



## **Session 5 – “NPSH Made Simple (well, Simpler Anyway!)”**

*Aimed at Process and Mechanical Engineers and Consultant Engineers specifying pumping equipment as well as Applications & Sales Engineers selecting and quoting them. Develop an understanding of the fundamentals and practical aspects of NPSH – probably the most difficult and misunderstood concept in pumping*

*Will cover such topics as Cavitation damage, Suction Specific Speed and the “11,000 limitation”, understanding the NPSH Curve*

# NPSH Made Simple

## Contents

1.  $NPSH_A$
2.  $NPSH_R$ 
  1. What it is
  2. How we measure and test for it
3. Cavitation
  1. NPSH induced Cavitation
  2. Low Flow Recirculation induced Cavitation (suction & discharge)
4. Suction Specific Speed ( $N_{SS}$ )
  1. What it is
  2. The 11,000 limitation



# NPSH<sub>A</sub>

## What it is.

At any given temperature, all liquids have a definite pressure at which they boil. Every day we witness the fact that a liquid boils at atmospheric pressure when it reaches a sufficiently high temperature. It is important to remember also that a liquid will boil at any temperature if the pressure is reduced sufficiently. While at sea level water boils at 100°C (212°F) at the top of Mount Everest it boils at 68°C (154°F)

It is the problem of the Process & Applications Engineers to make certain that there is enough pressure on the fluid being fed to the pump so that the liquid does not boil in the suction of the pump.



# NPSH<sub>A</sub>

## Definition

**N**et

**P**ositive (means head over and above the vapour pressure)

**S**uction (at the suction flange / centreline of impeller)

**H**ead

**A**vailable

“The net positive suction head available is the total suction head in feet (meters) of liquid absolute determined at the suction flange minus the vapour pressure of the liquid in feet (meters) absolute”

$$\text{NPSHA} = \text{Suction Pressure (ft or m)} - \text{Vapour Pressure (ft or m)}$$

# NPSH<sub>A</sub>

**Arithmetically:-**

$$\text{NPSHA} = H_a - H_{\text{vpa}} \pm H_{\text{st}} - H_f$$

Where:-

$H_a$  = the head from the absolute pressure acting on the surface of the liquid

(in an open suction system this will be atmospheric pressure, in a closed system it will be the pressure in the suction vessel acting on the surface of the liquid)

$H_{\text{vp}}$  = the head from vapour pressure (always a negative value)

$H_{\text{st}}$  = Static head above the pump impeller centerline (suction flange)

(this value is negative in the case of a suction lift)

$H_f$  = Friction head in pipework (always a negative value)



# NPSH<sub>R</sub> (or 3 or 1)

## Definition

**N**et

**P**ositive (means head over and above the vapour pressure)

**S**uction (at the suction flange / centreline of impeller)

**H**ead

**R**equired (historically by convention the same as NPSH<sub>3</sub>)

**NPSH<sub>3</sub>** is measured and defined as a **3%** reduction of pump TDH

**NPSH<sub>1</sub>** is measured and defined as a **1%** reduction of pump TDH

“NPSHR is the total suction head in feet (meters) of liquid absolute measured at the suction flange / centreline of the impeller that corresponds to a 3% (sometimes 1%) reduction in discharge pressure.”



# NPSH<sub>R</sub> Why a Pump Requires a Positive Suction Head

A pressure drop occurs between the pump suction flange and the minimum pressure point within the pump impeller because of:

- 1- An increase in the velocity between the suction flange and entrance to the impeller vanes.
- 2- Friction & turbulence between the suction flange and the entrance to the impeller vanes

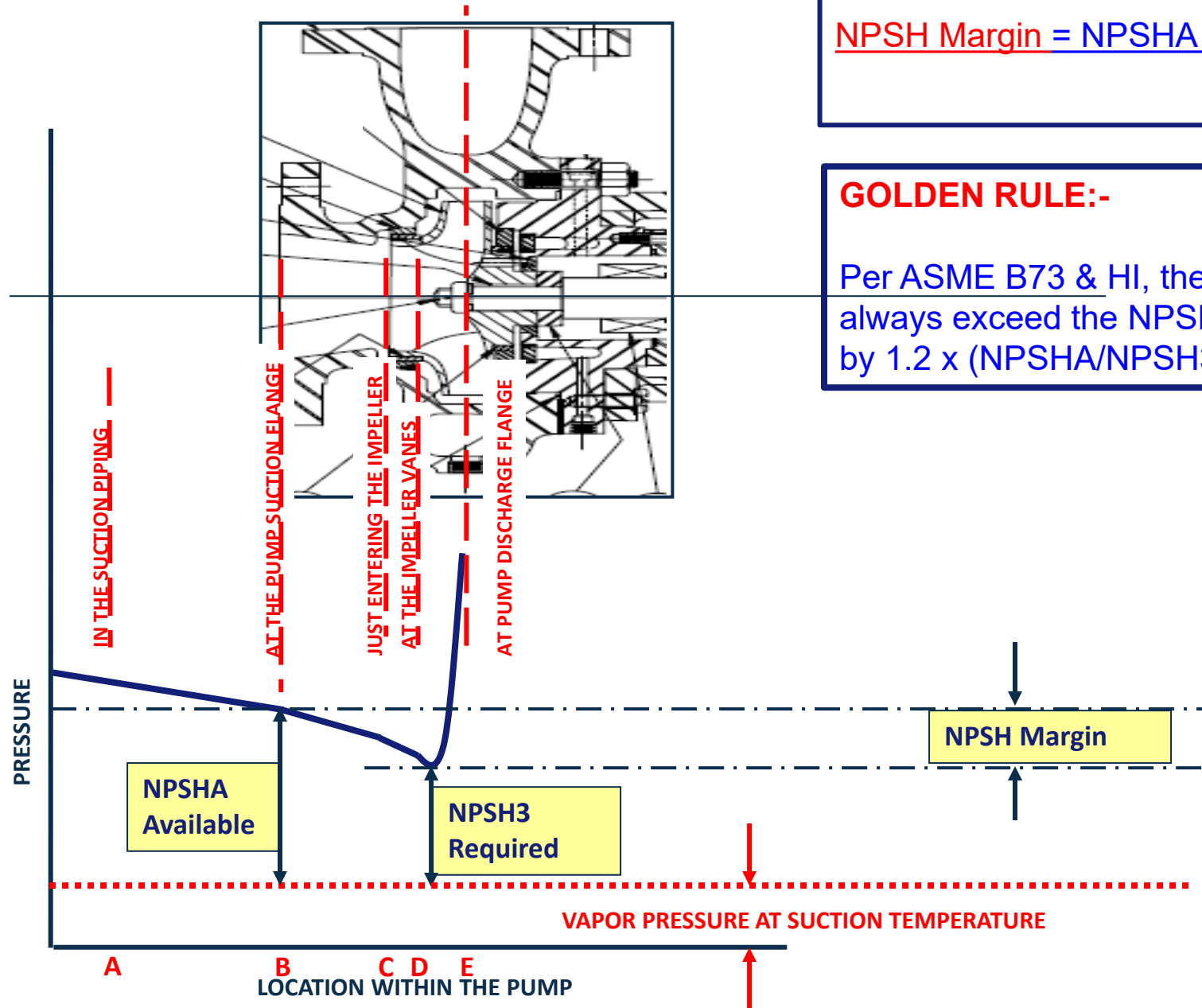
It is impossible to design a centrifugal pump in which there is no pressure drop between the suction flange and the entrance to the impeller vanes.

