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Design of Centrifugal Pumps with Non-Overloading Power Curve: A Review of the State of the Art

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Abstract

Non-overloading (NOL) power characteristics of a centrifugal pump is an important performance parameter that ensures safety and reliability in diverse industrial applications, particularly where the pump should continue to run without overloading or tripping the driver over the entire operating range. For most applications, pump designers try to optimize pump efficiency for a given set of rated duty conditions-flow, head and available Net Positive Suction Head. However, there are many applications where a centrifugal pump is required to operate over a wide range of flow and head. This operating requirement is often systemic, such as in municipal water distribution systems, where flow is regulated by turning on or off a number of pumps operating in parallel. There are other situations, such as in fire pump applications, where governing standards (Underwriters Laboratory, Factory Mutual Global) mandate operation of the pump over a wide range of flows. For applications such as these, the pump designer needs to try and optimize not merely efficiency at a single rated duty point, but also peak power consumption over the entire range of operating flows. However, attaining NOL power characteristics poses a significant challenge to the designers. Though the science of fluid movement is quite advanced and the recent years have witnessed a phenomenal improvement in CFD, design of a pump for a specific characteristic from the first principle of fluid mechanics is still a formidable task. Till date experts (pump manufacturers) follow a design strategy, where scientific principle is intervened by empiricism obtained through carefully conducted experiments. It need not be exaggerated that the design of pumps with NOL characteristics from the basic theory of fluid mechanics is a formidable task, if not impossible. In this paper, the importance of NOL power, parameters affecting attainment of the NOL power, different design approaches and modifications in the existing design to achieve NOL power have been discussed in detail. It has been observed that with the exception of Anderson’s area ratio method, very few of the design approaches deal with the NOL power characteristics. Considering the usefulness of Anderson’s area ratio method, a real-life example has been cited describing this approach for the benefit of the readers. Two dominant schools across the Atlantic have been using empirical design factors on the one hand and a concept known as “area ratio” on the other hand. Both approaches have solid theoretical foundations and they have produced successful pump designs in the second half of twentieth century. The review paper discusses these design approaches and their impact on the shape of the power characteristics of centrifugal pumps. This review paper has been prepared to document the accumulated study, design work and experimentation to achieve NOL power characteristics in centrifugal pumps. Finally, it can be concluded that systematic research endeavors, combining experiments and computation, are needed to achieve further improvement in NOL characteristics of centrifugal pumps.

Categories: Design and Manufacturing, Thermal and Fluid Systems
Keywords: non-overloading power (nol), area ratio, centrifugal pump design, firepumps, pump reliability

Introduction And Background

Introduction

Pump is a very special energy conversion device which, at the expense of mechanical energy, forces a liquid to flow in a particular path across a pressure gradient. Based on their working principles, the two main categories of pumps are positive displacement and rotodynamic. Among the rotodynamic pumps, centrifugal pumps are most versatile and are widely used for diverse applications. An impeller that converts the supplied mechanical energy into kinetic energy of the flowing fluid is the heart of the centrifugal pump. Subsequently, this kinetic energy of the fluid is converted mostly into pressure energy in the volute casing [1]. Depending on the type, size and capacity, an industrial centrifugal pump consists of various components apart from impeller and volute casing, such as wear rings, shaft, shaft sleeves, bearing housing and more as described in Figure *1* [2-5].

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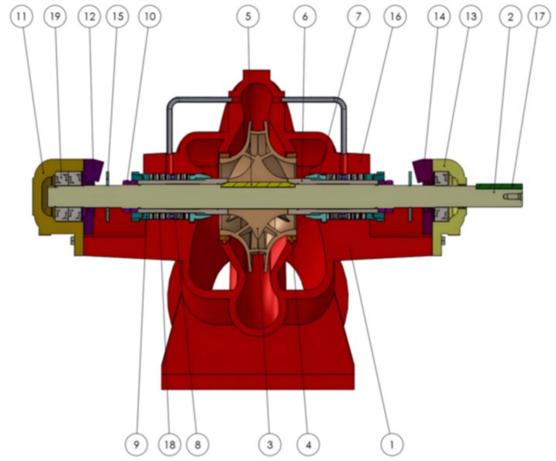


ITEM NO.	PART NUMBER	QTY.
1	CASING BOTTOM HALF	1
2	SHAFT	1
3	IMPELLER	1
4	WEAR RING	2
5	IMPELLER KEY	1
6	SHAFT SLEEVE	2
7	STUFFING BOX BUSH	2
8	LANTERN RING	2
9	SPLIT GLAND	2
10	SLEEVE NUT	4
11	BEARING HOUSING (NDE)	1
12	BEARING END COVER (NDE)	1
13	BEARING HOUSING (DE)	1
14	BEARING END COVER (DE)	1
15	WATER THROWER	2
16	CASING TOP HALF	1
17	COUPLING KEY	1
18	GLAND PACKING	10
19	BEARINGS (DE + NDE)	3

FIGURE 1: Cross-sectional drawing of a split case double suction pump

Centrifugal pumps are widely used in domestic and industrial sectors such as metallurgical industry, irrigation and drainage, water supply, building services, power generation, sewage treatment plants, oil and gas refining, marine and fire fighting, etc. [6]. An annual increase in the global pump market has been reported to be 13% in 2022 indicating a global market of US$64.4 billion for pumps [7]. This scale of market and environmental impact of pump usage justifies research in this field so that energy efficient and reliable designs are available for specific pump applications.

In recent times, one of the major design objectives of pump designers has been to attain a non-overloading (NOL) power characteristics in centrifugal pumps. Now, in order to proceed with this discussion, it may be helpful to do a brief review of the basics of a centrifugal pump and its performance characteristics, with specific emphasis on NOL power.

Centrifugal pumps are being used in industries and in other sectors of economy for many centuries. Flow (Q) through a centrifugal pump is identified as the main performance parameter on which the other parameters like head (H), power (P), efficiency (η) and Net Positive Suction Head (NPSH) depend. Accordingly, the four performance curves, namely, H-Q, P-Q, η -Q and NPSH-Q as shown in the Figure *2*, are of primary important in defining a centrifugal pump.

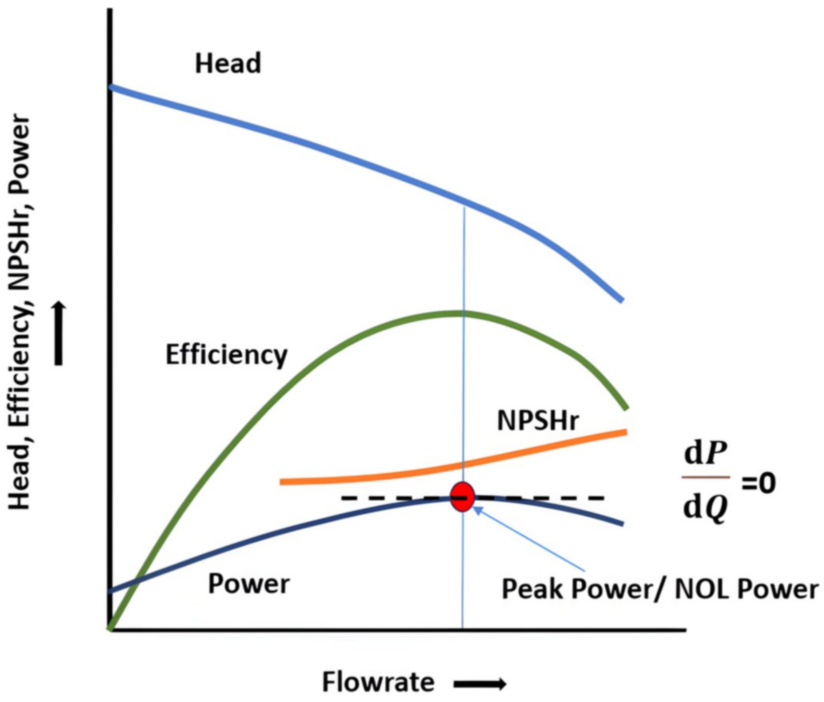


FIGURE 2: Performance characteristic curve of a centrifugal pump

NPSHr, Net Positive Suction Head Required; NOL, non-overloading

The shapes of the characteristic curves arise from different design (geometric) parameters selected for specific operating conditions. In particular, the vane geometry, which determines the inlet and outlet directions of the liquid, is of great importance. Depending on the magnitude of the outlet vane angle β2, the H-Q curve becomes rising, flat or steep, as shown in Figure *3*.

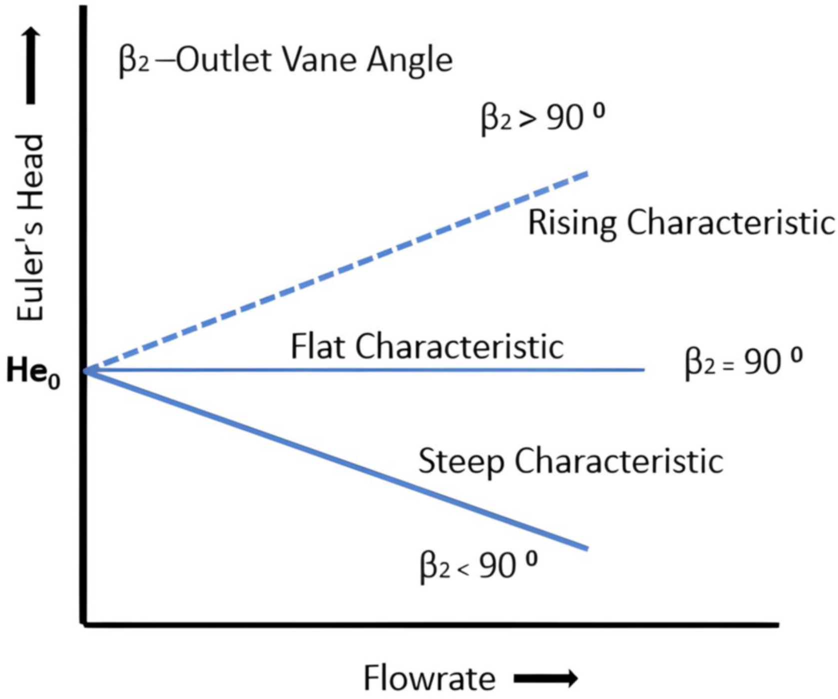


FIGURE 3: Euler's head (He) versus flowrate (Q) characteristics on different outlet vane angles (β2)

These basics are included in standard texts on centrifugal pumps and we have included some in the reference for the sake of completeness [2-5]. The ideal head can be determined using the inlet and outlet velocity vectors as $H_e = k_1 - k_2 Q$, where k_1 and k_2 are constants and k_2 depends on the outlet vane angle β_2. The vanes are also defined as backward curved (β_2 < 90°) or radial (β_2 = 90°) or forward curved (β_2 > 90°) based on different values of β_2. In centrifugal pumps, stable and continuously rising H-Q characteristics are preferred, and therefore, backward curved vanes are used in most designs. In practice, the outlet vane angle varies between 15° and 35°. The power characteristic curve depends on the outlet vane angle, and the shape of the curve is shown in Figure *4*.

