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WHAT CONSTITUTES "HIGH ENERGY" IN CENTRIFUGAL PUMPS?

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ABSTRACT

Every engineer has his own idea of what makes a centrifugal pump a high energy pump, but getting 2 or more engineers to agree on a definition of "High Energy" is difficult. Is a pump high energy because it runs at > 6000 rpm, or because it consumes more than 4MW, or because it produces > 6000m (20,000 ft) of head? Or, is it high energy because it moves more than 34,000 m3/h (150,000 GPM), but at lower pressure?

This is a collection of the results of recent literature searches in a quest to define this term; a term everybody knows and uses, but can't agree on the boundaries of its definition.

For most readers, it may be years between such applications. Experiences can get lost in organizations over time. We hope the following pages will help document some parameters for future reference, thus hopefully saving the reader's time and helping to avoid unpleasant experiences

INTRODUCTION

In general, we all know that centrifugal pumps with fairly narrow impellers (lower Ns or Nq) can run smoothly at heads approaching 300m (1000 ft) per stage at 3600 rpm. If we increase speed, we increase dP quickly, and thus increase chances of vane pass vibration.

For radial flow impellers, as we increase flow, we increase the Nq or Ns which causes the impeller to be wider. The vane thickness increases due to width and vane stress. When we machine the OD of the impeller, we have a rather wide strip of vane passing a volute lip or diffuser vane tip. That increases the vane pass pulse intensity, which can manifest itself in vane pass vibration. There are casting and machining variations which can exaggerate those issues.

Following the above in a general fashion, there is a relationship of dP and Nq or Ns, ie, head x fluid density or specific gravity vs Nq or Ns. Since head is produced by impeller peripheral velocity, there is also a relationship of Nq or Ns and impeller peripheral velocity.

"Pump Handbook" 4th edition (Karassik, Messina, Cooper, Heald), and Dr. Johann Gülich's "Centrifugal Pumps", 2008 provide guidance on the subject. Mick Cropper suggested a Hydraulic Institute reference 1.1-1.2 2000. Some days later, Mick also found a 1982 Bingham chart. We attempt to summarize the references and make some recommendations below.

Hydraulic Institute - 2000

Following is a quotation of H.I. 1.2.6.6 from Hydraulic Institute 1.1-1.2 2000, pages 59 and 60 which we understand has been withdrawn

1.2.6.6 High-energy pump
High-energy pumps are defined as those above a certain energy level. One parameter used in determining energy level is the total head and the density (specific gravity) of the pumped fluid. The other parameter is pump specific speed, which defines pump and impeller geometry in relative terms. Specific speed is used in conjunction with developed head and specific gravity to effectively define "high energy," while avoiding the many variables involved in other specific design and application parameters.
By using these terms and relating them to general pump operating experience, a measure of "high energy" versus "low energy" pumps is defined and graphically represented. This definition, as represented in Figures 1.45 and 1.46, shows that high-energy pumps can be of low specific speed design, with relatively high total head, or of high specific speed design, with relatively low total head. The curve separating "low" and

“high” energy pumps is of nearly constant energy level. It is not a definitive separating line, but rather a broad band and pumps falling close to this line — on either side — might be considered as low or high energy.

The following symbols, their definitions and units of measure apply to the figures:
Total Head × S = pump total head (per stage for multistage pumps) meters (feet) times specific gravity (S).

Specific Speed = RPM x Q^0.5 / H^0.75

NOTE: for double suction pumps (impellers), *Q* is the total pump flow in the H.I. charts
n = speed, rpm
Q = rate of flow, m3/h (GPM)
H = total head (per stage), meters (feet)