All pump systems must have a positive suction head sufficiently high to overcome this pressure drop within the pump and to **keep the fluid from boiling at the pumping temperature.**



# NPSH<sub>R</sub>

Visually



$$\text{NPSH Margin} = \text{NPSHA} - \text{NPSH3}$$

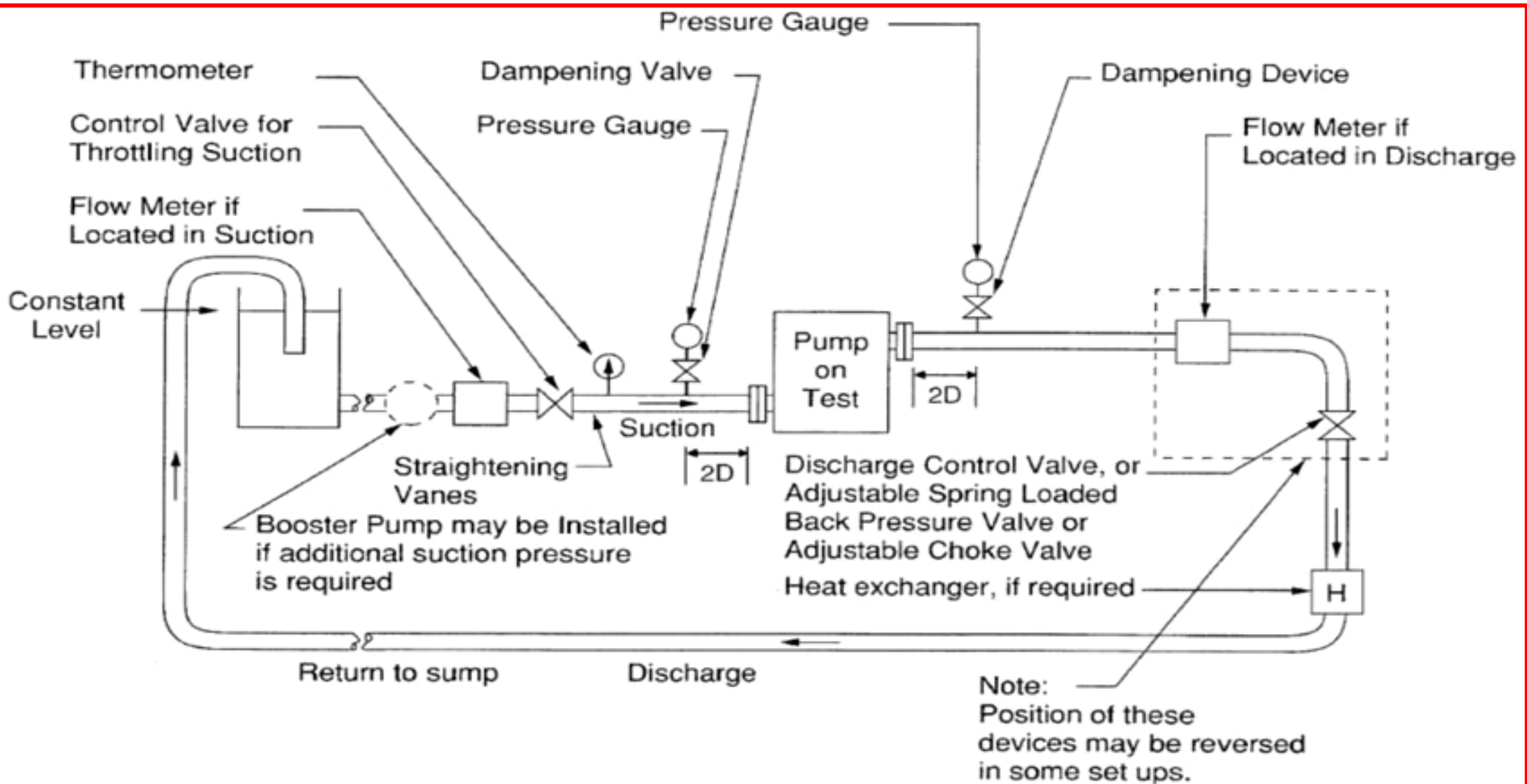
**GOLDEN RULE:-**  
Per ASME B73 & HI, the NPSHA must always exceed the NPSH3, by 3Ft (1m) or by 1.2 x (NPSHA/NPSH3).





