CASE STUDIES – CS 6

Segment – Affinity Laws

Topic/ case – A process pump in operation in Singapore -

Trimming the impeller for new duty - should the trimming be done only on the vanes or both on the shrouds and vanes?

Description of the case – The customer has posed the above question to the consultant for their existing end suction process pump. The pump consultant advises the user to trim both the vanes as well as the shrouds to save energy.

Questions for discussion:

2. How is the impeller trim carried out using the affinity laws?
3. Why is it not advisable to trim the vanes alone leaving the impeller shrouds intact?
4. Are there situations where trimming only the impeller vanes is beneficial?
5. What determines the maximum trim of an impeller as a percentage of its full diameter for a specific pump type?
6. What is “alternate vane trim” and how does it alter the shape of H-Q curve?

Impeller Trim for a New Duty

Use of Affinity Laws :

A Centrifugal Pump is designed for a fixed impeller geometry and a variety of impeller diameters to define its performance characteristics for a range of flows & head.

The head and capacity produced is dependent on the peripheral velocity of the liquid exiting from the impeller. Adjusting this velocity, one can get the desired output & this can be accomplished by two ways:

- By changing the speed of rotation
- By changing the impeller Diameter

Affinity law is a set of governing rules for pump performance that expresses the mathematical relationship between the pump output & the above two parameters.

WHEN ONLY IMPELLER DIA. CHANGES & SPEED REMAINS THE SAME	WHEN ONLY SPEED CHANGES & IMPELLER DIA. REMAINS THE SAME	WHEN BOTH DIA & SPEED CHANGE
$Q_2 = Q_1 \times (D_2/D_1)$	$Q_2 = Q_1 \times (N_2/N_1)$	$Q_2 = Q_1 \times (D_2/D_1) \times (N_2/N_1)$
$H_2 = H_1 \times (D_2/D_1)^2$	$H_2 = H_1 \times (N_2/N_1)^2$	$H_2 = H_1 \times \{ (D_2/D_1) \times (N_2/N_1) \}^2$
$BKW_2 = BKW_1 \times (D_2/D_1)^3$	$BKW_2 = BKW_1 \times (N_2/N_1)^3$	$BKW_2 = BKW_1 \times \{ (D_2/D_1) \times (N_2/N_1)\}^3$

- Q1, H1, BKW1, D1 & N1 ARE CAPACITY, HEAD, INPUT POWER IN KW, IMPELLER DIA. & SPEED AT INITIAL CONDITION.
- Q2, H2, BKW2, D2 & N2 ARE CAPACITY, HEAD, INPUT POWER IN KW, IMPELLER DIA. & SPEED AT CHANGED CONDITION.