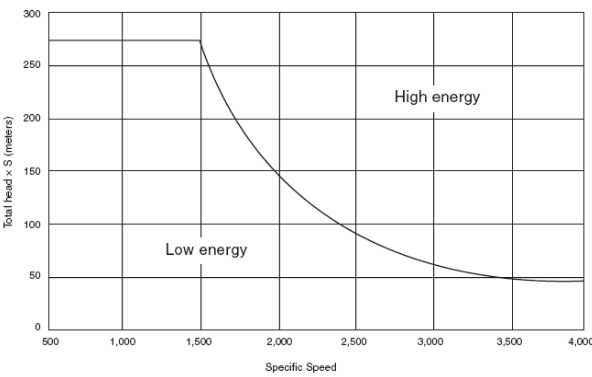


Figure 1.45 — High-energy versus low-energy pumps (metric)

Oddly, H.I. used m3/h for the Fig. 1.45 plot above, not m3/s which is normally used to calculate Nq

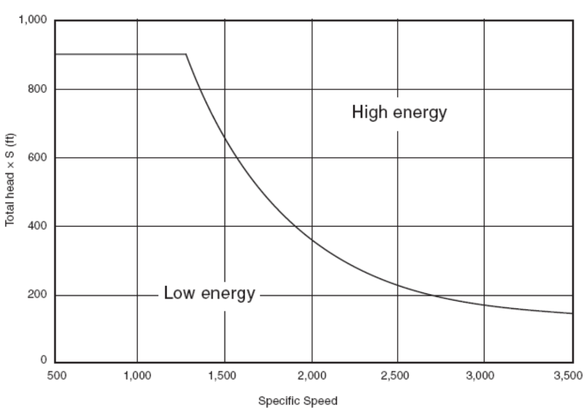


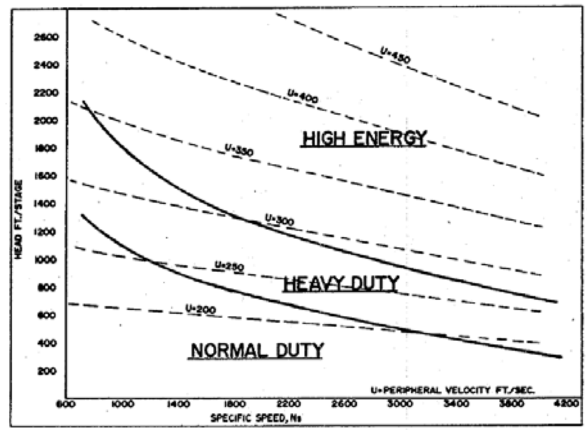
Figure 1.46 — High-energy versus low-energy pumps (US units)

Bingham 1982 Method

Mick Cropper was kind enough to forward a method for indicating high energy from a 1982 internal Bingham guideline. The intention of the guideline was to give the tendering teams an easy reference to determine when more engineering attention was needed before a pump design was proposed to customers. It is similar to all the methods above and has 3 areas instead of just 2

The chart includes impeller peripheral velocity. However, it did not multiply the head/stage x specific gravity which would seem prudent.

As noted in the H.I. paragraphs above, the lines between normal duty, heavy duty, and high energy should be changes of shading instead of lines to indicate subtle bands of change. However, the general concept seems sound.



Bingham Pump Energy Chart 1982

Ns = RPM * GPM^0.5 / (feet per stage)^0.75

Centrifugal Pumps – Johann Gülich, 2008,   Chapter 15.4

Dr. Johann Gülich kindly provides an equation (15.1 below) and plots it on his fig. 15.2 below. He uses the term "quality classes", which relate to energy density.

$$\frac{H_{st,opt}}{H_{Ref}} > 275\left(\frac{n_{q,Ref}}{n_q}\right)^{1.85}\left(\frac{\rho_{Ref}}{\rho}\right)$$