# NPSH<sub>R</sub>

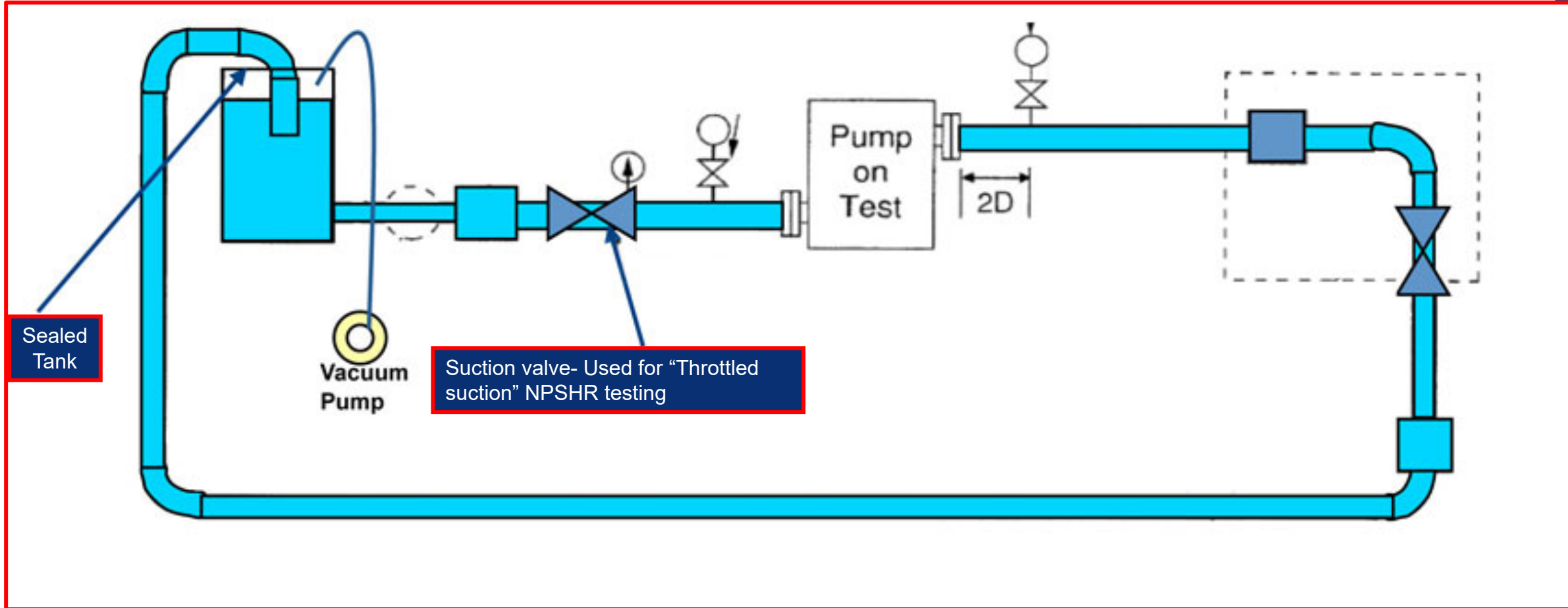
## Typical Pump Test Loop





# NPSH<sub>R</sub>

## NPSHR Testing Procedure



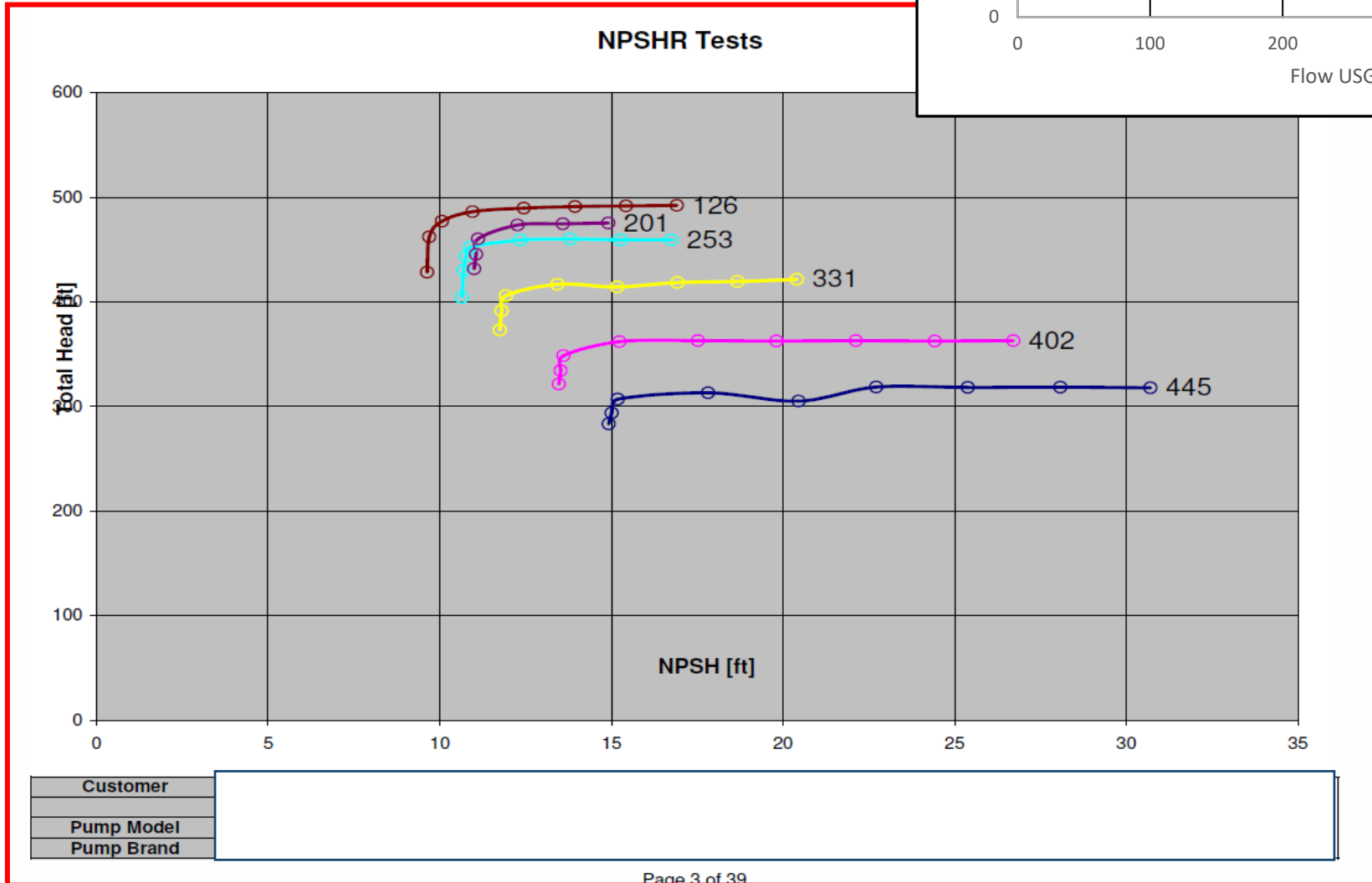
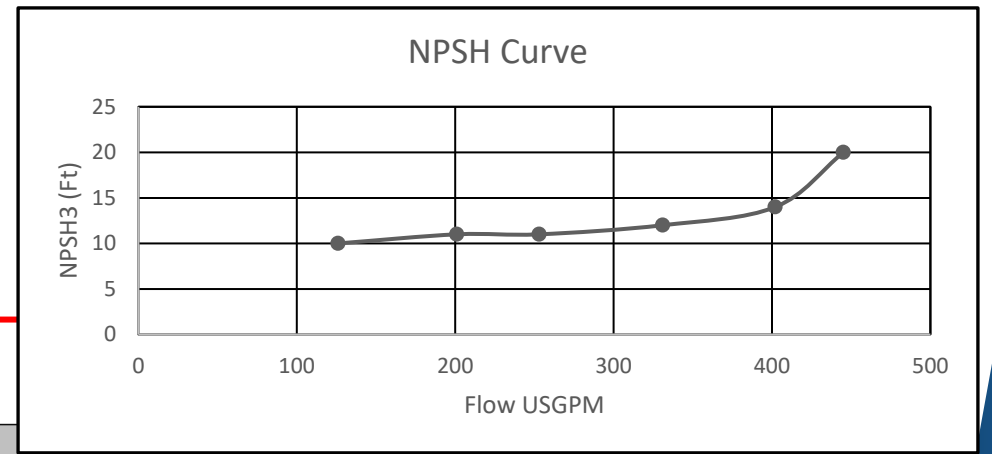


