llamada, denominadas rejillas primarias o troncos, cuya característica es que la apertura entre barras es grande y sirven para detener objetos grandes arrastrados por el flujo como troncos, llantas o cadáveres de animales. Otras rejillas pueden colocarse dentro del cárcamo para detener objetos pequeños, pero de menor diámetro que el paso de esfera de la bomba. Estas rejillas se denominan secundarias o finas.

VOLUMEN MÍNIMO DE UN CÁRCAMO.

En el caso de aguas negras, la retención de éstas en un cárcamo no se recomienda que sea mayor a 30 minutos. Por otro lado es conveniente diseñar un cárcamo cuyo volumen sea el mínimo posible, pero compatible con una buena operación.

Ello plantea la necesidad de establecer una relación conveniente volumen, caudal y tiempo de retención, así como las características del equipo de bombeo y el programa de operación (tiempos de arranque y paro) de dicho equipo.

El volumen de un cárcamo para una bomba, entre las elevaciones correspondientes al arranque y la parada, se puede calcular con la siguiente expresión:

$$V = \frac{QT}{4}$$

donde:

V: Es el volumen del cárcamo en m^3
Q: Gasto de la bomba en m^3/s
T: la duración mínima de un ciclo de bombeo en segundos.

La duración mínima de un ciclo de bombeo se presenta cuando el caudal de entrada es exactamente igual a la mitad de la capacidad de la bomba.; En estas condiciones, el tiempo que permanece encendida la bomba es igual al tiempo que permanece apagada. Si el caudal es mayor, la bomba permanecerá encendida por más tiempo y viceversa; en ambos casos, la duración del ciclo es mayor que el mínimo.

Para bombas y motores grandes, T no debe ser menor que 20 minutos. Para bombas menores, T puede ser reducida a 10 minutos, aunque lo recomendable es 15 minutos. Si esto conduce a un volumen excesivo de una estación de bombeo pequeña que tiene dos bombas idénticas, una de las cuales es de reserva, se puede reducir a la mitad del volumen del cárcamo, operando las bombas en forma alternada, ya que de este modo se logra que el valor de T asociado al cárcamo, sea igual a la mitad del valor efectivo de T asociado al equipo.

DIMENSIONAMIENTO DEL CÁRCAMO DE BOMBEO

Los parámetros geométricos fundamentales para el diseño exitoso de cámaras de succión para manejar caudales de hasta 1.3 m^3/s, e incluso mayores, se presentan a continuación:

Para una campana de succión contenida en un plano paralelo al fondo del cárcamo, la literatura recomienda respetar una distancia entre D/4 y D/2. De manera más precisa puede recomendarse una distancia del orden de 0.33D a 0.4D [ver Dicmas] para diámetros de succión $\phi_s \leq 355.6mm$ (14"). Conductos de succión de diámetro superior a 508 mm (20"), no se recomiendan debido al costo de la tubería. Para instalaciones mayores debe considerarse la succión de los equipos como una parte de la estructura de concreto, utilizando solamente tubería metálica de espesor mínimo a modo de molde. En este caso se recomienda el uso de placas supresoras de vórtices.

Velocidad de entrada en la campana de succión menor a 1.1 m/s.

Velocidad en el conducto de succión menor a 1.5 m/s.

Velocidades horizontales en la zona de aproximación a la campana de succión menores a 0.3 m/s.

La sumergencia de la succión debe afectarse de un factor de seguridad generoso debido a que la formación de vórtices depende de un gran número de factores.

La geometría de la cámara de succión no debe inducir rotación del fluido.

ARREGLO DEL CUARTO DE MÁQUINAS.

-Utilice los muros para soportar las pesadas válvulas y las tuberías.

-Evite arreglos de tuberías que impidan las maniobras de mantenimiento y el acceso del personal hacia las motobombas, controles, alimentadosres, etc.

-El arrancador, e instrumentos intrínsecos para la operación de una motobomba, deben instalarse de manera adyacente, de manera evidente, para facilitar las operaciones de mantenimiento.

-Nunca conecte la descarga de un equipo de bombeo a un múltiple, sin instalar una válvula de no retorno.

-Cuando se está manejando agua con sólidos o aguas residuales, nunca instale una válvula de no retorno en un conducto vertical, ya que puede atascarse la válvula, convirtiéndose rápidamente en un tapón.

-Garantice una distancia respetable de las juntas bridadas con cualquier muro, digamos de 0.6 a 1.2 m, para facilitar las maniobras de montaje y desmontaje de tuberías.

-Debe garantizarse el tránsito del personal entre los equipos de bombeo, sin obstrucciones u objetos que representen un riesgo de trabajo.

-No debe olvidarse la ventilación. Para garantizar la seguridad del personal, deben hacerse 12 cambios por hora del 100% del aire contenido en los recintos propios de la instalación de bombeo, e incluso 30 cambios por hora cuando se están manejando motores de combustión interna para impulsar los equipos de bombeo o existen gases combustibles o tóxicos, tales como el sulfuro de hidrógeno. Deben instalarse detectores a bajo nivel para el sulfuro de hidrógeno mientras su funcionamiento debe revisarse cada semana. La condición de alarma para el detector de gases combustibles ocurre cuando la concentración del gas, excede el 20 % del límite inferior explosivo (L.E.L. por sus siglas en inglés), o a 10 ppm de sulfuro de hidrógeno. Los detectores de gases combustibles deben revisarse y calibrarse cada 6 meses.

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CURSOS ABIERTOS

DISEÑO DE CARCAMOS DE BOMBEO

CA 387

TEMA
ANEXO I

EXPOSITORES: M. EN I. FRANCISCO ORDUÑA BUSTAMANTE
DEL 09 AL 30 DE JUNIO DE 2007
PALACIO DE MINERÍA

Net Positive Suction Head (NPSH)

$$NPSH = \left[\frac{N}{N_1}\right]^2 \times NPSH_1$$

where

$NPSH_1$ = Net positive suction head at test speed in feet

$NPSH$ = Net positive suction head at rated speed in feet

Note Refer to page 77, "NPSH-Experimental Deviation from the Square Law" for discussion of other factors which may affect this relationship

PLOTTING RESULTS

The total head, efficiency, and brake horsepower are usually plotted as ordinates on the same sheet with capacity as the abscissa as shown on Fig 63.

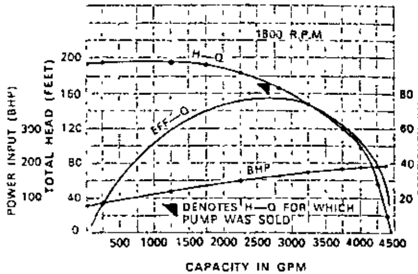


Fig 63 PLOTTED TEST RESULTS

Note For effect of compressibility of the fluid on the calculation of efficiency see page 71

SUMMARY OF NECESSARY DATA ON PUMPS TO BE TESTED

The following information should be furnished on pumps to be tested:

General:

1. Owner's name ________
2. Plant location ________
3. Elevation above sea level ________
4. Type of service ________

Pump.

1. Manufactured by ________
2. Manufacturer's designation ________
3. Manufacturer's serial number ________
4. Arrangement horizontal ____ vertical ____
5. Inlet ____ single ____ double ____
6. Number of stages ________
7. Size suction. nominal ________ in
   actual ________ in.
8. Size discharge nominal ________ in.
   actual ________ in

Intermediate Transmission:

1. Manufactured by ________
2. Type ________
3. Serial number ________
4. Speed ratio ________
5. Efficiency ________

Driver:

1. Manufactured by ________
2. Serial number ________
3. Type. motor ____ turbine ____ other ____
4. Rated horsepower ________
5. Rated speed ________
6. Characteristics (voltage, frequency, etc) ____
7. Calibration data ________

Specifying Rated Conditions

The following information is necessary in specifying rated conditions.

1. Liquid pumped (water, oil, etc) ________
2. Specific weight ________
3. Viscosity at pumping temperature ________
4. Temperature ________ F
5. Vapor pressure ________ psia
6. Capacity ________ gpm
7. Total suction lift (h_s) ________ ft
   head (h_s) ________ ft
8. Net positive suction head (NPSH) ________ ft
9. Total discharge head (h_d) ________ ft
10. Total head (H) ________ ft
11. Liquid horsepower (whp) ________ hp
12. Efficiency (η_p) ________ per cent
13. Brake horsepower (bhp) ________ hp
14. Speed ________ rpm

TEST INFORMATION

Test information should be listed substantially as follows

General:

1. Where tested ________
2. Date ________
3. Tested by ________
4. Test witnessed by ________

Capacity.

1. Method of measurement ________
2. Meter—Make and serial number ________
3. Calibration data ________

Head:

1. Suction gauge—Make and serial number ____
2. Calibration curve ________
3. Discharge gauge—Make and serial number ________
4. Calibration data ________

Power:

1. Method of measurement ________
2. Make and serial number of instrument ____
3. Calibration data ________

Speed:

1. Method of measurement ________
2. Make and serial number of instrument ____
3. Calibration data ________

Test Performance Corrected to Specified Conditions

Test performance of pumps handling non-viscous liquids shall be corrected to rated speed and rated specific weight. For correction values applying to centrifugal pumps handling viscous liquid, refer to page 111.

Model Tests

In many installations involving units of large size, model tests are of great utility. Even when it might be feasible to test the large unit in the factory, a model may often be tested with greater accuracy and thoroughness. By adopting a standard size of model for various pumps, properly comparable performances can be obtained. The model impeller should be not less than 12 inches outside diameter. The exact model to prototype ratio shall be selected by the pump builder. Comparisons between model tests are valid only when the model to prototype ratios are substantially the same.

Testing models in advance of final design and installation of a large unit not only provides advance assurance of performance but makes alterations possible in time for incorporation in the prototype unit.

Not all installations lend themselves to a practical model investigation. The pumping of water carrying considerable quantities of sand or other foreign material is not readily reproduced in model operation. This standard, therefore, is limited to the pumping of clear water, free from abnormal quantities of air or solids, both in field installations and factory tests. The effects of wear and deterioration, the effects of free surface disturbances in open channel sumps, interference between neighboring units, and peculiar problems caused by abnormal settings are covered by model sump tests (See page 129).

It is recommended that when model tests are to be conducted, the performance characteristics be specified for the model. It is not, in general, essential that the model test head be the same as that of the prototype. A model pump should be tested at such conditions that complete turbulent flow will be maintained in all flow passages at all times. In general, this means that the model head will be the same as that of the prototype.

The model should have complete geometric similarity with the prototype, not only in the pump proper, but also in the intake and discharge conduits as specified above for tests on full sized pumps. The model should be run at such speed that the specific speed remains the same as that of the prototype unit. If cavitation tests are not available, the suction head or lift should be such as to give the same sigma value as in the prototype. As previously explained, if the prototype sigma is known to be safely in excess of the critical sigma, then a higher sigma can be used for the model tests, although it is preferable to maintain the same value.

There is danger of air separation destroying similarity relationships if the absolute pressure is reduced too low. Consequently, condensate pumps should not be modeled.

If corresponding diameters of model and prototype are (D_1) and (D) respectively, then the model speed (N_1) and model capacity (Q_1), under the test head (H_1), must agree with the relationships

$$\frac{N_1}{N} = \left[\frac{D}{D_1}\right]\sqrt{\frac{H_1}{H}}$$

and

$$\frac{Q_1}{Q} = \left[\frac{D_1}{D}\right]^2\sqrt{\frac{H_1}{H}}$$

If a model vertical wet pit pump is tested in its corresponding model intake structure, it should be remembered that the conditions to satisfy the pump model relationship and the Froude sump model relationship (see page 129) cannot exist simultaneously. The velocities derived by the Froude law will be considerably less. The model system should be designed so that performance can be observed and measured through the entire range of velocities.

The efficiency of the model should not, in general, be assumed to be exactly equal to that of the prototype. In testing a model of reduced-size, the above conditions being observed, complete hydraulic similarity may not be attained because of certain influences. For example, complete geometric similarity will not be obtained unless the relative roughness of the impeller and pump casing surfaces are the same. With the same surface texture in both model and prototype, the model efficiency will be lower than that of the larger unit. Further, it is generally not practical to model running clearances or bearing sizes. When such is the case, the model efficiency will be reduced. To approximate prototype efficiency with a model, the impeller and diffuser surfaces must be considerably smoother on the model than on the prototype.

When a high degree of understanding exists between manufacturer and user relative to the comparison limitations encountered going from model to prototype, thought may be given to the practicability of increasing the prototype efficiency on the basis of model test results. However, this should be done only by mutual agreement before the job is let, on the basis of all the available test data of a similar nature.

Numerous comparisons of prototype and model efficiencies, with consistent surface finish of models and prototypes, are necessary for a given factory to establish a basis for stepping up model performance to field performance. This stepping up can be applied conveniently according to the formula in use for turbines, namely

$$\frac{1-\eta_1}{1-\eta} = \left[\frac{D}{D_1}\right]^n$$

The exponent (n) is to be determined from actual data as described above.

The values for the exponent (n) have been found to vary between zero and 0 26, depending on relative surface roughness of model and prototype and other factors.

For possible adjustment of a cold water test to hot water conditions see page 77.

Typical Example of Model Test

A single stage pump designed to deliver 90,000 gpm against a head of 400 feet at 450 rpm and with a total suction head of 10 feet has an impeller diameter of 6 8 feet. This pump is too large for a factory test and, in place of such test on the actual pump, a model is to be tested at a reduced head of 320 feet. The model impeller is to be 18 inches, or 1.5 feet in diameter.

EXAMPLE: Determine speed, capacity and suction head for the above model test.

Applying the above relationships:

$$\frac{N_1}{N} = \left[\frac{D}{D_1}\right]\sqrt{\frac{H_1}{H}}$$

or

$$N_1 = N\left[\frac{D}{D_1}\right]\sqrt{\frac{H_1}{H}} = 450\left[\frac{6.8}{1.5}\right]\sqrt{\frac{320}{400}} = 1825 \text{ rpm}$$

$$\frac{Q_1}{Q} = \left[\frac{D_1}{D}\right]^2\sqrt{\frac{H_1}{H}}$$

or

$$Q_1 = Q\left[\frac{D_1}{D}\right]^2\sqrt{\frac{H_1}{H}} = 90{,}000\left[\frac{1.5}{6.8}\right]^2\sqrt{\frac{320}{400}} = 3920 \text{ gpm}$$

The model pump should therefore be run at a speed of 1825 rpm delivering 3920 gpm against a head of 320 feet.

To check these results it will be noted that the specific speed of the prototype is

$$N_s = N\frac{\sqrt{Q}}{H^{3/4}} = 450\,\frac{\sqrt{90{,}000}}{(400)^{3/4}} = 1510$$

and the specific speed of the model will be

$$N_s = 1825\frac{\sqrt{3920}}{(320)^{3/4}} = 1510$$

Therefore, the specific speeds are the same as required.

The cavitation factor sigma for the field installation, which should be the same as in the test, assuming the usual water temperature of 80 F as a maximum probable value, will be

$$\sigma = \frac{NPSHA}{H}$$

where

$$NPSHA = \frac{144}{w}(p_a - p_{vp}) + h_s = \frac{144}{62.3}(14.7 - .5) + 10 = 32.8 + 10 = 42.8 \text{ feet}$$

therefore

$$\sigma = \frac{42.8}{400} = 0.107$$

With the water temperature of model and prototype approximately the same

$$\sigma = \frac{NPSHA_1}{H_1}$$

Thus

$$NPSHA_1 = \sigma H_1 = 0.107 \times 320 = 34.25 \text{ feet}$$

and

$$h_s = 34.25 - 32\ 80$$
$$= 1\ 45 \text{ feet}$$

The model should therefore be tested with a total suction head of 1 45 feet to reproduce the field condition

Test of Models at Increased Head

Under special and unusual circumstances, it may be desirable to carry out factory tests at higher heads than the prototype head This may be due, for example, to the limitations of available test motors or electrical frequency In this case, all of the above considerations continue to apply.

It should be pointed out, however, that with a reduced size model, coupled with an increase in head, the increase in speed corresponding to the head increase tends to minimize the change in Reynolds number, that is, the product of flow velocity and linear dimensions of the model tends to approach equality with the same product in the prototype This effect tends to restore dynamic similarity in model and prototype and to approach equality of efficiencies and other performance factors. With increased head, however, the preservation of the same sigma value in the model as in the prototype must still be observed, and this factor will assume increased importance, requiring an increase in submergence or reduction in suction lift in the factory test

The last mentioned requirement may result in another reason for the use of an increased head in the factory test Cases may arise in which the limitations of the factory test set-up may preclude the obtaining of sufficient suction lift to reproduce the prototype sigma In such cases, the required sigma can be obtained by an increase in the pumping head instead of by a reduction in suction head or increase in suction lift

Testing of Self-Priming Pumps

In addition to the standard performance tests, as outlined in preceding paragraphs, it is recommended that self-priming pumps be tested to determine the priming time For this test, the suction line shall be substantially the same as that shown in Fig 64 Static lift between the eye of the impeller and the liquid level shall not be less than ten feet. No check or foot valve shall be installed in the suction piping

In making this test, proceed as follows:

Fill priming chamber with liquid, or turn on the sealing water line to the vacuum pump if a wet-type vacuum pump is used

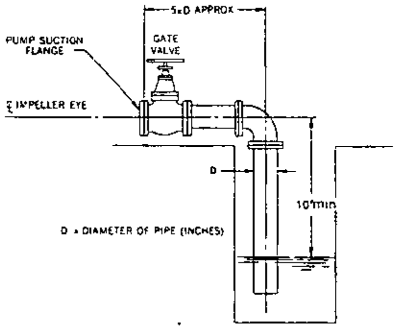


Fig 64 SUCTION LINE FOR STATIC LIFT TEST

Start the unit The priming time then shall be the total elapsed time between starting the unit and the time required to obtain a steady discharge gauge reading, or full flow through the discharge nozzle. During this phase of the test, the discharge pipe must be vented if the priming system is of the recirculating type This will prevent a back pressure being developed as the result of the accumulation of gas. If the unit is equipped with a priming pump of the separate type, it will be necessary that the discharge pipe be sealed with a column of water that will prevent air being drawn from the discharge side of the unit

Determination of Maximum Developed Vacuum by Means of Dry Vacuum Test

The test procedure is

With the unit in operation and delivering full flow, close gate valve in the suction line.