Href = 1m, Nq Ref = 25, rho ref = 1000 kg/m3

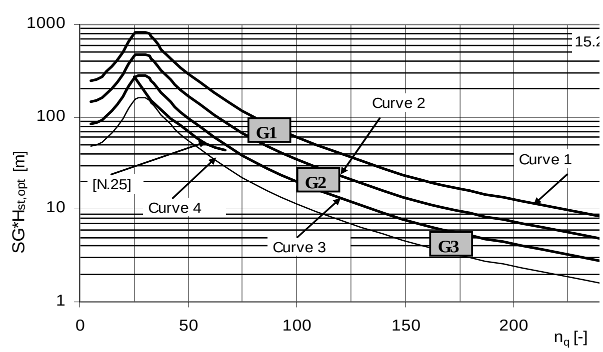


Fig 15.2   Nq = RPM*(m3/sec)^0.5 / m^0.75 where m is meters of head per stage

In fig. 15.3 chart below are his plots for quality classes for Nq vs U2 (impeller peripheral velocity in m/sec)

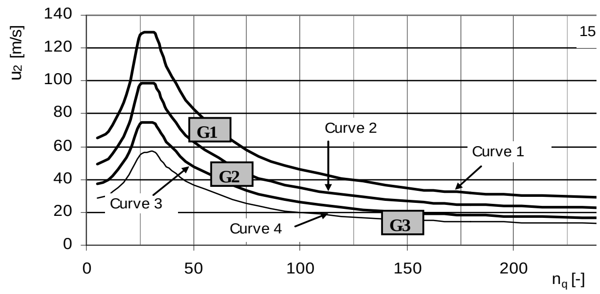


Fig 15.3 (cropped for clarity)

He proposes the idea of quality classes G1, G2, G3. He suggests close attention must be paid to impeller and volute or diffuser design and especially the final dimensions of the components, of pumps falling in the G1 or above G1 band. Obviously, the higher the energy density the less tolerant the pump will be of suboptimal dimension and vane / blade angle control. Paul Behnke's article on impeller tipping in January 2005 issue of Power Engineering provides more experiences and suggestions.

Equation 15.2 indicates what drives the rotor/stator interaction (RSI) forces. If we know an RSI force, we can then estimate the change in the RSI force when we change certain parameters. In the following equation, note that b_2* is the ratio of the impeller width to impeller diameter. Likewise, dP* is the ratio of dP to impeller peripheral head.

$$F_{dyn} \sim \frac{\rho}{2} d_2^{\ 2}\, b_2^{\ *} u_2^{\ 2} \Delta p^*$$

In other words, the RSI forces are a function of half of the density of the fluid, the square of the impeller diameter, the ratio of impeller width to diameter, the square of the impeller peripheral velocity, and the ratio of the differential pressure created per unit of impeller diameter. However, Dr. Philippe Dupont pointed out to me that

$u_2 = d_2$ x PI x N(rpm) / 60

By replacing the u_2 with d_2 x PI x N(rpm) / 60, we find that F_{dyn} ~K x d_2^4 x b_2* x N^2 * □P*
So we can also state that the RSI forces are proportional to the 4^{th} power of the impeller diameter, the square of the RPM, the width of the impeller / its diameter, and the dP / impeller diameter.

Using equation 15.1, a pump may pass under Curve 1 on chart 15.2. However, the pump also has to be checked against chart 15.3. In one of my checks, I found that some of the rather low Nq high energy water flood pumps did just that – they were under curve 1 on chart 15.2 but exceeded curve 1 in chart 15.3 due to their low Nq. In other locations some of the 1960's and 1970's steam generator feed pumps exceeded the chart 15.2 at higher Nq. Both examples are still running today albeit with improved technology.

Dr. Gülich describes the manufacturing quality aspects in more detail in the earlier Chapter 15.3.2 of his book. Tables 15.2 & 15.3 replicated below have 3 columns that pertain to the quality levels listed above: G1 (most critical), G2, or G3 (least energy density). Each of the 3 columns then provides tolerances for the following (G1 being the tightest tolerances):