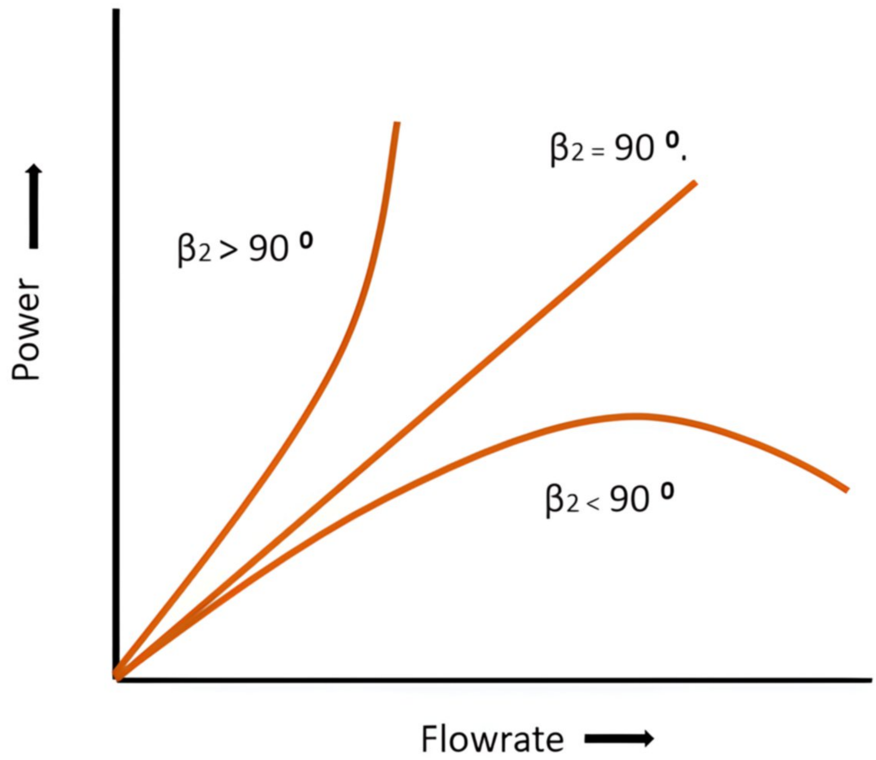
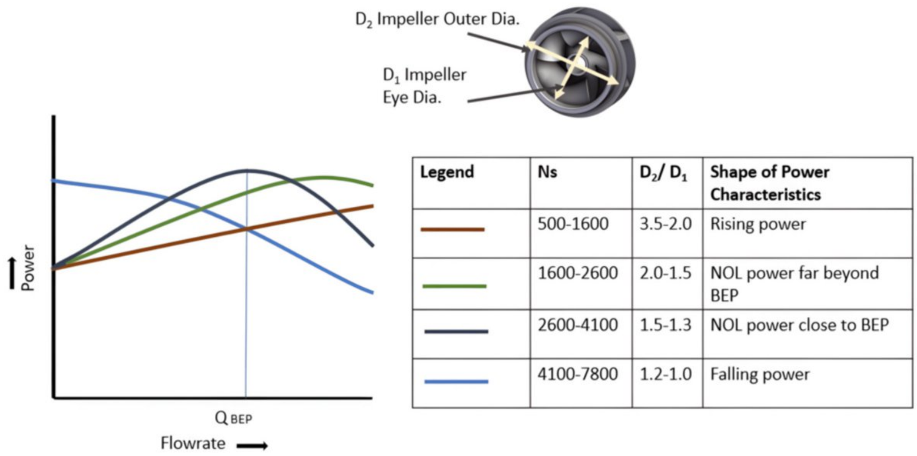


FIGURE 4: Power (P) versus flowrate (Q) characteristics based on different outlet vane angles (β2)

To categorize the performance of centrifugal pumps, it is convenient to introduce the concept of specific speed, derived by dimensional analysis. Specific speed (Ns) defines the impeller type, performance characteristics and achievable efficiency of a centrifugal pump. Specific speed (Ns) = US units, where N is the operating speed in rpm, Q is the flow in USGPM and H is the head developed by the impeller in feet. For centrifugal pumps, specific speed (Ns) ranges from 300 to 10000 US units. Power curves can be categorized based on the specific speed. For low specific speed pumps, the power curve is rising with the power increasing as the flowrate increases, for medium specific speed pumps, power curve is generally NOL, with a distinct peak power, and for high specific speed pumps, the power curve is falling in nature with maximum power at shut-off or zero flow condition. The relationship between the specific speed, impeller type and power characteristics curve has been illustrated in Figure *5*. Specific speed ranges of 4100 to 7800 US units is considered to be high, whereas the Ns range of 500 to 1600 US units is relatively low for centrifugal pumps [4].



Legend	Ns	D_2/D_1	Shape of Power Characteristics
(brown line)	500-1600	3.5-2.0	Rising power
(green line)	1600-2600	2.0-1.5	NOL power far beyond BEP
(dark line)	2600-4100	1.5-1.3	NOL power close to BEP
(blue line)	4100-7800	1.2-1.0	Falling power

FIGURE 5: Relationship between the specific speed (Ns), D2/D1 ratio and power curve

NOL, non-overloading; BEP, best efficiency point

NOL power characteristics have a particular peak or maximum power on its power characteristics. Peak power considered to occur at a point where the slope dP/dQ = 0, when a tangent is drawn on the power curve as shown in the Figure *2*. This is the highest power required by a pump running throughout its operating zone. If the power curve is not NOL, the driver may trip under overload conditions, and the pump can cease to operate. This can result in major production loss and compromise safety of the system. Centrifugal pump failure due to the tripping of the driver accounts for significant maintenance costs and downtime. The non-overload power curve helps the pump to run steadily at any flow rate [8,9].

Hydraulic losses have an effect on the shape of the power characteristics. Pump power consumption rises with the increase in the losses reducing the hydraulic efficiency of the pump. The objective of pump designers is to reduce the losses as much as possible to optimize pump efficiency. Designers consider these losses [10] to consist mainly of (a) shock loss or eddy loss, (b) impeller/volute skin friction loss, (c) disk friction loss and (d) re-circulation loss. It is observed that part flow recirculation loss has a significant effect on the power characteristics. The effect of different losses described by Gulich [5] has been illustrated in Figure *6*.

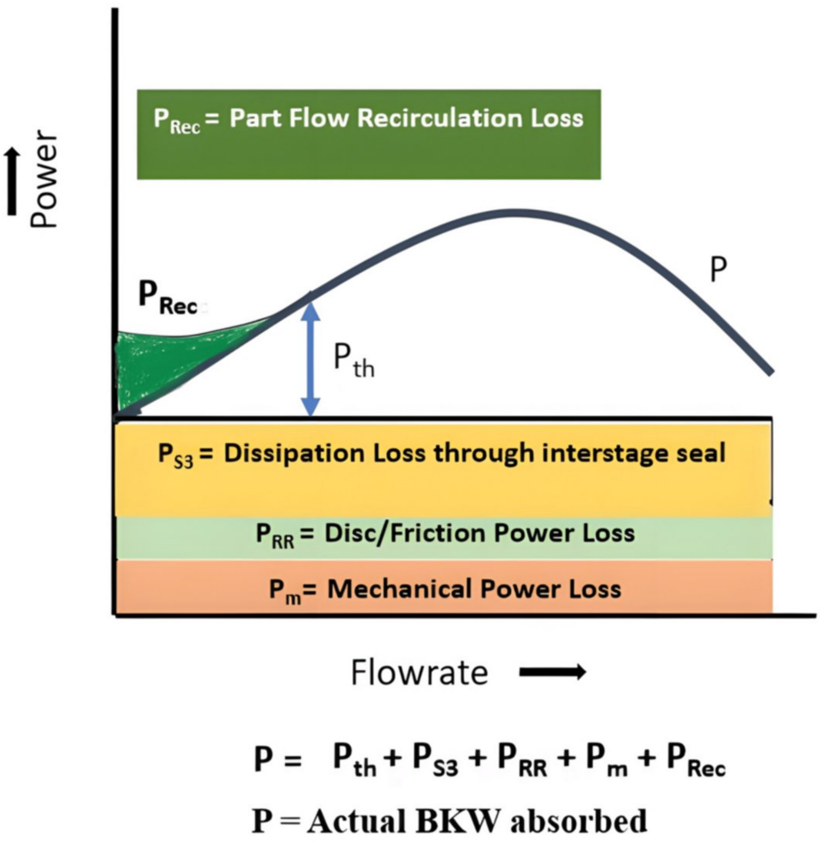


FIGURE 6: Different losses affecting the power of a centrifugal pump

BKW, Brake Horsepower

In recent times, one of the major design objectives of many hydraulic designers has been to attain the NOL power characteristics in centrifugal pumps, of medium and low specific speeds. In adverse conditions or emergency situations, it is important that the pump continues to run without overloading or tripping the driver over the entire operating range. The peak power or maximum power occurs at a particular flow rate. If the peak power can be accurately determined, the driver selection can be optimized, and therefore, the most desirable design philosophy is to bring the maximum power close to the best efficiency point (BEP). In case of fire pumps, the standards of Underwriters Laboratories (UL448) and of Factory Mutual (FM1311) mandate that the driver selection should be based on the peak power, considering the safety and reliability aspect during accidental fire events. While NOL power is certainly desirable, very few research studies have been conducted to find ways to limit peak power of centrifugal pumps. The behavior of the fluid inside the pump is complex in nature. It is difficult to establish a mathematical model to predict NOL power characteristics. Hence, empirical relationships obtained from actual test data of industrial pumps play a major role in the design approaches. It is a fact that the natural power curve of a low specific speed pump is rising in nature, and therefore, it is a challenge for the designers to obtain a peak power near the BEP without sacrificing efficiency, suction lift capabilities and mechanical reliability for this class of pumps. In this paper, the importance of NOL power, parameters affecting attainment of NOL power, different design approaches and modifications to the existing designs to achieve NOL power have been discussed in detail.

