STERLING
SIHI

ENGINEERING FUNDAMENTALS

STERLING FLUID SYSTEMS GROUP

STERLING SIHI ENGINEERING FUNDAMENTALS

The data contained within this book has been compiled from STERLING SIHI data and various other sources.

The purpose of this book is to present some basic technical data as it applies to the STERLING SIHI liquid ring vacuum pumps and compressors.

It is hoped that this book will illustrate the thermodynamic characteristics of the liquid ring pump and, by means of specific examples, will be of aid to those wishing to make basic pump selections from the STERLING SIHI data book or other STERLING SIHI published literature.

NORTH AMERICAN MANUFACTURING PLANTS:

U.S.A.

Sterling Fluid Systems (USA)
303 Industrial Blvd.
P.O. Box 460
Grand Island, NY 14072
Office: (716) 773-6450
FAX: (716) 773-2330
Email: mail@SIHI.com

Canada

Sterling Fluid Systems (Canada) Ltd.
225 Speedvale Ave. W.
P.O. Box 728
Guelph, Ontario N1H 6L8
Office: (519) 824-4600
FAX: (519) 824-7250
Email: mail@SIHI.com

www.sterlingfluidsystems.com

U.S.A.
Sterling Fluid Systems (USA)
Grand Island, N.Y.

Engineering Services, Research & Development, Manufacturer of Standard & Custom Packages and **SIHI**^{dry} Vacuum Pumps

Canada
Sterling Fluid Systems (Canada) Ltd.
Guelph/Ontario

Engineering Services, Research & Development, Manufacturer of Standard & Custom Packages and **SIHI**^{dry} Vacuum Pumps.

Germany
Sterling SIHI GmbH Itzehoe

Engineering Services, Manufacturer of Liquid Ring Vacuum Pumps & Compressors, **SIHI**^{dry} Vacuum Pumps, Air Ejectors, Blowers, Packages, and Research & Development.

CONTENTS

CONTENTS

LIST OF APPENDICES

LIST OF SYMBOLS

ACFM = Actual cubic feet per minute.

BHP = Normal power required when using water as service liquid.

BHP_x = Power required when using other than water as service liquid.

C_p = Specific heat of fluid.

C_f = Condensing correction factor.

C_Q = Correction for service liquid flow when using different viscosity liquid.

C_{SD} = Correction for capacity when using different specific gravity service liquid.

C_{SV} = Correction for capacity when using different viscosity service liquid.

C_{VHP} = Correction for horsepower when using different viscosity service liquid.

C_{HP} = Correction for horsepower required when using different specific gravity service liquid.

cSt = Centistokes

$\dot{H}_{ic}$ = Isothermal compression and friction heat.

$\dot{H}_c$ = Condensation heat.

$\dot{H}_{gc}$ = Gas cooling heat (enthalpy change).

Δh_v = Latent heat of vaporization.

h_{fg} = Enthalpy change for condensed vapor.

$\dot{H}_t$ = Total heat to be removed from system.

K = Kelvin degree

$\dot{m}$ = Weight flow rate.

MW = Molecular weight.

$MW_{avg.}$ = Average molecular weight of a mixture.

P_1 = Initial absolute pressure.

P_2 = Final absolute pressure.

P_{PG} = Partial pressure of dry gas in a mixture.

P_{PV} = Partial pressure of a vapor in a mixture (condensable).

P_t = Total pressure.

P = System pressure rise per unit time.

ΔP = Differential pressure rise in Torr.

$\dot{Q}$ = Flow of service water (USGPM).

$\dot{Q}_x$ = Flow of liquid other than water (USGPM).

$\dot{Q}_L$ = Leakage rate in Torr • L/Sec or Inch Hg • ft^3/Sec.

R = Rankine degree

Note: A period over a symbol is used to denote a rate.

LIST OF SYMBOLS (cont.)

°R = Universal gas constant (ft • lbf/lb • mole • °R).

$\dot{S}_A$ = Actual pumping speed or capacity.

$\dot{S}_g$ = Gas capacity.

$\dot{S}_V$ = Vapor capacity.

$\dot{S}_{avg.}$ = Average pump capacity.

$\dot{S}_{mixt.}$ = Capacity of saturated mixture.

$\dot{S}_{th}$ = Theoretical pumping speed or capacity.

$\dot{S}_{DA}$ = Listed pump capacity from data book based on 15°C (60°F) service water, 20°C (68°F) dry air.

$\dot{S}_x$ = Pump capacity with service liquid other than water.

$\dot{S}_1$ = Capacity at initial conditions.

$\dot{S}_2$ = Capacity at final conditions.

SCFM = Standard cubic feet per minute.

ACFM = Actual cubic feet per minute at operating temperature & pressure.

T_1 = Initial absolute temperature.

T_2 = Final absolute temperature.

T_m = Temperature of gas and liquid mixture leaving the pump.

T_c = Condensing temperature rise.

T_f = Final temperature.

t_{ev} = Evacuation time in minutes.

t_L = Lapsed time in seconds.

V_A = Volume of impeller cells.

V_S = System volume.

V_1 = Volume at initial conditions.

V_2 = Volume at final conditions.

W = Weight.

$\eta_{vol.}$ = Volumetric efficiency.

μ_w = Viscosity of water.

μ_x = Viscosity of liquid other than water.

∂_w = Specific gravity of water.

∂_x = Specific gravity of liquid other than water.

$\propto$ = Capacity reducing factor.

Note: A period over a symbol is used to denote a rate.

Section One 1

STERLING SIHI LIQUID RING VACUUM PUMPS & COMPRESSORS

1. FEATURES/BENEFITS/APPLICATIONS

STERLING SIHI liquid ring vacuum pumps are rotary displacement pumps of simple and durable construction that have found wide application in many fields.

STERLING SIHI liquid ring vacuum pumps and compressors have the following features:

- Reliable, low maintenance, and safe operation
- Low noise and vibration
- Practically isothermal, hence safe cool compression of flammable vapors and gases
- Capable of handling almost any gas and/or vapor
- Saturated gases can be pumped without difficulty
- Liquid carryover can be handled
- No sliding contact
- Can deliver oil free gases
- Available in a variety of materials to handle most applications

As a result of the above features, the STERLING SIHI liquid ring vacuum pumps and compressors are widely used in industry for operations such as drying, distilling, condensing, evaporating, flushing, handling corrosive and explosive gases, evacuating systems, and compressing oil free air for medical breathing and instrument air amongst others.

TYPICAL STERLING SIHI LIQUID RING VACUUM PUMPS

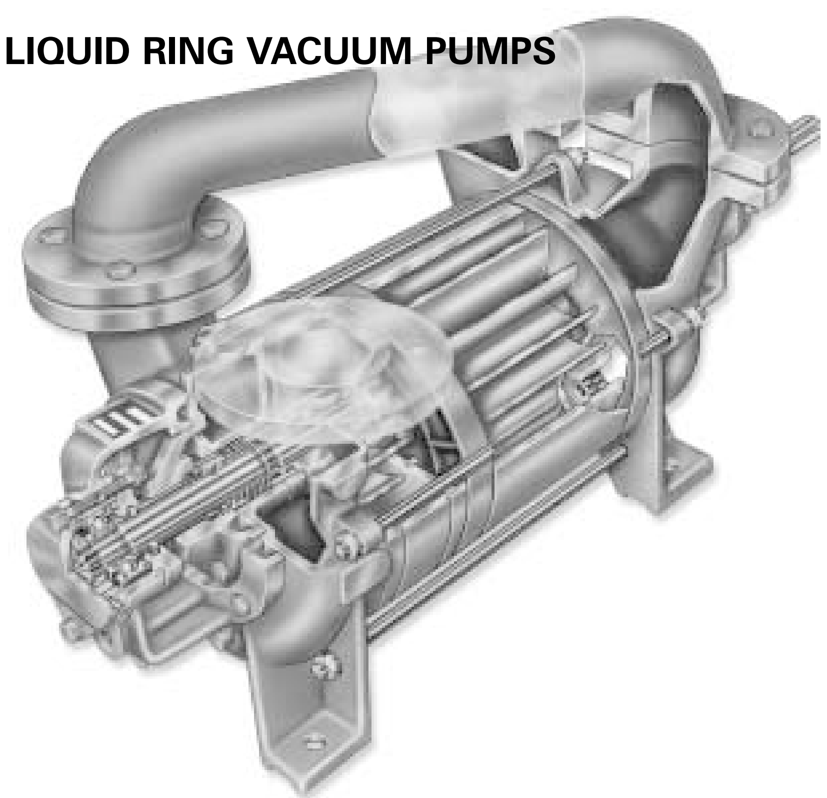


FIGURE 1 SECTION VIEW OF A STERLING SIHI TWO STAGE PUMP

FIGURE 2 TYPICAL STERLING SIHI PRODUCTS

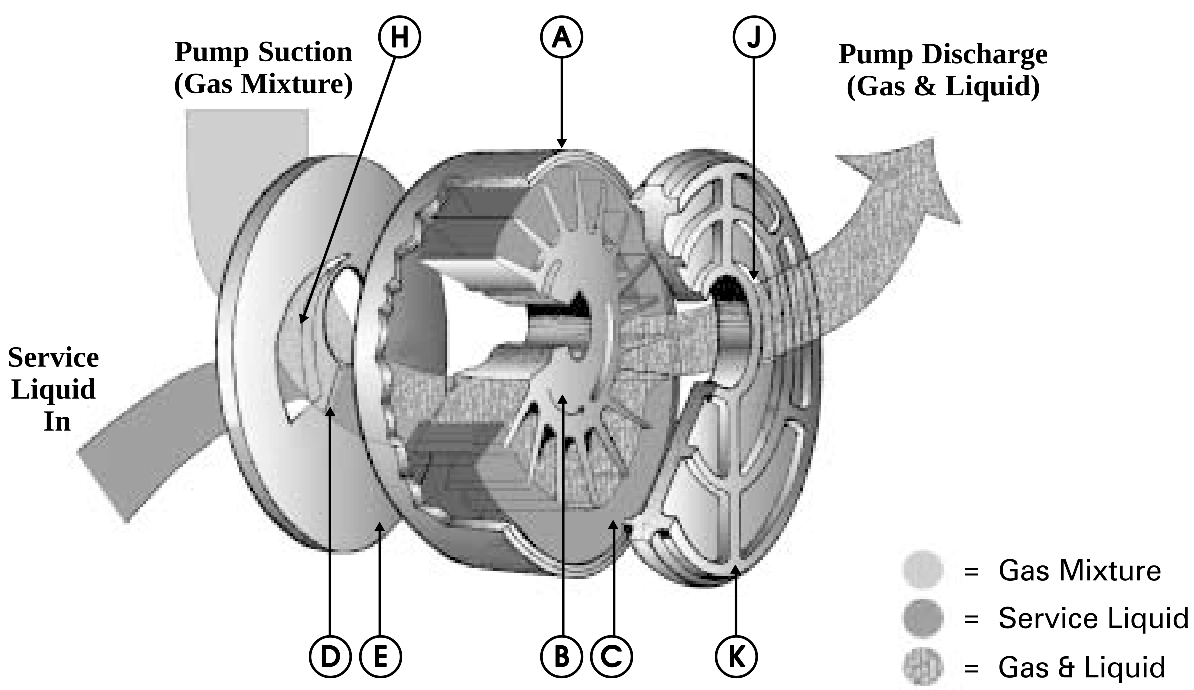
2. WORKING PRINCIPLE OF A STERLING SIHI SINGLE ACTING PUMP

In a round pump body (**A**), a shaft mounted impeller (**B**) is positioned at a point eccentric to the center-line of the pump body. The centrifugal action of the rotating impeller forces the service liquid introduced via channel (**D**) towards the periphery of the pump body forming the liquid ring (**C**).

When pumping action is achieved, the gas mixture being handled is introduced to the impeller through the suction port (**H**), in the intermediate plate (**E**), causing a vacuum at the pump suction. The gas mixture fills the impeller cavity between the inside diameter of the liquid ring and the root of the impeller blade. As the impeller rotates, the impeller blade immersion in the liquid ring increases reducing the volume between the liquid ring and the root of the impeller blade. The result is the compression of the gas mixture until it reaches the discharge port (**J**), located in the intermediate plate (**K**). The gas mixture exits through the discharge port.

During the compression cycle heat is being imparted to the liquid ring. In order to maintain a temperature below the vapor point of the service liquid, cooling must be applied. Cooling is achieved by continuously adding a cool supply of service liquid to the liquid ring. The amount of service liquid added is equal to that discharged through the discharge port (**J**) together with the compressed gas mixture. The gas mixture and service liquid is eventually passed through the pump discharge for separation.

FIGURE 3 EXPLODED VIEW OF A STERLING SIHI SINGLE ACTING PUMP



★ *These illustrations are intended to depict the operating principle of the Sterling SIHI liquid ring pumps only and should not be considered for engineering details of construction.*

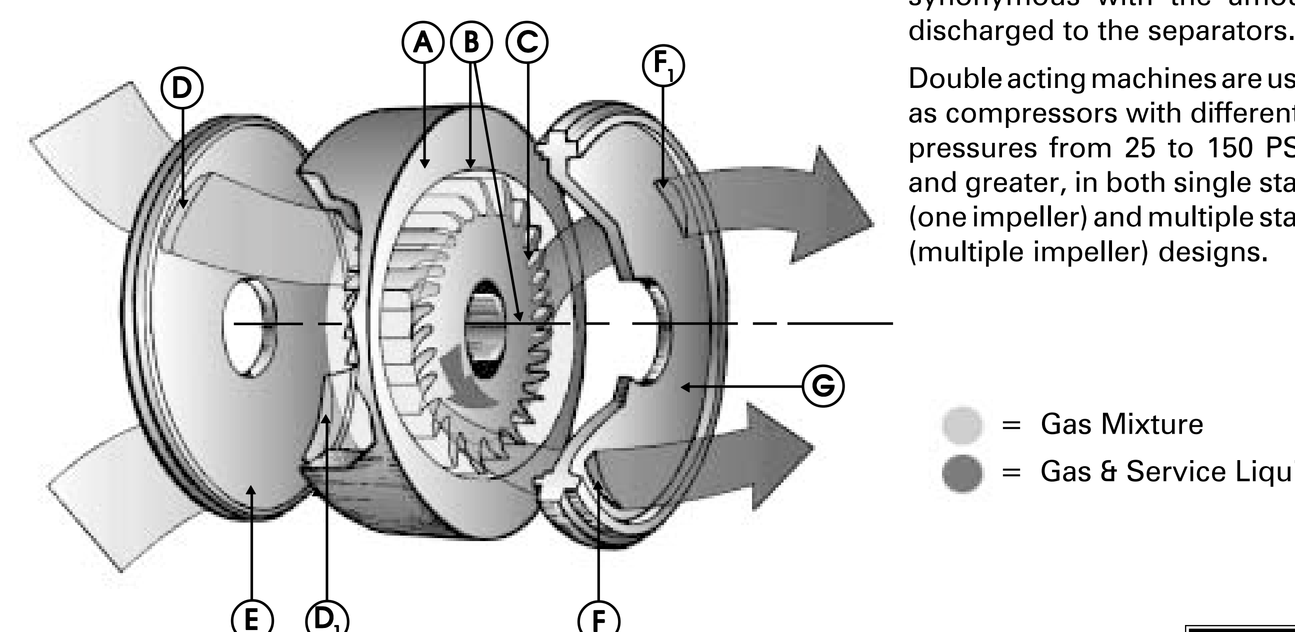
3. WORKING PRINCIPLE OF A STERLING SIHI DOUBLE ACTING COMPRESSOR

As in the STERLING SIHI single acting pump, the pump body (A) is round externally. However, note that the internal periphery of the body is elliptical (B) the center of which coincides with the center line of the shaft mounted impeller (C). Due to centrifugal action, service liquid introduced to the compressor assumes the elliptical shape of the internal casing. By controlling the depth of the service liquid, the impeller blades are totally immersed at six and twelve o'clock, whereas all but the blade tips are exposed at three and nine o'clock during each revolution. Half of the total gas (or vapor) entering the stage enters through suction port (D) at which point the service liquid is receding from the root of the passing impeller blades. This gas is carried between the impeller blades and as the service liquid (due to elliptical shape) commences to completely immerse the impeller blades to their root, the gas is compressed and discharged from the pump via discharge port (F).

Concurrently, a similar action takes place involving the remaining fifty percent of the gas through suction port (D_1) and discharge port (F_1). By locating the suction ports (D), (D_1) in a strategic position in the suction port plate (E) and the discharge ports (F), (F_1), in the discharge port plate (G), a suction/compression cycle is completed with each 180° of rotation. Since the points of highest pressure are diametrically opposed, radial shaft forces are balanced. Hence the double acting principle is used for high pressure compressors to reduce shaft deflection and increase mechanical seal life. There is no metal to metal contact during this cycle, thus the need for internal lubrication is eliminated. During the compression cycle, heat is being imparted to the service liquid which is carried away by the introduction of additional cool service liquid. The amount of coolant supplied is synonymous with the amount discharged to the separators.

Double acting machines are used as compressors with differential pressures from 25 to 150 PSIG and greater, in both single stage (one impeller) and multiple stage (multiple impeller) designs.

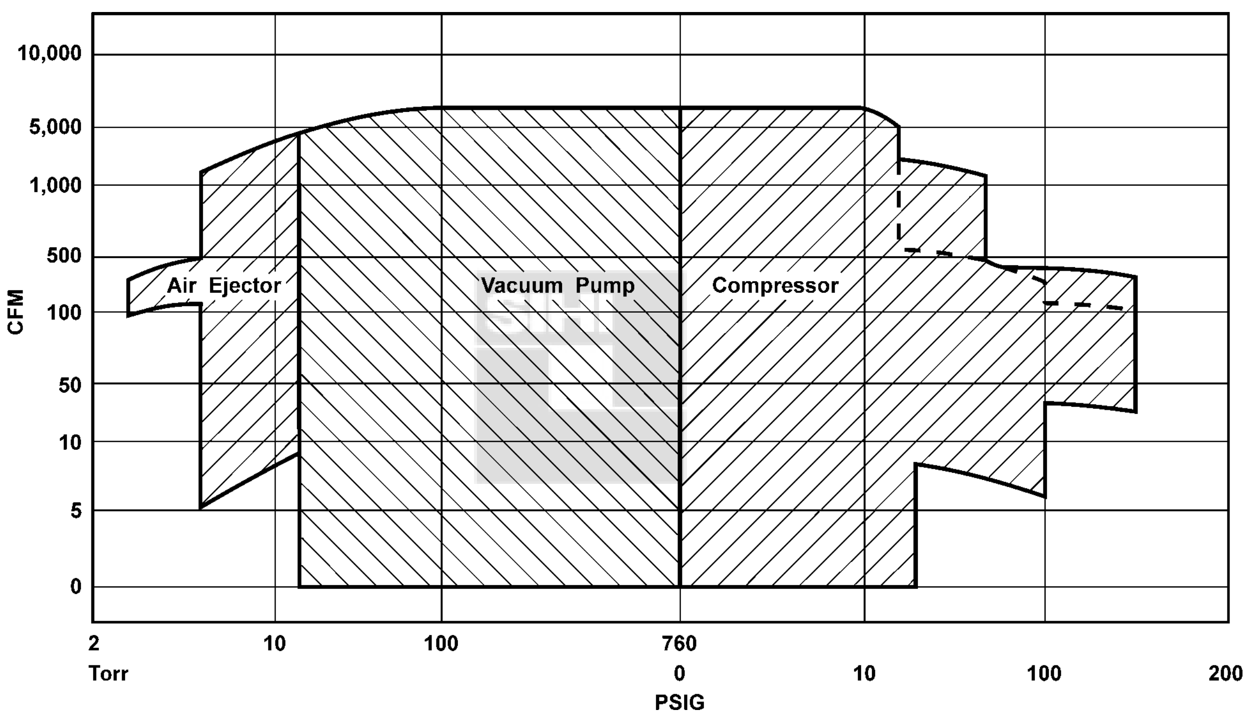
FIGURE 4 EXPLODED VIEW OF A STERLING SIHI DOUBLE ACTING COMPRESSOR



4. CAPACITY AND RANGE OF OPERATION

STERLING SIHI vacuum pumps are capable of 25 Torr (1" Hg Abs). STERLING SIHI atmospheric air ejectors can decrease this to about 3 Torr.