Table 15.2	Tolerances for Impellers, diffusers and volutes					
Parameter	**Symbol**	**G1**	**G2**	**G3**		
Criticality (Added)		Most Critical	Medium	Least Critical		
Impeller peripheral speed, m/s,	u2	>90	40-90	<40		
Impeller peripheral eye velocity, m/s	u1	>50	15-50	<15		
Head per stage, m	Hst	>400	80-400	<80		
Power per stage, kW	P	>3000	300-3000	<300		
Fig. 15.2, 15.3 and 15.5 on previous pages provide more specific criteria for head/stage, impeller tip speed and power per stage defining the quality class. **The most stringent criterion determines the quality class.**						
Parameter		**Tolerances**			**Check With**	**Significance**
Impeller outlet width	b2	± 2.5%	± 3.5%	± 5%	Slide Gauge, Calipers	Head Stability and Excitation of Vibrations
Impeller vane distance at outlet	a2	± 2.5%	± 3.5%	± 5%	Calipers, Circular Disks	
Impeller Vane outlet angle	Beta2b	± 4%	± 7%	± 10%	Gauge, Coordinate measuring machine	
Impeller Vane Inlet angle	Beta1b, Alpha3b	± 1%	± 2%	± 3%	Gauge, Coordinate measuring machine	Cavity length, NPSHi, NPSH3
Impeller vane distance at inlet	a1	± 3%	± 4%	± 6%	Calipers, Circular Disks	NPSH3 at Q > Qsf
Impeller Inlet profile		± 4%	± 8%	-	Profile Gauge	NPSHi
Impeller outlet profile		± 5%	± 10%	-	Profile Gauge	Head, pressure pulsations
Impeller vane thickness	e	± 7%	± 10%	± 15%	Slide Gauge, Calipers	Head stresses
Throat Area of diffuser or volute	A3q	± 5%	± 7%	± 10%	Calipers, Gauges, Circular Disks	Flow rate at BEP, Efficiency, Shutoff head, Stability
Diffuser or Volute Vane distance	a3	± 5%	± 7%	± 10%	Calipers, Gauges, Circular Disks	
Diffuser or Volute Vane width	b3	± 5%	± 7%	± 10%	Calipers, Gauges, Circular Disks	

Table 15.3	Quality Requirements for impellers & diffusers (volute areas)					
Impeller shrouds machined or not		Machined	Machined	As Cast		
Hydraulic passage and shroud roughness		N8	N8 or 9	N9 or 10		
Casting process to use for impellers		Invest. or Ceramic Core	Quality Cstg or Ceramic Core	Sand Castings		
Fillet radi crticality		Ck. Very Important	Dim. Ck	Visual Ck		
Thickness variation		Ck. Very Important	Dim. Ck	Visual Ck		
NDT level of impellers (VI = visual, or PT/MT)		PT/MT	VI	VI		

NPSHi = Incipient NPSH when the vapor cavity just starts to form on the vane
NPSH3 = NPSH with 3% reduction in first stage head
Qsf = shockless flow = zero incidence angle flow
Surface Roughness: N7 = 1.6 um (63u-in); N8 = 3.2 um (125 u-in); N9 = 6.3 um (250 u-in); N10 = 12.5 um (500 u-in)
m/sec * 3600 = m3/h; m3/h*4.4 = USGPM; m*3.281 = ft
kW * 1.34 = Hp

In addition to above, Chapter 5 "Partload Operation" captures some of the damaging effects in subchapter 5.4.4. Chapter 6 addresses suction capability and cavitation. Chapter 10 "Noise and Vibration" has still other aspects to be considered. Chapter 10.7 discusses the importance of understanding interaction of the number of impeller and diffuser blades. Chapter 10.8 contains his suggestions for designing pumps with low sensitivity to vibrations

On page 874, he offers a summary:

- The quality class is a measure for the level of engineering analysis and manufacturing effort required for a specific application in order to fulfill the needs of the pump owner.
- The various criteria applied to the selection of the quality class allow taking into account the many aspects which should enter this decision
- The most stringent criterion or requirement determines the quality class.