Two dominant schools across the Atlantic have been using empirical design factors on the one hand and a concept known as “area ratio” on the other hand. Both approaches have solid theoretical foundations and they have produced successful pump designs in the second half of the twentieth century. The review paper discusses these design approaches and their impact on the shape of the power characteristics of centrifugal pumps. Pump sizes become larger under the challenging demands of resource crunch, environmental degradation and climate change. It is also very important to focus on sustainability, carbon footprint and reliability of centrifugal pumps used for urban water supply, air-conditioning of large public spaces and for industrial cooling applications. This review paper has been prepared to document the accumulated study, design work and experimentation to achieve NOL power characteristics in centrifugal pumps. This paper will also be useful for other researchers working in the field of performance characteristics of centrifugal pumps.

For consolidating this review and for greater clarity for the readers, two small case studies involving pump design approaches were taken up. In the first case, factor law is applied to design a new pump with performance characteristics different from the available test data of a pump of similar specific speed. In the second case, the method of estimating the area ratio has been elucidated. These two examples are included in the appendices of this paper.

Importance of NOL power characteristics

A centrifugal pump can draw excessive power due to several hydraulic and mechanical reasons, and these reasons are explained in the standards of Hydraulic Institute [11]. Water horsepower, friction losses and leakage losses are the main components of the pump shaft power. Researchers have developed design theories to achieve NOL power by reducing the water horsepower [12]. Main design considerations for NOL power curve are as follows: (a) pump running at higher flow rate instead of rated flow conditions due to the decrease in the system head or overestimation of the system head during project stage and (b) increase in the clearances due to component wear and subsequent decline in hydraulic performance can cause opening of the control valve wider leading to an increase in both flowrate and power consumption [13].

In case of fire-fighting pumps, conforming to NFPA20, UL 448, FM 1311 and EN 12845, driver rating is selected based on the peak power of the pump. This application, therefore, demands the pump power should be NOL in order to avoid driver overloading in case of fire. In shipping industry, serial-parallel centrifugal pumps work under two different conditions and NOL power curve is recommended [14]. In sewage pumps, the application of solid-liquid two-phase flow becomes predominant. The solid particles can create blockage in the flow passage resulting in an increase of the input current leading to motor overload. Also, in case of low specific speed sewage pumps, due to change in the operating conditions the pump operates at overflow conditions and the power increases drastically with the increase in the flow rate and causes an overload to the driver, and therefore, NOL power is important in sewage applications. In solid-liquid two phase applications, such as coal industry, metallurgical industry, paper industry, etc. NOL power becomes essential. In the nuclear industry, where ensuring efficient cooling of the reactor core becomes priority, NOL power curve can improve the reliability of the system. For nuclear power plants, the reliability of the cooling system has to be ensured even for accident and post-accident scenarios. Therefore, NOL power characteristics of this class of pumps are very crucial.

Review

Parameters affecting NOL power

Anderson [15] suggested that the shape of pump power curves depend on the area ratio (Y). Area ratio is the ratio of the outlet area between vanes (OABVs) and the throat area of the volute or diffusers in the collector section of the pump.

$$(Area Ratio (Y) = (Outlet Area between Vanes)/(Throat Area) = (0.95 \Pi D_2 b_2 sin \beta_2)/A_3$$
---------- (1)

OABV depends on the impeller outlet diameter, impeller width, outlet vane angle and blockage. Blockage is a function of the number of vanes used in the impeller and vane thickness. Generally, an area ratio of less than 0.75 is required for achieving NOL power curve. According to area ratio method of pump design, the factors affecting the attainment of NOL power characteristics have been summarized in Figure 7.

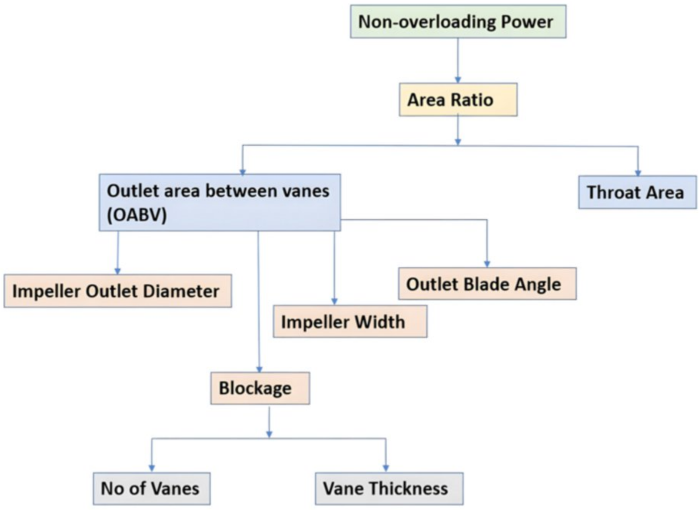


FIGURE 7: Factors affecting NOL power according to area ratio method

NOL, non-overloading

Impeller Outlet Diameter

Use of a low area ratio in order to achieve NOL power curve demands for low head coefficient (ψ), to be defined shortly. As a result, the peripheral velocity (u) increases by increasing the impeller diameter. Large impeller diameter, in turn, increases the disc friction losses and leads to lower efficiency [16]. As a result, the power consumption with large diameter impeller increases significantly. The stress at the tip of the impeller increases too, leading to higher strain and displacement. It has been observed that the selection of lighter impeller material can reduce power consumption. Material technology is evolving rapidly and researchers can work on high strength lightweight non-metallic materials for construction of the impellers. This can eliminate the design constraint of usable impeller tip speed and reduce power consumption of higher diameter impellers by reducing disk friction losses [17].

Impeller Outlet Vane Angle

Increase in outlet vane angle makes the H-Q curve flatter and P-Q curve becomes rising in nature. For low specific speed pumps, the hydraulic performance improves with the increase of the outlet vane angle. The effect on efficiency is prominent at higher flow rates [18]. The optimal outlet vane angle lies in the range of 22°-30° and produces the highest possible efficiency at the BEP [19]. The shaft power is proportional to the theoretical head, which on other hand is proportional to the whirl velocity. The whirl velocity decreases with the decrease in the outlet vane angle, limiting the increase of shaft power. The low outlet vane angle limits the increase of shaft power, and therefore, NOL power curve and steeper H-Q curve are often observed in pumps using low outlet vane angles. However, this reduction in the outlet vane angle increases the length of the impeller vanes and consequently, increases the friction losses leading to a decrease in pump efficiency. Selection of β_2, therefore, depends on, (a) desired steepness of the H-Q characteristics, (b) optimization of efficiency and (c) shape of the P-Q characteristics [20]. Very limited research work has been done on the effect of the outlet vane angle on the shape of the P-Q characteristics. Designers continue to face challenges to achieve NOL power characteristics without compromising on the pump efficiency.

Vane Thickness

As the vane thickness increases, the relative flow angle at the outlet of the impeller decreases and therefore, tangential component of absolute velocity at outlet (C_{u2}) becomes lower and H-Q curve becomes steeper. The BEP as well as the peak power shifts toward the left on the pump characteristics. Internal turbulence losses increase with the increase in vane thickness and both head and efficiency decline [21].

Impeller Width

An increase in impeller width results in an increase in the head at the BEP due to a decrease in the friction losses and also makes the H-Q curve flatter. For high specific speed pumps, the effect is more prominent [22]. However, the impeller width cannot be increased beyond a limit when the efficiency starts declining. Therefore, the determination of the optimal impeller width plays an important role in the pump design. It has been found that impeller width affects the steepness (K_h) of the H-Q characteristics. The steepness is defined as, $K_h = (Hmax - Href)/Href$;H_{max} is the maximum head at $0.5Q_{ref}$, and H_{ref} is the head at Q_{ref}. There is a definite relationship (equation 2) between the width of the impeller (b_2) and the steepness of the H-Q curve.

$$K_h = a(b_2/D_2)^{-k} \text{-----------------------(2)}$$

where, a and k are the statistical coefficients based on the specific speed of the pumps [23].

The steepness increases with the decrease in the impeller outlet width. Anderson's area ratio (Y) also decreases with the decrease in the impeller outlet width and this can possibly lead to NOL power characteristics.

Number of Vanes

Higher number of vanes improves head developed by the pump as it provides an improved guidance to the flow by decreasing the slip effect. It means that increasing the number of vanes improves uniformity of the flow through a particular blade passage and reduces the chance of building secondary flows, and therefore, it reduces the slip effect. But, as number of vanes increases there is a decrease in the area of flow passage and the local velocity of the fluid particle increases resulting in higher friction losses and reduction in pump efficiency. Usually, for stable H-Q characteristics, the number of vanes more than eight is not recommended. However, if the system requires a narrow zone of operation, impellers with up to nine vanes are also used. It has been observed that increasing the number of vanes increases head developed by the pump in the low flow region due to decreased secondary losses, but at higher flow regions the frictional loss is more dominant and therefore, reduces the head [24,25]. This phenomenon results in steeper H-Q characteristics and a sharper decline in pump efficiency at high flow zones. The chances of achieving NOL power characteristics increases due to relatively sharper decline in pump head in the high flow zone.