Determination of Approximate Impeller Diameter at Rated Duty

Procedure – 1

1. Extend the head scale to zero head zero flow point (0,0).

2. Plot required duty point (Q,H).

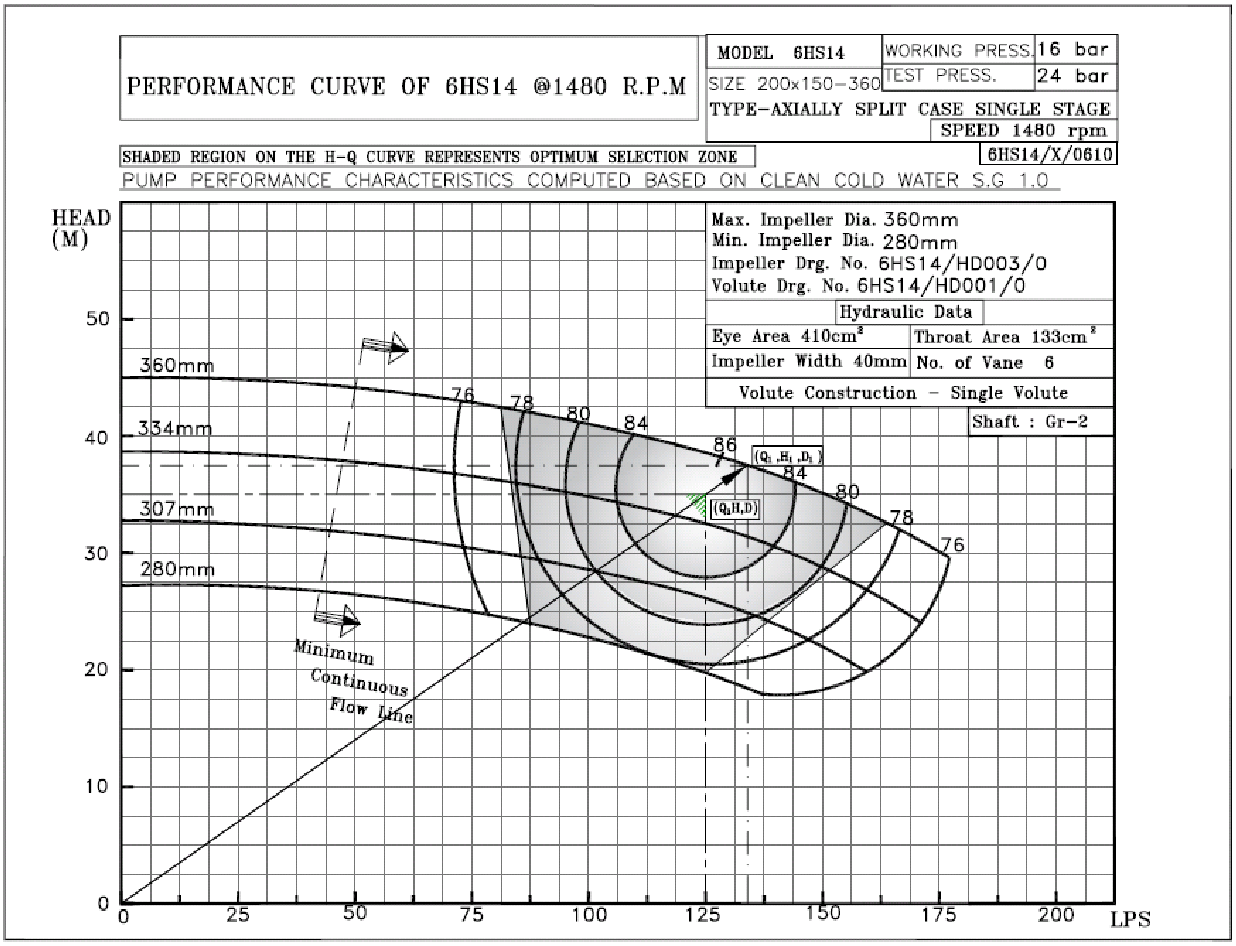
3. Join (0,0) to that specified duty point (given Q & H) and extend the straight line to next higher impeller diameter and get the Q & H at that point.

4. The required impeller diameter is approximated by

$D/D_1 = \sqrt{(H/H_1)}$, or $D = D_1 \sqrt{(H/H_1)}$

Example.

New Duty	
Capacity	125 lps
Head	35 m



<u>from the fig above</u>

D_1=360mm, Q_1=134 lps, H_1=37.5mt. D=?