# NPSH<sub>R</sub>      NPSHR Testing Procedure

- The pump is set to the rated speed.
- At each flow point (typically 5 or 6 flows), while the Suction Pressure is reduced, the Discharge Valve is opened slightly to maintain the constant set Flowrate.
- At each NPSHR Test point, Flow, Head, Suction pressure & Water Temperature readings are taken at progressively lower and lower Suction pressures. Whether the reduction in Suction pressure is due to a Vacuum Suppression Test or a Throttled Suction valve Test.
- At some point, as the Suction Pressure is reduced more and more, the Pump Differential Head will get closer to a 3% Head reduction. So now the Suction Pressure Reduction values are taken closer together.
- After the Head drop has reached 3% at every Flowrate, the NPSH3 test is completed.



# NPSH<sub>R</sub> NPSH Test Curve





# NPSH<sub>R</sub>

## Common Misconception

"Suction cavitation occurs when NPSHa (available) is less than NPSHr (required)."

This is completely **wrong** and a misunderstanding of the physics involved in a centrifugal pump. Here is why:

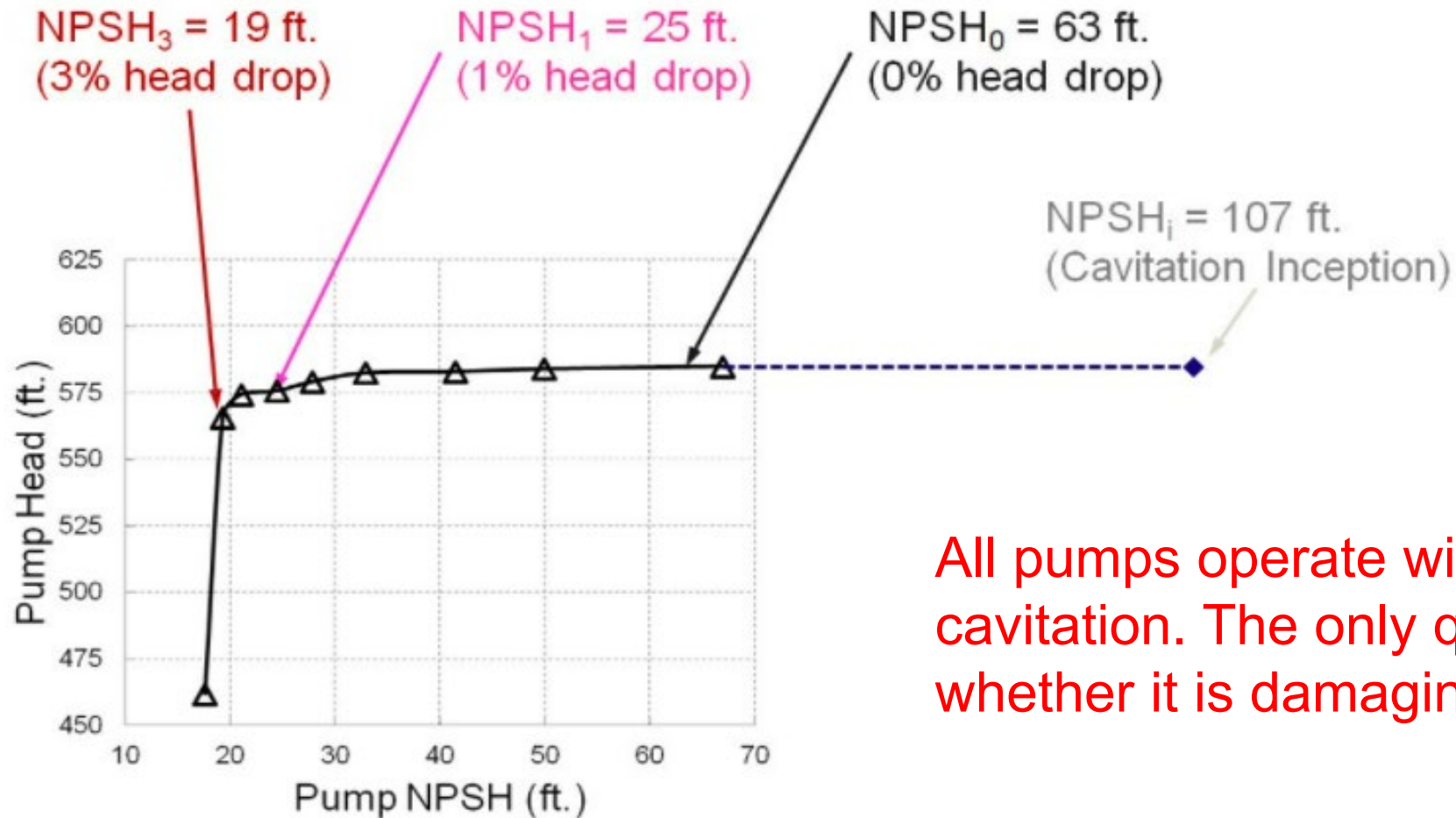
- Consider that NPSHr is normally determined where the pump head has already degraded by 3%. By that point there is already extensively developed cavitation in the impeller.
- Simply raising NPSHa slightly above NPSHr does not result in that cavitation magically disappearing. You need to go to much higher suction pressures to completely eliminate cavitation.
- For an "average" centrifugal pump, to **completely** eliminate cavitation, the NPSHa would need to be >500% of the NPSHr (specifically the NPSHi point shown on the example below).

Source – Simon Bradshaw - Director Engineering, Pumps Americas at CIRCOR



# NPSH<sub>R</sub>

## Onset of Cavitation



All pumps operate with suction cavitation. The only question is whether it is damaging cavitation.

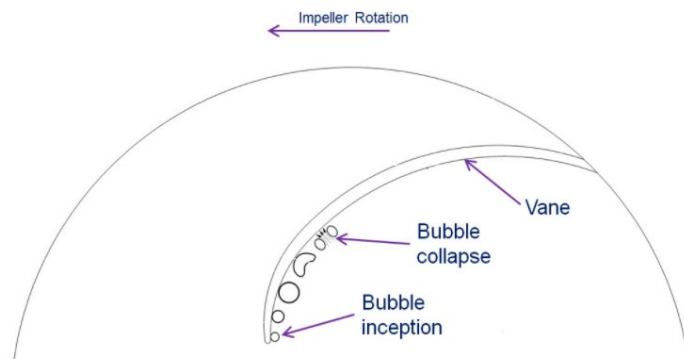


# Cavitation

As the suction head value gets closer to the  $NPSH_3$  value, vapour bubbles form on the underside of the inlet vanes of the impeller.