The reading on the vacuum gauge will then be the maximum developed vacuum

Priming Time Conversion Factors

If a suction line is used which is larger in size than is normal for the pump being tested, it is necessary to compute the performance for the normal size of pipe Referring to Table 6, select the size of suction pipe actually used in the test. Follow this line horizontally to the right, to the vertical column under the heading representing the normal suction pipe size of the pump The figure shown at the point of intersection is the conversion factor Divide the test time (in seconds) by this factor and then divide the resultant by the total length of empty test suction line in feet The resultant will then be the average time in seconds for air removal from a suction line of normal size, per foot of length.

TABLE 6

Priming Time Conversion Table

Size of Std. Pipe Actually Used	3/8"	1/2"	3/4"	1"	1-1/4"	1-1/2"	2"	2-1/2"	3"
1/2	1 59	—	—	—	—	—	—	—	—
3/4	2 79	1.75	—	—	—	—	—	—	—
1	4.53	2 85	1.62	—	—	—	—	—	—
1-1/4	7 85	4 93	2.81	1 73	—	—	—	—	—
1-1/2	10 7	6.70	3 82	2 35	1 36	—	—	—	—
2	17 6	11.0	6 29	3.88	2.24	1.64	—	—	—
2-1/2	25 1	15.8	8 98	5.53	3 20	2 35	1.42	—	—
3	38.7	24 3	13 9	8 55	4 94	3 62	2 20	1 54	—
3-1/2	—	32.6	18.6	11 4	6 61	4 85	2.95	2 07	1 34
4	—	41.8	23.9	14 7	8 50	6 25	3.79	2.66	1.72
5	—	66.0	37 5	23 2	13 4	9 83	5 96	4 18	2 70
6	—	—	54 2	33.4	19 3	14 2	8.61	6 03	3.90
7	—	—	72 6	44 8	25 9	19.0	11 5	8 10	5.24
8	—	—	93 8	57.9	33 4	24 5	14.9	10 4	6.77
9	—	—	—	72 6	41 9	30.8	18 7	13 1	8 49
10	—	—	—	91.3	52.7	38 7	23 5	16 5	10 7
11	—	—	—	—	63 5	46 6	28 3	19.8	12.9
12	—	—	—	—	75 5	55 5	33 7	23 6	15 3

Size of Std. Pipe Actually Used	3-1/2"	4"	5"	6"	7"	8"	9"	10"	11"
1/2	—	—	—	—	—	—	—	—	—
3/4	—	—	—	—	—	—	—	—	—
1	—	—	—	—	—	—	—	—	—
1-1/4	—	—	—	—	—	—	—	—	—
1-1/2	—	—	—	—	—	—	—	—	—
2	—	—	—	—	—	—	—	—	—
2-1/2	—	—	—	—	—	—	—	—	—
3	—	—	—	—	—	—	—	—	—
3-1/2	—	—	—	—	—	—	—	—	—
4	1 29	—	—	—	—	—	—	—	—
5	2 02	1.57	—	—	—	—	—	—	—
6	2 92	2 27	1 44	—	—	—	—	—	—
7	3 92	3 04	1 94	1 34	—	—	—	—	—
8	5 06	3.93	2 50	1 73	1.29	—	—	—	—
9	6 34	4 93	3 14	2 17	1.62	1 25	—	—	—
10	7 98	6 20	3 94	2.73	2 04	1.58	1 26	—	—
11	9.62	7.46	4 75	3.29	2.45	1.90	1 51	1.20	—
12	11 4	8 88	5 65	3 91	2 92	2 26	1 80	1.43	1 19

Object

The purpose of this section is to provide a guide for the application of centrifugal pumps for various services No attempt has been made to cover all phases of centrifugal pump application, but an endeavor has been made to point out some of the principal features of pumps and the precautions which should be taken in their use

Minimum Flow Limitation

All Centrifugal pumps have limitations on the minimum flow at which they should be operated. The most common limitation is to avoid excessive temperature build up in the pump because of absorption of the input power into the pumped fluid Other reasons for restrictions are

- Increased radial reaction at low flows
- Increased NPSHR at low flows resulting in cavitation
- Noisy, rough operation and possible physical damage due to internal recirculation
- Increased suction and discharge pulsation levels
- Increased axial reaction affecting thrust bearing loading

The size of the pump, the energy absorbed, and the liquid pumped, are among considerations in determining these minimum limitations For example, most small pumps on ordinary applications with good suction conditions such as domestic home circulators, service water pumps and chemical pumps have no limitations except for temperature buildup considerations. For small pumps, when NPSH is critical, limitations of about 25% of the capacity at the best efficiency point should be imposed Many large, high horsepower pumps have limitations as high as 70% of the best efficiency point capacity The manufacturer should be consulted in all cases where doubt exists regarding the allowable minimum flow

Boiler Feed Pump Applications

The boiler feed pump usually takes suction directly from the condensate pump, deaerating heater or feed booster pump. There may be feed water heaters in the suction line. The boiler feed pump discharges through a feedwater regulating valve to the boiler

It is extremely important for satisfactory operation that sufficient net positive suction head (NPSH) be available at the pump suction flange As a guide for determining the necessary NPSH, the values from Figs 68 and 69 are recommended (Pages 107 and 108)

In addition to providing sufficient NPSH when operating at rated conditions, precautions should be taken in the design of the feed system to prevent fluctuations of the pump suction pressure resulting from sudden station load changes and consequent reduction in the feed water heater pressure on the suction side of the pump In cases where it is not possible to maintain the heater pressure under these conditions, cold water should be introduced into the system to prevent flashing, preferably at the suction of the pump, or steam into the heater to prevent too rapid a rate of pressure decay

Provision must be made for recirculating a portion of the pump capacity from the discharge line back to the top of the heater on the suction side of the pump This by-pass should be opened wide during starting and stopping and during periods of low pump capacity operation in order to prevent overheating of the pump

Materials used in the construction of boiler feed pumps must be carefully selected to withstand the corrosive and erosive action of the feedwater Unless experience has shown that cast iron, bronze or steel are satisfactory for a particular installation, corrosion-erosion resisting materials such as the chrome or chrome nickel alloy types should be used

For temperatures above 300 F, the packing boxes should be provided with cooling jackets Smothering type glands are usually recommended In the larger horsepower applications, labyrinth type seals employing cold condensate injection are frequently used

It is important that suction and discharge piping be properly supported to avoid undue pipe strains on the pump and that the unit be aligned at the operating temperature

Chemical Pump Applications

Pumps used for handling corrosive liquids or slurries are commonly termed chemical pumps The materials of construction for the parts in contact with the liquid, including stuffing boxes or seals, must be selected to offer maximum resistance to corrosion and abrasion at the pumping temperature, with due consideration to the economy of such use

Each application must be carefully scrutinized to determine the severity of corrosion or abrasion, the viscosities at the extreme pumping temperatures, the hazard involved in the material to be pumped, changes in the composition of the liquid, vapor pressure, NPSH, prolonged operation at or near shut-off, or any other pertinent characteristics of the liquid or the application

The physical and chemical properties of materials, the available forms, and methods of fabrication, must be considered in the design of satisfactory equipment Dissimilar materials in contact with the liquid pumped should be avoided unless the combination is one which is known to be satisfactory for the particular service.

Special seals or deep stuffing boxes, with provision for lubrication or sealing by clear cold water, are frequently necessary. Large unobstructed liquid passages are desirable. The unit should be designed for easy and quick disassembly for inspection, cleaning or repair Water jackets, steam jackets, or smothering type glands, may be mandatory The need for these features can be determined only after careful consideration of application requirements

The manufacturer's instructions with reference to installation should be strictly followed. In many cases, such installation may be radically different from those for clear water pumps.

Regularly established schedules for periodic examination and maintenance are essential

Condensate Pump Applications

Pumps handling condensed steam from a condenser, or other form of surface heat exchange equipment, are commonly termed condensate pumps.

Most condensate pumps have special design impellers with low inlet velocities This special design feature allows small values of h_{sv} resulting in lower elevations and shorter pump settings and excavations.

Suction piping should be of ample size to provide low velocities It should be short and as direct as possible, with a minimum of fittings and horizontal runs

To prevent the accumulation of vapor at the impeller inlet, the first stage impeller eye should be vented back to the vapor space in the condenser. Vent piping should have a continuous upward slope toward the condenser Discharge check valves should always be located below hot well levels to prevent overpressure failure on startup.

Stuffing boxes operating under vacuum should be provided with lantern rings connected to a dependable supply of seal water Provision should be made to prevent air leakage between the shaft sleeve and the shaft. Lantern rings are not required in stuffing boxes which are under pressure at all loads.

Condensate pumps handle unbuffered water, but due to the low temperatures involved, the use of bronze fitted pumps is usually satisfactory.

Heater drip and drain pumps usually handle condensed steam at high pressures and temperatures and, therefore, may be considered as condensate pumps Material selection is similar to boiler feed pumps for equivalent water conditions.

Slurry Handling Pump Application

Centrifugal slurry pumps may be used for both in-plant process and pipeline applications where heads are not high enough to warrant the use of reciprocating or rotary units

The other factors which affect the use of centrifugal slurry pumps are-

- Capacity
- Abrasiveness
- Particle Size and Shape
- Pressure

Centrifugal pumps are commonly applied for capacities from 10 GPM to 20,000 GPM with heads up to 350 feet per stage Some pumps may be installed in series for high head applications Solids content of the slurries handled varies from a trace to approximately 70 per cent by weight The percentage depends upon the size of the solid, its weight and the carrying medium

There are many different slurry pump designs available to accommodate various industrial applications. Those applications include the pumping of solids encountered in mineral ore treatment, dredging, sewage handling, land reclamation, paper manufacture, solids transportation and chemical processing The carrying medium is generally water, though other liquids, such as brine and oil, are sometimes used

The pumps are normally made of either hard metals (abrasion resistant cast irons and steels) or elastomers, and are designed to resist abrasion These pumps may have replaceable liners or covers.

In operation, because the abrasive nature of the slurry tends to open up the running clearance between the impeller and the suction cover or liner, means may be provided to adjust the clearance without dismantling the pump If the proper clearance is not maintained, excessive internal leakage will take place, causing considerable wear and altering the head-capacity characteristics of the unit. Quick disassembly features may be designed into

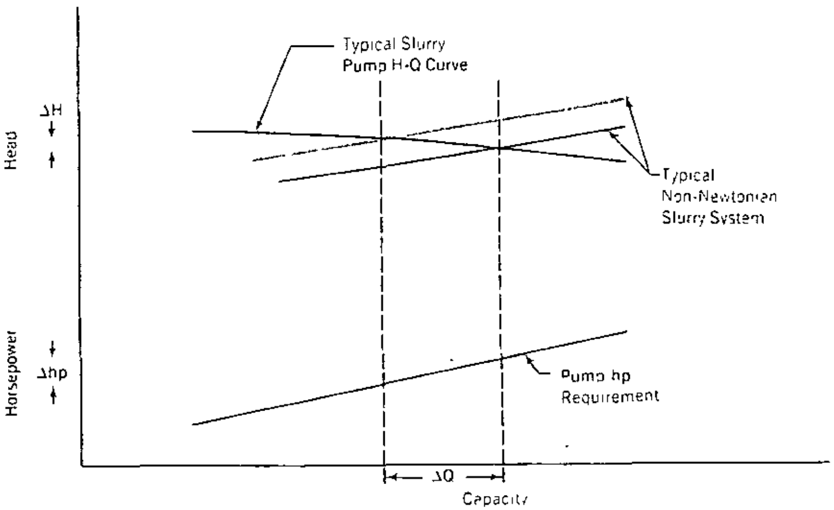


Fig 65 POWER VERSUS CAPACITY FOR A NON-NEWTONIAN SLURRY PUMP SYSTEM

the pump for ease of maintenance when it becomes necessary to replace worn parts

Hydraulic passages in centrifugal slurry pumps are usually larger than in a clear liquid pump for the same head and capacity. Such construction leads to higher radial loads requiring larger shafts and bearings It also tends to change the shape of the head-capacity curve, making it flatter and moving the peak efficiency to a capacity that is greater than considered normal for a water pump.

Clear liquids or greases are often used to seal the stuffing box of centrifugal slurry pumps. In addition to lubricating the shaft sleeve, the liquid must have sufficient pressure to keep solids from entering between the sleeve and the packing In order to reduce the required pressure, back vanes are frequently used on the back of the impeller hub shroud Some pumps are constructed with the inlet on the stuffing box side of the liquid end to reduce the seal water pressure requirement In some limited applications, mechanical seals are used in lieu of conventional packing Other pumps may have dynamic seals, which use an auxiliary set of vanes to prevent leakage through the box when the pump is operating During shutdown, leakage is prevented by a seal that is either mechanically or hydraulically operated

In applying centrifugal slurry pumps to handle settling slurries, one must be certain that the head requirements of the system at the critical carrying velocity would be met by the pump If the head delivered is insufficient, the capacity will be reduced and the solids will settle in the line, thus increasing the system head. Since the head-capacity curve of most slurry pumps has little slope, such an increase can make a large reduction in the volume pumped, further reducing the flow velocity and leading to plugging in the system pipe This situation can usually be avoided by using conservative values for the slurry critical carrying velocity.

The flatness of the centrifugal slurry pump head-capacity curve needs to be considered when applying the unit to a system handling a non-Newtonian slurry Such slurries generally exhibit a flat system curve Since the angle of intersection of these two curves is small, a modest change in the total head values selected (ΔH) would create a large deviation in the capacity being handled (ΔQ) See Fig 65 This could mean a corresponding increase in the horsepower requirements (Δhp) This situation is normally best handled by using a variable speed drive.

Large changes in power requirements could also

be created by changing of pipe sizes when handling non-Newtonian slurries Slurries of this type with modest to high yield values have a resistance to flow that is roughly inversely proportional to the pipe diameter for a given velocity. An increase in pipe size without a change in pump speed or impeller diameter could easily lead to an over-capacity condition. Throttling with a gate valve or similar device is only a temporary solution, since the high velocities at the point of restriction will cause wear and thus reduce the artificial head created It is advisable to use a smaller diameter pipe for a portion of the system rather than to attempt to increase the system head with a valve.

A possible exception to this is a diaphragm or pinch valve These devices do not wear as rapidly as metallic restrictions and could, on some occasions, offer a satisfactory solution.

Valves are generally used on the discharge side of a slurry pump. Most slurry pumps have suction lift capabilities, even though the majority of installations are made with the suction sump liquid level above the centerline of the pump Throttling on the suction side of a slurry pump can cause cavitation. The normal destructive effects of cavitation will be greatly increased in a slurry application The vapor bubbles that form would have fine particles as their nucleus, and implosion against the internal surfaces of the pump will be considerably more damaging than that experienced with a clear liquid

Fire Pump Applications

The National Fire Protection Association, NFPA, issues standards for the Installation of Centrifugal Fire Pumps. This standard is published in their pamphlet No. 20 which is revised periodically Always refer to the latest edition of this standard NFPA does not approve, inspect or certify any installation, procedures, equipment or materials.

The Factory Mutual System (FM) and Underwriters Laboratories, Inc. (UL) approve and list fire pumps which must be designed, manufactured and tested in accordance to their standards.

When applying fire pumps it is necessary to determine the governing standards from the authority having jurisdiction. The use of listed or approved fire pumps is usually mandatory.

Hot Oil Pump Applications

Pumps for handling oils within the range of 300 F to 850 F are commonly termed hot oil pumps

It is important that sufficient net positive suction head (NPSH) be available, as in practically all cases the liquid is near the boiling point

Provision should be made to allow self-venting of vapors from the impeller eye by venting the suction eye of the first stage except where the suction nozzle is in a vertical upward position.

The stuffing boxes and bearing housings should be provided with cooling jackets The glands should be of the smothering type If packing conditions require seal oil, lantern rings together with the necessary pipe connections should be provided During operation, the seal oil pressure in the lantern ring should be held to a minimum of 25 psi above stuffing box pressure.

The materials used for the construction of hot oil pumps should have a uniform coefficient of expansion, and should be selected with particular reference to the corrosive nature of the oil, as well as the actual pumping temperature

Due to the high pumping temperature, the support of the pump should be arranged in such manner as to permit expansion of the pump casing without adversely affecting the coupling alignment

It is important that the suction and discharge piping be supported to avoid pipe strains being imposed on pump nozzles The unit must be aligned at the operating temperature.

Hydraulic Pressure Pump Applications

These pumps are used for supplying water under pressure for scale removal from steel products, for operation of presses, leveling tables, hydraulic press service, elevators, etc.

The suction supply should be adequate to prevent parting of the liquid column during sudden demands for high capacity. If the suction line is long, a suction accumulator may be required.

The demand is frequently intermittent and the control valves are usually rapid in action. The sudden demand or cessation of demand causes accelerations and decelerations of water in the piping, resulting in pressure waves of great intensity. These waves are familiarly called "shock" or "water hammer." The waves originate at the point of valving and travel back through the line toward the pump

To protect the pump against damage from shock, an air-ballasted alleviator is recommended The alleviator should have a free liquid surface against which the shock can dissipate, with separate inlet and outlet connections so placed that shock waves cannot by-pass the alleviator. Alleviators mounted on side outlets of tees are of little value.

