How to Interpret and Use Performance Curves

Selecting regenerative turbine pumps and motors, and calculating NPSH

Reading the Graphs

All performance data in this section is plotted on a coordinate system. **Capacity** runs left to right along the bottom in gallons per minute (GPM). **Head** runs upward from the vertex in feet of liquid. Two secondary vertical scales show hydraulic efficiency (percent) and power input (brake horsepower).

Plate III shows that a regenerative turbine pump has a very steep curve. An increase of as much as 231 ft of head (100 PSI of water) reduces capacity by only 7 GPM, or 22%, when the pump operates at peak efficiency.

> **Select by head first.** Because the curve is so steep, always give total dynamic head (TDH) first consideration when you select a pump.

Pump Selection

Example: A pump must supply 25 GPM at 200 PSI differential pressure, handling a liquid with a specific gravity of 1.0.

1. Convert 200 PSI to feet of head: 200 × 2.31 = **462 ft**.
2. Find 462 ft on the head scale and trace horizontally to the performance curve labeled “Capacity–Head at 1 foot NPSH.” From that intersection, trace straight down to the GPM scale. The capacity at 462 ft is **28 GPM**.

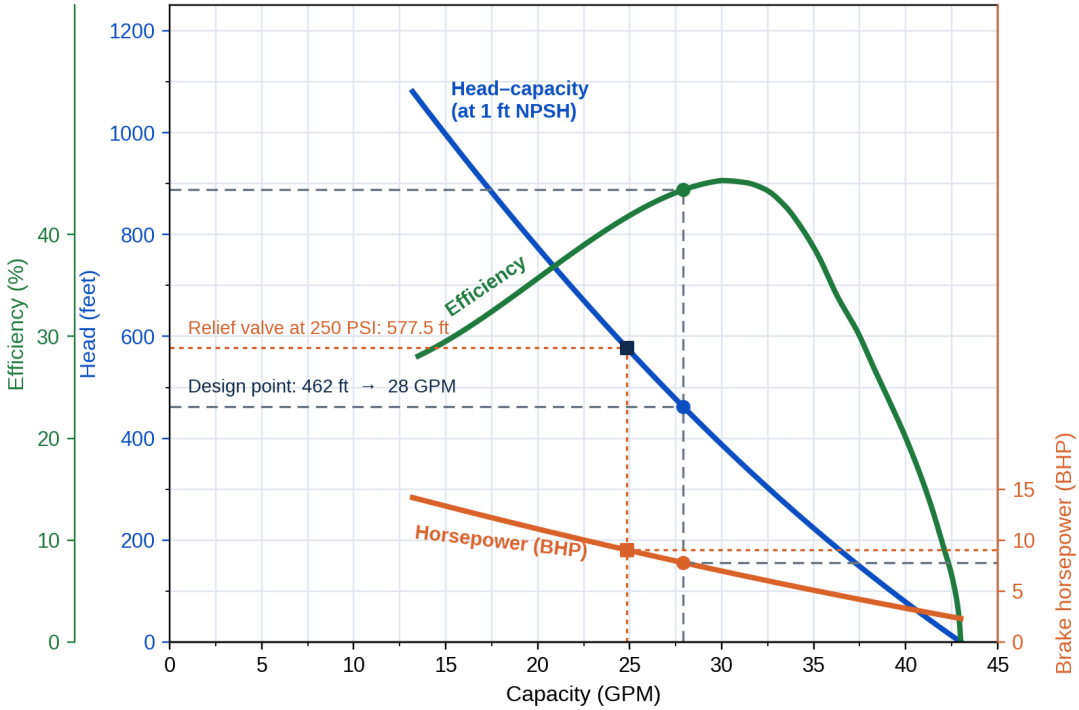
This selection is normally satisfactory because it allows about 10% surplus capacity. Most turbine pump applications run intermittently under float-switch, pressure-switch or manual control, so the extra 10% capacity increases pump life without harming the system.

Where an exact flow is required on continuous service, install a pressure-operated relief valve in the discharge line. It returns the surplus capacity to the source.