The closer you get to the  $NPSH_3$  value the more bubbles will form and over a larger area of vane.

As these bubbles are swept into higher pressure areas they collapse with a shock





# Cavitation

## The Cavitation Mechanism

When a Cavitation vapour bubble collapses, the instantaneous pressure of this small, high energy shock-wave is many thousands of PSI over an extremely small area.

There are two progressive shock waves that impact the metal surface against the bubbles:-

The initial MICRO-JET formed when the top surface of the bubble starts to collapse.

Immediately after this micro-jet, the whole surface of the bubble then collapses & returns to liquid form.

Shock waves are formed by collisions among the surrounding liquid molecules, that rush in to fill the void caused by the collapsing bubble.

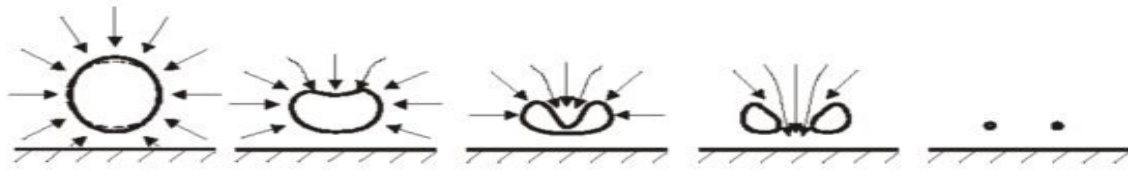
Research has shown that the life span of a Cavitation bubble from formation to collapse, is about two milliseconds (two one-thousands of a second), so this event occurs very rapidly.

The more rapidly the surrounding Liquid collides, the greater is the energy of the damaging shock-wave & micro-jet.



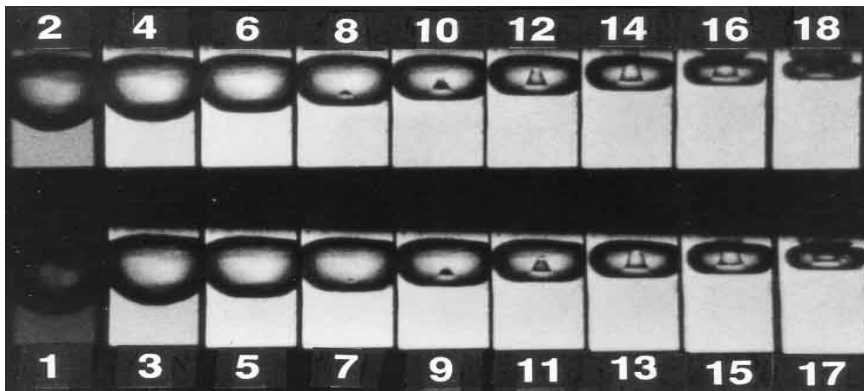
# Cavitation

## The Cavitation Mechanism



**This illustration shows the progression of the vapor bubble collapse.**

These ultra-high speed Laboratory photos below show this progression:-



Below is a photo showing the micro-burst jet, just before the final collapse:-





# Cavitation

## Extent of the Damage

The extent of the damage will depend on several factors:

- The size of the bubbles formed
- The Density of the fluid
- Thermodynamic effects (Enthalpy & Latent heat)

These combined effects comprise “Thermal Cavitation Criteria –B” (see next slide)

But simply put:

- Cold water forms big bubbles and has a high density, so the damage done when the bubbles collapse is high
- Hydrocarbons form small bubbles and have a lower density, so the damage done when the bubbles collapse is much less



# Cavitation

## Extent of the Damage – in More Detail

### COMPARITIVE VALUES OF THE THERMAL CAVITATION CRITERIA-B:-

Below we show a list of the various THERMAL CAVITATION CRITERIA-B values for Water which exhibits high cavitation damage, and Butane, a light hydrocarbon, which usually exhibits a much lower cavitation damage rate.

<u>Thermal Cavitation</u>	Water	Water	Water	Water	Butane
<u>Criteria B</u>	<u>70F</u>	<u>180F</u>	<u>212F</u>	<u>300F</u>	<u>70F</u>
<u>Value of B</u>	<b>253</b>	1.048	0.324	0.0223	<b>0.0202</b>

This shows why one of the worst liquids to produce damaging cavitation erosion is COLD WATER. As this has a very high ratio of vapour to liquid and a high density (relative to hydrocarbons which increases the kinetic energy of the cavitation micro-jet causing the erosion damage.