Mine Pump Applications

Pumps used for handling acid or gritty mine water and/or abrasive mixtures, slush, etc , are commonly termed mine pumps

The pump should be liberally designed, with heavy casing wall having ample corrosion allowance, and with provisions to keep corrosive liquids from pump shaft The design should provide for easy renewal of parts subject to corrosion or wear

The materials of construction for parts in contact with the liquid pumped must be selected for maximum resistance to corrosion and erosion

Non-Clog Pump Applications

Pumps designed to assure maximum freedom from clogging when handling liquids containing solids or stringy materials are commonly called non-clog pumps. They are also designated as sewage or trash pumps

Non-clog pumps are recommended for handling raw or unsettled sewage, activated sludge, industrial waste waters containing solids, and similar liquids where excessive clogging would otherwise be encountered The largest solid sizes that the pump will be required to handle in normal operation must be specified The term "sphere size" denotes the largest diameter ball which can be passed through the pump. Comminution and/or adequate bar screens must be provided to prevent larger solids from entering the pump. When used, bar screen openings should be sized to prevent clogging from irregular shaped solids

Storm water and/or domestic sewage may be handled successfully by mixed flow and axial flow pumps, using the preceding guidelines.

For domestic sewage service, pumps built to the individual manufacturer's material specifications are ordinarily used Corrosion and wear resistant shaft sleeves are desirable for maximum life. Inspection openings in the casing or adjacent piping, for access to the impeller, are recommended Stuffing boxes may be furnished with mechanical seals or packing, either water or grease lubricated When water is used for the stuffing box lubricant or flush, the supply line must be isolated from any potable water system

If the pumpage is corrosive and/or abrasive, the materials of construction for parts in contact with the liquid should be selected for resistance to the effects of the pumpage

Paper Stock Pump Applications

Pumps handling paper stock with consistencies over one per cent by weight are commonly termed stock pumps The pump should be designed to prevent clogging and air binding

When installing stock pumps, the suction line should be large and direct. Stock should flow freely to the impeller Friction head should be figured liberally

Allowances should be made for reduction in head, capacity and efficiency when handling heavy stock. The manufacturer should be consulted for correction values. It is important that drivers have ample power

The stuffing boxes should be provided with an outside seal connection to keep stock out of the packing

Self-Priming Pump Applications

Pumps designed to have the ability to prime themselves automatically, after being initially filled, when operating under a suction lift; to free themselves of entrained gas without losing their prime; and to continue normal pumping without attention, are commonly termed self-priming pumps Pumps in this class usually have single inlet impellers (Figs 66 and 67)

Self-priming pumps are used extensively by industry and elsewhere for the intermittent or continuous transfer of liquids where the self-priming feature is required The materials of construction vary with the requirements of the liquid or liquids pumped. Examples of such applications are tank car unloading, bulk oil station service, bilge service, etc. The majority of such pumps use electric motors as drivers and may be close coupled or separately connected, depending on the size and/or application involved

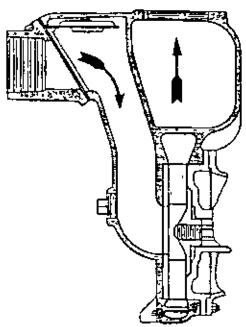


Fig. 66 SELF-PRIMING PUMP

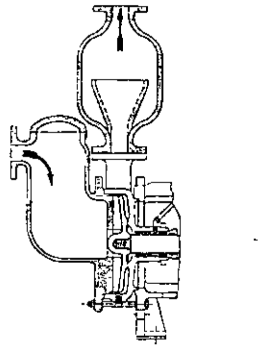


Fig 67 SELF-PRIMING PUMP

Contractors pumps are widely used for dewatering construction jobs and for other intermittent service where it is desired to have a pump which does not require manual priming prior to operation The materials of construction are usually cast iron, steel or bronze Examples of such applications are general construction work, irrigation, oil field gathering, etc The majority of such pumps use internal combustion engines for drivers and may be close coupled or separately connected, depending upon the physical size of the equipment. Some electrically driven or belt driven modifications are used Many manufacturers of this type pump build units to conform to the specifications established by the Contractors Pump Bureau of the Associated General Contractors of America, Inc In these cases, the pumps may carry an AGC rating plate and are built and rated to conform to AGC specifications

Sanitary Pump Applications

Pumps designed to handle foodstuffs and beverages are commonly called sanitary pumps. The materials of construction for parts in contact with the liquid pumped, including the stuffing box or shaft seal, are selected to prevent bacterial, chemical, color, or taste contamination Materials such as stainless steel, monel, porcelain, glass, etc , are frequently used. Sanitary pumps are constructed to permit ready access for cleaning, flushing, and draining If the liquid to be pumped contains solids, the maximum size solid must be specified.

The user is cautioned that many states prescribe regulations regarding sanitary pumps. Adherence to such regulations is mandatory. The responsibility for determining these requirements rests with the user

Volatile Liquid Pump Applications

Pumps for handling volatile petroleum products, or other liquids having similar properties, are commonly termed volatile liquid pumps.

The determination of the net positive suction head (NPSH) of pumps handling volatile, multicomponent liquids such as gasoline should be based, whenever possible, on the true vapor pressure of the particular liquid at the actual pumping temperature. The NPSH required by a pump at a given flow is a function of the individual pump proportions, and of the liquid pumped The available NPSH must exceed the NPSH required by the pump and can be established correctly only when the true vapor pressure is known.

For refinery process applications, the true vapor pressure is usually available. For the pumping of finished petroleum products, the Reid vapor pressure is usually the only information available This is the vapor pressure determined by the use of apparatus and procedure corresponding to the ASTM Standard D-323 Because of certain inadequacies of the test procedure, Reid vapor pressures are generally significantly lower than the true vapor pressures. Precautions must therefore be taken when the available NPSH must be determined on the basis of the Reid vapor pressure If the commercial grade of the liquid handled is known, the use of one of the standard handbook correction charts for conversion of Reid vapor pressure to true vapor pressure is indicated.

The suction piping should be arranged to avoid any accumulation of vapor and provision should be made to allow self-venting of vapors by venting the first stage impeller suction eye, except where the suction nozzle is in a vertical position and facing upwards.

Since the suction pressure may vary over a wide range and the liquid pumped is frequently flammable or toxic, the stuffing box may require the use of a mechanical seal, or, if packed, the use of one or more of the following elements· water jacketing, bleed-off connection, lantern ring for an oil or grease seal, or smothering type gland

The materials of construction should be selected with due regard to any corrosive action of the liquid pumped.

Suction Limitations

Among the more important factors affecting the operation of a centrifugal pump are the suction conditions. Abnormally high suction lift, insufficient submergence, or low NPSH beyond the suction rating of the pump (see following paragraphs) usually cause serious reduction in capacity and efficiency, often leading to serious trouble from vibration and cavitation.

Two other conditions should also be fulfilled for adequate supply to the pump. First, the suction bell must be well below the water surface, and second, the intake must be of a functionally correct design. The requirements apply both to submerged pumps and to the suction pipe for any type of pump. This is true for all specific speeds.

Use of Suction Condition Terms

To avoid pump application errors, care must be exercised in the selection of terms to describe system conditions affecting the pump suction.

Total Suction Lift or Total Suction Head are the preferred terms when the liquid pumped is cold water and when the system is uncomplicated by extraneous factors such as an artificially produced vacuum.

Net Positive Suction Head (NPSH) is a useful term for more complicated pumping problems usually associated with handling liquids at or near their boiling points. Examples might be condensate return systems or the pumping of liquified gases.

Submergence is a term used to relate liquid level to the setting of an immersed pump with a free air surface. It is a static dimension, partially describing a system, and cannot be substituted for a dynamic term such as NPSH.

NPSH cannot be used interchangeably with the term *Suction Head*. *Suction Head* refers to pressure above atmospheric while *NPSH* is related to pressure above vapor pressure on the absolute scale.

Whenever practical, the proper term should be accompanied by:

A pictorial description of the system to help in the determination of suction losses.

Pertinent data concerning the liquid, such as its temperature, specific gravity, viscosity and vapor pressure.

The static lift (vacuum) or head to be imposed on the pump.

Net Positive Suction Head Available (NPSHA)

Net Positive Suction Head Available (NPSHA) is the total suction head in feet of liquid absolute corrected to datum (see Fig. 46, page 70) less the vapor pressure of the liquid in feet absolute. Therefore, NPSHA is the pressure or head available above vapor pressure to move and accelerate the fluid into the impeller inlet.

Thus

$$NPSHA = h_{sa} - h_{vpa}$$

where

h_{sa} = total suction head in feet absolute
or $= h_p + h_{se} - h_f$

$$NPSHA = h_p + h_{se} - h_f - h_{vpa}$$

where

h_p = Absolute pressure on the surface of the liquid where the pump takes suction expressed in feet of liquid. In an open system, h_p equals atmospheric pressure, h_a, expressed in feet absolute.

h_{se} = static elevation of the liquid above the datum point of the pump expressed in feet. If the liquid level is below the pump datum, h_{se} is a negative value.

h_f = friction and entrance head losses in the suction piping expressed in feet. If suction piping is not used, $h_f = 0$.

h_{vpa} = vapor pressure of the fluid at the pumping temperature expressed in feet of liquid absolute.

When the absolute pressure and vapor pressure are expressed in psia, the following formula may be used:

$$NPSHA = \frac{144}{w}(p_p - p_{vpa}) + h_{se} - h_f$$

where

p_p = Absolute pressure expressed in psia. In an open system, p_p equals atmospheric pressure, p_a, expressed in psia.

p_{vpa} = vapor pressure expressed in psia.

w = specific weight of liquid at the pumping temperature in pounds per cubic foot.

If a pump takes its suction from a source where the absolute pressure on the surface of the liquid, p_p, is equivalent to the vapor pressure, p_{vpa}, the NPSHA is the difference in elevation between the liquid level and the datum, minus the entrance and friction losses in the suction piping.

Thus

$$NPSHA = h_{se} - h_f$$

The formulas shown above are commonly used for determining the NPSHA for proposed installations and for measuring the NPSHA in existing installations without suction piping. The formula commonly used for measuring the NPSHA in existing installations with suction piping is as follows.

$$NPSHA = h_a + h_g + \frac{V^2}{2g} - h_{vpa}$$

where

h_a = atmospheric pressure, expressed in feet absolute

h_g = gauge pressure at the suction flange of the pump corrected to datum and expressed in feet of liquid. h_g is a negative value if it is below atmospheric pressure.

$\frac{V^2}{2g}$ = velocity head at the point of measurement of h_g. (This is necessary since gauge readings do not include the velocity head.)

Corrections for Higher Temperature and Elevation

Applications with water in an open system at sea level with a pumping temperature of 85 F are common. Given W = 62.4 lbs per cu ft, p_a = 14.7 psi, p_{vpa} = 0.6 psia, h_{se} = 10.0 ft, and $h_f = 0$, we find:

$$NPSHA = \frac{144}{W}(p_a - p_{vpa}) + h_{se} - h_f$$

$$= \frac{144}{62.4}(14.7 - 0.6) + 10.0 - 0.0$$

$$= 42.5 \text{ ft}$$

To find the NPSHA for water of 180 F temperature (p_{vpa} = 7.51 psia and w = 60.53 lbs per cu ft), proceed as follows:

$$NPSHA = \frac{144}{W}(p_a - p_{vpa}) + h_{se} - h_f$$

$$= \frac{144}{60.53}(14.7 - 7.51) + 10.0 - 0.0$$

$$= 27.1 \text{ ft}$$

To find the NPSHA for water of 180 F temperature at 5,000 feet elevation (p_a = 12.25 psi), proceed as follows:

$$NPSHA = \frac{144}{W}(p_a - p_{vpa}) + h_{se} - h_f$$

$$= \frac{144}{60.53}(12.25 - 7.51) + 10.0 - 0.0$$

$$= 21.3 \text{ ft}$$

The correction for elevation is approximately one foot per 1,000 feet of elevation.

Net Positive Suction Head Required (NPSHR)

The net positive suction head required (NPSHR) is a performance characteristic of the pump, and is established by test (see Cavitation Tests, pages 73 to 75).

Any system must be designed such that the net positive suction head available (NPSHA) is equal to, or exceeds, the net positive suction head required (NPSHR) by the pump through the range of operation to prevent detrimental cavitation.

Suction Specific Speed Required

Suction Specific Speed Required, S, is an index number descriptive of the suction characteristics of a given pump. It is defined as:

$$S = \frac{N\sqrt{Q}}{(NPSHR)^{3/4}}$$

where

S = Suction Specific Speed Required
N = Rotative Speed in Revolutions Per Minute
Q = Flow in Gallons Per Minute (use half of the total flow for double suction pumps)
NPSHR = Net Positive Suction Head Required in feet

Normally, the highest value of S is at, or near, the capacity corresponding to the optimum efficiency. However, special designs may cause the highest value of S to shift away from the point of optimum efficiency.

Higher numerical values of S are associated with better suction capabilities. The numerical value of S is mainly a function of the impeller inlet and suction inlet design. For pumps of normal design, values of S vary from 6,000 to 12,000. In special designs, including inducers, higher values can be obtained, however, special materials may be required for continuous operation. These standards are not intended to cover such special designs.

Suction Specific Speed Available

Suction Specific Speed Available, SA, is an index number descriptive of the available suction conditions of the system from which the pump is receiving suction It is defined as,

$$SA = \frac{N\sqrt{Q}}{(NPSHA)^{3/4}}$$

where

SA = Suction Specific Speed Available
N = Rotative Speed in Revolutions Per Minute
Q = Flow in Gallons Per Minute Required of the pump (use half of the total flow for double suction pumps)
NPSHA = Net Positive Suction Head Available in feet

The Suction Specific Speed Required, S, must equal or exceed the Suction Specific Speed Available, SA, to prevent cavitation The difference between S and SA is the safety margin Some of the factors that affect the degree of margin necessary are pump size, power consumption, intake design, range and mode of operation, type of service and materials of construction

Rotative Speed Limitations

Increased pump speeds without proper suction conditions can result in abnormal wear and possible failure from excessive vibration, noise and cavitation damage. Suction Specific Speed Available, SA, has been found to be a valuable criterion in determining the maximum permissible speed The curves presented in this standard are based on a Suction Specific Speed Available of 8500, this represents a practical value Obviously, values may be lower

On special applications, it is possible that some pumps may exceed 8,500 In such cases where the characteristics of the pump are based on the manufacturer's experience and test data, the values may be exceeded

Given the amount of NPSHA and substituting 8,500 for SA, one can solve for the Rotative Speed, N; for a given capacity, Q; by rearranging the formula for SA as follows

$$N = 8{,}500\,\frac{(NPSHA)^{3/4}}{\sqrt{Q}}$$

The curves (Figs 68 and 69) are a graphical representation of the SA formula. The curves show speeds for normal operating conditions and are based upon the premise that the pump is operating at, or near, its point of optimum efficiency

The curve, (Fig. 68) covers single suction centrifugal pumps

The curve, (Fig. 69) covers double suction pumps

EXAMPLE: Single Suction Pumps

Given a capacity of 90,000 GPM, and NPSHA of 50 feet, what is the RPM limit for 8,500 Suction Specific Speed Available?

$$N = \frac{SA(NPSHA)^{3/4}}{\sqrt{Q}}$$

therefore

$$N = \frac{8{,}500(50)^{3/4}}{(90{,}000)^{1/2}}$$

or

$$N = \frac{8{,}500(18.8)}{300}$$

and

$$N = 533$$

Therefore, the recommended maximum operating RPM is 533

From Fig 68, note that the intersection of the vertical line for 90,000 GPM, and the horizontal line for 50 feet of NPSHA corresponds to 533 RPM

EXAMPLE: Double Suction Pumps

Given a capacity of 20,000 GPM and NPSHA of 17 feet, what is the RPM limit for 8,500 Suction Specific Speed Available?

$$N = \frac{SA(NPSHA)^{3/4}}{\sqrt{\frac{Q}{2}}}$$

therefore,

$$N = \frac{8{,}500(17)^{3/4}}{\left(\frac{20{,}000}{2}\right)^{1/2}}$$

or,

$$N = \frac{8{,}500\ (8.37)}{100}$$

and

$$N = 711$$

Referring to the curve, Fig 69, the intersection of the vertical line for 20,000 GPM, and the horizontal line for 17 feet of NPSHA corresponds to an RPM of 711