Moving back to Chapter 15.4, he goes on to describe what creates vibration in pumps. "The vibration velocity is assumed to be proportional to the excitation force divided by the mass of the pump and the angular velocity of the rotor."

$$v_v \sim \frac{F_{dyn}}{m_{pump}\ \omega} = \frac{\rho\ d_2{}^2 b_2{}^* u_2{}^2 \Delta p^*}{2\ m_{pump}\ \omega} = \frac{d_2{}^2 b_2{}^* u_2{}^2 \Delta p^*}{2\ m_{spu}\ \omega\ g\ H_{st,opt}\ Q_{opt}}$$

Equations 15.3 above indicate that the dynamic forces are a function of the density of the fluid, the square of the impeller diameter, the width of the impeller/diameter, the square of the peripheral velocity and the dP per stage/ impeller diameter (see the previous page for the u_2 substitution discussion).That is then divided by 2 times the mass of the pump x omega (radian/sec).

Equation 15.4 below defines the specific mass = mass of the pump divided by the useful power at BEP

$$m_{spu} = \frac{m_{pump}}{P_{u,opt}} = \frac{m_{pump}}{\rho\ g\ H_{opt,tot}\ Q_{opt}}$$

Or, the specific mass of the pump as the mass of the pump divided by rho x g x TH at BEP x Q = kg/kW, where rho is fluid density kg/m3, g = 9.81 m/s2, H is total head of the pump in m, and Q is in m3/s. He plots that on figure 15.4 and then plots the quality levels against kg/kW in fig. 15.5. Table 15.4 indicates a tabular format of High, Medium and Low energy levels for the quality levels. It also indicates a difference between water and seawater.

The challenge is to estimate how much the pump will weigh before it is designed. As noted in figure 15.4, for a large 10 MW pump, the specific mass could vary from 0.6 to 2.0 kg/kW.

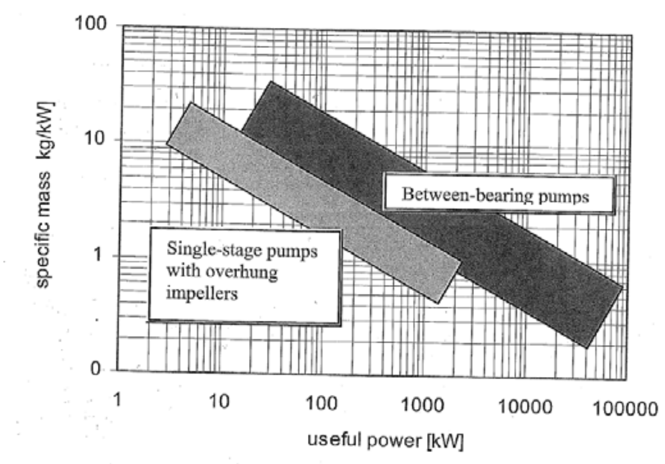


Fig. 15.4 Specific Mass (kg/kW) related to useful Power (kW)

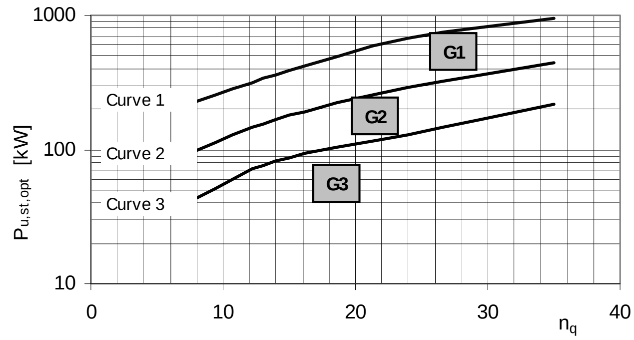


Fig. 15.5 Tentative definition of quality classes in terms of useful power per stage

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Pump Handbook, 4th Edition, 2008 (Karassik, Messina, Cooper, Heald)

Dr. Paul Cooper begins his high energy pump section on page 2.70. He discusses vane stress levels on page 2.72 and has a chart on page 2.73 that relates to energy levels. Equation (f) in table 12 on page 2.72 relates to the curve in fig. 32 on page 2.73.

dP/stg (PSI) = 78.72 / OmegaS^2 in psi, or dP/stg (Mpa) = 0.5427 / OmegaS^2. OmegaS = Ns/2733.