On the other hand, fewer number of vanes increases the diffusor loss or mixing loss due to non-uniform flow regime at the impeller exit. Number of vanes fewer than five, produces much non-uniformity due to large vane spacing at the outlet, leading to pressure pulsation, vibration and noise. However, the effect of blockage decreases with the number of vanes, and therefore, it reduces the frictional losses. As the flow increases, the mixing losses become predominant reducing the head of the pump and increasing the steepness of H-Q characteristics.

Since the mixing loss increases with lower number of impeller vanes, decreases when the number of vanes is increased and increases again if the number of vanes is increased more, it is a challenge for the designer to select optimum number of vanes depending on the application of the pump. Apparently, NOL power can be achieved by using either large number of vanes or very few vanes for low specific speed pumps. However, further studies should be conducted to clearly understand the effect of number of vanes on NOL power curve.

Design approaches on power characteristics

For generations, different pump manufacturers have adopted a number of design approaches using partly semi empirical and partly graphical techniques for design of centrifugal pumps. In the pump industry, these design approaches are still widely practiced although the use of CFD techniques is becoming increasingly popular for validation of design and for numerical testing of pump performance. It will be a very useful exercise to explore the existing design methods to examine how the issue of NOL power characteristics can be addressed. According to Salisbury [16], principal methods of pump design are as follows:

1. Modeling

2. Modification

3. Inlet design

4. Free vortex in the volute casing

5. Method of design coefficient

6. Method of area ratio

Table *1* depicts the design principles and the respective pump manufacturers [26-31].

Author	Design approaches	Pump manufacturers
JH Bunjes and JAV Woerden	Slip and loss analysis	Stork Pompen BV
EW Thorne	Area ratio method	Worthington-Simpson
EA Chiappe	Graphical technique and empirical formulae	NEI-APE Limited
J Richardson	Size, shape and similarity	Hayward Tyler Limited
AN Neal	Theoretical methods-combination of through-flow analysis with vane-to-vane technique	National Engineering Laboratory
TE Sterling	NEL methods	National Engineering Laboratory

TABLE 1: Design principles used by different pump manufacturers (I Mech E Conference 1982)

NEL, Numerical and Experimental

As we are exploring the means for achieving NOL power characteristics, in this review, we have placed emphasis on the design approaches that explain different power characteristics in terms of shape of the power curve, peak power and shut-off power.

Modeling

For similar specific speeds, the pump performance of a new pump can be predicted from the performance characteristics of an existing pump either larger or smaller in size by using model laws or factor laws. The new pump is called the prototype and the existing pump is known as the model. The power characteristics can, therefore, be estimated using the new flow and head and the power can be expressed as $P = \rho * Q * H/3.67 * \eta$. In this design approach, it is seen that the mechanical sizes are scaled in a different way compared to the hydraulic passages. Any change in the hydraulic passages, pump geometry or mechanical scaling has significant effect on the shape of the power characteristics. Also, the Euler's number needs to be identical while using the dynamic similarity for the pump applications [32,3]. It is difficult to predict the absolute value and position of the peak power and shut-off power by modeling due to unscalable nature of some of the losses involved and consequent destruction of similitude. Researchers have proposed different model laws for (a) pumps running at different speeds, (b) pumps handling fluids of different properties and (c) pumps running under various suction conditions [33]. These laws are grossly empirical and there exists large differences in the results when these laws are applied for modeling. However, pump manufacturers generally have a large body of information from the pump test data, and they still use these laws to get a baseline to initiate the new design. Generally, high specific speed pumps where empirical relationships and conventional design approaches are known to be inadequate, designers still prefer to follow the model laws validated additionally using CFD analysis. Since the power prediction depends on various factors with complex unquantifiable losses, further studies can be made in order to obtain a good guideline for a realistic prediction of the power characteristics using the factor laws. Factor laws can be written as [32]:

$$\frac{Q}{w * D^3} = \text{Constant} \text{ --------------------------------(3)}$$

$$\frac{Q * H}{w^2 * D^2} = \text{Constant--------------------------------(4)}$$

$$\frac{P}{\rho * w^3 * D^5} = \text{Constant --------------------------(5)}$$

If the speed remains unaltered, one can simplify the above equations to the following [3]:

$$Q_n = Q_m * f^3 \text{----------------------------------(6)}$$

$$H_n = H_m * f^2 \text{----------------------------------(7)}$$

$$\frac{D_n}{D_m} = f \text{---(8)}$$

where, f = reduction factor or multiplier

Q_n = Capacity of new pump

H_n = Head of new pump

D_n= Impeller diameter of new pump

Q_m = Capacity of model pump

H_m = Head of model pump

D_m = Impeller diameter of model pump

The application of factor laws to predict the geometry of a prototype from a model pump is shown in Appendix 1. Though this design concept is used in the industry, established empirical relationships based on the specific speed of the pumps to predict pump performance are still missing.

Modifications

This approach focuses on the modifications of the existing design parameters of a pump to achieve a desired performance. The critical design parameters [16] of the pump characteristics are as follows: (a) Impeller outlet diameter, (b) Impeller eye diameter, (c) Impeller hub diameter, (d) Number of vanes, (e) Inlet vane tip positions, (f) Impeller vane inlet angles, (g) Impeller width, (h) Impeller vane outlet angles, (i) Impeller vane shape at inlet and outlet, (j) Impeller vane length, (k) Impeller symmetry and concentricity, (l) Volute throat-shape and area, (m) Volute vane inlet tip radius, (n) Volute vane inlet tip angle, (o) Volute passage shape, (p) Impeller and volute surface finish.

Apart from the above, inlet area between the vanes, OABVs, vane surface of revolution, manufacturing tolerances are among many factors that influence the hydraulic performance of a pump. Power characteristics depend dominantly on the impeller width, outlet vane angle, OABVs, vane thickness, wear ring clearances, throttling effects and presence of pre-rotations. Modifications of these parameters, such as impeller trimming, volute modifications, under-filing and over-filing of the vanes, adding splitter vanes etc. can change the power curve of the pump. For example, if during the installation of a pump, it is found that the actual head of the system is much lower than the specified head, the existing impeller can be trimmed down to meet the actual system requirement more efficiently with lower power consumption. The pump performance can also be modified by the removal of material from the vane tips at the impeller outer diameter. The removal of material at non-pressure side of the vane is known as under-filing, whereas removal of material from the pressure side is known as over-filing. Under-filing improves head generated at higher flows reducing the steepness of the H-Q curve. The effect of this modification on the power characteristics is not conclusive from the published information [34]. These modifications depend largely on the experience and intuition of the pump designers and in most cases trial and error is adopted.

Inlet Design

Impeller inlet design is mostly related to NPSHr (Net Positive Suction Head Required) and recirculation characteristics of a pump. Impeller eye diameter optimization based on the axial and peripheral velocities at the impeller eye is a common practice in the pump industry. To achieve better inlet recirculation and cavitation characteristics, suction specific speed can be changed through a reduction of the impeller eye diameter and changes in the inlet vane angles. Proper inlet design reduces vibration, noise, erosion damages and reduces large forces on the impeller [16]. Documented studies on the effect of inlet design on power characteristics is negligible, and there exists significant scope for research in this area. The relationship between inlet area between vanes and OABVs can be one of the determining factors affecting power characteristics, and further studies can be made. In this modification, method too, NOL power characteristics cannot be obtained directly.

Design Coefficient

Two main reference books on pump design, one by Stepanoff [4] and the other by Lobanoff & Ross [3] mainly prescribe the use of design coefficients for designing centrifugal pumps. These are dimensionless velocity ratios based on the impeller specific speed and impeller outlet vane angles. Head Coefficient and Flow Coefficient are two major ratios used in sizing of new impellers [3-5].

$$HeadCoefficient = (\frac{2 * g * H}{U_2^2})\text{---------------}(9)$$

$$FlowCoefficient = (\frac{C_{m2}}{U_2^2})\text{-----------------------}(10)$$

Depending upon the pump type and application, head and flow coefficients may be selected from empirical charts provided by these authors. Design coefficients are extensively used for quick estimations and references.

Coefficient	Design parameters	Formula	Chart
Head coefficient K_u	Impeller diameter (D_2)	D_2 = (1840 × K_u × $H^{0.5}$)/RPM	Refer Figure 8
Flow coefficient K_{m2}	Impeller width (b_2)	C_{m2} = K_{m2} * (2 g H $)^{0.5}$ b_2 = (GPM × 0.321)/C_{m2} × (Л D_2 – Z t_u)	Refer Figure 9
Volute velocity constant K_3	Volute area (A_3)	A_3 = 0.04 × GPM/K_3 × $H^{0.5}$	Refer Figure 10