# Cavitation

## Hydrocarbon Correction Factor

Allowed by HI

Not allowed by API since 6<sup>th</sup> edition

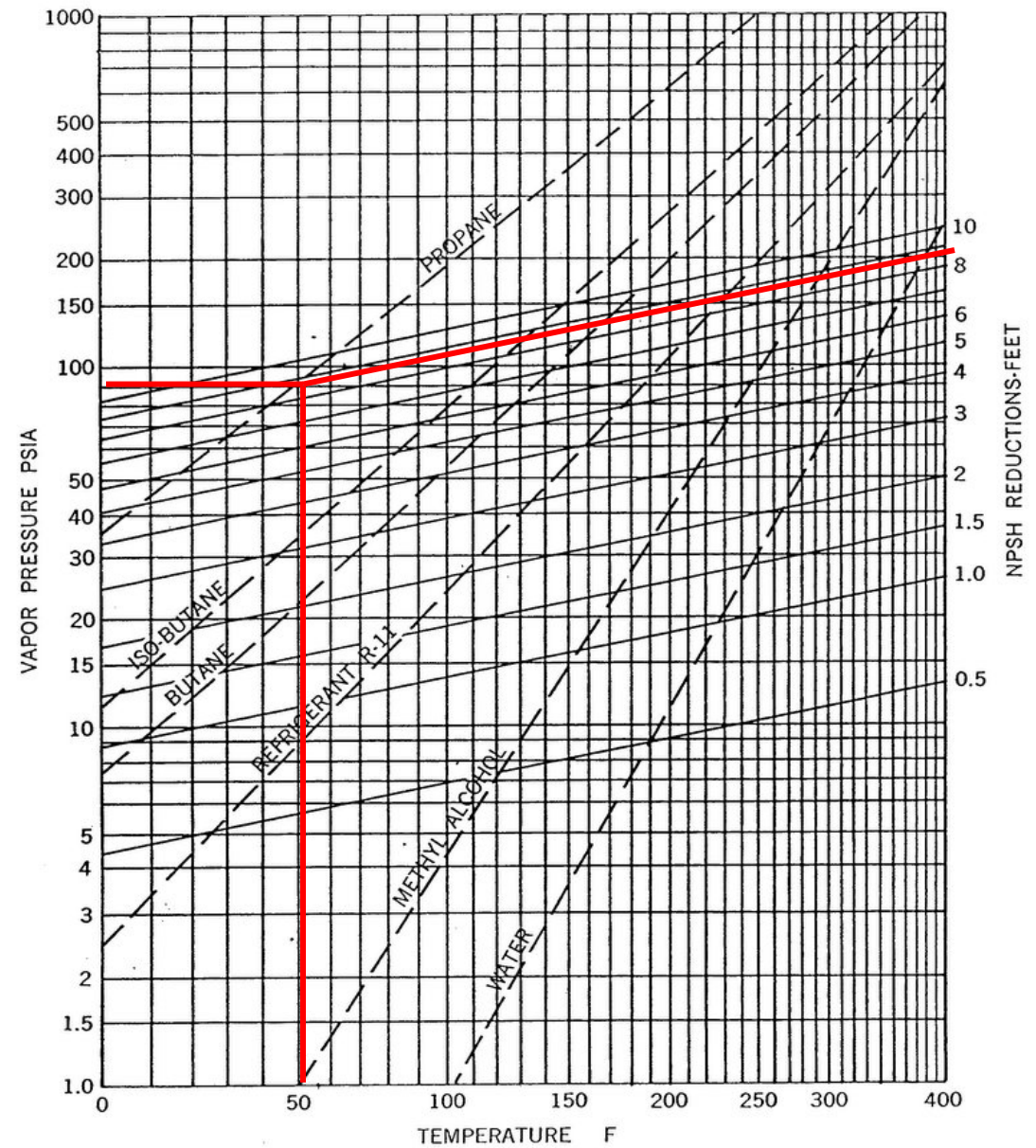


Fig. 61 NPSH REDUCTIONS FOR PUMPS HANDLING HYDROCARBON LIQUIDS AND HIGH TEMPERATURE WATER



# Cavitation

## NPSH Suggested Margins

ANSI / Hydraulic Institute Standard Para 9.6.1 addresses NPSH Margin

API Pumps in hydrocarbon service

NPSH Margin ( $NPSH_A / NPSH_3$ ) = 1.1 or 1m (3.3ft)

in the allowable operating range (AOR)

Chemical Process Pumps

with  $N_{ss} < 11,000$ , NPSH Margin ( $NPSH_A / NPSH_3$ ) = 1.1 or 0.6m (2ft)

with  $N_{ss} > 11,000$ , NPSH Margin ( $NPSH_A / NPSH_3$ ) = 1.2 or 1.0m (3.3ft)

in the allowable operating range (AOR).

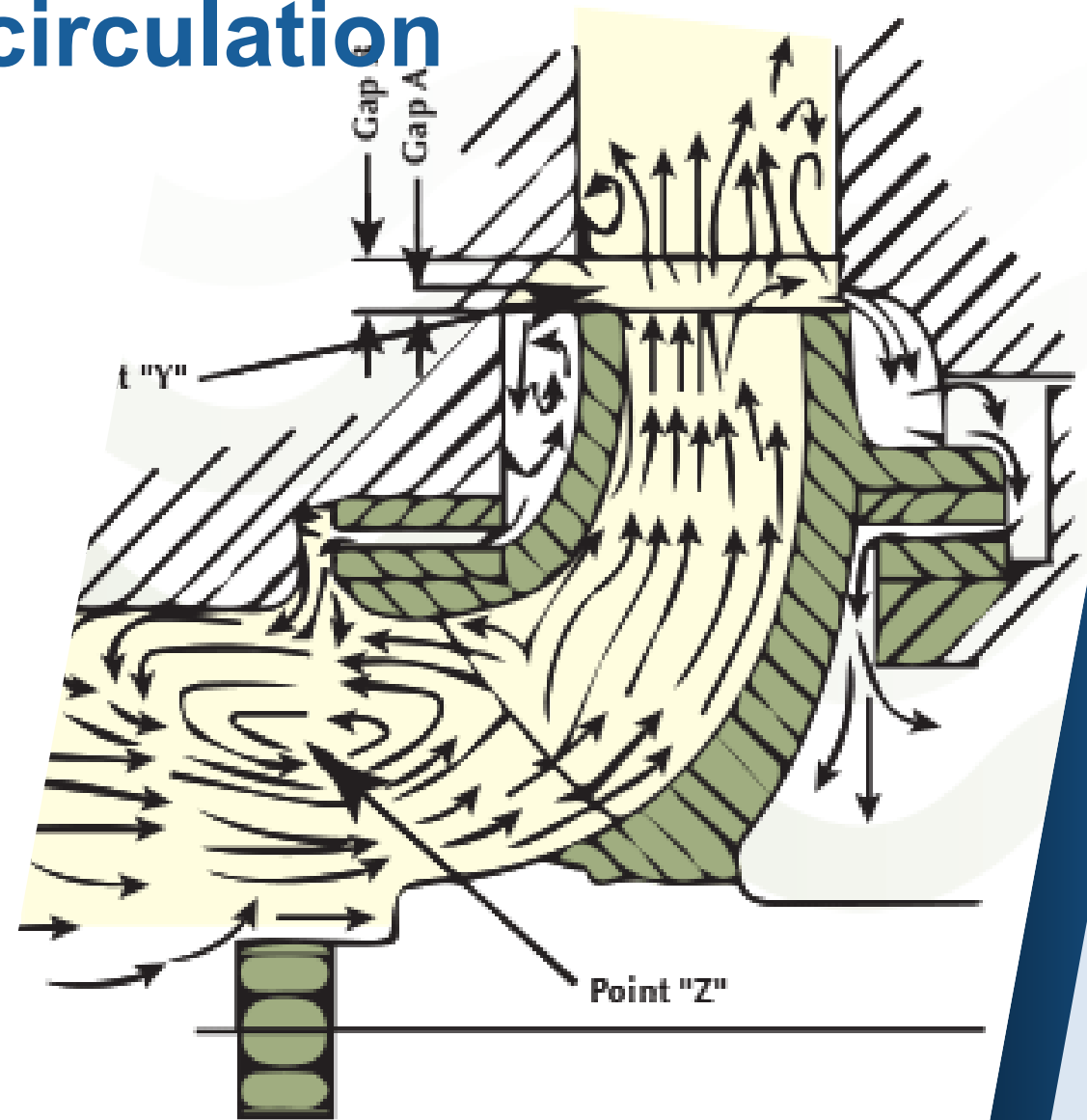
# Suction & Discharge Recirculation

## A Different kind of Cavitation

Occurs when pumps operate back on the curve from BEP

When two flow paths within a fluid are moving in opposing directions and in close proximity to each other, vortices form.