Motor Selection

1. The vertical line you traced in step 2 crosses the BHP curve directly below the head–capacity point. Trace horizontally from that crossing to the brake horsepower scale on the right. The reading is **8.0 BHP**.
2. For intermittent water pumping with an open drip-proof motor, conventional practice allows a **7½ HP** motor protected against overload by a thermal relay in the magnetic starter. The NEMA service factor for this type and size is 1.15, which permits loads up to 8.62 BHP at voltage fluctuations of ±5%. Good economics says to use the full service factor, and field experience confirms this practice is satisfactory.
3. If you need a totally enclosed motor (service factor 1.0), or the pump will run continuously, select a **10 HP** motor protected by a thermal overload relay.
4. If valving in the discharge line could restrict flow, the pump will develop higher differential pressure, as the capacity–head curve shows. Protect both pump and motor with a pressure-operated relief valve that opens and diverts flow at a set pressure. In that case you must select the larger motor.

Relief valve example (Plate III). Assume a relief valve set at 250 PSI. Convert to feet: 250 × 2.31 = 577.5 ft. Trace horizontally to the “Capacity–Head 1 foot NPSH” curve, then down to the BHP curve, then right to the BHP scale. The reading is **9.2 BHP**, so even an open drip-proof motor must be 10 HP.



Redrawn from the original Plate III (Curve No. 3998). Curves start at about 13 GPM.

PLATE III — Performance curve for the example selection (redrawn from Curve No. 3998)

Low NPSH Selection

You do not need to calculate available NPSH exactly for the equipment covered by this bulletin. A specification reading **“Pump must be suitable for 1 foot NPSH or higher”** is adequate for all models.

Give special attention to system design wherever liquefied gases are pumped below normal temperatures. Refrigerant and cryogenic pumps are in this class. The same problems can occur when you pump night-chilled liquids by day, or pump liquefied gas for storage, transfer or aerosol service in a warm building.

These problems start with vaporization in the pump, suction line or discharge line during idle periods between pumping cycles. Vapor pockets in a non-vented pump or suction line can block or interrupt liquid flow to the pump. Vapor pockets in the discharge line cause vapor binding only if they back up as far as the pump.

Design principles

- Size all pipe and fittings in the suction line for liquid velocities **below 3 feet per second**.
- Locate the pump **below the supply source** and as near to it as possible. Long suction lines cause operating problems.
- Allow **no vertical connections or risers** of any kind along the top of the suction line.
- Install a **compound gauge** in the suction line.

- Vent the pump back to the vapor chamber of the supply vessel, with both automatic and manual venting. Ask Roth for drawing A-4120 for vent pipe recommendations.
- Install an **automatic bypass line** back to the supply vessel in the discharge line.

NPSH Calculation

There are three general cases to consider when you calculate NPSH.

Case I	Supply liquid level is **above** the pump centerline (static suction head).
Case II	Supply liquid level is **below** the pump centerline (suction lift).
Case III	Liquefied gases in refrigeration systems.

Case I — static suction head

Case I is by far the most common in industry. Hot water and other liquids are pumped out of a vessel above the pump. You normally need only four elements: static suction head (h_s), vapor pressure (h_{vp}), atmospheric pressure (h_a) and friction loss (h_f). Express all values as absolute pressure converted to feet of liquid. For turbine pump capacities up to 250 GPM, you can ignore velocity head.

CASE I

$$NPSH = h_s + h_a - (h_{vp} + h_f)$$

CASE IA — closed vessel under vacuum, or liquid at raised temperature

$$NPSH = h_s - h_f$$

Here the vapor pressure equals the pressure on the liquid, so those two terms cancel.

When you plan the piping, enlarge the suction line until friction loss is under **0.5 ft** of liquid. At that level you can disregard h_f, and the formulas simplify:

CASE I (simplified)

$$NPSH = h_s + h_a - h_{vp}$$

CASE IA (simplified)

$$NPSH = h_s$$

NPSH equals the static suction head when the liquid is at its boiling point and the suction piping is large enough to hold total friction loss below 0.5 ft.