FIGURE 5 below illustrates the present STERLING SIHI capacity range.



Note: Products outside shaded area are available (consult factory).

FIGURE 5 CAPACITY VS PRESSURE RANGE FOR VARIOUS STERLING SIHI EQUIPMENT

Auxiliary equipment such as rotary lobe blowers and steam jets can extend this range to 1 Torr or less. Discharge pressures from atmospheric to approximately 25 PSIG are attained with single stage, single acting pumps.

Higher pressures require the use of double acting multi-stage compressors.

The lowest suction pressure attainable with the liquid ring pump is a function of the physical properties of the service liquid. If water at 60°F (15°C) is being used as service liquid, the continuous suction pressure of 25 Torr (1 inch Hg Abs) is easily obtained with most STERLING SIHI two-stage models.

Lower suction pressures can be achieved by using service liquids with lower vapor pressures (oils, certain hydrocarbons, etc.) or by installing other equipment such as steam ejectors with after condenser, air ejectors, rotary lobe pumps, or combinations thereof in series with the liquid ring pump.

As previously noted, single stage pumps are normally selected if suction pressures to 100 Torr with atmospheric discharge pressure are desired. If inlet pressures lower than 100 Torr are needed, a two-stage vacuum pump can be employed. It must also be remembered that a two-stage vacuum pump is also advantageous when using service liquids with high vapor pressures (less capacity loss), when higher than atmospheric discharge pressures are desired, and when handling gases which are soluble in the service liquid.

In some applications air ejectors are also used with the two-stage pump for pressures of 15 to 60 Torr or to allow operation with high vapor pressure liquids.

FIGURE 6 PERFORMANCE CURVES FOR LPH SINGLE STAGE PUMPS

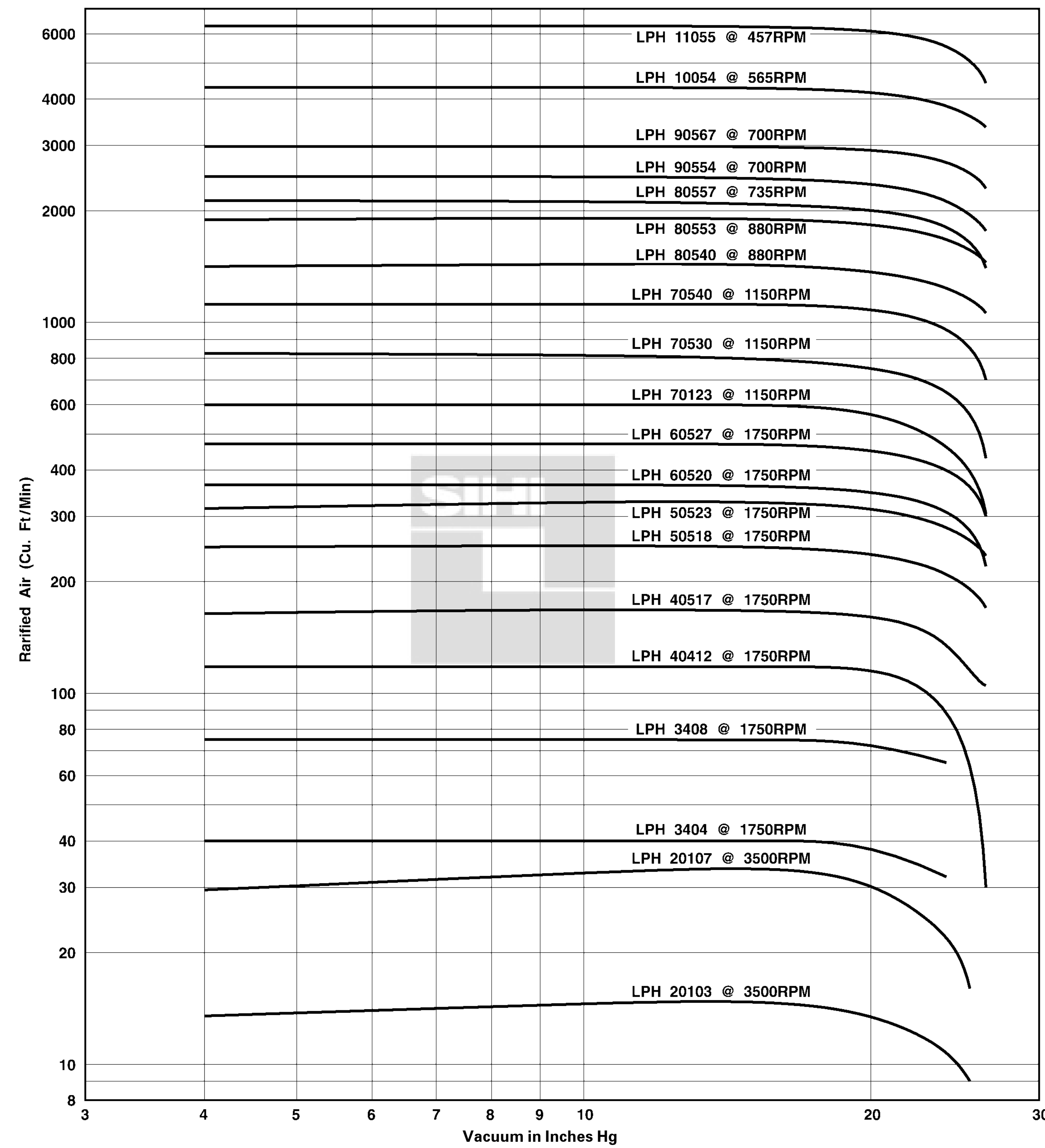
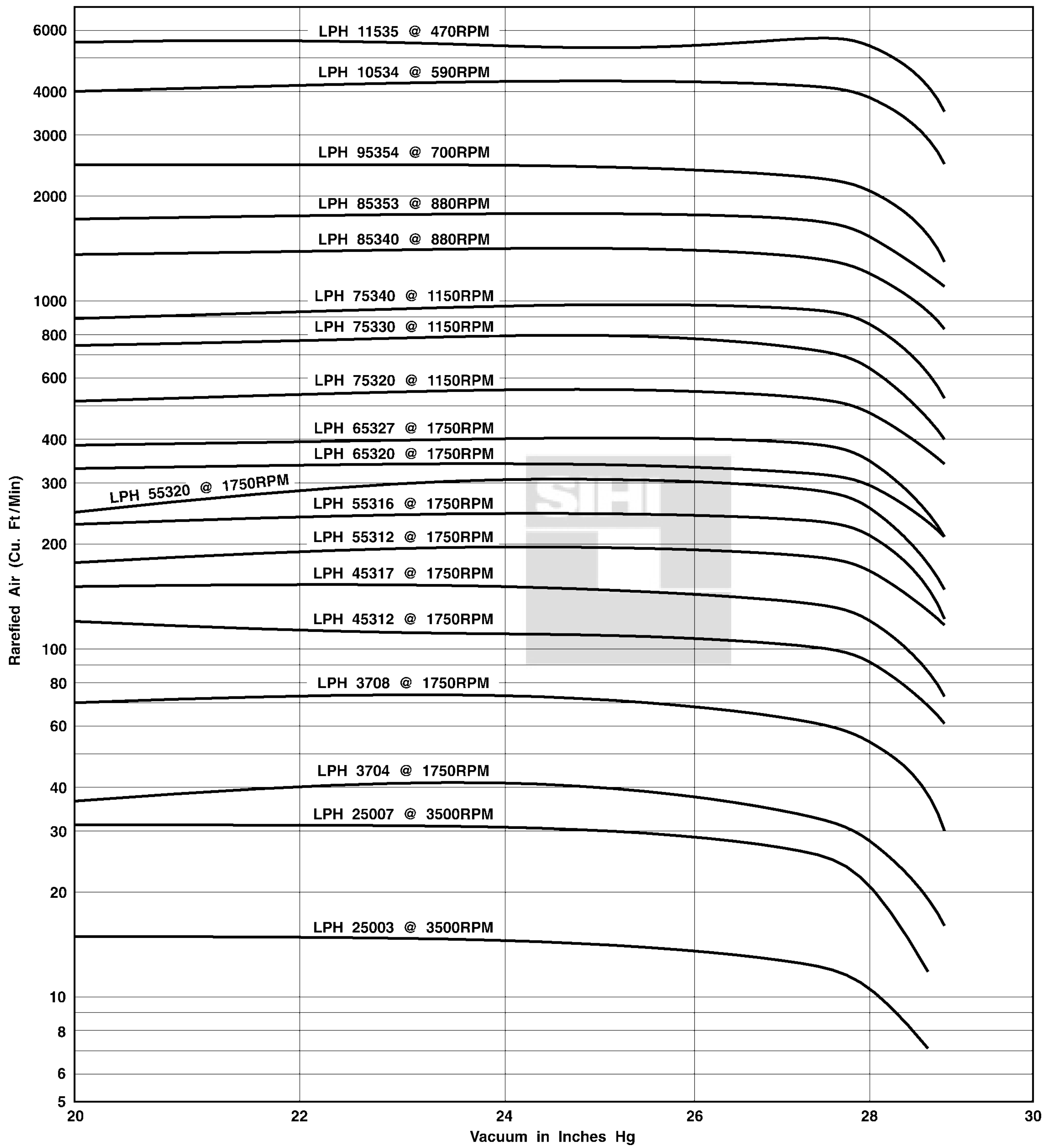


FIGURE 7 PERFORMANCE CURVES FOR LPH TWO STAGE PUMPS

PERFORMANCE CURVES FOR SIHI TWO STAGE PUMPS



Note: For Single Stage High Vacuum Pumps (LEM & LEH contact factory).

5. INDUSTRIAL APPLICATIONS OF STERLING SIHI LIQUID RING VACUUM PUMPS AND COMPRESSORS

BATTERY MANUFACTURE
- Vacuum drying of plates

BOTTLING EQUIPMENT
- Filling of bottles
- Air drying of bottles
- Air cleaning of bottles

BRICK & TILE MANUFACTURERS
- De-aeration of clay in extruders

CANDY
- Vacuum cooking
- Flash cooling by vacuum

CHEMICALS
- Distillation and evaporation
- Solvent recovery
- Vacuum stripping

COFFEE
- Manufacture instant coffee - vacuum distillation
- Oil free air
- Vacuum packaging

COSMETICS
- Bottling
- Vacuum distillation

DAIRY EQUIPMENT
- Compressed air - aeration and agitation
- Vacuum deodorizing
- Evaporated and powdered milk
- Container filling (see bottling)
- Milking machinery

ELECTRICAL EQUIPMENT INDUSTRY
- Transformer filling
- Coil impregnation
- Turbine and gland exhaust

EXPLOSIVES
- Vacuum transfer of liquids
- Vacuum de-aeration of solutions
- Vacuum drying
- Vacuum filters
- Handling of explosive gases and vapors

FILM MANUFACTURE & PROCESSING
- Oil free air for drying and handling vacuum processes in film manufacturing

FILTERS
- Vacuum filters used in manufacture of fertilizers, foods, chemicals, and ore processing

FISH PROCESSING
- Vacuum deodorizing
- Vacuum drying of fish meal
- Vacuum flash cooling
- Vacuum eviscerating
- Vacuum pumping of live fish

FOOD PRODUCTS
- (see bottling applications)
- Deodorizing of product
- De-aeration of product
- Drying, cooking, distillation
- Oil free air for agitation, cleaning etc.
- Vacuum canning & packaging
- Meat & poultry processing
- Steam sterilization of vacuum dryers

GLASS PRODUCTS
- Clean air for coating mirrors
- Vacuum holding of glasses & bottles during manufacturing
- Vacuum lifting of plate glass
- Clean air for lens manufacture
- Vacuum chucking
- Mold degassing

HOSPITAL AND MEDICAL
- Vacuum for sterilizers
- Hospital vacuum systems
- Compressed air for surgical instruments
- Compressed air for patient treatment

LABORATORIES
- Vacuum for research in university and industrial labs

MARINE
- Vacuum & condenser exhaust (see Thermal Power Plants)
- Vapor recovery (barge unloading)
- Vacuum sewage systems
- Vacuum priming of pumps

INVESTMENT CASTING & DIE CASTING
- Vacuum curing of plaster molds
- Removal of air from dies and molds

5. VACUUM PUMPS AND COMPRESSORS - APPLICATIONS (CONT.)

OILS - VEGETABLE
- Vacuum deodorizing
- Differential distillation of oils
- Vacuum transport of product
- Oil free air for agitation, etc.
- Hydrogen compression

PETROLEUM INDUSTRY
- Flue gas CO_2 recovery
- Vacuum filling and cleaning
- Vacuum filters for dewaxing
- Vacuum priming of pumps
- Recovery of light ends - oil ring compressors
- Vapor recovery
- Well point evacuation

PHARMACEUTICALS
- Instrument air
- Vacuum stripping, vacuum cooling, drying etc.

PLASTICS
- Vacuum molding
- De-aeration of mixers and extruders
- Handling gases such as vinyl chloride
- Vacuum handling of sheets
- Reactor evacuation
- Vacuum sizing of extruded products

PLATING
- Air agitation of solutions
- Compressed air for water removal from parts
- Vacuum chucking

POULTRY PROCESSING
- Eviscerating
- Packaging
- Drying of egg products

PRINTING
- Vacuum handling of paper & folding (especially envelopes)

PULP AND PAPER
- Vacuum for removal of moisture on paper machines

RUBBER PRODUCTS
- (see Chemicals)
- De-aeration of liquid rubber and butyls
- Removal of steam from molds
- Drying of tire cords (textile)
- Vacuum holding
- Oil free air for instruments

SOAP MANUFACTURE
- (See Chemicals and Bottling)
- Packaging applications
- De-aeration of soap prior to molding

SUGAR REFINING
- De-aeration, evaporation, filters and crystallizers,
- CO_2 compressors

TEXTILES
- Many applications for blowing, drying, and de-aeration for dyeing

TOBACCO
- Vacuum drying
- Vacuum packaging
- Humidification

THERMAL POWER PLANTS
- Condenser evacuation
- Water de-aeration and degassing
- Turbine gland exhausters
- Priming centrifugal pumps

TRANSPORTATION INDUSTRIES
- Evacuation of chemical tankers
- Solvent vapor recovery from barges/rail cars

WATER/SEWAGE TREATMENT
- Flue gas compressors for CO_2
- Air agitation
- Vacuum distillation of sea water
- De-aeration
- Priming pumps

WIRE
- Vacuum coating of wire with insulation

WOOD
- Vacuum impregnation
- Vacuum handling of plywood

Section Two

BASIC GAS LAW CALCULATIONS

1. CONVERSION OF DATA FOR USE WITH IDEAL GAS LAWS

The capacities of STERLING SIHI vacuum pumps are stated in ACFM, handling dry air at 68°F at the pump operating inlet pressure, using water at 60°F as the service liquid. Discharge pressure on all standard curve data is sea level atmospheric (29.92" Hg Abs or 760 Torr).

STERLING SIHI compressor curves, on the other hand, follow the normal industry practice of stating capacity in SCFM at standard temperature and pressure, (inlet pressure 29.92" Hg Abs, inlet temperature 68°F) and discharge pressure as required.

In the majority of applications, potential customers will provide data for selections under different conditions. In these cases, it is necessary to first convert the data to curve conditions. The Ideal Gas Laws are utilized to perform the conversions.

To utilize gas laws, all data must be in absolute units. Typical absolute units employed are listed below.

Absolute Units:

Pressure: Torr (mm Hg Abs), inches Hg Abs, pounds per square inch absolute (PSIA), KPa, bar and millibar.

Temperature: Rankine (°R) or Kelvin (K)

STP - standard temperature and pressure conditions are:

520°R (60°F) and 29.92" Hg Abs in English units
288K (15°C) and 760 Torr (760 mm Hg) in SI units

Note: Standard temperature is 60°F (15°C) in North America and 32°F (0°C) in Europe. If using 0°C, one pound mole of gas occupies 359 cubic feet compared to 379 cubic feet at 15°C (see page 13).

Temperature Conversions

The Ideal Gas Laws require use of absolute temperature scales. The two scales used are the Kelvin and the Rankine scales.

1 Kelvin degree = 1 Celsius degree
1 Rankine degree = 1 Fahrenheit degree

i.e., The size of each unit of the absolute scales is the same as its corresponding normal scale unit.

However, to convert from standard temperature readings, the difference between the zero points of the standard and absolute scales must be added to the standard reading.

0° Celsius = 273 Kelvin
0° Fahrenheit = 460° Rankine

Example 1: Convert 15°C to Kelvin
Kelvin = 15 + 273 = 288 K

Convert 60°F to °Rankine
Rankine = 60 + 460 = 520°R

Conversion of Vacuum Units to Absolute Pressure Units

Vacuum is a negative gauge pressure, usually referenced to the existing standard barometric pressure where the equipment will operate. This means vacuum is a differential reading between the surrounding atmospheric pressure and the pressure in the system evacuated. In all instances when given a vacuum condition, the question should be asked, at what elevation the pump will operate since the barometric pressure varies with altitude above or below sea level.

To convert vacuum units to absolute units simply use the formula:

Absolute Pressure = Actual Barometric Pressure - Vacuum

Example 2: Convert 20" Hg vacuum to absolute pressure assuming the pump will operate at sea level. The absolute barometric pressure at sea level is 760 Torr or 29.92" Hg Abs

Absolute pressure = 29.92 - 20 = 9.92" Hg Abs

Note: Any Torr or mm Hg pressure reading can be converted to inches Hg by dividing by 25.4.

1 inch Hg = 25.4 mm Hg (Torr)

Normally, vacuum is not given in Torr or millimeters of mercury. Torr is defined by convention to be an absolute unit. However, in some instances, people unfamiliar with these conventions will provide data such as 508 Torr vacuum or 508 mm Hg vacuum.

If the vacuum unit was incorrectly given in terms of Torr or mm Hg vacuum, we can convert to absolute pressure by using the above formula and noting that the barometric pressure at sea level is 760 mm Hg Abs (760 Torr).

Hence, absolute pressure corresponding to 508 mm Hg vacuum equals

760 - 508 = 252 mm Hg Abs (252 Torr).

2. BAROMETRIC PRESSURE CORRECTIONS FOR ALTITUDE

The absolute pressure (barometric pressure) decreases with altitude, hence, if vacuum levels are given at altitude, conversion to absolute pressure must be done as shown above, using the barometric pressure at the site rather than at sea level. Refer to Figure 8 to obtain the expected barometric pressure at elevations higher than sea level.

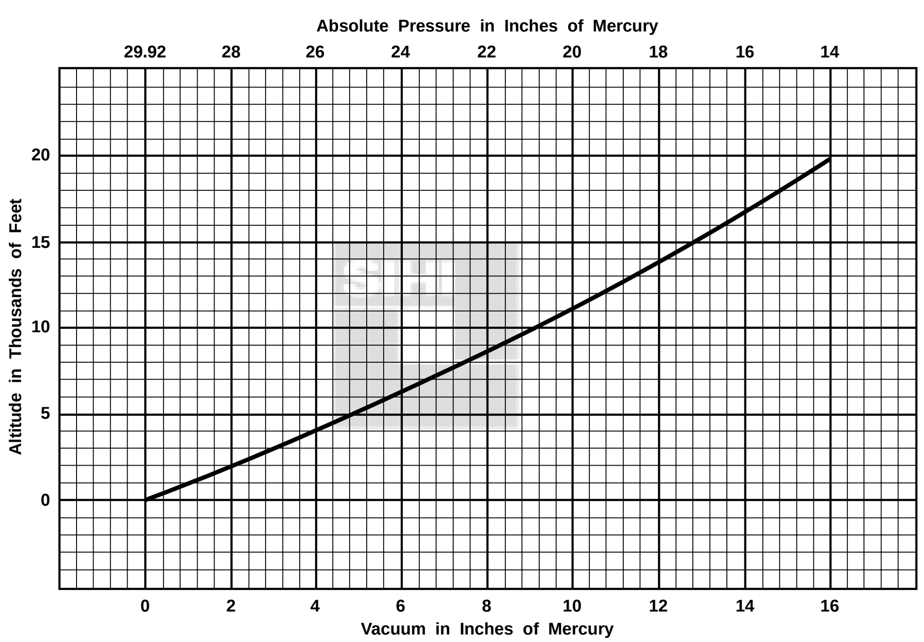


FIGURE 8 BAROMETRIC PRESSURE RELATIVE TO ALTITUDE

3. IDEAL GAS LAWS

The Ideal Gas Laws are used to convert between different pressures and temperatures and to convert mass flows to volume flows as summarized in the following sections.