These vortices result in low pressure areas (where bubbles form) and high pressure areas (where they collapse).



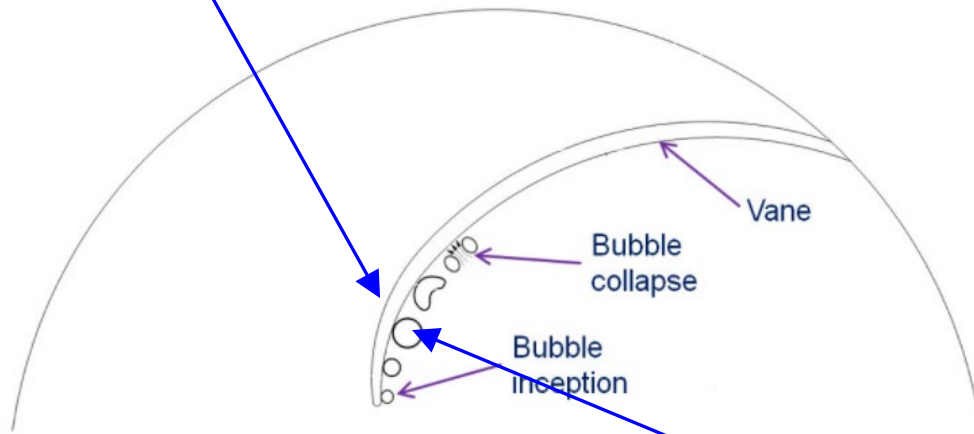
*Recirculation vortices at impeller suction eye and at vane tips (source Handbook, Igor J. Karassik and Joseph P. Messina; ISBN-10 007033)*



# Suction Recirculation

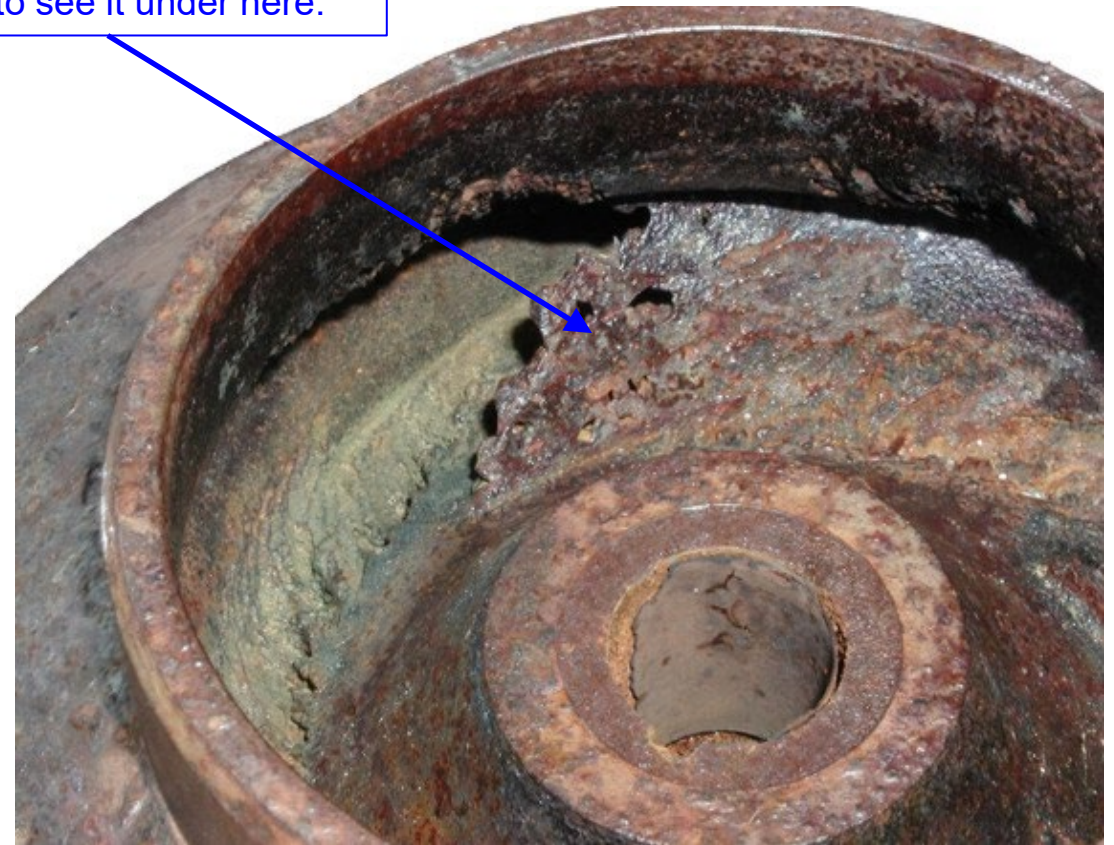
## A Different kind of Cavitation

Suction Recirculation Cavitation Damage appears on back side (pressure side) of vane



NPSH Cavitation Damage appears on front side (suction side) of vane

Suction Recirculation Cavitation Damage appears on back side (pressure side) of vane. You need a dentist's mirror to see it under here.



# Discharge Recirculation

## A Different kind of Cavitation



The central core of the Discharge Recirculation Vortex, is such a low pressure that the liquid starts to vapourise & as the bubbles move to the outer area of the vortex, where the pressure is much higher, they Cavitate (implode) against the Impeller Shrouds, and cause the hole damage shown.





# Cavitation – Which Type Have I Got?

The quickest and easiest way to identify the real cause of the noise, loss of performance, reduced power & vibration, is merely to throttle the discharge valve...

This will cause the pump flow to reduce, so that one of three events will occur, which allows the true cause to be found:-

**Event 1:-** If the rumbling & rattling sound reduces, or in fact may be eliminated entirely....

*The pump is now running at a lower flow with lower NPSHR requirements. **So this event indicates that NPSH Cavitation was the real culprit.***

**Event 2:-** If the rumbling & rattling sound, and the vibration increases.....

*This tells us that the pump is moving into a lower flow condition causing the situation to get worse. **So this event indicates that Suction and/or Discharge recirculation was the real culprit.***

**Event 3:-** OR- If the result, is that there is no change in the sound or vibration.....

*This tells us the events are unaffected by flow. **So this event indicates that Air or Gas entrainment in the Liquid was the real culprit.***



# Cavitation Damage

How can I Minimise or Mitigate the Damage?