Case II — suction lift

Case II never involves liquid at its boiling point, because the result would be negative NPSH and the liquid would vaporize. Every suction-lift calculation therefore involves a liquid below its boiling point. Atmospheric pressure is the only positive force. Vapor pressure and suction lift both work against you.

CASE II

$$NPSH = h_a - h_{vp} - h_s$$

Where h_s is the suction lift. Subtract friction loss (h_f) as well if the suction line loses more than 0.5 ft.

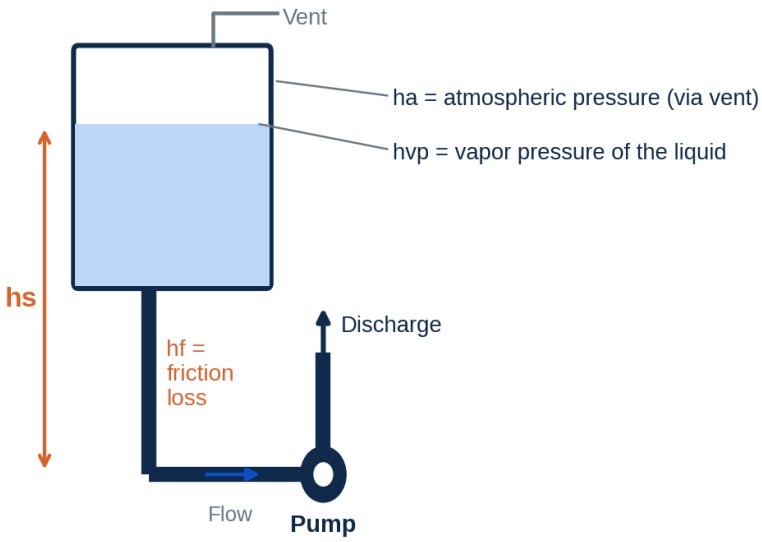


PLATE IV — Case I: liquid supply above the pump

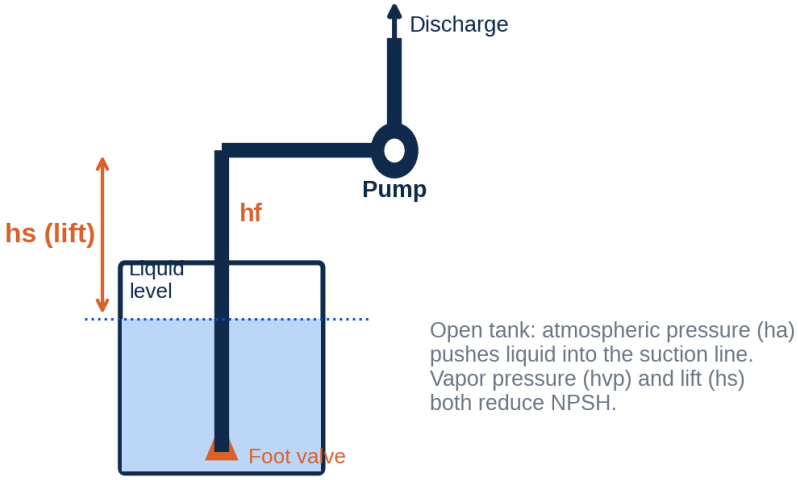


PLATE V — Case II: suction lift with a foot valve

Case III — liquefied gases and refrigerants

Case III almost contradicts the theory of NPSH. Liquid refrigerants are nearly always at their boiling point, and the supply vessel sits above the pump, so static heads of four to twelve feet are usually available. But when the compressor starts, it lowers the pressure below the boiling point and violent boiling occurs. The pump and suction line are often uninsulated, so heat from the surroundings raises the vapor pressure and increases boiling throughout the supply line, pump and vessel. The pump must therefore often handle entrained vapor.