If you wish to use Nq to get to Mpa instead of Ns, the equation becomes dP/stg = 0.51686/OmegaS^2 in Mpa since Omega S = Nq/52.919 per equation 38.b on page 2.22

For an Ns of 1100 (Nq 21.3) which is common for multistage pumps and 1.0 sp gr water, OmegaS would be 0.402 and dP = 486 psi = 1123 feet or 342 m per stage. That seemed reasonable to me.

At Ns = 1500 (Nq 29), OmegaS would be 0.549 and dP = 261psi which is only 604 feet (184m). In his example of the 60 MW 20x25CA -4 stage boiler feed pumps, they are making around 1200 psi (8.27 Mpa) or about 2900 ft (885m) per stage at that specific speed. Those pumps were installed in 1970's in various plants and are still running today. As with any high energy pump, they have been modified to reflect improvements in technology.

The dashed line on the lower left indicates the upper limit of the low energy level due to other than diffuser inlet issues which prompted fig. 32. This is a similar indication as found in Dr. Gülich's treatment of the same issues.

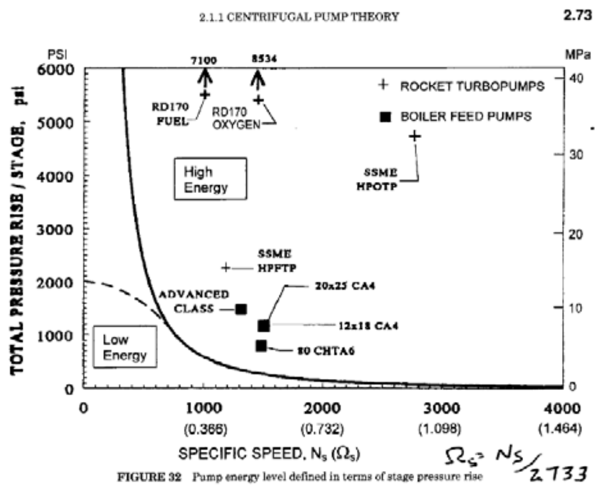


Fig. 32

Pump energy level defined in terms of stage pressure rise. Derived from EPRI CS-5857 pg. 263, 1988 and EPRI 1025730, pg 409, 1991

Dr. Cooper points out that if fig. 32 were extended to higher specific speed, relatively low head pumps would be considered high energy and thus candidates for full stress and modal analysis. From EPRI CS-5857 page 262, Fig 32 main curve is a plot of dP = 300 psi x (1400/Ns)^2

Fluid / Structure interaction is discussed further since it manifests itself in higher vibration which emanates from

a) Blade – Vane interaction - see pages 2.76 and 2.77 related to Dr. Bolleter's and later work

b) Recirculation – see pages 2.77 and 2.78. Recirculation and rotating stall can cause fatigue failure in a number of hydraulic components. The Berten – Hentschel – Kieselbach - Dupont 2011 paper referenced in the bibliography expands upon this subject. They used a 32 channel data acquistion system to capture actual pressure and strain in both the impeller and diffuser during actual full speed test conditions on water. They then were able to simulate and correlate the results using CFD and finite element models

c) Anomalous axial thrust behavior – see page 2.78 and 2.79 Thrust value and even axial thrust direction changes are possible as a high energy pump is throttled back on its curve.