Boyle's Law: If a unit volume of gas is expanded or compressed without change in temperature, the absolute pressure will vary inversely with the volume.

$$\frac{P_1}{P_2} = \frac{V_2}{V_1}$$

Charles' Law: If pressure is held constant during expansion or compression of a gas, its volume will vary directly as the absolute temperature.

$$\frac{V_1}{V_2} = \frac{T_1}{T_2}$$

and if volume is kept constant, the pressure will vary directly as the absolute temperature.

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}$$

General Gas Law: The combination of Charles' and Boyle's Law yields the more useful general equation.

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

Where:

V_1 = Volume at condition 1, usually standard temperature and pressure (STP)

V_2 = Volume at conditions specified

P_1 = Barometric pressure at condition 1, usually sea level (29.92" Hg Abs)

P_2 = Design operating pressure

T_1 = Temperature at condition 1, usually standard temperature (520°R)

T_2 = Design operating temperature

This formula is useful to correct for both temperature and pressure.

Example 3: Determine the actual volume ACFM of 10 SCFM (volume at standard pressure and temperature) when its temperature is 100°F and expanded to 2" Hg Abs.

$$\frac{29.92 \times 10}{(460 + 60)} = \frac{2 \times \dot{S}_2}{(460 + 100)}$$

$$\therefore \dot{S}_2 = 161 \text{ ACFM}$$

Note: $\dot{S}$ = A volume rate of flow in cubic feet per minute (CFM).

Avogadro's Law: Where the gas load is given by weight flow rather than volume flow, Avogadro's Law is used. Avogadro's Law states one pound mole of any gas when at standard conditions of temperature and pressure (60°F or 520°R and 29.92" Hg Abs) occupies 379 cubic feet. In SI units, Avogadro's Law states one gram mole of any gas when at standard conditions of temperature and pressure (0°C or 273 K and 1.013 Bar) occupies 22.41 liters.

Note: In countries using the SI system of measurement, one pound mole of gas occupies 359 cubic feet at 0°C (273 K).

Therefore, the gas specific volume is given by:

$$\frac{379}{MW}$$

Since the molecular weight of air is 29,

Air specific volume is $\frac{379}{29} = 13.07$ cu. ft/lb

Should the gas flow rate be given in lb/hr ($\dot{m}$), the ACFM will be as follows:

(Avogadro's Law) $\text{SCFM or } \dot{S}_1 = \frac{\dot{m}}{60} \times \frac{379}{MW}$

(using general gas law formula) $\frac{\dot{S}_1 \times P_1}{T_1} = \frac{\dot{S}_2 \times P_2}{T_2}$

$$\therefore \text{ACFM} = \dot{S}_2 = \frac{\dot{S}_1 \times P_1 \times T_2}{P_2 \times T_1} = \frac{\dot{m} \times 379 \times P_1 \times T_2}{60 \times MW \times P_2 \times T_1}$$

Example 4: Find the actual volume of 45.9 lb/hr dry air when heated at 100°F and expanded at 2" Hg Abs

$$\dot{S}_1 = \frac{45.9}{60} \times \frac{379}{29} = 10 \text{ SCFM}$$

$$\dot{S}_2 = \frac{10 \times 29.92 \times (460 + 100)}{2 \times (460 + 60)} = 161 \text{ ACFM}$$

4. GAS MIXTURES

To calculate the volume of gas mixtures, it is necessary to calculate the average molecular weight ($MW_{avg.}$) or as an alternative calculate the number of moles of each gas then total these and use the general gas law with Avogadro's Law to obtain the ACFM.

Calculation Of The Average Molecular Weight ($MW_{avg.}$)

$$MW_{avg.} = \frac{W_t}{\frac{W_1}{MW_1} + \frac{W_2}{MW_2} + \cdots + \frac{W_n}{MW_n}}$$

Where: W_t = Total Gas Weight

$W_1 \ldots W_n$ = Weight of each Component

$MW_1 \ldots MW_n$ = Molecular Weight of each Component

Knowing the average molecular weight, we can then proceed as if it were a single dry gas.

Example 5: 11 lb/hr of gas to be handled and composed of:

3 lb/hr Air	MW = 29
1 lb/hr H_2O	MW = 18
5 lb/hr O_2	MW = 32
2 lb/hr N_2	MW = 28

gas temperature 100°F.
operating pressure 2" Hg Abs

$$MW_{avg.} = \frac{11}{\frac{3}{29} + \frac{1}{18} + \frac{5}{32} + \frac{2}{28}} = 28.4$$

$$\text{SCFM} = \frac{11}{60} \times \frac{379}{28.4} = 2.45$$

$$\therefore \text{ACFM} = 2.45 \times \frac{29.92}{2} \times \frac{(460 + 100)}{(460 + 60)} = 39.5$$

Alternatively by finding the sum of the molar flows of all the gases present:

Air lb mole/hr $= \frac{3}{29} = 0.1034$

H_2O lb mole/hr $= \frac{1}{18} = 0.0556$

O_2 lb mole/hr $= \frac{5}{32} = 0.1563$

N_2 lb mole/hr $= \frac{2}{28} = 0.0714$

Total: lb mole/hr $= 0.3867$

$$\therefore \text{ACFM} = 0.3867 \times \frac{379}{60} \times \frac{29.92}{2} \times \frac{(460 + 100)}{(460 + 60)} = 39.5$$

Section Three 3

EFFECTS AND CORRECTIONS FOR VARIOUS LIQUID AND GAS PROPERTIES ON PUMP PERFORMANCE

1. THEORETICAL AND ACTUAL PUMP CAPACITY

STERLING SIHI liquid ring gas pumps are rotary positive displacement machines and as such their theoretical capacity is given by:

$$\dot{S}_{th} = V_A \times RPM$$

Where $\dot{S}_{th}$ = Theoretical pumping speed or capacity in CFM

V_A = Volume of impeller cells in CF

The impeller cells are filled with a mixture of incoming gas and evaporated vapor from the service liquid. The portion of the volume occupied by this vapor will reduce the theoretical displacement accordingly.

According to the Dalton Gas Law, such displacement is given by:

$$\propto = \frac{V_{GAS}}{V_{TOTAL}} = \frac{P_{PG}}{P_t} = \frac{P - P_{PV}}{P}$$

Where $\propto$ = Reducing factor

P = Inlet pressure of pump

P_{PV} = Vapor pressure of service liquid

But the gas in the impeller cells will not be entirely discharged, also there are flow losses in the inlet and outlet ports as well as internal leakage losses which further reduce the pump theoretical displacement.

Therefore, a volumetric efficiency must be introduced giving us the final formula:

$$\dot{S}_A = \eta_{Vol.} \times \propto \times \dot{S}_{th}$$

Where $\dot{S}_A$ = Actual capacity (ACFM)

η_{Vol} = Volumetric efficiency

2. VAPOR PRESSURE EFFECTS

As previously stated, the vapor pressure of the service liquid will have a direct influence on the gas handling capability of the liquid ring pump.

It will be noticed from the above theoretical discussion and pictorially from Figure 9 below that the total of $\dot{S}_G$ (gas capacity) and $\dot{S}_v$ (vapor capacity) could then be the optimum pump capacity.

The actual capacity will, therefore, increase when the $\dot{S}_v$ portion decreases, i.e. when service liquids with very low vapor pressures are used (oils). Conversely, when the service liquid vapor pressures are higher, the pump gas handling capability will be reduced.

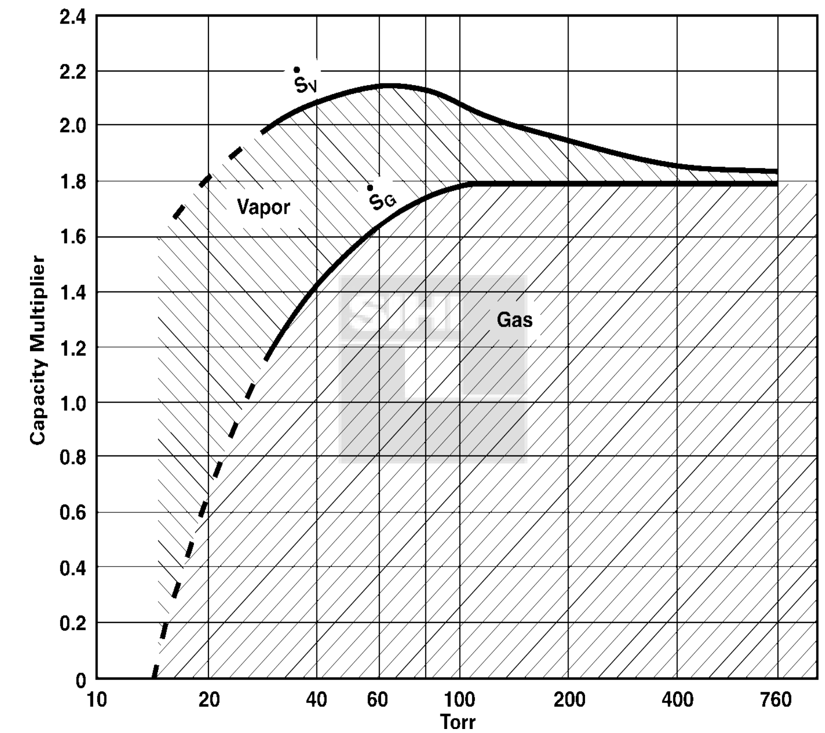


FIGURE 9 VOLUME OCCUPIED BY THE VAPOR VS VOLUME OCCUPIED BY THE ENTRAINED GAS

We can say then that the suction capacity (pumping speed) of a liquid ring vacuum pump is dependent upon the vapor pressure of the service liquid. The listed capacity of STERLING SIHI liquid ring vacuum pumps are based on the use of service water at 15°C (60°F). Therefore, when the service liquid has a vapor pressure different than water at 60°F, capacity must be corrected accordingly. The applicable correcting factors for water are obtainable from the Appendix 1 and 2.

When service liquids other than water are used, correction for vapor pressure can be made by matching the liquid's vapor pressure with that of water (from steam tables); finding at what temperature the water would have the same vapor pressure and applying the correction factors as per Appendix 1 or 2.

3. SERVICE LIQUID EFFECTS

If the liquid ring vacuum pump is to handle gases containing water vapors, then the use of water as service liquid may be the best choice. Most of the incoming vapors will condense in the pump and be discharged as condensate together with the service water and the non-condensables. Normally, this mixture is discharged from the pump into a gas/liquid separator where the gases are separated from the liquid by gravity.

The gases, which will be water saturated at the discharge pressure and temperature, may be vented to atmosphere or directed to other areas as the process requires.

The separated water may be drained or returned to the pump after it has been cooled via a heat exchanger or after a fresh make-up has been added in order to remove the heat imparted by compression and condensation. Since the service liquid must always be compatible with the process, the use of water as a service liquid is not always advantageous or possible. When the gases contain condensables other than water vapor, service liquids which are chemically compatible with these vapors must be selected. The physical characteristics of the chosen liquid are important.

Density, viscosity, vapor pressure, as well as solubility of the handled gases in the service liquid will be significant.

In many applications, it is possible to select a service liquid which will help in the condensation of the incoming vapors and will separate by gravity from the non-condensables in the separator just as we have seen in the case of air and water. However, in some instances, the chosen service liquid when mixed with the condensables may create a new mixture in the pump. This new mixture after being discharged from the pump must be treated so that the pump will reuse the clean liquid as originally selected and not a contaminated liquid which may have different physical properties.

4. DENSITY EFFECTS

The compression of a gas is obtained from the rotating liquid ring which must have at least an energy equal to the given isothermal compression energy. The amount of energy needed varies with the impeller rotating speed (RPM), the density of the service liquid used, and the volume of service liquid. The inner contour of the liquid ring is influenced by the absorbed energy which, in turn, will effect the suction capacity and, therefore, pump performance. Since energy needed varies directly with density, a correction for design power must be made if using a service liquid other than water. Since specific gravity is a measure of the density of a compound relative to water, this is a convenient property to relate performance.

Specific gravity is defined as: $\dfrac{\text{Mass of 1 cc of X}}{\text{Mass of 1 cc water @ 4°C}} = \dfrac{\text{Density}_X}{\text{Density}_W}$

Since density of water = 1 g/cm³ or 1 g/ml @ 4°C

Then specific gravity = Density

For liquids having specific gravity between 0.8 and 1.2, the following may be applied for quick correction only.

Power requirement:

$$\frac{BHP_x}{BHP} = \sqrt{\frac{\partial_x}{\partial_w}}$$

$$C_{HP} = \frac{BHP_x}{BHP}$$

$$BHP_x = C_{HP} \times BHP$$

Where BHP = Normal power required when using water

BHP_x = Power required for different specific gravity liquid

C_{HP} = Correction for horsepower required when using different specific gravity service liquid

∂_w = Specific gravity of water = 1.0

∂_x = Specific gravity of proposed liquid

Capacity correction: $\dot{S}_x = C_{SD} \times \dot{S}_{DA}$

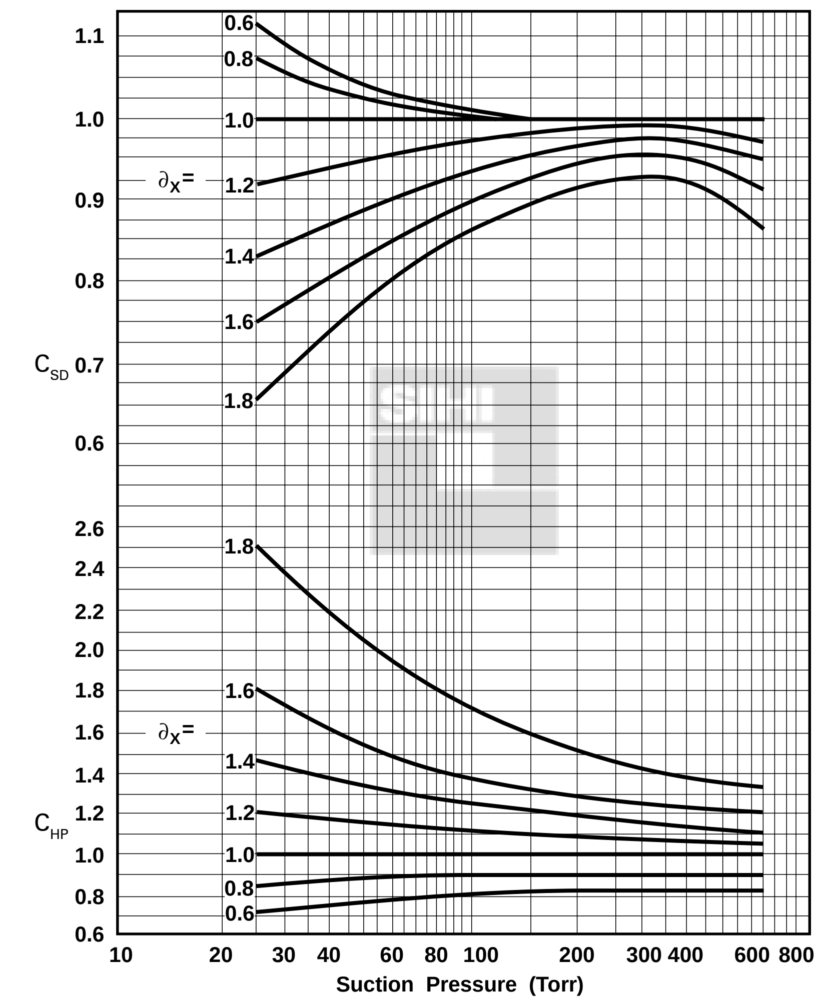
Where $\dot{S}_x$ = Pump capacity with service liquid other than water

C_{SD} = Correction for capacity when using different specific gravity service liquid

$\dot{S}_{DA}$ = Normal pump capacity with water as service liquid

The following graph illustrates corrections which must be made in the case of a medium size vacuum pump (LPH 45000 Series).

Note: These values, especially the capacity correction C_{SD}, cannot be used for every pump, as they are dependent upon the impeller diameter, rotational speed, and other factors governing such corrections. Values should be used for illustrative or quick approximations only. Should accurate calculations be desired, contact the factory.



$C_{SD} = \dfrac{\dot{S}_X}{\dot{S}_{DA}}$

$C_{HP} = \dfrac{BHP_X}{BHP}$

Service Liquid: Vp = 12.8 Torr

Viscosity: 1°E

Discharge Pressure: 760 Torr

FIGURE 10 DENSITY EFFECT ON PUMP PERFORMANCE

5. VISCOSITY EFFECTS

The pump capacity and especially power requirements are greatly affected by the viscosity of the service liquid.

The influence of viscosity on the suction capacity is normally relatively small and depends above all on the sealing attainable between the impeller and intermediate plate.

Capacity changes when using service liquid with viscosity of **2 to 20 Centistokes** may be obtained from:

$$\frac{\dot{S}_x}{\dot{S}_{DA}} = 1 - 0.01 \times \frac{\mu_x}{\mu_w}$$

$$C_{SV} = \frac{\dot{S}_{DA}}{\dot{S}_x} \quad \text{or} \quad \dot{S}_x = \dot{S}_{DA} \times C_{SV}$$

Where

$\dot{S}_{DA}$ = Normal pump capacity with water as service liquid

$\dot{S}_x$ = Pump capacity with service liquid other than water

C_{SV} = Correction for capacity when using different viscosity service liquid

μ_w = Viscosity of water = 1.15 cSt at 60°F (15°C)

μ_x = Viscosity of viscous liquid in cSt

Viscosity of liquids from **2 to 20 centistokes** will change the power requirements as follows:

$$\frac{BHP_x}{BHP} = 1 + (0.01 \text{ to } 0.02) \times \frac{\mu_x}{\mu_w}$$

$$C_{VHP} = \frac{BHP_x}{BHP} \quad \text{or} \quad BHP_x = C_{VHP} \times BHP$$

Where

BHP = Power required when using service water

BHP_x = Power required when using viscous liquid

C_{VHP} = Correction for horsepower when using different viscosity service liquid

μ_w = Viscosity of water = 1.15 cSt at 60°F (15°C)

μ_x = Viscosity of actual liquid in cSt

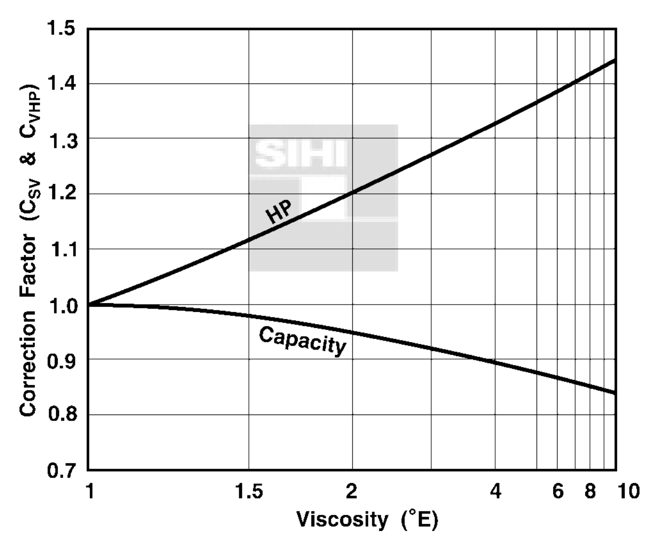
The influence of viscosity on the absorbed power will depend upon the Reynolds number.

The BHP increases as the viscosity increases. The following graph is offered as an example and is only applicable for a medium size unit (LPH 45000).

DO NOT USE THESE VALUES FOR OTHER PUMP MODELS.

FIGURE 11 TYPICAL VISCOSITY EFFECT ON PUMP PERFORMANCE CURVES

Note: Consult factory as necessary.