CASE III

NPSH = 1.0 ft

Select pumps for liquid refrigerants with this characteristic in mind. Experience shows that 1.0 ft is the only NPSH you can count on in liquids at boiling point below ordinary ambient temperatures, **regardless of the static**

suction head in the physical layout. Even so, provide a **minimum static suction head of 4 ft** so gravity helps feed the vapor–liquid mixture to the pump. A few industrial pump models at 1200 RPM (Group A) also suit this service, and Roth's low NPSH pumps are excellent selections.

Efficiency

To find efficiency in Plate III, trace upward from the point where 462 ft of head meets the head–capacity curve to the efficiency curve. Then trace left to the efficiency scale. The reading is **43.5%**.

Specific Gravity Conversion

To select a pump for a liquid with a specific gravity other than 1.0, account for the liquid's density when you convert differential PSI to TDH in feet of liquid.

> **Example:** A pump is required for 10 GPM at 50 PSI differential, handling propane (sp. gr. 0.46).
>
> 50 × 2.31 = **115.5 ft** of water (sp. gr. 1.0)
>
> 115.5 ÷ 0.46 = **251 ft** of propane

You can read selections given in feet of liquid straight from the performance curves, with no conversion for specific gravity. Power input, however, comes from the curve and then needs a density correction.

Example: Model 5531 at 3500 RPM suits the propane example because it delivers the required capacity with 1 ft NPSH. The BHP reading is 2.0, so 2.0 × 0.46 = **0.92 BHP**, and a 1 HP, 3500 RPM motor works.

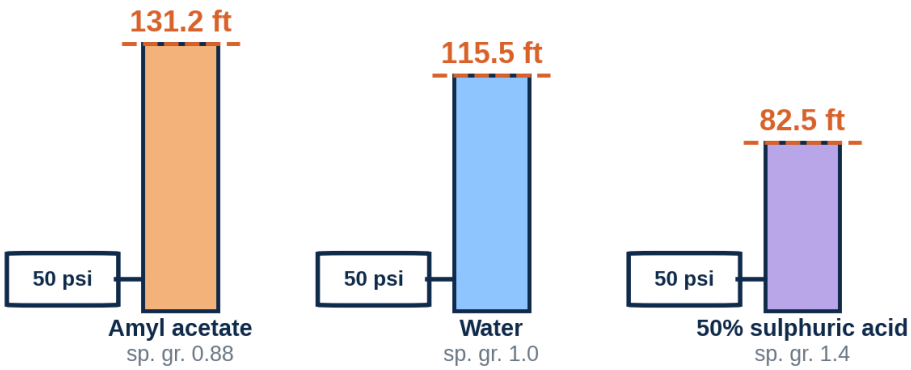


PLATE VI — Feet of head for 50 PSI at different specific gravities

NPSHA Table for Feed Water at Sea Level

Column headings 0–10 give the static suction head in feet. Table values are the available NPSH in feet, with friction loss ignored.

Temp. °F	Vapor pressure PSIA	Equiv. ft of water	0	1	2	3	4	5	6	7	8	9	10
212	14.7	34	0	1	2	3	4	5	6	7	8	9	10
210	14.1	32.6	1.4	2.4	3.4	4.4	5.4	6.4	7.4	8.4	9.4	10.4	11.4
208	13.6	31.4	2.6	3.6	4.6	5.6	6.6	7.6	8.6	9.6	10.6	11.6	12.6
207	13.3	30.7	3.3	4.3	5.3	6.3	7.3	8.3	9.3	10.3	11.3	12.3	13.3
206	13.0	30	4.0	5.0	6.0	7.0	8.0	9.0	10.0	11.0	12.0	13.0	14.0
205	12.7	29.3	4.7	5.7	6.7	7.7	8.7	9.7	10.7	11.7	12.7	13.7	14.7
204	12.5	28.9	5.1	6.1	7.1	8.1	9.1	10.1	11.1	12.1	13.1	14.1	15.1
200	11.5	26.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5
190	9.3	21.5	12.5	13.5	14.5	15.5	16.5	17.5	18.5	19.5	20.5	21.5	22.5

Decrease available NPSH by **1 ft** (to a minimum of the static suction head) for each 1,000 ft above sea level.