**THE FOLLOWING MATERIALS ARE PROGRESSIVELY MORE RESISTANT TO EROSION & CAVITATION DAMAGE. STARTING WITH CAST IRON AS THE LEAST RESISTANT:-**

1. Cast Iron..... Least Resistant.
2. Leaded Bronze.
3. Cast Carbon Steel.
4. Manganese Bronze.
5. Monel.
6. Chrome and Stainless Steels:-  
    CA15 & **CA6NM (Martensitic)** & CF8M (Austenitic).
7. Cast Duplex Stainless Steels.
8. Cast Nickel Aluminum Bronze.
9. Alloyed Titanium.
10. Cast Carburized 12% Chromium Stainless and Chrome Manganese  
    Austenitic Steels.....Very Resistant.
11. Stellite coating.....Most Resistant.



# Suction Specific Speed ( $N_{SS}$ )

## Definition

You may think of it as a dimensionless number \* that describes the NPSH capability of an impeller

Defined as:-

$$NSS = N(\text{RPM}) \times Q(\text{BEP Full Dia})^{0.5} / \text{NPSH}(\text{BEP Full Dia})^{0.75}$$

$$\text{NSS}(\text{Metric}) = \text{NSS}(\text{US}) \times 1.16 \text{ (m}^3/\text{hr, m, rpm)}$$

Almost universally expressed in USGPM, Ft, RPM units

The higher the number the lower the  $\text{NPSH}_R$

Most specifications limit the allowable value to 11,000 (US units) or 12,760 (m<sup>3</sup>/hr, m, rpm)

WHY?

\* Except that it isn't actually dimensionless!



# Suction Specific Speed ( $N_{ss}$ )

## The 11,000 Limit

In the 1950's to 1980's the impeller design methods available to pump designers were more limited than they are today. Impeller designs from that era were notable for their achievement of good suction performance through the deployment of large impeller inlet diameters. It was not understood until later that the enlarging of the impeller inlet diameter caused impairment of the impeller performance at flow rates lower than the best efficiency point (BEP). This impairment exhibited itself as significantly increased vibration, Suction Recirculation, and in some extreme cases an unstable NPSHr characteristic.

- Warren Fraser – “Flow Recirculation in Centrifugal Pumps” (1981)

<https://oaktrust.library.tamu.edu/handle/1969.1/163728>

In 1981 Warren Fraser published a paper which brought the consequences of relying on large impeller inlet diameters into focus.

Pump users had already become increasingly concerned that while such designs minimized plant 1st cost, it was at the price of reliability and overall life cycle cost.



# Suction Specific Speed ( $N_{ss}$ )

## The 11,000 Limit

- Jerry Hallam – “Centrifugal Pumps: Which Suction Specific Speeds are Acceptable?”, Hydrocarbon Processing, April 1982

In 1982 Jerry Hallam published the results of a large scale reliability study of 480 pumps over a 5 year period at the Amoco Texas City refinery. He found that the reliability of a pump was meaningfully related to its suction specific speed ( $N_{ss}$ ). Specifically pumps with a  $N_{ss} > 11,000$  ( $S > 12760$ ) failed twice as often compared to lower suction specific speed pumps.

Hallam concluded: "This study indicates that caution should be exercised when purchasing hydrocarbon or small water pumps with a  $N_{ss}$  greater than 11,000 unless operation is closely controlled near BEP."



# Suction Specific Speed ( $N_{ss}$ )

## The 11,000 Limit

- Lobanoff & Ross, “Centrifugal Pumps: Design & Application 2<sup>nd</sup> Edition Fig 8-7”

Lobanoff & Ross carried out testing in 1985 which supported this limitation based on the then current state of impeller design.

They tested a range of impellers with differing suction specific speeds from  $N_{ss} = 7000$  to  $N_{ss} = 20,000$ . For each impeller the flow was varied until the pump vibration level exceeded the API 610 allowable level of 0.3 inches/sec (7.6 mm/s) peak.

The testing showed a strong correlation of  $N_{ss}$  & vibration.

The limit of 11,000 was widely adopted as a hard limit in the oil and gas industry and it is still rare to see a specification that does not invoke it.



# Suction Specific Speed ( $N_{ss}$ )

## The 11,000 Limit

Impeller design has improved by leaps and bounds since the 1980s and many authors have published papers to this effect.

Central to their claim was the premise that modern impeller design techniques allowed the attainment of higher suction specific speeds without relying solely on enlargement of the impeller eye.

- Bradshaw, Simon; Cowan, David; Liebner, Thomas (2013). “Influence Of Impeller Suction Specific Speed On Vibration Performance.”

<https://oaktrust.library.tamu.edu/handle/1969.1/162554>

In 2013 Bradshaw, Cowan & Liebner of ITT Goulds repeated the Lobanoff & Ross study from 1985 using modern impeller designs and pump construction standards. It is a very interesting read and I would recommend it to you all. I provide a link at the end of this presentation.

It would certainly suggest a limit of 13,000 rather than 11,000 and probably as high as 14,776.



# Sources & References

1- Simon Bradshaw - Director Engineering, Pumps Americas at CIRCOR, formerly Sulzer Pumps & ITT Goulds Pumps

“No You Can’t Prevent Cavitation”

<https://www.linkedin.com/pulse/you-cant-prevent-cavitation-simon-bradshaw/?trackingId=T9h7yxZWQFmQiQJ3IISIPA%3D%3D>

2- Ian James – Engineering Manager variously at David Brown Bingham, Dresser Rand, HMD Pumps, RuhRPumpen, Best Pump Works, AMEC, and DXP & Pumpworks.

3- Warren Fraser – “Flow Recirculation in Centrifugal Pumps” (1981)

<https://oaktrust.library.tamu.edu/handle/1969.1/163728>





# Sources & References

4- Jerry Hallam – “Centrifugal Pumps: Which Suction Specific Speeds are Acceptable?”, Hydrocarbon Processing, April 1982

5- Lobanoff & Ross, “Centrifugal Pumps: Design & Application 2<sup>nd</sup> Edition Fig 8-7”

6- Bradshaw, Simon; Cowan, David; Liebner, Thomas (2013). “Influence Of Impeller Suction Specific Speed On Vibration Performance.”

<https://oaktrust.library.tamu.edu/handle/1969.1/162554>



## Coming Attractions 😊

There will be a break in these Short Courses during the summer holiday period of July and early August. We will recommence at the end of August.

### “Mechanical Seals & Seal Systems”

Thur 26<sup>th</sup> August – 08.00 (UK BST) (Eastern Hemisphere) & 17.00 (UK BST) (Western Hemisphere)

*Aimed at Process and Mechanical Engineers and Consultant Engineers specifying pumping equipment as well as Applications & Sales Engineers selecting and quoting them. Develop an understanding of the fundamentals of sealing technology, the types of seals available and their associated sealing support systems (piping plans).*

Future subjects in preparation include:

- Double Case Pumps (Barrel Pumps – BB5)
- Fire Pump Systems & Packages