d) Cavitation – Cavitation is not just the NPSHa vs NPSHr at BEP. In higher energy pumps it is often worse at lower flows. Over the years, a number of different minimum continuous stable flow calculation methods have been proposed. Gopalakrishan, Heald-Palgrave, Bingham (Sulzer -not published) are all methods that give a much better indication of MCSF than we had in 1970's. Flow visualization studies for industrial pumps (as opposed to hydroelectric turbines, pumped storage and rocket fuel pumps) became more prevalent in late 1970's. Those studies helped us all understand how to lessen cavitation damage and prolong impeller life. In high energy pumps, 40,000 hour suction stage impeller life is now common, as are incipient NPSHi curves on proposals. If one uses the equations in Dr. Gülich's book and those in Dr. Cooper's book, one will arrive at similar incipient NPSHi values – which are often measurably higher than NPSH3 values at flows less than BEP. Dr. Gülich may then suggest additional safety factor multipliers for various commercial and technical risks which could cause them to diverge. Computational Fluid Dynamics (CFD) has replaced much of the actual flow visualization studies prominent in 1970's. CFD is a computer program so will provide a response to 4 decimal places with erroneous data. Thankfully, many companies have spent years correlating the CFD code to match the observed pump test data from flow visualization and other actual tests.

Starting on page 12.397, Dr. Cooper's table of liquid rocket propellant pump performances indicates that the Space Shuttle main engine 3 stage hydrogen pump consumes 57MW (77,000 hp) but only weigh about 352 kg (775 lbs). It makes 61,000m (220,000 ft) of head at a flow of 3700 m3/h (16,300 USGPM). Its 305mm (12 in) impellers have a tip speed of about 600 m/s (1960 fps). It runs on hydrogen cooled ball bearings at 37,400 RPM, which is 533 times the API 610 limit for such a bearing system. A friend pointed out that these pumps were not designed for API 610's suggested 20 year pump life or 40,000 hour impeller life.

The Other End of the Spectrum

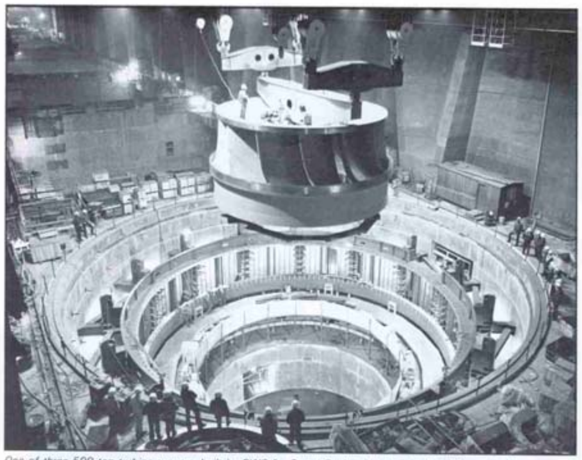
All of the above has been about lower flow, very high head, typically high speed, multistage pumps. But what about very large, relatively low head pumps at relatively low speeds?

Dr. Gülich's charts go up to Nq 250 (Ns 12,900) which is in the axial flow pump regime. At those high specific speeds, the head is relatively low, but the flow is high. Flow times head relates to power, so power levels can be still be very large.

All good papers on centrifugal pumps have to reference Stepanoff at least once. In his Centrifugal and Axial Flow Pumps, second edition, page 277 address pumped storage units as they existed up to the 1957 publish date. Connecticut Power & Light Rocky River had 2 pumps rated at 6MW (8100 hp) built in 1930. Colorado River Authority's Buchanan Dam in central Texas was rated at 10MW (13,450 hp) and built in 1949. On page 279 he has a Sulzer 2 stage cross section rated at 18.7MW (25,100 hp). On page 280, he discusses the TVA Hiwassee Dam pumped storage pumps rated at 76MW (102,000 hp) with a 22 ft diameter impeller.

The 6 irrigation pumps installed at Grand Coulee dam in 1951 were rated at 48.5MW (65,000 hp) each to lift water 85m (280 ft). Total flow was about 1,600 ft^3/sec (163,000 m3/h) (718,000 USGPM). See the internet site in the bibliography. I believe Byron Jackson (now Flowserve) built those pumps.

Some years later the turbine runner below was fabricated in the Bingham (Sulzer) shop in Portland, Oregon, USA for one of the Grand Coulee hydroturbines. It weighed about 500 tons. Each turbine was rated at about 612MW (820,000 hp). While not a pump per se, it is a hydraulic machine so the issues associated with impellers or runners are still important. As a scale reference, notice that there is a man standing on the runner.



