Master's Thesis

Product standardization as a strategy for a globalized and digitalized market

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Case study based on a Vertical Line Shaft Pump

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STATUTORY DECLARATION

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Abstract

In todays globalized and prospective digitalized markets, companies must enhance ways to approach change. Due to globalization, more competitors emerge and market prices decrease. Digitalization increases efficiency and automatizes parts of the value creation chain.

This thesis aims to demonstrate that the product strategy – standardization – is a good market approach, to use the cost advantages of a globalized market and to implement the product within the corporate digitalization strategy.

First, a market analysis was conducted, to get a clearer picture of how big the future market potential is and for which sizes standardization will be most promising. A competitor analysis helped to identify the current common designs and in which ranges the competitors have standardized their portfolios. The internal cost analysis examined the most cost-intensive parts of the product and helped to determine the future cost saving potentials. By means of an analysis of the internal hydraulics, the usability for standardization was decided. These hydraulics were the basis for a Standard Pump Mapping. Finally, a standard calculation tool and standard pump parts were established.

The results of the product standardization are a shorter lead time, lower internal costs and thus, lower achievable sales prices and higher competitiveness in a dynamic and price-driven market. According to standard calculations and based on a parametrized design, the engineering process can be accelerated and shortened. Due to standardized parts, the variant management and the automatized Bill of Materials creation can be pursued easier. The standardization enables a clear and fixed product structure.

Kurzfassung

Aufgrund der Globalisierung und Digitalisierung der Märkte, müssen Unternehmen ihre Strategien anpassen, um diesen Wandel erfolgreich zu meistern. Durch die Globalisierung gibt es mehr Mitbewerber und die Marktpreise sinken. Die Digitalisierung steigert die Effizienz und viele Teile der Wertschöpfungskette sind automatisiert.

Diese Masterarbeit soll aufzeigen, dass die Produktstrategie – Standardisierung – ein guter Ansatz ist, um die Kostenvorteile eines globalisierten Marktes zu nutzen und das Produkt in die Digitalisierungsstrategie des Unternehmens zu integrieren.

Zunächst wurde eine Marktanalyse durchgeführt, um ein klares Bild zu bekommen, wie groß das zukünftige Marktpotenzial ist und welche Größen für eine Standardisierung am wichtigsten sind. Eine Konkurrenzanalyse half dabei, die aktuell üblichen Designs zu vergleichen und festzustellen in welchen Bereichen die Mitbewerber ihre Portfolios standardisiert haben. Die interne Kostenanalyse untersuchte die kostenintensivsten Bauteile des Produkts und half das zukünftige Kosteneinsparpotenzial zu ermitteln. Anhand einer Analyse der firmeninternen Hydrauliken wurde die Verwendbarkeit dieser für die Standardisierung bestimmt. Diese Hydrauliken waren die Basis für ein Standard-Pumpen-Mapping. Schließlich wurde ein Standard-Berechnungstool erstellt und es wurden Standardgrößen für diverse Pumpenteile festgelegt.

Durch Einführung dieses Standardberechnungstools und eines parametrisierten Designs kann der Entwicklungsprozess verkürzt werden. Das Ergebnis der Produktstandardisierung werden kürzere Lieferzeiten und niedrigere Entwicklungskosten sein, wodurch niedrigere Verkaufspreise erzielbar sind. Die Wettbewerbsfähigkeit dieses Produkts wird als Resultat in einem dynamischen und vom Preis bestimmten Markt erhöht. Durch standardisierte Bauteile kann das Variantenmanagement und die automatisierte Erstellung von Stücklisten einfacher verfolgt werden. Die Standardisierung ermöglicht eine klare Produktstruktur.

1 Table of contents

2 Introduction

2.1 Current situation

The Andritz AG, Division Pumps, is a leading vendor for different industries, such as: pulp and paper, water and waste water, mining and much more.

The extensive product portfolio comprises several standard pumps like end-suction centrifugal pumps, axially split case centrifugal pumps, medium consistency centrifugal pumps, submersible pumps and -motors and others.

Furthermore, the portfolio consists of engineered pumps for customized applications and higher head or flow rate pumps. These are vertical volute pumps, axially split case multistage pumps, high pressure pumps and vertical line shaft pumps.

In this thesis, the vertical line shaft pump (VLSP) is analysed in detail and a standardization concept with prospected cost savings is examined.

The vertical line shaft pump is already in the company's product portfolio for a long time. This product is currently managed at two locations with different market approaches. At the headquarter in Graz, the VLSP is an engineered pump especially for bigger sizes. For smaller sizes the company has a vendor for which the Andritz AG delivers a standardized range. Currently every pump is engineered from its subsidiary in China.

At present the Andritz AG is not competitive enough with this product, especially with smaller sizes because competitors have certain standardized sizes and produce in low cost countries. By means of vendoring a standardized range, some know-how was lost.
Due to that, and in order to control and have influence on the costs, the demand for standardization in-house is a hot topic already since quite some time.

2.2 Target of the thesis

This thesis aims at examining in which areas it is most promising to standardize the VLSP and how this can be executed. The cost saving potential will be evaluated to analyse if the VLSP will be competitive again after the standardization. This product should not be positioned as a low-cost product; it will still be a highly technological product since the company offers high level hydraulic know-how and hence, the best solution can be provided. Furthermore, the company offers different options like adjustable blades to ensure best performance for different operating points, multi-stage design for higher heads or pull-out design. With these features and a competitive price this product should be a more important product for the future.

The following 3 targets are crucial:

- Use of cost advantages due to globalization and based on its subsidiaries
- Alignment of the product to the divisional digitalization strategy
- Reducing lead time

Additionally, to have a comprehensive global product strategy, in future the company also wants to produce the smaller sizes in-house and thus gain the gross margin of the vendor as its own gross margin.

Furthermore, as Industry 4.0 is a big topic nowadays and to align the product to the divisional digitalization strategy standard tools will be established, such as a calculation tool, and the standard sizes will be integrated into the variant management. Standard hydraulic sizes will be established. These measures will help to fasten up processes and reduce lead time. Due to that and as a follow up, a parametric design of the vertical line shaft pump will be established to reduce the design time for such a product.

In Chapter 3, the product strategy, the facts for standardization and the vertically suspended centrifugal pumps are introduced in general. Also, the main product in this thesis – the vertical line shaft pump – is described in detail and important issues referring to each part of the pump are examined. In Chapter 4, the current market situation is analysed and the different application possibilities are shown. In Chapter 5, the competitive situation is described in detail and the standardization level of the competitors is analysed. In Chapter 6, the cost of the product is examined and potential savings are identified. Chapter 7 analyses the different hydraulics and a standard hydraulic mapping is developed. Finally, in Chapter 8, the planned standardization concept with the standard calculation tool is described and standard sizes are established.

3 General

3.1 Product strategy

A company usually intends to grow and to develop its business. The growth strategy implies that the company can use economies of scale, economies of experience, the increase of the market power and is protecting jobs. (Vorbach, 2015, p. 199)
To grow naturally, a strategy based on products should be examined.

According to Ansoff, there are 4 strategic directions for growth. Ansoff`s model is used to derive the right strategy for divisional growth (Vorbach, 2015, p. 199). Usually a company starts with its existing products in existing markets. Afterwards, the company has the possibility to penetrate the current market with the current product or to move further with product development within the existing market, develop a new market with the existing products or to go a complete new way with new products in new markets called diversification (Johnson, et al., 2009, p. 174).

For the VLSP the strategy is to grow in the existing markets with a current product. This means that the VLSP is supposed to grow by means of the market penetration strategy.

The following advantages accompany this strategy:

- Use of existing company capabilities
- Higher sales with existing customers
- Divisional orientation stays the same
- New customers by exploiting weaknesses of competitors
- Scaling and experience advantages (Vorbach, 2015, p. 200)

This strategy implies a further product development as well but not to such an extent as with the product development strategy. (Johnson, et al., 2009, p. 174)

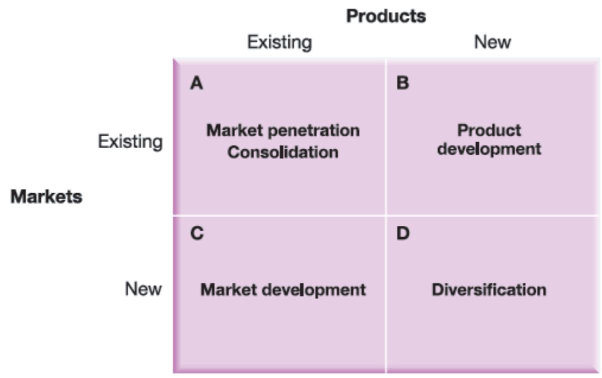


Figure 1: Strategic directions Ansoff matrix (Johnson, et al., 2009, p. 174)

For the market penetration, the strategy of product standardization was chosen.
The advantages and disadvantages, according to literature, are shown in the next chapter.

3.2 Product standardization

To achieve the goal described in the last chapter, the product standardization was selected. According to the relevant literature the product standardization will provide the following advantages and improvements:

- Easier and cheaper procurement costs
- Increasing recycling chances
- Price reduction according to higher quantity
- Standardized services (Hofbauer & Hellwig, 2009, p. 325)
- Cost savings, the R&D engineer will be available for other tasks
- Stable quality
- Higher efficiency within the development process (Hebenstreit, 2009, p. 12)
- Faster time to market
- Improvement of single functions
- Better bargaining position when buying bigger amounts
- Standard products are tested less (Hebenstreit, 2009, p. 21)

Disadvantages of product standardization:

- Initial investment without a financed project as background
- Less flexibility regarding customer preferences (Hebenstreit, 2009, p. 13)

The product development is classified in different categories. In Figure 2, on the next page, the standardization level is diminishing with increasing standardization class. The first class, which contains standard parts, refers to parts which are standards such as screws, washers and many others. The last class is the class of engineered parts, whereas all products are specified according to customer requirements. The classes in between are standardized up to a certain extent e.g. the parametric design derives the design of a template but stays flexible according to customer requirements for certain parts.

The decision to which extent products shall be standardized should be taken according to how often a product or assembly is used with the same specifications, hence, customer preferences are the main determinant factor for companies. The fact that development costs rise the higher the customer-specific level is, must be considered as well. This is illustrated in Diagram 1.

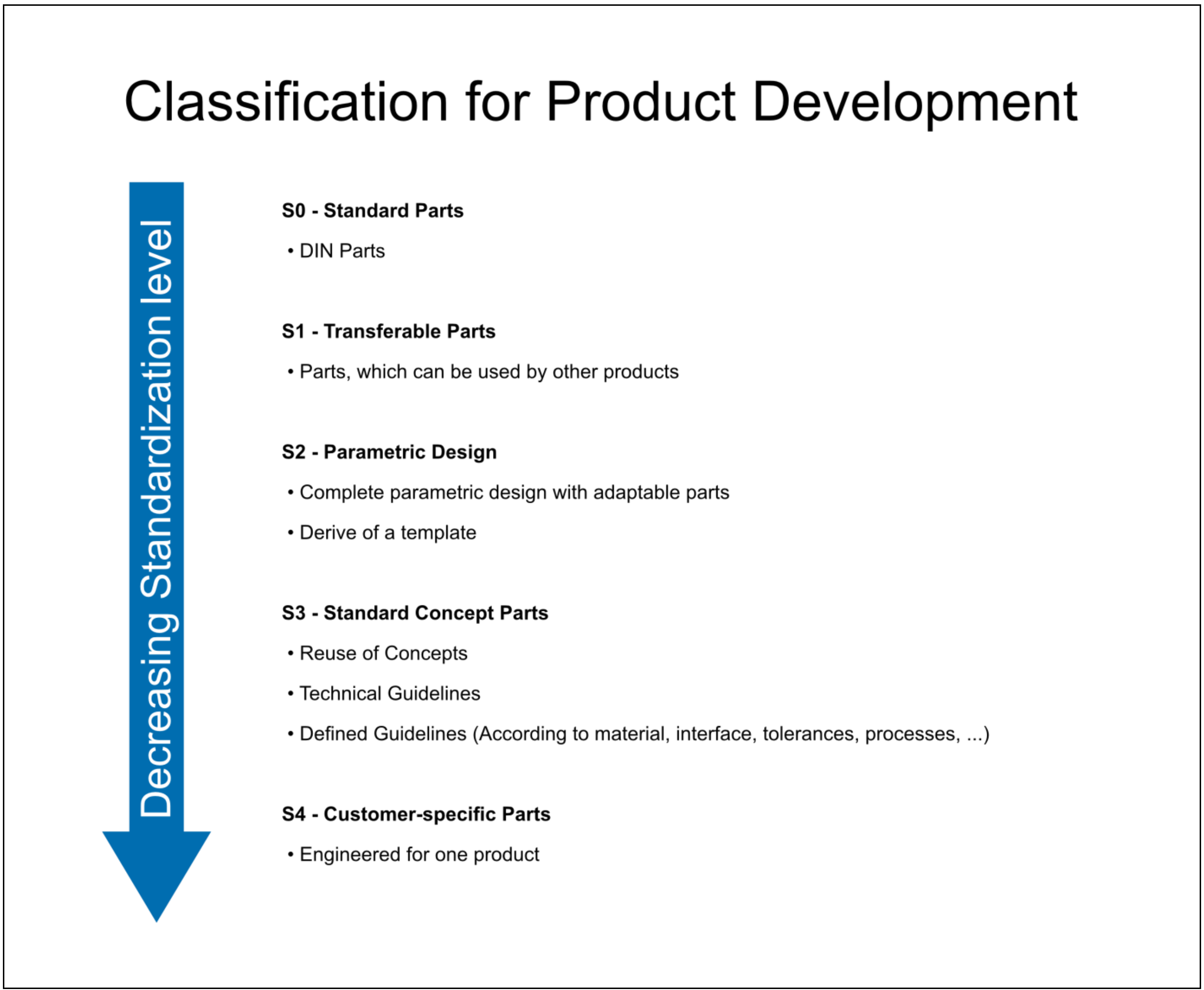


Figure 2: Standardization classes (Hebenstreit, 2009, p. 17) – (source modified)

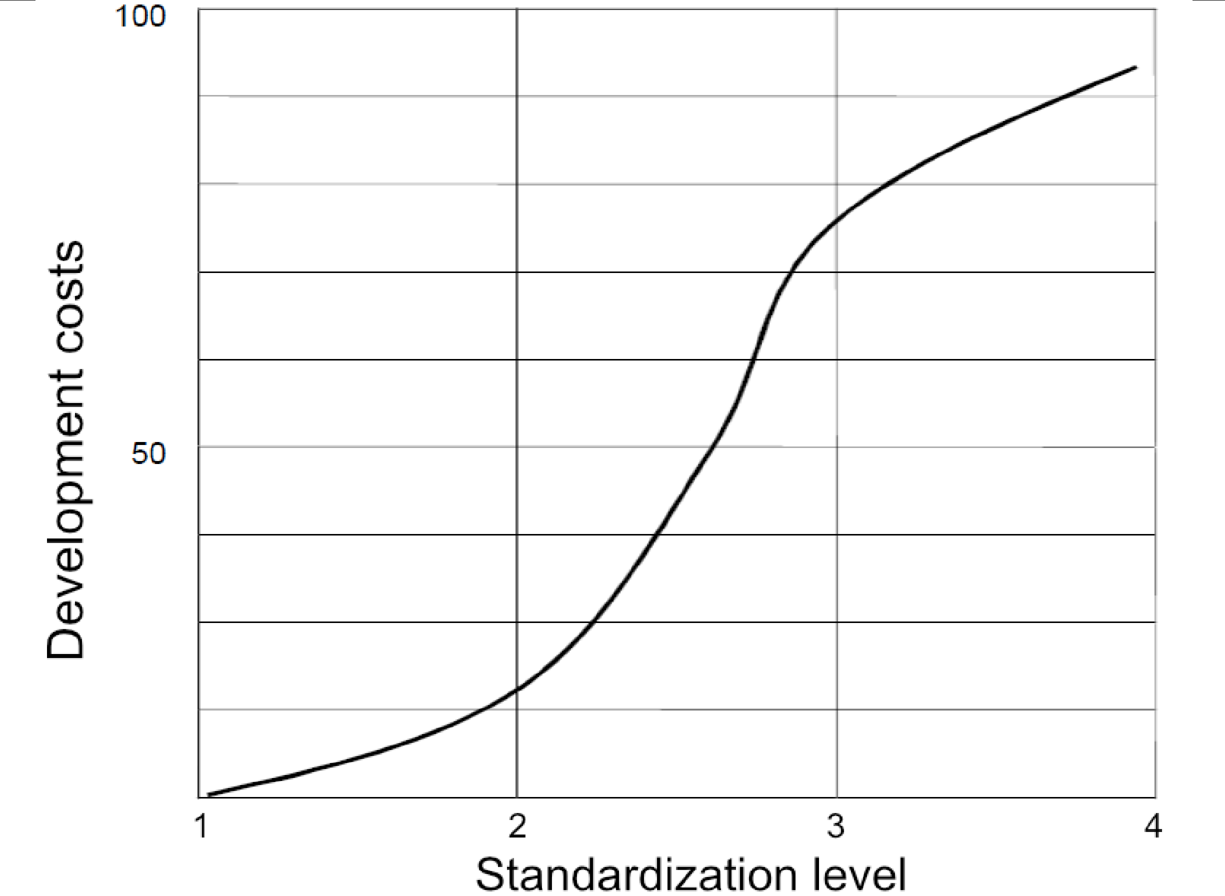


Diagram 1: Development costs (Hebenstreit, 2009, p. 19) – (source modified)

In some industries, the platform strategy is quite famous, also across different brands, e.g. the automotive industry. With that strategy, additional savings are generated. (Hofbauer & Hellwig, 2009, p. 325)

In the hydraulic industry, particularly the pump industry, some standards are used especially for the most common industries, such as the water or the chemical industry. Often, the same pumps are used for different applications and the design is adjusted to customer needs. Equal parts or assemblies, such as bearing supports, sealing systems or hydraulics, are used crosswise, for several products, hence, the advantages mentioned apply. The main task of product standardization is to use the same parts in different products or the same sizes for different applications.

3.3 Vertically suspended centrifugal pumps

Vertically suspended centrifugal pumps are used where less space is available and self-priming operation is required. Either clean- or contaminated fluids are pumped.

There are several different vertically suspended pumps for industrial applications available. These are classified below according to Figure 3.

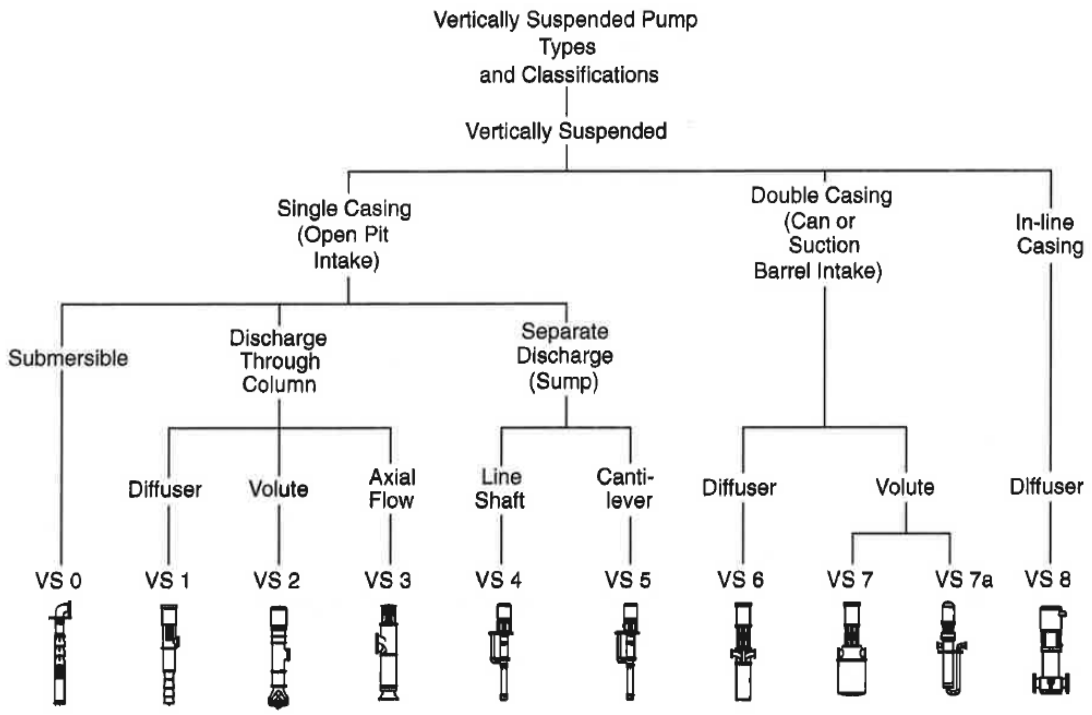


Figure 3: Vertically suspended pump types and classifications (ANSI/HI 1.3, 2013)

The different vertically suspended pumps are briefly described on the next page in Table 1. Figures 4 - 7 illustrate the different pumps.

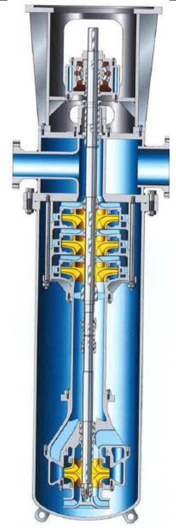
Vertical sump pumps (separate discharge)	**Vertical line shaft pumps (discharge through column)**	**Submersible pumps**	**Double casing pumps (Condensate pump)**
VS4, VS5	VS1, VS2,VS3	VS0	VS6
 **Figure 4: Sump pump (Andritz, internal source)**	 **Figure 5: Vertical line shaft pump (Andritz, internal source)**	 **Figure 6: Submersible pump (Andritz, internal source)**	 **Figure 7: Condensate pump (Hellmann, 2011, p. 381)**
Separate discharge pipe	Discharge through column pipe	Discharge through column pipe	Discharge through column pipe
Volute casings	Guide vanes	Guide vanes	Guide vanes
Clean or slightly contaminated water, fibrous slurries and liquids containing large solids from the deep sumps. (Sulzer – Vertical sump pumps, 2017)	Clean water or slightly contaminated water, salt water	Clean water or contaminated water	Clean condensate
Dry installed motor	Dry installed motor	Wet installed motor	Dry installed motor
Water- and waste water applications	Water intake applications	Mining- and well applications	Power plant applications
+ Direct installation in the pump sump (Sterling SIHI, 2000) + Different impeller modifications possible for different slurry and sump applications with e.g. single vane impellers + No suction or inflow pipe needed – Limited length (Sterling SIHI, 2000)	+ Is radially smaller designed than with a separate discharge pipe + Slide bearings are mostly lubricated with the pumped liquid + Direct installation in the pump sump (Sterling SIHI, 2000) + Easier access to wear parts + Less area required + No suction or inflow pipe needed – Higher costs for the building (Sterling SIHI, 2000) – The motor must be installed flood safe (Sterling SIHI, 2000)	+ Small diameters to fit into mining holes + Slide bearings are mostly lubricated with the pumped liquid + Easy heat removal with the pumped fluid (Hellmann, 2011, p. 293) + High submergence possible, up to several kilometres. (Hellmann, 2011, p. 293) + No shaft transmission (Sterling SIHI, 2000) – The hydraulic is designed at the expense of the diameter – No access to the wear parts	+ The $NPSH_{av}$ can be easily increased by extending the barrel + First stage double suction for low $NPSH_{req}$ + Operation close to steam pressure + Used when closed circuits are needed – Higher costs

Table 1: Comparison of the vertically suspended centrifugal pumps

3.4 Vertical line shaft pump

According to Figure 3 above, this kind of pump is classified as VS1 to VS3, the discharge through column vertically suspended centrifugal pumps. These are named vertical line shaft pumps company internal. Vertical line shaft pumps (VLSP) are also often called vertical turbine pumps (VTP).

This type of pump was used as a deep well pump. The outer diameter of these pumps is important and often the impeller is a mixed- and axial flow type. However, this type provides certain benefits for other industries as well. (Bloch & Budris, 2010, p. 354) These benefits are described on the next page.

The water intake is at the bottom at the suction bell. The conversion from speed to pressure is realized in guide vanes. Thus, this pump is often called a diffusor-pump. Afterwards the pumped fluid is directed through the column pipe.
The radial bearings are sliding bearings and mostly lubricated with the pumped liquid.
The discharge elbow directs the fluid to the pressure gate. The shaft is sealed at the discharge elbow with a stuffing box or a mechanical seal. The motor stool is located above the discharge elbow, and inside of that the axial thrust bearing and the motor coupling. The motor is mounted on top of the motor stool. The axial thrust bearing is either a tilting pad bearing or a roller bearing. (Hellmann, 2011)

The pump efficiency in centrifugal pumps is measured between the pressure port and the suction port. In typical end-suction pumps the efficiency includes the impeller and the volute casing. In comparison, a VLSP includes the impeller, the guide vanes, column pipes, diffusors and elbows. Thus, the column pipe, diffusors and the discharge elbow must be taken into consideration when designing such a pump, to ensure high efficiency.

The vertical line shaft pump is designed hydraulically with an axial-, mixed- or radial flow impeller. This depends on the flow rate and the head. Which impeller is used for which configurations is described in detail in Chapter 3.5.3.

The VLSP can be realized in different design configurations. This is described and illustrated in Chapter 3.5 – Design options. The main design is shown in the Figure 8 on the next page.

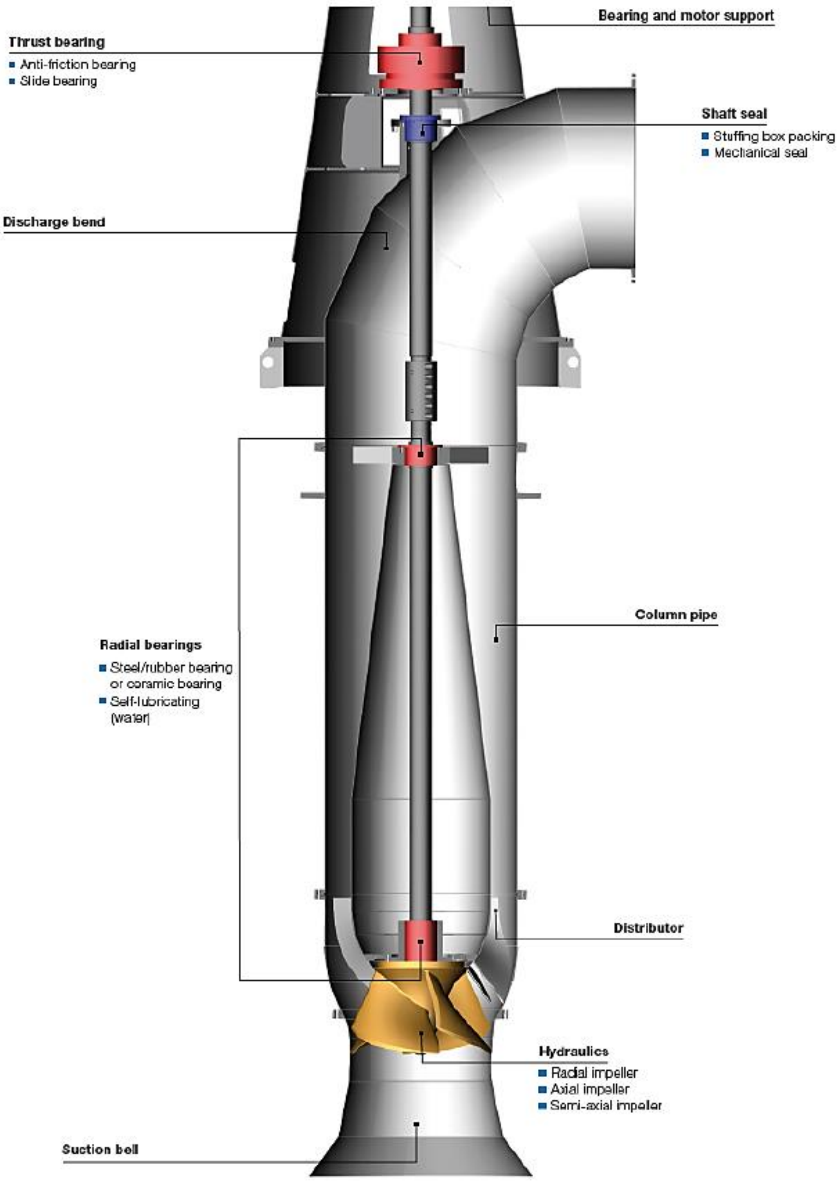


Figure 8: Main design (Andritz – Vertikale Rohrgehäusepumpen)

The advantages in comparison to a horizontal pump are:

- VLSP pumps are submerged for a certain overlay → thus these types are operating in self-priming operation.
- Due to the fact that the motor stool with the motor is above the floor, it occupies minimal floor space.
- The installation and the foundation design are rather straightforward and cheap.
- $NPSH_{available}$ can easily be increased by extending the column pipe length.
- High efficiency is obtained with a multistage design.
- This pump can be designed in a modular way according to its application. (Bloch & Budris, 2010, p. 354)

3.5 Design options

3.5.1 Different discharge configurations

The different discharge options depend on the site conditions available, such as space and limitations of floor load. For this pump, 4 different discharge configurations are possible.

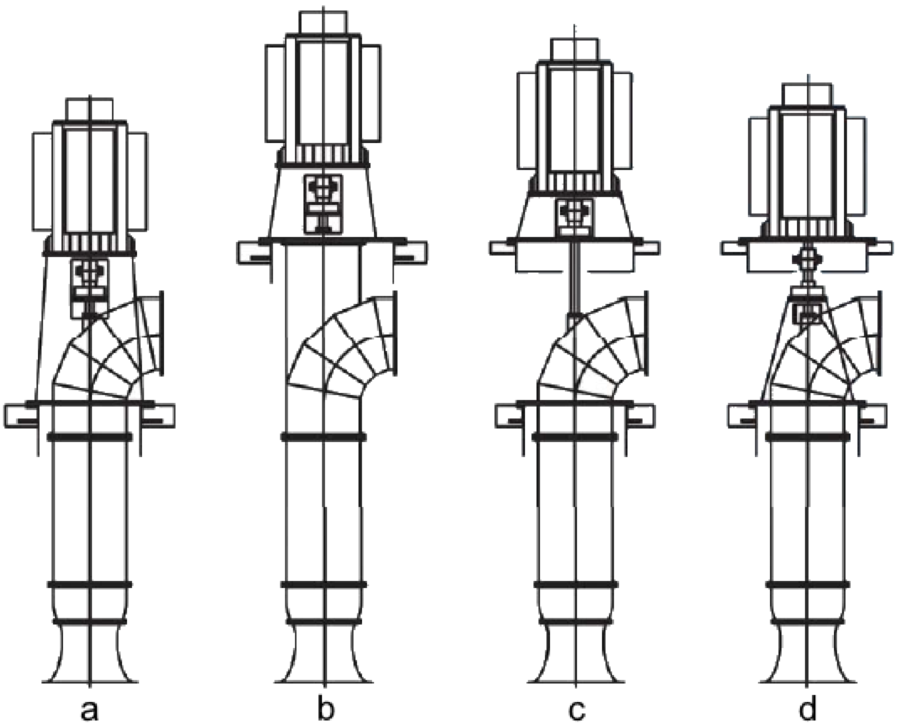


Figure 9: Discharge configurations (Andritz – Vertikale Rohrgehäusepumpen) – (source modified)

<u>a) Above ground</u>

The discharge head is located above the ground. This is the most common configuration and is especially used if the pressure pipe and armatures must be installed dry.
With this configuration, the whole pump and motor weight, water filling included, is carried by the fundament.

<u>b) Below ground</u>

The discharge head is located below the ground. This is used when the space above the floor is limited and the pressure pipes and the armatures must be installed wet. With this configuration, the whole pump and motor weight, water filling included, must be borne by the fundament.

c) d) Double deck installation

This configuration is used when the motor floor cannot take the complete weight of the pump. Both configurations are double deck whereas the location of the axial thrust bearing differs. In c) the axial thrust bearing is located in the motor and in d) it is located at the discharge head. Thus, in c) the axial load is carried by the motor floor and in d) by the pump floor.
Thus, the weight is separated and the pump floor has just to bear the pump weight.
These configurations relate to higher requirements regarding the sealing of the building.

Other extending discharge configurations are available as well e.g. a horizontal one or, as shown below in the Figure 10, a slanting configuration. Such slanting configurations are used because of service reasons – when the pipes are axially split, maintenance of the inner parts is easier.

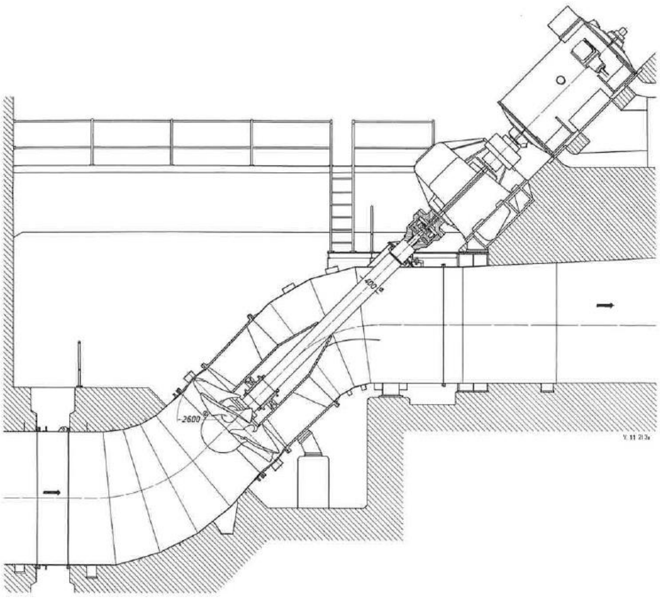


Figure 10: Slanting configuration (Voith – Schrift 2269)

3.5.2 Pull-out design

This design option enables removing only the runner unit without pulling out the whole pump unit. The ease of service is the determinant factor using this design. This design option is often preferred by customers for bigger diameter pumps, hence, for higher flow rates. The slide bearings and other wear parts can be easily maintained and the standstill time is reduced. In addition to that, the hall cranes lifting capacity can be reduced and less floor space for storing the whole pump is required. Of course, this design requires more investment costs and the weight is a bit higher but less operating costs occur due to easier maintenance. A pull-out VLSP is illustrated in Figure 11 on the next page.

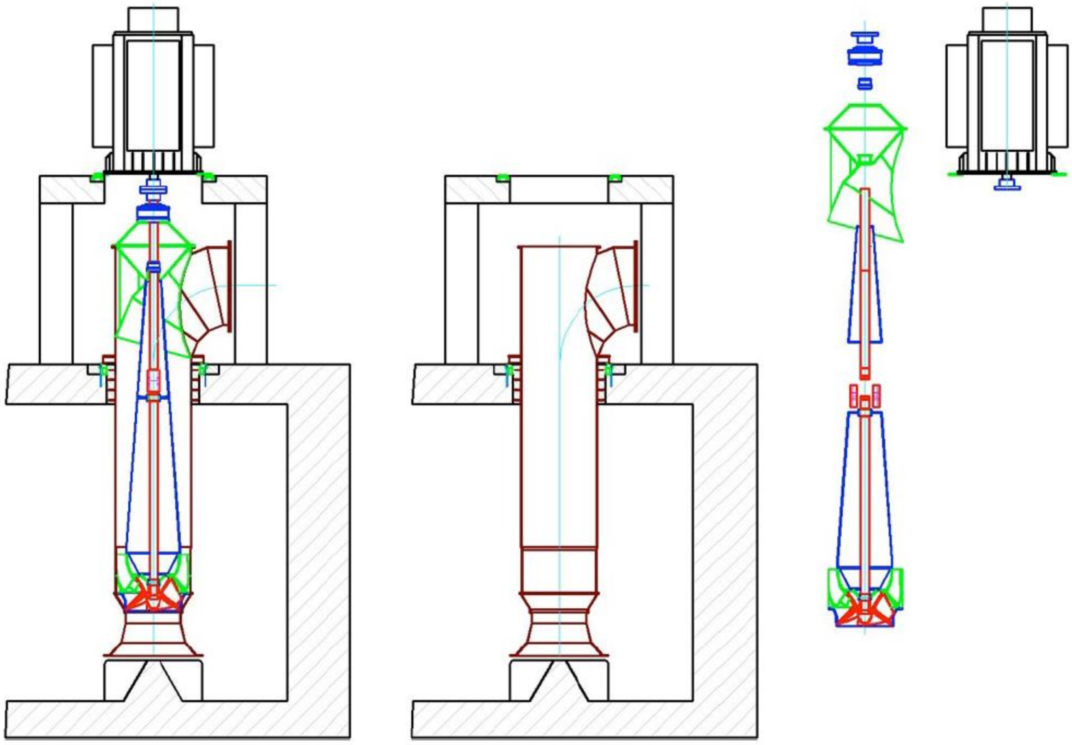


Figure 11: Pull-out design (Andritz – Vertikale Rohrgehäusepumpen)

3.5.3 Impeller

The impeller transfers the rotating energy to the fluid and increases total pressure. Depending on the specific speed (nq) and the characteristics of the streamlines through the impeller, the impeller design is classified as radial, semi-axial (mixed) or axial. With increasing specific speed, the flow through the impeller changes from radial to axial. In Figure 12 the different designs are illustrated.