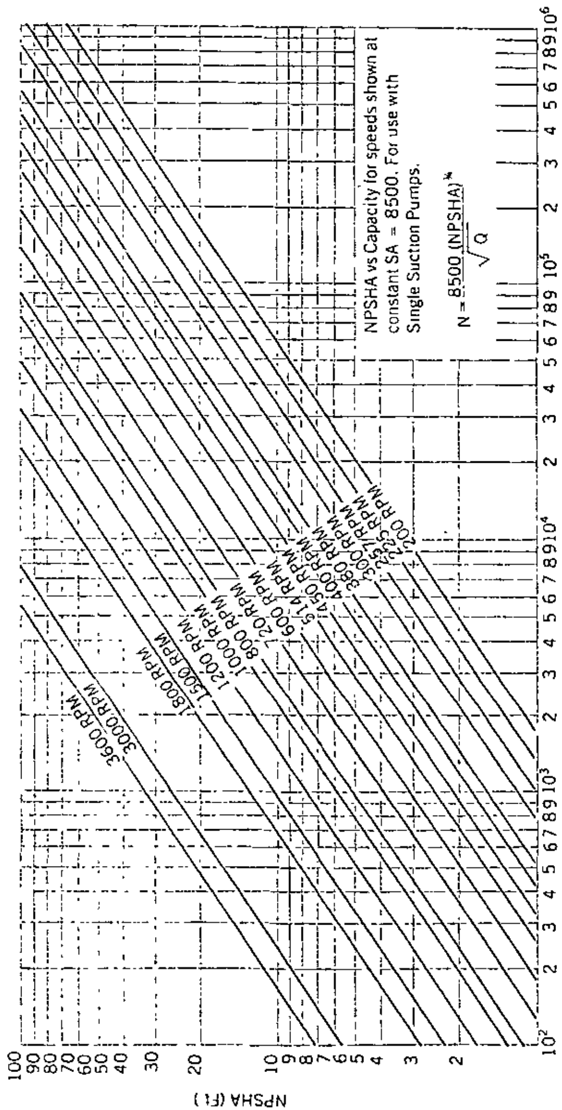


Fig. 68 RECOMMENDED MAXIMUM OPERATING SPEEDS FOR SINGLE SUCTION PUMPS

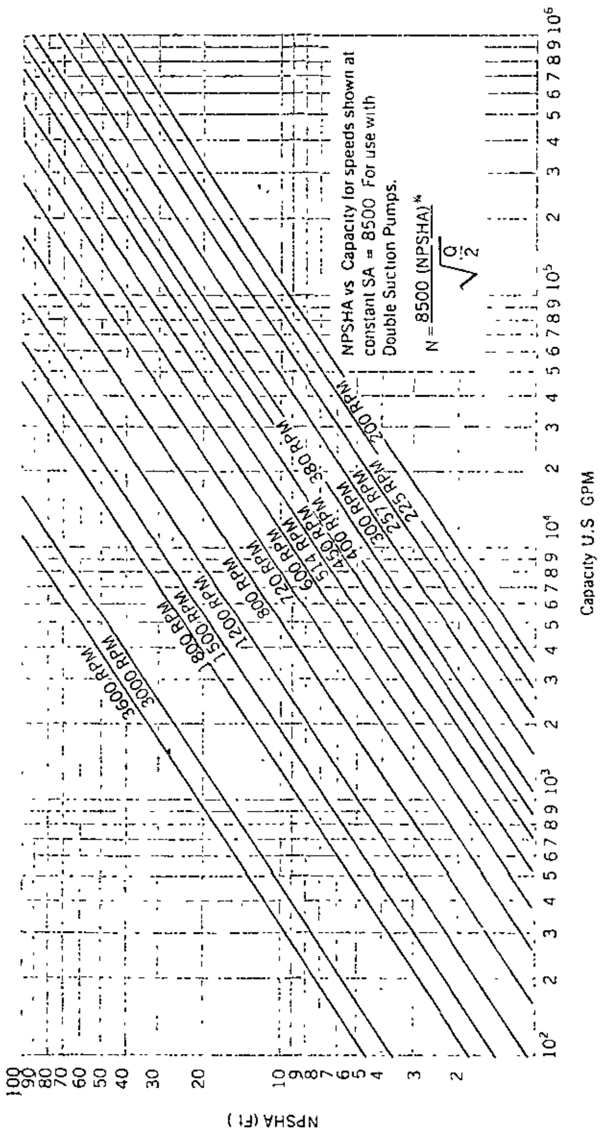


Fig. 69 RECOMMENDED MAXIMUM OPERATING SPEEDS FOR DOUBLE SUCTION PUMPS

EXAMPLE Single Suction Pumps

Given a capacity of 3,000 GPM, and NPSHA of 30 feet, what is the RPM limit for 8,500 Suction Specific Speed Available?

$$N = \frac{SA(NPSHA)^{3/4}}{\sqrt{Q}}$$

therefore

$$N = \frac{8{,}500(30)^{3/4}}{(3000)^{1/2}}$$

or

$$N = \frac{8{,}500(12\ 82)}{54\ 77}$$

and

$$N = 1990$$

Therefore, the recommended maximum operating RPM is 1990

From Fig. 68, note that the intersection of the vertical line for 3,000 GPM, and the horizontal line for 30 feet of NPSHA corresponds to 1990 RPM.

Net Positive Suction Head for Pumps Handling Hydrocarbon Liquids and Water at Elevated Temperatures

The NPSH requirements of centrifugal pumps are normally determined on the basis of handling water at or near normal room temperatures Operating experience in the field has indicated, and a limited number of carefully controlled laboratory tests have confirmed, that pumps handling certain hydrocarbon fluids, or water at significantly higher than room temperatures, will operate satisfactorily with less NPSH available than would be required for cold water

Figure 70 is a composite chart of NPSH reductions which may be expected for hydrocarbon liquids and high temperature water, based on available laboratory data from tests conducted on the fluids shown, plotted as a function of fluid temperature and vapor pressure at that temperature

Limitations for Use of Chart for Net Positive Suction Head Reduction (Fig. 70)

The following limitations and precautions should be observed in the use of Fig. 70

Until specific experience has been gained with operation of pumps under conditions where this chart applies, NPSH reduction should be limited to 50% of the NPSH required by the pump for cold water

This chart is based on pumps handling pure liquids Where *entrained* air or other noncondensable gases are present in a liquid, pump performance may be adversely affected even with normal NPSH available (see below) and would suffer further with reduction in NPSH available Where *dissolved* air or other noncondensables are present, and where the absolute pressure at the pump inlet would be low enough to release such noncondensables from solution, the NPSH available may have to be increased above that required for cold water to avoid deterioration of pump performance due to such release

For hydrocarbon mixtures, vapor pressure may vary significantly with temperature and specific vapor pressure determinations should be made for actual pumping temperatures.

In the use of the chart for high temperature liquids, and particularly with water, due consideration must be given to the susceptibility of the suction system to transient changes in temperature and absolute pressure, which might necessitate provision of a margin of safety of NPSH far exceeding the reduction otherwise available for steady state operation.

Because of the absence of available data demonstrating NPSH reduction greater than ten feet, the chart has been limited to that extent and extrapolation beyond that limit is not recommended

Instruction for Using Chart for Net Positive Suction Head Reduction (Fig. 70)

Enter Fig 70 at the bottom of the chart with pumping temperature in degrees F and proceed vertically upward to the vapor pressure in psia From this point follow along or parallel to the sloping lines to the right side of the chart, where the NPSH reduction in feet of liquid may be read on the scale provided If this value is greater than one half of the NPSH required on cold water, deduct one half of the cold water NPSH to obtain corrected NPSH required If the value read on the chart is less than one half of the cold water NPSH, deduct this chart value from the cold water NPSH to obtain corrected NPSH required

EXAMPLE: A pump that has been selected for a given capacity and head requires a minimum of 16 feet NPSH to pump that capacity when handling cold water In this case the pump is to handle propane at 55 F, which has a vapor pressure of 100 psia Following the procedure indicated above, the

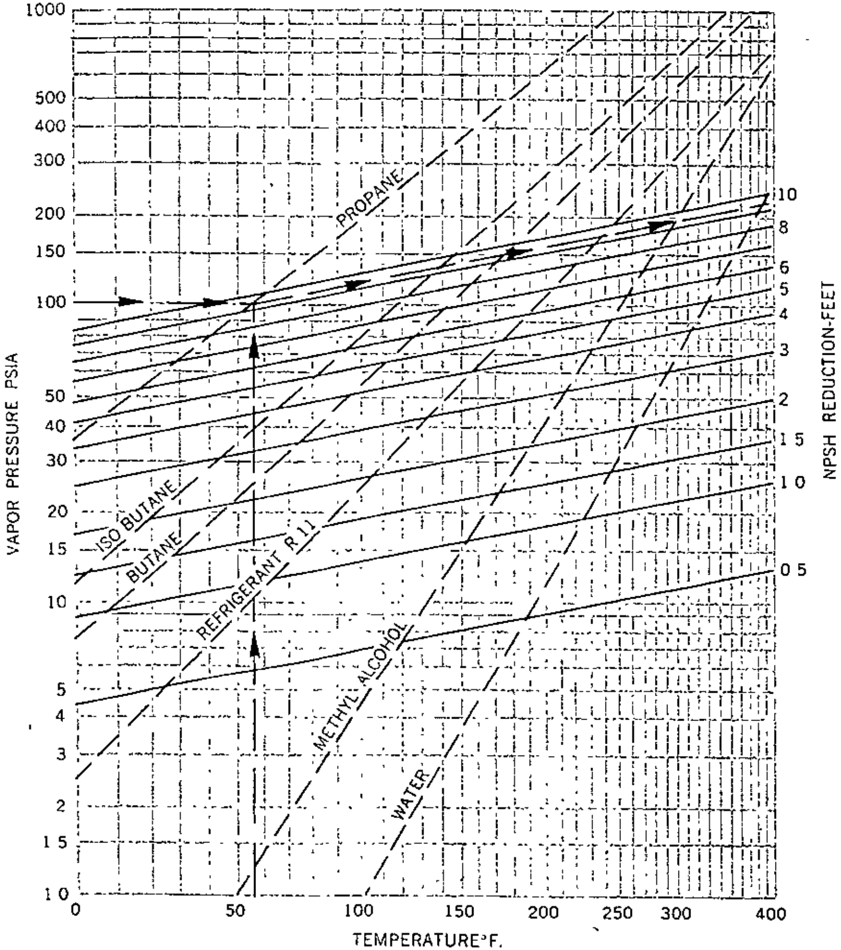


Fig. 70 NPSH REDUCTION FOR PUMPS HANDLING HYDROCARBON LIQUIDS AND HIGH TEMPERATURE WATER

Note This chart has been constructed from test data obtained using the liquids shown For applicability to other liquids refer to the text.

chart yields an NPSH reduction of 9 5 feet, which is greater than one half of the cold water NPSH required The corrected value of NPSH required is therefore one half the cold water NPSH required or 8 feet

EXAMPLE: The pump of example above has also been selected for another application to handle propane at 14 F, where it has a vapor pressure of 50 psia In this case the chart shows an NPSH reduction of 6 feet, which is less than one half the cold water NPSH The corrected value of NPSH is therefore 16 feet less 6 feet, or 10 feet.

Use of Chart for Net Positive Suction Head Reduction (Fig. 70) for Liquids Other Than Hydrocarbons or Water

The consistency of results which have been obtained on tests which have been conducted with both water and hydrocarbon fluids suggests that NPSH required by a centrifugal pump may be reduced when handling any liquid having relatively high vapor pressure at pumping temperature However, since available data are limited to the liquids for which temperature and vapor pressure relationships are shown on Fig 70, application of this chart to liquids other than hydrocarbons and water is not recommended except where it is understood that such usage can be accepted on an experimental basis:

Centrifugal Pumps Handling Entrained Air or Gas

Under a number of different circumstances, centrifugal pumps may be required to handle a mixture of air and water or similar mixtures It is known that this reduces the head, capacity and efficiency of a centrifugal pump, even when relatively small percentages of air or gas are present.

Deterioration of performance for a given percentage of air or gas varies from pump to pump depending on rotating speed, specific speed, pump size, suction pressure, discharge pressure, number of stages and various special design features. These mixtures may also have a detrimental effect on the mechanical operation of the pump. An explanation and evaluation of the effect of these factors is beyond the scope of this Standard

Determination of Pump Performance When Handling Viscous Liquids

The performance of centrifugal pumps is affected when handling viscous liquids A marked increase in brake horsepower, a reduction in head, and some reduction in capacity occur with moderate and high viscosities.

Figures 71 and 72 provide a means of determining the performance of a conventional centrifugal pump handling a viscous liquid when its performance on water is known Figs. 71 and 72 can also be used as an aid in selecting a pump for a given application The values shown in Fig 72 are averaged from tests of conventional single stage pumps of 2-inch to 8-inch size, handling petroleum oils. The values shown in Fig 71 were prepared from other tests on several smaller pumps (1 inch and below) The correction curves are, therefore, not exact for any particular pump.

When accurate information is essential, performance tests should be conducted with the particular viscous liquid to be handled.

Limitations on Use of Viscous Liquid Performance Correction Chart

Reference is made to Fig. 71 and Fig 72. Since these charts are based on empirical rather than theoretical considerations, extrapolation beyond the limits shown would go outside the experience range which these charts cover and is not recommended

Use only for pumps of conventional hydraulic design, in the normal operating range, with open or closed impellers Do not use for mixed flow or axial flow pumps, or for pumps of special hydraulic design for either viscous or non-uniform liquids.

Use only where adequate NPSH is available in order to avoid cavitation.

Use only on Newtonian (uniform) liquids. Gels, slurries, paper stock and other non-uniform liquids may produce widely varying results, depending on the particular characteristics of the liquids.

Symbols and Definitions Used in Determination of Pump Performance When Handling Viscous Liquids

These symbols and definitions are·

Q_{vis} = Viscous capacity in gpm—The capacity when pumping a viscous liquid.

H_{vis} = Viscous head in feet—The head when pumping a viscous liquid.

η_{vis} = Viscous efficiency in per cent —The efficiency when pumping a viscous liquid

bhp_{vis} = Viscous brake horsepower—The horsepower required by the pump for the viscous conditions.

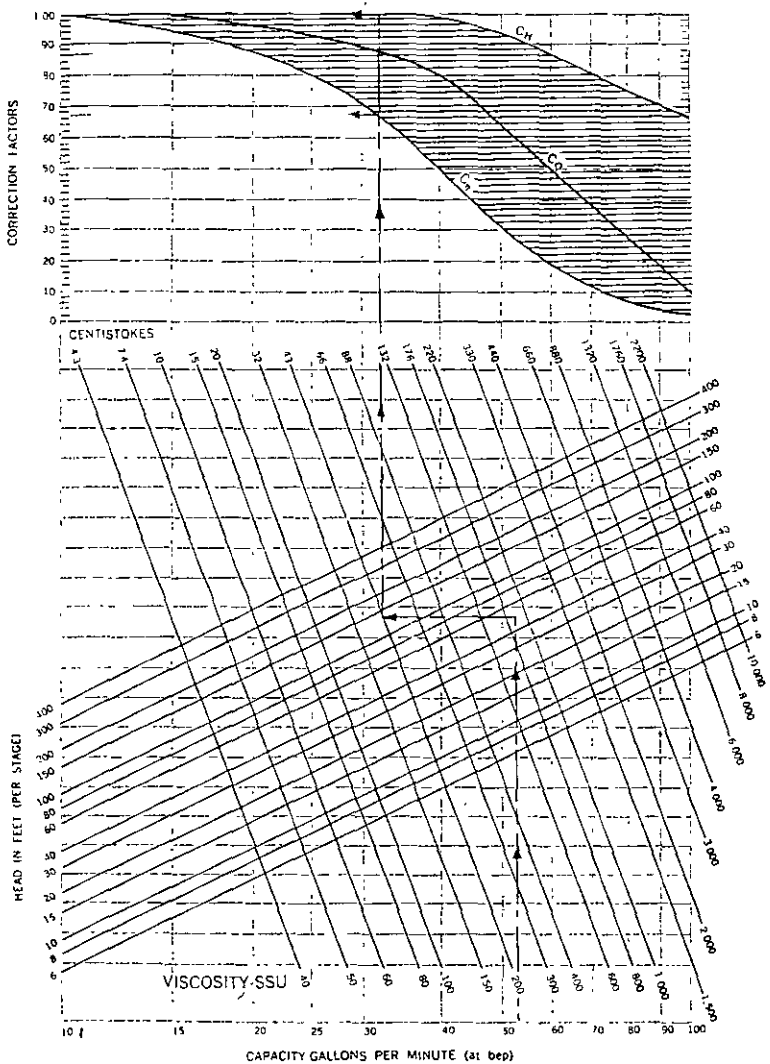


Fig. 71 PERFORMANCE CORRECTION CHART FOR VISCOUS LIQUIDS

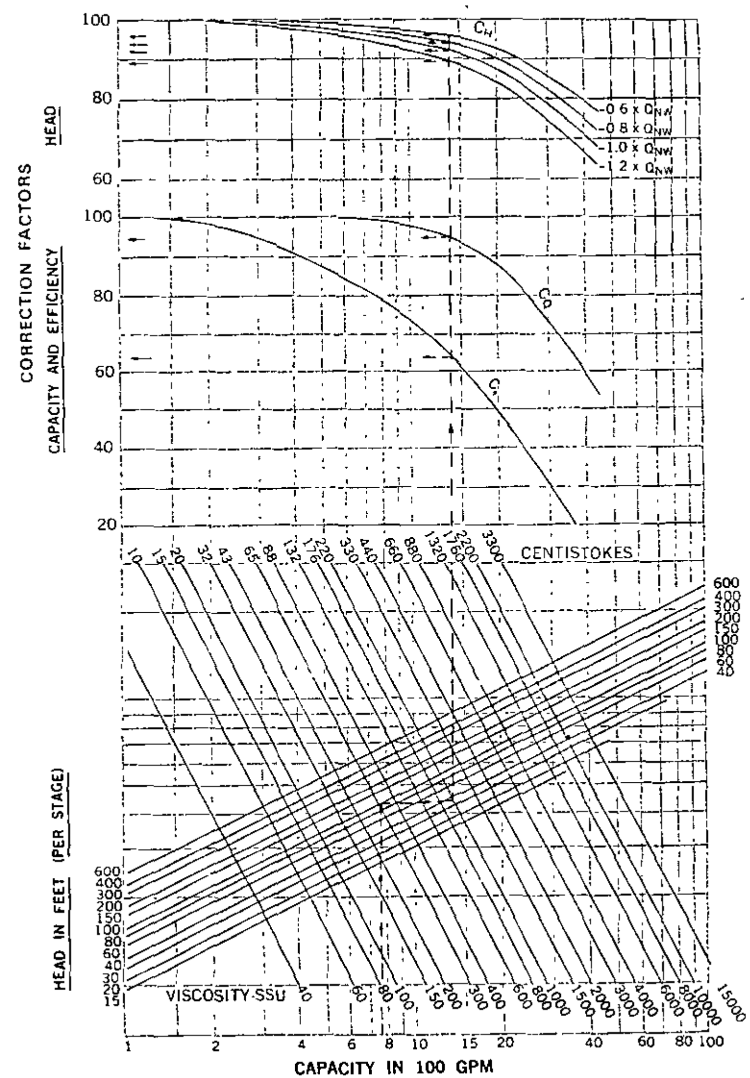


Fig. 72 PERFORMANCE CORRECTION CHART FOR VISCOUS LIQUIDS

Q_w = Water capacity in gpm —The capacity when pumping water
H_w = Water head in feet—The head when pumping water
η_w = Water efficiency in per cent—The efficiency when pumping water
s = Specific gravity
C_Q = Capacity correction factor
C_H = Head correction factor
C_η = Efficiency correction factor
Q_{NW} = Water capacity at which maximum efficiency is obtained

The following equations are used for determining the viscous performance when the water performance of the pump is known.

$$Q_{vis} = C_Q \times Q_w$$
$$H_{vis} = C_H \times H_w$$
$$\eta_{vis} = C_\eta \times \eta_w$$
$$bhp_{vis} = \frac{Q_{vis} \times H_{vis} \times s}{3960 \times \eta_{vis}}$$