TABLE 2: Use of design coefficient according to Lobanoff and Ross

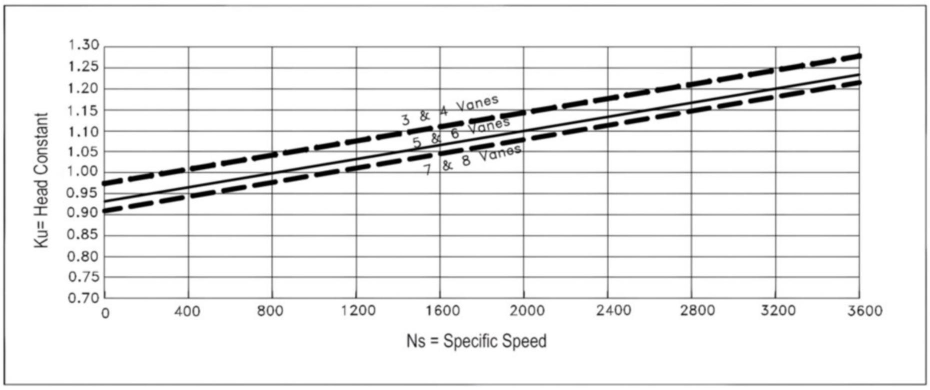


FIGURE 8: Head coefficient versus specific speed

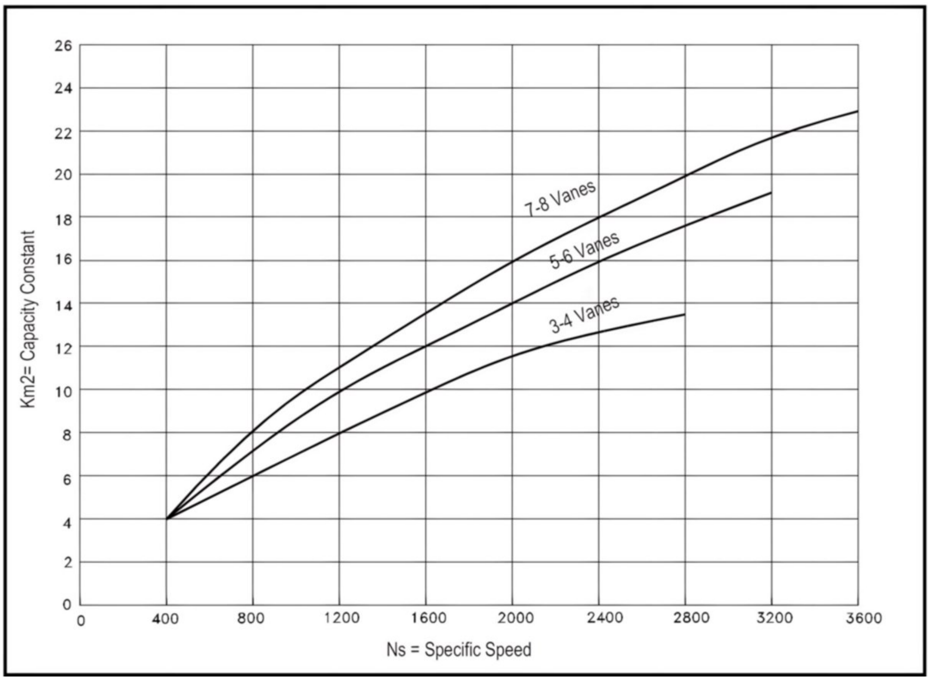


FIGURE 9: Flow coefficient versus specific speed

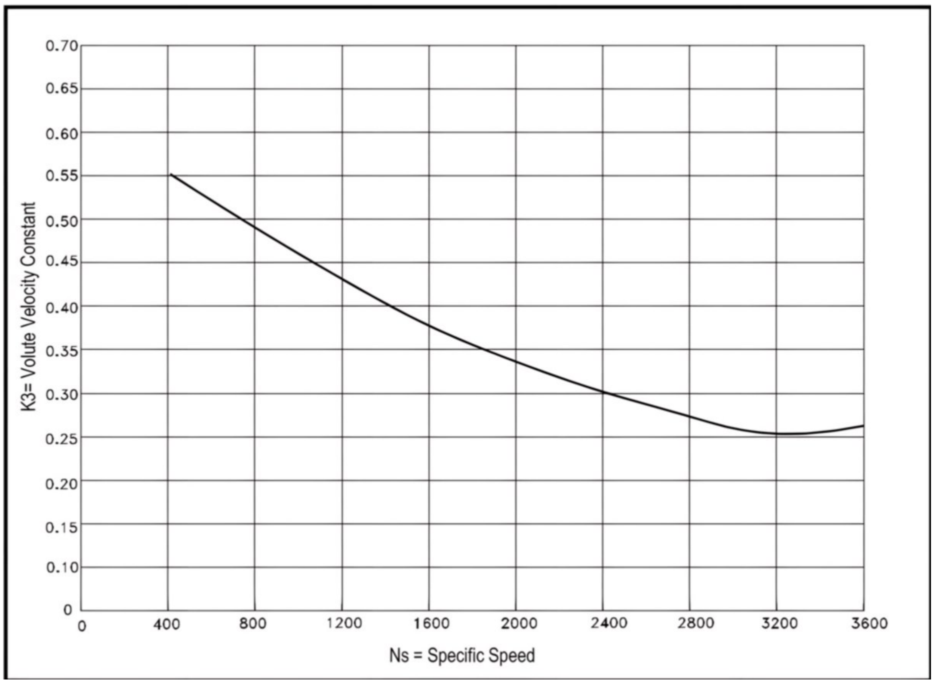


FIGURE 10: Volute velocity coefficient versus specific speed

Lobanoff and Ross method [3] elaborates the use of design coefficients for calculating the major design parameters. These are summarized in Table *2*.The above charts were generated from numerous performance test data of pumps of different specific speeds. Similarly, Stepanoff's [4] design method also uses flow and head coefficients in a chart (Figure *11*), which plots these coefficients against specific speed.

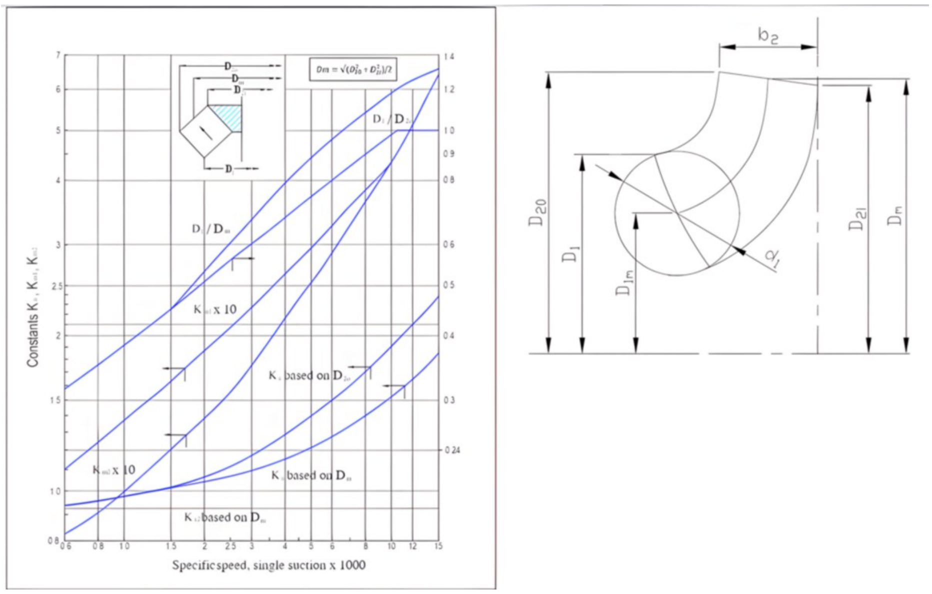


FIGURE 11: Speed constants against specific speed

Almost universally, design coefficients of Stepanoff are used today to design a new impeller when no model impeller is available. Table 3 sums up the definitions of these coefficients.

Coefficient	Definition	Use of coefficient
Speed constant K_u	$K_u = u_2/(2\ g\ H\ )^{0.5}$	Head coefficient $\psi = 1\ /\ (2^*\ K_u^2)$
Capacity constant K_{m2}	$K_{m2} = C_{m2}/(2\ g\ H\ )^{0.5}$	Capacity coefficient $\phi = K_{m2}/\ K_u$
Entrance velocity constant K_{m1}	$K_{m1} = C_{m1}/(2\ g\ H\ )^{0.5}$	$C_{m1} = Q/\ \pi\ D_{1m}\ d_1$, D_1/D_m and D_1/D_{20} ratios are used

TABLE 3: Use of design coefficient according to Stepanoff

These coefficients are derived through both experiments and actual experience of building pumps in the last century. The scope, therefore, remains to improve them through numerical and test data analysis and further experimentation. Unfortunately, these coefficients cannot be directly used to influence the shape of the power characteristics of a pump. However, the shape of the power characteristics will alter based on the steepness of the Q-H curve and Q-efficiency curves that these coefficients influence. Therefore, the method of design coefficient is not of limited direct use for designing a pump with a desired power characteristics. Using this method, therefore, only limited success may be obtained in achieving a desired power characteristics after a large number of design trials.

Free Vortex

Free vortex flow is a common assumption in centrifugal pump design for fluid flow through the volute casing. Free vortex follows the law of conservation of angular momentum. The increase in the pressure is significantly less compared to the energy of the forced vortex. The pressure distribution is given by [5]:

$$p - p_0 = \frac{\rho}{2} C_{u,0}^2 (1 - \frac{r_{0^2}}{r^2}) \text{---------------(11)}$$

Using this approach, the BEP of the pump depends on the impeller and volute characteristics. Impeller characteristics has a negative slope with head declining with increasing flow, whereas the volute characteristic has a positive slope. The point of intersection of these two characteristics is defined as BEP (Figure 12).