When using a viscous liquid, normal flow of the service liquid is reduced as follows:

$$\frac{\dot{Q}_x}{\dot{Q}} = 1 - 0.015 \times \frac{\mu_x}{\mu_w}$$

$$C_Q = \frac{\dot{Q}_x}{\dot{Q}}$$

Where

$\dot{Q}$ = Normal flow of service water

$\dot{Q}_x$ = Flow of viscous liquid

C_Q = Correction for service liquid flow when using different viscosity service liquid

μ_w = Viscosity of water = 1.15 cSt at 60°F (15°C)

μ_x = Viscosity of viscous liquid in cSt

6. SOLUBILITY OF GASES

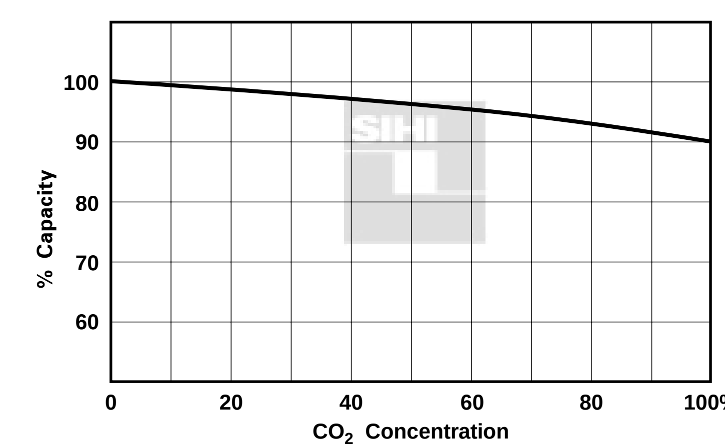
The solubility of the inlet gases in the service liquid must be taken into consideration when selecting a liquid ring pump.

Gas compressed will dissolve in the service liquid at the discharge pressure. When this enriched mixture returns to suction side, outgassing will occur at the reduced pressure. The "outgas" will take some of the space in the impeller cells which was available for the incoming gas. Hence, a reduction in pump capacity will be experienced.

Generally, the decrease in capacity connected with this phenomena **is not as great as the theoretical calculations would suggest.** The fluid is exposed to this low pressure area for a very short time, hence complete outgassing of the dissolved gas is never fully reached.

Tests have shown, for example, that when handling CO_2 with water as service liquid, the drop in capacity increases as the inlet pressure decreases with a maximum drop in capacity of about 10% when operating at 30 Torr.

FIGURE 12 CAPACITY DROP VERSUS CO_2 CONCENTRATION



The decrease in pump capacity is accentuated when handling gases with greater solubility such as ethylene oxide or SO_2 with water as service liquid.

7. HEAT OF COMPRESSION

During compression of any gas, most of the energy used for compression is converted into heat.

In liquid ring gas pumps, most of the heat generated is absorbed by the service liquid and hence, discharged with the liquid. The compression process is, for all practical purposes, isothermal (constant temperature).

The quantity of heat in BTU/hr is given by $\dot{H}_{ic}$ = 2545 BTU/hr. x BHP.

It can be assumed that about 10% of the quantity of heat $\dot{H}_{ic}$ is dispersed due to heat transfer, to the surroundings and the balance (approx. 90%) is passed on to the service liquid. As a rule, the incoming gas has low heat value which has little effect on the temperature of the service liquid. Hence, heat added or removed from the gas is ignored in the following formula.

Since the gas is so thoroughly mixed with the service liquid during compression, it can be assumed that the gas at pump discharge has the same temperature as the liquid. When condensables are not present, discharge temperature is given by:

$$T_m = T_1 + \frac{0.9 \times 2545 \times BHP}{\dot{Q} \times 8.34 \times 60 \times \partial \times C_p}$$

Where T_m = Temperature of gas and liquid mixture leaving the pump (°F)

T_1 = Temperature of service liquid entering the pump (°F)

∂ = Specific gravity of service liquid (1.0 for water)

C_p = Specific heat of service liquid (1.0 BTU/lb • °F or 1.0 cal/gm • °C for water)

$\dot{Q}$ = Flow of liquid in USGPM

8.34 = Approximate weight of 1 gallon of water (lbs) @ 60 °F

When the entrained gases are condensable, there will be additional heat to be removed by the service liquid due to condensation of these gases. Therefore, its temperature will be:

$$T_f = T_m + T_c$$

Where T_f = Final temperature (°F)

T_c = Condensing temperature rise in °F

$$T_c = \frac{\text{lb/hr} \times h_{fg}}{\dot{Q} \times 8.34 \times 60 \times C_p \times \partial}$$

h_{fg} = Enthalpy change for condensed vapor

Note: This is not completely correct since it assumes the entire condensable load condenses in the pump. The actual temperature rise will be somewhat less based on the mass of vapor which does not condense and is discharged as vapor. If a more exact discharge temperature is required, contact the factory.

When recirculating the service liquid, the heat to be removed from the service liquid $\dot{H}_t$ in BTU/hr is given by:

$$\dot{H}_t = \dot{H}_{ic} + \dot{H}_c + \dot{H}_{gc}$$

Where $\dot{H}_{ic}$ = Isothermal compression and friction heat = 0.9 x 2545 x BHP

$\dot{H}_c$ = Condensation heat = lb/hr (condensed vapor) x enthalpy change = lb/hr x h_{fg}

$\dot{H}_c$ = lb/hr (condensed vapor) x Δh_v

Or $\dot{H}_{gc}$ = Gas cooling heat = enthalpy change = lb/hr (gas) x C_p x $(T_1 - T_2)$

T_1 = Incoming gas temperature

T_2 = Outgoing gas and liquid temperature

h_{fg} = Enthalpy change for condensed vapor

Δh_v = Latent heat of vaporization

C_p = Specific heat

Note: $\dot{H}_{gc}$ is normally very small, hence can be neglected.

8. LIQUIDS IN THE SUCTION LINE

Liquid ring vacuum pumps are capable of handling moderate liquid flows over and above the normal service liquid flow.

This will, however, cause a reduction in pump capacity and increase in horsepower. Hence, it is advisable to limit the incoming liquid flow to about 1 to 2% of the gas volume flow (this will depend on the pump model).

Usually, the entrained liquid is continuous, hence reducing the normal service liquid flow by the same amount of entrained liquid is a good practice. When dealing with gases containing larger liquid flows, it is recommended a separator with corresponding liquid pump be installed before the vacuum pump.

9. PUMP METALLURGY

When handling corrosive gases and/or liquids, proper material selection for parts in contact with the media is required.

Premature pump failures are often the result of incompatibility of the chosen pump material with the process fluids. Therefore, it is imperative that attention is paid to pump metallurgy.

Pump parts are subject to various forms of wear such as corrosion including pitting, galvanic, intercrystalline, crevice, spot corrosion, and catalytic, among others. The pump internals are subjected to the corrosive media flowing at relatively high velocity and various conditions of pressure which, when combined, will lead to an intensification of the above given kinds of corrosion due to cavitation, abrasion and erosion.

It is possible that the severity of some applications is such that no readily available materials can be offered. In these instances, different service liquids may be considered in an attempt to render the process corrosion free.

Section 4 Four

VACUUM PUMP SIZING

Very often, the most difficult part in selecting a vacuum pump lies in the determination of the gas flow quantity and operating suction pressure.

When operating conditions are doubtful and a safe selection is desired, the following design rule must be remembered:

DO NOT DESIGN FOR LOWER ABSOLUTE SUCTION PRESSURE, RATHER INCREASE THE PUMP CAPACITY AT THE OPERATING PRESSURE DESIRED.

1. DETERMINATION OF PUMP CAPACITY FOR DRY GAS FLOW

A rather simple problem because the pump capability may be considered the same as for handling dry air. The capacity tables in the sales data book will apply. This ideal situation is, in practice, uncommon. However, because it can be expressed simply and is very convenient for testing, manufacturers of liquid ring vacuum pumps specify the pump capacity in terms of dry air at 20°C (68°F).

Service water temperature 15°C (60°F)

When we have 15°C service water available and handling dry inert gases, the pump selection is straight forward. Consulting the sales data book, we simply select a pump model to meet the requirements.

A. SERVICE WATER TEMPERATURE CORRECTION

Service water temperature other than 15°C (60°F)

The suction capacity of the pump varies in accordance with the correction curves shown in Appendix 1 and 2. It follows from the discussion of the effects of service liquid vapor pressure in Section III, parts 1 & 2, that if service liquid temperature increases, its vapor pressure increases, thus lowering pump efficiency. Through extensive in-house testing, STERLING SIHI has derived vapor pressure correction factors.

Example 6: What capacity will the LPH 45317 pump driven at 1750 RPM have when operating with 28°C (82°F) service water at 28" Hg?

From Appendix 2 for a two stage pump,

$$\frac{\dot{S}_A}{\dot{S}_{DA}} = 0.72$$

From Appendix 12, $\dot{S}_{DA} = 125$ ACFM

Hence, $\dot{S}_A = 125 \times 0.72 = 90$ Corrected Dry ACFM

Note: The selection is not complete until we have considered material requirements, shaft sealing arrangements, maximum allowable casing pressure and in some cases, solubility of gas and low molecular weights. **Refer to factory with low molecular weight gases such as Helium or Hydrogen.**

2. HANDLING OF A GAS/VAPOR MIXTURE

The liquid ring vacuum pump operates as a displacement compressor, gas cooler, and as a condenser. Consequently, when handling saturated gases, the pump capacity will increase in comparison to its capacity when handling dry gases. STERLING SIHI's Research Department, through extensive testing, has determined condensing correction factors which are applicable to STERLING SIHI liquid ring single, and two stage vacuum pumps.

Appendix 3 through 9 illustrate condensing correction factors (C_f) when the service water ranges from 10° to 40°C (50° to 104°F) in increments of 5°C (9°F).

For other service water and/or gas temperatures, extrapolation may be used with a reasonable degree of accuracy. **However, exact values may be obtained by contacting STERLING SIHI's engineers, giving full details of the application.**

A. SATURATED AIR/VAPOR MIXTURES

Determination of the inlet capacity required under saturation conditions when only dry gas rate is known. Since water vapor (or any other condensables) can be assumed to follow the ideal gas laws, calculations can be made using the following information:

From Dalton's Law, we know that two different gases (we treat water vapor as a gas) when stored in a common container, will fill the container completely.

The total pressure P_t in the container is the sum of the partial pressure of each gas in the container.

$$P_t = P_{P1} + P_{P2} + P_{P3} + \ldots\ldots + P_{PN}$$

If the designation (P_{PG}) is equal to the dry non-condensable gas and (P_{PV}) to the vapor or condensable gas, then:

$$P_t = P_{PG} + P_{PV}$$

But from the Ideal Gas Law, $P\dot{S} = \dot{m}RT$

Where
P = Absolute pressure
$\dot{S}$ = Volume flow rate
$\dot{m}$ = Mass flow rate
T = Absolute temperature
R = Gas constant $= \frac{1545}{MW}$ = ft. • lb_f/lb • mole • °R

And from Dalton's Law the following is also true:

a. The partial pressure of a gas in a mixture is the pressure exerted by that gas on the total volume. It is the pressure the gas would exert if it occupied the total volume by itself.

b. The partial volume of a gas in a mixture is considered at the total pressure of the mixture. The total pressure of the mixture is equal to the sum of all the partial pressures.

From this it can be concluded:

$$\dot{S}_{mixt.} = \frac{\dot{m}_g \times R \times T}{P_{PG}} \quad \text{or} \quad \dot{S}_g = \frac{\dot{m}_g \times R \times T}{P_t}$$

Solving both equations simultaneously for $\dot{m}_g$:

$$\dot{m}_g = \frac{\dot{S}_{mixt.}\, P_{PG}}{R\ T} \qquad \dot{m}_g = \frac{\dot{S}_g\, P_t}{R\ T}$$

$$\dot{m}_g = \dot{S}_{mixt.} \times P_{PG} = \dot{S}_g \times P_t$$

But:

$$P_t = P_{PG} + P_{PV}$$

Substituting:

$$\dot{S}_{mixt.}\,(P_t - P_{PV}) = \dot{S}_g \times P_t$$

Or:

$$\dot{S}_{mixt.} = \dot{S}_g \times \frac{P_t}{(P_t - P_{PV})} \quad \text{or} \quad \dot{S}_{mixt.} = \dot{S}_v \times \frac{P_t}{P_{PV}}$$

Example 7: What capacity must the pump be designed for if an air flow of 20 lb/hr dry air must be handled at an absolute pressure of 75 mm Hg when saturated with water vapor at a temperature of 40°C (104°F). From the steam tables, the vapor pressure of water vapor is:

P_{PV} at 40°C = 55.3 mm Hg Abs (Torr)

The standard temperature and pressure conditions are:

60°F at 760 mm Hg Abs

$$\text{SCFM} = \frac{\text{lb/hr}}{60} \times \frac{379}{\text{MW}}$$

$$\text{ACFM} = \text{SCFM} \times \frac{P_1}{P_2} \times \frac{T_2}{T_1}$$

$$\text{ACFM} = \dot{S}_{DA} = \frac{20}{60} \times \frac{379}{29} \times \frac{760}{75} \times \frac{(460 + 104)}{(460 + 60)} = 47.9 \text{ or} \cong 48 \text{ ACFM}$$

$$\text{ACFM} = \dot{S}_{mixt.} = \frac{P_t}{(P_t - P_{PV})} \times \dot{S}_{DA} = \frac{75}{(75 - 55.3)} \times 48 = 182.4 \text{ or} \cong 182 \text{ ACFM}$$

The volume of the mixture is almost four times the volume of dry air! At higher temperatures and lower pressures, the mixture will contain larger amounts of vapor. Cooling and condensing before the pump is usually advantageous since there is no pump more economical than a condenser. As a rule of thumb, condensers should be seriously considered when the partial pressure of the vapor is more than half the total operating pressure.

If $\frac{P_{PV}}{P_t} > 0.5$, **Then consider using a condenser.**

B. CORRECTION OF PUMP CAPACITY FOR SATURATED VAPORS

Service liquid: water at 15°C (60°F)

If the inlet gas stream is partially or fully saturated at the inlet temperature and pressure, the capacity of the pump will be higher than the dry air curve value. This occurs since the closer the inlet gas stream is to being saturated, the less service liquid evaporation can occur and hence the closer the useful or actual capacity is to the theoretical capacity. Further, if the inlet gas is saturated at a temperature above the service liquid temperature, gas cooling and condensation prior to and in the inlet of the pump will occur, causing a further increase in capacity. STERLING SIHI, through extensive in-house testing has derived condensation factors. These results are provided in the curves in Appendices 3 through 9.

Example 8: continuing with Example 7,

working pressure (P_t) = 75 Torr

volume of mixture saturated ($\dot{S}_{mixt.}$) = 182 ACFM

temperature of mixture ($T_{mixt.}$) = 40°C (104°F)

Solution: from Appendix 4

$$C_f = \frac{\dot{S}_{mixt.}}{\dot{S}_{DA}} = 1.75$$

$$\dot{S}_{DA} = \frac{\dot{S}_{mixt.}}{C_f} = \frac{182}{1.75} = 104 \text{ ACFM}$$

Therefore, the pump can then be selected to handle 104 ACFM at 75 Torr.

Pump selection: LPH 45312 at 1750 RPM. (From Appendix 11)

C. MINIMUM NON-CONDENSABLES REQUIRED

When handling gas mixtures with large amounts of condensables, we must consider the effect of cavitation at the pump discharge side due to lack of non-condensables (the condensables will condense in the pump during compression). The minimum amount of non-condensables should be controlled at all times and should correspond at least to the listed minimum flow (of the particular pump model) at the lowest suction pressure. This is to ensure that sufficient non-condensables are present at the lowest suction pressure to prevent cavitation.

Example 9: Continuing with Example 8, during compression from 75 to 760 Torr, most of the water vapor will condense. Assuming the pump is capable of handling the previous 20 lb/hr of dry air (and this is the actual leak rate) we can check for cavitation conditions by simply expanding the given 20 lb/hr dry air to its volume at 25 Torr, and then comparing this value with the curve capacity @ 25 Torr.

$$\dot{S}_{DA} = \frac{20}{60} \times \frac{379}{29} \times \frac{760}{25} \times \frac{(460 + 68)}{(460 + 60)} = 134.5 \text{ ACFM}$$

The LPH 45312 @ 1750 RPM and 25 Torr has a dry air flow of 61 ACFM (from Appendix 11). Since 134.5 ACFM dry air actually will be pumped, the LPH 45312 should be able to operate without an air bleed.

D. SERVICE LIQUID TEMPERATURE CORRECTION (REFER TO PAGES 16 & 17)

Service Liquid: Water at temperatures other than 15°C (60°F)

Example 10: From the conditions in Example 7 and service liquid temperature = 35°C (95°F).

We have C_f = 1.6 condensing correction factor (from Appendix 8).

From Appendix 2: A further correction for service liquid vapor pressure must be considered.

$$\frac{\dot{S}_A}{\dot{S}_{DA}} = 0.7 \text{ correction factor for 35°C service water}$$

Hence: $$\dot{S}_{DA} = \frac{182}{1.6 \times 0.7} = 162.5 \text{ ACFM at 75 Torr}$$

the pump selection will now be LPH 55312 at 1750 RPM. (From Appendix 13)
from the performance curve at 75 Torr (26.96 in. Hg vac)

$$\dot{S}_A = 191 \times 1.6 \times 0.7 = 213.9 \text{ ACFM}$$

E. CONDENSING PRIOR TO PUMPING

Assume coolant temperature at 15°C (60°F)

The most efficient method of handling condensible vapors is by using a condenser. This can be illustrated by using the data from Example 7.

Example 11: Assuming coolant at 60°F (15°C) is available to the condenser, it is possible to have the gas temperature at the condenser discharge in the region of 68°F minimum.

Considering the conditions per Example 7:

P_t = 75 Torr T_1 = 104°F

$\dot{S}_{DA}$ = 48 ACFM (@104°F) $\dot{S}_{mixt.}$ = 182 ACFM

Assuming we condense and cool to 68°F, the partial pressure of vapor after condenser from steam tables:

P_{PV} @ 68°F = 17.5 Torr

$$\dot{S}_{mixt.} = \frac{P_t}{(P_t - P_{PV})} \times \dot{S}_{DA}$$

$$\therefore \dot{S}_{mixt.} = 48 \times \frac{75}{(75 - 17.5)} \times \frac{(460 \times 68)}{(460 + 104)} = 58.6 \text{ ACFM}$$

This is about 1/3 the original design capacity!

Effect of condensation inside the pump: since the gas exiting any condenser is saturated at the condensing pressure and temperature, a further increase due to condensation in the pump will apply.

service water temperature = 15°C (60°F)

saturated gas temperature (exiting the condenser) = 20°C (68°F)

C_f = 1.18 condensing correction factor for 15°C service water (Appendix 4)

$$\dot{S}_{DA} = \frac{58.6}{1.18} \cong 50 \text{ ACFM}$$

Pump selection: LPH 3708 at 1750 RPM from the pump performance curve (Appendix 10)

$$\dot{S}_{mixt.} = 66 \times 1.18 = 78 \text{ ACFM at 75 Torr}$$

Amount of water vapor per unit time ($\dot{m}$) condensed in the condenser is given by:

$$P\dot{S}_{mixt.} = \dot{m}RT$$

$$\dot{m}_1 \text{ before condenser} = \frac{P \times \dot{S}_{mixt.} \text{ before}}{R \times T \text{ before}}$$

$$\dot{m}_2 \text{ after condenser} = \frac{P \times \dot{S}_{mixt.} \text{ after}}{R \times T \text{ after}}$$

$$\Delta\dot{m} = \dot{m} \text{ condensed} = \dot{m}_1 - \dot{m}_2$$

$$\Delta\dot{m} = \frac{P}{R} \times \left[\frac{\dot{S}_{mixt.} \text{ before}}{T \text{ before}} - \frac{\dot{S}_{mixt.} \text{ after}}{T \text{ after}}\right]$$

Note: R = gas constant = $\frac{1545}{MW}$ and MW = 18 (water vapor)

Therefore, $\frac{1545}{18} = 85.83$

Converting P to lb/ft.²:

1 Torr = 0.01934 PSI

Therefore, 0.01934 lb/in.² x 144 in.²/ft² = 2.785 lb/ft²

$$\Delta\dot{m} = \frac{75 \times 2.785}{85.83} \times \left[\frac{182}{(460 + 104)} - \frac{58.6}{(460 + 68)}\right] = 0.503 \text{ lb/min}$$

If a surface condenser is used, in most instances, it is possible to remove the condensate directly through the suction flange of the pump. This will depend upon the amount of condensate, specific size and operating point of the vacuum pump.