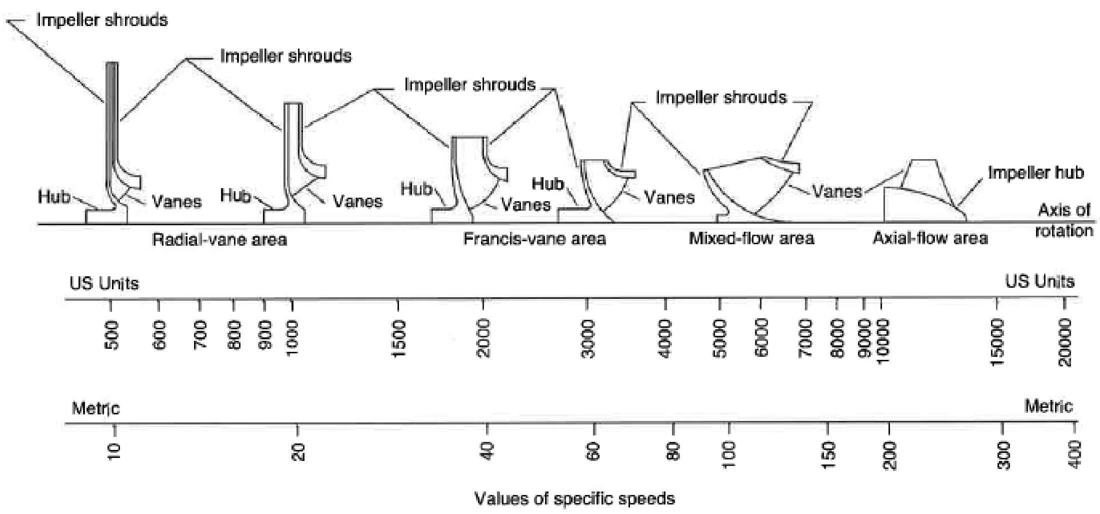


Figure 12: General impeller types (ANSI/HI 1.3, 2013, p. 12)

The characteristics of the pump curve mainly is a function of the specific speed, hence, of the design of the impeller. With increasing specific speeds the steepness of the pump curve increases, which is illustrated in Figure 13. The axial type (high specific speed) has a steeper pump curve than the radial one (low specific speed). The high specific speed has a distinct instability in the pump curve, hence, this limits the operating range when impellers with high specific speeds are used.

The efficiency curve of a low specific speed pump at the maximum efficiency is flatter than a high specific speed pump where the efficiency decreases fast after the maximum. This systematic is illustrated in Figure 13.

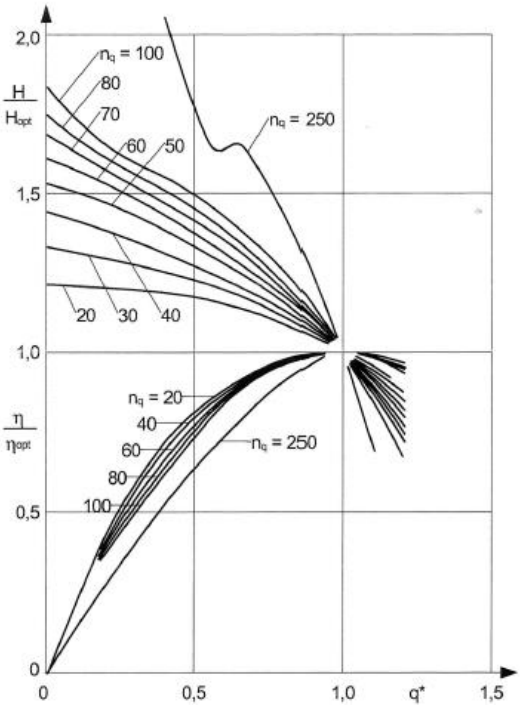


Figure 13: Pump- and efficiency curves as a function of the nq (Jaberg, 2012, p. 403)

In Figure 14 the power consumption as a function of the specific speed and the specific flow are illustrated. The most important information in this picture is that low specific speed pumps have the lowest power consumption at flow rate zero. Contrary, high specific speeds cause the highest power consumption at flow rate zero.

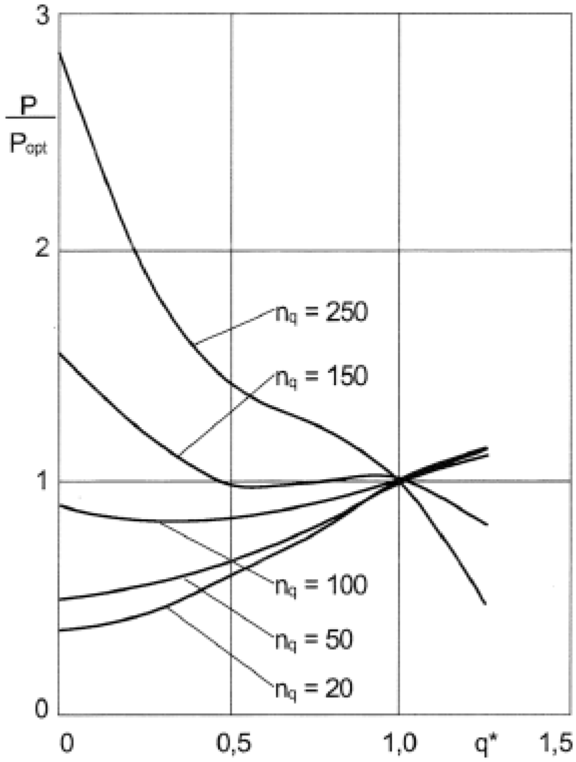


Figure 14: Power curves depending on nq (Jaberg, 2012, p. 403)

3.5.3.1 Fixed blade

3.5.3.1.1 Radial

Radial flow impellers are designed to be used for high head and low flow rate applications. The fluid leaves the impeller in perpendicular direction to the shaft. This type has lower specific speeds in the range from nq10 to nq50. (ANSI/HI 1.3, 2013, p. 12)

3.5.3.1.2 Semi axial (mixed)

This type is the transition between the radial and the axial impeller types. The semi axial hydraulics are from nq 50 (ns182,5) to nq 170 (620,5) (Gülich, 2013, p. 108).

Mixed flow impellers are often used because of economic reasons, to suit the needs of applications where these kinds of pumps are used, wet pit or deep well applications. Hence, the pump is built in smaller shape and no onion shaped, or so called pot design. The guide vane housing is directly connected to the column pipe.

The tangential speed of the impeller is limited to 25-30m/s due to cavitation reasons. Thus, a mixed flow impeller maximum head is 60m, hence for higher heads the pump must be designed in a multistage way. (Hellmann, 2011)

Adjustable blades are not possible with a mixed flow impeller; hence, pre-swirl regulation is used in this case. This is described in detail in Chapter 3.5.3.3.3.

3.5.3.1.3 Axial

As the flow rate increases and the head decreases, axial impellers are used for such applications. The fluid flow is in parallel to the shaft centerline. (ANSI/HI 1.3, 2013, p. 12) This impeller type starts from nq160 (ns586), hence, this type has the highest specific speed. The axial impeller is an impeller which can be designed with fixed blades or adjustable blades with manual or automatical adjustment. The difference and the functionality of the adjustable blades are described in Chapter 3.5.3.3.

Pumps with a high specific speed start operation with open valves – to avoid high power consumption at the zero-flow point and an overload of the motor. This is illustrated in Figure 14. Thus, bypass regulation or a frequency converter is often used to start the operation of the pump. Especially the suction conditions are crucial for axial impellers. If the suction design is not correct, the power consumption and the head at zero flow can increase substantially, which results in an overload of the motor. The importance of the suction conditions and the different possibilities are described in Chapter 3.5.5.

3.5.3.1.4 Closed / Semi-open / Open impeller

Radial impellers are mainly **closed** types. This means that a shroud is attached to both sides of the impeller vanes, these are the most common types. The high-pressure side is separated to the low-pressure side with narrow clearances at the so-called wear rings. (ANSI/HI 1.3, 2013, p. 13) This reduces leakage in the pump and increases efficiency.

The disadvantage of a closed impeller is the unequal pressure distribution on each shroud. (Hellmann, 2011)

The mixed impeller is either semi-open or closed. Up to heads of 15m and a clean fluid the mixed impeller can be designed **semi-open.** Hence, the front shroud is relinquished. (Hellmann, 2011, p. 160) To maintain efficiency comparable to the one of a closed impeller, the impeller should have narrow clearances between the open surface and the so-called front liner. (ANSI/HI 1.3, 2013, p. 14) This gap is set with the adjusting nut, which is described in the next Chapter 3.5.3.2. To reduce the gap losses, for higher heads a closed Impeller is recommended. (Hellmann, 2011, p. 160)

In case that the pumped fluid is soiled, a semi-open impeller is recommended. The free ball passage diameter is important for bigger containments, for non-clogging applications. (ANSI/HI 1.3, 2013, p. 14)

The axial impeller is only available in **open design**.
This means that this type has no front and back shroud, hence the axial force is the lowest of all types. (ANSI/HI 1.3, 2013, p. 14)

3.5.3.2 *Impeller gap adjustment*

To adjust the impeller axially and set the clearance to the desired value, an adjusting nut or a rigid adjustable coupling is used. Adjusting the Impeller clearance is especially important for mixed-open impellers.

The rigid adjustable coupling is described in Chapter 3.5.11.

The location of the adjusting nut depends on the location of the axial thrust bearing. If the axial thrust bearing is located inside the pump head, the adjusting nut is on top of the axial thrust bearing casing, see Figure 15. This adjusting nut is mainly a lock nut with a lock washer.

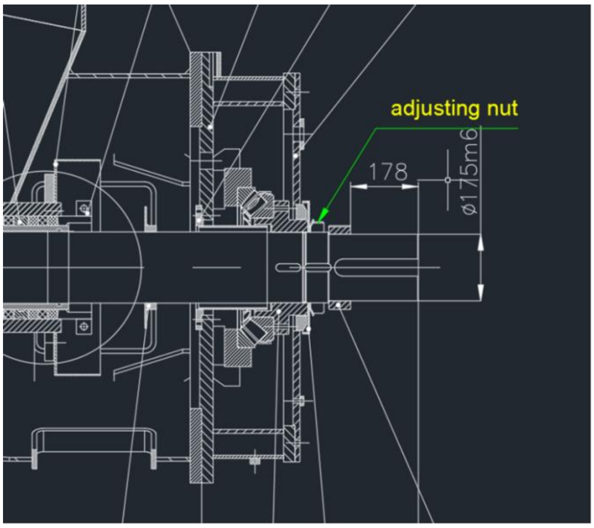


Figure 15: Adjusting nut (Andritz, internal source)

If the axial thrust bearing is integrated into the motor, the adjusting nut is located on top of the motor. This is illustrated in Figure 16. The motor has a hollow shaft and the pump shaft extends through the motor. The self-release coupling and the locking arm ensures that no reverse rotation occurs.

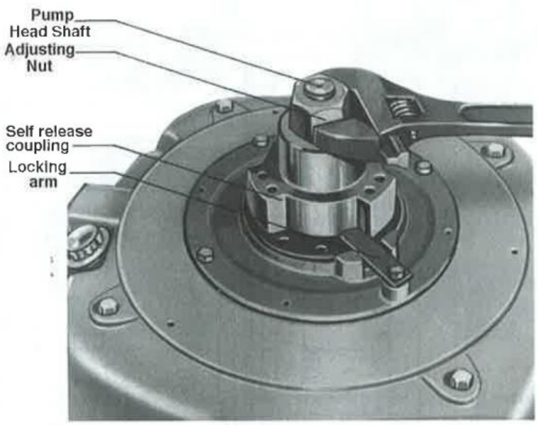


Figure 16: Hollow shaft through the motor (Bloch & Budris, 2010, p. 307)

3.5.3.3 *Change of the pump curve*

To change the characteristic pump curve, three ways are possible to achieve flexibility – by adjusting the blades, by means of a frequency converter to control the speed or the pre-swirl regulation. Adjusting the blades is described in Chapter 3.5.3.3.1, the speed control is described in Chapter 3.5.3.3.2 and the pre-swirl regulation in Chapter 3.5.3.3.3. Additionally hydraulic torque converter and spur or planetary gears are possible to use.

Flexibility is needed to adapt the duty point as of tidal fluctuations, changes between day/night or summer/winter operation.

To compare adjusting blades with the frequency converter, Diagram 2 illustrates both characteristic maps. It can be seen, that speed control is better for variations in head and blade adjustment for higher changes in the flow rate.

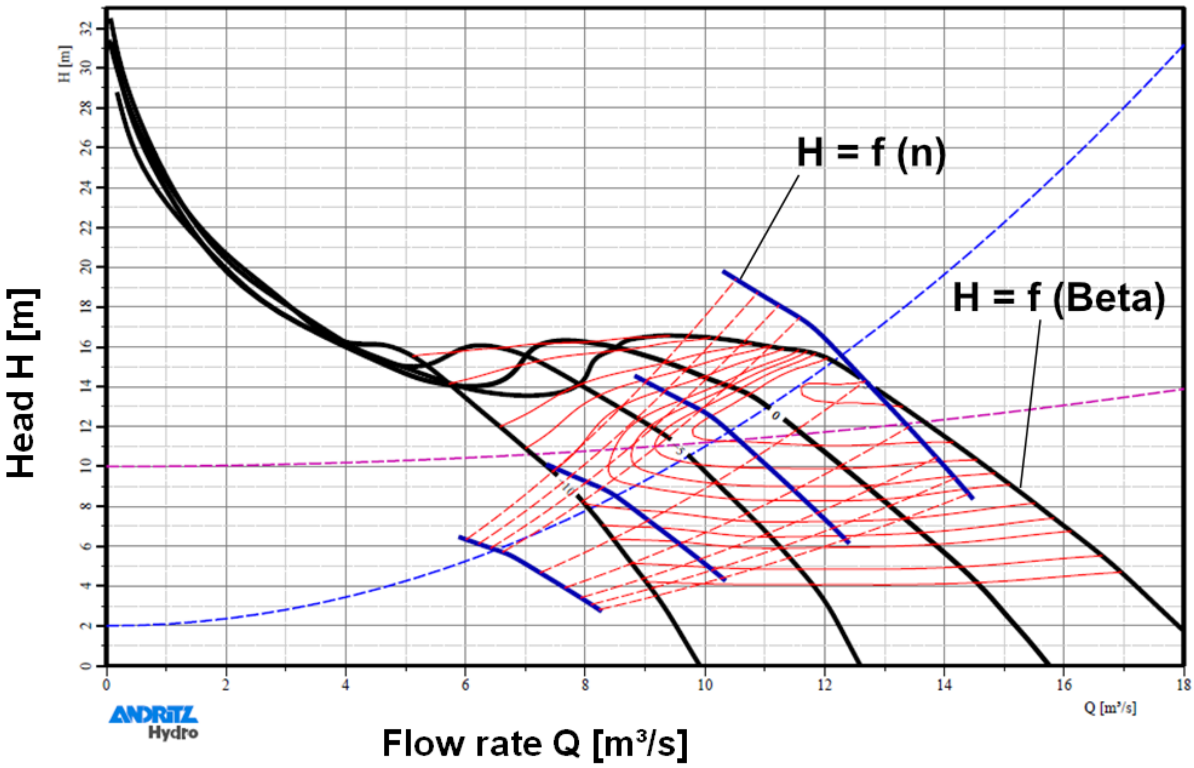


Diagram 2: Characteristic map for adjustable blade and speed control (Andritz, internal source) – (source modified)

In the following chapters the different flexibility options are described in detail.

3.5.3.3.1 Manually/Automatically adjustable blades

Diagram 3, illustrates a characteristic map of adjustable hydraulics. In fact, with adjustable hydraulics it is easy to react to flow rate changes. Each pump curve, which are shown in black, is a function of the blade angle. Due to the change of the angle, the NPSH and the efficiency change as well.

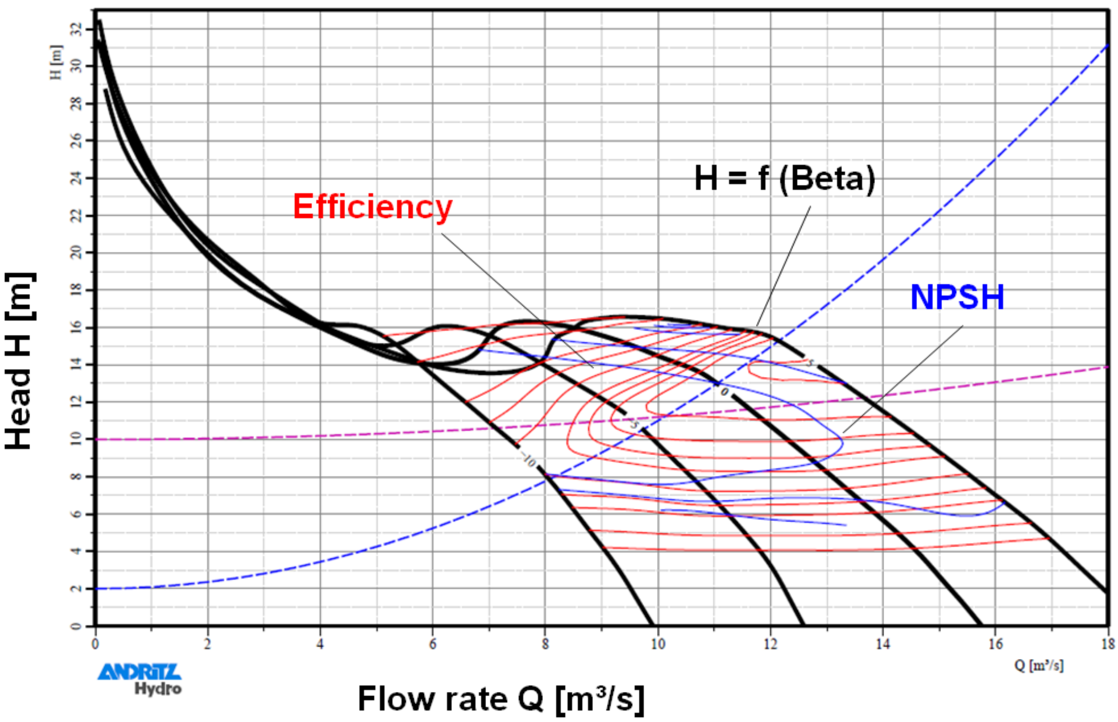


Diagram 3: Characteristic map of an adjustable blade hydraulic (Andritz, internal source)

The adjustable blades are either manually adjustable blades or during operation adjustable blades - so called automatically adjustable blades.

3.5.3.3.1.1 Manually adjustable blades

The manually adjustable blades are fixed with a pin after the desired angle is adjusted, illustrated in Figure 17. The angle can be adjusted, when the machine is not running – in case the duty point has changed. This is mainly used for standard pumps to create a characteristic map with one blade diameter.

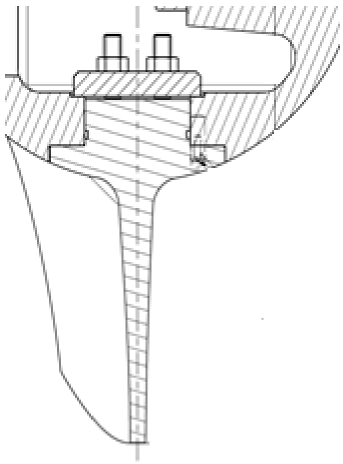


Figure 17: Separately mounted impeller adjustable blades (Andritz, internal source)

3.5.3.3.1.2 Automatically adjustable blades

This is to change the blade angle of the impeller during operation, to react to changing conditions. This option is often used in power plants for cooling water purposes because of changes in water flow rate and head, especially within day and night operation or tidal fluctuations.

The automatic adjusting device has clear advantages when there are substantial changes in delivery flow rate. The impeller blade angle varies up to 15° between minimum and maximum. Each blade position accompanies an own pump curve because of changed velocities. The biggest advantage is the possibility to react to changing conditions during operation. According to that the efficiency can be optimized or the duty point changed. (Andritz – Vertical Line Shaft Pump)

To enable the adjustment, the outer and the inner blade profiles, as well as the housing and the hub, are shaped in concentric bowls. This is hydraulically not the best shape and the gaps are bigger off the duty point, which reduces the efficiency. (Hellmann, 2011, p. 164)

The disadvantage is the higher design and manufacturing effort and according to that the higher costs. Another disadvantage is that adjustable blades are only possible for high specific speeds, thus for axial- and mixed flow impellers. The advantage is the gained flexibility and that there is no permanent energy input needed for the adjusting mechanism.

There are 2 ways to adjust the blades, with a hydraulic piston or with an axially adjustable rod within a hollow shaft.

The first option is altering the blades with a **hydraulic piston** inside the blade hub. The shaft has axial holes, where the oil is pumped into either hole, to adjust the servo cylinder axially. The adjusting lever of the blades is attached to the servo cylinder with linear guides. These linear guides convert the axial movement into a rotation of the blades by angular positioned slots. The piston stands still and is pressurized on both sides of the piston. Each part is described and illustrated in Figure 18.

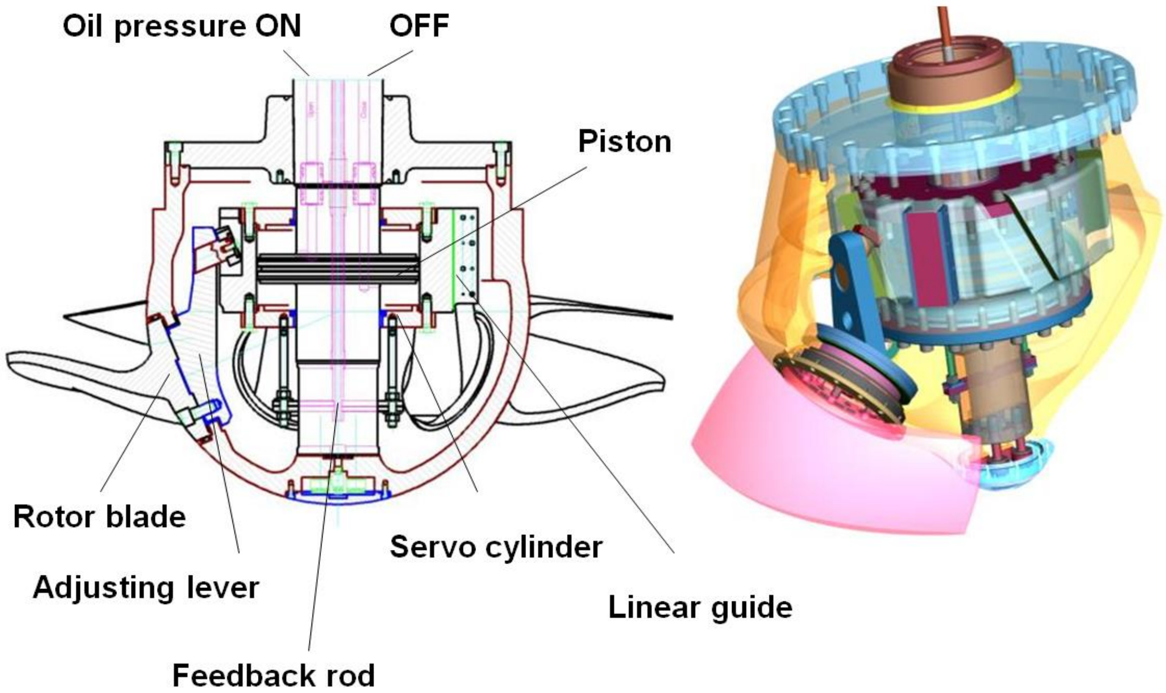


Figure 18: Impeller adjustment with piston (Andritz, internal source)

The second option is shown in the following Figure 19 and 20. The **axially adjustable rod** is incorporated inside a hollow shaft and is commonly powered by a mechanical thread drive or a hydraulic piston at the top of the pump. (Hellmann, 2011, p. 164)

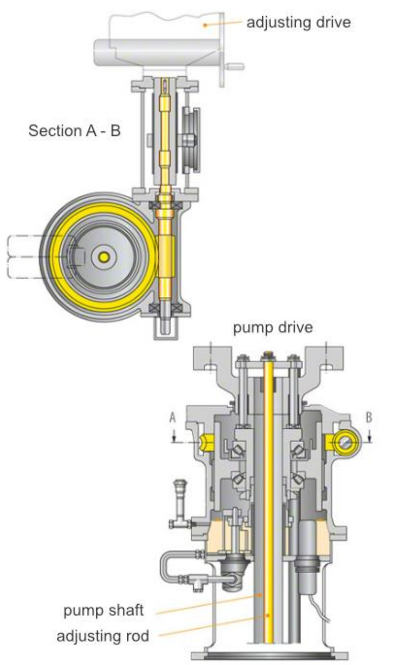


Figure 19: Drive options for adjusting rod (Hellmann, 2011, p. 164) – (source modified)

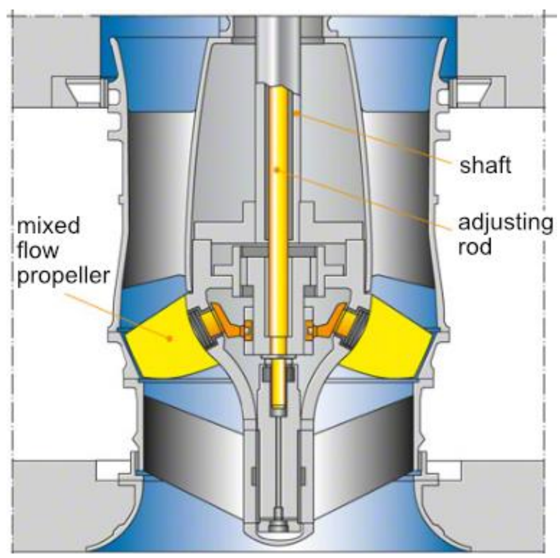


Figure 20: Adjustable blades (Hellmann, 2011, p. 164) – (source modified)

3.5.3.3.2 Speed variation

As already mentioned, the speed variation with a frequency converter is another option to react flexible and change the duty point according to changed conditions and to use the best efficiency point but the speed variation is, on the contrary to the adjustable blades, only used when the water level varies. In Diagram 4, there is a typical characteristic map for pump curves changed with speed variation illustrated. Each pump curve and the NPSH is a function of the speed.

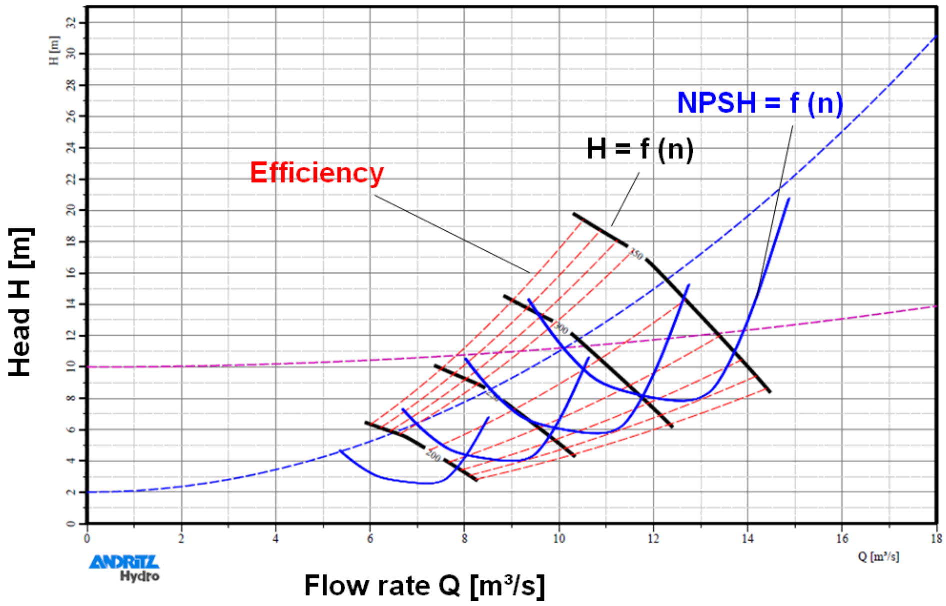


Diagram 4: Speed controlled pump curves (Andritz, internal source)

The speed variation is appropriate for pump curves with mainly dynamical portion. (Jaberg, 2012)

The advantage is that speed control can be used for all specific speeds. The disadvantage of the speed variation is the high cost, especially for high power consumption, the permanent power consumption for the speed controlling unit, the higher required space and the required air conditioning. (Andritz, internal source)

There are 3 options possible:

- Change of the pair of poles
- Change of the slip
- Change of frequency

The options of changing the pair of poles and changing the slip are not the preferred methods, because of too high changes in speed between the different numbers of pole-pairs and by changing the slip the losses a pretty high. (Hellmann, 2011, p. 61)

The most common option is the change of the frequency with the frequency converter. In Diagram 5, the torque depending on the shaft speed for different frequencies is illustrated. With change of the frequency the torque stays the same.

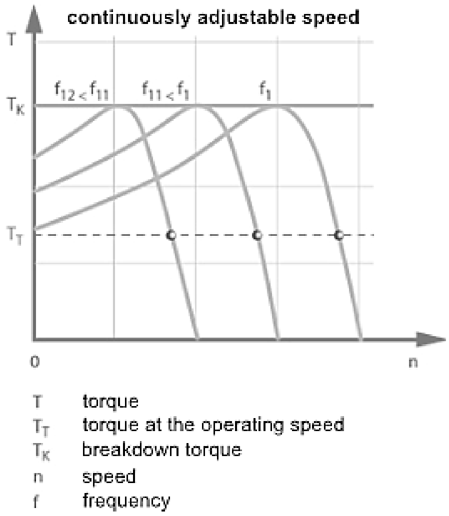


Diagram 5: Speed variation options (Hellmann, 2011, p. 61) – (source modified)

3.5.3.3.3 Pre-swirl regulation

A pre-swirl regulation enables a variation of the head. This is realized with a guide vane with adjustable blades in front of the impeller. This is used for all pumps, but high specific speed pumps have the highest effect. The pre-swirl regulations result in lower efficiency, higher cavitation-sensitivity and a higher blade load. (Jaberg, 2012, p. 428) The design of a pre-swirl regulation is shown below in Figure 21.

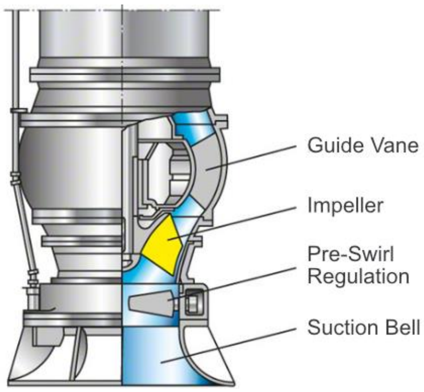


Figure 21: Pre-swirl regulation (Hellmann, 2011, p. 78) – (source modified)

3.5.3.4 Impeller assembly

The impeller is either with a feather key or with tapered collets assembled.

The tapered collets are used for lower cost VLSP. (Bloch & Budris, 2010, p. 357) But the impeller is not clear axially positioned. In Figure 22, the mounting with a tapered collet is illustrated.

The common assembly of an impeller is with a feather key and a split thrust ring or a shaft shoulder. This is due to longer maintenance free operation (Bloch & Budris, 2010, p. 357). A keyed impeller is illustrated below in Figure 23.

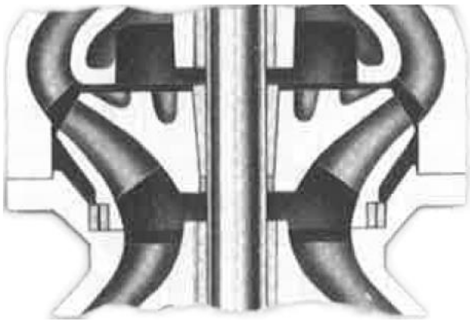


Figure 22: Tapered collet (Bloch & Budris, 2010, p. 357) – (source modified)

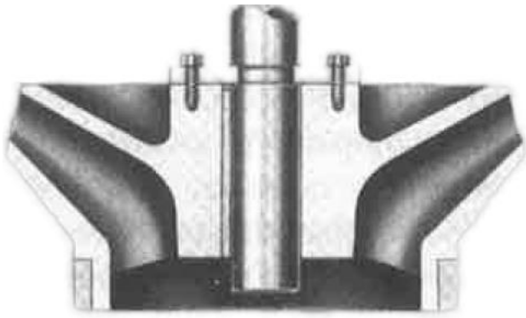


Figure 23: Keyed impeller (Bloch & Budris, 2010, p. 357) – (source modified)

3.5.4 Single- and multistage design

The VLSP is designed as a single stage as well as a multistage pump. The criterion if a multistage pump is used are the NPSH and the total head. The NPSH limits the maximum head per stage. If the $NPSH_{req}$ is higher than the $NSPH_{available}$, a multistage pump with a smaller impeller diameter must be used, hence, with a lower $NPSH_{req}$.

The total head of a multistage pump is the sum of the head per stage, while the flow remains the same.

3.5.5 Suction

To ensure a good operation of the pump the inflow must be consistent and free of any interference, hence, the suction design is a very important issue for vertically suspended pumps.

The energy transfer of specific low speed pumps is conducted by an energy transfer according to the centrifugal effect. For specific high speed pumps this is done by current redirection at the blades. Therefore, especially axial vertically suspended pumps react sensitive to bad suction conditions. (Hellmann, 2011, p. 329)

Following 3 criteria are relevant for a good suction inflow:

- Swirl free suction
- Consistent velocity distribution
- Vortex free (Hellmann, 2011, p. 329)

The swirl at the entry is either a counter- or an identical swirl. Both are harmful for the efficiency, increase the head and the power consumption. To have a **swirl free suction** anti-vortex ribs (Figure 28 and 29), comb flow straightener, floor cones or floor splitter blades at the bottom of the entry are used (Figure 25). (Hellmann, 2011, p. 329)

The initial pump impeller design assumes a **consistent velocity distribution**. This means that the actual distribution is between a rectangular velocity distribution and a fully developed turbulent current. With different interferences this distribution is disturbed, for example due to depression at built-in installations, any flow redirection (elbows) or flow separation. The more the distribution is disturbed, the simulated and measured pump data will not occur. Even if the distribution is not rotational-symmetric, vibrations will arise. (Hellmann, 2011, p. 330) A consistent velocity distribution is achieved with a nozzle, which accelerates the fluid and even-distributed resistances in the current area. This is accomplished in a vertical pump with an acceleration suction elbow (Chapter 3.5.5.2) or a suction bell (Chapter 3.5.5.3). Regardless, the fluid redirection is an issue for a vertical pump and the disturbance by that get minimized with the sump design.