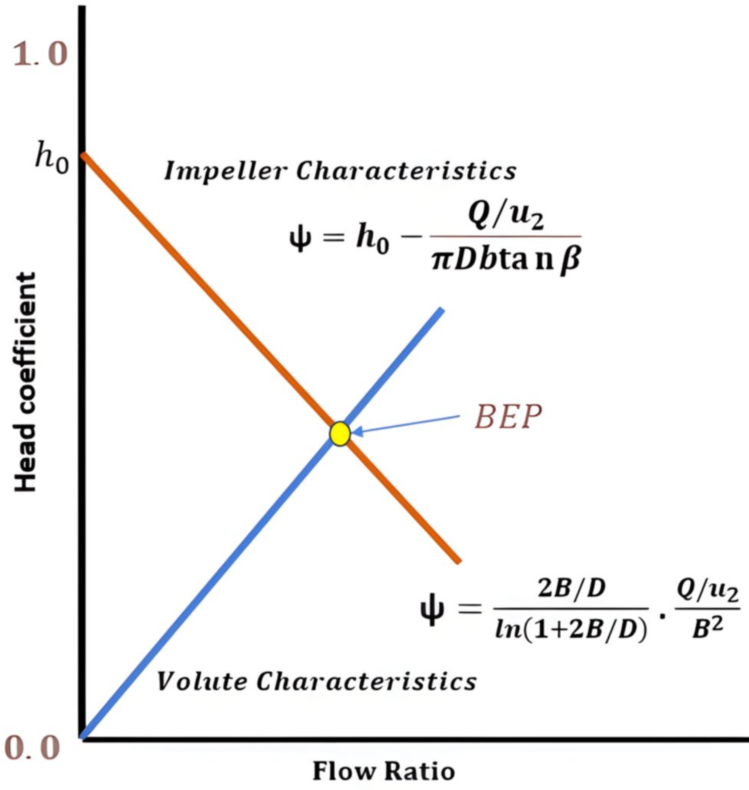


FIGURE 12: Intersection of volute and impeller characteristics giving the best efficiency point

The impeller characteristics using Busemann head coefficient (h_0) and no pre-rotation is given by:

$$\frac{gH_{imp}}{U_2^2} = h_0 - \frac{C_{m2}}{U_2} cot\beta_2 \text{---------------(12)}$$

The impeller head is equal to the input head in the volute characteristic. When we consider hydraulic losses, the equation is modified to:

$$\frac{gH}{\eta_h U_2^2} = h_0 - \frac{\frac{Q}{U_2}}{\pi D_2 b tan\beta_2} \text{---------------(13)}$$

The head and flow at BEP are determined based on the following equations derived by Worster [35].

$$\psi = \frac{gH}{\eta_h U_2^2} = \frac{h_0}{1 + \frac{ln(1+\frac{2B}{D_2})B^2}{(\frac{2B}{D_2})\frac{B^2}{\pi D_2 b tan\beta_2}}} \text{------------------(14)}$$

$$t_q = \frac{Q}{U_2 B^2} = \frac{h_0}{\frac{\frac{2B}{D_2}}{ln(1+\frac{2B}{D^2})} + \frac{B^2}{\pi D_2 b tan\beta_2}} \text{------------(15)}$$

Sometimes designers use the free vortex approach in impeller blading for semi-axial flow impellers. Constant vane work is always transmitted along each streamline and radial equilibrium is maintained because the axial velocity remains constant.

Worster concludes that pump volute plays a major role in determining the BEP for low and moderate specific speed units. However, for high specific speed pumps, the impeller is the determining factor [35].

The free vortex approach does not directly address power characteristics of a pump. The position of the peak power and the shape and slope of the power curve are areas for further research with this design approach.

Area Ratio

Anderson was the first designer to propose a correlation between the shape of pump performance curves with the pump geometry in his area ratio approach. Area ratio is defined as the ratio of total outlet area of all vanes and throat area of the pump volute.

$$Y = \frac{0.95\pi D_2 b_2 sin\beta_2}{A_3} \text{----------------(16)}$$

The blockage factor is taken as 0.95 to account for the impeller vane thickness. Alternately, the blockage effect can be estimated as: $1 - \frac{Zt}{\pi D_2 sin\beta_2}$

If the area ratio is low, the pump will generate a lower head and the BEP will shift toward left in the flow regime, where the flow velocity at the volute throat is small compared to the peripheral velocity of the impeller. On the other hand, with a high area ratio, the pump will generate a head, which is approximately twice the velocity head and the flow velocity at volute throat becomes very close to the peripheral velocity of the impeller. Figure *13* describes the shape of the pump characteristics with low and high area ratios [15].

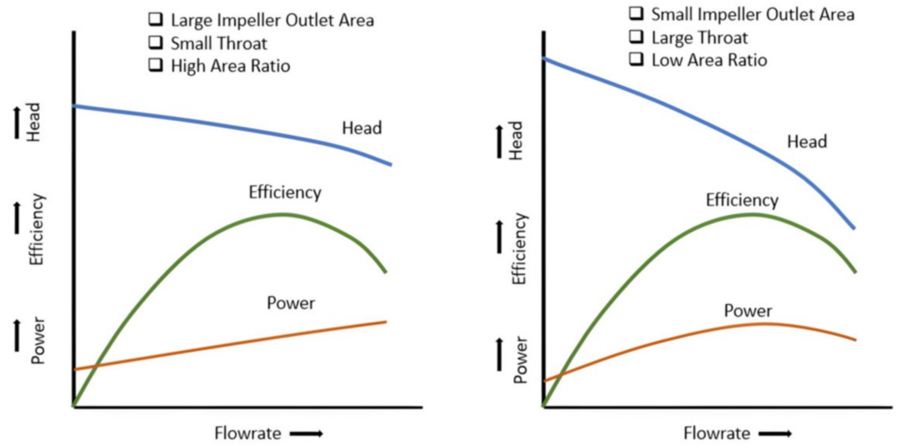


FIGURE 13: Shape of characteristics and area ratio

This approach specifically deals with the power characteristics based on different specific speeds and area ratios. Two coefficients, (a) quantity coefficient and (b) head coefficient, are incorporated and plotted against the area ratio from experimental data to determine the pump impeller and volute parameters (Figure *14*). If the area ratio is smaller, the head-capacity curve becomes steeper and power-capacity reaches a maximum value at a certain flow and then begins to drop. On the other hand, if the area ratio used is larger, the head/capacity curve becomes flat and power/capacity curve keeps rising with increasing flow [15]. Worster [35] has clearly shown that the area ratio is a significant parameter for analyzing the effect of different shapes of pump volute.

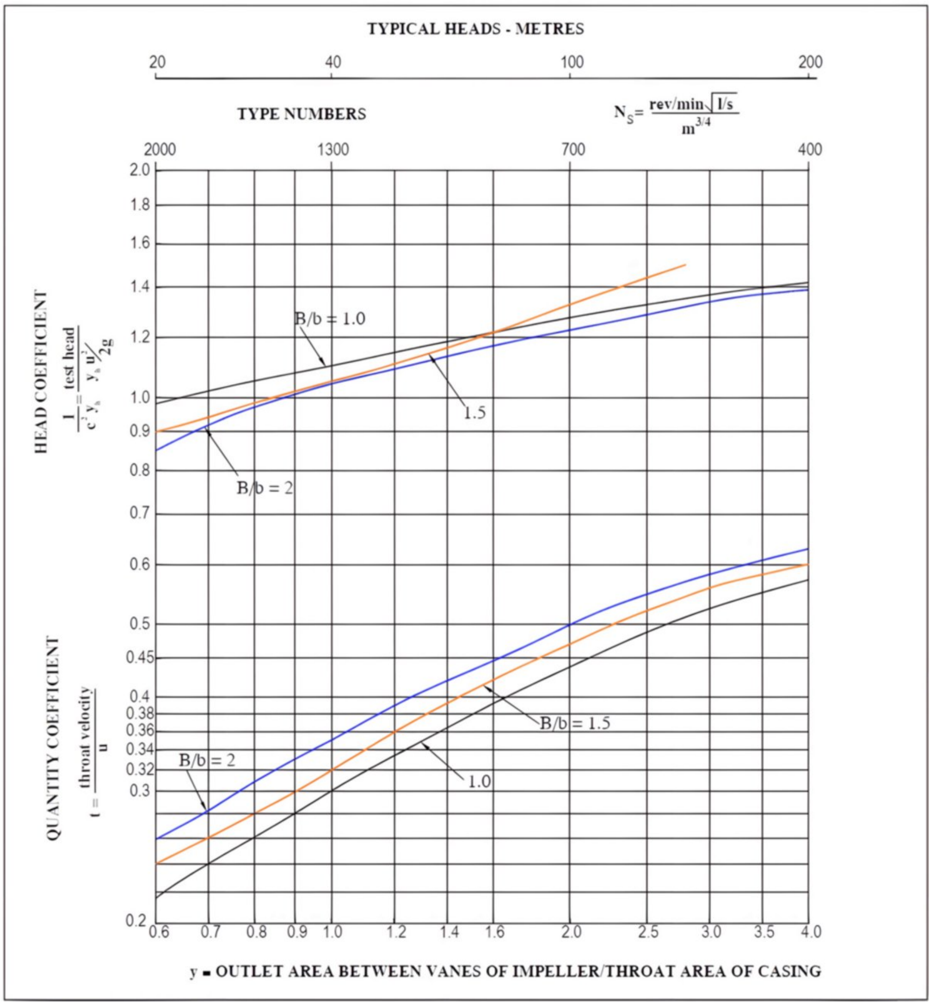


FIGURE 14: Relationship between area ratio, type number, flow and head coefficients

For predicting pump performance, area ratio method offers advantages compared to the other methods discussed so far. Experimentally, it has been found that H-Q and P-Q curves vary as shown in Table 4 depending on the area ratio used. An example of the actual variations in the pump curves based on different area ratios has been included as Appendix 2.

Y	<<	1	Steep head/flow curve
Y	<	1	Stable head/flow curve, Ns: 0.4 to 1.0
Y	<	0.75	Non-overloading power/flow curve, Ns: 0.8 to 1.2
Y	>	1	Small diameter machine, flat head/flow

TABLE 4: Relationship between area ratio and pump characteristics

Thorne defined the relative outlet area of the impeller as the area of the passage. He also suggests that the power/capacity curve depends on the area ratio. If the area ratio is more than three, the power curve will rise continuously. From Figure 15, the prediction of the peak power (P_{max}) and the flow (Q_{pmax}) at which peak power occurs can be calculated. However, a detailed study still needs to be made to validate these predictions.