Assuming coolant temperature other than 15°C (60°F)

If coolant to the vacuum pump and condenser is other than 60°F (15°C), both condensation and service liquid vapor pressure corrections must be considered.

Example 12: Assume water available at 25°C (77°F), we estimate gas temperature at condenser discharge to be 30°C (86°F)

$\dot{S}_{mixt.}$ before condenser = 182 ACFM (20 lb/hr air saturated at 40°C from previous information in Example 7)

partial pressure of vapor after condenser from steam tables:

P_{PV} @ 86°F = 31.8 Torr

$$\dot{S}_{mixt.} \text{ after condenser} = 48 \times \frac{75}{(75 - 31.8)} \times \frac{(460 + 86)}{(460 + 104)} = 81 \text{ ACFM}$$

Correction for condensation inside the pump:

Following with the information in Examples 10 & 11,

service water temperature = 25°C (77°F)

gas mixture temperature (exiting the condenser) = 30°C (86°F)

P_t = 75 Torr

C_f = 1.3 condensing correction factor for 25°C service water (Appendix 6)

Correction for service liquid temperature:

$$\frac{\dot{S}_A}{\dot{S}_{DA}} = 0.88 \text{ vapor pressure correction factor for 25°C service water (Appendix 2)}$$

$$\therefore \dot{S}_{DA} = \frac{81}{1.3 \times 0.88} = 71 \text{ ACFM}$$

pump selection LPH 45317 direct driven at 1750 RPM (Refer to Appendix 12)

$$\dot{S}_{mixt.} = 84 \times 1.3 \times 0.88 = 96.1 \text{ ACFM at 75 Torr}$$

Amount of water vapor per unit time ($\dot{m}$) condensed in condenser:

$$\Delta\dot{m} = \frac{75 \times 2.785}{85.83} \times \left[\frac{182}{(460 + 104)} - \frac{81}{(460 + 86)}\right] = 0.42 \text{ lb/min}$$

The above examples are very common and reveal that condensation before the vacuum pump can be beneficial, most often resulting in the selection of a smaller vacuum pump.

F. ADDITIONAL INFORMATION

Mixtures of gases and vapors from liquids which are not mutually soluble has been defined by the general formula:

$$\dot{S}_{mixt.} = \frac{P_t}{(P_t - P_{PV})} \times \dot{S}_{DA}$$

Once we know $\dot{S}_{mixt.}$, the individual component can be determined either by their density at the respective partial pressure or by:

$$\dot{m}_1 = \frac{P_1 \times \dot{S}_{mixt.}}{R \times T_{mixt.}}$$

Where P_1 = lb/ft²

R = ft • lb_f/lb • mole • °R

Generally, the law of partial pressures for gases and vapors 1, 2, 3, etc. applies:

$$P_t = P_{PG} + P_{PV1} + P_{PV2} + \ldots + P_{PVn}$$

and

$$\frac{\dot{m}_{V1}}{\dot{m}_{gas}} = \frac{MW_{V1}}{MW_{gas}} \times \frac{P_{PV1}}{(P_t - P_{PV1})}$$

$$\dot{m}_{V1} = \dot{m}_{gas} \times \frac{MW_{V1}}{MW_{gas}} \times \frac{P_{PV1}}{(P_t - P_{PV1})}$$

Where MW = Molecular weight

$\dot{m}_{gas}$ = Weight flow rate of gas (non-condensable)

$\dot{m}_{V1}$ = Weight flow rate of vapor (condensable)

Example 13: Given the LPH 55312 from Example 10, calculate the discharge temperature.

$\dot{S}_{DA}$ = 191 x 1.6 x .07 = 213.9 ACFM

BHP = 13.5

service liquid requirement = $\dot{Q}$ = 10 GPM

$$\dot{m}_{v1} = \dot{m}_{gas} \times \frac{MW_{v1}}{MW_{gas}} \times \frac{P_{pv1}}{(P_t - P_{pv1})}$$

$$\text{vapor entering the pump} = 20 \times \frac{18}{29} \times \frac{55.3}{75 - 55.3} = 34.7 \text{ lb/hr}$$

(From page 21) $T_f = T_m + T_c$

T_m = Temperature of gas and liquid mixture leaving the pump (°F)

T_c = Condensing temperature rise (°F)

T_1 = Temperature of service liquid entering the pump (°F)

$$T_m = T_1 + \frac{0.9 \times 2545 \times BHP}{\dot{Q} \times 8.34 \times 60 \times \partial_w \times C_p}$$

Since ∂_w and C_p = 1.0 for water,

$$T_m = 95 + \frac{0.9 \times 2545 \times 13.5}{10 \times 8.34 \times 60} = 101.2°F$$

$$T_c = \frac{\text{condensed vapor (lb/hr) x enthalpy change (condensed gases)}}{\dot{Q} \times 8.34 \times 60 \times \partial_w \times C_p}$$

$$T_c = \frac{\text{condensed vapor (lb/hr)} \times h_{fg}}{10 \times 8.34 \times 60} = \frac{34.7 \times 1035}{10 \times 8.34 \times 60} = 7.2°F$$

$$^* T_f = T_m + T_c = 108.4°F$$

***Note:** per page 21 gas cooling not included

This example demonstrates that calculating the discharge temperature using the total vapor condensed will be satisfactory for estimating purposes, and will conservatively estimate discharge temperature for vapor carry-over calculation as well as heat load in the pump.

Should more accurate calculations of the gas discharge temperature be required, contact the factory.

Checking for cavitation:

$$\dot{S}_{DA} = \frac{20}{60} \times \frac{379}{29} = 4.36 \text{ SCFM (@ 60°F)}$$

LPH 55312 at 1750 RPM handles 117 ACFM at 25 Torr (from Appendix 13)

Hence, $117 \times \frac{25}{760} \times \frac{(460 + 60)}{(460 + 68)} = 3.85$ SCFM dry air is required.

No air bleeding is necessary since we have 4.36 SCFM available.

3. DETERMINATION OF PUMP CAPACITY FROM SYSTEM LEAK RATE

A. BASED ON THE LENGTH OF SEALING FACE

Inward leakage of a system can be calculated rather simply, using the following empirical values.

Condition Of The Seal	lb/hr Of Air Per Foot of Seal
Excellent	0.0020
Good	0.0067
Normal	0.0134

Values are applicable for pressures less than or equal to 400 Torr (mm Hg Abs), i.e. after exceeding critical pressure conditions.

Example 14: Find the inward leakage across a seal surface of total length 100 feet with normal seal quality.

air at 90°F

absolute pressure 40 Torr

air leak = 100 x 0.0134 = 1.34 lb/hr of air

$$\text{ACFM} = \frac{1.34}{60} \times \frac{379}{29} \times \frac{760}{40} \times \frac{(460 + 90)}{(460 + 60)} = 5.87 \text{ or approximately 6 ACFM}$$

B. BASED ON SYSTEM VOLUME

Where system volume is known, the empirical data has resulted in the leakage rates per Figure 13.

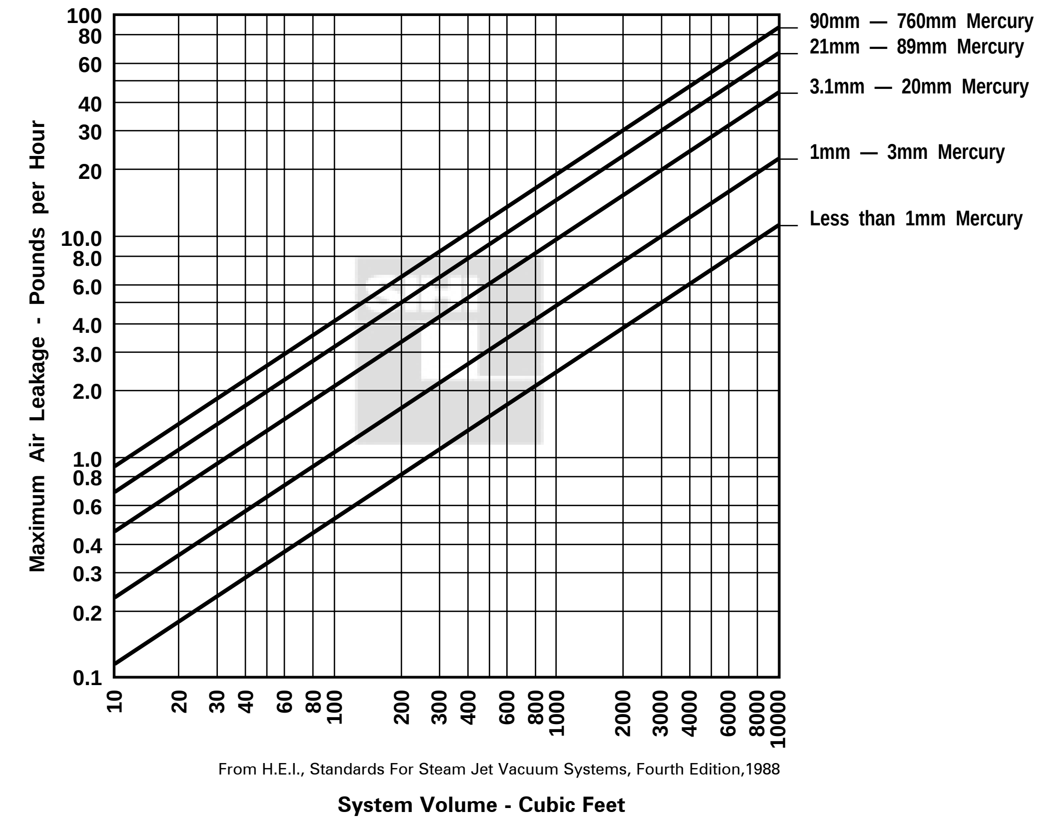


FIGURE 13 MAXIMUM AIR LEAKAGE VALUES FOR COMMERCIALLY TIGHT SYSTEMS

Example 15: System volume: 1000 cu. ft
Air at 90°F.
Absolute pressure 40 Torr (40 mm Hg Abs).
from figure 13: Leak Rate = Approx. 15 lb/hr

$$\text{ACFM} = \frac{15}{60} \times \frac{379}{29} \times \frac{760}{40} \times \frac{(460 + 90)}{(460 + 60)} = 65.7 \text{ ACFM}$$

Note: In the examples above, a leak rate was calculated. In order to determine the pump capacity, the system volume flow must be added to the leak rate. This gives the total volume flow through the pump.

C. BASED ON PRESSURE DROP TEST

When using metric units, i.e. Torr, liters and m³/hr

$$\dot{S}_{avg.} = 3.6 \frac{\dot{Q}_L}{P}$$

Where $\dot{S}_{avg.}$ = Volume flow rate in m³/hr

and $$\dot{Q}_L = \frac{\Delta P \times V_s}{t_L}$$

Where $\dot{Q}_L$ = Leakage in Torr-Liter/sec

P = Design operating pressure in Torr

ΔP = Differential pressure rise in Torr

V_s = Total system volume in liters

t_L = Lapsed time in seconds

When using English units, i.e. inch Hg, cubic feet, and cu. ft/min

$$\dot{S}_{avg.} = 60 \frac{\dot{Q}_L}{P}$$

Where $\dot{S}_{avg.}$ = Volume flow rate in cu. ft/min

and $$\dot{Q}_L = \frac{\Delta P \times V_s}{t_L}$$

Where $\dot{Q}_L$ = Leakage in inch Hg - cu. ft/sec

P = Design operating pressure in inches Hg Abs

ΔP = Differential pressure rise in inches

V_s = Total system volume in cubic feet

t_L = Lapsed time in seconds

Example 16: A 350 cu. ft system has been leak tested by evacuation to 2.0 inches Hg Abs sealed and monitored over time. After 10 minutes, it is found the pressure has risen to 2.36 inches Hg Abs If it is desired to operate at 1 inch Hg Abs, what pump capacity is necessary to maintain that pressure.

$$\dot{S}_{avg.} = 60 \times \frac{\dot{Q}_L}{P} \quad \text{and} \quad \dot{Q}_L = \frac{\Delta P \times V_s}{t_L}$$

therefore $$\dot{S}_{avg.} = 60 \times \frac{\Delta P \times V_s}{t_L \times P}$$

$$\dot{S}_{avg.} = 60 \times \frac{(2.36 - 2) \times 350}{10 \times 60 \times 1}$$

$$\dot{S}_{avg.} = 12.6 \text{ ACFM}$$

This obviously will provide the best method of determining optimum pump size and is highly recommended when replacing existing vacuum equipment.

4. PUMP DOWN OF A LEAK TIGHT SYSTEM

In installations operating intermittently (batch) the evacuation time (t_{ev}) is the most important factor.

To make the preliminary selection of a pump to evacuate a leak free system of known volume and specified evacuation time, the following formula is used:

$$\dot{S}_{avg.} = \frac{V_S}{t_{ev}} \times \ln\left(\frac{P_1}{P_2}\right)$$

Where t_{ev} = Evacuation time in minutes

V_S = Volume of system to be evacuated in cubic feet

$\dot{S}_{avg.}$ = Average capacity of vacuum pump in ACFM

P_1 = Initial absolute pressure

P_2 = Final absolute pressure

$\ln \frac{P_1}{P_2}$ per Figure 14

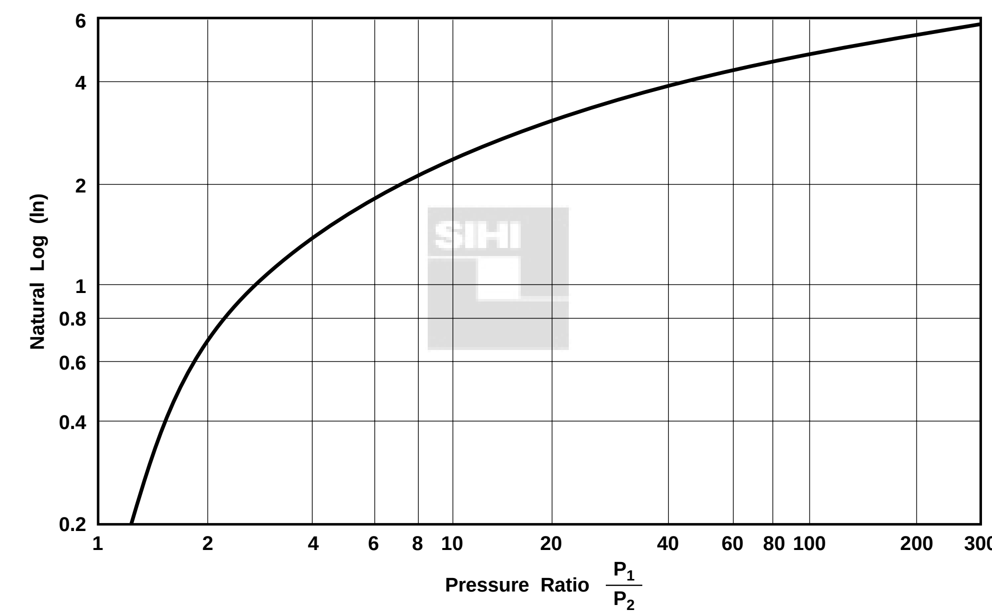


FIGURE 14 NATURAL LOG OF PRESSURE RATIO

Example 17: What average capacity is needed to evacuate a 100 cu. ft system from 760 Torr to 50 Torr in 2.25 minutes.

$$\dot{S}_{avg.} = \frac{100}{2.25} \times \ln\left(\frac{760}{50}\right) = 120.95 \text{ ACFM}$$

∴ Pump selection: LPH 45317 @ 1750 RPM

Once the pump size is selected, we must recalculate the evacuation time by using that pump's average capacity. This is done using the same formula as above but in the following form:

$$t_{ev} = \frac{V_S}{\dot{S}_{avg.}} \times \ln\left(\frac{P_1}{P_2}\right)$$

Example 18: Find evacuation time using pump model LPH 45317 driven at 1750 RPM.
system volume 100 cu. ft
initial pressure (atmospheric) 760 Torr.
final pressure 50 Torr.
service water 15°C (60°F)

Find average pump capacity from 760 to 50 Torr by estimating area under the curve such that the area above the line equals that below: Average Pump CFM = 120

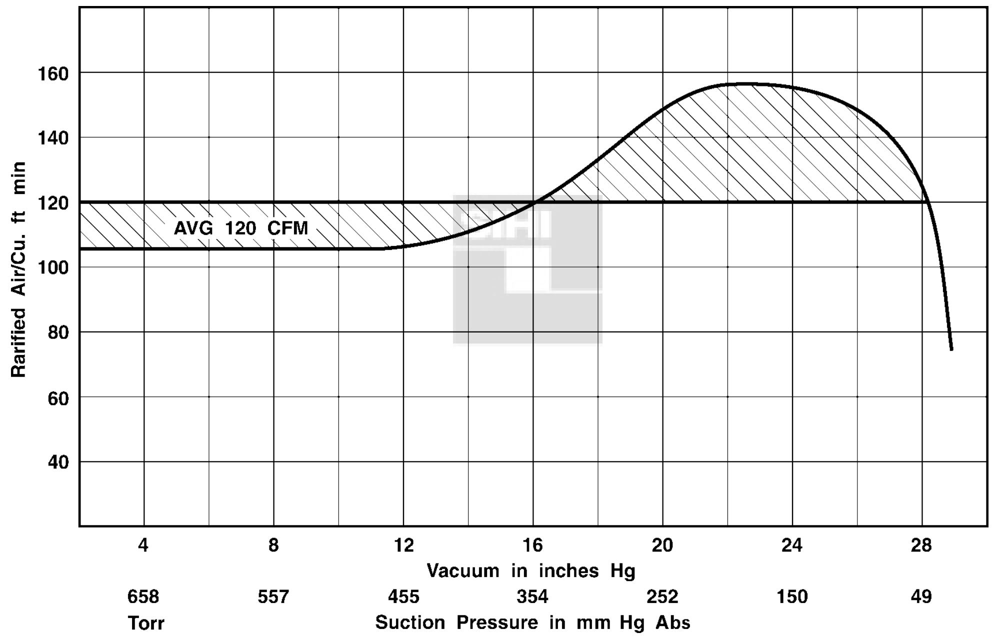


FIGURE 15 ESTIMATED AVERAGE PUMP CAPACITY FROM A PERFORMANCE CURVE

$$t_{ev} = \frac{100}{120} \times \ln\left(\frac{760}{50}\right) = 2.26 \text{ min}$$

A CLOSER APPROXIMATION OF EVACUATION TIME FOR A LEAKTIGHT SYSTEM USING A PARTICULAR PUMP CAN BE DONE BY INCREMENTAL SUMMATION AS SHOWN IN THE FOLLOWING EXAMPLE.

Example 19: Using the same pump model as in the previous example, from Fig. 15 we have:

Pump performance is constant at 105 CFM from 760 to 455 Torr, therefore:

$$t_{ev1} = \frac{100}{105} \times \ln\left(\frac{760}{455}\right) = 0.49 \text{ min.}$$

pump performance from 455 to 252 Torr is approximately 125 CFM, therefore:

$$t_{ev2} = \frac{100}{125} \times \ln\left(\frac{455}{252}\right) = 0.47 \text{ min.}$$

from 252 to 100 Torr average capacity is approximately 152 CFM, therefore:

$$t_{ev3} = \frac{100}{152} \times \ln\left(\frac{252}{100}\right) = 0.61 \text{ min.}$$

from 100 to 50 Torr average capacity is approximately 133 CFM, therefore:

$$t_{ev4} = \frac{100}{133} \times \ln\left(\frac{100}{50}\right) = 0.52 \text{ min.}$$

$$t_{ev} = t_{ev1} + t_{ev2} + t_{ev3} + t_{ev4} = 2.09 \text{ min.}$$

More accurate calculation will be obtained with shorter steps. However, since this is a theoretical time, a safety factor must be added to allow for air leakage, (usually 10 to 20%). Therefore, we may accept Example 19 as being sufficiently accurate.