Vortices sprout in shear flows and positions with a high velocity gradient. These are classified in sub-surface- and free surface vortices. (Hellmann, 2011, p. 331)

The strength of the free surface vortices varies from a surface swirl to a full air core. The sub-surface vortices are classified in swirl, dye core or air core (bubbles). (Illustrated in Figure 24)

Vortices which last into the pump intake disturb the performance and the operation of the pump. These are full core vortices terminated as a free surface- or a sub-surface vortex. A swirl disturbs the performance, the head and the flow rate of a pump. In addition to that, the vortices are mostly transient and result with varying performance in vibration, sound and in mechanic stress at the impeller. Depending on the rotational velocity, even a vapour or gas core may occur. (Hellmann, 2011, p. 331)

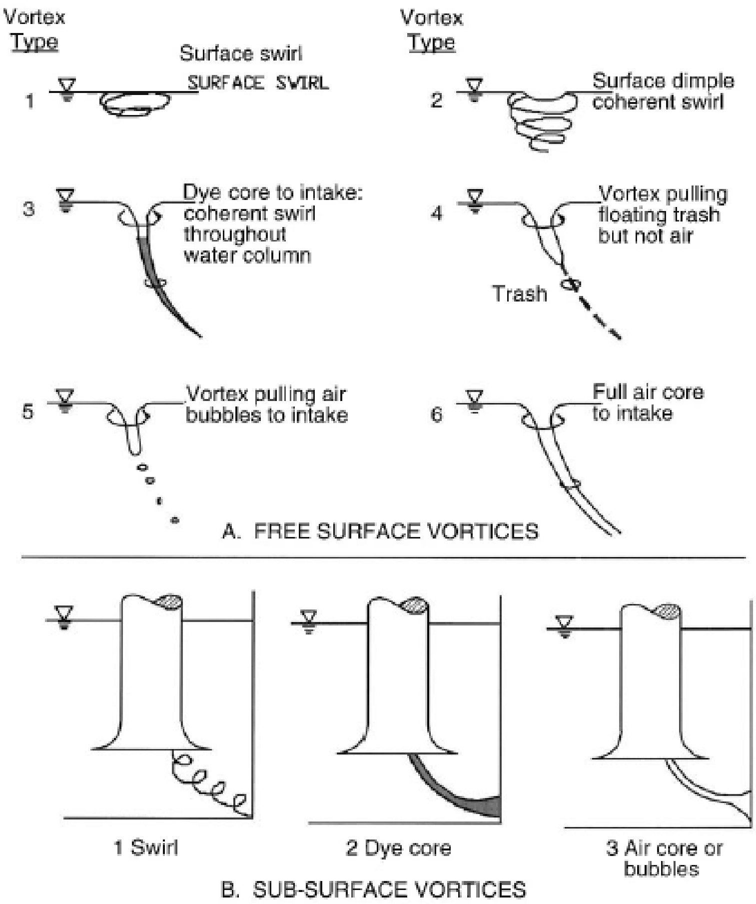


Figure 24: Classification of free surface and sub-surface vortices (Hydraulic Institute – 9.8, 1998)

To avoid full air core free surface vortices and sub-surface vortices:

- the submergence S is increased (see Figure 26)
- improvement of the inflow to avoid rotation and velocity gradients
- covering the vortex vulnerable area with a ceiling
- installation of swirl avoiding parts such as splitters, rips at the entry, with floor cones at the bottom (see Figure 25) (Hellmann, 2011, p. 331)

With installations inside the sump, covering the vulnerable area and/or a suction elbow the required submergence can be reduced. (Hellmann, 2011, p. 331)

The suction of air swirls (Free surface vortices) depends on the **submergence** as already mentioned. In Diagram 6, the principle of the minimum submergence is illustrated. The minimum submergence in area I ensures that the bottom bearing is already lubricated with water during start-operation. This depends mainly on the pump design. In area II the minimum submergence secures the pump against the air core vortices. In this area, the minimum submergence depends on the fluid velocity at the suction. At the area III the $NPSH_{req}$ is decisive, to ensure a cavitation free operation. (Sterling SIHI, 2000, p. 113)

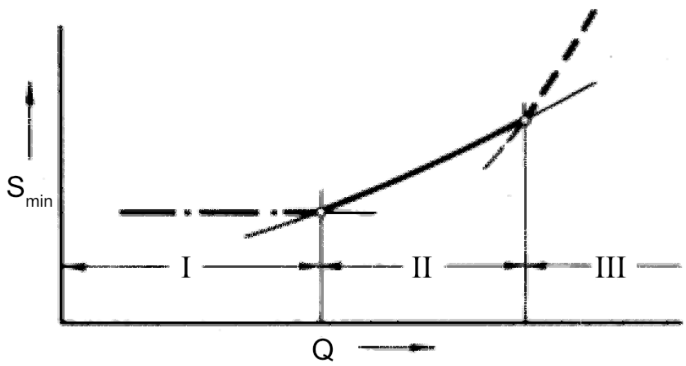


Diagram 6: Min. submergence / flow rate (Sterling SIHI, 2000, p. 113)

The Hydraulic Institute has already carried out some investigations about minimum submergence. To define the diameter of the suction bell inlet d_E, the Hydraulic Institute has experimentally examined that, depending of the flow rate and a recommended speed at the inlet of the suction bell, a certain bell inlet diameter is recommended. This is illustrated in Diagram 7 and 8.

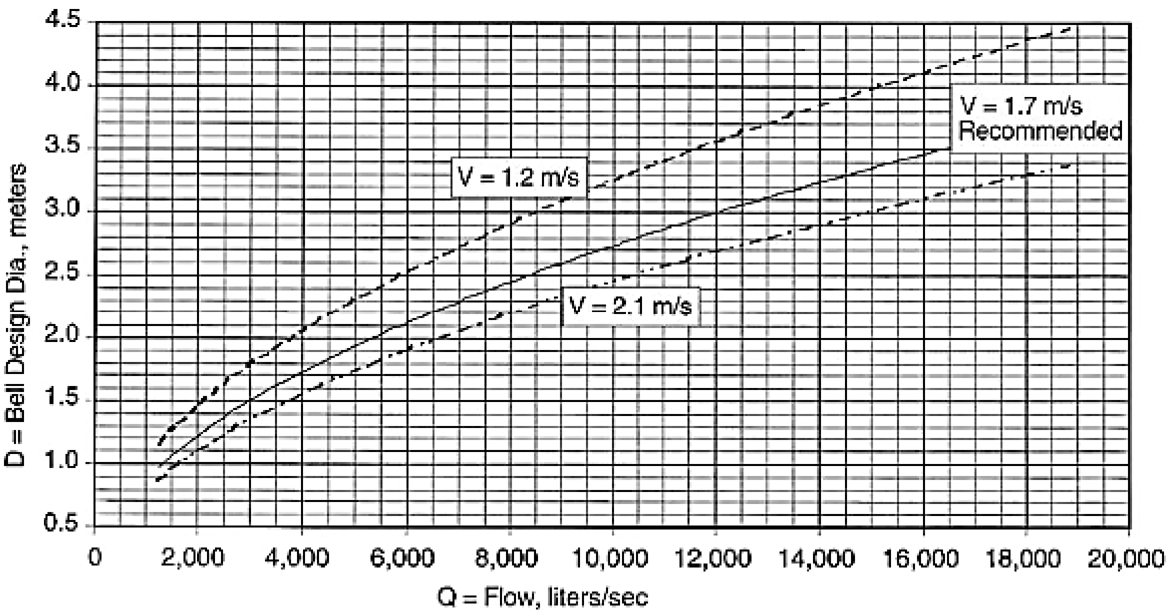


Diagram 7: Recommended suction bell inlet diameter 1 (Hydraulic Institute – 9.8, 1998, p. 30)

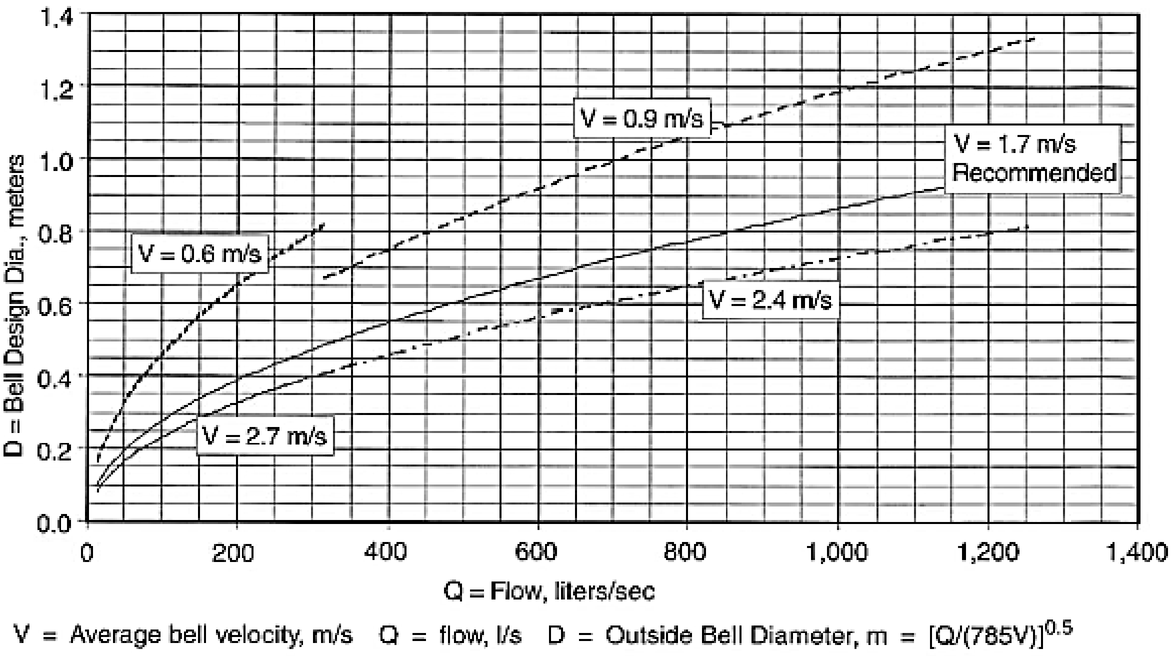


Diagram 8: Recommended suction bell inlet diameter 2 (Hydraulic Institute – 9.8, 1998, p. 30)

The calculation of the minimum submergence according to HI is executed with Equation 1 to 3 below. This systematic is also illustrated with Diagram 9.

The v_s is calculated with the Equation 1 below:

$$v_S = \frac{Q}{\frac{{d_E}^2 \times \pi}{4}}$$

Equation 1: Fluid velocity

v_s	Fluid velocity	[m/s]
Q	Flow rate at the optimum	[m^3/s]
d_E	Outer diameter at the suction bell inlet	[m]

To calculate the minimum submergence the following formula according to HI is used:

$$S_{min} = d_E \times [1 + 2{,}3 \times F_D]$$

Equation 2: Minimum submergence (Hydraulic Institute – 9.8, 1998, p. 32)

S_{min}	Minimum submergence	[m]
F_D	Froude number	[-]
g	Gravitational constant	[m/s^2]

The Froude number is calculated according Equation 3.

$$F_D = \frac{v_s}{(g \times d_E)^{0{,}5}}$$

Equation 3: Froude number (Hydraulic Institute – 9.8, 1998, p. 32)

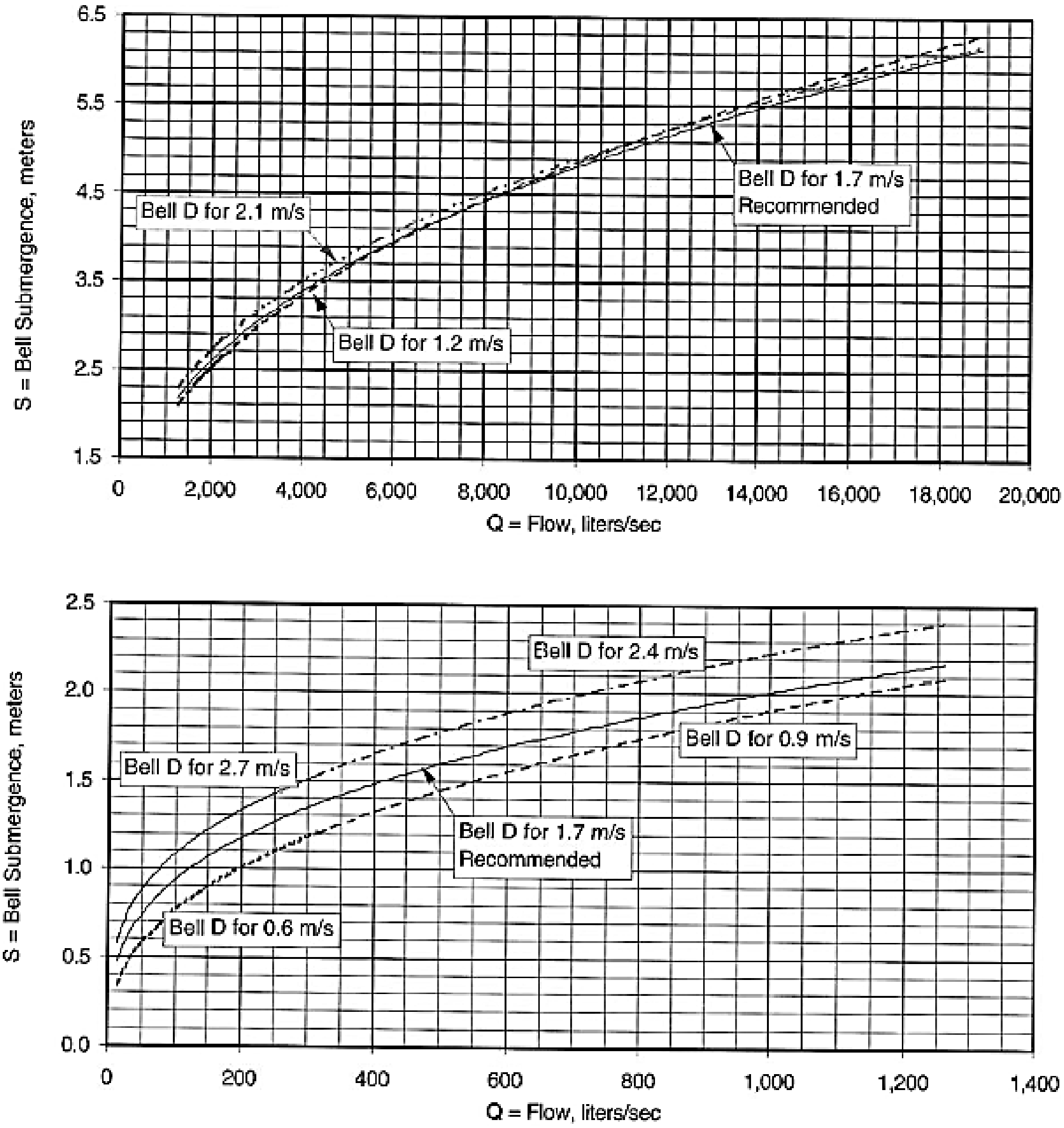


Diagram 9: Minimum submergence to minimize free surface vortices (Hydraulic Institute – 9.8, 1998, p. 33)

The sub surface vortices are not avoided with the proper submergence. The determinant factor for sub surface vortices is the water volume or the clearance below the suction bell. If the clearance is too high, the water volume below will excite by the impeller rotation and start to swirl. Sub surface vortices are only avoided with a proper sump design and installations below the suction bell, such as a floor cone, splitter blades and others, which are illustrated in Figure 25.

3.5.5.1 Sump

The design of the pump sump is very important. It is a chamber to accommodate a swirl free and consistent suction from all sides and a proper design will prevent sub surface vortices. A wrong designed or a suction without a chamber result in cavitation, vibration, less efficiency and higher zero-flow head (Hellmann, 2011, p. 79).

The chamber is designed with different installations on the floor or wall. The different possibilities are illustrated in Figure 25. All these installations can be used in combination or separately. These installations prevent rotating of the water volume below the suction bell and must be used individually according to the site conditions. When installations such as corner fillets are placed in the chamber, the recommended submergence can be reduced. The ground view of the chamber is either rectangular or round.

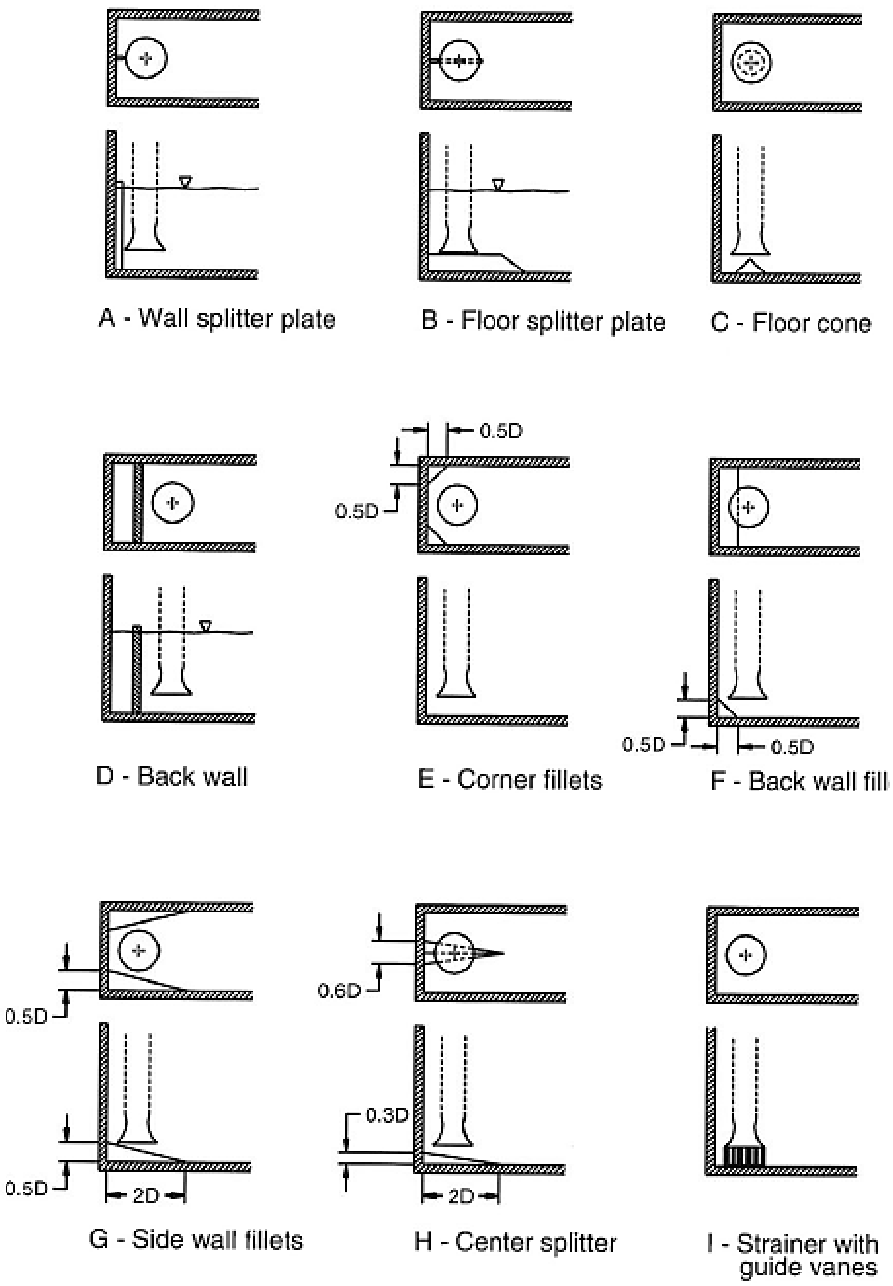


Figure 25: Installations to prevent sub surface vortices (Hydraulic Institute – 9.8, 1998, p. 51)

For the chamber itself, the Hydraulic Institute has investigated with model witness tests the best dimension for a sump design. The main dimensions are shown in Figure 26. An optional “travelling through flow screen” prevents large compartments entering the sump chamber. The chamber width is defined with twice the suction bell diameter but this can be adjusted by case, if the velocity extends the mentioned 0,5 m/s. S as the minimum submergence is, as already mentioned, calculated with Equation 2. The distance B from the back wall is recommended with 0,75 times the suction bell mouth diameter. The clearance which is also important for sub surface vortices is recommended with 0,3 to 0,5 times the suction bell diameter.

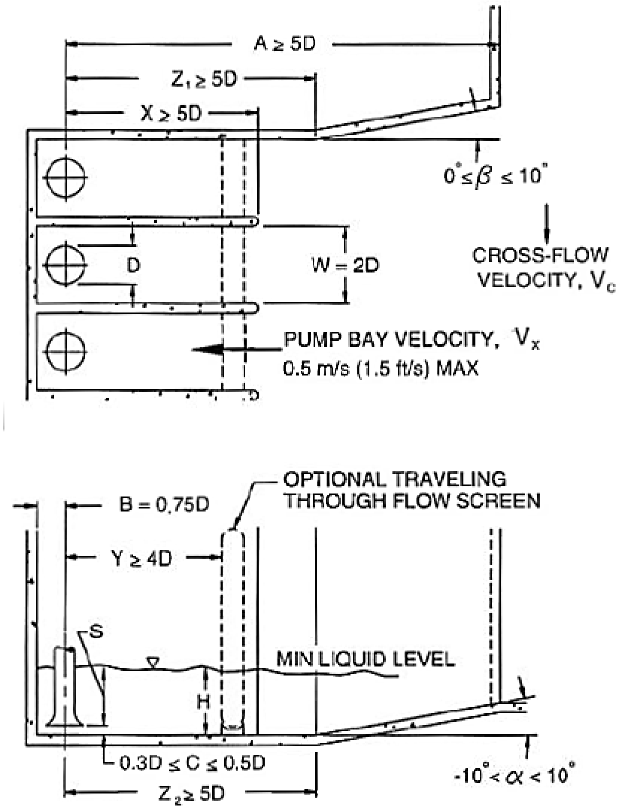


Figure 26: Sump dimensions (Hydraulic Institute – 9.8, 1998, p. 3)

3.5.5.2 Suction elbow

The suction elbow is designed as an acceleration elbow to have a consistent velocity at the inflow and to reduce disturbances. The best inflow condition is when inside the elbow the fluid velocity is increased by 2-4 times of the pipe current. Such acceleration elbows are also realized with transition-profiles starting with a rectangular and ending with a circle. (Hellmann, 2011, p. 332) This is shown in Figure 27.

Usually a formed metal sheet suction elbow is used (Figure 27) but for bigger VLSP’s a concrete suction hose, or so called Kaplan suction elbow, is applied as well (Figure 33).

These designs are used when the suction chamber width is lower than twice the suction bell diameter, as recommended in Figure 26.

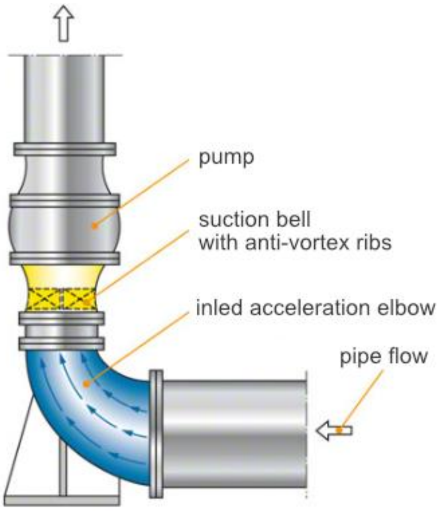


Figure 27: Inflow acceleration elbow (Hellmann, 2011, p. 330) – (source modified)

3.5.5.3 Suction bell

The suction bell is designed as a nozzle with the purpose to accelerate the fluid. The acceleration of the fluid reduces irregularities of the velocity distribution. Especially the velocity distribution is very important for good inflow conditions. The high specific speed hydraulic is susceptible for irregularities of the inflow. (Hellmann, 2011, p. 78) The loss coefficient of the inflow through a suction bell is ζ=0,05. (KSB, 2005, p. 24)

The suction bell is either casted or fabricated.

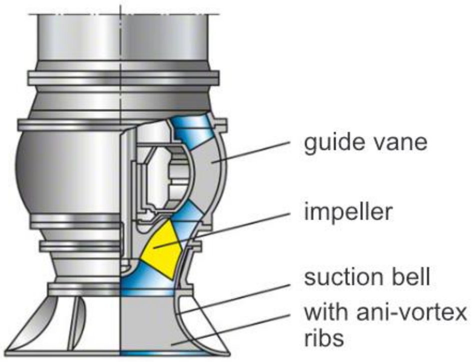


Figure 28: Suction bell with anti-vortex ribs (Hellmann, 2011, p. 78) – (source modified)

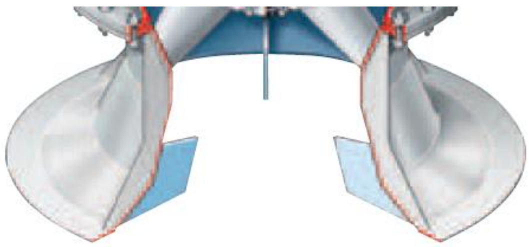


Figure 29: Anti-vortex vanes (Flowserve VCT, 2008, p. 4)

If the inflow has a swirl, anti-vortex ribs are needed to reduce the swirl and improve the inflow conditions. The anti-vortex ribs work as a flow straightener. The anti-vortex ribs design is as a spider (Figure 28) or only as vanes inside the suction bell (Figure 29).

3.5.5.4 *Bell bearing bushings*

Typically the radial bearings are located between the motor and the impeller. The impeller deflection is a critical factor for the shaft design. To reduce the impeller deflection, a bottom bearing (bell bearing) can be additionally used. This is illustrated in Figure 30. Additionally, the support of the bearing works as an anti-vortex rib. Often additionally a sand collar is installed to prevent grit entering the bell bearing.

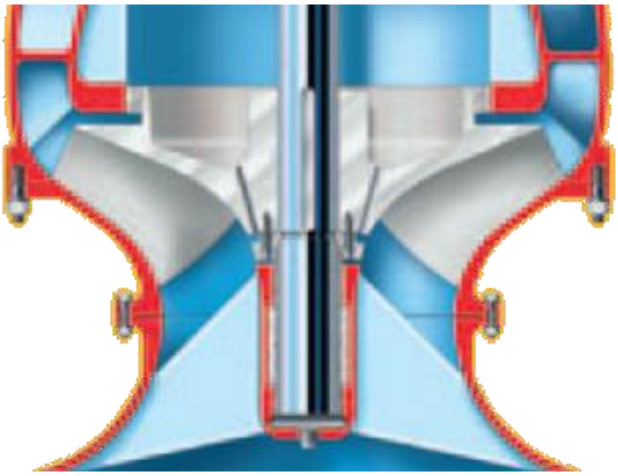


Figure 30: Bell bearing bushings (Flowserve VCT, 2008, p. 5)

Another advantage of such design is, due to reduced flow area, the higher velocity of the fluid and a more stable pump curve. The in-stationary vortex swirls are carried on with the higher velocity but to use this effect the inflow area must be significantly reduced.

3.5.5.5 *Strainer (basket)*

Strainers are optionally mounted at the suction bell to prevent bigger compartments entering the suction bell. Cone strainers are used for deep well applications. Such a basket is illustrated in Figure 31.

Figure 31: Basket (Flowserve corporation – VTP, 2014, p. 5)

3.5.6 Pressure casing

The pressure casing incorporates the guide vanes and directs the fluid to the column pipe. The purpose of the guide vanes is to transfer the speed of the fluid into pressure. The pressure casing is mostly casted but is also available fabricated.

3.5.7 Column pipe (Riser pipe)

The column pipe conveys the pumped fluid to the discharge elbow, connects the guide vane housing with the discharge elbow and supports the integrated slide bearing which transmits the torque to the impeller. Bearing spiders are located at the column pipe connection (Bloch & Budris, 2010, p. 361). The length of each column pipe section is equal to the length of each shaft and the shaft sections are connected with rigid shaft couplings. Details about the shaft are described in Chapter 3.5.8.

The column pipe is available in different designs and constructions. The column pipe is either a steel- or a concrete column pipe, this is illustrated in Figure 32 and 33. The steel column pipe design is more usual. The concrete column pipe design is optionally used for bigger sizes. When a pull-out design is required a concrete column pipe is a good solution. The concrete option limits the allowable head to about 40m, hence, for high head applications the steel column pipe is recommended.

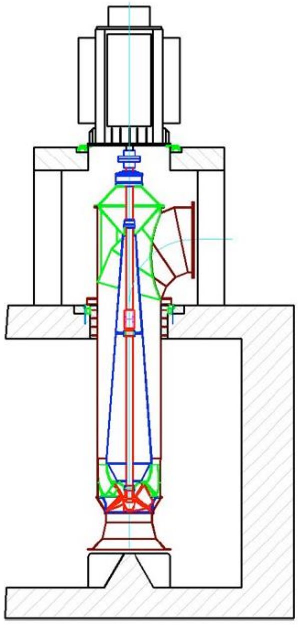


Figure 32: Steel column pipe (Andritz – Vertikale Rohrgehäusepumpen)

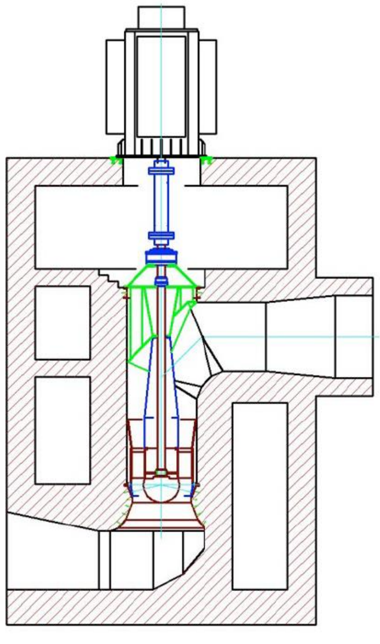


Figure 33: Concrete column pipe (Andritz – Vertikale Rohrgehäusepumpen)

The column pipe is either flanged (Figure 34) or threaded (Figure 35). The flanged version is the common one. The threaded version is used for applications in wells, where small diameters are needed.
According to literature the threaded version is cheaper than the flanged one. (Bloch & Budris, 2010, p. 354)

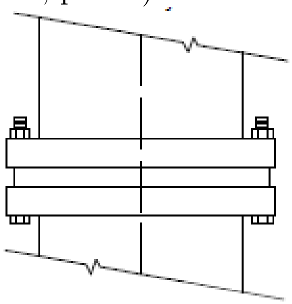


Figure 34: Column pipe flanged (Flowserve corporation – VTP, 2014)

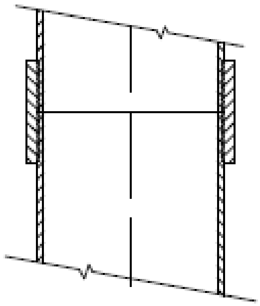


Figure 35: Column pipe threaded (Flowserve corporation – VTP, 2014)

In column pipes, diffusors are used to change the velocity inside the column pipe to the restricted velocity. This is either done with an outer riser pipe diffusor (Figure 36) or an inner riser pipe diffusor, or so called, hub diffusor (Figure 37). Both diffusors are flow separation jeopardised. The angle as a function of the length of the diffusor is determinant. A 3,5° to 4° half opening angle is recommended.

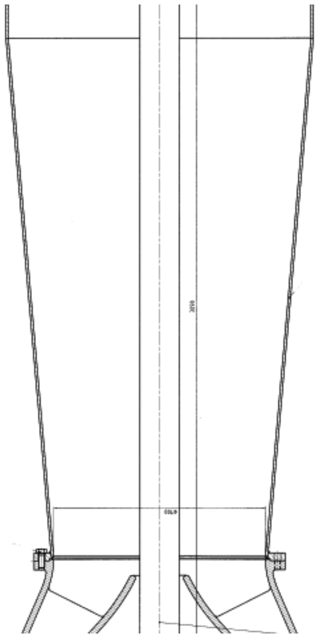


Figure 36: Outer riser pipe diffusor (Andritz, internal source)

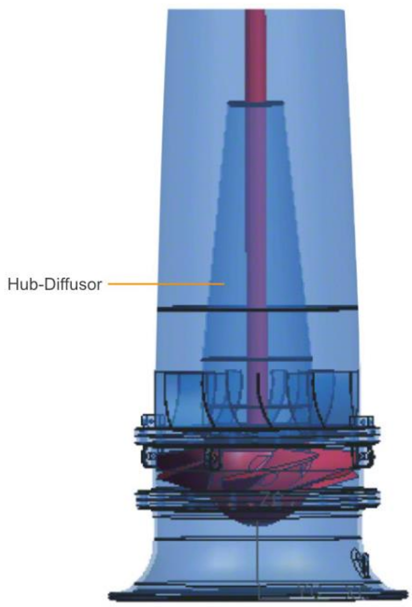


Figure 37: Inner riser pipe diffusor (Hellmann, 2011, p. 186) – (source modified)

The crucial factor for the diameter of the column is the velocity inside the pipe. To manufacture a cheaper pump, the velocity must be high, to lower the diameter of the column pipe. The minimum velocity speed is set to 2m/s. The maximum velocity is set to 4,5m/s. Otherwise the losses in the elbow and the column pipes are too high. The column pipe is not the defining factor according to head loss. The discharge elbow is more critical according to head loss. The losses in elbows are examined in detail in Chapter 3.5.10. Furthermore, the customer uses concrete piping after the pump and therefore, the velocity is restricted to 1-2m/s, hence, after the pump a diffusor must be installed to reduce the speed, according to customer requirements.

The efficiency loss depending on the head and the flow velocity inside the column pipe is illustrated in Diagram 10. For this the loss coefficient inside the elbow and the column pipe was taken into consideration with ζ=0,6. This picture illustrates clearly that if the pump head is just 10m, the speed inside the column pipe is critical because with a speed of 3,8m/s the efficiency loss is already 4%. On the contrary, a pump with a head of 55m can use the range between 2 - 4,5m/s and even with 4,5m/s the loss is only 1%. By using a higher speed the column pipe is up to 20% smaller in diameter. Thus, the column pipe speed depends on the head and the acceptable efficiency loss.

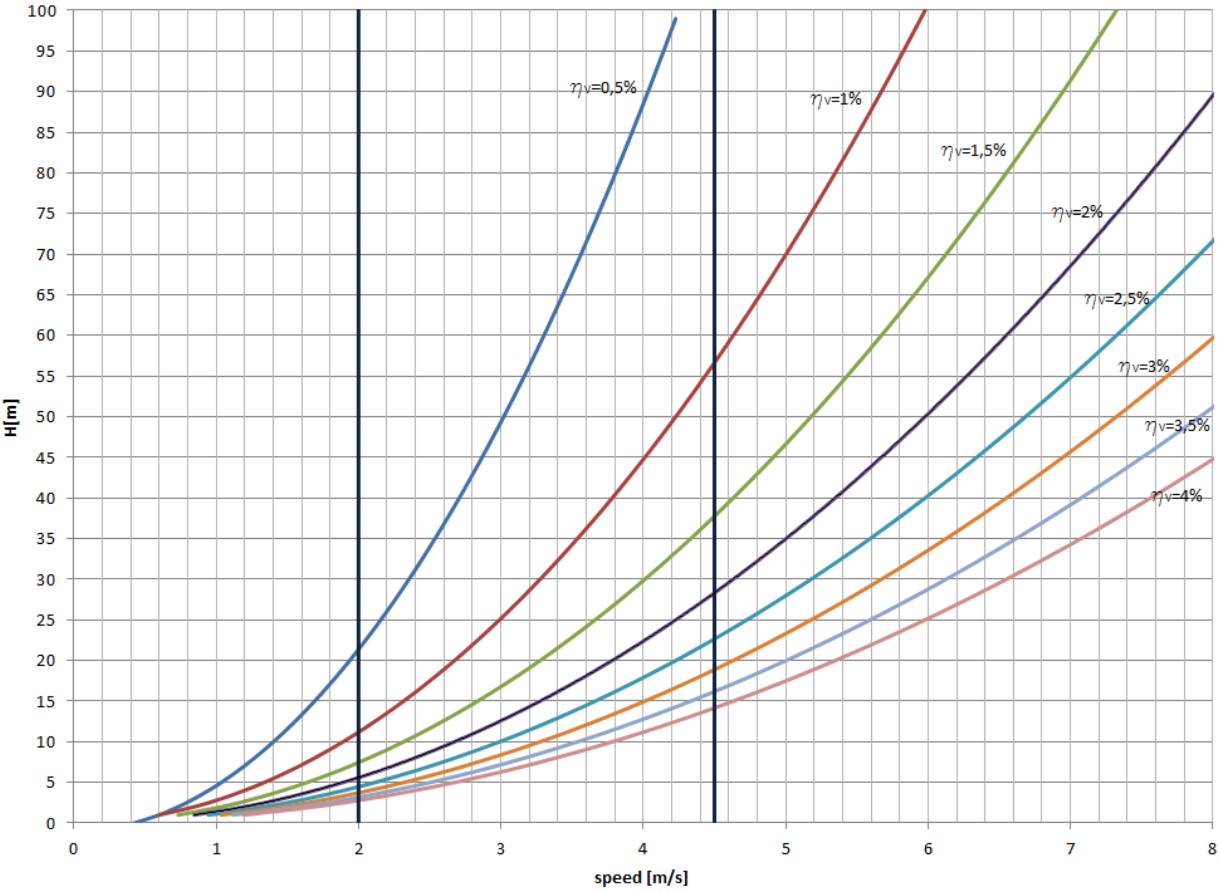


Diagram 10: Headloss / speed inside the column pipe

The head loss in a pipe is calculated according to Equation 4.

$$h_{v\_pipe} = \lambda \frac{L}{D} \times \frac{c^2}{2 \times g}$$

Equation 4: Elbow head loss

h_{v_pipe}	Pipe head loss	[m]
λ	Pipe loss coefficient	[-]
L	Length of the pipe	[m]
D	Inner diameter of the pipe	[m]
c	Velocity inside the pipe	[m/s]
g	Gravitational constant	[m/s²]

The loss coefficient λ is a function of the Reynolds-number determined with the Moody diagram. The Moody diagram is illustrated in Diagram 11. The Reynolds number determines if a current is laminar or turbulent. Furthermore, which curve to choose in the Moody diagram depends on the pipe inner surface structure, if it is smooth or rough. If it is rough, the roughness extends into the turbulent layer and disturbs this. The Reynolds number Re is calculated according to Equation 5. The kinematic viscosity ν is a function of the fluid type, -temperature and -pressure.

$$Re = \frac{c \times D}{\nu}$$

Equation 5: Reynolds number

Re	Reynolds number	[-]
c	Velocity	[m/s]
D	Inner diameter of the pipe	[m]
ν	Kinematic viscosity	[m²/s]

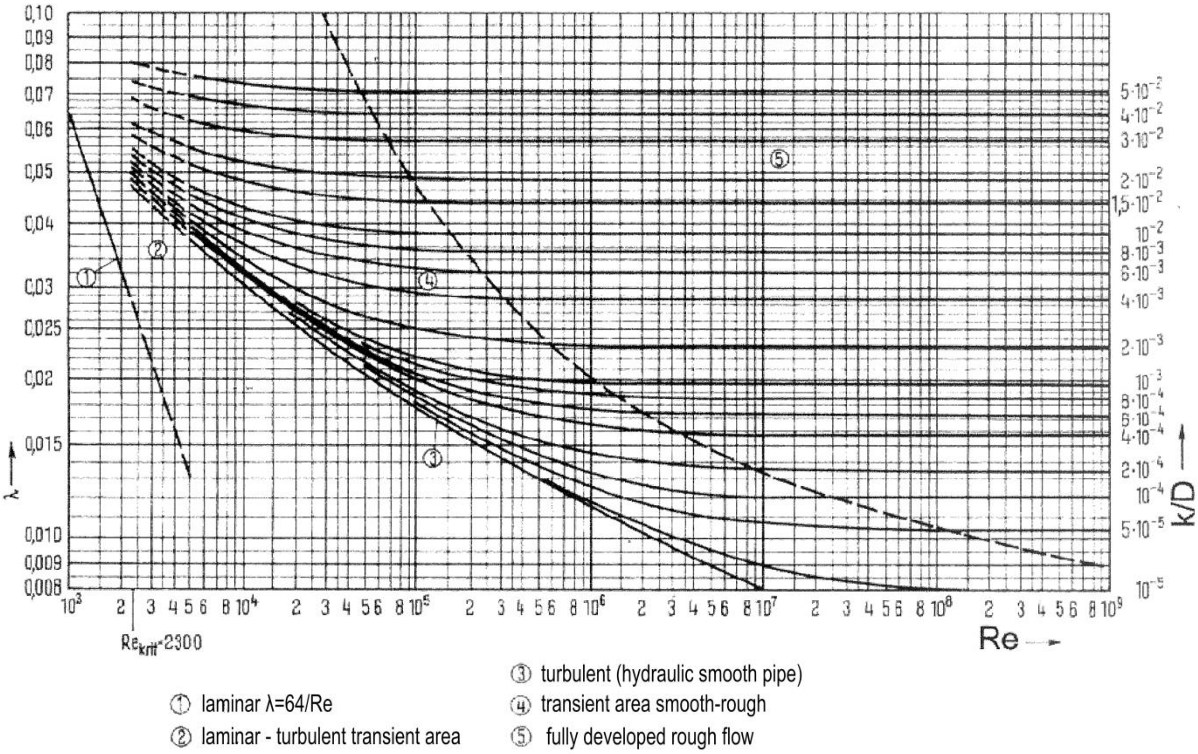


Diagram 11: Moody diagram (Brenn & Meile, 2013)

3.5.8 Shaft

The pump shaft transmits the torque from the motor to the impeller. Each shaft length is equal to the column pipe length. The shaft sections are connected with rigid shaft couplings. Rigid shaft coupling requires precise shaft alignment. If so used, misalignment will result in vibration, wear and less mechanical seal life. (Bloch & Budris, 2010, p. 306)

Rigid shaft couplings are either threaded, flanged or a sleeve coupling. The sleeve coupling is either split (clamp coupling) or not (muff coupling). The advantage of the muff coupling is cheaper manufacturing but for service reasons the split sleeve coupling is the better solution.