C_Q, C_H and C_η are determined from Fig 71 and Fig 72 which are based on water performance Fig 71 is to be used for small pumps having capacity at best efficiency point of less than 100 gpm (water performance)

The following equations are used for approximating the water performance when the desired viscous capacity and head are given and the values of C_Q and C_H must be estimated from Fig 71 or 72 using Q_{vis} and H_{vis} as.

$$Q_w \text{ (approx)} = \frac{Q_{vis}}{C_Q}$$

$$H_w \text{ (approx)} = \frac{H_{vis}}{C_H}$$

Instructions for Preliminary Selection of a Pump for a Given Head-Capacity-Viscosity Condition

Given the desired capacity and head of the viscous liquid to be pumped, and the viscosity and specific gravity at the pumping temperature, Figs 71 or 72 can be used to find approximate equivalent capacity and head when pumping water

Enter appropriate chart at the bottom with the desired viscous capacity, (Q_{vis}) and proceed upward to the desired viscous head (H_{vis}) in feet of liquid For multistage pumps use head per stage Proceed horizontally (either left or right) to the fluid viscosity, and then go upward to the correction curves Divide the viscous capacity (Q_{vis}) by the capacity correction factor (C_Q) to get the approximate equivalent water capacity (Q_w approximately) Divide the viscous head (H_{vis}) by the head correction factor (C_H) from the curve marked "$1.0 \times Q_{NW}$" to get the approximate equivalent water head (H_w approximately) Using this new equivalent water head-capacity point, select a pump in the usual manner The viscous efficiency and the viscous brake horsepower may then by calculated

This procedure is approximate as the scales for capacity and head on the lower half of Fig. 71 or Fig 72 are based on water performance. However, the procedure has sufficient accuracy for most pump selection purposes Where the corrections are appreciable, it is desirable to check the selection by the method described below

EXAMPLE: Select a pump to deliver 750 gpm at 100 feet total head of a liquid having a viscosity of 1000 SSU and a specific gravity of 0 90 at the pumping temperature

Enter the chart (Fig 72) with 750 gpm, go up to 100 feet head, over to 1000 SSU, and then up to the correction factors

$C_Q = 0.95$
$C_H = 0.92$ (for $1.0\ Q_{NW}$)
$C_\eta = 0.635$

$$Q_w = \frac{750}{0.95} = 790 \text{ gpm}$$

$$H_w = \frac{100}{0.92} = 108.8 = 109 \text{ feet head}$$

Select a pump for a water capacity of 790 gpm at 109 feet head The selection should be at or close to the maximum efficiency point for water performance If the pump selected has an efficiency on water of 81 per cent at 790 gpm, then the efficiency for the viscous liquid will be as follows

$$\eta_{vis} = 0.635 \times 81\% = 51.5 \text{ per cent}$$

The brake horsepower for pumping the viscous liquid will be.

$$bhp_{vis} = \frac{750 \times 100 \times 0.90}{3960 \times 0.515} = 33.1 \text{ hp}$$

For performance curves of the pump selected, correct the water performance as discussed below.

Instructions for Determining Pump Performance on a Viscous Liquid When Performance on Water is Known

Given the complete performance characteristics of a pump handling water, determine the performance when pumping a liquid for a specified viscosity.

From the efficiency curve, locate the water capacity ($1.0 \times Q_{NW}$) at which maximum efficiency is obtained

From this capacity, determine the capacities ($0.6 \times Q_{NW}$), ($0.8 \times Q_{NW}$) and ($1.2 \times Q_{NW}$).

Enter the chart at the bottom with the capacity at best efficiency ($1.0 \times Q_{NW}$) go upward to the head developed (in one stage) (H_w) at this capacity, then horizontally (either left or right) to the desired viscosity, and then proceed upward to the various correction curves.

Read the values of (C_η) and (C_Q), and of (C_H) for all four capacities

Multiply each head by its corresponding head correction factor to obtain the corrected heads. Multiply each efficiency value by (C_η) to obtain the corrected efficiency values which apply at the corresponding corrected capacities

Plot corrected head and corrected efficiency against corrected capacity Draw smooth curves through these points The head at shut-off can be taken as approximately the same as that for water.

Calculate the viscous brake horsepower (bhp_{vis}) from the formula given on page 114.

Plot these points and draw a smooth curve through them which should be similar to and approximately parallel to the brake horsepower (bhp) curve for water

EXAMPLE. Given the performance of a pump (Fig. 73) obtained by test on water, plot the performance of this pump when handling oil with a specific gravity of 0 90 and a viscosity of 1000 SSU at pumping temperature

On the performance curve (Fig 73) locate the best efficiency point which determines Q_{NW} In this example it is 750 gpm Tabulate capacity, head and efficiency for (0.6×750), (0.8×750), (1.0×750) and (1.2×750) (See Table 7, Sample Calculations, on page 116)

Using 750 gpm, 100 feet head and 1000 SSU, enter the chart and determine the correction factors. These are tabulated in Table 7, Sample Calculations Multiply each value of head, capacity and efficiency by its correction factor to get the corrected values Using the corrected values and the specific gravity, calculate brake horsepower These calculations are shown on page 116. Calculated points are plotted in Fig 73 and corrected performance is represented by dashed curves.

Figure 71 is used in the same manner as Fig. 72 except that only the best efficiency point corrected performance is obtained Through the corrected head-capacity point, draw a curve similar in shape to the curve for water performance and having the same head at shut off. The corrected efficiency point represents the peak of the corrected efficiency curve, which is similar in shape to that for water The corrected brake horsepower curves are generally parallel to that for water

Radial Thrust in Single Volute Pumps

Single volute pump casings in the specific speed range between 500 and 3500 may be designed for uniform pressure around the volute casing at the design (best efficiency point) capacity For pumps in applications normally operating at or near the best efficiency point capacity, the thrust factor may approach zero On either side of the best efficiency point capacity, pressure distribution is not necessarily constant, resulting in radial thrust The radial thrust at shutoff may be approximated, for this type of design, using the following expression

$$R_{so} = K_{so} \times \frac{H_{so} \times s}{2.31} \times D_2 \times B_2$$

Thrust values R at capacities other than shut-off may be approximated by the following formula

$$R = \frac{K}{K_{so}} \times \frac{H}{H_{so}} \times R_{so}$$

where

$$K = K_{so}\left[1 - \left(\frac{Q}{Q_N}\right)^x\right]$$

and

R_{so} = Radial thrust in pounds, at shutoff
R = Radial thrust in pounds, at operating condition
K_{so} = Thrust factor at shutoff from Fig 74
K = Thrust factor at operating condition
H_{so} = Total head at shutoff in feet
H = Total head at operating condition in feet
s = Specific gravity of the liquid
D_2 = Impeller diameter in inches
B_2 = Impeller width at discharge including shrouds in inches
Q = Capacity at operating condition, in gpm
Q_N = Capacity at best efficiency point in gpm
x = Exponent, varying between 0.7 and 3.3 established by test. In the absence of test data, the exponent may generally be assumed to vary linearly between 0.7 at specific speed 500 and 3 3 at specific speed 3500

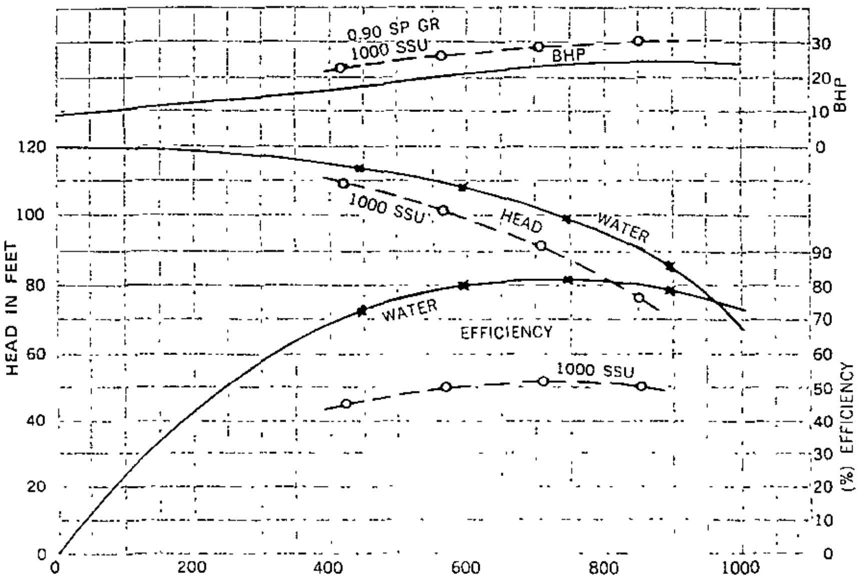


Fig. 73 SAMPLE PERFORMANCE CHART

TABLE 7
Sample Calculations

Sample Calculations

	$0.6 \times Q_{NW}$	$0.8 \times Q_{NW}$	$1.0 \times Q_{NW}$	$1.2 \times Q_{NW}$
Water capacity (Q_w)	450	600	750	900
Water head in feet (H_w)	114	108	100	86
Water efficiency (η_w) (%)	72 5	80	82	79 5
Viscosity of liquid	1000 SSU	1000 SSU	1000 SSU	1000 SSU
C_Q—from chart	0 95	0 95	0 95	0 95
C_H—from chart	0.96	0 94	0 92	0.89
C_η—from chart	0 635	0.635	0 635	0 635
Viscous capacity—$Q_w \times C_Q$	427	570	712	855
Viscous head—$H_w \times C_H$	109.5	101 5	92	76.5
Viscous efficiency—$\eta_w \times C_\eta$	46 0	50.8	52 1	50.5
Specific gravity of liquid	0 90	0 90	0 90	0 90
bhp viscous	23 1	25 9	28 6	29 4

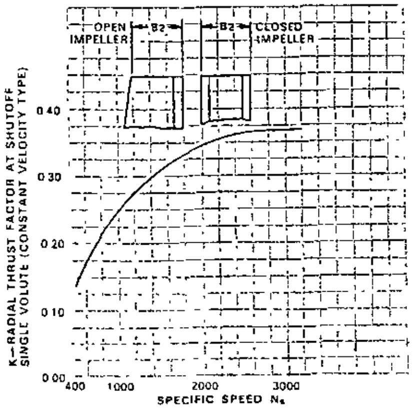


Fig. 74 K_{so} = RADIAL THRUST FACTOR AT SHUTOFF SINGLE VOLUTE (CONSTANT VELOCITY TYPE)

Radial thrust factors (K_{so}) at shutoff are shown on Fig 74. Individual manufacturers may have data on specific designs demonstrating other values than those shown

Balancing of Centrifugal Pump Rotors

Unbalance of a pump's rotor exists whenever the center of mass is radially displaced from the center of rotation in at least one plane which is at right angles to the shaft axis Unless the extent of such unbalance is kept within specific limits. (Figs 75–78) the pump may vibrate excessively and suffer premature failure

The establishment of realistic limits of residual unbalance for pump rotors is a complex problem which must take into account several factors, among which are maximum operating speeds, ratio of rotor weight to pump weight, type of bearings with which the unit is to be equipped, dynamic response characteristics of the rotor, mass and relative rigidity of the pump supporting structure, ratio of operating to critical speeds, and the nature of the environment in which the pump is to be installed. Since these factors will vary over extreme ranges for pumps covered by these standards, it is impractical to establish here numerical limits of residual unbalance.

Accordingly, methods of balancing and limits of residual unbalance are to be established by the pump manufacturer, subject to the requirement that the field vibration limits specified in the following section of these standards can be satisfied

Radial Vibration of Centrifugal Pumps

Vibration of a centrifugal pump is related to the rigidity of the support structure It is, therefore, necessary to discuss the subject based on the type of structure in question.

Rigid Structure

A rigid pump structure or mounting is defined as one for which the pump structure has a fundamental natural frequency higher than the shaft maximum rotative speed for the installation. It is possible for any pump manufacturer to calculate or determine by test this natural frequency In a given installation, the fundamental natural frequency of the base and foundation on which the pump is mounted must be well above the pump rotational speed; otherwise the condition of a "rigid structure" is not satisfied

Assuming a rigid structure, some of the sources of vibration are imbalance of rotating parts, hydraulic forces produced between the impeller and casing volute or guide vane, and coupling misalignment Responsibility for the elimination of vibration due to the first two sources lies with the pump manufacturer, whereas the third one lies with the installer

Experience shows that hydraulic forces in low specific speed pumps running near peak points of efficiency, are not sufficient to have a great effect on a rigid type structure. In a rigid structure, therefore, the principal consideration must be given to the proper balance of rotating parts

Figures 75 and 76 show recommended upper limits of vibration for pumps that are classified as rigid structure

Non-Rigid Structure

Non-rigid or compliant pump structures herein defined are those for which the shaft rotational speed is near the structure's first natural frequency The term "Reed Frequency" has been used to identify the first natural frequency of a unit structure in a given installation. This natural frequency is unique to a given installation and may not be measurable in any other mounting such as in a shop test. Normally, a flexibly supported pump unit will have a lower "Reed Frequency" than when it is installed in a typical permanent system.

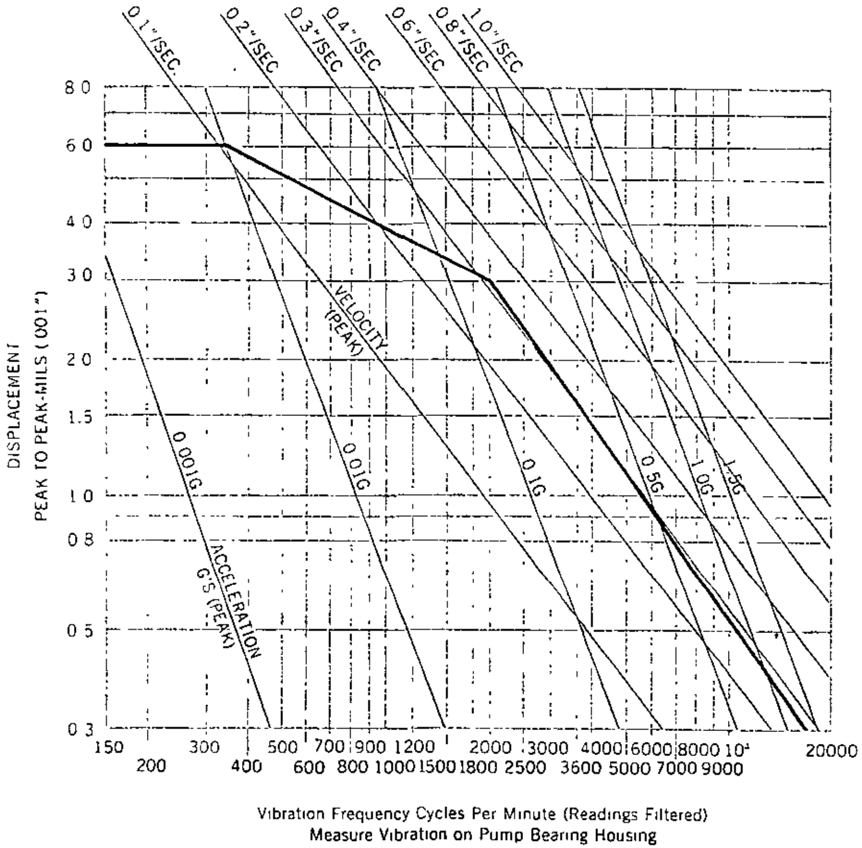


Fig. 75 ACCEPTABLE FIELD VIBRATION LIMITS FOR HORIZONTAL PUMPS—CLEAR LIQUID (RIGID STRUCTURES)

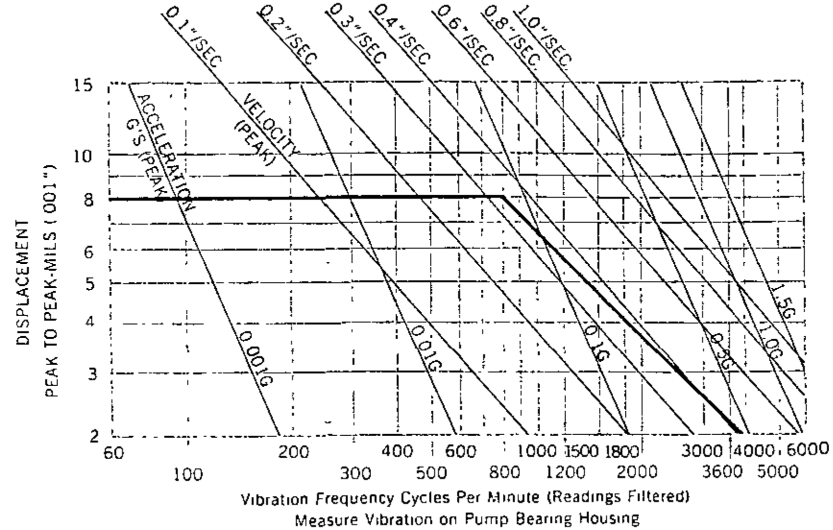


Fig 76 ACCEPTABLE FIELD VIBRATION LIMITS FOR HORIZONTAL NON-CLOG PUMPS (RIGID STRUCTURES)

"Reed Frequency" is best determined by exciting the structure with a variable speed frequency exciter. A vibration pick-up such as a vibration analyzer with filtered input is used to determine the resonant frequency as exciter frequency is varied. Alternately, the "Reed Frequency" may be obtained approximately by the "bump" method. In this technique, the structure is excited by striking with a "soft hammer" (a heavy piece of wood is ideal). This will cause the structure to vibrate at its "Reed Frequency." A vibration analyzer with a "frequency finder" capability or other more sophisticated equipment may be used to measure the frequency of this vibration

In a non-rigid system, units may operate below or above the "Reed Frequency." A manufacturer can design or modify his structure to control the resonant frequency of his structure only Accepted practice is to avoid the structure's natural frequency by approximately 25% above or below Since resonance of the installed unit is the responsibility of the system designer, coordination between pump manufacturer and system designer is necessary to avoid operation at or near the "Reed Frequency" of the installed unit

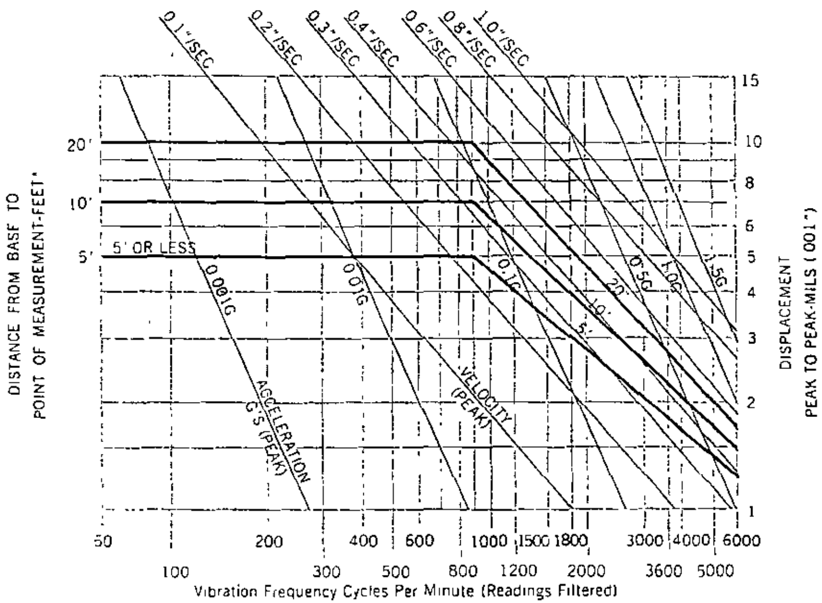
Figures 77 and 78 provide vibration limits for non-rigid structure pumps Vibration limits are not easily classified for a large range of products Forces on bearings and structural components of varying stiffness are the factors to be considered rather than displacement levels.