These methods can also be used in instances where the customer may already have a pump that he wants to use and needs to know the evacuation time for his system.

5. TIME REQUIRED TO PUMP DOWN A SYSTEM WITH LEAKS

Given a system with known volume and known pressure change with time, the evacuation time may be calculated using:

$$t_{ev} = \frac{V_s}{\dot{S}_{avg.}} \times \ln \left[\frac{\left(\dfrac{P_1 \dot{S}_{avg.}}{V_s} - \dot{p} \right)}{\left(\dfrac{P_2 \dot{S}_{avg.}}{V_s} - \dot{p} \right)} \right]$$

Where
- P_1 = Initial pressure
- P_2 = Final pressure
- $\dot{S}_{avg.}$ = Average pump capacity between P_1 and P_2
- $\dot{p}$ = Pressure rise per unit time
- V_s = System volume
- t_{ev} = Time for evacuation

Example 20: The same system as Example 17 must be evacuated with the LPH 45317 pump, but now knowing that when isolated under vacuum, the pressure in the vessel rises 60 Torr per hour or 1.0 Torr per minute as a result of leakage.

$\dot{p}$ = System pressure rise in units of pressure per unit time. The units of pressure and time being the same as used elsewhere in the equation.
From Example 18, $\dot{S}_{avg.}$ = 120 CFM, therefore:

$$t_{ev} = \frac{100}{120} \times \ln \left[\frac{\left(\dfrac{760 \times 120}{100} - 1.0 \right)}{\left(\dfrac{50 \times 120}{100} - 1.0 \right)} \right] = 3.57 \text{ min.}$$

∴ The pump will require an extra 1.31 minutes to evacuate the vessel to 50 Torr due to added air leakages.

Note: This method can be used during system design by estimating system volume then estimating a leak rate (as shown previously) determining the pressure leak rate ($\dot{p}$), by assuming a time and calculating the change in pressure which would occur in this system assuming the estimated leakage rate actually did occur. Please contact the factory should assistance be required.

6. CORRECTION OF PUMP CAPACITY FOR ALTITUDE

All STERLING SIHI performance curves and technical data are referenced to barometric pressure at sea level. When operating at higher altitudes, the barometric pressure is always lower, therefore some calculations are required to correct for this pressure variation. Caution must be taken when specifying a pump at higher altitudes; the amount of vacuum desired cannot exceed the barometric pressure at that altitude.

Since the pump performance of a positive displacement pump is a function of pressure ratio:

$$\left(\frac{P_1}{P_2}\right)_{\text{at sea level}} = \left(\frac{P_1}{P_2}\right)_{\text{at altitude}}$$

$$(P_2)_{\text{at sea level}} = \frac{(P_1)_{\text{at sea level}}\ (P_2)_{\text{at altitude}}}{(P_1)_{\text{at altitude}}}$$

Example 21: Select a vacuum pump for 200 CFM to be installed at an altitude of 3000 meters (approximately 10,000 feet), and to operate at a vacuum of 13 in. Hg at that altitude using 15°C water as service liquid.

From Fig. 8, barometric pressure at 3000 meters is 525 mm Hg Abs, or 20.7 inch Hg absolute. Hence, a vacuum of 13 inches at this altitude means an absolute pressure of 20.7 - 13.0 = 7.7 in. Hg Abs

The actual pressure ratio of the unit is:

$$\left(\frac{P_1}{P_2}\right)_{\text{at altitude}} = \frac{20.7}{7.7} = 2.688$$

Because pump performance basically is a function of pressure ratio, the selection should be made at the similar pressure ratio at sea level.

i.e. $\frac{29.92}{2.688}$ = 11.13" Hg absolute

or 29.92 - 11.13 = 18.79 in. Hg vacuum

Pump selection: LPH 50518 at 1750 RPM

Always check the minimum absolute pressure required at the inlet with the minimum absolute pressure on the standard STERLING SIHI performance curve. If the actual operating pressure is less than the normal minimum shown on the standard curve, please verify pump selection with the factory.

Section Five

SERVICE LIQUID SUPPLY SYSTEMS

THE SERVICE LIQUID MAY BE SUPPLIED IN THREE (3) BASIC SYSTEM ARRANGEMENTS.

1. ONCE-THROUGH INSTALLATION

The service liquid enters the pump and is normally discharged to the drain after being separated from the gas.

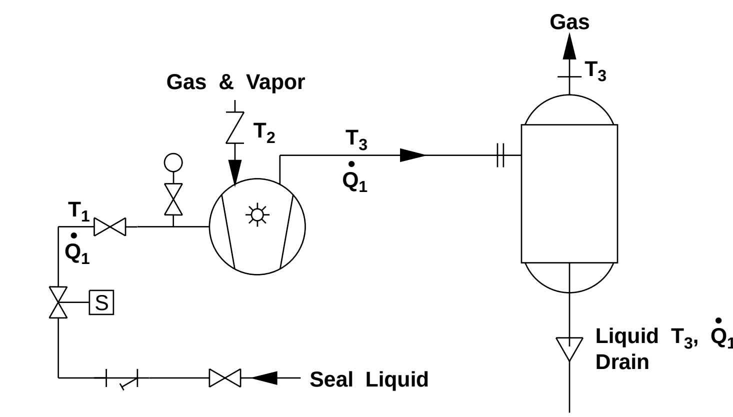


FIGURE 16 TYPICAL ONCE-THROUGH LAYOUT

Temperature rise calculation

Temperature rise is calculated using the heat of compression ($\dot{H}_{ic}$), heat of condensation ($\dot{H}_c$), and heat of gas cooling ($\dot{H}_{gc}$).

From the discussion on pages 21/22, final temperature rise (gas & liquid outlet temperature) is given by:

$$T_3 = \frac{(\dot{H}_{ic} + \dot{H}_c + \dot{H}_{gc})}{\dot{Q}_1 \times 8.34 \times 60 \times \partial \times C_P} + T_1$$

$\dot{Q}_1$ = Service liquid flow in GPM from data book

T_1, T_2 and T_3 are temperatures in °F.

2. PARTIAL RECIRCULATION INSTALLATION

The service liquid enters the pump and is discharged to the recirculation tank.

An additional controlled flow of cool service liquid is introduced (make-up) while an equal amount of liquid (plus any condensate) is discharged from the separator tank via an overflow connection to maintain the working level in the same horizontal plane as the pump shaft center line. The cool make-up removes heats of compression and condensation from the recirculated liquid.

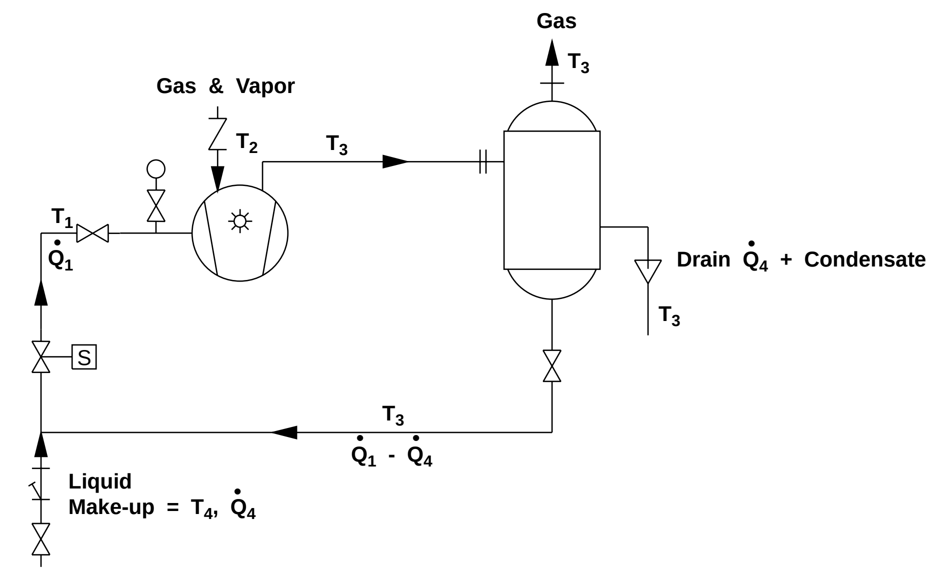


FIGURE 17 TYPICAL PARTIAL RECIRCULATION LAYOUT

Temperature rise calculation

When partial recirculation is employed, two conditions become important: 1) the gas/liquid outlet temperature and 2) the amount of cool make-up required. The amount of cool make-up required in turn depends on the required design capacity of the pump unit versus the actual pump capacity with the actual service liquid supply temperature.

Calculation of gas/liquid outlet temperature (T_3).

T_3 is calculated from isothermal heat ($\dot{H}_{ic}$), condensation heat ($\dot{H}_c$) and gas cooling heat ($\dot{H}_{gc}$) from pages 21/22.

$$T_3 = \frac{(\dot{H}_{ic} + \dot{H}_c + \dot{H}_{gc})}{\dot{Q}_1 \times 8.34 \times 60 \times \partial \times C_p} + T_1$$

For optimum (least quantity of cool make-up required) conditions T_1 is the important temperature. If minimum make-up is desired, T_1 should be selected at the highest temperature at which the pump capacity meets the required design capacity.

$$\dot{Q}_4 = \left[\frac{(T_3 - T_1)}{(T_3 - T_4)}\right] \bullet \dot{Q}_1$$

Where $\dot{Q}_1$ and $\dot{Q}_4$ are liquid flows in USGPM;
T_1, T_3, T_4, are temperatures in °F.

3. TOTAL RECIRCULATION INSTALLATION

The service liquid enters the pump, is discharged to the recirculation tank, cooled in a heat exchanger, and returned to the vacuum pump.

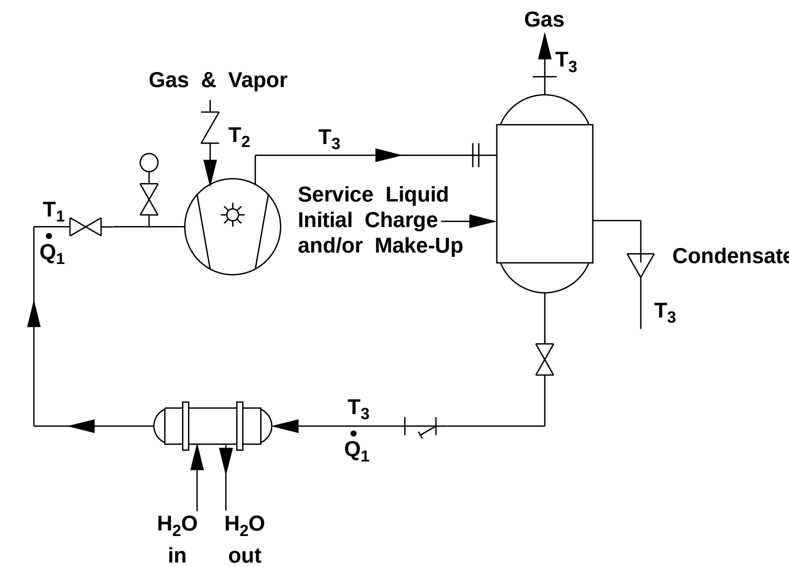


FIGURE 18 TYPICAL TOTAL RECIRCULATION LAYOUT

Temperature rise calculation

When totally recirculated systems are utilized, again two considerations are important: 1) the outlet gas/liquid temperature (T_3) and 2) the design service liquid supply temperature to the pump (T_1).

In order to minimize the coolant flow rate to the heat exchanger, T_1 should be selected at the highest temperature at which the selected pump model will be equal to the design capacity (with the warmest coolant to be supplied) including any required safety factor.

The heat load to the heat exchanger or service liquid cooler is as follows:

$$\dot{H}_t = \dot{H}_{ic} + \dot{H}_c + \dot{H}_{gc} \text{ from pages 21/22.}$$

Temperature out of the pump or into the cooler is calculated per page 21/22 as T_3.

$$T_3 = \frac{(\dot{H}_{ic} + \dot{H}_c + \dot{H}_{gc})}{\dot{Q}_1 \times 8.34 \times 60 \times \partial \times C_p} + T_1$$

$\dot{Q}_1$ = Service liquid flow in GPM from data book

T_1, T_2 and T_3 are temperatures in °F.

A booster pump for the recirculation of the service liquid is not required if the friction losses between the discharge separator and the vacuum pump service liquid inlet are not higher than 15% of the differential pressure between the pump discharge and the pump suction providing the normal continuous operating vacuum is greater than 10" Hg vacuum.

GLOSSARY OF TERMS
Used In Vacuum Technology

ABSOLUTE PRESSURE
Pressure measured from absolute zero, i.e., from an absolute vacuum.

ABSOLUTE TEMPERATURE
The temperature above absolute zero (point where molecular activity ceases) expressed as degrees Rankine (°R) in the English system of units or degrees Kelvin (K) in the S.I. system of units.

ACFM - Actual Cubic Feet per Minute
Actual cubic feet per minute of a volume of gas at operating pressure and temperature conditions.

AIR EJECTOR
A device used in conjunction with a liquid ring vacuum pump to develop pressures as low as 6 mm Hg Abs Principle of operation based on a venturi.

ATMOSPHERIC PRESSURE
The ambient pressure of the atmosphere typically expressed in inches Hg absolute. At sea level this value is defined as 29.92 in. Hg Abs

BAROMETRIC PRESSURE
Term synonymous with atmospheric pressure.

CAVITATION
Erosion of the pump components caused by the formation and sudden collapse of vapor bubbles in a liquid. This usually occurs near the discharge side of the pump.

COMPRESSION RATIO
Ratio of discharge pressure to inlet pressure.

CONDENSABLE GAS
A gas at a temperature below the critical temperature, enabling liquification by compression, without lowering the temperature. Also called a vapor.

DISPLACEMENT
The geometric volume swept out per unit time by the working mechanism of mechanical pumps at normal frequency.

DRY AIR
Pure air theoretically containing no condensable vapor at the temperatures and pressures handled. Practically, under vacuum conditions some vapor may be present but at insignificant quantities compared to that possible under saturation conditions.

EXPANDED AIR
Air at a pressure lower than atmosphere.

FREE AIR
Air at atmospheric pressure.

IDEAL GAS
A gas which obeys Boyle's Law and Charles' Law. Also known as perfect gas and can be represented by the equation $PV = nRT$.

INLET PRESSURE
Pertaining to liquid ring vacuum pumps, "Total Static Pressure" measured at the inlet flange of the pump or air ejector. Also called suction pressure.

LEAK
Relating to vacuum, a hole, opening or porosity in a system of piping, vessels and valving capable of passing gas from the outside to the inside of the system adding to the total mass flow rate.

LIQUID RING VACUUM PUMP
A rotary displacement pump using a liquid in the pump to compress the incoming gas stream.

LIQUID-SEALED MECHANICAL PUMP
A mechanical pump in which a liquid (usually oil) is used to seal the gap between parts which move with respect to one another and to reduce the free space in the compression chamber at the end of the compression cycle. The liquid also usually serves to lubricate and reduce wear.

MICRON OF MERCURY
A unit of pressure equal to 1/1000th of one millimeter of mercury pressure (1/1000th of a Torr); abbreviated as μ of Hg or μ Hg

MILLIMETER OF MERCURY
A unit of pressure corresponding to a column of mercury exactly one millimeter high at 0°C under standard acceleration of gravity.

MOLECULAR WEIGHT
A summation of the atomic weights of the atoms that make up a molecule.

NONCONDENSABLE GAS
A gas at a temperature higher than its critical temperature; a gas that cannot be liquefied solely by an increase in pressure.

OUTGASSING
The emission of gas from a liquid or a solid under vacuum.

PARTIAL PRESSURE
The pressure that would be obtained if the same mass of individual gas were alone in the same total volume at the same temperature.

RAREFIED AIR
Expanded air at a pressure lower than atmospheric.

RATE OF RISE
The rate of pressure increase over a given time period in a vacuum system which has been isolated from the pump.

ROUGH VACUUM
Range of absolute pressure from 760 mm Hg Abs to 1mm Hg Abs

ROUGHING PUMP
Pump used to reduce system pressure to a point at which another stage of vacuum equipment such as a pump, blower or jet can be utilized to reduce the pressure further to reach the operating condition or reduce the pressure in a given period of time.

SEAL
A joint or closure between two elements of a vacuum system which is effective in maintaining leakage at or below a required level.

SCFM - Standard Cubic Feet per Minute
Standard cubic feet/minute of a gas at standard pressure and temperature conditions.

THROUGHPUT
The quantity of gas in pressure-volume units, at a specified temperature, flowing across a specified open cross section.

TORR
A unit of absolute pressure defined at 1/760th of a standard atmosphere. One Torr is equivalent to 1 mm Hg or 133.3 Pascals in the SI unit of measurement.

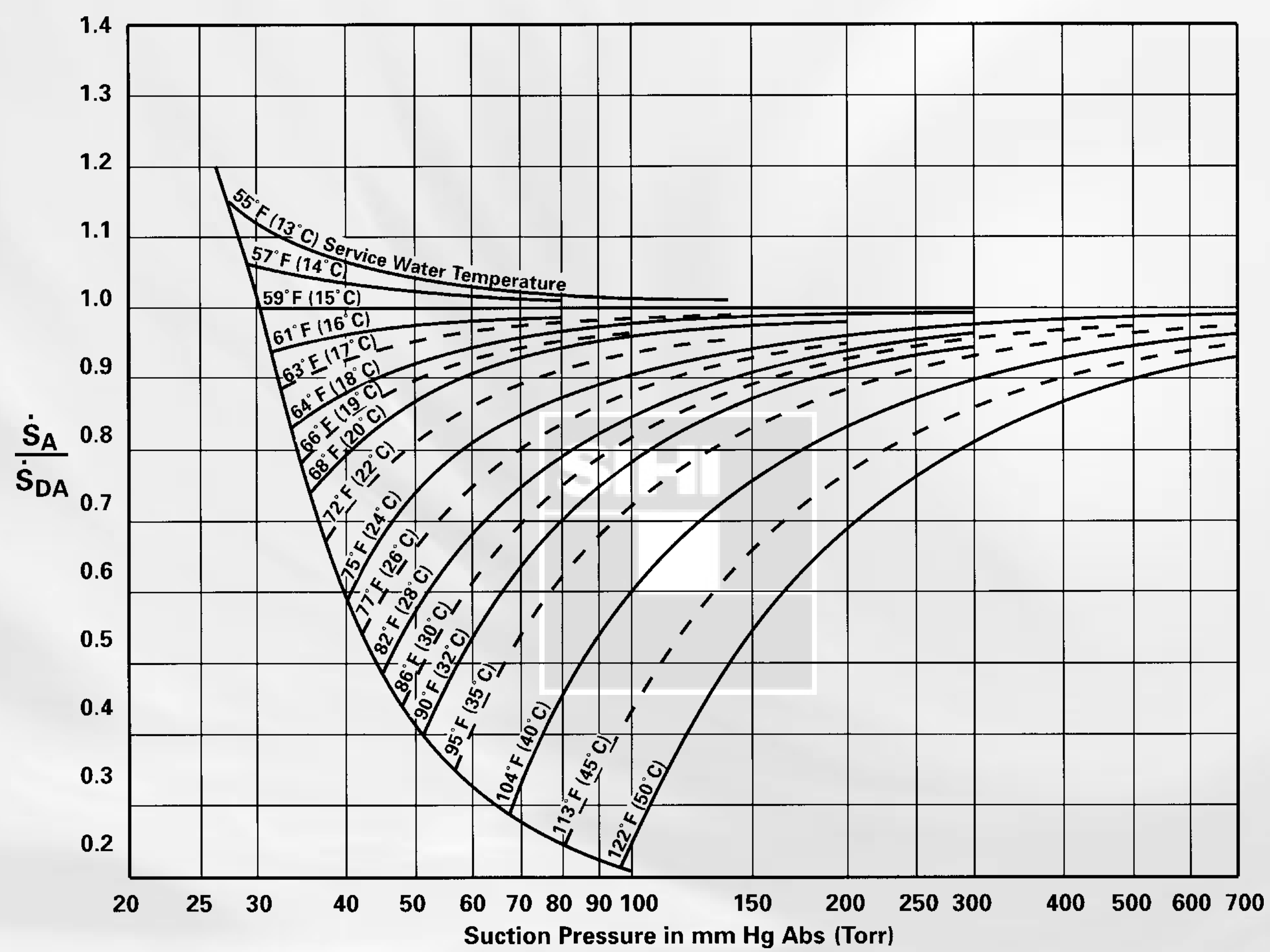
ULTIMATE PRESSURE
The limiting pressure approached in a vacuum system after a sufficient pumping time has elapsed to establish that further reduction in pressure will be negligible. Sometimes referred to as blank-off pressure.