A threaded coupling requires high axial thrust to align the shafts properly. If the impeller is hydraulically balanced the shaft coupling must be a sleeve coupling to avoid additional unappreciated bearing loads resulting from misalignment. (Bloch & Budris, 2010, p. 362)

The critical speed of a shaft depends on the shaft diameter, the spacing between the bearings and the axial thrust. The critical speed increases with the shaft diameter and the axial thrust, hence, with the tension. The critical speed decreases with increasing bearing spacing. A shaft has more than one critical speed, regardless, the pump should operate at least 15% below the first critical speed to ensure a sufficient operation. (Bloch & Budris, 2010, p. 362)

Deep well applications usually design the bearing spacing with 3m but other applications should not extend the suggested bearing spacing's according to API610. Diagram 12 illustrates the maximum spacing (Y) as a function of the shaft diameter (X) and the curves of rotational speed in rpm. The values in brackets are in inches.

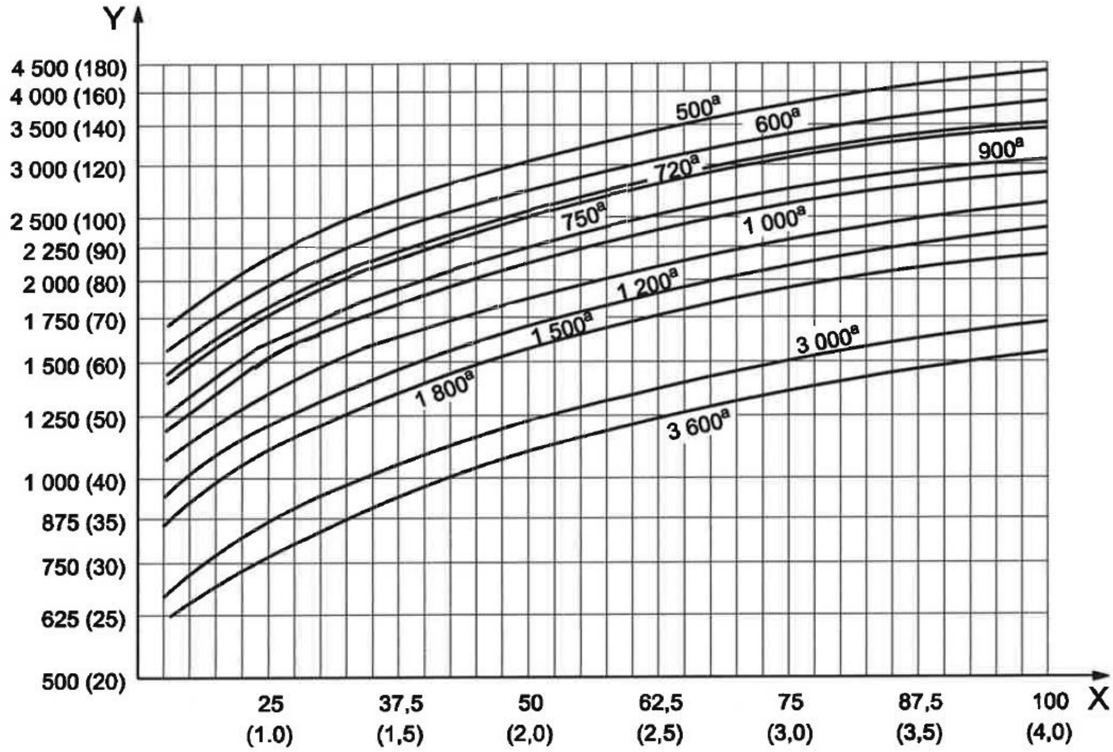


Diagram 12: Maximum bearing spacing (American Petroleum Institute, 2010, p. 95)

3.5.9 Discharge head

The discharge head accommodates the shaft sealing, the motor coupling, the discharge elbow, supports the motor, and depending on the design the axial thrust bearing. The discharge head is connected with screws at the foundation plate and deviates the force to the ground. In Figure 38, a typical discharge head with all the integrated components is illustrated. The steel foundation plate can also be integrated to the discharge head as illustrated in Figure 38. A separate foundation plate allows levelling of the pump discharge head without permanently anchoring it. The pump is removed without disturbing the foundation. (Flowserve corporation – VTP, 2014, p. 5)

The discharge head is either casted, which is used for smaller units, or welded. The motor is either directly screwed to the discharge head, or for bigger motors, a separate motor stand is also possible.

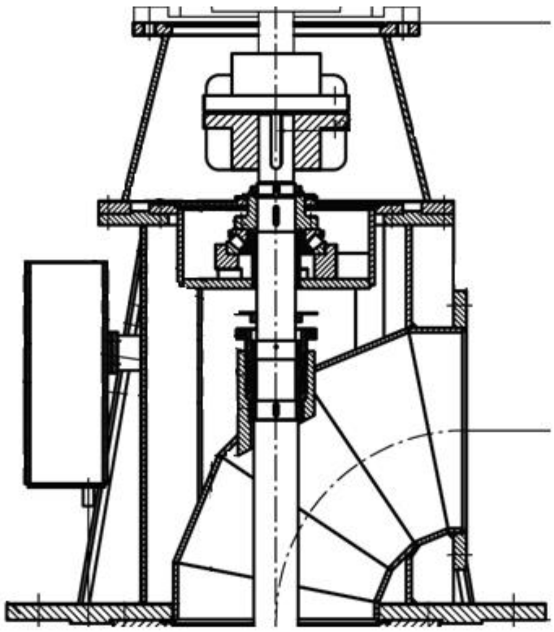


Figure 38: Discharge head (Andritz, internal source)

The discharge head must be designed stiff to increase the natural frequency and to eliminate reed resonance. The reed resonance arises when the natural frequency of the pump is close to the operation frequency. Even when all rotating parts are well balanced, excessive vibration may occur. (Bloch & Budris, 2010, p. 354) The discharge head may do its part to increase the natural frequency of the pump assembly. When the natural frequency and the running speed coincide, the vibration amplitudes are at its maximum and may damage parts of the pump.

3.5.10 Discharge elbow

The discharge elbow redirects the fluid flow. This redirection causes losses and disturbances, as already mentioned in Chapter 3.5.5. Since the efficiency of the pump is not only the hydraulic efficiency (bowl efficiency) - in this pump the efficiency counts until the end of the discharge elbow - the losses inside the elbow jeopardize the pump efficiency.

The design of the discharge elbow is according to flow-optimised and cost effectiveness aspects chosen.

There are some aspects which should be taken into consideration. The following factors are crucial:

- Curvature radius of the elbow
- Smooth or segmented elbow
- Pipe smoothness

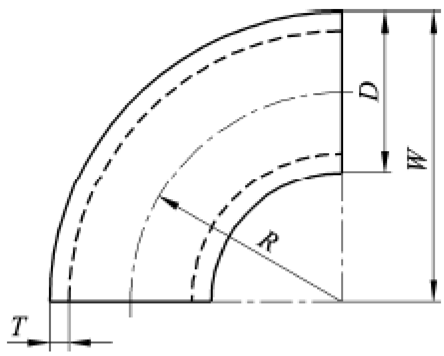


Figure 39: Elbow R/D ratio (DIN EN 10253-2, 2008, p. 24)

The definition of the **curvature radius** is R/D. The standard curvature radius R/D is one. With increasing R/D the disturbances according to consistent velocity distribution are reduced. Additionally, the higher R/D ratio, the less the head losses are. According to Stepanoff the loss coefficient ζ of a **smooth elbow** is decreasing with increasing R/D ratio. The values for smooth elbows for different R/D ratios are shown in Table 2.

R/D	ζ
1	0,27
1,25	0,22
1,5	0,17
2	0,13

Table 2: Losses in a smooth elbow depending on R/D (Stepanoff, 1957, p. 14)

The law of theoretical velocity distribution states that the product of velocity and radius is constant. Practically this is just possible with low fluid speed and / or with high R/D ratios. With normal fluid speeds and little R/D ratios the current starts flow separation at the inner side of the elbow and vortex occur. (Stepanoff, 1957, p. 6) This is illustrated in Figure 40.

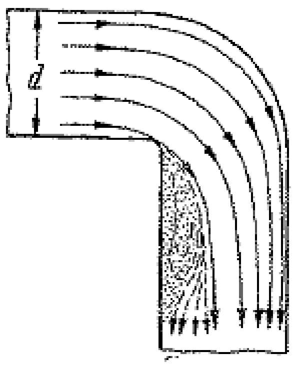


Figure 40: Current streamlines in an elbow (Stepanoff, 1957, p. 6)

To reduce disturbances and to suppress flow separation, the elbow should have a big curvature radius. In addition to that, a higher curvature radius reduces the head losses in the elbow. On the contrary, the higher the curvature radius the more expensive such an elbow is.

Up to DN800 smooth 90° angle elbows are available. Above DN800 only welded elbows are available.

On the contrary, an elbow designed with a sharp 90° angle (Figure 41a), has a loss coefficient of ζ=1. With installations inside the elbow (Figure 41b) the loss coefficient is reduced to ζ=0.136 - 0.24, which results in the same loss coefficient as an elbow with a curvature radius of 1.25. This means the design- and manufacturing effort of installations inside a sharp 90° elbow is not worth in comparison to a usual elbow with a curvature radius of 1.25. (Stepanoff, 1957, p. 14)

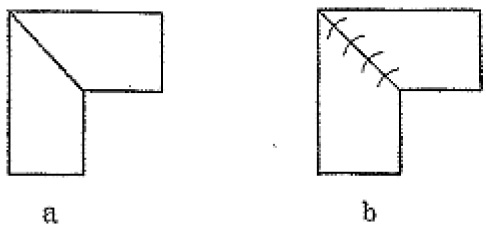


Figure 41: Sharp 90° elbow without (a) or without installations (b) (Stepanoff, 1957, p. 14)

Welded elbows are designed with a certain number of segments. The loss coefficient of segmented elbows depends on the R/D ratio, the medially length of each element and the number of segments. For different R/D ratios and l_{el}/D ratios the local loss coefficients ζ_{loc} are shown in Tables 15 to 17 with 3 to 5 welded elements in the appendix. The total loss coefficient ζ is the sum of the local loss coefficient ζ_{loc} and the friction loss ζ_{fr}. To summarize, the higher the numbers of welded segments, the higher the R/D ratio and the higher the medially length l_{el} is, the less the local loss coefficient is.

The head loss in an elbow is calculated according to Equation 6.

$$h_{v\_elbow} = \xi \times \frac{c^2}{2 \times g}$$

Equation 6: Elbow head loss

h_{v_elbow}	Elbow head loss	[m]
ζ	Elbow loss coefficient	[-]
c	Velocity inside the pipe	[m/s]
g	Gravitational constant	[m/s^2]

The discharge nozzle is either flanged (Figure 43) or with a plain end (Figure 42), according to customer requirements.

Figure 42: Discharge nozzle configuration – plain end (Sulzer SJM, 2015, p. 3)

Figure 43: Discharge nozzle configuration – flanged end (Sulzer SJM, 2015, p. 3)

3.5.11 Motor couplings

The determinant factor if the motor coupling is either rigid or flexible, is where the axial thrust bearing is located. If the axial thrust bearing is inside the motor, the coupling must transmit the axial thrust, hence, only rigid motor couplings are possible to use. Thus, flexible couplings are used when the axial thrust bearing is located inside the pump head.

3.5.11.1 Rigid motor couplings

These couplings cannot compensate any misalignment of the shafts, hence, if misalignment occurs, this will result in higher vibrations and wear (Bloch & Budris, 2010, p. 306).

A rigid flanged coupling, as seen in Figure 44, cannot be used for axial adjustment.

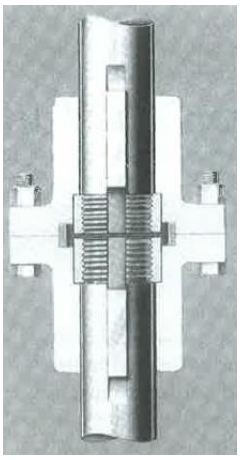


Figure 44: Flanged shaft coupling (Bloch & Budris, 2010, p. 306)

A rigid adjustable coupling allows adjusting the axial position of the shaft by simply turning the plate. This plate is either positioned between the coupling halves, which are illustrated in Figure 45, or below the coupling, which is illustrated in Figure 46.

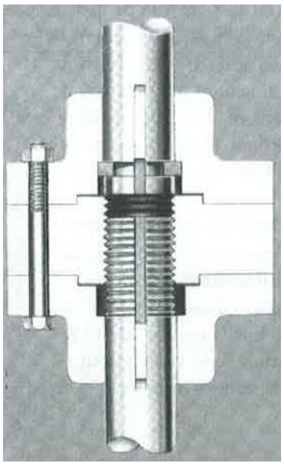


Figure 45: Rigid adjustable coupling (Bloch & Budris, 2010, p. 306)

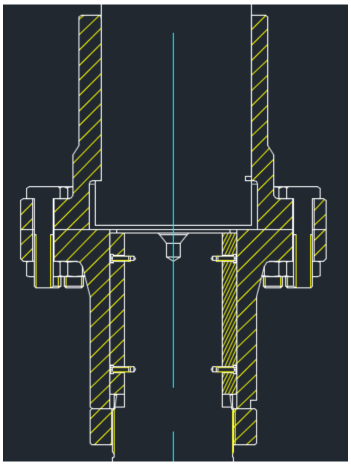


Figure 46: Rigid adjustable coupling 2 (Andritz, internal source)

An adjustable spacer coupling may be used for removal of the mechanical seal. During service the spacer part is removed and the mechanical seal can be maintained without removing the motor. Additionally, this coupling is axially adjustable with a threaded plate. This coupling is shown in Figure 47.

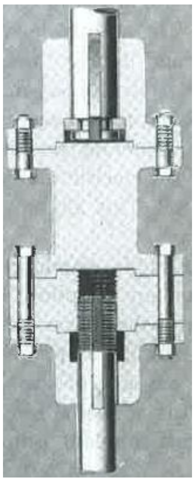


Figure 47: Rigid adjustable spacer coupling (Bloch & Budris, 2010, p. 306)

3.5.11.2 *Flexible motor couplings*

These types can compensate misalignment, which is either caused by angular, radial, length or combinations of it. Flexible couplings may have shear pins for limiting the torque e.g. if a short circuit occurs. These flexible couplings are available in different designs.

3.5.12 Sealing options

The shaft sealing of a VLSP is located at the discharge elbow, where the shaft is extending through the elbow. This shaft sealing is either a stuffing box or a mechanical seal, which are rotational sealings.

3.5.12.1 Stuffing box sealing

The limitation of a stuffing box is the heat conduction. For cooling reasons the stuffing box must leak. If the stuffing box is not leaking, the packing is burnt. The stuffing box is only used for environmentally friendly fluids due to permanent leakage. If the leakage is too high, the stuffing box must be adjusted with the stuffing box gland. The packing material is either PTFE, graphite, aramid or wool. The stuffing box is used for low pressure applications and low sliding velocities but there are several elaborate designs available for high pressure applications as well. High pressure stuffing boxes use cooling water or a cooling chamber. The stuffing box is usually cheaper than other shaft sealing options, which is illustrated in Figure 48. All the components of a stuffing box are illustrated in Figure 48. (Hellmann, 2011, p. 312)

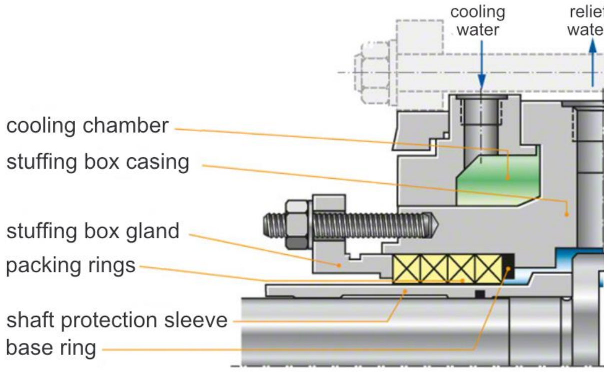


Figure 48: Stuffing box (Hellmann, 2011, p. 312) – (source modified)

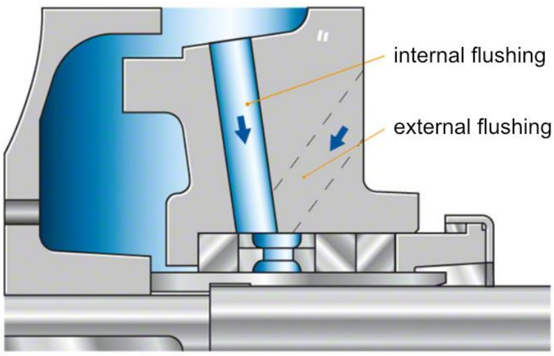


Figure 49: Stuffing box flushing (Hellmann, 2011, p. 313) – (source modified

The flushing of the stuffing box is either with an internal or an external fluid, which is illustrated in Figure 49.

3.5.12.2 Mechanical sealing

The mechanical sealing has, in comparison to the stuffing box, a sealing gap perpendicular to the shaft. Additionally, the mechanical seal requires less space and no maintaining in comparison to the stuffing box sealing. The mechanical seal is applied for all sealing pressures and tangential speeds. (Hellmann, 2011, p. 314)

The different components of a standard mechanical seal are illustrated in Figure 50. It consists of a stationary ring - fixed with a pin, a rotating ring which is tightened to the stationary ring by a compression spring and the hydraulic force. Between the finished-machined sliding surfaces is a liquid lubrication film.

The disadvantage of a mechanical seal is when using fluids with particles, the sealing surface gets damaged. (Hellmann, 2011, p. 314)

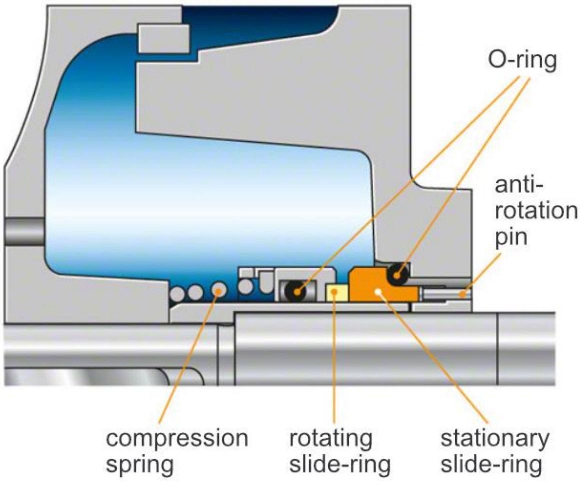


Figure 50: Mechanical sealing (Hellmann, 2011, p. 315) – (source modified)

3.5.13 Axial thrust

The axial thrust in a VLSP is generated due to the hydraulic force of the impeller and the weight force of the impeller and the shaft. In Figure 51, the pressure distribution of a closed impeller is shown, whereas the resulting axial thrust is downwards because of lower suction pressure, the weight force and because of higher pressure at the back area of the impeller.

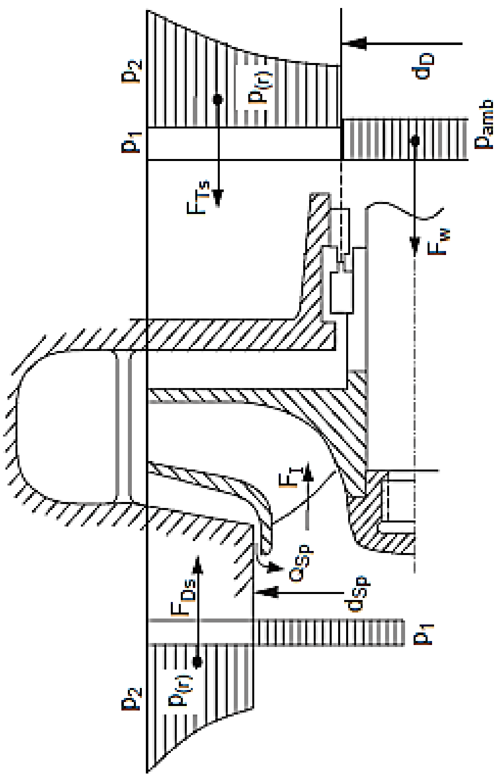


Figure 51: Pressure distribution on an impeller (Gülich, 2013, p. 607)

The resulting axial thrust force consists of the hydraulic force, the weight force, the momentum at the impeller, the force of the suction pressure on the shaft. This is shown with Equation 7.

$$F_{ax} = F_{Hydr} + F_W - F_I + F_S$$

Equation 7: Total axial force

F_{ax}	Axial thrust force	[N]
F_{Hydr}	Hydraulic force	[N]
F_W	Weight force	[N]
F_I	Momentum force	[N]
F_S	Suction force	[N]

The **weight force** is calculated according to Equation 8, where the weight of the impeller, the shaft and the water inside the impeller is summed up and multiplied by the gravitational constant, which is 9,81m/s^2.

$$F_W = \left(m_{IP} + m_{shaft} + m_W\right) \times g$$

Equation 8: Weight axial force

m_{ip}	Impeller weight	[kg]
m_{shaft}	Shaft weight	[kg]
m_w	Water weight inside the impeller	[kg]
g	Gravitational constant	[m/s^2]

The **hydraulic axial force** depends on the design of the impeller, hence, of the specific speed nq. The axial force of a radial impeller is higher than the axial force of an axial impeller, mainly because of higher possible heads of radial impellers.

For closed low specific speed impellers, hence radial and mixed flow impellers, the axial force is calculated as mentioned with Equation 9. This equation is only an estimation formula because the pressure characteristic is not considered. The head H is, depending on the specific speed, the highest possible head which occurs during operation, to incorporate the worst-case scenario. This is mostly the head at zero flow. This equation is just applicable for axially-uncompensated impellers. All the dimensions which are used in Equation 9 are illustrated in Figure 52.

$$F_{Hydr} = 0{,}9 \times \rho \times g \times H \times \frac{({d_{sp}}^2 - {d_D}^2) \times \pi}{4}$$

Equation 9: Axial force of radial/mixed flow impeller (Gülich, 2013, p. 613)

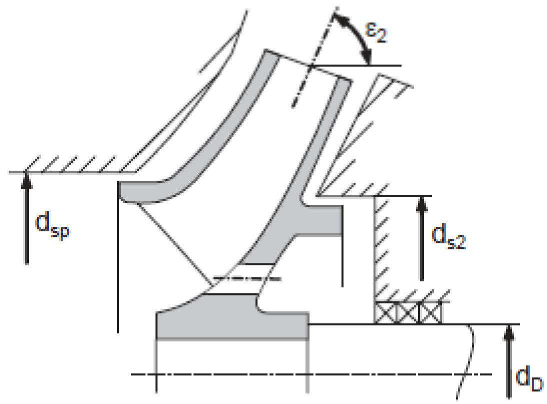


Figure 52: Dimensions for a radial/mixed flow Impeller (Gülich, 2013, p. 612)

For high specific speed impellers, hence axial ones, the axial force is calculated according to Equation 10, where d_2 is the outer diameter and d_D is the shaft diameter.

$$F_{Hydr} = 1{,}1 \times \rho \times g \times H \times \frac{(d_2^2 - d_D^2) \times \pi}{4}$$

Equation 10: Axial force of an axial impeller (Gülich, 2013, p. 613)

The **momentum force** due to redirection of the fluid is calculated with Equation 11.

$$F_I = \rho \times \vec{c} \times A \times (\vec{c} \times \vec{n})$$

Equation 11: Momentum force

$\vec{c}$	Fluid velocity in axial direction	[m/s]
A	Flow area	[m^2]
$\vec{c} \times \vec{n}$	Fluid velocity perpendicular to the flow area	[m/s]

The **suction force** is considered with the static head before the impeller, the dynamical pressure due to the speed and the loss ahead the impeller. The suction force is calculated as mentioned in Equation 12. The static head is represented for the worst case by the $NPSH_{req}$. To consider the proper static head the $NPSH_{av}$ minus the losses should be used.

$$F_S = \left(NPSH_a - \frac{v_S^2}{2 \times g} - \xi \times \frac{v_S^2}{2 \times g} \right) \times \rho \times g \times \frac{d_D^2 \times \pi}{4}$$

Equation 12: Suction force

$NPSH_a$	Net positive suction head available	[m]
v_S	Suction velocity	[m/s]
ζ	Loss coefficient	[-]
ρ	Density	[kg/m^3]

The pressure distribution, as shown in Figure 52, is a result of the currents in the back and the front of the impeller.

The suction side current is the same for single- and multistage pumps. The gap current to the suction side accelerates the rotation with $\frac{\omega}{2}$, hence, the pressure decreases in the direction to the shaft. This is illustrated in Figures 53 and 54.

The pressure side current for a single-stage pump accelerates in the direction to the shaft with $\frac{\omega}{2}$, hence it decreases the pressure (Figure 53). The pressure side current for a multi-stage pump throttles in the direction to the impeller outer diameter the rotation with $\frac{\omega}{2}$ and raises the axial thrust (Figure 54). This systematics is just applicable without any axial thrust compensation options. The influence on the characteristic of the pressure distribution behind, and in front of the impeller, is shown in Figure 55.

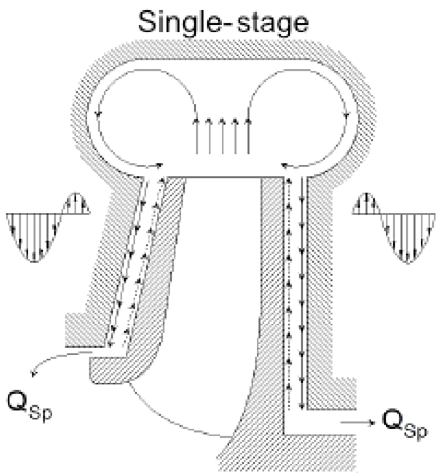


Figure 53: Currents in back and front area of the impeller for a one stage pump (Jaberg, 2012, p. 135) – (source modified)

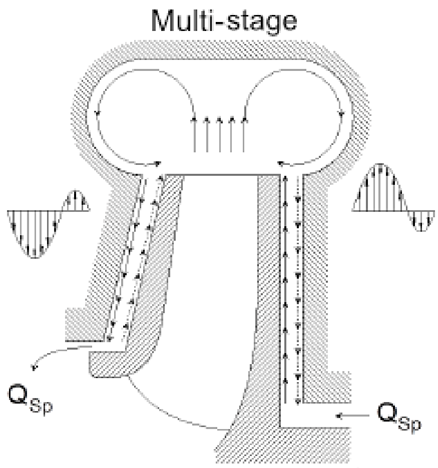


Figure 54: Currents in back and front area of the impeller for a multistage pump (Jaberg, 2012, p. 135) – (source modified)

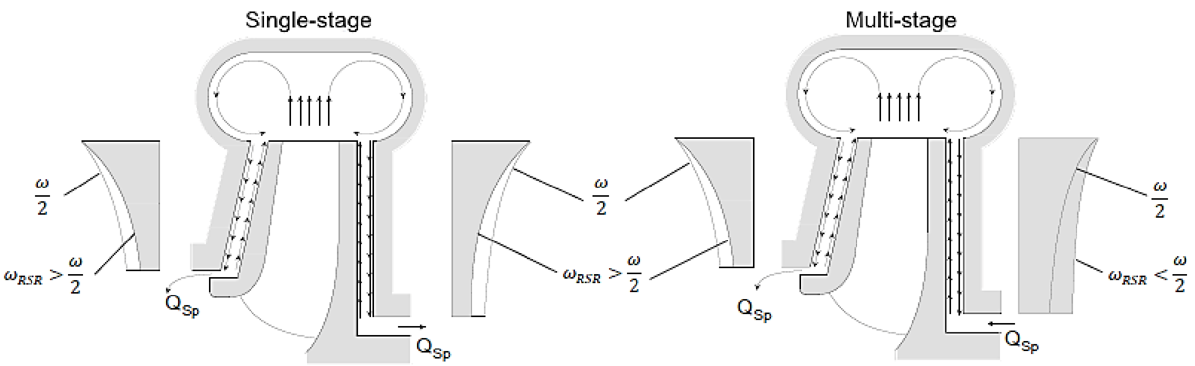


Figure 55: Influence of the currents on the axial thrust (Jaberg, 2012, p. 135) – (source modified)

3.5.13.1 Axial thrust bearings

The axial force is absorbed by the axial thrust bearing. This is designed in different variations.

The axial thrust bearing is either integrated inside the motor or inside the pump-head. This depends on the customer requirements and economic reasons. The axial thrust bearing integrated in the motor is a standard in the USA and results in a more expensive motor. In Europe, an extra axial thrust pot inside the pump head is preferred.

For the axial thrust bearing located inside the pump head the following bearings are used:

- Angular contact bearing
- Spherical roller bearing
- Tilting pad bearings

The tilting pad as an anti-friction bearing is used for high thrust applications.

The bearing is either oil or grease lubricated.

3.5.13.2 Axial thrust compensation

- No compensation

  The whole axial force is absorbed by the axial thrust bearing. The pump has the best efficiency without any axial thrust compensation (Hellmann, 2011, p. 36). Due to the vertical installation of the pump, the axial thrust is sometimes rather high. Thus, axial thrust compensation should be considered.
- Back vanes

  One option to lower the axial thrust is to attach back vanes at the back of the impeller, this is illustrated in Figure 56. The back vanes accelerate the fluid at the back of the impeller and reduce the static pressure. The disadvantage of this design is the higher power consumption since these back vanes work as a pump at zero flow and decrease the efficiency up to 3%. Back vanes are not applicable with abrasive fluids. (Jaberg, 2012, p. 138)

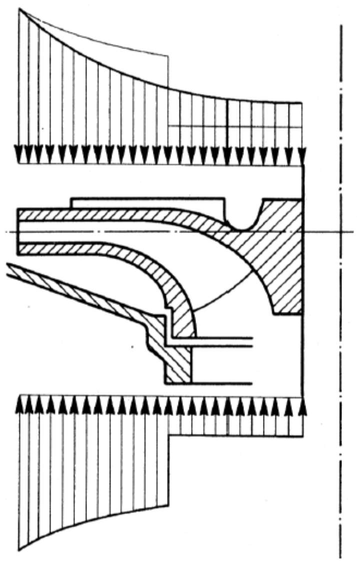


Figure 56: Back vanes (Jaberg, 2012, p. 138)

- <u>Balance holes</u>

  The second option is to use back-wear rings and balance holes, as illustrated in Figure 57. The area of the back-wear ring (xxx) throttles the pressure and inside the area A the suction pressure is applied with the connection of the balance hole B. Thus, the area where the high pressure is applied is reduced, resulting in less axial thrust. Investigations have shown that this is about 75% less. The disadvantage is less pump efficiency. (Bloch & Budris, 2010, p. 356)

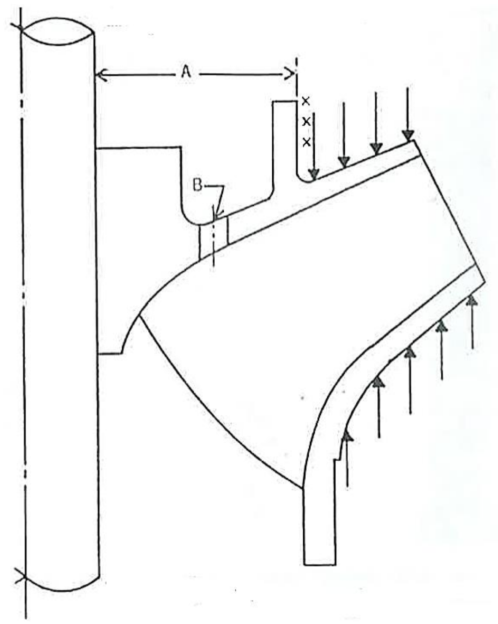


Figure 57: Balance holes (Bloch & Budris, 2010, p. 356)

3.5.13.3 Lubrication

The lubrication is either with grease or oil. This is done with an oil sump. In Figure 58 such an oil sump is illustrated. Additional oil cooling systems can be applied. The oil cooling is either executed with a cooling coil or a mantle cooling system of the casing.