Other Factors Affecting Vibration

There are a number of factors besides physical unbalance of the rotating parts which may cause vibration Among these are-

Resonance between the unit and its foundation or piping Resonant vibrations caused by driver or other equipment in operation in the area

Operation at or near a critical speed (The amount of vibration observed will depend on the degree of



*Measure Vibration at top motor bearing

Fig. 77 ACCEPTABLE FIELD VIBRATION LIMITS FOR VERTICAL PUMPS & HORIZONTAL PUMPS WITH PIGGYBACK MOUNTED MOTORS (NON RIGID STRUCTURES)

unbalance and damping present. Normal design practice is to avoid a critical speed by approximately 25 per cent.)

Vibrations due to hydraulic disturbances caused by improper design of the suction piping or sump. Disturbances may also be caused by improperly designed valves, piping supports, piping and other components exterior to the pump (Such vibrations are usually at random frequencies.)

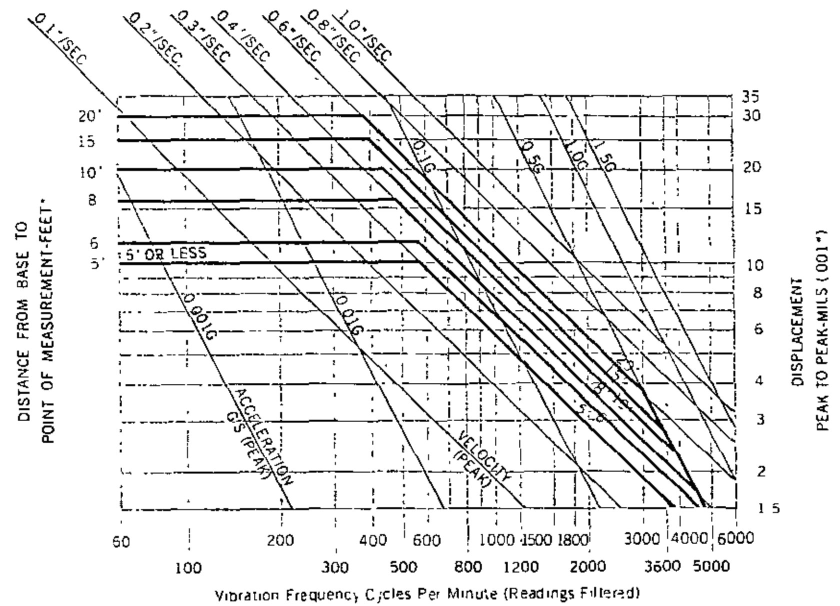
Torsional vibrations resulting from a combination of driver, coupling, and pump which has a natural torsional frequency at, or near, a harmonic of the pump or driver rotating speed.

For the non-clog pumps, sudden increases in the vibration levels may be due to the passage of large solids through the pump (If the vibration condition persists, solids may be lodged in the impeller, and remedial measures should be taken to clear it.)

In using these curves, the following conditions apply:

Pumps must be operating in a non-cavitating and non-separating condition Suction piping should be arranged so as to provide a straight, uniform flow to the pump Piping should be connected in such a way so as to avoid undo strains on the pump. Shaft couplings are aligned to within the pump manufacturer's recommendations

Figures 75 and 76 should be used as a general guide with recommendations that vibrations in excess of the curve values may require investigation



*Measure vibration at top motor bearing or at top pump bearing if motor is not integrally mounted to pump

Fig. 78 ACCEPTABLE FIELD VIBRATION LIMITS FOR VERTICAL NON-CLOG PUMPS (NON-RIGID STRUCTURES)

and correction Often, more important than the actual vibration itself, is the change of vibration over a period of time Vibrations in excess of the curve values may be acceptable if they show no increase over long periods of time, and if there is no other indication of damage, such as increase in bearing clearance or noise level.

Conversion Formulas

Symbols

D = Displacement Mils (.001"), Peak to Peak

V = Velocity Inches per second, Peak

A = Acceleration: G's, Peak

CPM = Cycles per minute

$$D = (1.910 \times 10^4)\frac{V}{CPM}$$

$$V = (3.696 \times 10^3)\frac{A}{CPM}$$

$$A = (2.704 \times 10^{-4})(CPM)(V)$$

Mechanical Seal Flush Loops

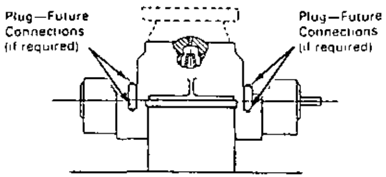
The purpose of a mechanical seal flush loop system is to provide the pump mechanical seal chamber with clean liquid of the proper temperature to establish an environment suitable for mechanical seal operation

The flush loops illustrated (pages 122–124) represent those most commonly used Other systems may be specified by the pump user, giving flushing fluid characteristics including pressure, temperature and viscosity.

DOUBLE SUCTION, DOUBLE BEARING TYPE PUMP

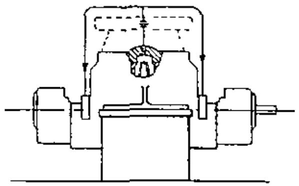
CLEAN PUMPAGE

PLAN 1



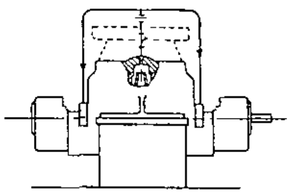
Dead end seal boxes with no circulation or flush fluid. Water cooled box jackets and throat bushings required unless otherwise specified

PLAN 4 (BASIC SYSTEM)



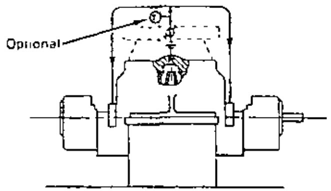
Recirculation from pump case through orifice to seals

PLAN 2



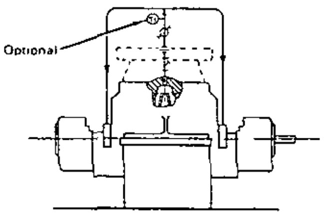
Recirculation from pump case through strainer and orifice to seals

PLAN 5



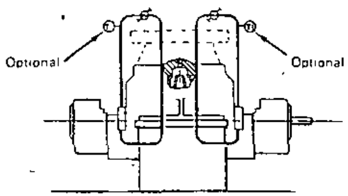
Recirculation from pump case through orifice and cooler to seals

PLAN 3



Recirculation from pump case through strainer, orifice and cooler to seals

PLAN 6



Recirculation from seals with pumping ring through cooler and back to seals

Fig. 79 (Part One) FLUSH LOOP SYSTEMS

Notes

These plans are representative of commonly used systems. Variations in specific arrangements and materials as long as they meet the intent of the system are permissible.

For plans 8, 9, 10, and 12, purchaser shall specify the fluid characteristics and vendor shall specify the required GPM and PSIG. When supplemental seal fluid is provided, vendor shall specify the required GPM and PSIG where these are factors (such as when auxiliary seal is outside mechanical type)

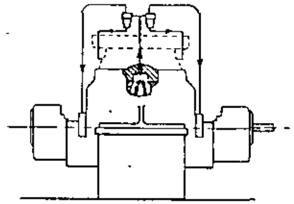
Piping materials shall follow case metallurgy unless otherwise specified by the customer

The pumps depicted are representative and not intended for any specific service

DOUBLE SUCTION, DOUBLE BEARING TYPE PUMP

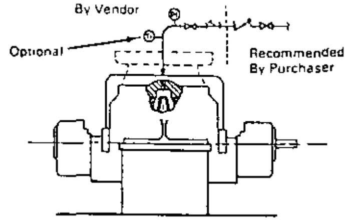
DIRTY OR SPECIAL PUMPAGE

PLAN 7



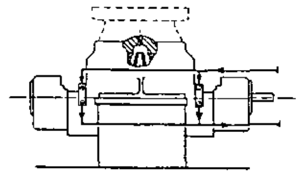
Recirculation from pump case through cyclone separator delivering clean fluid to seals, and fluid with solids back to pump section.

PLAN 10



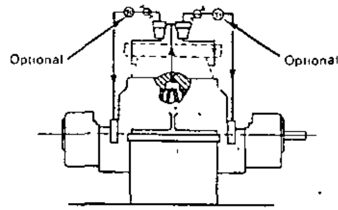
Injection to seals from external source of clean cool fluid. (See Notes)

PLAN 8



Circulation of clean cool fluid to double seals from an external circulation system (See Notes)

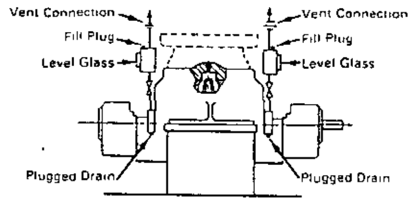
PLAN 11



Recirculation from pump case through cyclone separator delivering clean fluid through cooler to seals and fluid with solids, back to pump suction

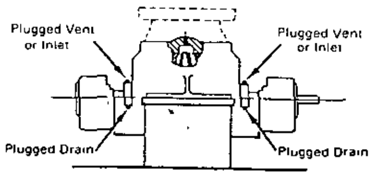
PIPING FOR THROTTLE BUSHING OR AUXILIARY SEAL DEVICE

PLAN 9



Dead Ended Blanket Usually Methanol (See Notes)

PLAN 12



Tapped connections for purchaser's use. Shall apply when purchaser is to supply fluid (steam, gas, water, other) to auxiliary sealing device (See Notes)

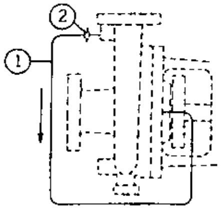
LEGEND

- Orifice
- Y-Type Strainer
- Cooler
- Separator
- (PI) Pressure Gauge W / cock
- Valve-Regulating
- Valve-Block
- Valve-Check
- (TI) Dial Thermometer
- (PS) Pressure Switch

Fig. 79 (Part Two) FLUSH LOOP SYSTEMS

SINGLE SUCTION PUMP

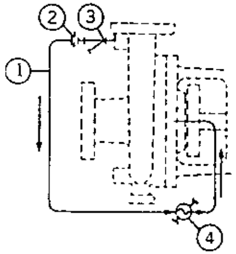
A—For Clean Pumpage
Recirculation from pump discharge connection through an orifice to flush connection



	Part
1	Tubing
2	Orifice

Terminal end of loop feeds to
☐ Stuffing Box Connection
☐ Mechanical Seal Gland Connection

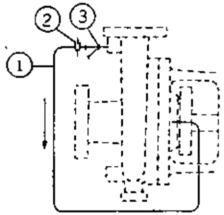
B—For Clean Pumpage
Recirculation from pump discharge connection through a "Y" strainer, the orifice, through a heat exchanger to seal flush connection



	Part
1	Tubing
2	Orifice
3	"Y" Strainer
4	Heat Exchanger

Terminal end of loop feeds to:
☐ Stuffing Box Connection
☐ Mechanical Seal Gland Connection

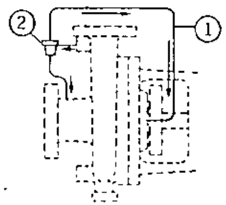
C—For Clean Pumpage
Recirculation from pump discharge connection through a "Y" strainer, through an orifice to flush connection.



	Part
1	Tubing
2	Orifice
3	"Y" Strainer

Terminal end of loop feeds to
☐ Stuffing Box Connection
☐ Mechanical Seal Gland Connection

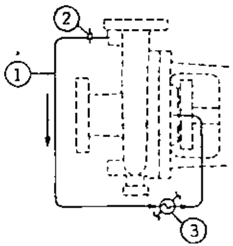
D—For Dirty or Special Pumpage
Recirculation from pump discharge connection through cyclone separator delivering clean pumpage to flush connection and pumpage with back to pump suction.



	Part
1	Tubing
2	Separator

Terminal end of loop feeds to:
☐ Stuffing Box Connection
☐ Mechanical Seal Gland Connection

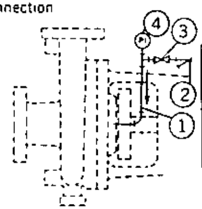
E—For Clean Pumpage
Recirculation from pump discharge connection through an orifice, through a heat exchanger to flush connection



	Part
1	Tubing
2	Orifice
3	Heat Exchanger

Terminal end of loop feeds to
☐ Stuffing Box Connection
☐ Mechanical Seal Gland Connection

F—For Dirty or Special Pumpage
Injection to seal flush connection from external source through "Y" strainer, globe valve, past pressure gauge to flush connection



	Part
1	Piping
2	"Y" Strainer
3	Globe Valve
4	Pressure Gauge

Terminal end of loop feeds to·
☐ Stuffing Box Connection
☐ Mechanical Seal Gland Connection

Fig. 80 FLUSH LOOP SYSTEMS

Intake Design

The function of the intake, whether it be an open channel or a tunnel having 100 per cent wetted perimeter, is to supply an evenly distributed flow of water to the suction bell An uneven distribution of flow, characterized by strong local currents, favors formation of vortices and with certain low values of submergence, will introduce air into the pump with reduction of capacity, accompanied by noise. Uneven distribution can also increase or decrease the power consumption with a change in total developed head There can be vortices which do not appear on the surface, and these also may have adverse effects

Uneven velocity distribution leads to rotation of portions of the mass of water about a center-line called vortex motion This centerline may also be moving. Uneven distribution of flow is caused by the geometry of the intake and the manner in which water is introduced into the intake from the primary source

Calculated low *average* velocity is not always a proper basis for judging the excellence of an intake High *local* velocities in currents and in swirls may be present in intakes which have very low *average* velocity Indeed, the uneven distribution which they represent occurs less in a higher velocity flow with sufficient turbulence to discourage the gradual build-up of a larger and larger vortex in any region. Numbers of small surface eddies may be present without causing any trouble

The ideal approach is a straight channel coming directly to the pump. Turns and obstructions are detrimental since they may cause eddy currents and tend to initiate deep-cored vortices

Water should not flow past one pump to reach the next if this can be avoided If the pumps must be placed in line of flow, it may prove necessary to construct an open front cell around each pump or to put turning vanes under the pump to deflect the water upward

All possible streamlining should be used to reduce the trail of alternating vortices in the wake of the pump or of other obstructions in the stream flow

The amount of submergence for successful operation will depend greatly on the approaches to the intake and the size of the pump. While specific design is generally beyond the scope of the pump manufacturer's responsibility, he may comment while the intake layout is still preliminary if he is provided with the necessary intake drawings reflecting the physical limitations of the site.

Complete analysis of intake structures is best accomplished by scale model tests

Subject to the qualifications of the foregoing statements, Figs. 81, 82, & 83 have been constructed for single and simple multiple pump arrangements to show suggestions for basic sump dimensions They are for pumps normally operating in the capacity range of approximately 3,000 to 300,000 gpm. Since these values are composite averages from a great many pump types and cover the entire range of specific speeds, they must not be thought of as absolute values but rather as basic guides subject to some possible variations For pumps normally operating at capacities below approximately 3,000 gpm, refer to Sump or Pit Designs (small pumps) page 129.

All of the dimensions in Figs 81, 82 & 83 are based on the rated capacity of the pump at the design head Any increase in capacity above these values should be momentary or very limited in time. If operation at an increased capacity is to be undertaken for considerable periods of time, the maximum capacity should be used for the design value in obtaining sump dimensions

The Dimension C is an average, based on an analysis of many pumps. Its final value should be specified by the pump manufacturer.

Dimension B is a suggested maximum dimension which may be less depending on actual suction bell or bowl diameters in use by the pump manufacturer The edge of the bell should be close to the back wall of the sump. When the position of the back wall is determined by the driving equipment or the discharge piping, Dimension B may become excessive and a "false" back wall should be installed.

Dimension S is a minimum for the sump width for a single pump installation. This dimension can be increased, but if it is to be made smaller, the manufacturer should be consulted or a sump model test should be run to determine its adequacy.