VACUUM
The condition of a gaseous environment in which the gas pressure is below atmospheric pressure.

VOLUME FLOW RATE
Flow rate of gas at the actual pressure and temperature existing.

APPENDIX 1

EFFECT OF SERVICE WATER TEMPERATURE ON SINGLE STAGE LIQUID RING VACUUM PUMPS.

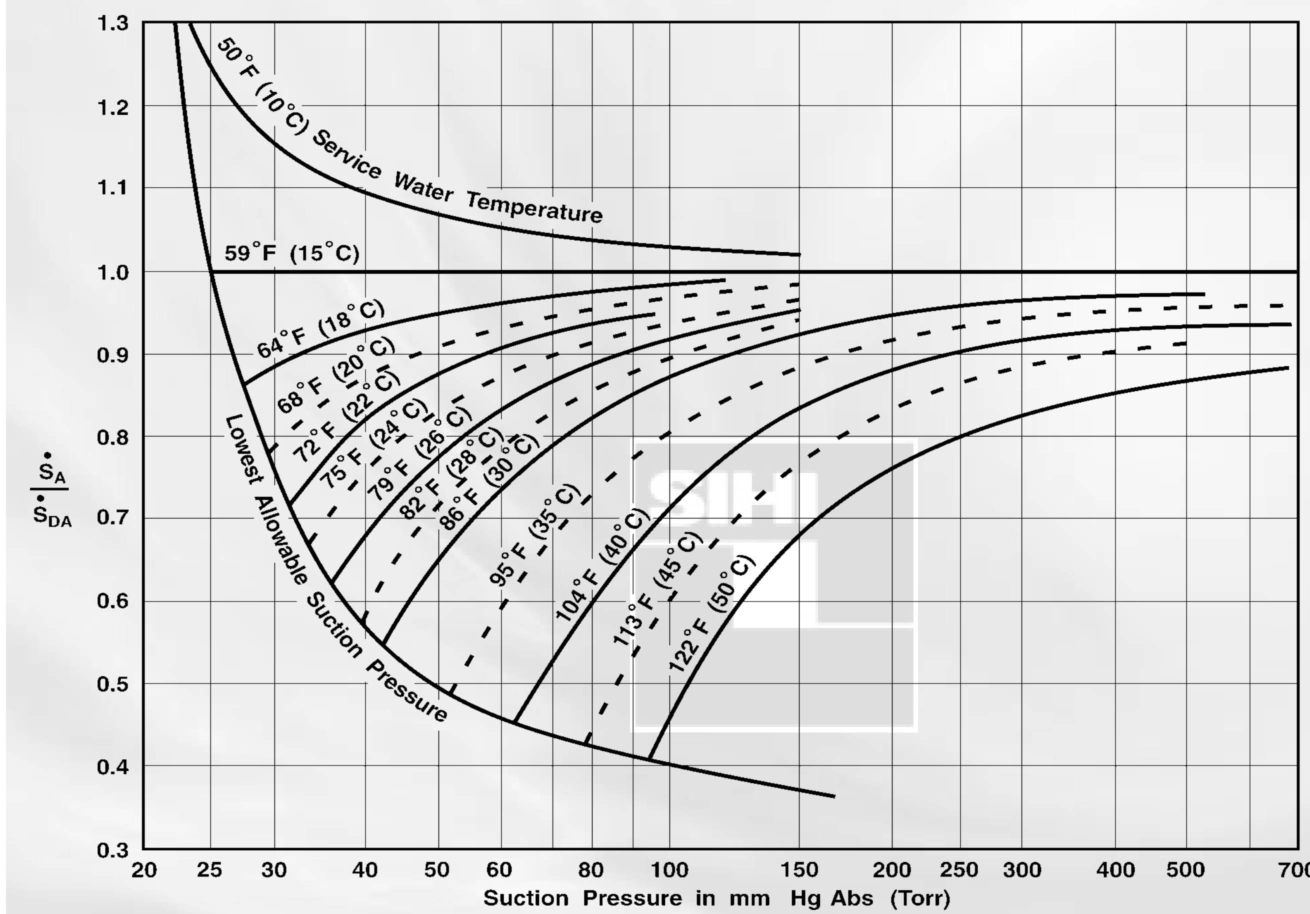


$\dot{S}_A$ = Actual Capacity

$\dot{S}_{DA}$ = Listed pump capacity from data book based on 15°C service water, 20°C air

APPENDIX 2

EFFECT OF SERVICE WATER TEMPERATURE ON TWO-STAGE LIQUID RING VACUUM PUMPS.

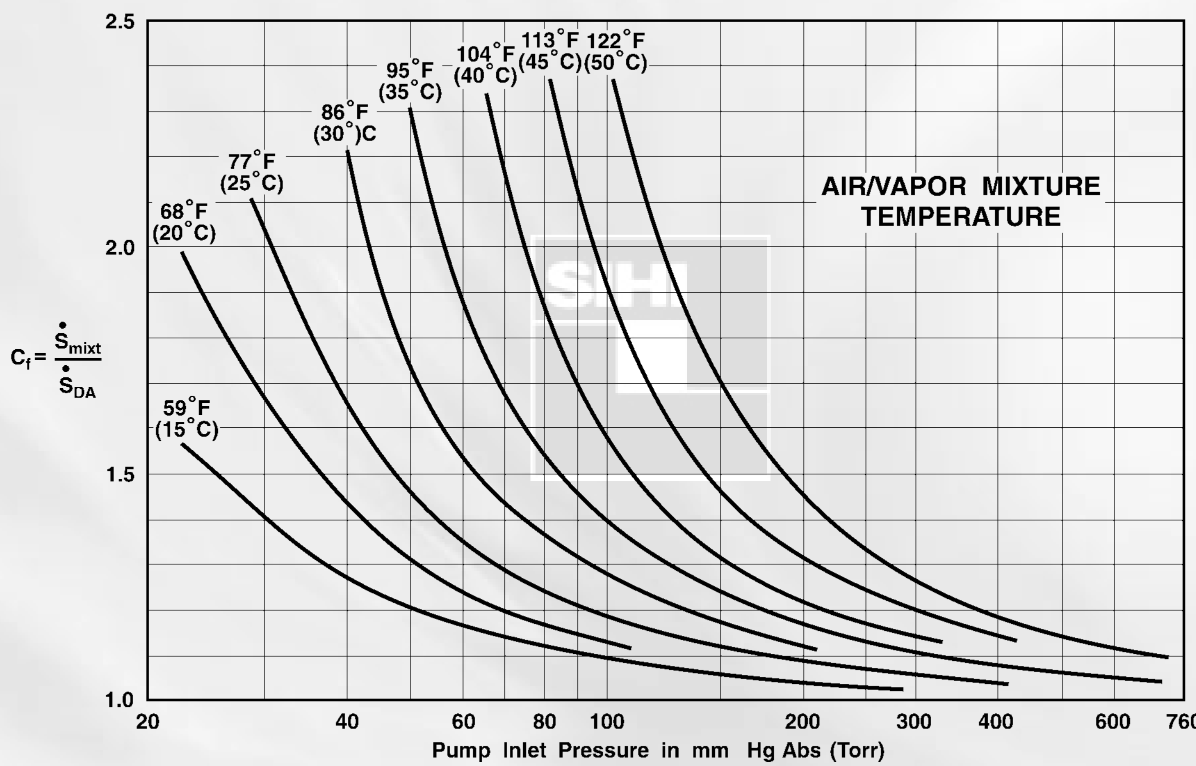


$\dot{S}_A$ = Actual Capacity

$\dot{S}_{DA}$ = Listed pump capacity from data book based on 15°C service water, 20°C air

APPENDIX 3

STERLING SIHI AVERAGE CONDENSING CORRECTION FACTORS FOR SATURATED AIR SERVICE USING 50°F (10°C) SERVICE WATER



C_f = Condensing correction factor, when service water is 10°C

$\dot{S}_{mixt.}$ = Air/vapor mixture capacity

$\dot{S}_{DA}$ = Dry air capacity, from data book, based on 10°C service water and 20°C air

$\dot{S}_{mixt.} = \dot{S}_{DA} \times C_f$

Example: STERLING SIHI pump model LPH 65320 driven at 1750 RPM is rated at 325 ACFM ($\dot{S}_{DA}$) dry air at 49 mm Hg Abs (28" Hg vac) and when using 10°C service water.

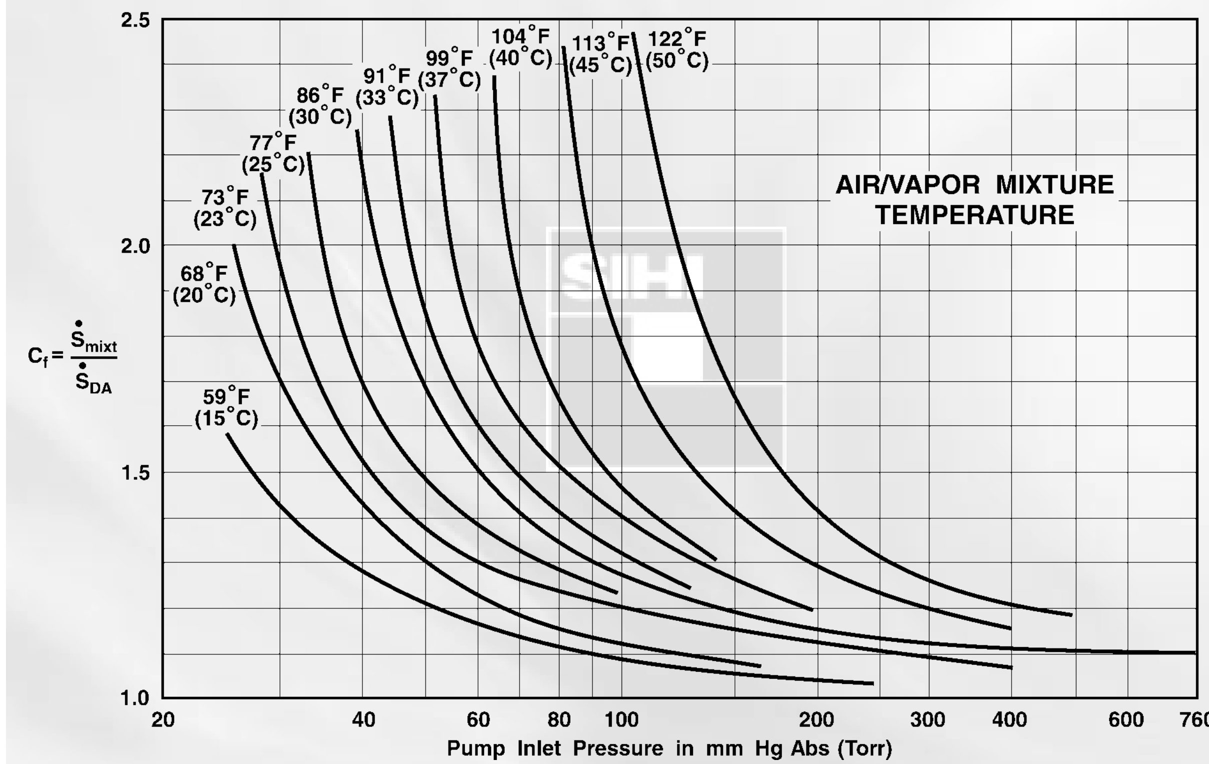
Find pump capacity ($\dot{S}_{mixt.}$) when handling saturated air at 25°C under same condition

C_f = 1.48 at 49 mm Hg Abs

$\dot{S}_{mixt.}$ = 325 x 1.48 = 481 ACFM

APPENDIX 4

STERLING SIHI AVERAGE CONDENSING CORRECTION FACTORS FOR SATURATED AIR SERVICE USING 60°F (15°C) SERVICE WATER



C_f = Condensing correction factor, when service water is 15°C

$\dot{S}_{mixt.}$ = Air/vapor mixture capacity

$\dot{S}_{DA}$ = Dry air capacity, from data book, based on 15°C service water and 20°C air

$\dot{S}_{mixt.} = \dot{S}_{DA} \times C_f$

Example: STERLING SIHI pump model LPH 65320 driven at 1750 RPM is rated at 305 ACFM ($\dot{S}_{DA}$) dry air at 49 mm Hg Abs (28" Hg vac) and when using 15°C service water.

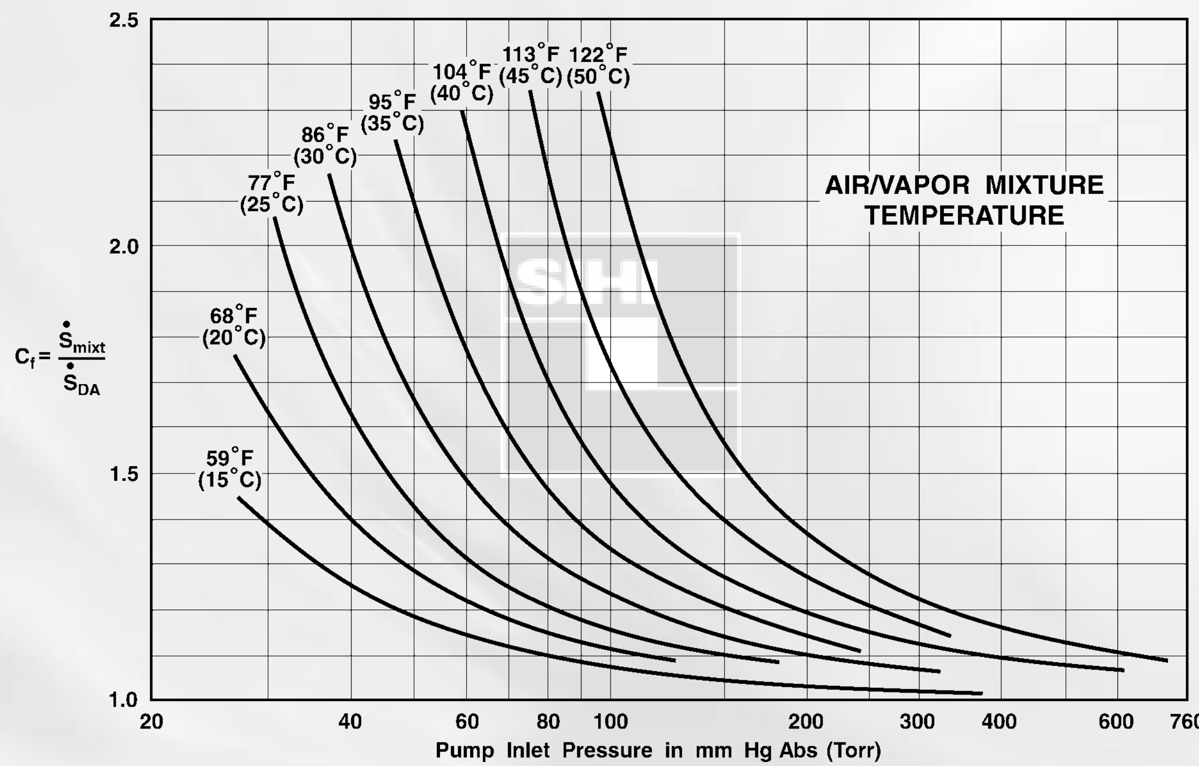
Find pump capacity ($\dot{S}_{mixt.}$) when handling saturated air at 25°C under same condition

C_f = 1.5 at 49 mm Hg Abs

$\dot{S}_{mixt.}$ = 305 x 1.5 = 457.5 ACFM

APPENDIX 5

STERLING SIHI AVERAGE CONDENSING CORRECTION FACTORS FOR SATURATED AIR SERVICE USING 68°F (20°C) SERVICE WATER



C_f = Condensing correction factor, when service water is 20°C

$\dot{S}_{mixt.}$ = Air/vapor mixture capacity

$\dot{S}_{DA}$ = Dry air capacity, from data book, based on 20°C service water and 20°C air

$\dot{S}_{mixt.} = \dot{S}_{DA} \times C_f$

Example: STERLING SIHI pump model LPH 65320 driven at 1750 RPM is rated at 276 ACFM ($\dot{S}_{DA}$) dry air at 49 mm Hg Abs (28" Hg vac) and when using 20°C service water.

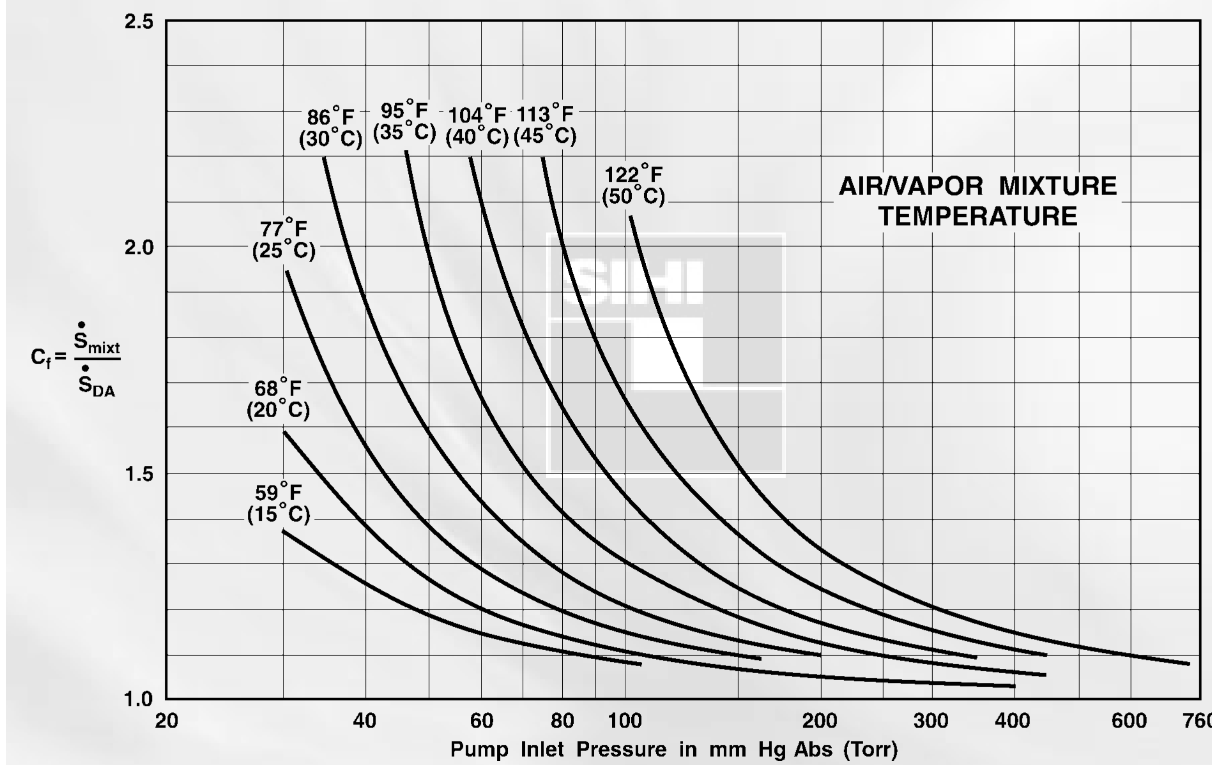
Find pump capacity ($\dot{S}_{mixt.}$) when handling saturated air at 25°C under same condition

C_f = 1.43 at 49 mm Hg Abs

$\dot{S}_{mixt.}$ = 276 x 1.43 = 395 ACFM

APPENDIX 6

STERLING SIHI AVERAGE CONDENSING CORRECTION FACTORS FOR SATURATED AIR SERVICE USING 77°F (25°C) SERVICE WATER



C_f = Condensing correction factor, when service water is 25°C

$\dot{S}_{mixt.}$ = Air/vapor mixture capacity

$\dot{S}_{DA}$ = Dry air capacity, from data book, based on 25°C service water and 20°C air

$\dot{S}_{mixt.} = \dot{S}_{DA} \times C_f$

Example: STERLING SIHI pump model LPH 65320 driven at 1750 RPM is rated at 289 ACFM ($\dot{S}_{DA}$) dry air at 75 mm Hg Abs (27" Hg vac) and when using 25°C service water.