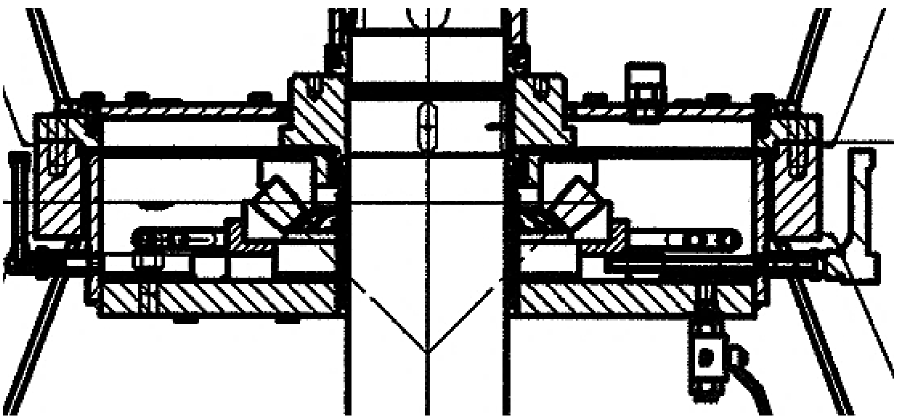


Figure 58: Oil sump (Andritz, internal source)

3.5.14 Radial force

Theoretical the radial force of a guide vane pump is zero, because the fluid flow at the circumference is symmetric. But due to tolerances and manufacturing errors at the impeller, a radial force occurs. (Hellmann, 2011, p. 221)

The radial force in a guide vane pump is calculated below with Equation 13. All the dimensions used for the calculation are shown in Figure 59. The coefficient k_{R0} is for $q^*=0$ $k_{R0} = 0{,}02 - 0{,}09$ and for $q^*=1$ it is $k_{R,opt} = 0{,}01 - 0{,}06$. For axial flow pumps b_2 corresponds to d_2 and $k_{R,D}$ is for this case 0,02 for $q^*<1{,}2$. (Gülich, 2013, p. 658)

$$F_R = k_R \times \rho \times g \times H \times d_2 \times b_2$$

Equation 13: Radial force (Gülich, 2013, p. 658)

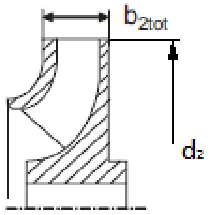


Figure 59: Radial force dimensions (Gülich, 2013, p. 658) – (source modified)

3.5.14.1 Radial bearing types

The radial bearings work with the slide principle; hence, these are slide bearings. The friction between the bearing and the shaft is minimized with a medium which should separate the two parts. For slide bearings 2 different types are possible:

- Hydrostatic slide bearings
- Hydrodynamic slide bearings

In hydrostatic slide bearings, the pressure of the fluid to separate the parts, is generated by an external pump. The pressure in hydrodynamic slide bearings is created with the relative motion between the shaft and the slide bearing. For VLSP's hydrodynamic slide bearings are usually used. When hydrostatic slide bearings are used, the shaft is designed as an enclosed type, which is described later in this chapter. For the start-up condition, when the slide bearing is not externally lubricated, special dry start condition slide bearings must be applied. The usual width to diameter ratio of such radial slide bearing is 1 to 1,5.

Which slide bearing material is applied depends on the conditions. The different characteristics of bearing materials are described and assessed in Table 3. 0 means "not qualified", 1 "limited qualified", 2 "qualified", 3 "good qualified" and 4 "very good qualified".

	Cast iron	Sintered metal	Co-Sn cast	G-CoPb alloys	Pb Sn alloys	Plastics	Rubber	Carbon graphite
Sliding properties	2	2	3	4	4	4	4	4
Emergency running properties	2	4	2	3	3	4	0	4
Wear resistance	4	2	4	2	1	2	0	1
Stat. load capacity	4	2	3	1	1	1	0	1
Dyn. load capacity	3	1	3	1	1	1	0	0
High slide velocity	1	0	3	4	4	0	0	3
Insensitive for edge pressure	0	0	3	3	4	4	4	2
Heat conducting properties	2	2	3	2	1	0	0	3
Small dilatation	4	4	3	2	2	0	0	4
High temperature resistance	2	2	2	0	0	0	0	4
Oil (-grease) lubrication	4	4	4	4	4	4	2	4
Water lubrication	0	0	0	0	0	4	4	4
Dry running conditions	0	0	0	0	0	4	0	4
Corrosion resistance	1	2	2	2	1	4	4	4

Table 3: Slide bearing selection and the qualification (Wittel , et al., 2009, p. 536)

Which bearing material is selected depends on how the radial bearing is designed and which lubrication is used. If the slide bearing is lubricated with the pumped liquid, which is the most common solution, the bearing material is mainly plastic or rubber. This is due to the properties of these materials, the good sliding properties, and the usability for water lubrication and hence the radial force is low, this type has not to take a high load. The good dry running conditions of plastic slide bearings are an advantage. The slide bearings are assembled at the bearing spiders, between the column pipe sections. This design is called open line-shaft and is illustrated in Figure 60.

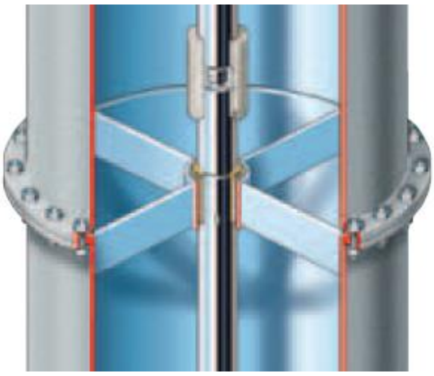


Figure 60: Open line-shaft design (Flowserve VCT, 2008)

Figure 61: Enclosed line-shaft (Flowserve corporation – VTP, 2014)

If the slide bearing is lubricated with oil, grease or clean water, a special design for the slide bearings must be used. This design is called enclosed line-shaft and is illustrated in Figure 61. This design isolates the shaft from the pumped liquid, which is important, when abrasive fluids are pumped, hence, reducing required maintenance of the slide bearings. With this design, often copper cast materials and alloys are used, especially with higher speeds. This design is also used when the pump extends a certain length and the used bearings are not proper for dry start-up or cannot be lubricated fast by the pumped liquid.

When using a hydraulically balanced impeller a cut-less rubber bearing should not be used inside the bowl. The recirculation flow through the bearing, due to the pressure difference, will harm the balancing effect, hence, the diameter of the balance holes has to be increased, which leads to less pump efficiency. Rubber bearings are also not the best solution for bowl bearings since this type needs higher clearances (Bloch & Budris, 2010, p. 361).

Due to recirculation inside the bowl slide bearing, the pressure distribution at the back of the impeller is comparable to a multistage-pump pressure side, resulting in higher axial thrust.

3.5.15 API conform design

The American Petroleum Institute (API) is an association of the most important companies in the oil- and gas sector. The API establishes and maintains the norm API610, to which all centrifugal pumps, applied in the oil- and gas market, must be aligned. The vertically suspended centrifugal pumps are classified as mentioned in Figure 3.

To fulfil the API requirements to use a pump for oil- and gas applications, certain design rules must be considered. Sometimes a customer even requires an API conform design for other applications as well.

Generally, all pump curves must be devalued according to the losses in the column pipe and the elbow. Pumps which handle with explosive fluids should have vent connections for suction barrels and seal chambers. For the clearances of the impeller and the slide bearings only certain values are allowed. If a semi-open impeller is used pumps must use a replaceable casing liner. To reduce shaft deflection the impeller must be located between bearings, hence, a bearing in the suction bell according Figure 30 should be applied. (American Petroleum Institute, 2010, p. 93)

The axial thrust bearing, either located in the pump head or the motor, must endure at least 25000 hours bearing life with continuous operation and 16000 hours with varying operation. The thrust bearing must be designed to carry the maximum load at any operating point. (American Petroleum Institute, 2010, p. 69) Each pump should accommodate two radial bearings and one double-acting axial thrust bearing of one of the following types:

- Rolling-element radial and thrust
- Hydrodynamic radial and rolling-element thrust
- Hydrodynamic radial and thrust (American Petroleum Institute, 2010, p. 56)

All the bushings must be corrosion-resistant and abrasion-resistant. The maximum spacing between the bearings must be according Diagram 12, to ensure to stay below the first critical speed of the shaft. (American Petroleum Institute, 2010, p. 94)

Vertical pumps are susceptible for vibration, especially the critical speeds must be analysed and the pump must be designed with a 20% safety margin between the operating speed and the natural frequency of the motor support. (American Petroleum Institute, 2010, p. 94)

The radial bearings are usually lubricated by the pumped liquid, if the pumped liquid is not suitable an enclosed line-shaft should be used (American Petroleum Institute, 2010, p. 94).

The pump should be provided with a device which prevents reverse rotation. (American Petroleum Institute, 2010, p. 95) This reverse rotating devise is illustrated in Figure 16, with a locking arm and the self-release coupling.

If the pump is without motor-integrated thrust bearing, rigid adjustable couplings should be used. If a mechanical seal is used, the rigid coupling must be a spacer type, to replace the sealing without removing the motor. (American Petroleum Institute, 2010, p. 95) Vertical pumps should be mounted on a separate foundation plate. (American Petroleum Institute, 2010, p. 96)

For the shaft sealing system, the API recommends special API-plans.

The pump must be tested fully assembled. A test with only the pressure casing and the impeller is not sufficient. If required, the pump must be tested according to resonance as well. (American Petroleum Institute, 2010, p. 96)

The standard, API610 delivers in-depth information.

4 Market analysis

4.1 Internal market analysis

To conduct a proper market analysis, the most important data is the internal collected data. This internal data is derived according to experience of the sales force, or due to documentation with a customer relationship management tool.

Referring to the company's customer relationship management tool all the inquired, offered and sold pump sizes are documented and this data is used for an internal analysis. Pump size means the flow rate and head of a pump. All these different sizes are illustrated in Diagram 13 and in more detail in Diagram 14. Overall these are 778 different flow rates and head combinations with in sum 2690 number of pumps over a certain time-period.

It is the fact that not all inquiries, especially of the company's subsidiaries, are considered in these analyses and the pictures below. This means that these figures mainly consider the European market.

It is apparent, that the VLSP is used for different Q and H combinations – the low head and low flow rate applications with a single stage VLSP pump, the high head applications with a multistage design, as well as high flow rate applications. The maximum inquired head is approximately 260m and the maximum flow rate is approximately 18m^3/s. However, mostly this pump is used for heads up to 100m and flow rates up to 6m^3/s, which also is obvious according to the dots in the diagrams on the next page.

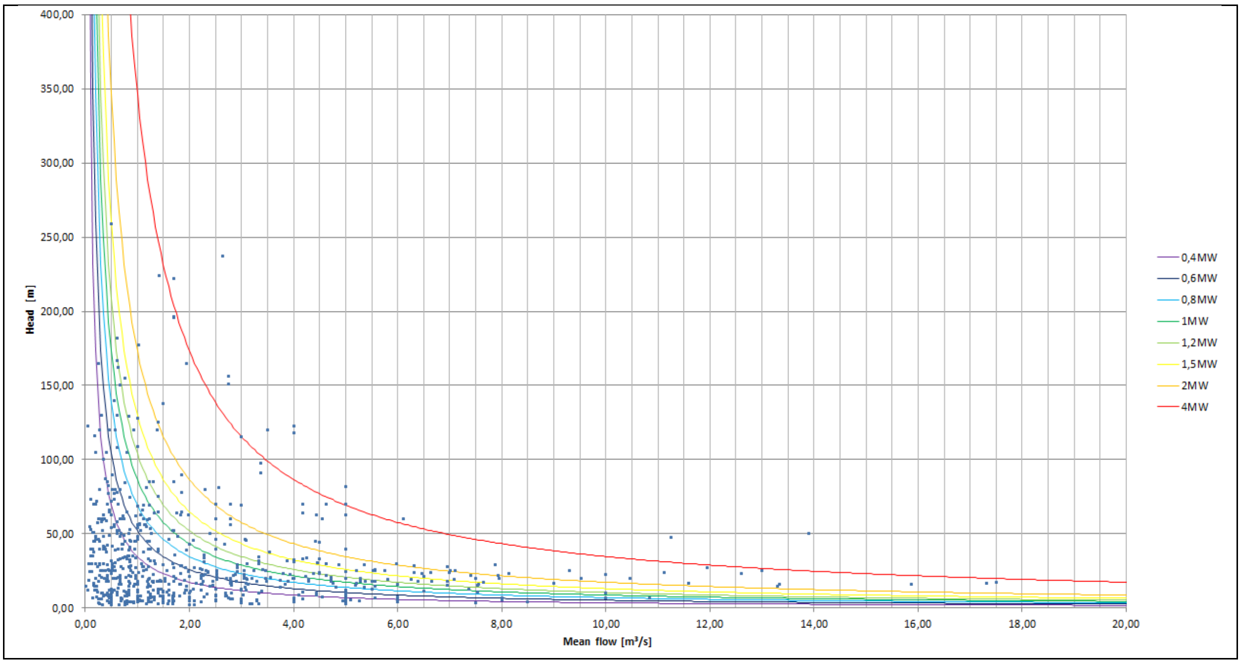


Diagram 13: Inquired, offered and sold sizes

To evaluate in which range the most sizes occur, the amount of different pump sizes, according to the power were analysed. In fact, about 612 different pump sizes have had less than 1 megawatt. These are 78% of all pumps which are inquired and offered. In Diagram 14 it is visible that most of the pumps are below 60m head and 3,5m³/s flow rate.

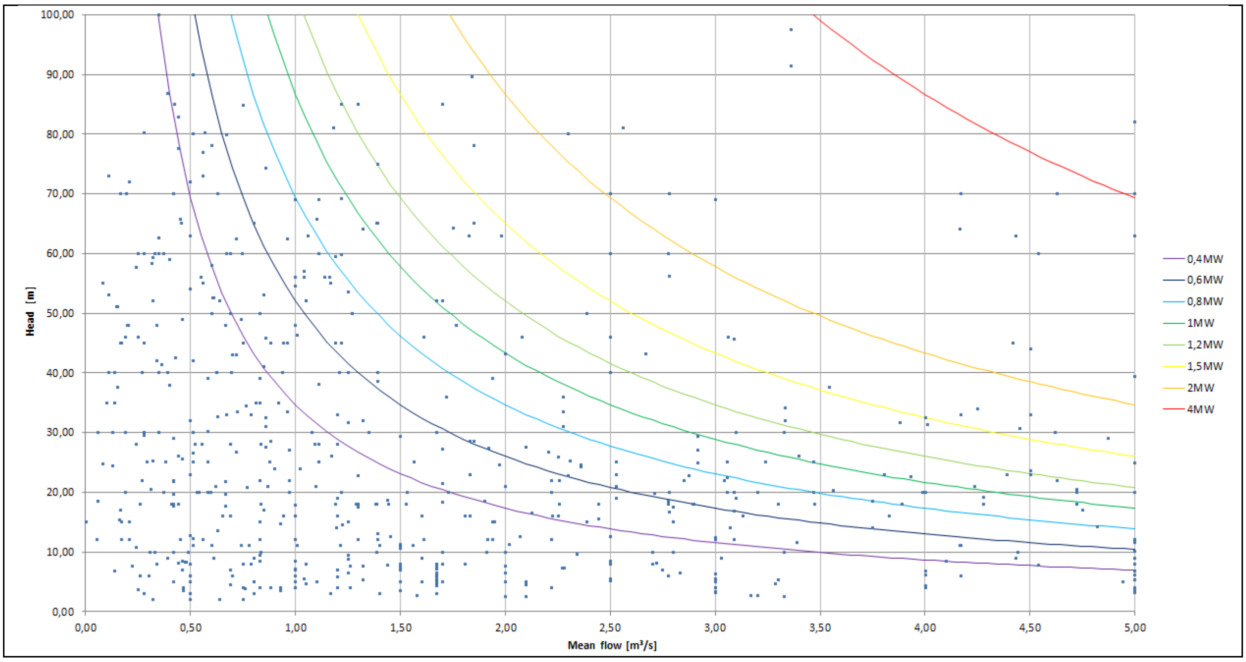


Diagram 14: Inquired, offered and sold sizes 1

Diagram 15 shows in detail the quantity of pumps inquired with a third dimension. The size of the bubbles illustrates qualitatively the amount which was inquired. It is apparently visible that below 1 megawatt the highest number of pumps are demanded.

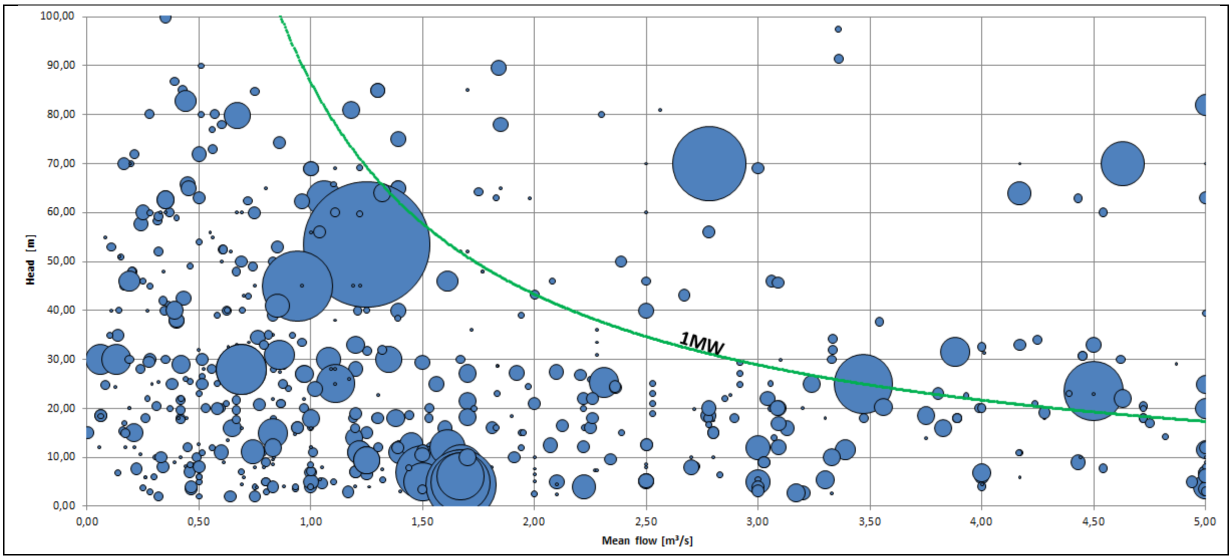


Diagram 15: Amount of pumps below 1MW

The distinct Q and H areas for different assigned applications are illustrated in Chapter 4.3 for each.

Secondly, the experience of the sales force according to market's requirements is taking into consideration. The internal analysis depicts the most important things for customer decisions for VLSP pumps. Especially for the smaller sizes, the efficiency is not the highest valued decision criterion. The most important criterion for selling VLSP pumps are the price and lead time of the pump. Also, the availability of spare parts, the company's reputation and quality are determinant factors. The bigger and higher the power consumption of the pump, the higher the efficiency and the companies' references are valued.

4.2 External market analysis

To validate and supplement the internal analysis, an external market study was conducted by a market research company. The main results are analysed and outlined in detail below.

In 2016, about 158.000 vertically suspended pumps were globally sold. The vertically suspended pumps are split up in different products, which is also mentioned in Chapter 3.3, Figure 3. In Diagram 16 the global pump market for vertically suspended centrifugal pumps is illustrated. For this market analysis, these are split up in submersible pumps, discharge through column, double casing and others. Submersible pumps take the biggest portion of the vertically suspended pump market with 84% of all sold pumps in 2016. The company also has different submersible pumps and motors in its product portfolio. The submersible centrifugal pumps have the highest market portion since the pump itself is rather cheap and standards are well established. One drawback is that the submersible motors are quite expensive, the higher the power output is.

The double casing vertically suspended centrifugal pumps have a share of 2% of the total market. These are mainly used in power plants and are NPSH optimized and especially designed for closed circuits.

The others capture a 10% market share. This segment includes vertical sump pumps and vertical in-line pumps.

The vertical suspended discharge through column pump or in this thesis called 'vertical line shaft pump', has a 4% share of all globally sold vertically suspended pumps. This type is often used for higher depths, higher flow rates and when a dry installed motor is needed, which is a cheaper pump for a higher output rate.

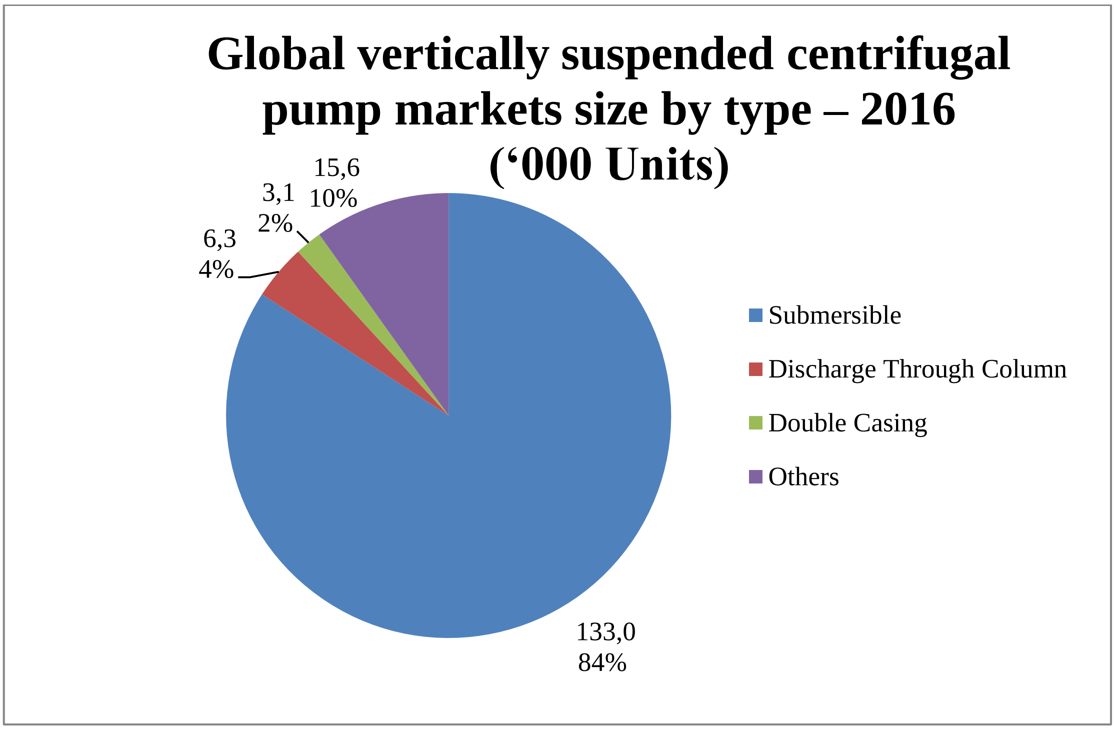


Diagram 16: Global vertically suspended centrifugal pump market size by type (Infiniti Research Ltd., 2017)

The global growth rate of discharge through column vertically suspended centrifugal pumps from 2017-2020 is 7,3%, which means on average an annually growth rate of 2,4%, which is rather promising. This product growth rate lies above many western countries annual prospected Gross Domestic Product growth rates.

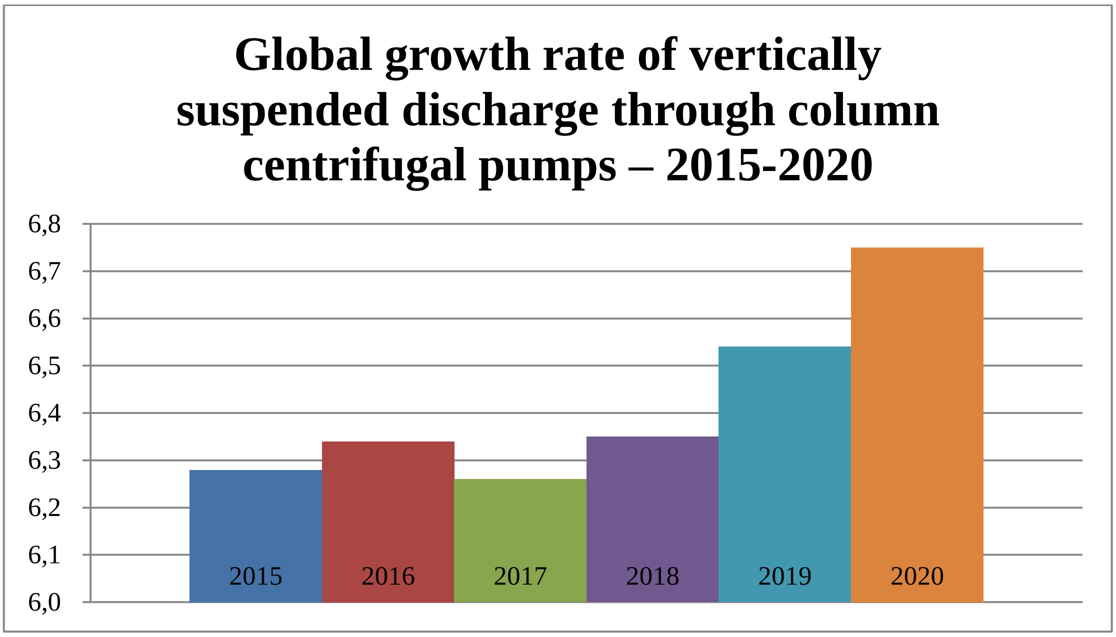


Diagram 17: Growth rate of vertically suspended discharge through column centrifugal pumps – 2015-2020 (Infiniti Research Ltd., 2017)

Vertically suspended pumps are used across different industries and applications. The major applications are, according to Diagram 18, water and waste water applications in sum. This is the fresh water supply, -transport, irrigation and drainage, flood control and waste water. These overall are standing for 50% of all applications. Unfortunately, an application share for VLSP's is difficult to evaluate and to have validated data. The different applications for VLSP's are analysed in Chapter 4.3 with the processes and the typical Q-H ranges assigned.

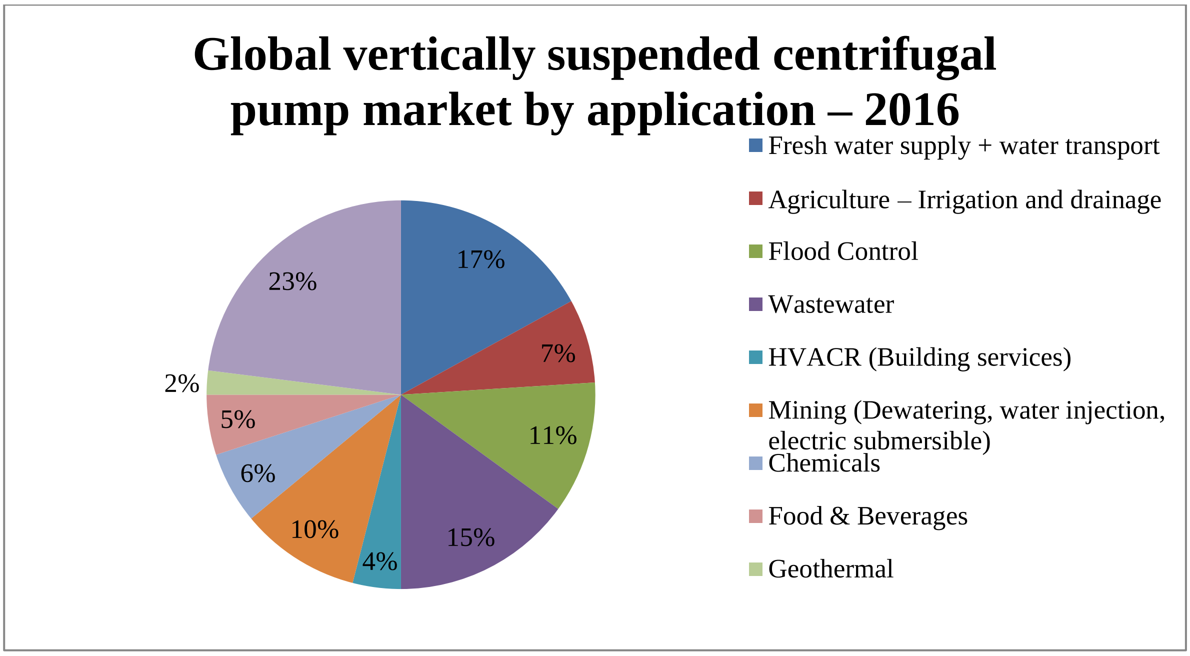


Diagram 18: Global vertically suspended centrifugal pump market by application – 2016 (Infiniti Research Ltd., 2017)

Comparing the yearly potential of approximately 6300 number of pumps with the internal amount of inquiries, it can be stated that the company might not have access to all markets where VLSP's are sold, but bearing in mind that not all inquiries globally in this analysis were considered.

The segmentation according to power of the vertically suspended discharge through column centrifugal pumps is executed as illustrated in Diagram 19. The number of pumps below 1000kW has the biggest portion of all sold VLSP's. This accounts for 3298 number of pumps of in total 6300 sold last year. Hence, more than 50% of all sold VLSP's are below 1 megawatt.

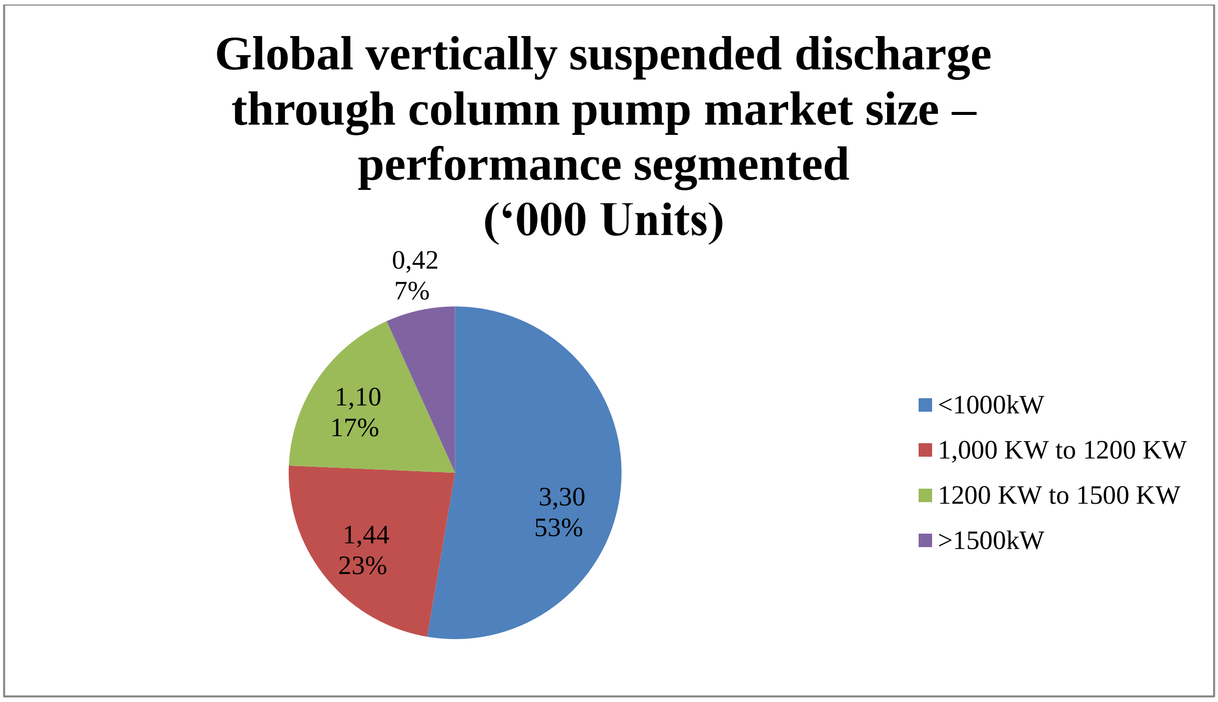


Diagram 19: Global vertically suspended discharge through column centrifugal pump market size – 2017 (Infiniti Research Ltd., 2017)

4.3 Applications

The VLSP is installed in a wide range of different industrial applications. It is used for different water intake applications, such as power plants, flood control or desalination. But also as an application for water transportation, for irrigation and drainage as well as for drinking water and industrial water supply. In Figure 62 below, an overview is illustrated, where VLSP's typically are applied. The blue circles illustrate a VLSP application. In the following sub-chapters and pictures, the different applications are described in more detail.

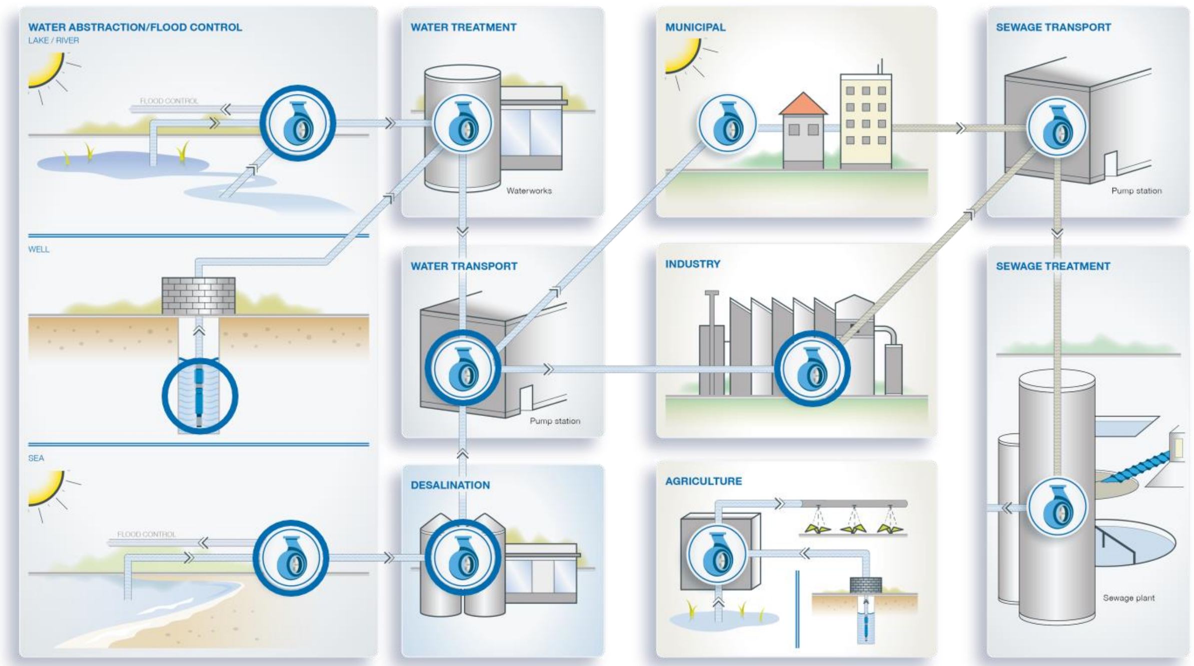


Figure 62: Applications (Andritz – Applications, 2017) – (source modified)

4.3.1 Irrigation and water transport

The market concentration for vertically suspended centrifugal pumps will mostly be in the emerging markets, such as India, China, Brazil, where water applications will be on the rise, according to the huge population and the increasing standard of living. However, also "western" countries, like the USA and Japan are in need to increase the water supply. This is shown with the figures in Diagram 20. Also according to the population increase, agriculture must be used in areas where it is not actually possible to grow crops. Climate change causes an additional stint, many agricultural areas would have been dried out if no irrigation had been used. Diagram 20 below considers all types of pumps, not only the VLSP, but it gives a hint which countries will invest in the future for these applications.

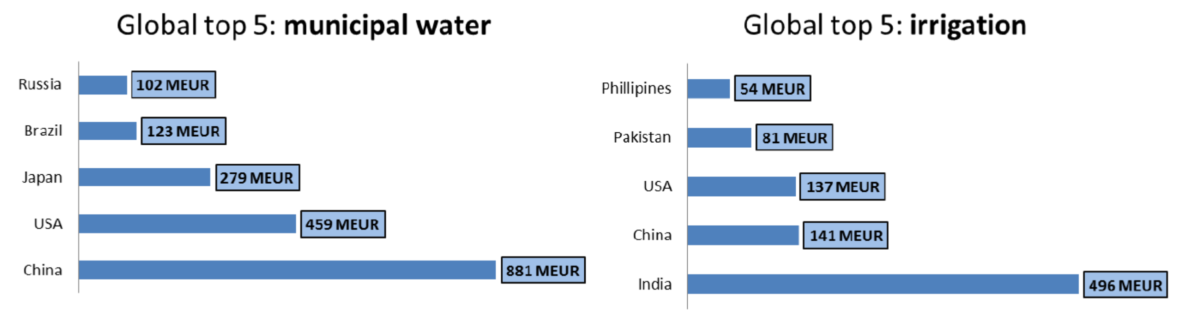


Diagram 20: Industry overview water municipal water and irrigation (Andritz, internal source)

Figure 63 below illustrates the water intake, the water transportation and irrigation pump, signalised with a dark blue pump sign. The water is saved in intermediate reservoirs to release the water when it is needed.