By the above formula D =360 $\times\sqrt{(35/37.5)}$ =348 mm

Procedure – 2

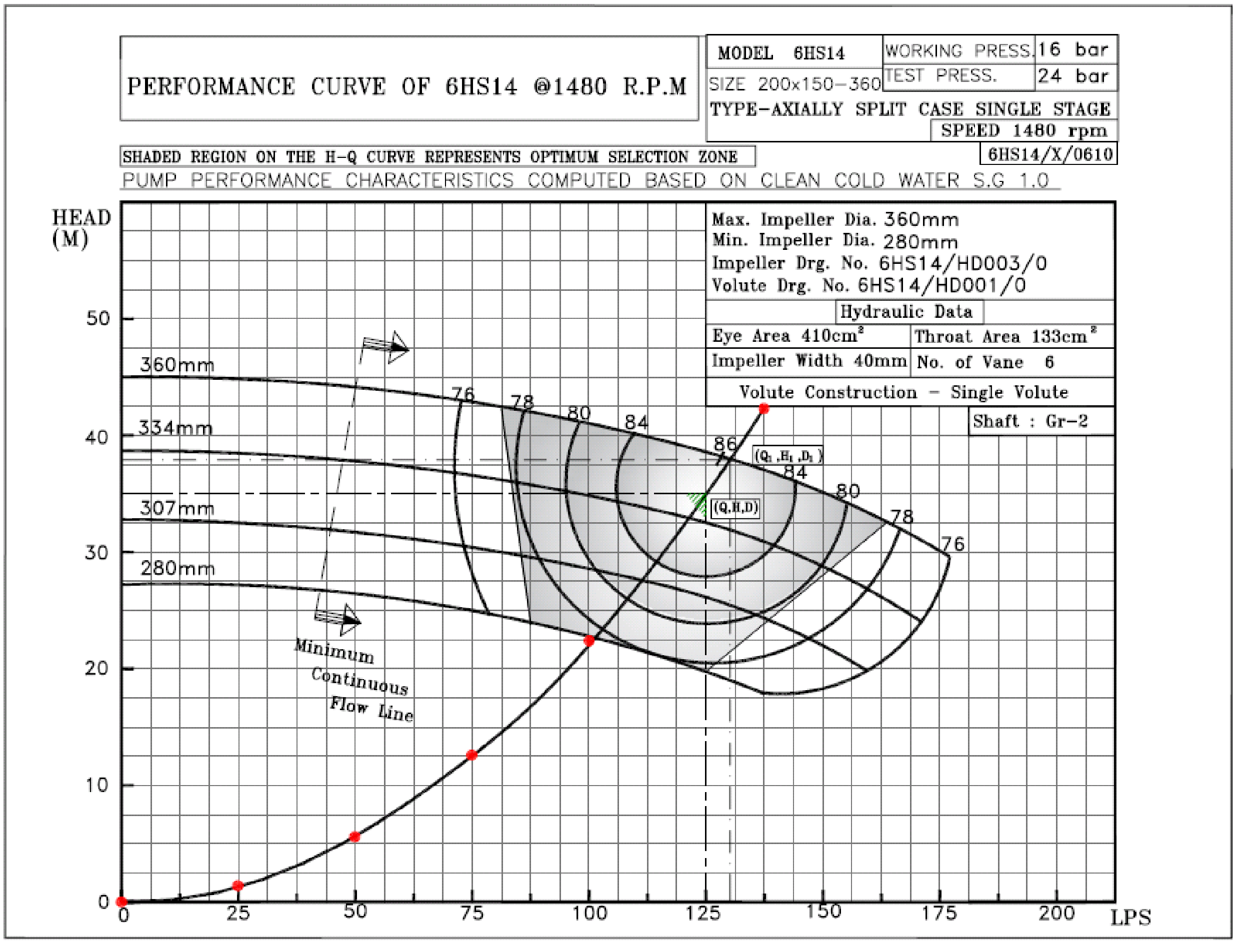
This is a relatively more accurate procedure for estimating the rated impeller diameter.

1. $H=f(Q^2)$ or $H=KQ^2$. From rated Q and H, the constant K is calculated and the curve $H=f(Q^2)$ is plotted.