Dimension H is a minimum value based on the "normal low water level" at the pump suction bell, taking into consideration friction losses through the inlet screen and approach channel. This dimension can be considerably less momentarily or infrequently without damage to the pump It should be remembered, however, that this does not represent "submergence." Submergence is normally quoted as dimension H minus C. This represents the physical height of water level above the bottom of the suction inlet The actual submergence of the pump is something less than this, since the impeller eye is some distance above the bottom of the suction bell, the magnitude being a function of pump size and style For the purposes of sump design in connec-

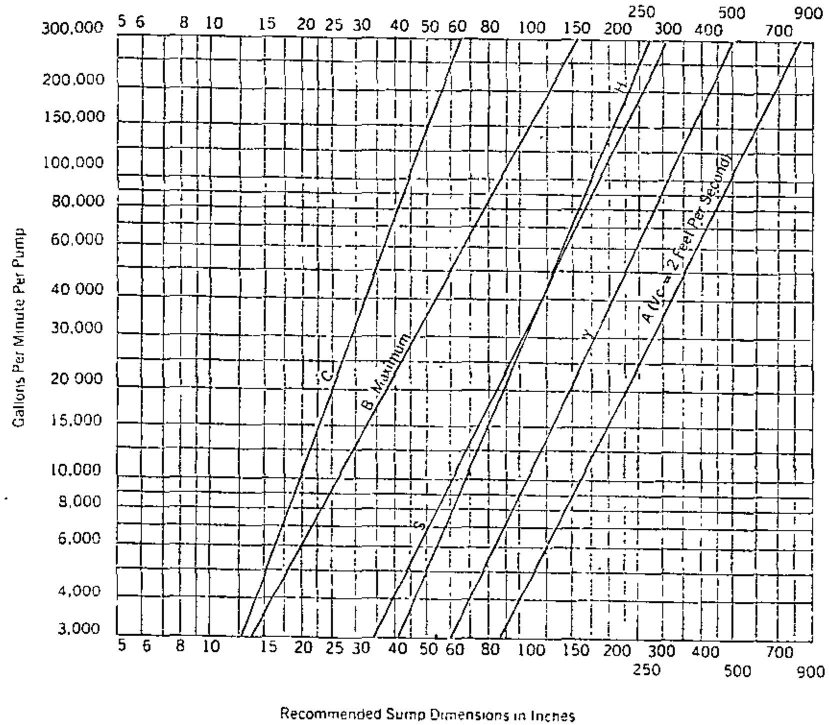


Fig. 81 SUMP DIMENSIONS VERSUS FLOW

Note: Recommended value of Y equals approximately 30 for most bell designs

tion with this chart, it is understood that the pump has been selected in accordance with maximum speeds charts, Figs 68 and 69, the submergence referred to herein having to do only with vortexing and eddy formations

Dimensions Y and A are recommended minimum values These dimensions can be as large as desired but should be limited to the restrictions indicated on the curve If the design does not include a screen,

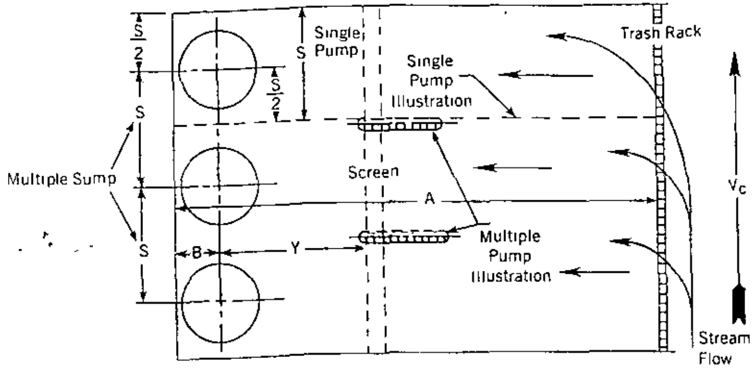


Fig. 82 SUMP DIMENSIONS PLAN VIEW

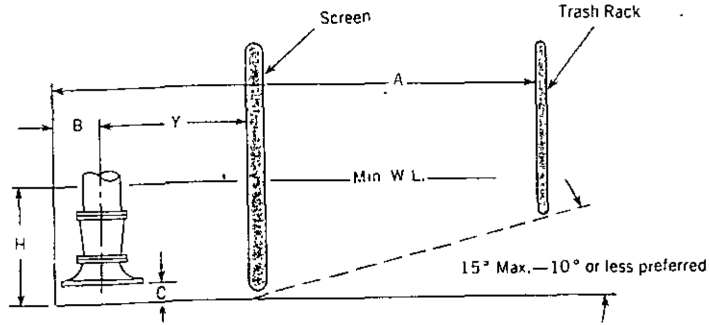


Fig. 83 SUMP DIMENSIONS ELEVATION VIEW

or if the channel has a sloping approach, dimension A should be considerably longer, even as much as twice the value shown If the channel approach has a downslope, the angle should be not more than 15 degrees and preferably 10 degrees with the horizontal The channel floor should be level for at least distance Y upstream before the slope begins. The screen or gate widths should not be substantially less than S, and heights should not be less than H. If the mainstream velocity is more than 2 feet per second, it may be necessary to construct straightening vanes in the approach channel, increase dimension A, conduct a sump model test of the installation, or work out some combination of these factors

Dimension S becomes the width of an individual pump cell or the center-to-center distance of two pumps if no division walls are used

On multiple pump installations, the recommended dimensions in Figs 81, 82 and 83 also apply as noted above, and the following additional determinants should be considered.

Fig 84A. Low velocity and straight-line flow to all units simultaneously is the first recommended style of pit Velocities in pump area should be approximately one foot per second. Some pumps with velocities of 2 feet per second and higher have given good results. This is particularly true where the design resulted from a model study Not recommended would be abrupt change in size of inlet pipe to sump or inlet from one side introducing eddying

Fig 84B. A number of pumps in the same sump will operate best without separating walls unless all pumps are always in operation at the same time, in which case the use of separating walls may be beneficial If walls must be used for structural purposes, and pumps will operate intermittently, leave flow space behind each wall from the pit floor up to at least the minimum water level.

If walls are used, increase dimension S by the thickness of the wall for correct centerline spacing Round or "ogive" ends of walls NOT recommended is the placement of a number of pumps around the edge of a sump with or without dividing walls

Fig 84C. Abrupt changes in size from inlet pipe or channel to pump bay are not desirable. A relatively small pipe emptying into a large pump pit should connect to the pit with a gradually increasing taper section The angle should be as small as possible, preferably not more than 15 degrees With this arrangement, pit velocities much less than one foot, per second are desirable Especially not recommended is a small pipe directly connected to a large pit with pumps close to the inlet Flow will have excessive change of direction to get to most of the pumps Centering pumps in the pit leaves large "vortex areas" behind the pumps which will cause operational trouble.

Fig 84D If the pit velocity can be kept low enough (approximately one foot per second), an abrupt change from inlet pipe to pit can be accommodated if the length equals or exceeds the values shown. As ratio W/P increases, the inlet velocity at p may be increased up to an allowed maximum of eight feet per second at W/P = 10 Pumps "in line" are not recommended unless the ratio of pit to pump size is quite large, and pumps are separated by a generous margin longitudinally A pit can generally be constructed at much less cost by using a recommended design

Fig 84E It is sometimes desirable to install pumps in tunnels or pipe lines A drop pipe or false well to house the pump with vaned inlet elbow facing upstream will be satisfactory in flows up to eight feet per second Without the inlet elbow, the pump section bell should be positioned at least two pipe (vertical) diameters above the top of the tunnel, not hung into the tunnel flow, especially with tunnel velocities of two feet per second or more. There should be no signs of air along the top of tunnel It may be necessary to lower the scoop or insist on minimum water level in vertical well

Note: The foregoing statements apply to sumps for clear liquid. For fluid-solids mixtures refer to the pump manufacturer

CORRECTION OF EXISTING SUMPS

It is well established that vortexing in pump suction pits is harmful to pumps and intake structures. It is equally true that a very small force will actually begin generating a vortex. While this phenomenon can be avoided in a new design, for existing structures where problems are already apparent or where expansion is required, corrective measures may be necessary. Possible revisions to correct particular sump problems are shown in Fig 85. In many cases, field modifications are expensive with no guarantee of success. It is recommended that a sump model test be considered to prove the effectiveness of the proposed changes.

Fig 85A—Reduce inlet velocity by spreading the inflow over a larger area, or change the direction and velocity of inflow by suitable baffling (The baffle may be floor mounted, extending above the minimum flow level, or may be hung from above, extending close to the floor)

Fig 85B—Change the location of pumps in relation to the inflow A suitable baffle may be necessary in front of inlet

Fig 85C—A cone may be added to reduce the possibility of submerged vortex formation

Fig 85D—Provide break-thru to "no-flow" bays in multiple pump pits and round or "ogive" ends of separating walls, or

Fig. 85E—Eliminate separating walls.

Fig. 85F—Eliminate sharp corners at gates, screens, etc , by filling in for smooth flow contour (fairing)

Fig 85G—Reduce the velocity of flow and eliminate vortexing by adding bell extension suction plate and splitter to pump bell Splitter must be in line with the flow.

Fig 85H—Use floating rafts around the pump column to prevent surface vortices

Fig. 85I—Use large spheres to prevent surface vortices

Fig. 85J—Reduce the clearance between the pump inlet and back wall This will improve velocity pattern to the pump to reduce the possibility of vortex formation

Fig. 85K—Change inlet flow direction gradually by means of parallel turning vanes.

In General:

Keep inlet flow below two ft per second

Keep flow in pit below one ft per second

Avoid changing direction of flow from inlet to pump, or

Change direction gradually, smoothly, independently

In addition to the above, other modifications such as 1) Suction Cone, 2) Horizontal perforated plate, or grids; 3) Horizontal beam with bottom flange submerged to control water surface, and 4) other flow straighteners may be used to correct the existing sumps

Any of these alterations, singly or in combination, may help to create a better flow pattern in the sump If troubles persist, it may be necessary to limit the total flow or change pump size and speed

Model Tests of Intakes

Often the analysis of a proposed intake design can only be made by use of a scale model of the intake The engineers responsible for the design of the pumping station should consult with pump manufacturers to establish one or more intake arrangements A sump model test can then be conducted by an independent laboratory or by the pump manufacturer The sump model test may show modifications of structure or baffling arrangement to be necessary, and sometimes sump model tests show how considerable savings can be made in the intake structure The model should be extensive enough to include all parts of the channel likely to affect the flow near the pump, including screens and gates

Deviations may occur between model and prototype, since all considerations of similarity cannot be produced simultaneously Consequently, the range of levels in velocities to be explored should be as broad as possible in order to disclose any markedly unfavorable tendencies which might only be incipient at mathematically analogous conditions.

Comparable flow in the model is generally considered to be obtained at equal Froude numbers

On this basis,

$$V_m = V_p \times \sqrt{R}$$

where

V_m = Velocity of water in the model
V_p = Velocity of water in the prototype
R = Linear scale ratio of model to prototype

or

$$\frac{L_m}{L_p}$$

where

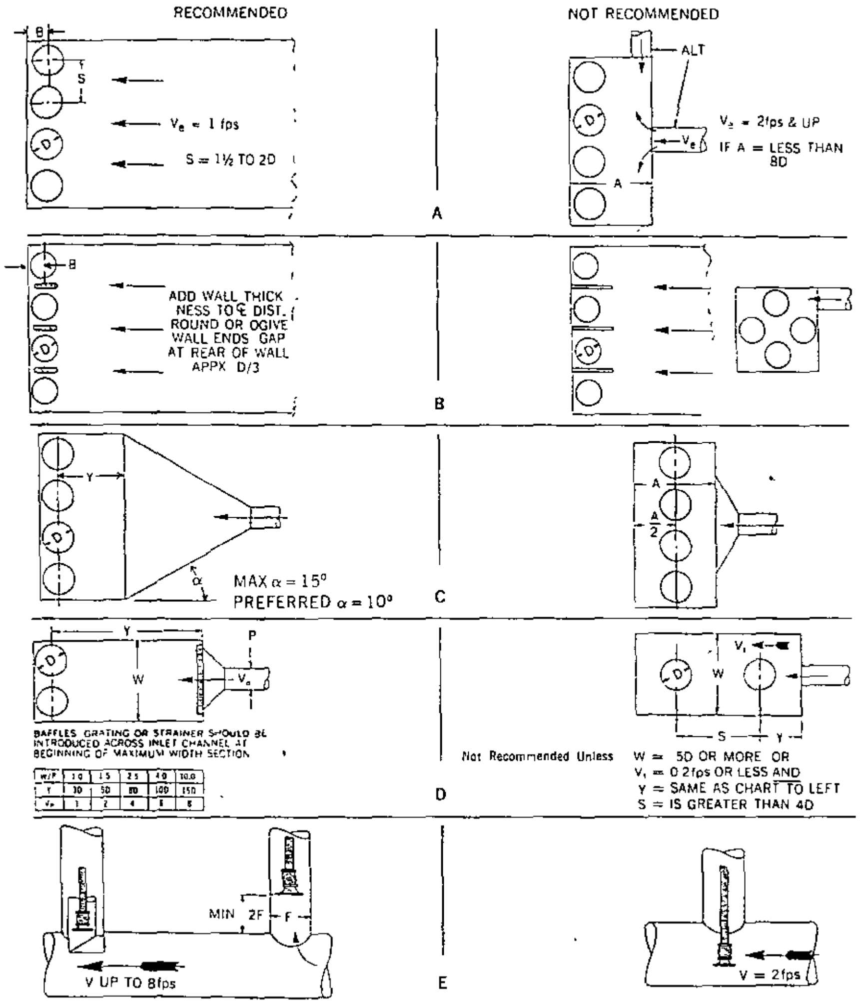
L_m = Any linear dimension of the model
L_p = The dimension on the prototype corresponding to any dimension L_m on the model.

Several investigators have found better agreement between model and prototype when velocities are equal than when velocities are in accord with the Froude number. In the present stage of the art, caution suggests that this entire range of velocities be explored in the model test

Sump or Pit Design (Small Pumps)

The design of sumps for small pumps (less than approximately 3,000 gpm, normal discharge capacity per pump) should be guided by the same general principles as outlined.

However, since there is a large variety of geometric configurations for these small units, recommended



W/P	1.0	1.5	2.5	4.0	10.0
Y	30	50	80	100	150
V_P	1	2	4	6	8

The Dimension D is generally the diameter of the suction bell measured at the inlet. This dimension may vary depending upon pump design. Refer to the pump manufacturer for specific dimensions.

Fig 84 MULTIPLE PUMP PITS

Note. Figures apply to sumps for clear liquid. For fluid-solids mixtures refer to the pump manufacturer.

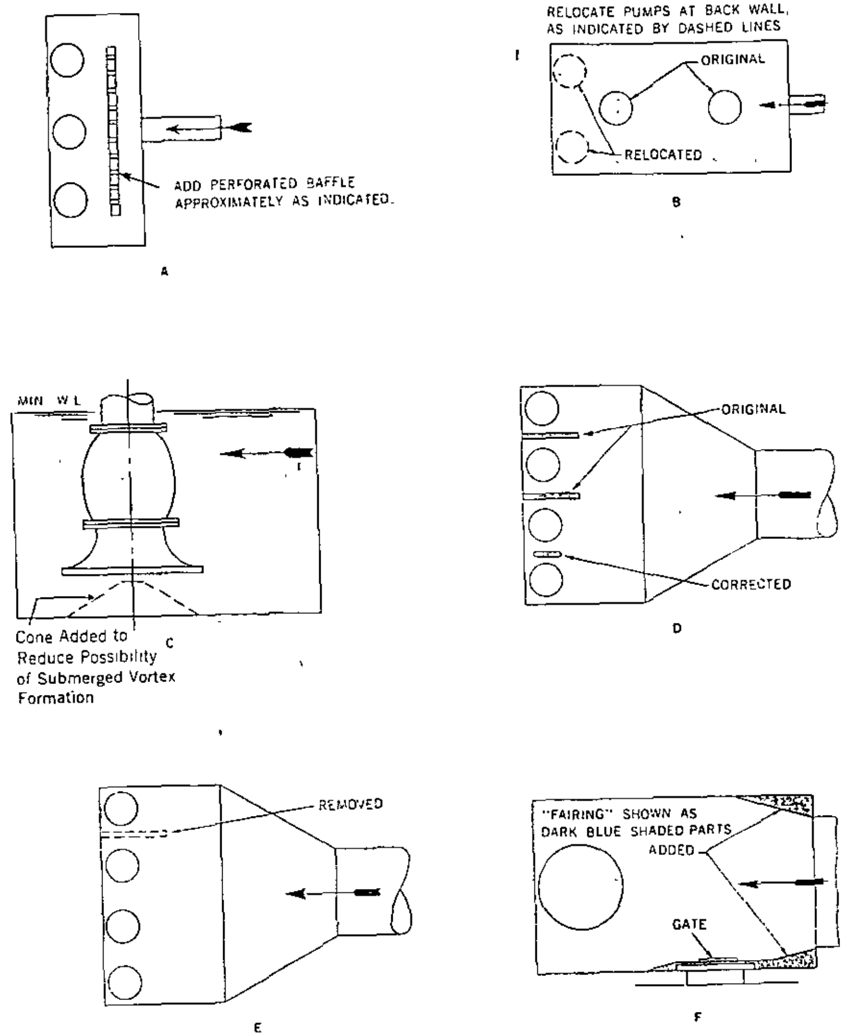
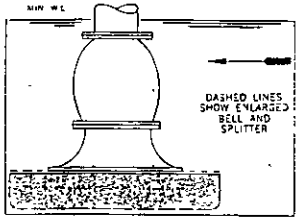


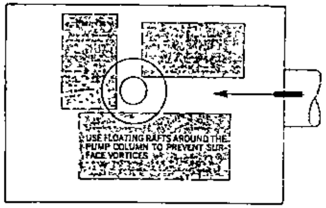
Fig. 85 CORRECTION OF EXISTING SUMPS (PART ONE)

Note. Figures apply to sumps for clear liquid. For fluid-solids mixtures refer to the pump manufacturer.