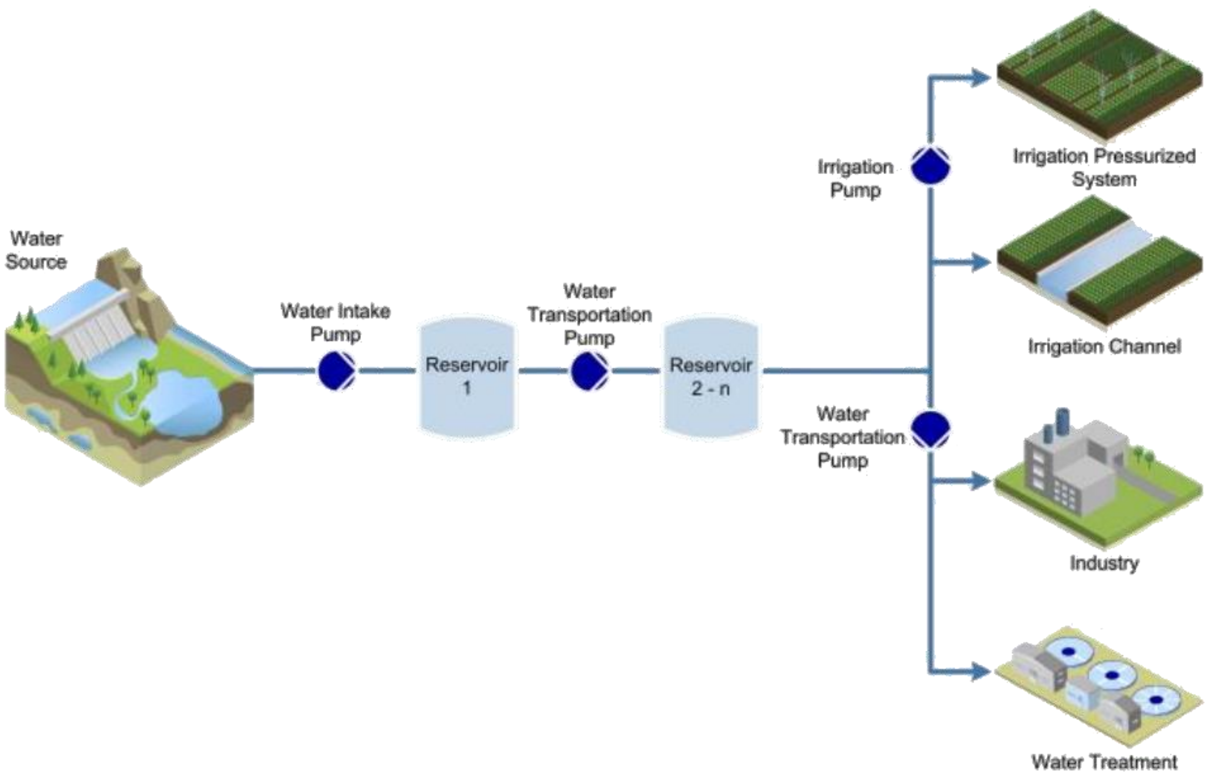


Figure 63: Water intake, -transportation, irrigation (Sulzer – Water intake, -transportation and Irrigation, 2017)

The typical water transport, flood control and irrigation application areas according to Q and H are shown in Diagram 21. All these typical areas are determined by selected inquired and sold VLSP's in a certain time-period.

The usual water transportation head is high, but this depends on the geodetically height. The water is pumped in reservoirs, this is done continuously, and that is why the flow rate maximum is 3m³/s.

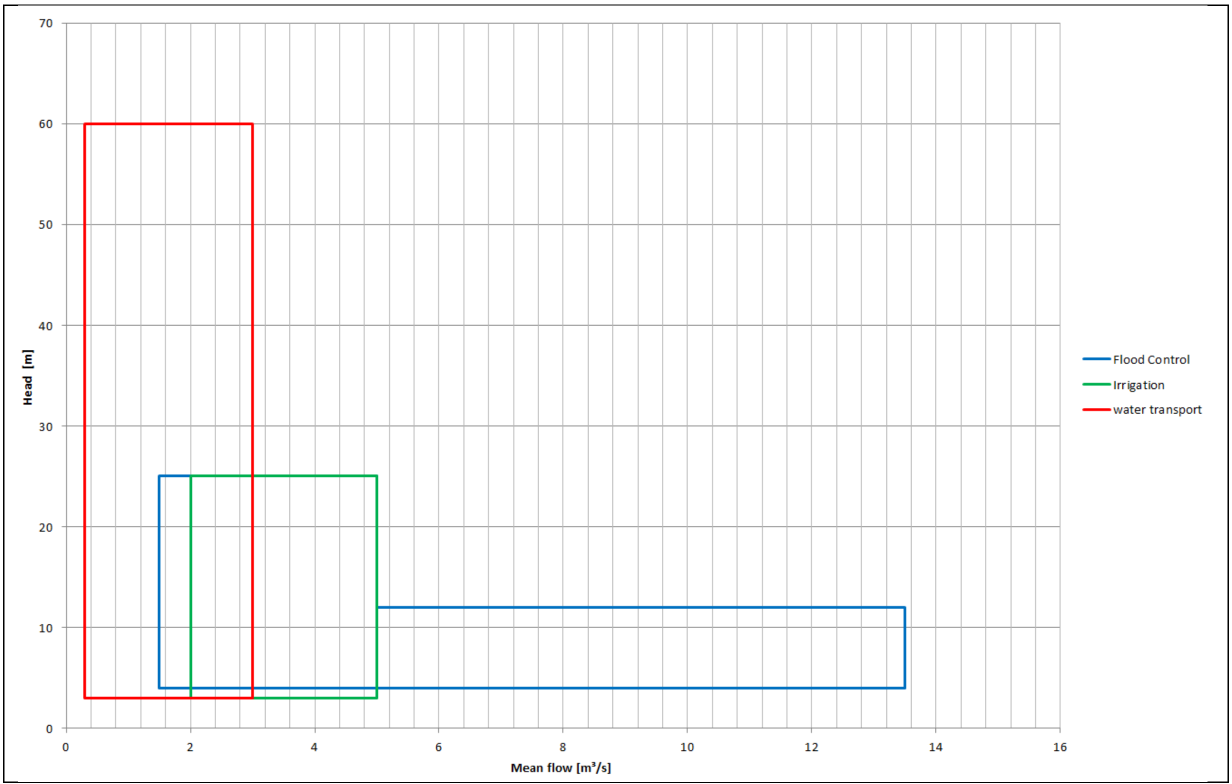


Diagram 21: Water transport, Flood control and irrigation Q-H areas

The irrigation head depends predominantly on the geographical location, but since agriculture is principally possible in flat areas, the geodetical heads are low. The irrigation is used in areas, where long dry periods during the year occur, thus here artificial irrigation basins are used.

When taking a closer look at the Chinese irrigation market, 2 regions can be indicated, where high head and low head VLSP's are mostly used. The coast and Eastern region of China (bright green and yellow area) is a chiefly flat area, whereas low head VLSP's (<15m) are mostly used for irrigation. The western and central region (red area) is more an alpine area, where high head VLSP pumps are used for irrigation. This topography map of China is shown in Figure 64. This information is according to the internal sales manager and is based on their experience. The influence of the piping system, hence the head loss in the pipes, is already taken into consideration. Sometimes an irrigation application will extend, according to the different piping systems, the illustrated flow rate and head range in Diagram 21.

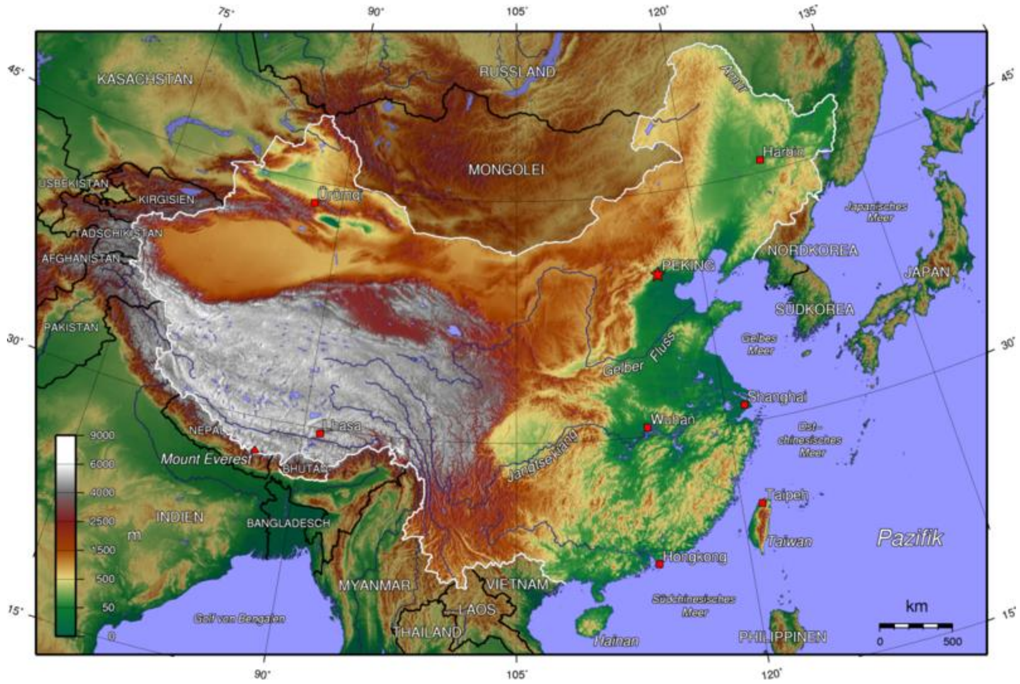


Figure 64: Chinese map – topography (weltkarte.at, 2017)

4.3.2 Flood control

The increase of disastrous hurricanes, monsoons and more devastating rain events due to climate change, flood control applications will a have higher demand in future. The biggest flood control markets are, as shown in Diagram 22, China, USA and India. As a note, the application flood control is also covered with other pump types, but it obviously illustrates the future flood control markets.

The head for flood control is usually low and this is mainly depending on the geographical location. The water is pumped into the sea or a basin, this also explains the low needed elevation. The flood control assigned Q-H range is illustrated in Diagram 21. The flow rate can be rather high since the water has to be transported off fast.

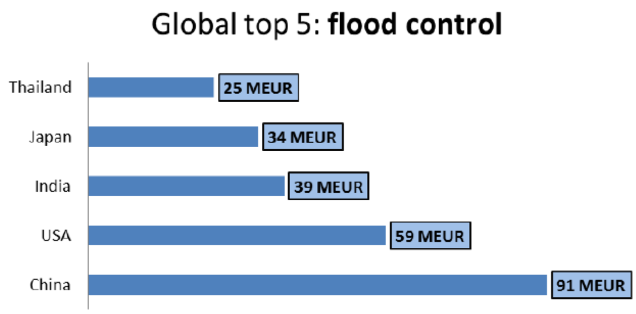


Diagram 22: Global top 5 flood control markets (Andritz, internal source)

The VLSP is typically used for flood control application, because it is a cheap pump according to pump- and site costs and requires low floor space.

4.3.3 Desalination

Seawater desalination is getting more important, since drinking water is becoming a rare good, especially in the emerging markets this will be an application on the rise. According to companies internal source the Middle East market is the most promising desalination market, followed by East Asia and Africa.

The VLSP is used as a water intake pump, marked with the blue pump sign in Figure 65. In addition, different high pressure multistage-, axially split case-, end suction pumps are used for seawater reverse osmosis application.

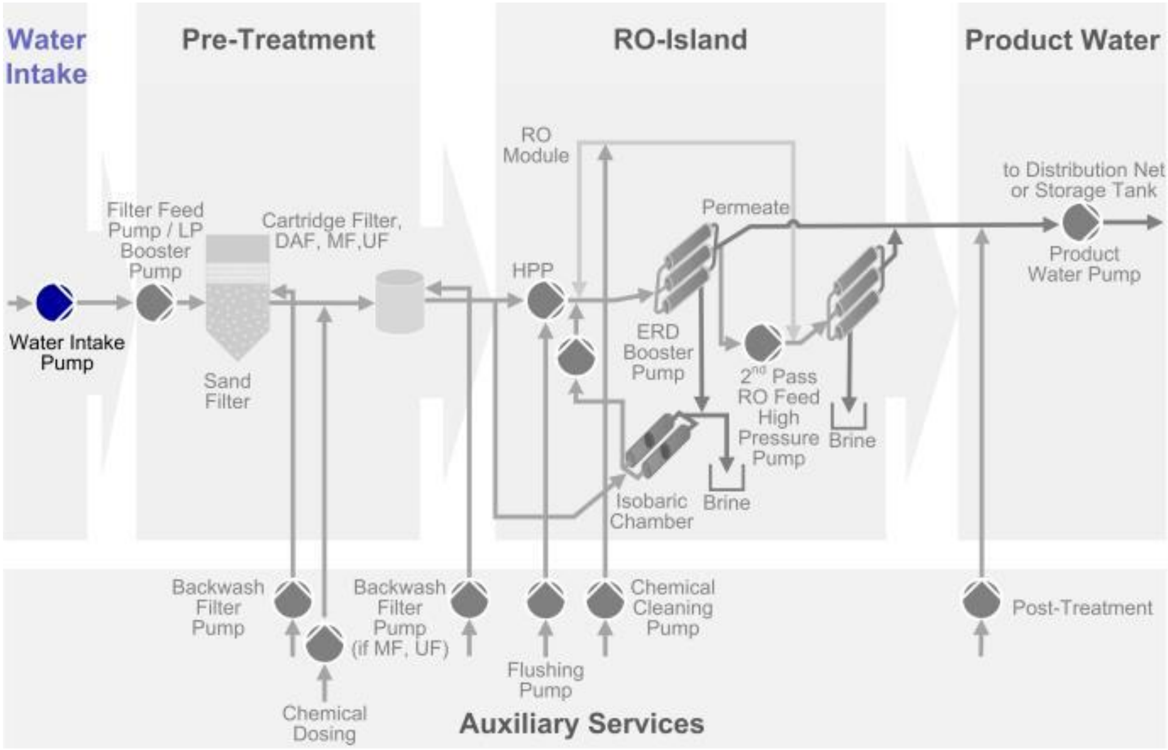


Figure 65: Water intake SWRO (Sulzer – Desalination, 2017)

In Diagram 23 it is illustrated in which area VLSP's are used for water intake for desalination. To determine the certain area where VLSP's for desalination are applied, selected inquiries of a certain time-period were taken into consideration. The dots in Diagram 23 symbolize each inquiry. The required head is determined by the geographical area and the process itself. If an additional booster pump is used, the VLSP can also have lower heads - according to the process. The pre-treatment is also an assertive factor for the needed pump head. If for the pre-treatment, a sand filter is used, the required head is lower, because the water travels through the sand gravitationally. If a micro filter or a Nano filter is used the required head is higher. The flow rate mainly depends on the plant size.

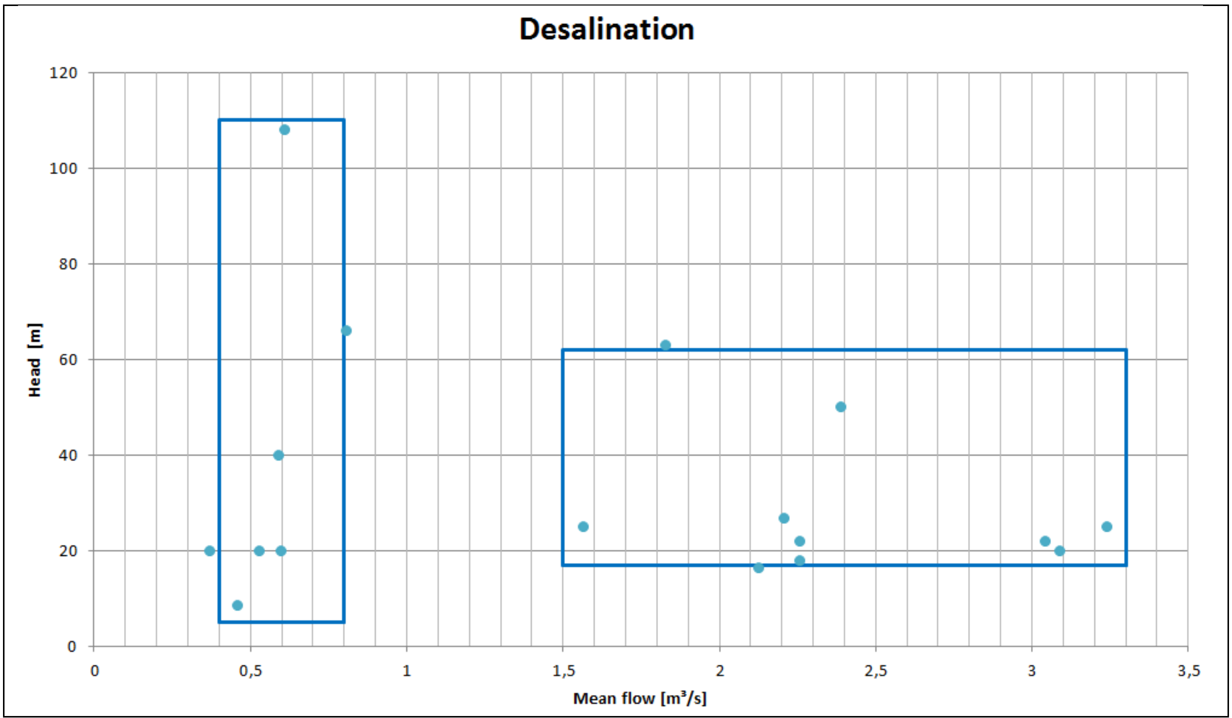


Diagram 23: Desalination Q-H areas

4.3.4 Cooling water pump - water intake

The VLSP is often used for the power industry, the Chinese market is here the most promising one, which is illustrated in Diagram 24 below.

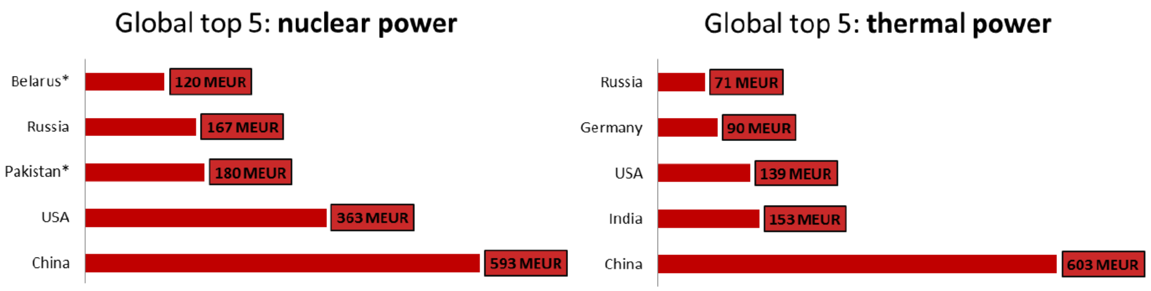


Diagram 24: Industry overview - power industry (Andritz, internal source)

The VLSP is a pump, which is often used for cooling water pumps, as water intake for coal fired-, gas fired- and biomass power plants, for geothermal plants and solar-thermal plants. In Figures 66 to 69 the different power plant applications are shown and marked with the blue pump sign and the CWP abbreviation. The VLSP is used in this application for the secondary cycle, the cooling cycle, and pumps the condensated water to the condensator and dissipates the heat, transferred with water to the cooling tower. This all power plant applications have in common.

The typical flow rate and head area, where cooling water pumps are used, is illustrated in Diagram 25. The flow rate depends on the required cooling power. Though usually power plants need a high flow rate of cooling water, hence the flow rate up to 18m³/s is justified. The heads mainly depend on where the plant is located, in comparison to the water intake and on the cooling system itself.

For the cooling system 2 different systems are used:

- Wet cooling
- Dry cooling

The dry cooling system works as an indirect one with air cooling. The wet cooling system is used with fresh water. The heads in a wet cooling system are approximately 5-15m and in a dry cooling system (cooling tower) 20-35m. (Hellmann, 2011, p. 155)

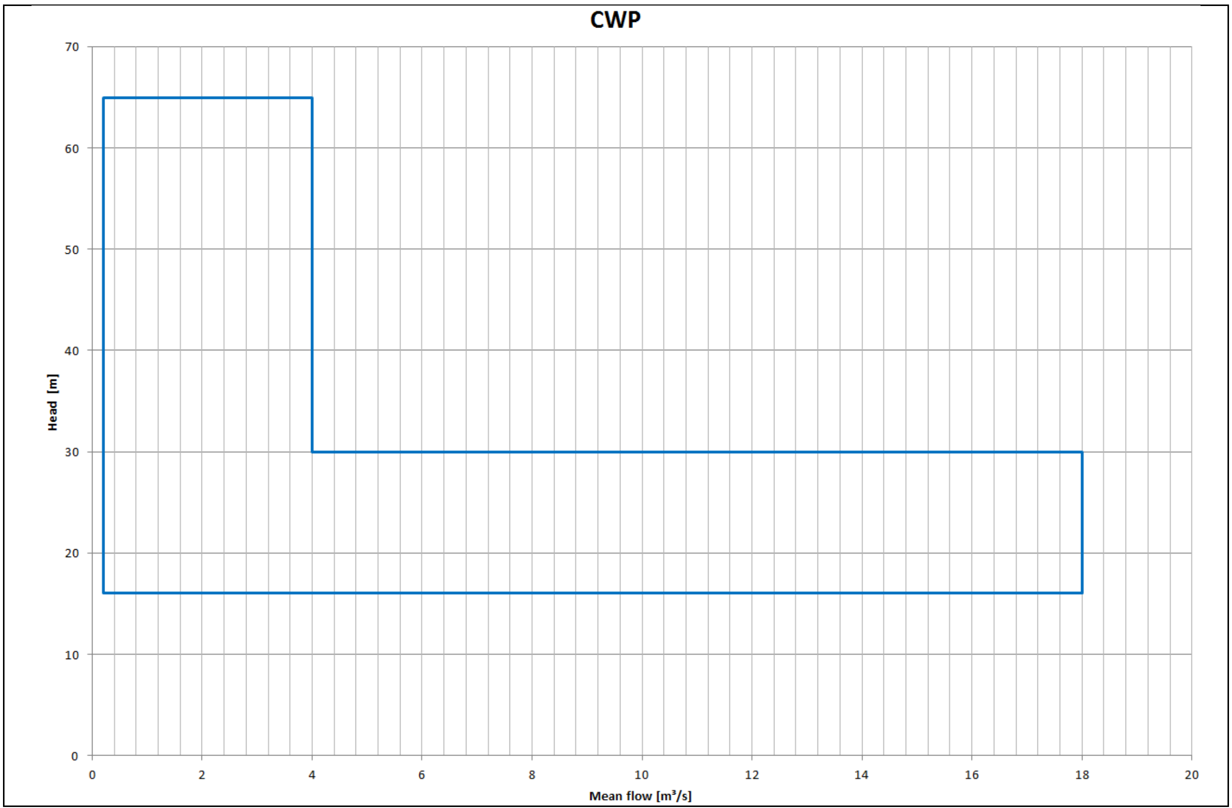


Diagram 25: CWP – Q-H area

The processes for a gas-fired- and a biomass power plant are illustrated in Figure 66 and 67. The condensate extraction pump (CEP) extracts in both processes the condensate water from the condenser and pumps it through the deaerator feed water tank. This CEP pumps are mainly double casing vertical suspended centrifugal pumps (Condensate-/ Barrel pumps). (Sulzer – CWP for coal- and oil-fired power plants, 2017)

The feed water pump (FWP) or boiler feed pump (BFP) is a multistage pump for high heads. These pumps drive the feed water from the deaerator through the heaters. (Sulzer – CWP for coal- and oil-fired power plants, 2017) The process figure for a coal-fired power plant is illustrated in appendix (Figure 85).

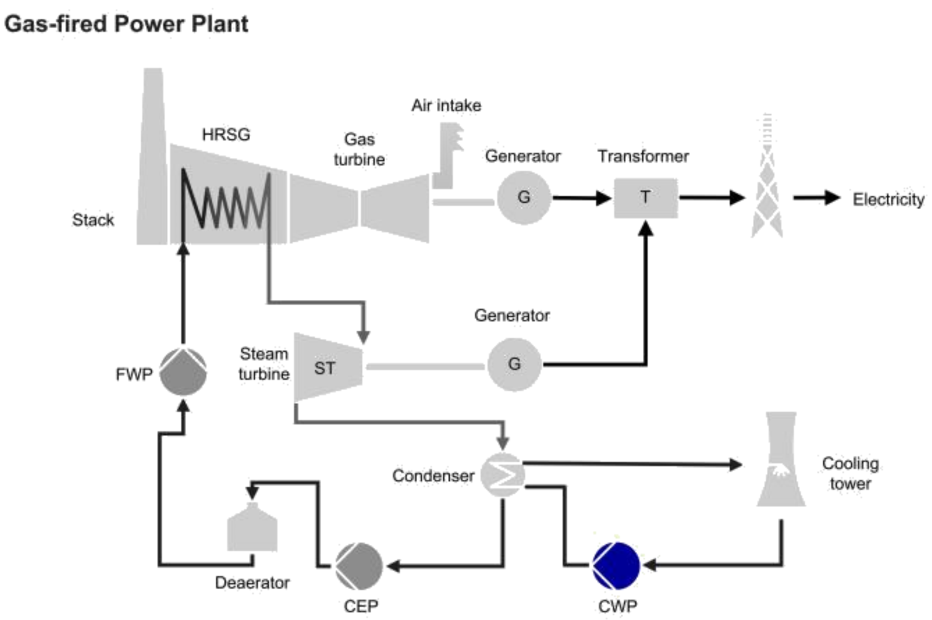


Figure 66: Cooling water pump for gas fired power plant (Sulzer – CWP for gas-fired power plants, 2017)

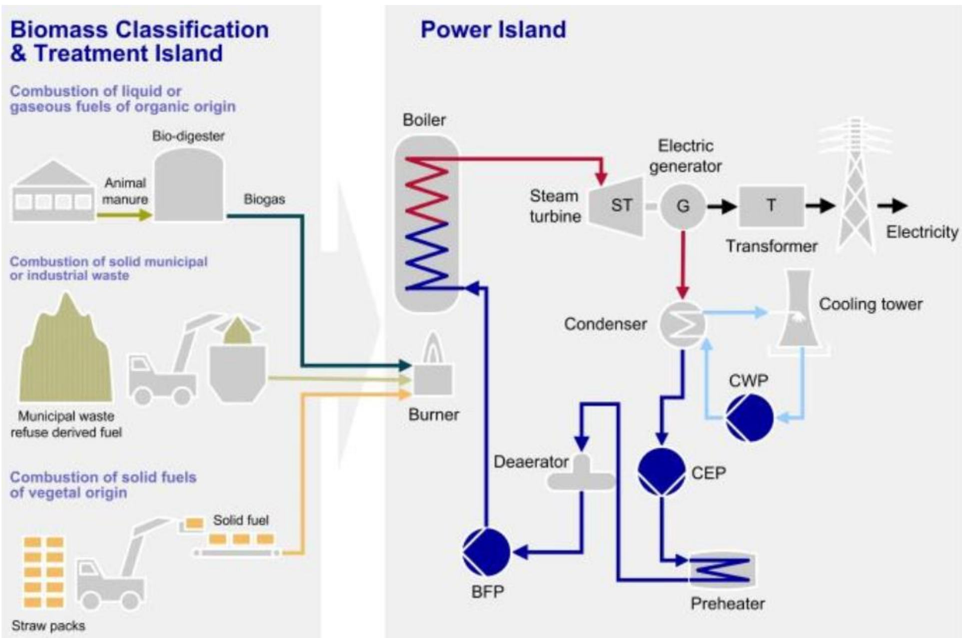


Figure 67: Biomass power plant (Sulzer – Biomass power plant, 2017)

Geothermal plant pumps take the water from a hot thermal source with a so-called production pump, which can be a special designed VLSP, and transfers it to the heat exchanger. The cold brine is injected back into the ground by a brine re-injection pump, which can be a high pressure- or axially split case pump. The hydrocarbon feed pump or in other applications called CEP, is a double casing vertically suspended centrifugal pump. The CWP is - as in other power plant applications - a VLSP for water intake, for the secondary cycle. The whole process is illustrated in Figure 68. (Sulzer – CWP for geothermal, 2017)

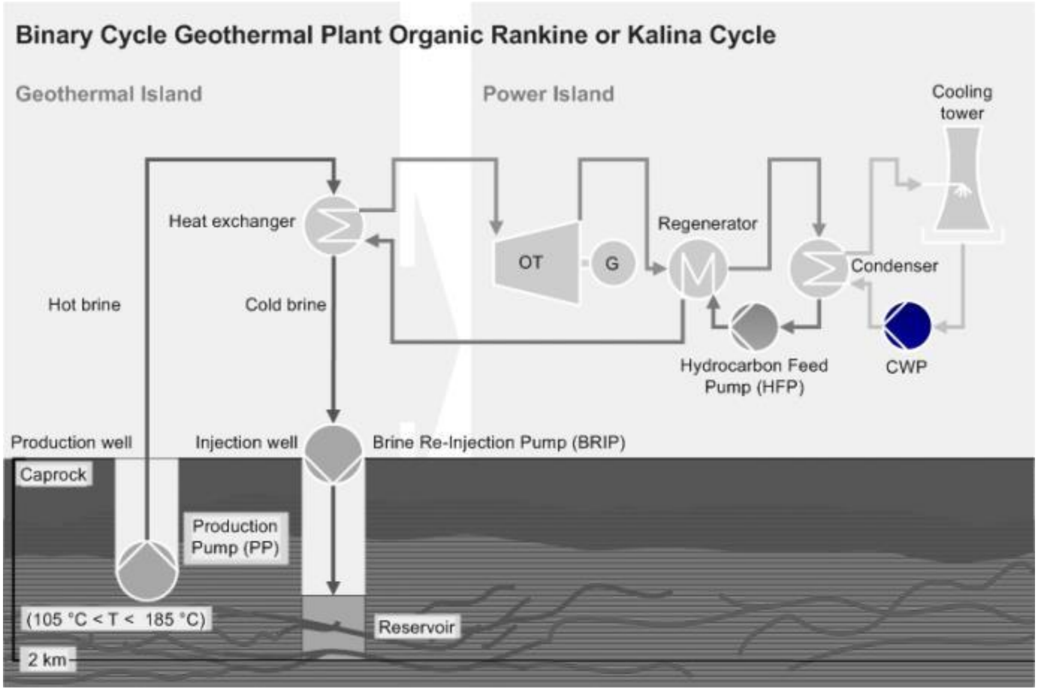


Figure 68: Geothermal plant CWP (Sulzer – CWP for geothermal, 2017)

There are different types of solar power plants, such as heliostat shown in Figure 69 and 86 in the appendix, or parabolic through (Figure 87). For the power station itself, CEP and FWP are also used. The circulation pump (CP) - Figure 69 - is a usual end-suction pump. (Sulzer – Solar power generation, 2017)

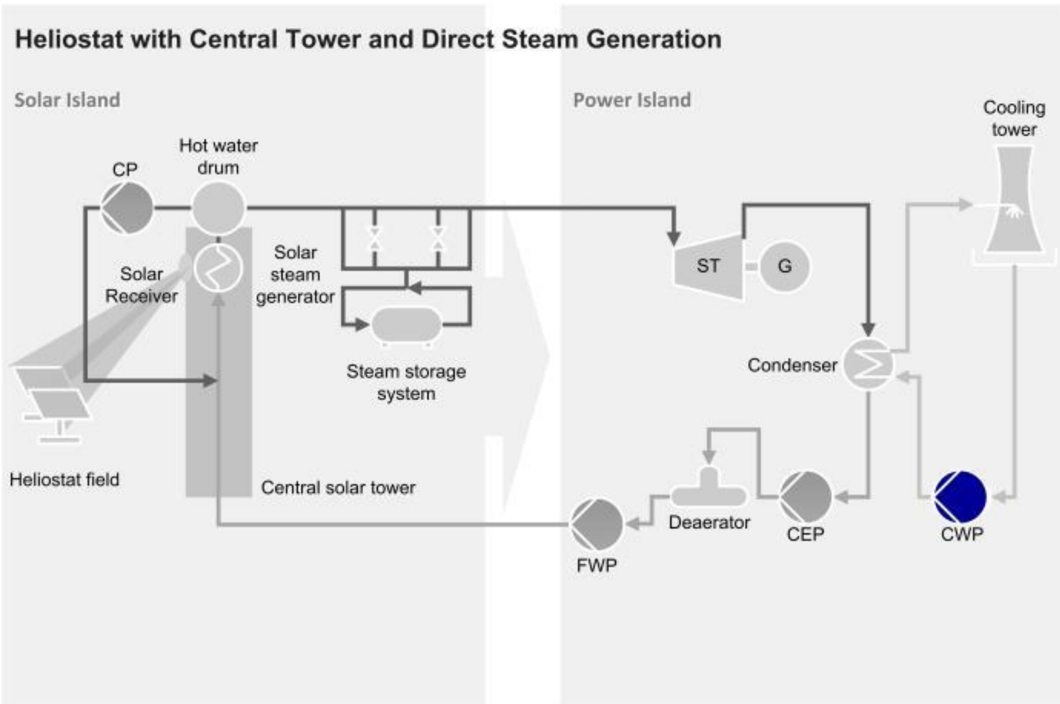


Figure 69: Heliostat with central tower (Sulzer – Solar power generation, 2017)

The hot/cold salt pump in Figure 86 for the molten salt heat storage, is a special designed VLSP for this application.

The heat transfer pump (HTF) in Figure 87 is commonly a double suction, volute-, end suction-, between bearings single stage- or overhung single stage pump. (Sulzer – Solar power generation, 2017)

4.3.5 Flue gas desulphurization

For flue gas desulphurization, a VLSP is also used as a water intake pump for seawater. The flue gas desulphurization is used in coal-fired power plants, waste incineration plants and others, which burden the environment with flue gas. (KSB – Rohrgehäusepumpen von KSB, 2017)

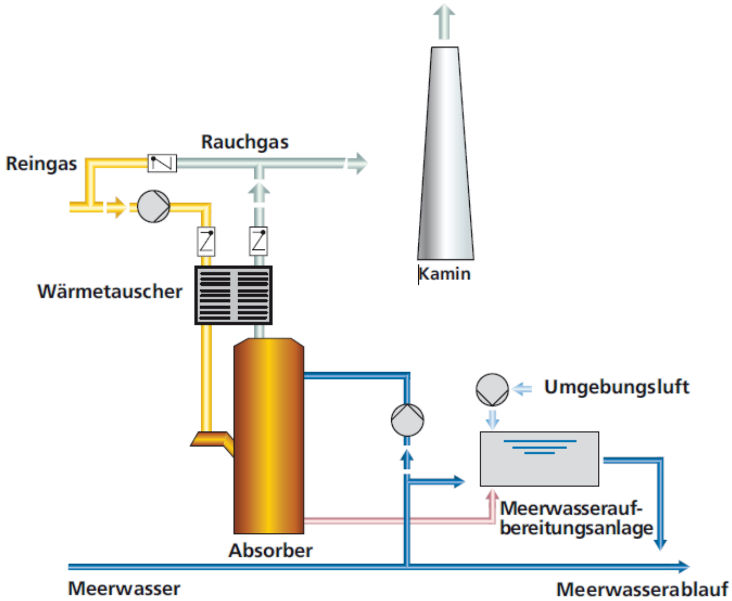


Figure 70: Flue gas desulphurization (KSB – Rohrgehäusepumpen von KSB, 2017)

The typical application area according to flow and head is illustrated in Diagram 26. The pumped water must travel through the absorber with a higher head loss, thus the heads result up to almost 40m. The process is also illustrated in Figure 70. The illustrated area was determined by selected inquired and sold pumps with this application over a certain time-period.