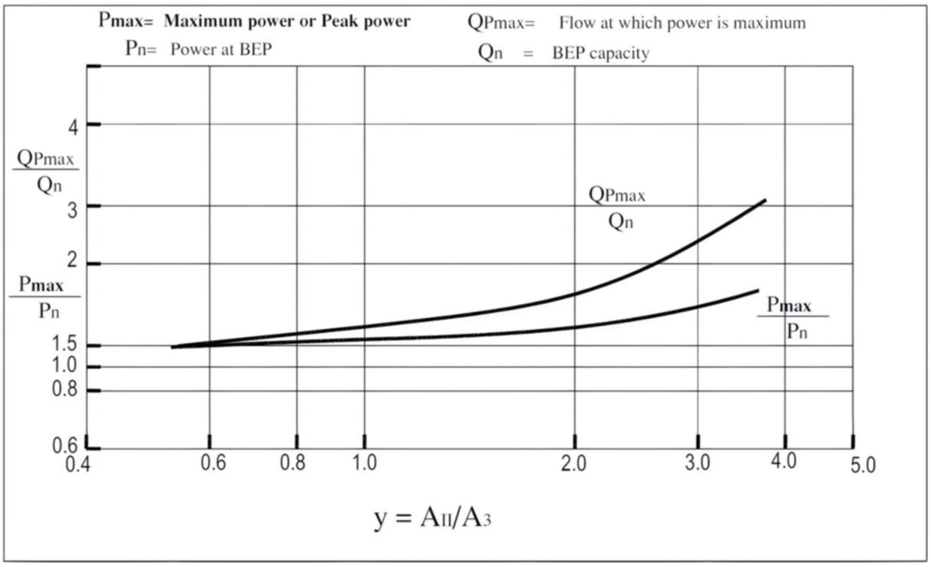


FIGURE 15: Effect of Y on the power curve (EW Thorne)

BEP, best efficiency point

Modifications in design to achieve NOL power characteristics

Modifications in the existing impeller design can affect the power characteristics of a centrifugal pump. Researchers have conducted different analytical studies and experiments in order to obtain NOL power characteristics. The prominent methods suggested so far are discussed below.

Contraction Type Impellers

The shaft power increases with the increase in the flowrate for low specific speed pumps. The area ratio theory suggests that by reducing the OABVs, the maximum power will be achieved at the theoretical head. In contraction type impeller [36], the suction surface thickness is altered to change the effective flow area. This does not vary the impeller inlet and pressure surface of the vane and helps to maintain the similar suction performance. The slope of the H-Q characteristic curve increases and the peak power appears on the P-Q curve.

However, thickening of the vane makes the velocity distribution non-uniform at the volute inner circle leading to high mixing losses and reducing efficiency.

In pump applications where driver rating depends on the peak power and this peak power is far right of the BEP, contraction type impellers can be used to shift the peak power to the left. Figure *16* shows diffusion and contraction type impellers on the same planar graph.

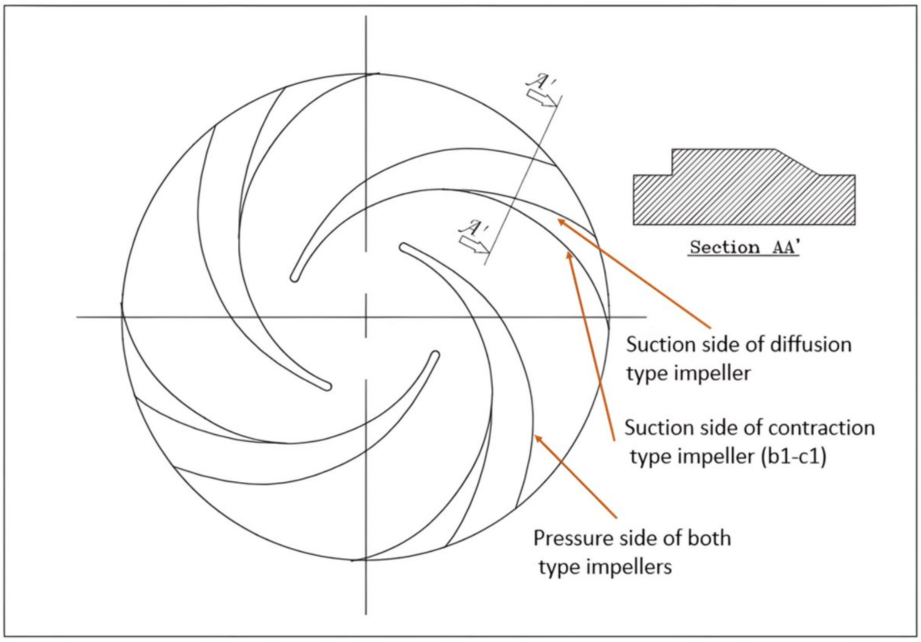


FIGURE 16: Diffusion and contraction type impellers on the same planar graph

Blockage of the Impeller Channel

The blockage of the impeller channel [37] has effect on the impeller outlet area (Figure *17*). It reduces the effective flow area and therefore the area ratio decreases. The reduction in turbulent kinetic energy in the impeller passage makes the H-Q curve steeper. The efficiency curves fall sharply at the right of the BEP and the power curve becomes NOL.

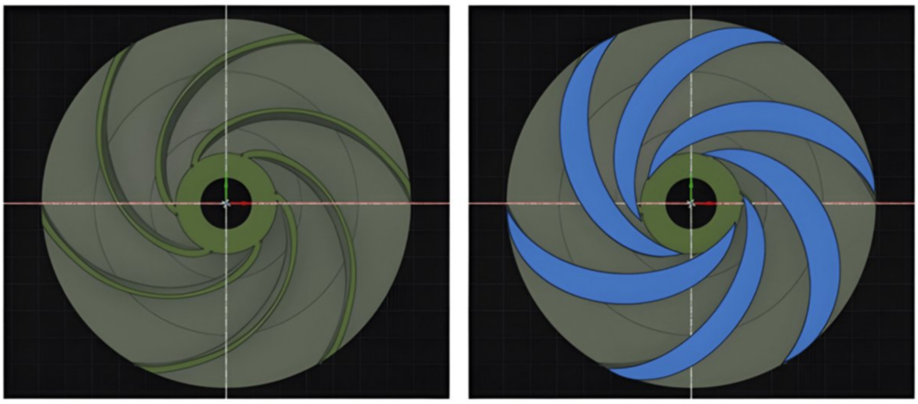


FIGURE 17: Normal impeller (A) and blocked impeller (B)

Groove and Gap Vane Design

While working on the performance characteristics in the low specific speed pumps, researchers developed two new impeller vanes: groove and gap vanes [38]. Compared to the conventional impellers, the new design improves the pump performance significantly.

Cutting a groove between the pressure and suction side of the impeller will prevent the boundary layer separation by allowing additional energy to the blocked flow. Efficiency of pump is improved, and therefore, the power is reduced.

Separation of boundary layer also occurs due to the angle of attack of the incoming flow. In order to control the angle of flow attack, a width gap was cut. The incoming flow enters the flow separation point through this gap and prevents separation. This has beneficial effect on pump efficiency and power.

However, the effect of gap and grooved vanes on the shape of the power curve and peak power has not been discussed in the published literature.

Introduction of Splitter Vanes

Different design modification in the impeller has been suggested by a number of researchers to modify the performance of the pumps. Among these modifications, inclusion of splitter vanes is worth mentioning. Splitter vanes are additional shorter vanes, which are equally spaced in between the main vanes. The inlet position of these vanes starts at a distance Res, which is above the inlet edge Re of the main vanes as shown in Figure *18*. It has been found that splitter vanes provide better guidance to the flow without any significant reduction in the flow passage and, therefore, increase the head due to better guidance. The flow increases marginally and the BEP shifts toward right [39]. Although there is an increase in the head, the efficiency remains unchanged due to an increase in the hydrodynamic losses. The splitter vane will reduce the effective impeller outlet area and reduce the area ratio without changing other impeller and volute design parameters. This can make the power characteristics NOL. The effect of the splitter vane on the peak power can be an area of further research. If the power curve is NOL type, addition of splitter vanes in the conventional impeller can shift the peak-power position.

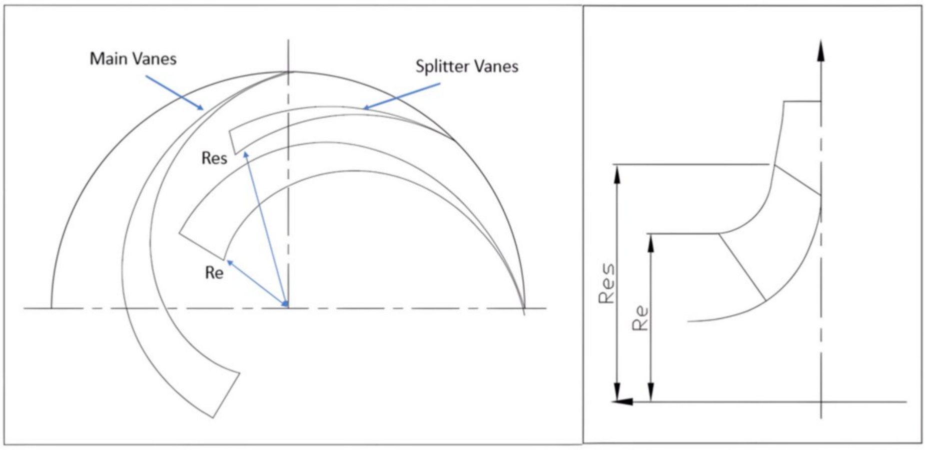


FIGURE 18: Polar view showing splitter and main vanes of the pumps

Conclusions

Concluding remarks

Historically, centrifugal pumps have been designed to achieve a specific design duty (capacity and head) with optimum efficiency. The power requirement of a pump depends on its capacity and head. Depending on the pump design, the power requirement may increase continuously with the flowrate overloading the motor and ultimately causing its trip. This is not acceptable in many applications, such as firefighting, nuclear industry, etc., for the protection of life and property against catastrophes. In such cases, special designs of pumps are required to ensure their unhindered operation for any flowrate. Such pumps are known as NOL pumps as they restrict the overload of the motor. The present communication provides a critical review of designing NOL pumps from a scientific point of view and points out the future research direction for obtaining this important goal.

While assessing various design approaches, it is observed that except Anderson's area ratio method, very few of the design approaches speak about NOL or inverted parabolic power curves. This classical area ratio method can be used to obtain NOL power characteristics and there are experimental evidences in support of this design approach. It has also been observed that low area ratio ($Y < 0.75$) results in a larger impeller diameter and throat area, a stable head-capacity characteristics and NOL power characteristics. However, larger impeller diameter increases disc friction and, therefore, reduces the efficiency.