G

SPLITTER MUST BE IN LINE WITH FLOW
SPLITTER IS TO PREVENT SUBMERGED VORTEXING



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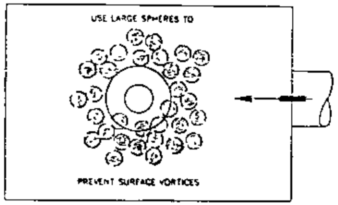
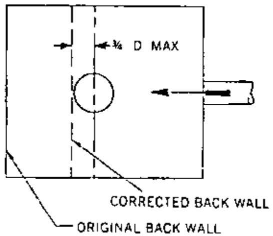


I

IMPROVE VELOCITY PATTERN TO THE PUMP TO REDUCE THE POSSIBILITY OF VORTEX FORMATION



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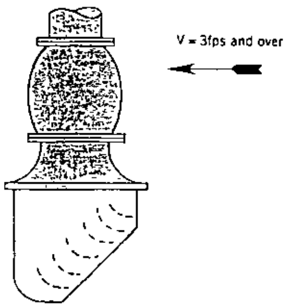

K

Fig. 85 CORRECTION OF EXISTING SUMPS (PART TWO)

Note- Figures apply to sumps for clear liquid For fluid-solids mixtures refer to the pump manufacturer

limiting dimensions, such as shown in Fig. 81–83, cannot be sufficiently generalized and so presented. Where specific pit or sump dimensions are required, the manufacturer's recommendations should be requested

In addition to the general design principles outlined, for single and multiple pump settings in large sump designs, the following factors are pertinent to the design of small sumps or pits

Inlet Opening (Pit Type Sumps)

The sump inlet should be below the minimum liquid level, and as far away from the pump as the sump geometry will permit The influent should not impinge against the pump, jet directly into the pump inlet, or enter the pit in such a way as to cause rotation of the liquid in the pit Where required, a distribution nozzle can be used to prevent jetting, and baffling can be used to prevent rotation

Sump Volume (Pit Type Sumps)

The usable pit volume should equal or exceed the maximum capacity to be pumped in two minutes If units operate on float switch control, pit should be sized to allow no more than three or four starts per hour per pump. These guides generally insure pits of adequate size to dissipate the inflow turbulence and to assure reasonable life of the starting equipment

Minimum Liquid Level

Minimum liquid level should be adequate to satisfy the particular pump design The pump manufacturer's specific dimensions should be used.

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División de Educación Continua, Facultad de Ingeniería, UNAM.

CURSOS ABIERTOS

DISEÑO DE CARCAMOS DE BOMBEO

CA 387

TEMA

ANEXO II

EXPOSITORES: M. EN I. FRANCISCO ORDUÑA BUSTAMANTE
DEL 09 AL 30 DE JUNIO DE 2007
PALACIO DE MINERÍA

Palacio de Minería, Calle de Tacuba No. 5, Primer piso, Delegación Cuauhtémoc, CP 06000, Centro Histórico, México D F ,
APDO Postal M-2285 • Tels· 5521.4021 al 24, 5623.2910 y 5623 2971 • Fax: 5510 0573

To minimize the viscous effects in modeling pumps, the Hydraulic Institute standards (6) recommend that the size of the model be such that the model impeller is not less than 30 cm in diameter. These same standards state that "the model should have complete geometric similarity with the prototype, not only in the pump proper, but also in the intake and discharge conduits."

Even with complete geometric similarity, one can expect the model to be slightly less efficient than the prototype. An empirical formula proposed by Moody that is used for estimating prototype efficiencies of radial- and mixed-flow pumps and turbines from model efficiencies is

$$\frac{1 - e_1}{1 - e} = \left(\frac{D}{D_1}\right)^{1/5} \tag{8-36}$$

where e_1 is the efficiency of the model, and e is the efficiency of the prototype.

EXAMPLE 8-11 A model having an impeller diameter of 45 cm is tested and found to have an efficiency of 85%. If a geometrically similar prototype has an impeller diameter of 1.80 m, estimate its efficiency when it is operating under conditions dynamically similar to those in the model test ($C_{Q\,\text{model}} = C_{Q\,\text{prototype}}$).

SOLUTION We apply Eq. (8-36) with the condition that $e_1 = 0.85$ and $D/D_1 = 4$. Then

$$e = 1 - \frac{1 - e_1}{(D/D_1)^{1/5}}$$

$$= 1 - \frac{0.15}{1.32} = 1 - 0.11 = 0.89$$

The efficiency of the prototype is estimated to be 89%. ■

8-14 Other Types of Pumps

The pumps and turbines we have discussed so far in this chapter are all classified as turbomachines. In turbomachines, the exchange of energy is accomplished by means of hydrodynamic forces developed between a moving fluid and the rotating and stationary parts of the machine. For example, in the axial-flow pump, the lift force of the rotating blades of the impeller produces the pressure increase of the pump.

Another entirely different class of pump is the positive displacement type. All positive displacement pumps have parts that interact in such a way that definite volumes of fluid are conveyed in the desired pumping direction essentially in proportion to the speed of operation of the pump. One of the simplest

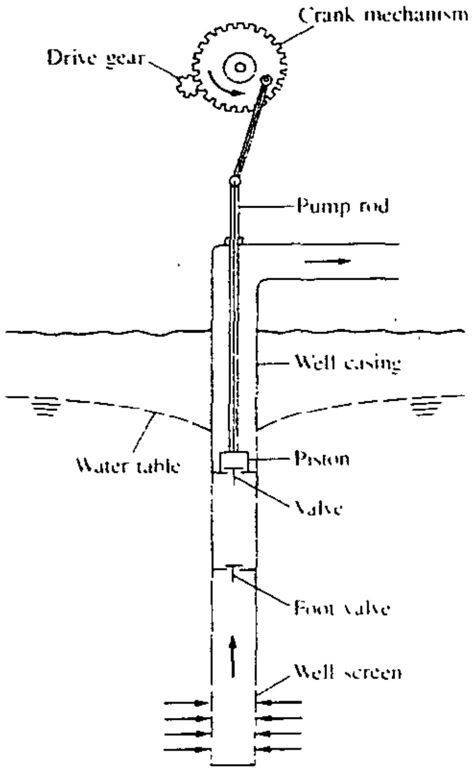


Figure 8-37 Reciprocating piston pump

positive displacement pumps is the reciprocating piston pump shown in Fig. 8-37. In this pump, as the pump rod and piston are raised by the crank mechanism, the valve in the piston is closed so that the piston draws water into the well and up through the well pipe (well casing). On this upstroke, the foot valve remains open. On the downstroke of the piston, the foot valve closes, but the valve in the piston opens Thus one can see that for each cycle of the crank that drives the piston a definite volume of water will be "lifted" from the well and through the outlet pipe. If the speed of the crank is doubled, the rate of pumping would also be doubled (neglecting leakage past seals of the piston). The work required to pump the water can be expressed in terms of the essentially static force applied to the piston to lift the water times the distance through which the piston acts when water is being lifted.

Many other types of positive displacement pumps have configurations different from that of the simple piston pump. Several of these pumps are the *gear pump, two-lobe rotary pump,* and *screw pump.*

Besides the broad categories of turbomachines and positive displacement pumps, *jet pumps* and *hydraulic rams* have limited but important use in special situations.

Descriptions of the aforementioned pumps are given under separate headings below.

Gear Pump

Figure 8-38 is a section through a spur-gear pump. The gears rotate in the direction indicated, and these gears have very close clearance with the casing of the pump. Where the gear teeth contact, they form a tight liquid seal. Thus as the gears rotate, liquid flows in between the gear teeth on the suction side in very much the same way that liquid is drawn into the cylinder of a piston pump when the piston is on the suction stroke. As the gears rotate, the liquid is trapped between the teeth and the casing and is carried around to the discharge side of the pump, where the liquid is forced out as the teeth of the gears mesh together.

Gear pumps are just one class of rotary pumps that are used for pumping various kinds of liquids over a wide range of pressure, viscosities, and temperatures. Several applications of rotary pumps are

1. Chemical processing
2. Food handling
3. Tank truck loading and unloading
4. Machine tool coolants
5. Pressure lubrication
6. Hydraulic power transmission
7. General transfer of liquids

The efficiency depends on the viscosity of the liquid being pumped, but it may be as high as 70% for low viscosity liquids. Rotary pumps can be designed to develop pressures up to 5000 psi, and some have capacities as high as 5000 gpm.

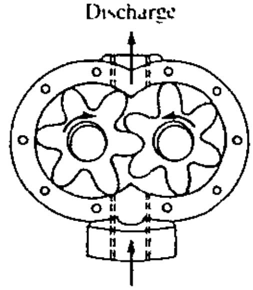
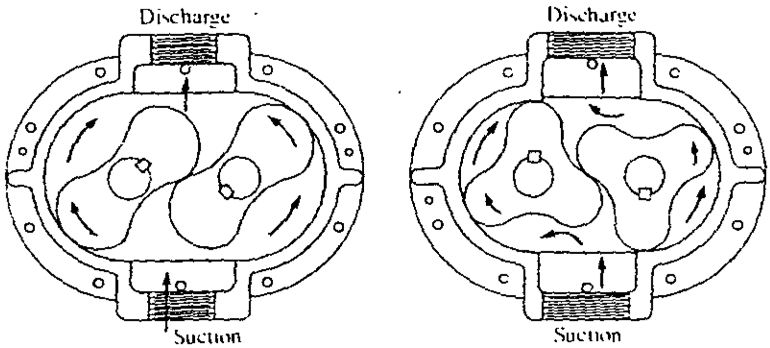


Figure 8-38 Spur-gear pump



(a) Two-lobe rotary pump (b) Three-lobe rotary pump

Figure 8-39 Lobe pumps

Lobe Pumps

Figure 8-39 shows two-lobe and three-lobe rotary pumps. These pumps operate exactly like gear pumps. However, because of the smaller number but larger volume of "chambers" that produce the pumping action, there may be more of a pulsating flow from the lobe pump than from the gear pump.

Screw Pumps

Screw pumps are similar to gear and lobe pumps in that pumping occurs as the elements of the pump rotate and mesh. In the screw pump, the liquid is carried between screw threads on one or more rotors and is displaced axially as the screws rotate.

Disadvantages of Rotary Pumps

The main disadvantages of rotary pumps are their cost is quite high because of the precise machining required to produce close tolerances, and they are not suited for pumping liquids that have abrasives in them. Because of the close clearances in rotary pumps, liquids containing abrasives (such as sand) will usually cause rapid wear of the surfaces.

Jet Pumps

Jet pumps derive their pumping action from a high velocity jet of fluid that then becomes entrained with the fluid it is pumping. The high momentum

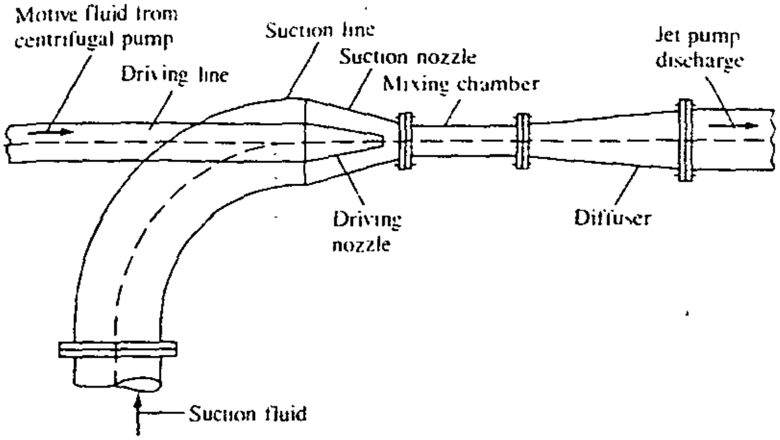


Figure 8-40 Jet pump

of the jet is converted to pressure in a diffuser. Liquid jet pumps are sometimes also called *eductors*. Figure 8-40 shows the essential features of a jet pump. There are many advantages of the jet pump, for example, it is self priming; it has no moving parts; and it can be made from any machinable materials, glass, and fiberglass. The main disadvantage of the jet pump is its relatively low efficiency. The entrainment process inherent in its operation produces large head losses that account for this low efficiency. Despite its low efficiency, it has several uses, including

1. Deep-well pumping
2. Bilge pumping on ships
3. Providing circulation in rearing tanks of fish hatcheries (absence of moving mechanical parts do not injure fish)
4. Chemical processing mixing
5. Pumping out wells, pits, sumps where there is an accumulation of sand or mud

The deep well application is illustrated in Fig. 8-41. The jet pump and centrifugal pump act as a two-stage pumping unit. In the pumping process, the jet pump near the bottom of the well produces enough pressure so that the pressure on the suction side of the centrifugal pump is well above the vapor pressure of the liquid. Thus the centrifugal pump provides the remaining necessary head to yield the desired results. Without the jet pump, the centrifugal pump alone at the surface of the ground would not be able to pump water from a well more than about 30 ft deep because the water would vaporize when the suction pressure reaches the vapor pressure of the water (equivalent to about -33 ft of head at normal temperatures). The jet pump is well suited for this kind of application because it can be designed to be a relatively compact unit that can be easily installed in a well. A typical commercial deep well unit has a 1-in. pressure pipe

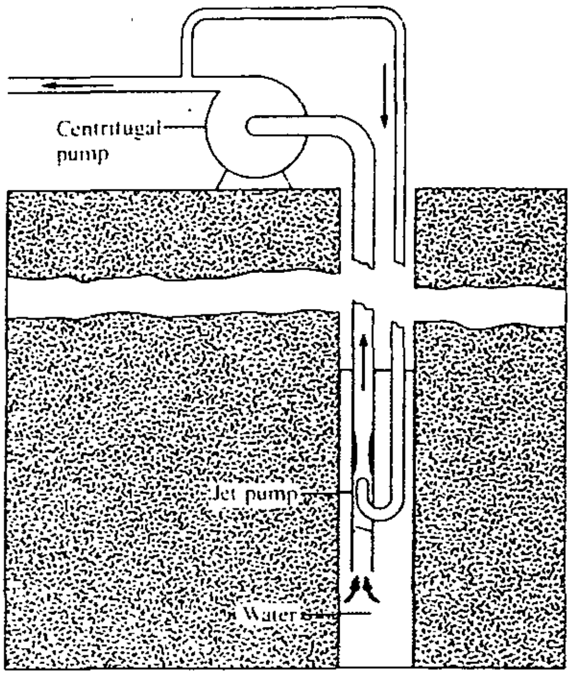


Figure 8-41 Jet pump in combination with a centrifugal pump for pumping water from a well

with a $1\frac{1}{4}$-in. discharge pipe and, depending on the depth of the well, several available nozzle and diffuser combinations (7).

Hydraulic Ram

The hydraulic ram was first developed in England in about 1800. It uses a relatively large flow of water under low head to pump a much smaller amount of water to a much higher elevation. Figure 8-42 shows the essential features of a hydraulic ram. Valve W is the waste valve, and valve C is a check valve. Assuming the cycle of operation starts with zero velocity in the drive pipe with

Table 8-1 Data on Selected Hydraulic Rams in the United States

Location	Discharge (cfs) to ram	Discharge (cfs) to reservoir	Head (ft) drive pipe	Head (ft) pump head	Strokes per minute
U.S. Naval Coaling Station, Bradford, Rhode Island	1.29	0.52	37	84	130
Seattle Water Works, Seattle, Washington	1.63	0.55	49	131	65

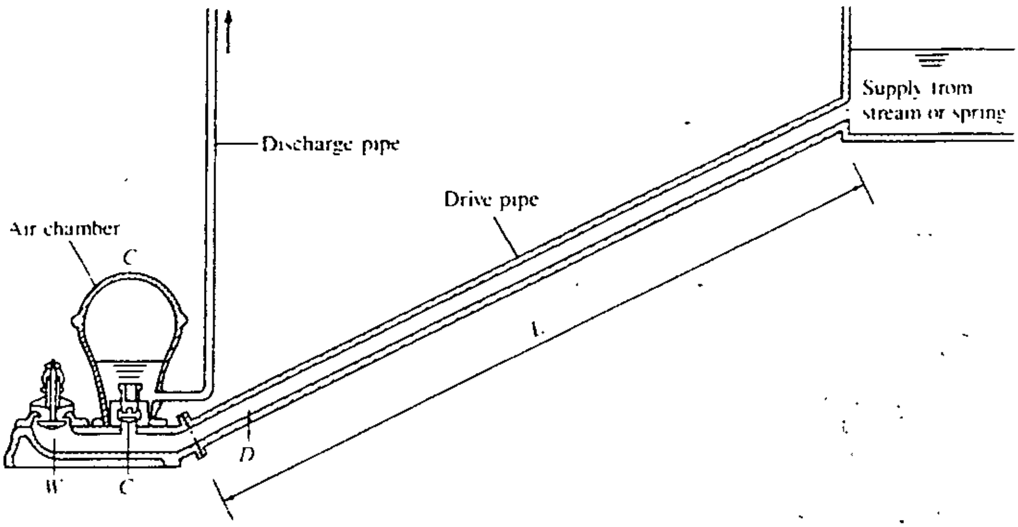


Figure 8-42 Hydraulic ram

valve W open and valve C closed, flow starts past valve W and, because of continuity, water also starts moving in the drive pipe. The flow through valve W and the drive pipe will accelerate until the velocity past the valve is so great that valve W closes quickly. The valve closure is initiated because the drag on the valve overcomes the weight of the valve that tends to keep it open. Once valve W closes, it produces a water hammer pressure in the drive pipe and in the body of the ram. This pressure will be large enough to open the check valve C, and some water will be forced into the air chamber. The pressure in the air chamber increases and further compresses the air. More water goes into the air chamber and, because of this increase of pressure, flow occurs in the discharge pipe. After a short time, the water hammer pressure in the drive pipe and ram subsides so that valve C closes. A short time later, the pressure in the drive pipe and ram is further relieved (the relief wave of water hammer starts) and valve W opens. Once valve W is opened, the cycle repeats itself.

In the early 1900s, many hydraulic rams were used for municipal water supplies as well as for individual farms. Table 8-1 gives data for two of these early rams. A study of some of these early rams (11) indicated that if the drive pipe was three times as long as the pumping head, it would be long enough to develop water hammer pressures to operate satisfactorily.

PROBLEMS

8-1 If the pump having the characteristics shown in Fig. 8-4, page 439, has a diameter of 40 cm and is operated at a speed of 1000 rpm, what will be the discharge when the head is 3 m?