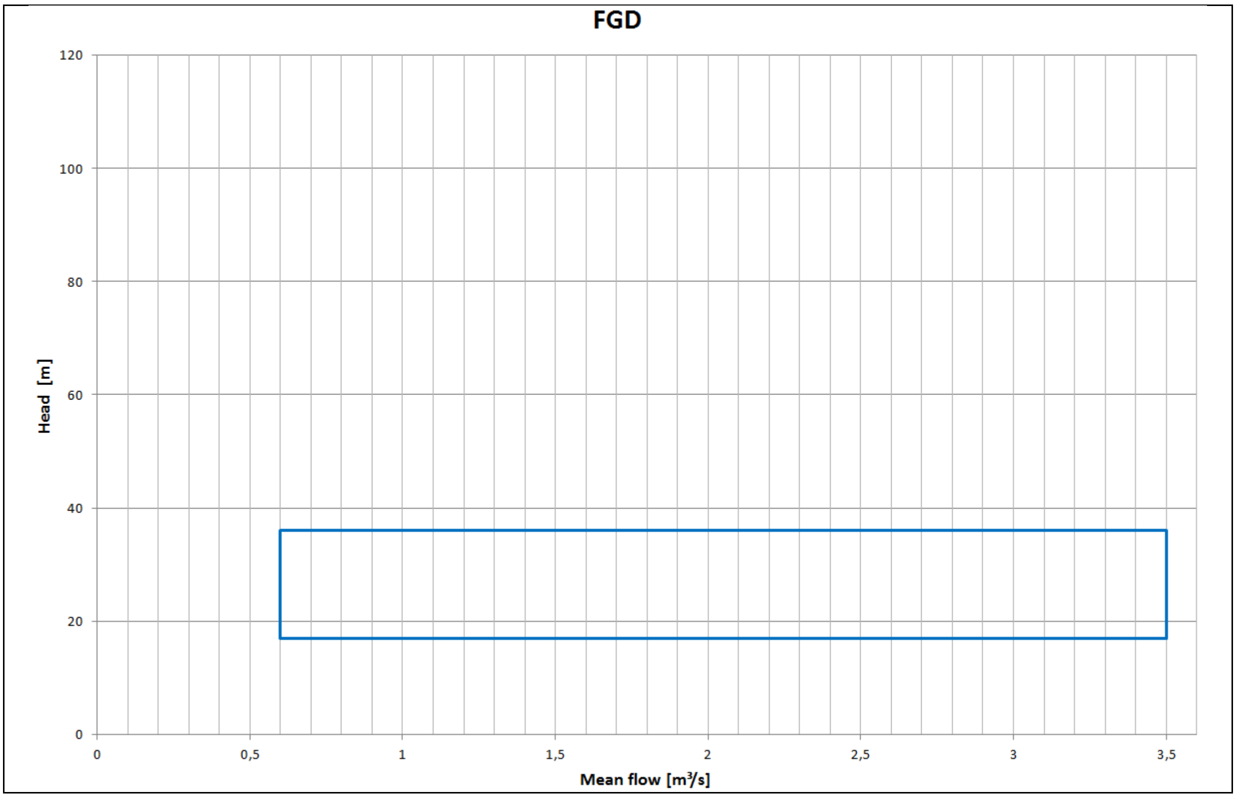


Diagram 26: Flue gas desulphurization Q-H area

The above described applications dedicate the different market potentials of the VLSP. The product will be focused for water and seawater applications, which are the most promising, as the market analysis shows as well. VLSP's will be mainly used for the following applications:

- Water supply
- Irrigation
- Flood control
- Desalination
- Industrial water
- Water intake for cooling water
- Water intake for flue gas desulphurization

5 Competitor analysis

The competitor analysis carves out the hydraulic mappings and the different available designs of the competitors.

The internal market analysis, as already mentioned, depicts that for standard sizes and usual designs the VLSP manufacturers distinguish with low price and low lead time. To achieve that, many pump manufacturers have a low-cost manufacturing facility and have standardized their design and the hydraulic sizes.

The standard hydraulic mappings and the standard series of the competitors KSB, Sulzer and Flowserve were analysed in detail. The standard products and hydraulic sizes of KSB and Sulzer are described in the following Chapters 5.1 and 5.2. The standard mappings of Flowserve are shown in appendix 5. The competitors have already hydraulically standardized a bigger range, up to higher heads and flow rates.

To compare the competitors different design options for VLSP's and to see whether a design is standard or optional, three VLSP manufacturers were analysed - Flowserve, Sulzer and KSB. The result is illustrated in a matrix in appendix 5

The "X" inside the matrix depicts a standard design and if there is "(optional)" added it is an optionally design feature. If the table is empty, no specific information was gathered for this design feature, or it is not available as a design option. All this information was collected with publicly available information from competitors' product folders, or companies' homepages.

The main message of the competitor analysis is that manufacturers of VLSP's produce all common designs that are required. The way how different parts are manufactured is distinct between the competitors e.g. if a discharge head is fabricated or casted. Another distinct design is how to execute the design e.g. the enclosed line shaft and the sealing of it. Hence, a company can only distinguish from the competitors with a high technological product. This especially applies for the design of adjustable blades, multistage and Pull-out designs, because only a few competitors can handle these. Also, API or NFPA conform designs are highly recommended. Sometimes a customer requires just the API conform design without all the documentation and the elaborate certifying process. Since the certifying process is the most challenging issue in having an API design, it is important to provide at least the API conform design.

Since the focus of this master thesis was not on the API market, the API conform designs of the competitors were not especially screened. The analysis, which is needed to have a VLSP with an API conform design, was executed and is mentioned in Chapter 3.5.15 (page 55).

The NFPA is a fire protection association and establishes the norm, where the regulations for firefighting pumps are included as well.

The selected standard design or optional parts, which were determined for the VLSP within this thesis, are described in Chapter 8.2.

The sales department has gathered, due to their comprehensive market knowledge, which amount of VLSP's the competitors manufacture per year worldwide. The Andritz AG is included in others. This is illustrated in Table 4.

Changsha	1000 pumps
Gaoyou	500 pumps
Wuxi	500 pumps
KSB	500 pumps
Flowserve	500 pumps
Gorman Rupp	500 pumps
Sulzer	300 pumps
Sigma	200 pumps
Others	1500 pumps
Sum	5500 pumps

Table 4: Competitor's number of pumps (Andritz, internal source)

These numbers correlate well to the yearly number of pumps sold which arise in the external market analysis.

Several other small competitors manufacture VLSP's in small amounts. Due to large amounts of competitors, the competition is rather fierce.

5.1 KSB

KSB has 5 different series for VLSP's, where 2 series are standardized products for smaller VLSP's. These are the SNW and the PNW series. The other series (SEZ, PHZ, and PNZ) have different additional design options, such as pull-out option, which can also be seen in the comparison of the different series in appendix 5. The hydraulic mapping of each pump series is illustrated in Diagram 27. The SEZ, PHZ, PNZ are designed for higher flow rates in comparison to the SNW and PNW series. None of these vertical pumps are available in multistage design.

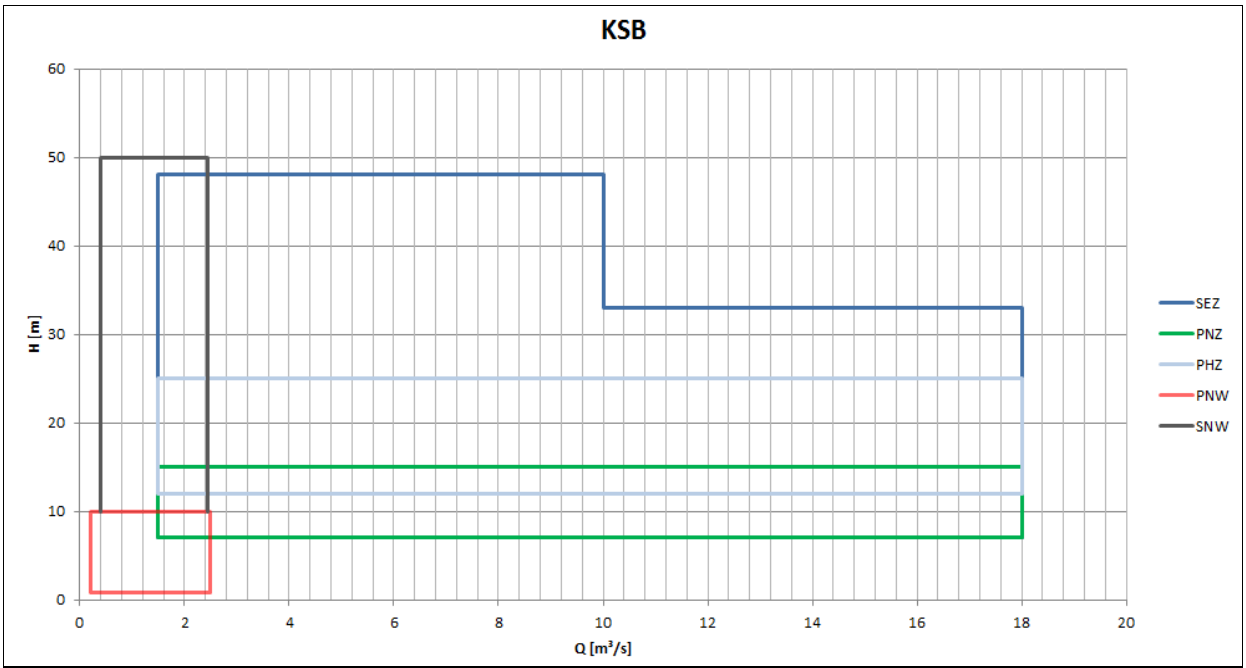


Diagram 27: KSB VLSP series

For example, the SEZ is a VLSP with a mixed flow impeller. This series has available in addition to the discharge configurations, above- and below floor, the between the floor option. This series is possible as a pull-out design.

The PHZ series has a mixed flow impeller with manually adjustable blades and as option hydraulic adjustable blades. This series is mainly used for CWP purposes and seawater intake for desalination plants.

The PNZ series is a VLSP with an axial propeller and is equipped with manually adjustable blades and optionally with hydraulically adjustable blades. This series is also mainly used for CWP purposes and seawater intake for desalination plants.

The SNW and the PNW are described in more detail, because for those series also the different sizes and detailed mappings are available.

5.1.1 KSB-SNW

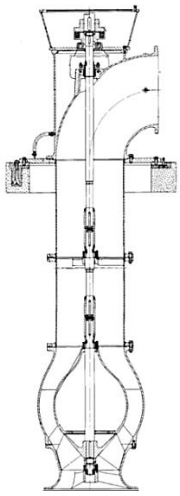


Figure 71: SNW-series (KSB – SNW, 2000, p. 4)

The SNW series is a VLSP with a mixed flow hydraulic up to 50m head and 2m³/s flow rate. The impeller is either an open or closed design. The basic design is shown in Figure 71. This series is just available as single stage pump and without pull-out design. The series is mainly used for water applications and is not API conform.

The hydraulic mapping of the SNW series is illustrated in Diagram 28. The blue pump characteristic curves are operating with 1450rpm speed, the green with 980rpm or 960 rpm speed and the orange are operating with 725rpm speed.

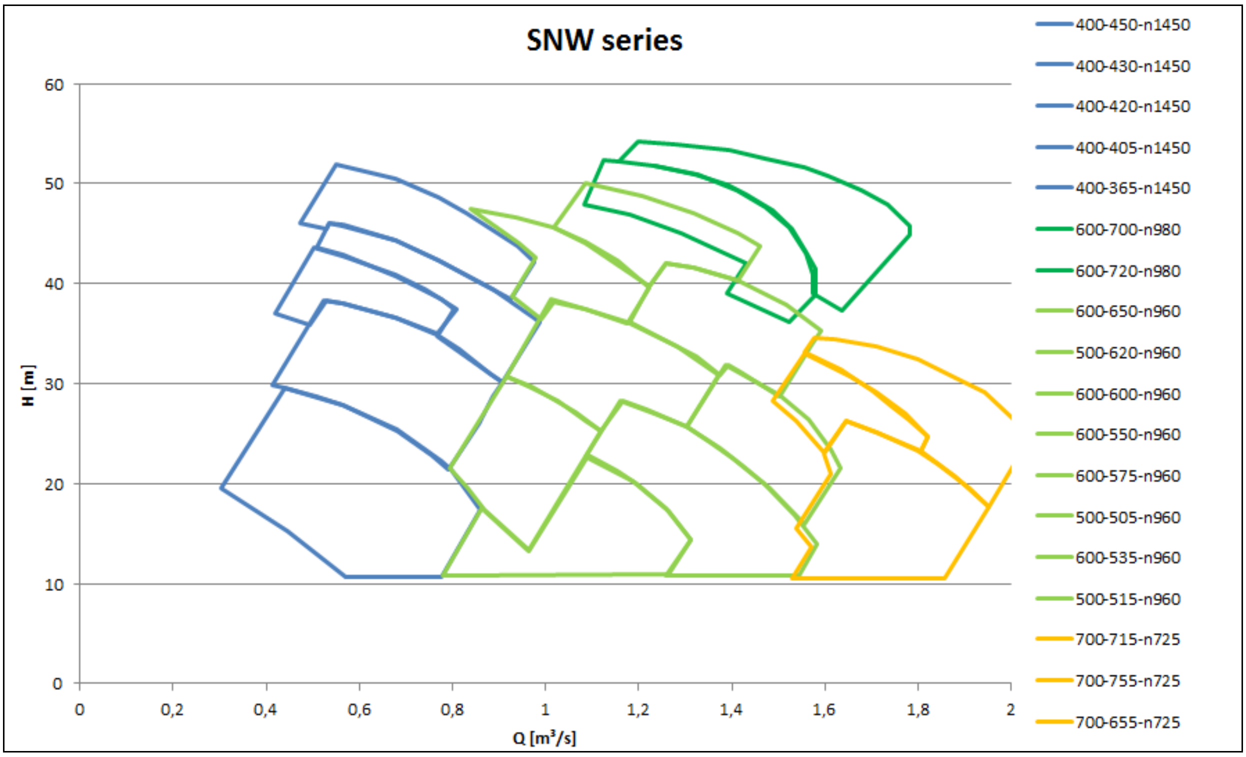


Diagram 28: Mapping SNW series (KSB – SNW, 2000, p. 2) – (source modified)

For this pump series, standardized dimensions are available. The first number in the specification (mentioned in the legend) is the column pipe diameter and the second number is the impeller pressure diameter.

The pump performance chart is resulting by trimming the impeller. This is illustrated as an example in Diagram 29, where the numbers 1 to 4 represent a different diameter.

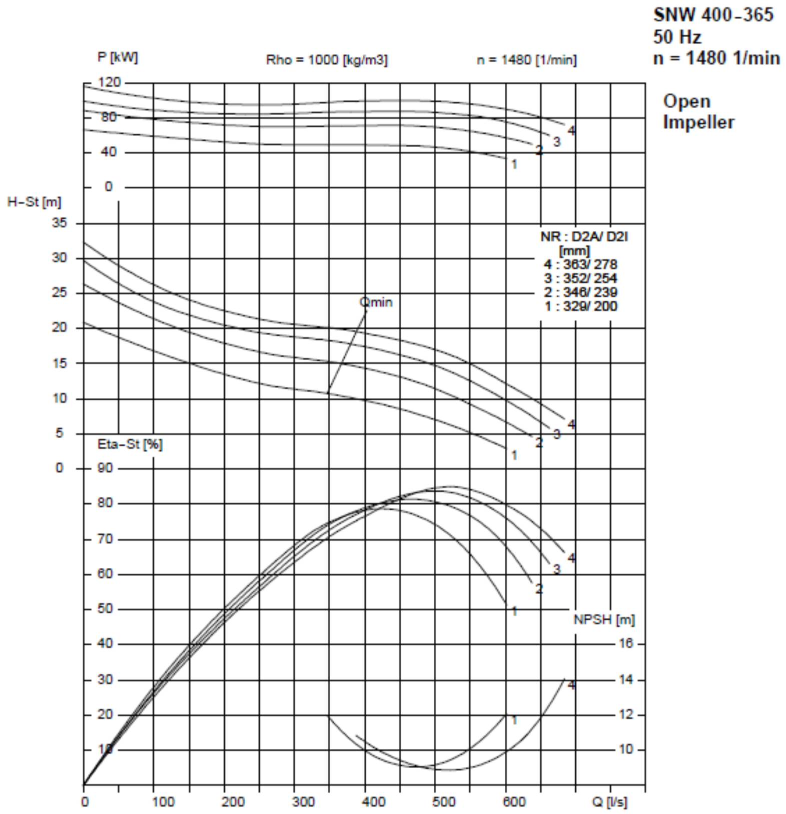


Diagram 29: Pump characteristic curve SNW 400-365 (KSB – SNW, 2000, p. 21)

5.1.2 KSB-PNW

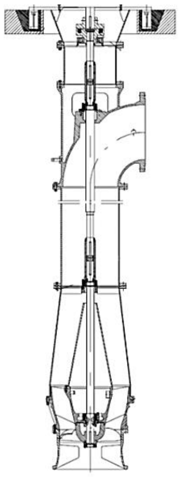


Figure 72: PNW-series (KSB – PNW, 1999, p. 4)

The PNW series is a VLSP with an axial flow hydraulic up to 14m head and 2,5m³/s flow rate. It is mainly used for water applications and is not API conform. The basic design is shown in Figure 72.

The hydraulic mapping of the PNW series is illustrated in Diagram 30. The blue pump characteristic curves are operating with 1450 rpm speed, the green with 960 rpm, the orange with 725rpm and lastly, the pink ones with 580rpm.

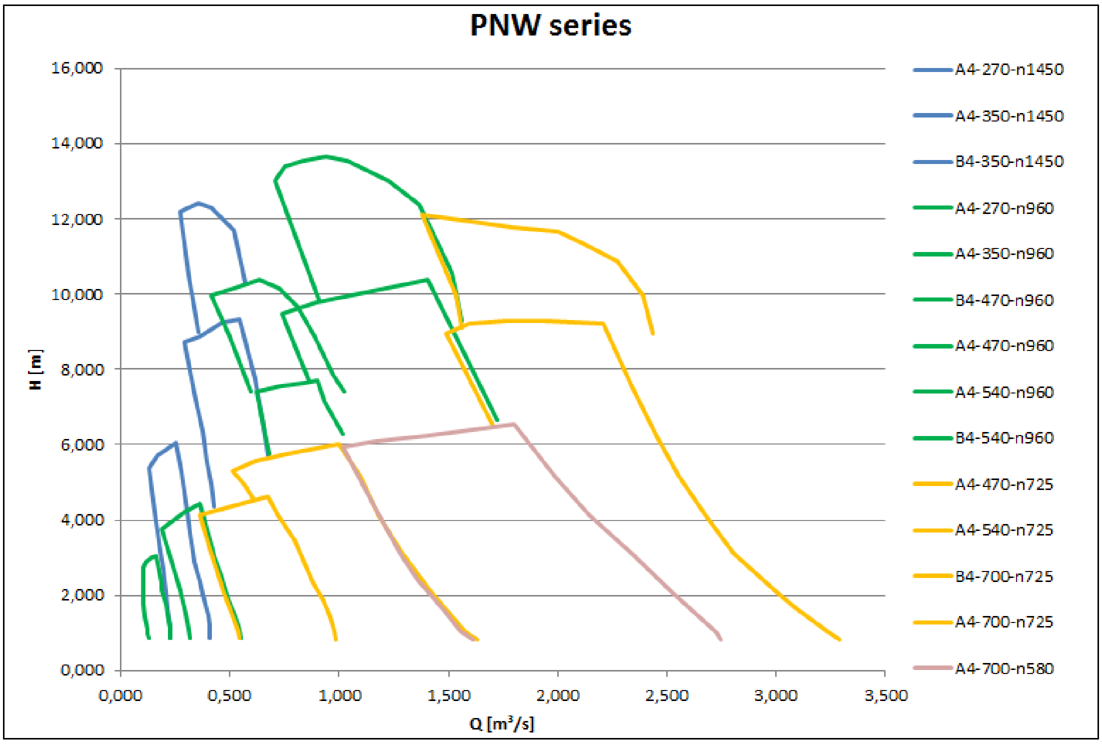


Diagram 30: Mapping PNW series (KSB – PNW, 1999, p. 2) – (source modified)

The specification A or B indicates the propeller type and the number describes the impeller diameter. The pump performance charts are the result of manually adjustable blades. An example of this is given in Diagram 31, where the numbers 1 to 7 next to the curves represent a certain angle, which is also listed in the picture.

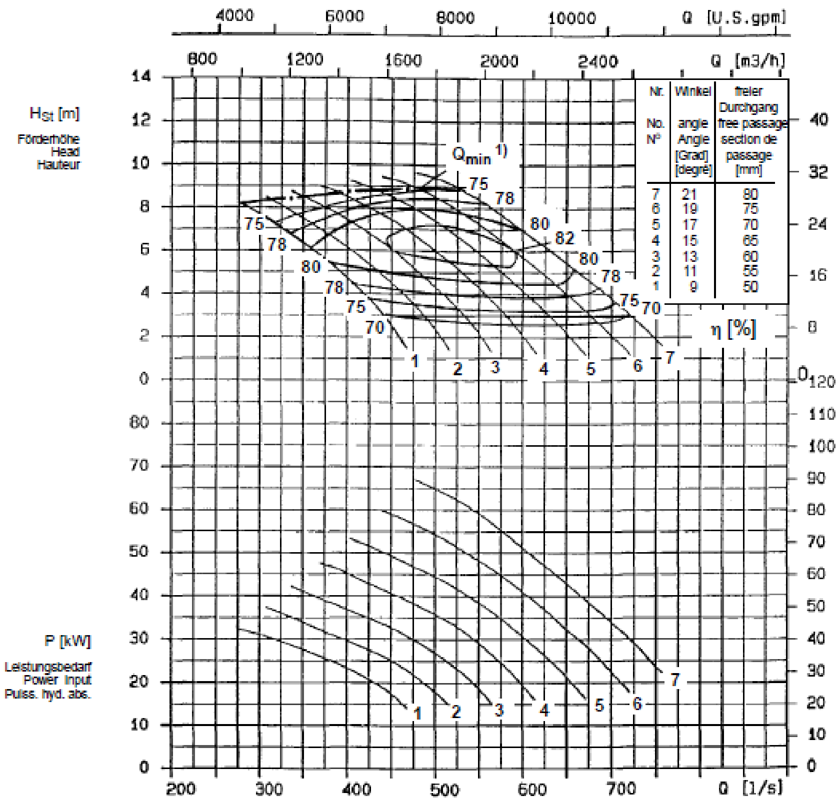


Diagram 31: Pump characteristic curve PNW A4 400-350 (KSB – PNW, 1999, p. 24)

5.2 Sulzer

Sulzer has 4 different VLSP series, the SJT, SJM and the SJP. The hydraulic mappings of these 4 types are illustrated in Diagram 32. These types can be upgraded to fulfil the API.

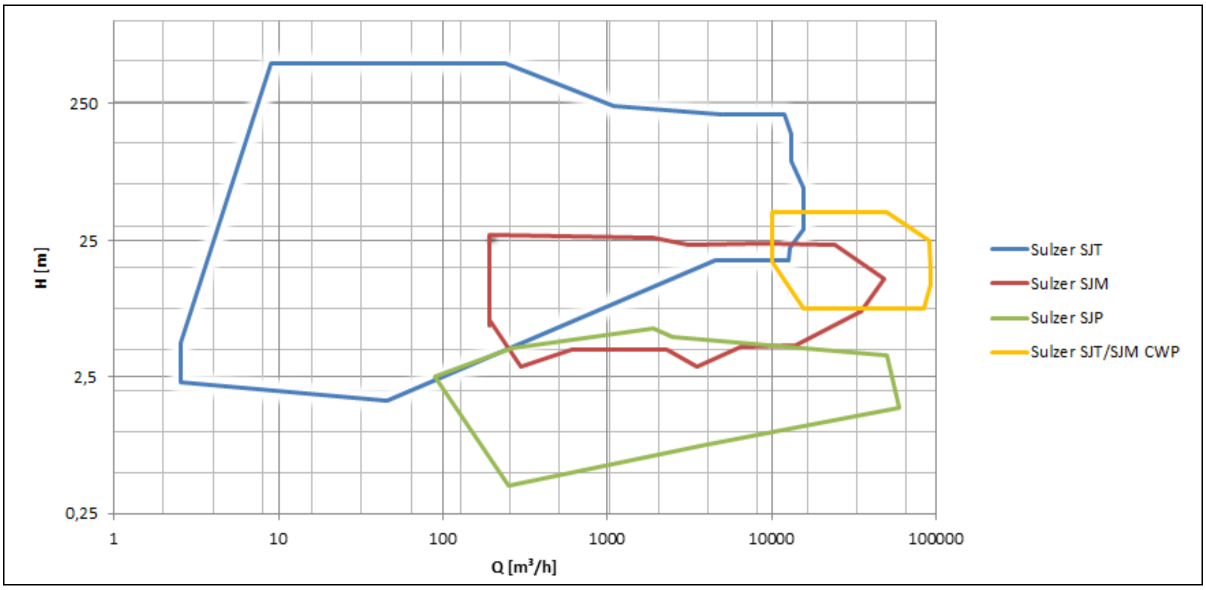


Diagram 32: Sulzer VLSP series (Sulzer – sales presentation) – (source modified)

5.2.1 Sulzer-SJT

The SJT is a radial/mixed flow impeller. The specific speed nq is between 35 and 110. The SJT is available as multistage configuration and the pull-out type as an option. The flow rate is up to 4,3m³/s and the head is up to 110m per stage, which is illustrated in Diagram 33. The bowl diameter design is up to 3000mm. This type is used for different water and wastewater applications. The characteristic curve of each pump size is created by trimming the impeller diameter, which is illustrated as an example in Diagram 34. The SJT can be supplied with fulfilling the API requirements as well, this series is then called JTS.

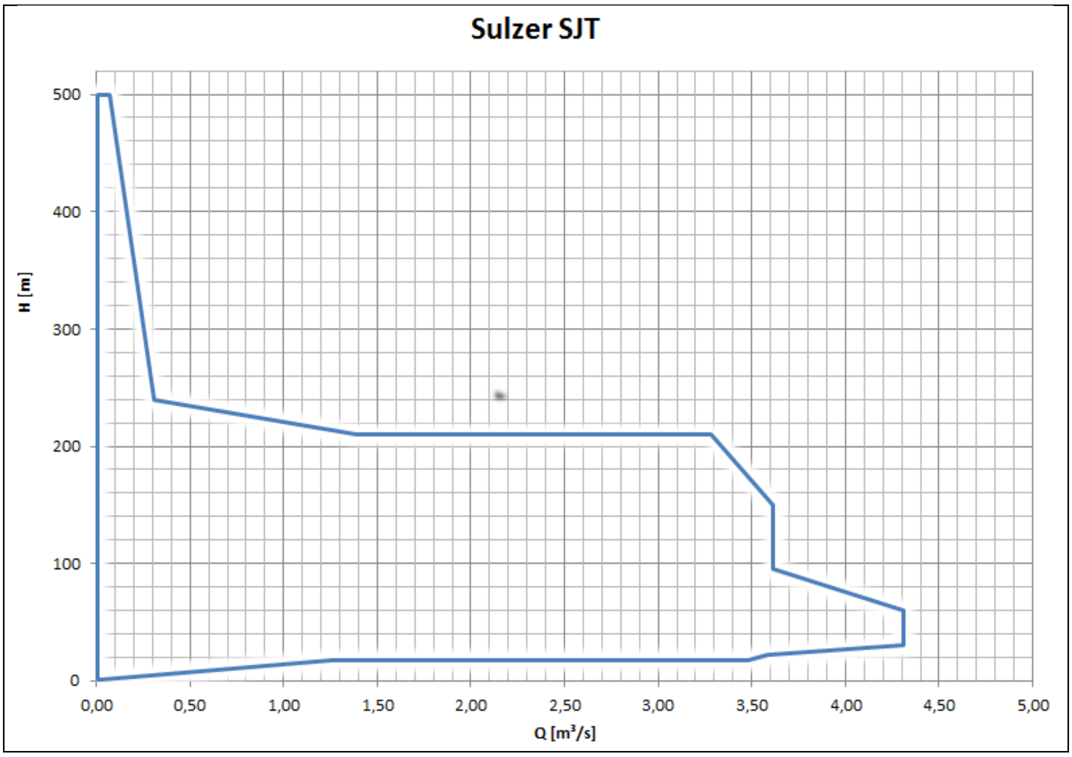


Diagram 33: Sulzer SJT (Sulzer SJT, 2008, p. 6) – (source modified)

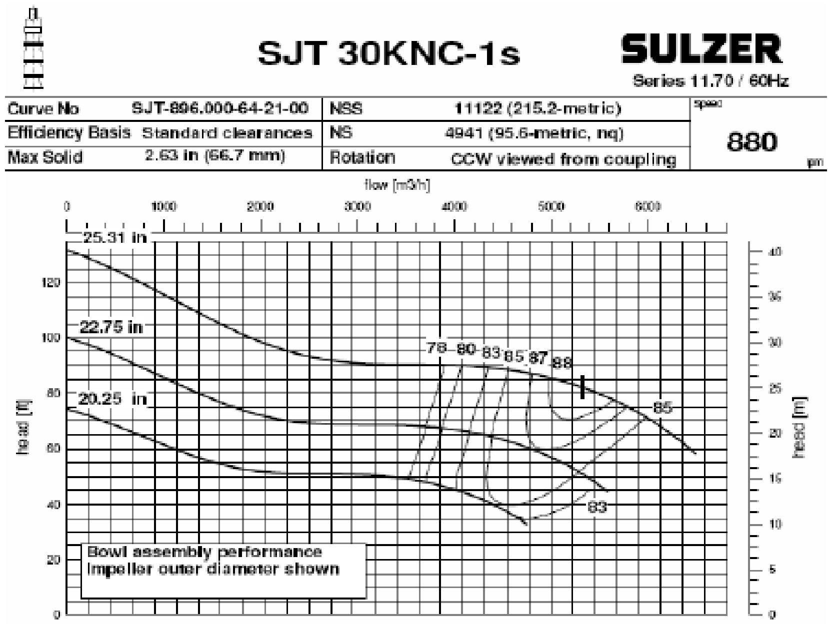


Diagram 34: SJT example pump curve (Sulzer – sales presentation, p. 9)

5.2.2 Sulzer-SJM

The SJM is a vertical mixed flow pump. The specific speed range is from 113 to 161. This pump is mainly used for water and wastewater applications. The flow rate is up to 13 m^3/s and head is up to 25m per stage. The bowl diameter design is up to 2390mm. The SJM is available as multistage configuration (max. 2 stages) and the pull-out type as an option.

The hydraulic mapping of the SJM is shown in red in Diagram 35. The characteristic curve of the SJM is enlarged by trimming the outflow angle of the impeller. An example of this is given in Diagram 36.

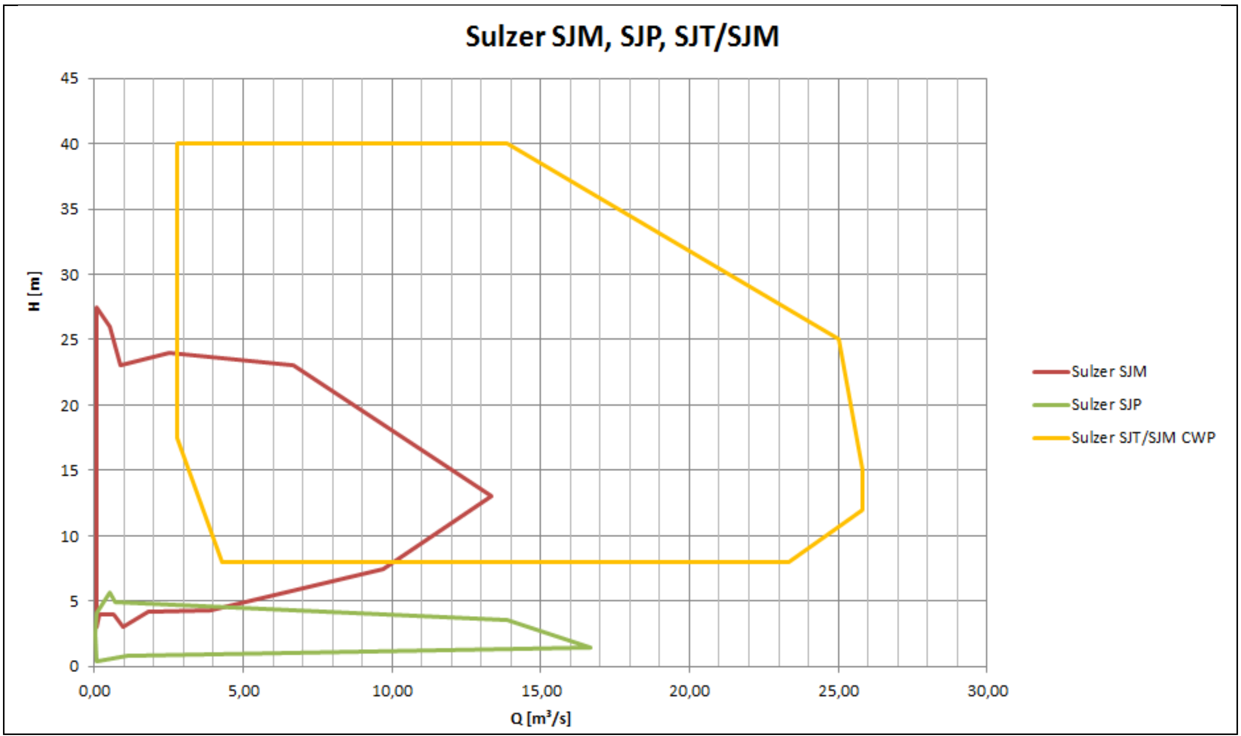


Diagram 35: Sulzer SJM, SJP, SJT/SJM series (Sulzer – sales presentation) – (source modified)

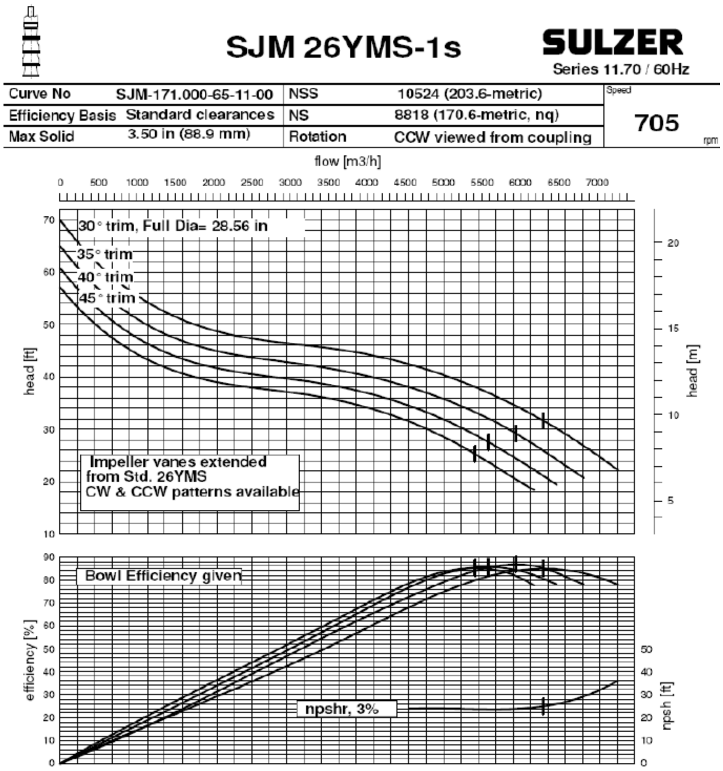


Diagram 36: SJM example pump curve (Sulzer – sales presentation, p. 15)

5.2.3 Sulzer-SJT/SJM CWP

The SJT/SJM concerns the hydraulical mapping, custom-made for the application as a cooling water pump in power plants. This hydraulic mapping is illustrated in Diagram 35. The main range, which also almost fits to the internal inquiries for this application (Diagram 25), is up to 40m in head and up to 26m³/s in flow rate. The impeller diameters are designed up to 1800mm. This series is optionally a pull-out design.

5.2.4 Sulzer-SJP

The SJP is a vertical propeller pump. The specific speed of this series is up to nq280. This series is mainly used for different water applications. With manually blade pitch adjustment, the characteristic pump curve is enlarged, which is given as an example in Diagram 37. The capacity of the type is up to 17m^3/s in flow rate and 6m in head. The maximum pump size is 2120mm. The hydraulic mapping is shown in Diagram 35.