2. The point of intersection of this curve with the next higher impeller diameter (D_1) is noted (H_1,Q_1).

3. The diameter for the rated duty is calculated using affinity law as follows

$$D= D_1 \sqrt{(H/H_1)} \text{--------------------} (1)$$



Example --- rated Duty –125 lps, 35mt

1. $H=K\,Q^2$ or 35= K x (125)2 or K=0.00224

Plot $h=KQ^2$ using the above value of K

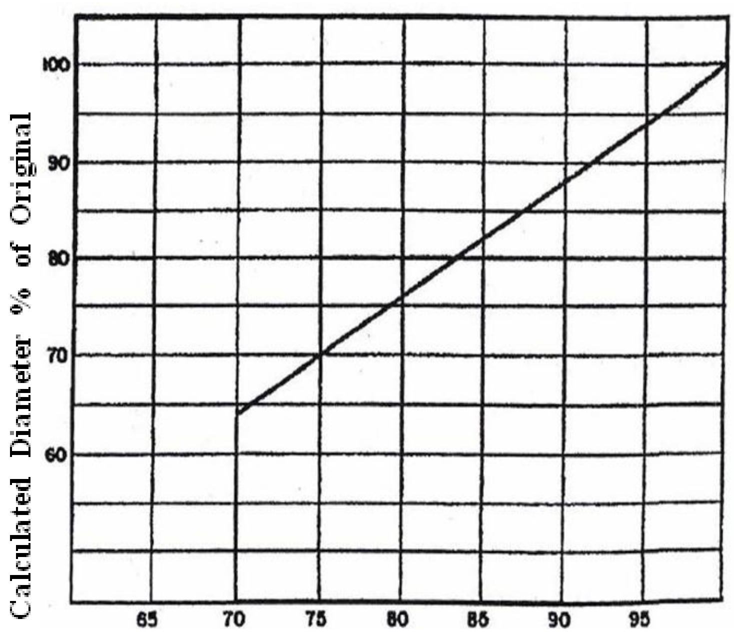
Q(lps)	H(mt)
25	1.4
50	5.6
75	12.6
100	22.4
137.5	42.4

2. The point intersection of this curve with D1=360mm, H_1 =38 mt, Q_1=130lps.

3. Applying affinity law, we get,

theoretical $D=D_1 \sqrt{(H/H_1)}=360 \sqrt{(35/38)}=345$mm

Impeller Trim Correction for Affinity Laws



Reasons for using Correction Chart

The method given above is based on affinity laws and they assume complete similarity in both cases.

In practice this is not achieved for following reasons:

-----Vane angle changes

-----Outlet width changes (hence head coefficient also cha

-----Ratio of impeller outlet area to volute throat area changes.

-----Radial cut water clearance is increased

Consequently the impeller reduction is corrected as shown in figure.

Here, reduction factor is 345/360=0.96 by affinity law. Actual reduction factor from chart1= 0.975.

Theoretical turndown is then corrected using the correction chart (fig above).

The required diameter =0.975 x 360=351mm

Since the performance of an impeller deviates more from the affinity laws as the impeller cut increases it is a good practice to estimate the impeller trim diameter from a reference performance curve obtained at a diameter closest to the estimated diameter.

Trimming Impeller Vanes with or without Shrouds ?

The relative proportion of disk friction losses varies with different types and specific speeds of pumps. For low specific speed pumps (low flow, high head), disk friction accounts for the major part of the pump losses. On the other hand, for high specific speed pumps (high capacity, low head), casing losses account for major portion of pump losses. Therefore, trimming impeller with shrouds is always better for volute pumps with radial flow impellers. For low impeller diameter multistage diffuser pumps often only the vanes are trimmed.

This is the reason why for bolted segment multistage pumps with diffusers, impeller shrouds are often left uncut and only the vanes are trimmed. This procedure ensures proper guidance of the liquid into the diffuser passages. Savings in recirculation and diffusion losses more than compensates additional disk friction losses for these multistage pumps with small diameter impellers.

For all radial flow volute pumps with shrouded impellers, the trim should be both on the shrouds and vanes. If only impeller vanes are cut down in diameter, the useful work of the impeller will be reduced as desired, but the disk friction power would remain unchanged and overall pump efficiency will be unnecessarily reduced.