A quick internet search provided info on more recent pumped storage projects. Duke Power commissioned the last of the Bad Creek Pumped Storage units in 1991 in Oconee County, SC, USA. Each of the 4 units is rated at 266 MW (357,000 hp) in a single stage. A Francis vane impeller lifts the water about 366m (1200 ft) when running as pumps. I recall that design work began in the 1970's. Since it is impractical to test the full size machine before it is installed, a great deal of effort went into the model design and model testing of those units and the structures surrounding them.

Each of the 6 Dominion Energy Bath County Virginia single stage pumped storage units are rated at 462MW (620,000 hp) each. When running as turbines, total flow can reach 3.3 million m3/h (14.5 million USGPM). They are the largest pumped storage units in the world according to the website and have been running since 1985.

Suggestions

The API 610 taskforce is considering an annex in the 12th edition to address these issues but will probably focus on lower specific speed multistage pumps. The head per stage, power per stage and overall discharge pressures approaching 1000Bar (14,500 psi) for deep sea water injection service, is prompting this annex. There are obviously refining and power plant services that could benefit from these approaches as well.

Until the pump is designed, it is not easy to estimate its weight. Dr. Gülich's kW vs Specific Mass (kg/kW) chart in fig. 15.4 can get us into the ballpark. Once the pump is tendered, it would be fairly straight forward to use equation 15.1 and plot it in fig. 15.2 and 15.3 to assess what quality level may be appropriate.

The ability to calculate and then plot against known criteria, helps to reduce worry and confusion, and focus attention. The authors address the criteria in slightly different ways but all add value.

Dr. Gülich's quality level check list in Table 15.2 provides a good checklist. The actual values will be supplier specific. A datasheet that the supplier could fill out with the tender may be useful to indicate specific values for each criterion.

At low Nq / high dP per stage, the vane pass pulsation energy can become troublesome. The user is urged to discuss this with the vendor and pay close attention to rotor – stator interactions, gaps and clearances. Rotating stall, expected onset of suction recirculation, NPSHi, etc, all need to be discussed. Impeller failure can occur if these subjects are not clearly addressed. Several different manufacturers can attest to this fact.

First stage impeller design can be a challenge in high speed, high energy multistage pumps. More power requires a larger shaft diameter. That drives out the hub diameter, which drives out the eye diameter (so there is enough area to pass the flow) which increases the peripheral eye velocity and speeds the onset of inlet recirculation as the pump is run back on its curve from BEP flow. Be sure to understand the NPSHi and size booster pumps to meet those needs at reduced flows and runout. Obviously, the minimum flow values will increase as well, often regardless of NPSH margin.

Understand and keep reminding yourself that these charts and graphs contain lines. The lines should really be shades of grey. The problem with equations and charts with lines is that once the value crosses the "line," the formulae may be different for no other reason than the value exceed some value by 0.01. This can lead to the wrong conclusions. Pump design has become much more exact, but there is still relevant art and experience that must be employed.

An aspect which I have not included is the impact of the pumped liquid on the material fatigue limits. With corrosive liquids (eg. seawater with high chlorine and maybe a little H2S is corrosive), the head per stage that is considered to be the limit for the high energy region should be reduced because of the reduction of the fatigue limit due to corrosion. Impellers and diffusers may have to be replaced due solely to number of hours of operation. In such cases, consultation with your trusted metallurgist is recommended.

CONCLUSION

There is no universal definition of "High Energy" in centrifugal pumps. Power to weight ratio is often used if one knows the weight of the pump. Dr. Cooper suggests it boils down to dP and torque per unit of flow. However, there are specific design aspects that need to be considered as the engineer approaches his own limit of when a pump may be considered high energy. Those may include rotor – stator interaction, NPSH3, NPSHi, suction impeller life, minimum continuous flow, rotor design, RPM, and pressure boundary design, to name a few. Since few engineers purchase such pumps routinely, the literature citied will hopefully provide some guidance. Please consult the referenced literature for updated editions, as well as other references.

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