Since one of the purposes of this communication has been to assess various design approaches that can be considered for achieving NOL power characteristics in centrifugal pumps, the findings have been summarized.

Modeling: No established empirical relationships to predict or achieve NOL power characteristics

Modification: No direct correlation with NOL power characteristics; relies on experience and intuition of pump designers.

Inlet design: No proven relationships with the NOL power characteristics based on published literature.

Free vortex in the volute casing: Does not directly address NOL power characteristics.

Method of design coefficient: Has limited success in achieving NOL power characteristics; requires large number of trial-and-errors, no direct predictive relationship.

Method of area ratio: The only classical approach that directly addresses the NOL power characteristics; supported by empirical models and experimental validation.

It has also been observed that design modifications like creating blockages in the impeller channel, contraction type impellers and induction of splitter vanes help to achieve NOL power characteristics but increases the hydraulic losses leading to lower pump efficiency. However, these design modifications require extensive research including both simulation and experimental results to find ways of controlling pump power characteristics in a more rigorous scientific approach.

The performance characteristics of a pump depend on the rather complex flow behavior of the fluid, which, in turn, depends on a large number of design parameters and operating conditions. In the present article, we have identified the effect of the design parameters on the loading (power) characteristics of a centrifugal pump from a scientific point. Then based on the practical design methodology, it has been debated how one can achieve the design of NOL pumps in the background of state-of-the-art knowledge.

Optimization of the pump efficiency is of great importance in the pump industry. But the question is whether the shape of the pump power characteristics curve can address the safety of the pumping system as well as the optimization of the overall efficiency of the system by reducing the driver rating. Therefore, the challenging dilemma for the hydraulic designers is to design a pump with NOL power characteristics without sacrificing the efficiency. This review shows that inadequate research initiatives have been taken for establishing broad guidelines in this direction. Systematic research endeavor combining experiments and computation is needed to achieve NOL characteristics of centrifugal pumps.

Appendices

Appendix 1: application of model law in centrifugal pumps

Let us consider an example of how the model law is applied to an existing 5” × 6” (Q_m = 252m^3/hr, H_m = 33m, N = 1480rpm) size pump to make a new pump of size 10” × 12” (Q_m = 1430m^3/hr, H_m = 105m, N = 1480rpm).

Specific speed (N_s)

Specific speed N_s defines the impeller type, performance characteristics and achievable efficiency of a centrifugal pump.

Specific speed (N_s) = $\dfrac{NQ^{0.5}}{H^{0.75}}$ US units

where,

N-operating speed in RPM

Q-flow in USGPM

H-head developed by each impeller in ft.

For centrifugal pumps, specific speed (Ns) ranges from 300 to 10000 US units

Whenever a new pump is manufactured, the pump manufacturer always tries to compare with the test reports of similar specific speed pumps to predict the performance and design parameters of the new pump.

In the example (Table 5), we have shown calculation of some of the major geometric parameters of the prototype pump. The other geometric parameters can be scaled up using the multiplier f. Figure 19 illustrates the scaled comparison of the prototype pump and the model pump.

Parameters	Model pump (existing) (5” × 6” - 13”)	Prototype pump (new) (10” × 12” - 24”)
Capacity (Q—m^3/hr)	252	1430
Head (H—m)	33	105
Speed (N—rpm)	1480	1480
Specific Speed (Ns—US units)	1469	1469
f (Q)	31460252 = 1.78	
f (H)	210533 = 1.78	
Impeller diameter (mm)	334	334 × 1.78 = 594
Impeller width (mm)	30	30 × 1.78 = 54
Impeller eye diameter (mm)	150	150 × 1.78 = 267
Throat area (sq cm)	57	57 × 1.78^2 = 181
Power (KW)	26	26 × 1.78^5 = 464.5

TABLE 5: Example of model law on a centrifugal pump

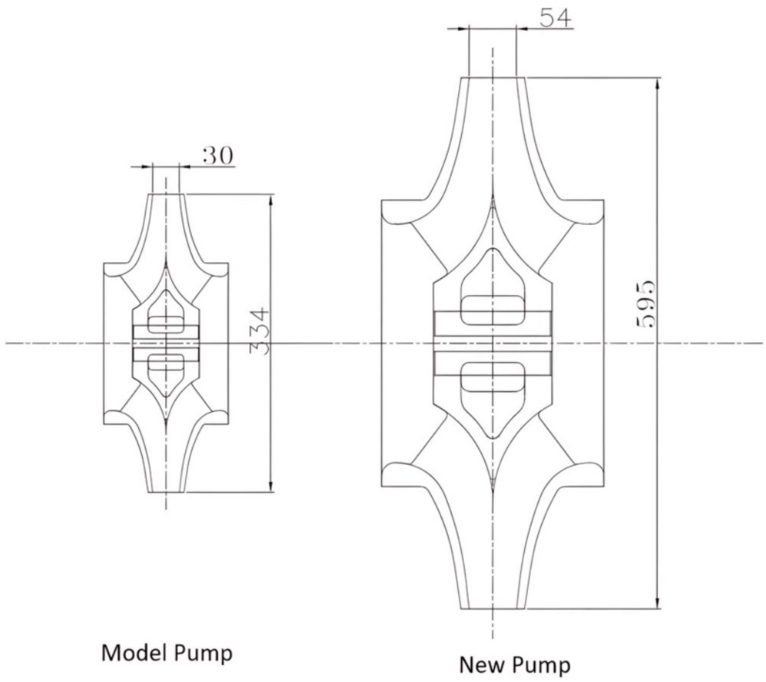


FIGURE 19: Impeller of model pump and the new pump

Appendix 2: procedure for calculation of area ratio

Area ratio can be used to determine the critical hydraulic design parameters of a centrifugal. The major hydraulic parameters of the casing and an impeller are shown in Figure 20. Similarly, if the area ratio is known, an approximate prediction of the shape of the H-Q and P-Q characteristics can be drawn.

Example of Area Ratio Calculation

In order to calculate the area ratio, the design parameters, such as impeller diameter, impeller width, outlet vane angle, throat area, etc., need to be known. On the other hand, if we need any specific pump characteristics, we can choose an area ratio and determine the design parameters of a centrifugal pump.

Table 6 shows the calculation of the area ratio of two pumps.

Parameters	Pump model 8HS12	Pump model 8HS22
D_2 mm	305	560
b_2 mm	51	37
β_2 mm	22o	18o
A_3 cm^2	173	143
Outlet area between vanes ($0.95\ \pi\ D_2\ b_2\ \sin\beta_2$)	173.81	190.99
Area ratio (Y)	1.0	1.33
Head rise to shut-off	133%	119%
Shape of power curve	Flat/non-overloading type	Rising type
H-Q curve steepness	Steeper	Less steep

TABLE 6: Example of area ratio calculation of a centrifugal pump

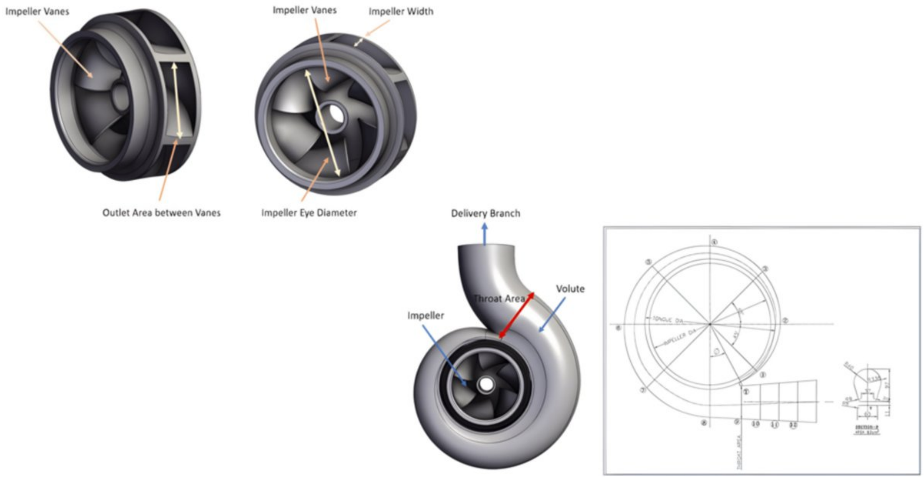


FIGURE 20: Hydraulic parameters of an impeller and a volute

Appendix 3: notations

Appendix 3 provides a list of notations, including the corresponding parameters and their descriptions, as shown in Table 7.

Parameters	Descriptions
Y	Area ratio
OABV	Outlet area between vanes
D_2	Impeller outer diameter
b_2	Impeller width
β_2	Outlet vane angle
A_3	Throat area
Ψ	Head coefficient
u_2	Peripheral velocity of impeller at outlet
u_1	Peripheral velocity of impeller at inlet
Q	Capacity
P	Power
η	Efficiency
H	Head
NPSH	Net positive suction head
C_{u2}	Tangential component of absolute velocity at outlet
BEP	Best efficiency point
K_h	Steepness
ω	Rotational speed
C_{m2}	Radial velocity at impeller outlet
C_{m1}	Radial velocity at impeller inlet
Z	Number of vanes
t	Vane thickness
t_u	Vane thickness at D2
P	Static pressure
P_0	Atmospheric pressure
t_q	Quantity coefficient
ϕ	Capacity coefficient
B	Volute width
h_o	Busseman’s head coefficient
He	Euler's head

TABLE 7: Notations

Additional Information

Author Contributions

All authors have reviewed the final version to be published and agreed to be accountable for all aspects of the work.

Concept and design: Sandipan Trivedi, Prasanta Kumar Das

Acquisition, analysis, or interpretation of data: Sandipan Trivedi, Himadri Sen, Prasanta Kumar Das

Drafting of the manuscript: Sandipan Trivedi, Himadri Sen

Critical review of the manuscript for important intellectual content: Sandipan Trivedi, Himadri Sen, Prasanta Kumar Das

Supervision: Himadri Sen, Prasanta Kumar Das

Disclosures

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