This reduction in efficiency can be quite significant because the disk friction power varies with the fifth power of the impeller diameter. The relationship is as follows:

Disk Power, $DF = KN^3D^5$ -------------------- (1)

Where,

K = Experimental Factor

N = Pump Speed in RPM

D = Impeller Diameter in mm.

The photograph sent by the customer (shown below) suggests a low specific speed radial flow process pump.

<u>Photograph for Pump Sent by the Customer</u>

Photograph for Pump Sent by the Customer

For this class of pumps the impellers are "tall and thin" and disk friction accounts for the largest part of the power losses in the pump.

Power Loss Calculation

Pump Type: End Suction ISO 2858 or API Process

Pump Size – 125x150-400

Rated Capacity = 300 m3/hr

Rated Head = 55 m

Speed = 1450 rpm

Efficiency = 81%

Power Absorbed = 55.5 KW

Pump Specific Speed, Ns = 1070 US units

Impeller Diameter = 417 mm

Impeller Width at outlet, b2 = 25 mm

Delivered Power (Water KW) = 45 KW

Total Power Loss = 55.5 KW – 45 KW = 10.5 KW

For a pump of this type and specific speed, the above power loss will have the following approximate components:

Loss	Description of loss	Loss in KW	Loss as a % of input power
Mechanical Loss	**Losses in bearings, mechanical seals, etc**	**0.55**	**1%**
Impeller Loss	**Shock losses at entry, skin friction losses**	**1.25**	**2.25%**
Disk Friction	**Power loss in running the impeller shrouds in the fluid**	**4.30**	**7.75%**
Leakage Loss	**Losses through wear rings, balancing holes**	**2.75**	**5%**
Casing Loss	**Skin friction, diffusion and recirculation losses**	**1.65**	**3%**
	Total Losses	**10.50**	**19%**

Let us now consider a revised duty for this pump:

Q = 280 m^3/hr

H = 40 m

New Impeller Diameter = 362 mm

If we trim the shrouds along with vanes, the reduction in disk friction from formula (1) will be:

4.30 X (362/417)^5 = 2.1 KW

The efficiency for this particular pump will remain unchanged at 81% if shrouds are trimmed along with the impeller vanes.

Water Power = (280x40)/367 = 30.5 KW

Input Power when shrouds are trimmed = 30.5/0.81 = 37.6 KW

Input power when shrouds are not trimmed = 37.6 + 2.1 = 39.7 KW

Therefore, pump efficiency, when shrouds are not trimmed = 30.5/39.7 = 76.5%

You will, therefore, observe that if we do not trim the shrouds, we lose efficiency by 4.5 units.

After the trim, the impeller vane tips must be filed and reshaped to resemble the original vane tip. This is difficult in a narrow impeller when the shrouds are left intact.

Besides, the added weight of the uncut shrouds may increase the shaft deflection specifically in overhang horizontal pumps

Maximum Allowable Impeller Trim of an Impeller

Efficiency reduces when the impeller is trimmed. Efficiency reductions can range from one to six points by trimming the minimum diameter. High specific speed pumps usually have greater reductions in efficiency due to trim than low specific speed pumps.

The maximum trim of an impeller usually depends on the type & geometry of an impeller.

Typically, the allowable minimum diameter, as a percentage of its maximum diameter, is as follows:

- Radial Impellers 80%
- Mixed Flow Impellers 90%
- Axial Flow Impellers 95%

Alternate Vane Trim & its Effect on Pump Performance Curve -

This procedure is sometimes followed in multistage diffuser pumps. Alternate vanes are trimmed and the shroud is left intact. The procedure corrects instabilities in the H-Q curve by introducing additional losses in the low flow zone. An unstable H-Q curve becomes stable in the process although the pump also loses a few points in efficiency.