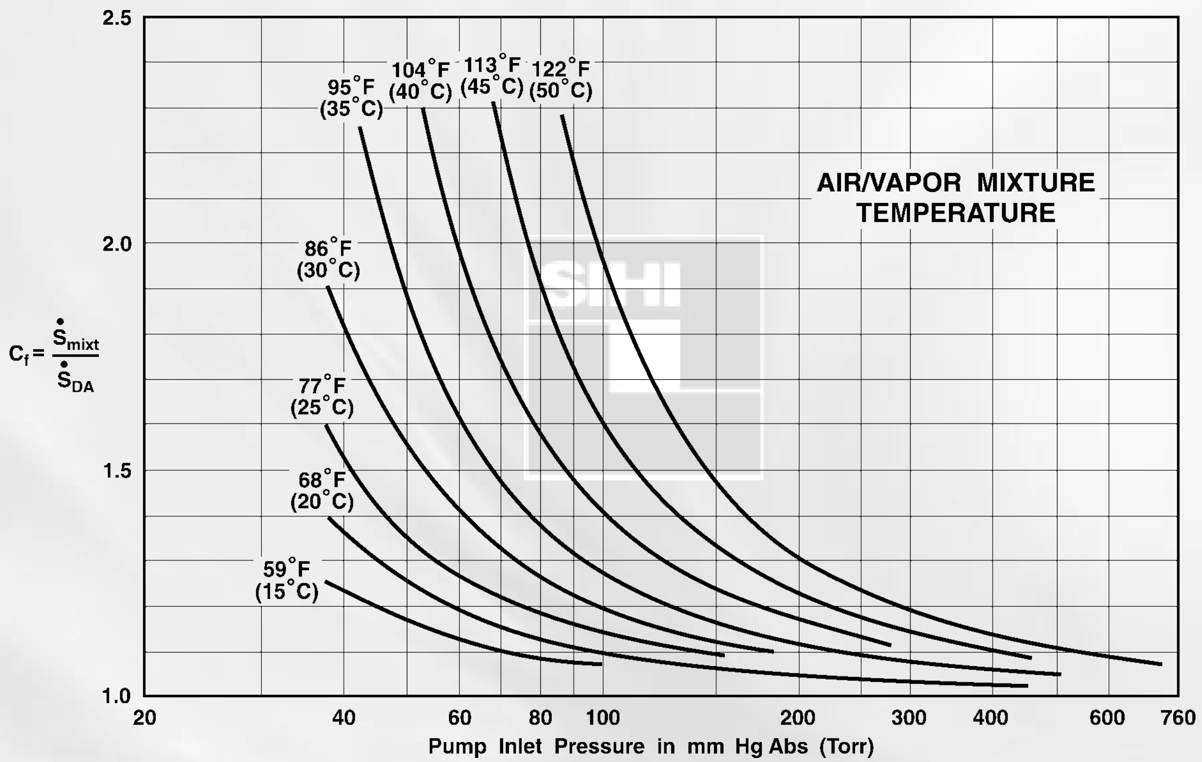
Find pump capacity ($\dot{S}_{mixt.}$) when handling saturated air at 30°C under same condition

C_f = 1.32 at 75 mm Hg Abs

$\dot{S}_{mixt.}$ = 289 x 1.32 = 381 ACFM

APPENDIX 7

STERLING SIHI AVERAGE CONDENSING CORRECTION FACTORS FOR SATURATED AIR SERVICE USING 86°F (30°C) SERVICE WATER



C_f = Condensing correction factor, when service water is 30°C

$\dot{S}_{mixt.}$ = Air/vapor mixture capacity

$\dot{S}_{DA}$ = Dry air capacity, from data book, based on 30°C service water and 20°C air

$\dot{S}_{mixt.} = \dot{S}_{DA} \times C_f$

Example: STERLING SIHI pump model LPH 65320 driven at 1750 RPM is rated at 335 ACFM ($\dot{S}_{DA}$) dry air at 75 mm Hg Abs (27" Hg vac) and when using 30°C service water.

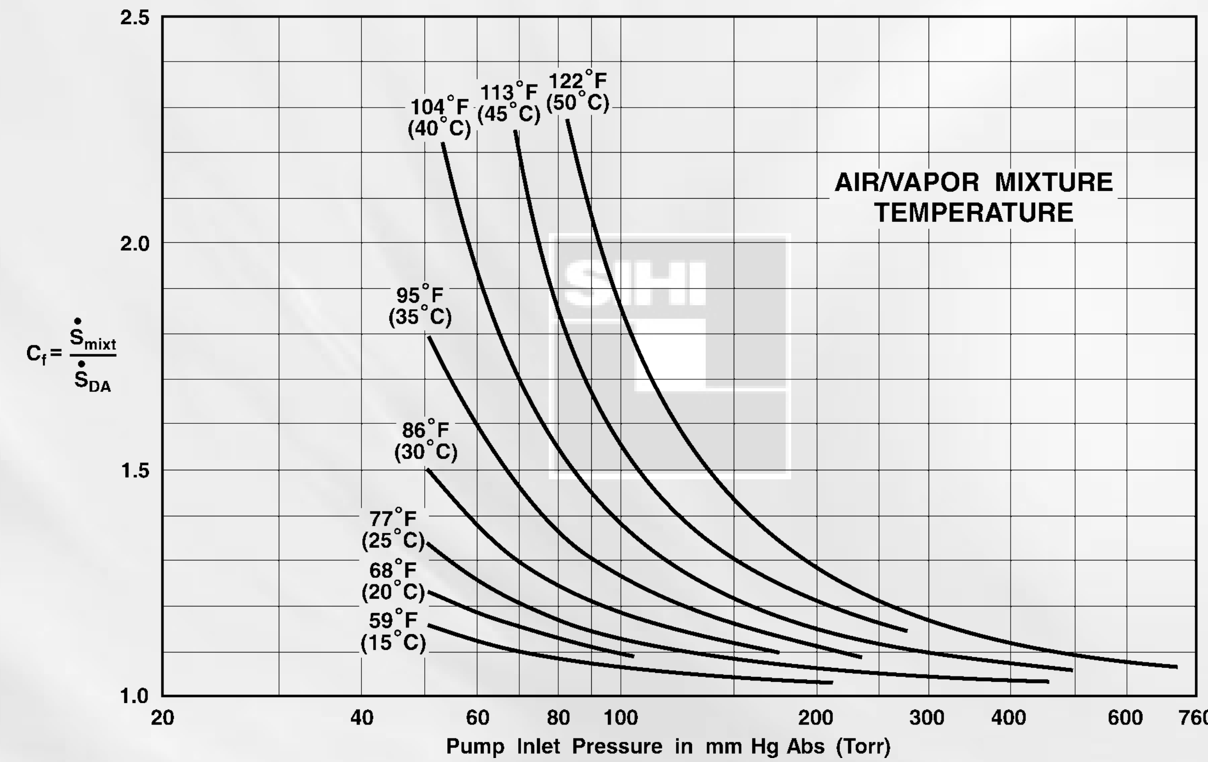
Find pump capacity ($\dot{S}_{mixt.}$) when handling saturated air at 30°C under same condition

C_f = 1.3 at 75 mm Hg Abs

$\dot{S}_{mixt.}$ = 335 x 1.3 = 435.5 ACFM

APPENDIX 8

STERLING SIHI AVERAGE CONDENSING CORRECTION FACTORS FOR SATURATED AIR SERVICE USING 95°F (35°C) SERVICE WATER



C_f = Condensing correction factor, when service water is 35°C

$\dot{S}_{mixt.}$ = Air/vapor mixture capacity

$\dot{S}_{DA}$ = Dry air capacity, from data book, based on 35°C service water and 20°C air

$\dot{S}_{mixt.} = \dot{S}_{DA} \times C_f$

Example: STERLING SIHI pump model LPH 70540 driven at 975 RPM is rated at 828 ACFM ($\dot{S}_{DA}$) dry air at 252 mm Hg Abs (20" Hg vac) and when using 35°C service water.

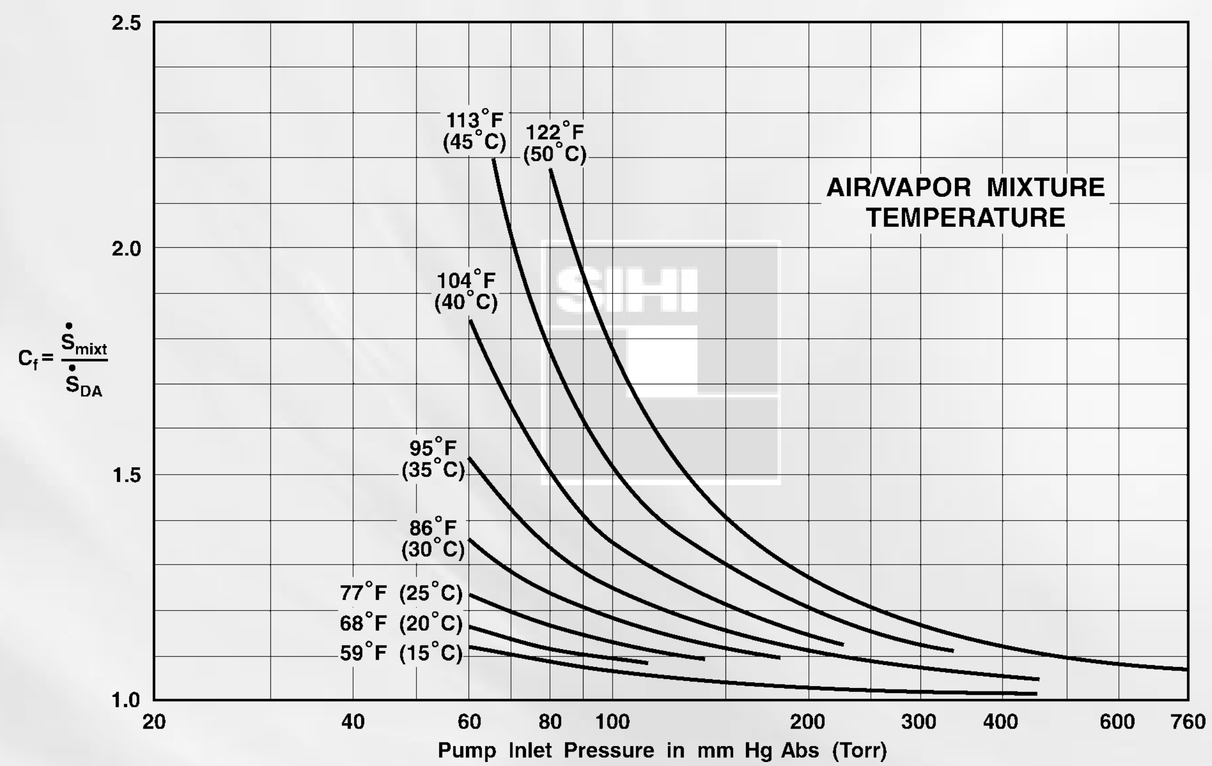
Find pump capacity ($\dot{S}_{mixt.}$) when handling saturated air at 40°C under same condition

C_f = 1.12 at 252 mm Hg Abs

$\dot{S}_{mixt.}$ = 828 x 1.12 = 927 ACFM

APPENDIX 9

STERLING SIHI AVERAGE CONDENSING CORRECTION FACTORS FOR SATURATED AIR SERVICE USING 104°F (40°C) SERVICE WATER



C_f = Condensing correction factor, when service water is 40°C

$\dot{S}_{mixt.}$ = Air/vapor mixture capacity

$\dot{S}_{DA}$ = Dry air capacity, from data book, based on 40°C service water and 20°C air

$\dot{S}_{mixt.} = \dot{S}_{DA} \times C_f$

Example: STERLING SIHI pump model LPH 60520 driven at 1750 RPM is rated at 308 ACFM ($\dot{S}_{DA}$) dry air at 252 mm Hg Abs (20" Hg vac) and when using 40°C service water.

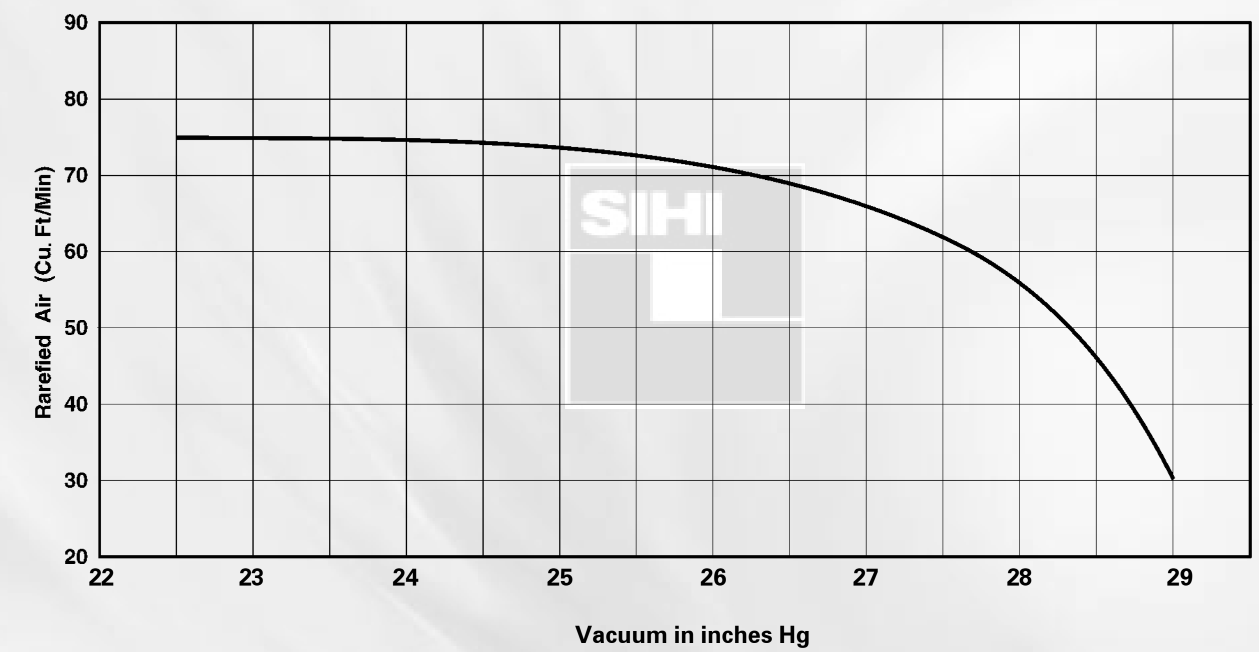
Find pump capacity ($\dot{S}_{mixt.}$) when handling saturated air at 40°C under same condition

C_f = 1.11 at 252 mm Hg Abs

$\dot{S}_{mixt.}$ = 308 x 1.11 = 342 ACFM

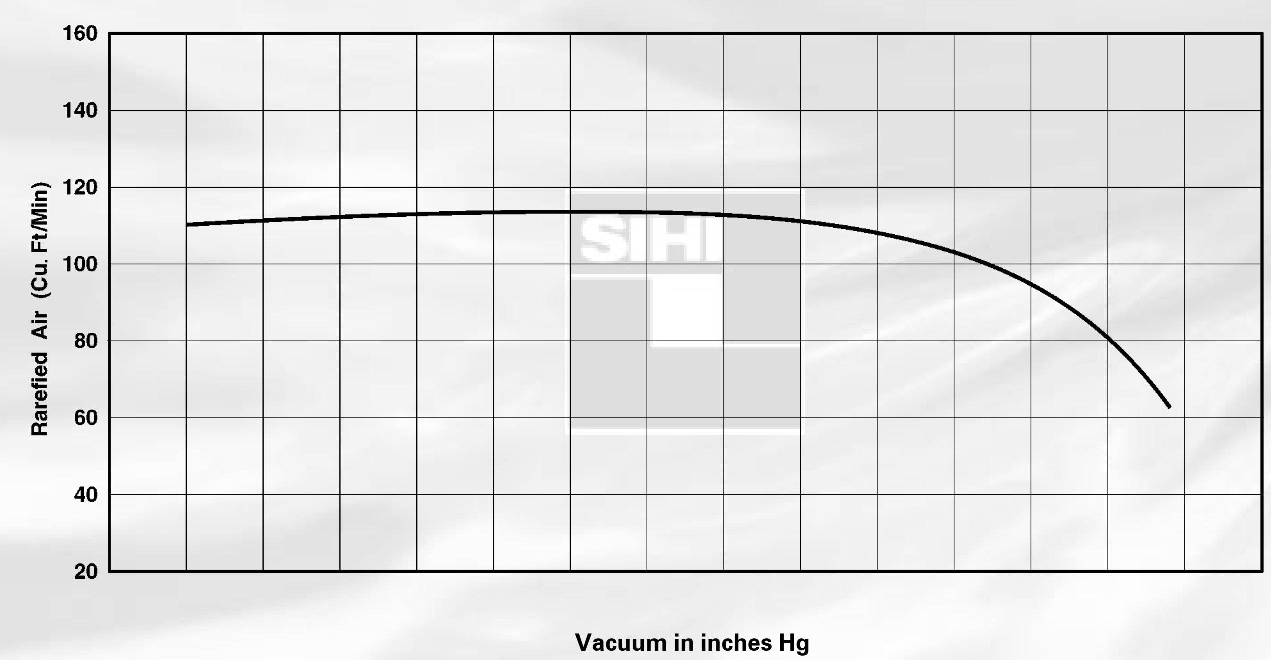
APPENDIX 10

PERFORMANCE CURVE FOR LPH 3708 @ 1750 RPM



APPENDIX 11

PERFORMANCE CURVE FOR LPH 45312 @ 1750 RPM



APPENDIX 12

PERFORMANCE CURVE FOR
LPH 45317 @ 1750 RPM

APPENDIX 13

PERFORMANCE CURVE FOR
LPH 55312 @ 1750 RPM

APPENDIX 14

PRESSURE CONVERSION CHART

STARTING UNIT	TO CONVERT STARTING UNIT MULTIPLY BY:					
	PSI	kPa	Bar	in. Hg	mm Hg	atm
PSI	1	6.895	6.895×10^{-2}	2.036	51.715	6.805×10^{-2}
kPa	0.145	1	.01	0.295	7.50	9.87×10^{-3}
Bar	14.504	100	1	29.5	750	.987
in. Hg	0.491	3.39	0.0339	1	25.4	3.342×10^{-2}
mm Hg	1.93×10^{-2}	0.133	1.333×10^{-3}	3.94×10^{-2}	1	1.316×10^{-3}
mbar	.014504	.1	.001	.0295	.75	9.869×10^{-4}

STERLING

NOTES:

MANUFACTURING AND SALES PROGRAM:

- **Liquid Ring Vacuum Pumps**
  - Single Stage
  - Two Stage
- **Close Coupled Liquid Ring Vacuum Pumps**
- **Air Ejectors**
- **Liquid Ring Compressors**
  - Single Stage
  - Multi Stage
- **SIHI*dry* Vacuum Pumps**
- **Standard Vacuum & Compressor Packages**
  - Once Through
  - Partial Recirculation
  - Total Recirculation
- **Oil Sealed Vacuum Packages**
- **Custom Vacuum & Compressor Packages**
  - Hybrid Steam Jets
  - Hybrid Blowers

ENGINEERING FUNDAMENTALS

Sterling Fluid Systems (USA)
303 Industrial Blvd.
Grand Island, NY 14072
Telephone: (716) 773-6450
Fax: (716) 773-2330

Sterling Fluid Systems (Canada) Ltd.
225 Speedvale Avenue W.
Guelph, Ont. Canada N1H6L8
Telephone: (519) 824-4600
Fax: (519) 824-7250

STERLING SIHI

Members of the Sterling Fluid Systems Group
www.sterlingfluidsystems.com