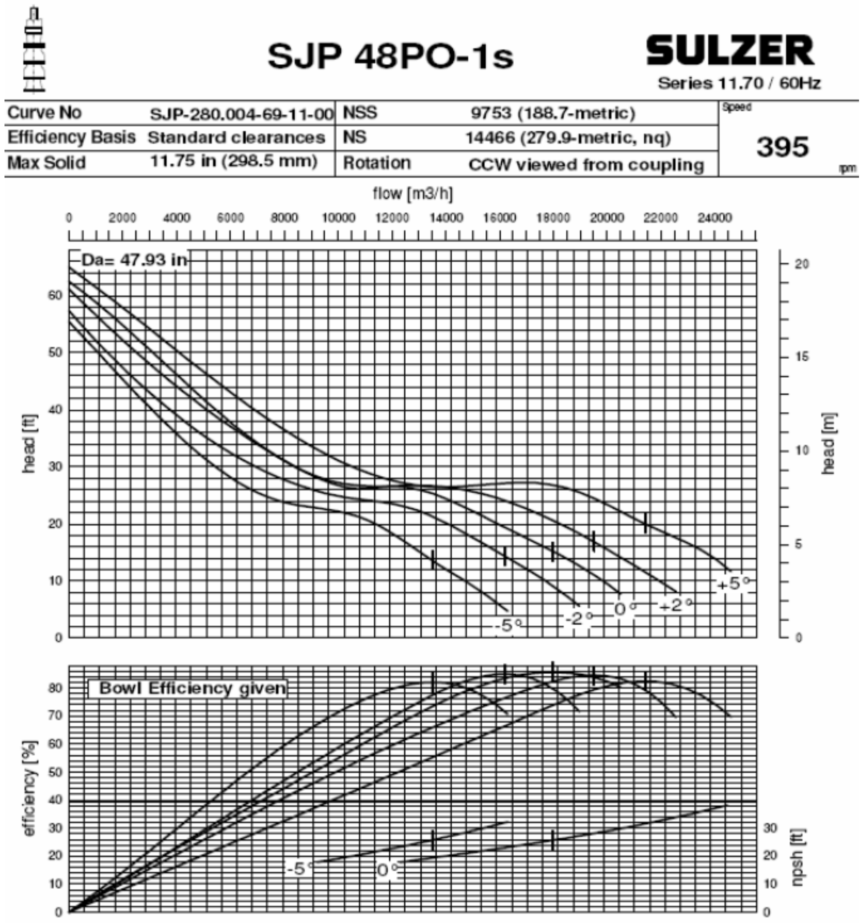


Diagram 37: SJP example pump curve (Sulzer – sales presentation, p. 21)

6 Cost analysis

To get a clear picture which costs on project basis consumes the most and how cost intense different assemblies of a VLSP are, a cost analysis for different realized projects was conducted. One project was picked as an example and the following diagrams are according to this project. This chosen project fulfils and represents the below mentioned constraints. The cost per assembly as percentage of the overall production costs is shown in Diagram 38. The ABC analysis shall examine which assemblies consume most of the production and material costs, and to know where the biggest cost saving potential will be.

The constraints for the analysed and figured costs are:

- No pull-out type
- No adjustable hydraulic
- Single stage
- Materials: impeller – stainless steel or duplex
- Shaft – stainless steel
- Other parts – carbon steel

Pull-out type pumps are more expensive because of the elaborate design and manufacturing. The pumps with adjustable hydraulics would not be comparable with normal fixed blades according to costs and the additional required parts.

The specifications of the selected project are listed in table 5. All the costs below are costs at the subsidiary.

Project X	
Q [m³/s]	3,06
H [m]	46
P_{motor} [kW]	1800
Pull-out	no
Adjustable blades	no
Materials	
Impeller	Duplex
Pressure casing	Cast iron
Shaft	Stainless steel
Other parts	Carbon steel

Table 5: Project specifications

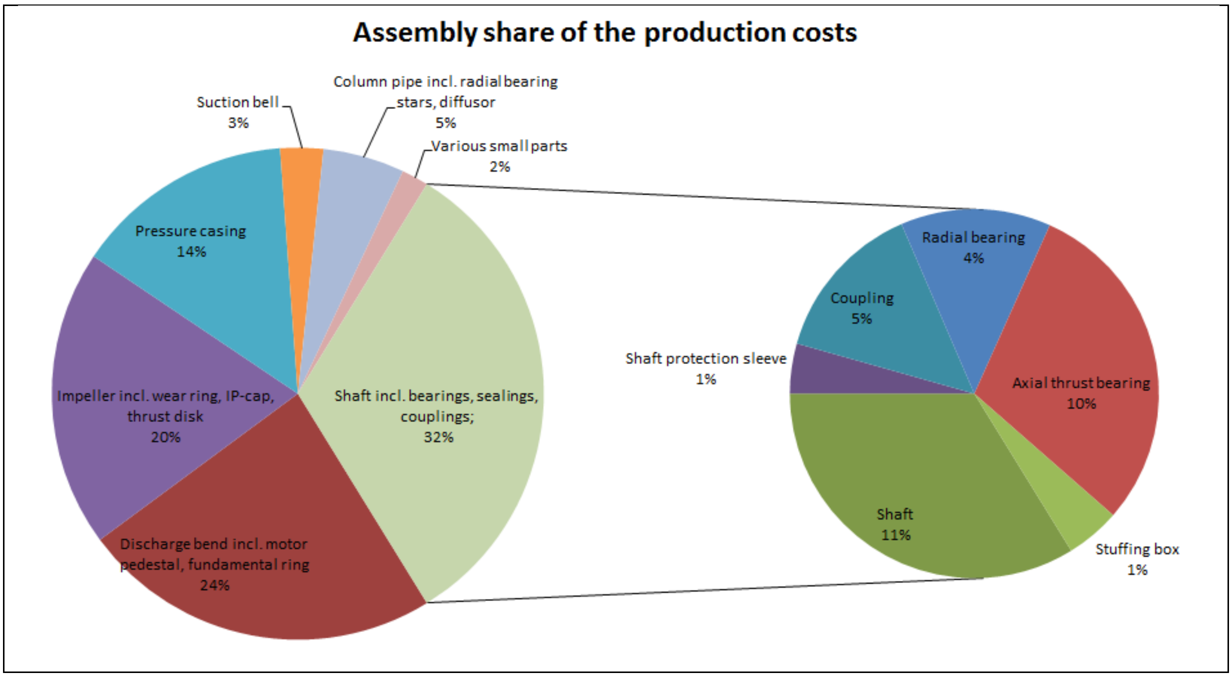


Diagram 38: Assembly share of the production costs

The costs illustrated in Diagram 38 above, represents the production costs per piece. The costs for patterns are not considered in this figure. These are mentioned in the total project costs in Diagram 41.

Actually, the percentages on the production costs vary from project to project, but due to the analysis of several projects, the investigation was that the percentage fluctuation is only ± 4% per assembly for the same specifications as mentioned in Table 5.

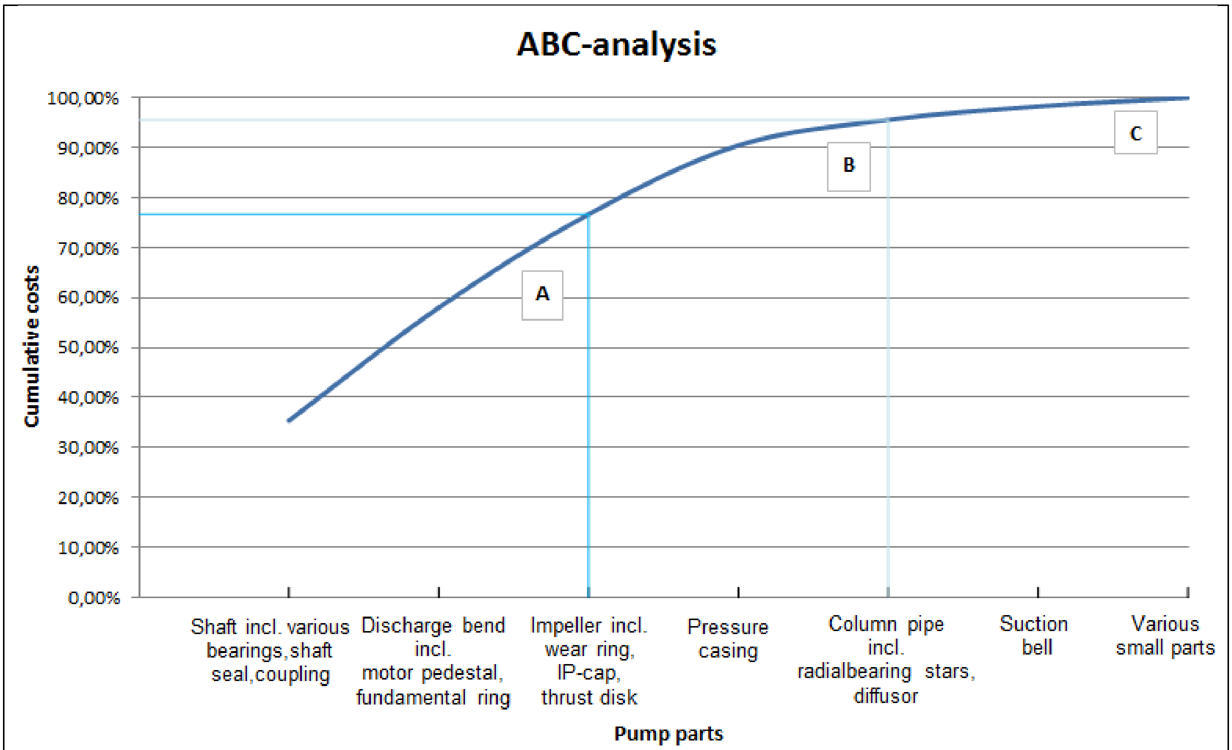


Diagram 39: ABC – analysis

An additional cost saving potential can be investigated when analysing the weight per assembly. This is illustrated with the chart below in Diagram 40.

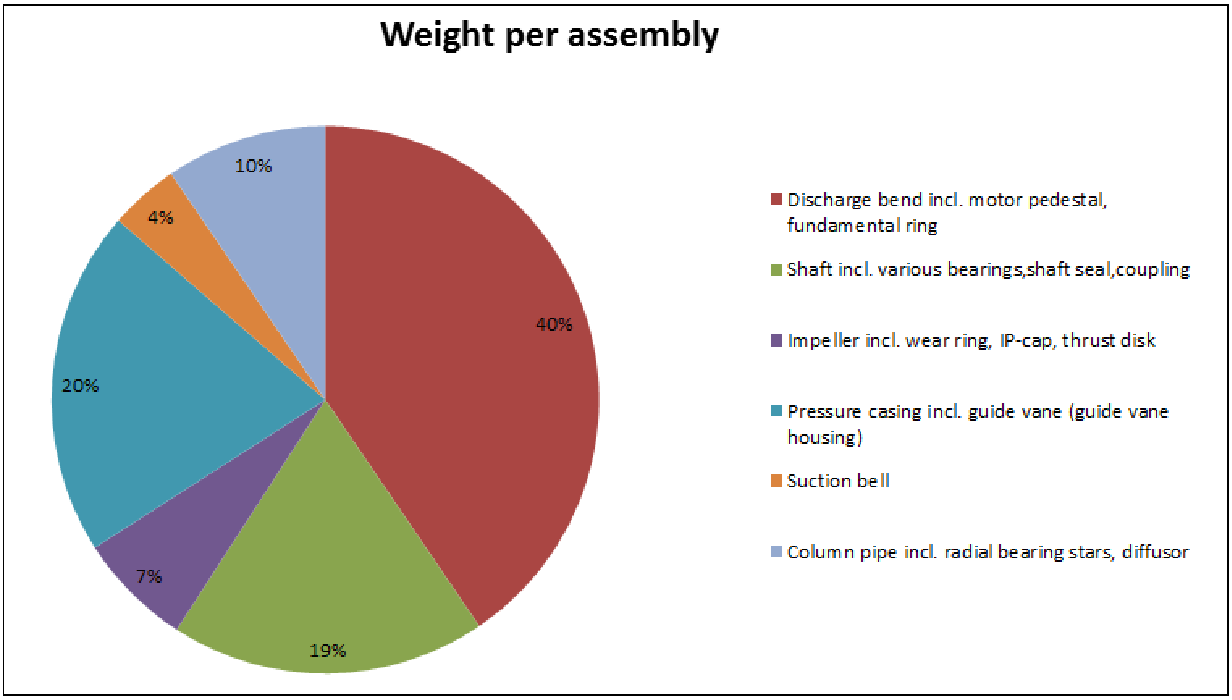


Diagram 40: Weight per assembly

To sum up the results of Diagrams 38-40, the most cost consuming assemblies are the shaft with couplings and bearings, the discharge bend, and the impeller assembly. This contributes to almost 80% of the production costs per piece, according to the ABC analysis in Diagram 39. This can be justified with the weight of the discharge elbow with a share of 40% of the total weight and the shaft assembly with 19% of the weight. The assemblies in category B, the pressure casing and the column pipe assembly, cost approximately 21,6%. The suction bell and the various small parts are not considered to be high cost contributing only 4,4% of the overall production cost.

Taking a closer look at the shaft assembly (Diagram 38) arises that the shaft itself consumes the highest cost. The whole axial thrust bearing assembly has the second highest cost with 10% of total production costs.

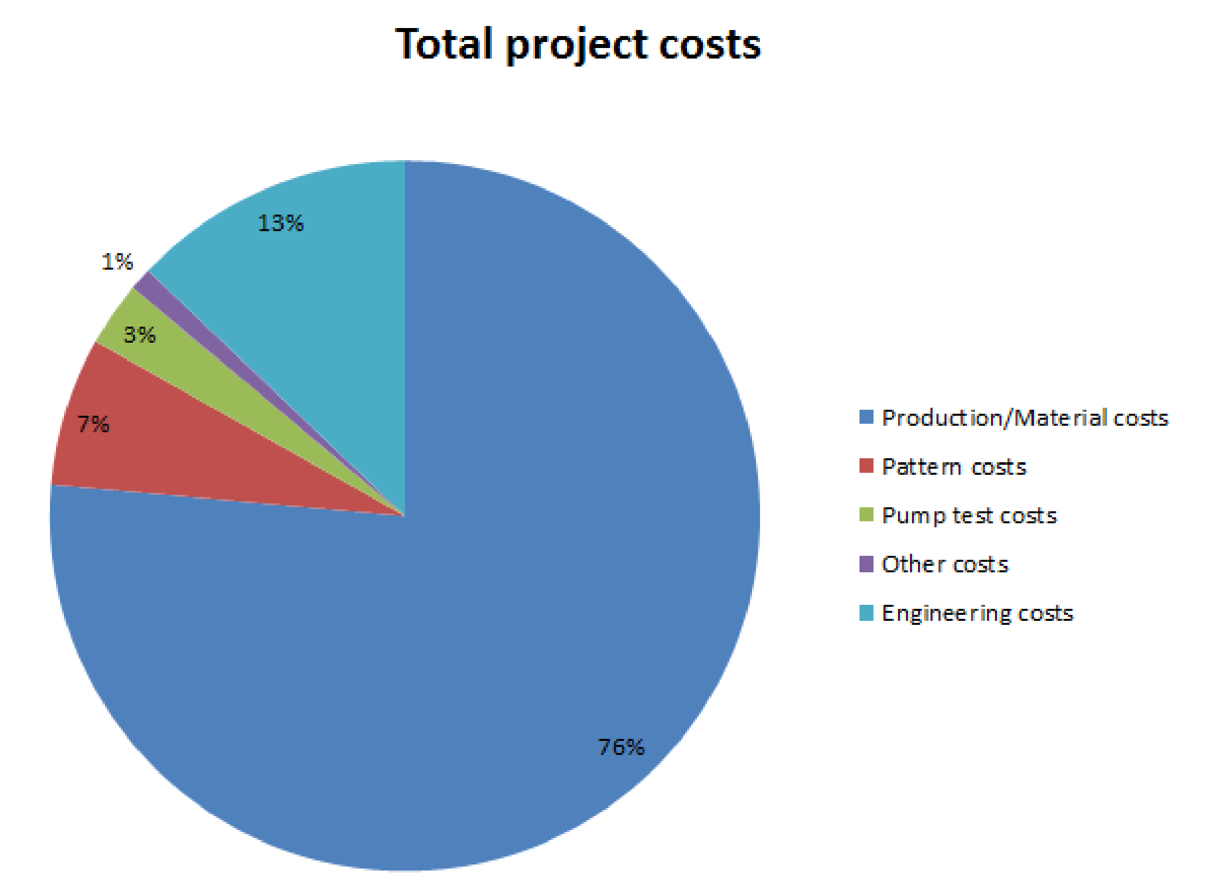


Diagram 41: Total project costs

The total project costs, as seen in Diagram 41, are split up in production and material costs, pattern-, pump test-, engineering and other costs. The production and material costs are naturally the biggest portion of a project. The details of the production and material costs on basis of different assemblies were already illustrated in Diagram 38. Other major cost factors in a project are the engineering costs with 13% and the pattern costs with 7% of the total project costs.

6.1 Cost saving potential

To examine the cost saving potential of the project, the last chapter will constitute as the basis. As already illustrated in Diagram 41, the main part of the total project costs are the production and material costs.

The discharge bend takes 24% of the production costs and consumes 40% of the total pump weight. Thus, there must be a cost saving potential but considering the resonant frequency of the pump assembly, the discharge bend is a determinant part. Hence, the high weight is needed that the discharge bend is stiff enough and that there is at least a 20% margin to the operating speed, to be sure that resonance will not occur at any point. None, or only a marginal cost saving will be possible.

At the moment, the discharge elbow itself is mostly welded in segments. A cost saving potential is possible, when using a smooth elbow which is bought at a supplier. According to internal company information this amounts to almost 20% savings of the elbow. On a project-level this accounts for a **0,35%** cost saving potential.

The shaft assembly has the highest portion of the production costs. The stuffing box takes just 1% of the costs, but currently the stuffing box is designed for each project specifically, considering using the stuffing box design of other internal pump series, would save some though not quantifiable costs, due to the series effect of these other pump series. Comparing the size of this project with an almost identical size of another Andritz pump series, a potential saving of 7% for the stuffing box could be possible. Thus, on total project cost basis it is **0,05%.**

For the shaft coupling currently clamp type couplings are used. If using a so-called muff coupling, which is like a clamp coupling but not cut into 2 pieces and without all the holes, there could be a saving of 15%, according to internal company information. This is on project basis, with a share of the coupling of 5% of the production costs and a share of the production costs of 76% of the total costs, then **0,57%.**

If for the radial bearing and for the other shaft assembly parts higher amounts are ordered, the prices may be reduced as well. By standardizing the shaft itself, there could be an additional potential possible which is not yet quantifiable.

All the casting parts, such as the impeller and the pressure casing may reduce the costs when higher amounts of the same size are casted. Thus, a standard hydraulic mapping will help that only certain sizes are casted. Alternative supply chains for casted parts are not considered within this cost saving potential. Additionally, the pattern costs account for 7% of the project costs. This can be reduced to zero when having a standard hydraulic mapping, because then the pattern costs will be investment costs and will not debit the project costs. Furthermore, trimming/adjusting the impeller to a certain degree, will help to reduce the needed patterns. Thus, **7%** cost savings are possible with these standard hydraulic sizes.

Also, the engineering costs debit the project usually rather high e.g. with 13%. Currently, this product is customer specific engineered and this accounts', according to Diagram 1, for the highest development costs. The engineering costs are mainly design-, calculation- and project management costs. The distribution, which type shares which costs, is project and size constrained. For this example, the distribution is assumed that the design and calculation costs account for 66% of the engineering costs, 70% of these costs may be saved with a standardization of the design and with a standard calculation tool. The potential 70% savings in development costs are in line with the illustrated savings in Diagram 1, when the parametric design is chosen. The remaining 30% will still be needed to adjust the product according to customer requirements, since the VLSP is still an engineered pump. The 33% account for the project management costs, which will be still debiting the project, but to a lower extent, with standardized project management procedures, such as BOM list creation.

Elbow cost saving	0,35%
Stuffing box unit cost saving	0,053%
Shaft coupling cost saving	0,57%
Pattern cost saving	7,0%
Engineering cost saving	6,5%
Potential cost savings	**14,5%**

Table 6: Cost saving potential

The different cost saving potentials are summarized in Table 6, resulting in a preliminary total cost saving potential of approximately 14,5%. This represents the potential at the company's subsidiary. The potential of a different supply chain for casted parts and design to cost was not taken into consideration in this calculation.

7 Mapping

7.1 Internal hydraulics

The company has several VLSP hydraulics, non-adjustable and adjustable, to cover all customer requirements to the best extent. These are listed below in Table 7 and 8.

For standardization, the hydraulics must fulfil several requirements, which are the following:

- Good operation range (approximately 0,65 Q_{opt} - 1,1 Q_{opt})
- None, or at least no distinct instability
- Hydraulics which are trim-able / manually adjustable to a certain extent to enlarge the pump characteristic curve to a pump performance chart
- Low NPSH basin
- High flat efficiency curve

All these specifications are not possible in one hydraulic because of physical constraints, but it shall work as a guideline. For different hydraulics, it is also important that these are tested and additionally different important values for calculations and dimensioning are measured. These are the following:

- Different NPSH - criterion such as incipient, 0% efficiency loss and 3% head loss
- Runaway speeds for shaft calculations
- Axial thrust force with and without compensation
- Trimming curves to a certain extent
- Wear ring gap variation analysis

The company has 7 fixed blade hydraulics, which are listed in Table 7, and 5 adjustable hydraulics, which are listed in Table 8. The specific speed definition ns come from the turbine design and hence is used for large scale pumps as well. This is defined with 3,65 times nq. All fixed blade hydraulics are illustrated with Ψ, η, σ values at Diagram 42. The different colours depict the different specific speeds. These colours are also assigned to each ns in Table 7. The range of each hydraulic represents the operating area. The lower boundary is set by the instability limit or the efficiency loss. The upper boundary is determined by the efficiency loss. The Diagram 42 shows clearly that the range up to ns500 is covered with the fixed blade hydraulics.

Trimming is for enlarging the pump characteristic curve to a pump performance chart. Currently, the company scales each pump size to the desired operation point and does not use the advantage of trimming. An experimental trimming test was only conducted for a few hydraulics but not to the maximum extent.

The trimming is conducted different for each hydraulic. For example, for a radial hydraulic like the ns145, the trimming can be done by trimming the outer diameter. This is also done by the competitors, which can be seen at the KSB PNW series in Diagram 31 and at Sulzer SJT series in Diagram 34.

Often also just the area between the two shrouds is trimmed at guide vane pumps to avoid instabilities at the pump curve, increase the axial thrust at part load and to reduce the momentum exchange between the main current and the current in the back of the impeller to the minimum. However, the efficiency loss is bigger when not trimming the whole surface because the effect of the impeller-friction is a function of d^5. (Gülich, 2013, p. 194)

The mixed flow hydraulics, which are the ns220, ns262, ns346,5, ns382 and ns444, are mainly trimmed with changing the angle at the outflow area. This can be seen at the Sulzer SJM series in Diagram 36. Here the efficiency decrease is modest, it only changes the optimum point of the pump characteristic curve to smaller flow rates. In the KSB Series SNW the outer diameter is trimmed as it can be seen in Diagram 29, but here a significant efficiency decrease is visible. Thus, for mixed flow hydraulics different trimming guidelines are recommended.

Axial hydraulics, as the ns486, cannot be trimmed since the outer diameter is fixed, otherwise the gap increases and the efficiency will decrease significantly. It is possible to change the angle of the outflow area by dragging the angle. A significant enlargement of the pump performance chart is not possible. To create a pump performance chart, the mixed and axial flow hydraulics are executed as manually adjustable hydraulics.

ns	nq	flow	
145	39,7	Radial	
220	60,3	Mixed	
262,2	71,8	Mixed	
346,5	94,9	Mixed	
382	104,7	Mixed	
444	121,6	Mixed	
486	133,2	Axial	

Table 7: Fixed blade hydraulics

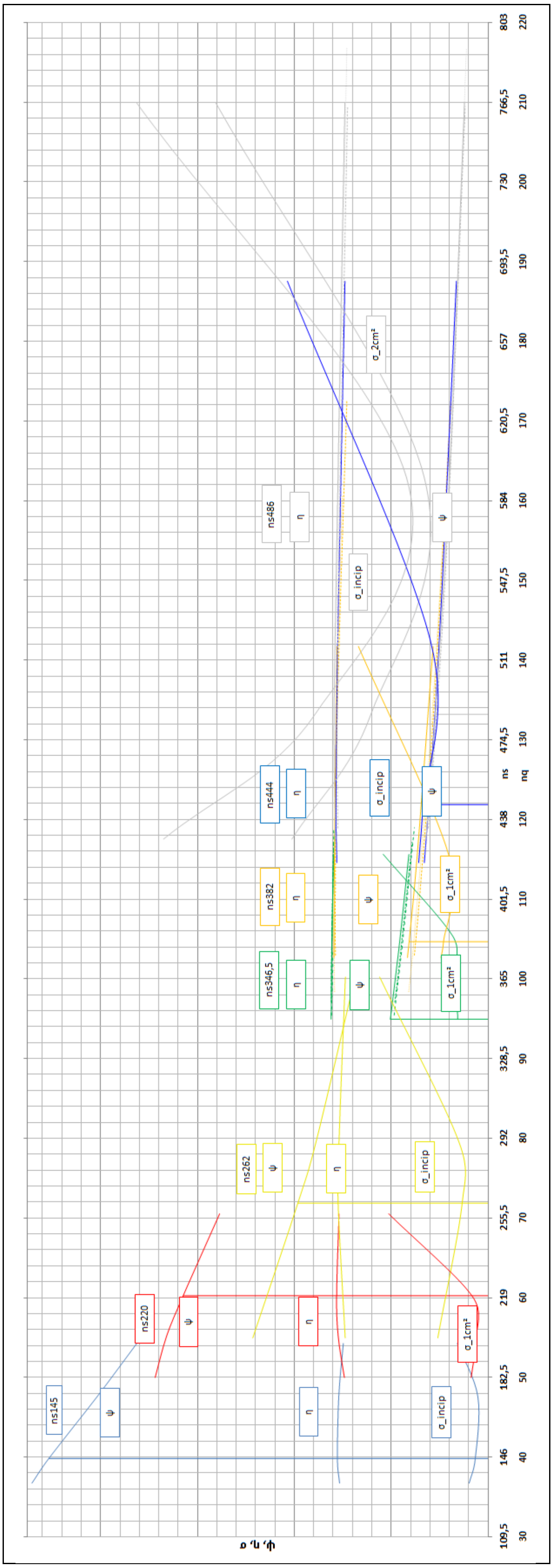


Diagram 42: Overview fixed blade hydraulics

Ψ is a dimensionless figure and represents the pressure figure. This figure is calculated according to the Equation 14, where u is the tangential speed with $u = \frac{d \times n \times \pi}{60}$. The pressure value decreases as seen in Diagram 42 with increasing specific speed. This connotes that with increasing specific speed the possible heads are decreasing.

$$\Psi = \frac{2 \times \mathrm{g} \times \mathrm{H}}{u^2}$$

Equation 14: Pressure figure

The value η represents the efficiency of each hydraulic.

The value σ represents the Thoma-figure and is used to describe the cavitation. This figure can be defined with the same criterion as the NPSH. In Equation 15 the definition of σ is shown. Ψ_c is also a dimensionless figure for cavitation and is calculated with the Equation 16. In Diagram 42 it is distinct apparent that σ is increasing with the specific speed, which is described physically with increasing flow rates.

$$\sigma = \frac{\Psi_C}{\Psi} = \frac{NPSH}{H}$$

Equation 15: Thoma figure

$$\Psi_C = \frac{2 \times \mathrm{g} \times \mathrm{NPSH}}{u^2}$$

Equation 16: Cavitation figure

A typical Cavitation characteristic curve, with the different cavitation criteria, is qualitatively shown in Diagram 43. The criterion incipient (incip), $1\mathrm{cm}^2$, 0% efficiency loss (0η), 1% efficiency loss (1η) and 3% head loss (3H), are shown at which pre-pressure these qualitatively occur. The criterion incipient determines when the cavitation starts. The criterion $1\mathrm{cm}^2$ is a company internal criterion and describes the cavitation where $1\mathrm{cm}^2$ of the impeller area is covered with vapour.

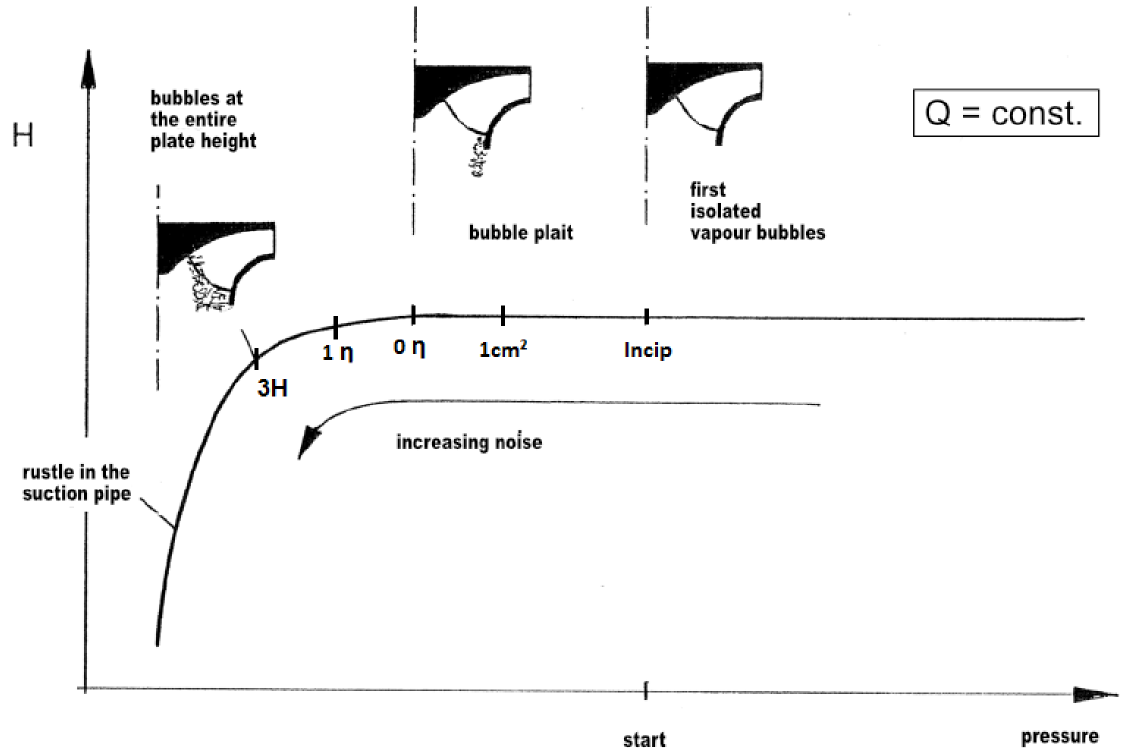


Diagram 43: Cavitation characteristic curve (Jaberg, 2012, p. 326) – (source modified)

These 5-adjustable hydraulics are mainly used for cooling water pumps to cover the required cooling water flow rate for the entire process. The different processes are described in Chapter 4.3.4 (page 68). The hydraulic adjusting systematic is described in Chapter 3.5.3.3.1.2 (page 20).

ns	nq
317,8	87,1
311,8	85,4
416,7	114,2
536,0	146,8
812,0	222,5

Table 8: Adjustable blade hydraulics

The adjustable blade hydraulics are automatic hydraulic adjustable ones and can also be used as manually adjustable hydraulics.

7.2 Preliminary standardization range

According to the collected market and competitor data, a preliminary standardization range is determined. The internal and external market analysis depicts that most of the VLSP pumps are below 1 megawatt, inquired and sold globally. Globally, 3298 pumps were sold in the last year below 1 megawatt, these are, as already mentioned in the market analysis, more than 50% of all sold VLSP's.

In comparison, the internal market analysis with the PNW and SNW series of KSB is shown in Diagram 44. This illustrates well that these two series of KSB would cover most of the internally demanded range. Additionally, the other series of KSB, as mentioned in the previous chapter, would cover the range up to higher flow rates. However, these series are not taken into consideration in the figure below because of additional design features; such as pull-out unit.

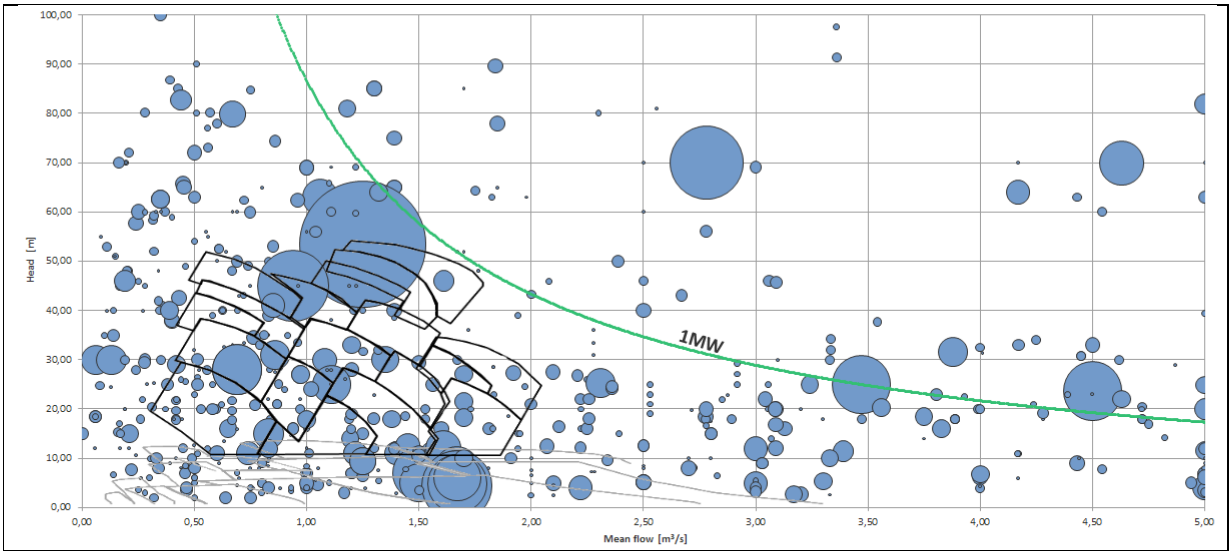


Diagram 44: Comparing with KSB

In Diagram 45 the internal inquiries are compared to the hydraulic mapping of Sulzer. Sulzer covers with its standardized range the complete area of the company's inquiries. This comparison shows that Sulzer has already hydraulically standardized a bigger range, up to higher heads and flow rates. This may also be a result of different or additional market accesses.

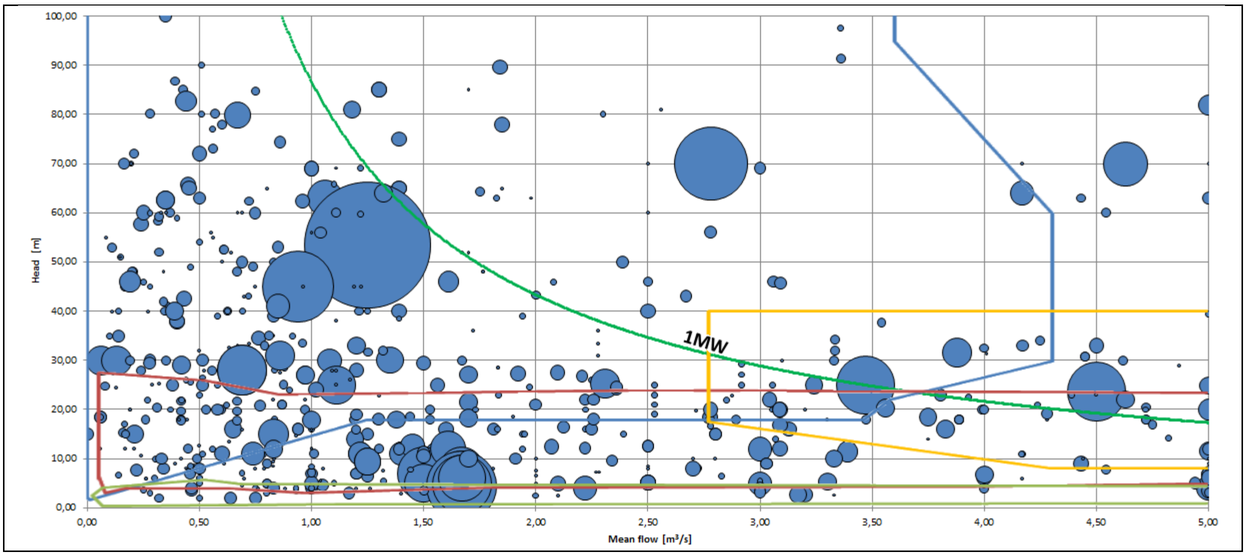


Diagram 45: Comparing with Sulzer

Furthermore, the fact that above 60m head pumps usually are in multistage design and this limits the standardization up to 60m. Another issue is that above 1m column pipe diameter, customers often prefer according to service reasons a pull-out type, because otherwise the required space for maintaining the pump needs to be enlarged and the limitation of the hall crane must be increased as well. This results in a maximum flow rate of approximately 2,3m^3/s when the fluid travels through the column pipe with a velocity of 3m/s. In fact, below 0,2m^3/s and 10m are almost no inquiries, this section is not dedicated to being standardized.

Taking all these facts into account, a specific area where it is most promising to standardize the pump characteristic curves of the VLSP can be examined. This standardization range is illustrated in Diagram 46. For determination of this range, the internal inquiries were prioritized higher than the competitors standardized ranges, since the internal inquiries reveal own market access. With this range, approximately 80% of sold pumps below 1 megawatt are covered. This is a potential of about 2600 pumps per year only within the preliminary standardization range.

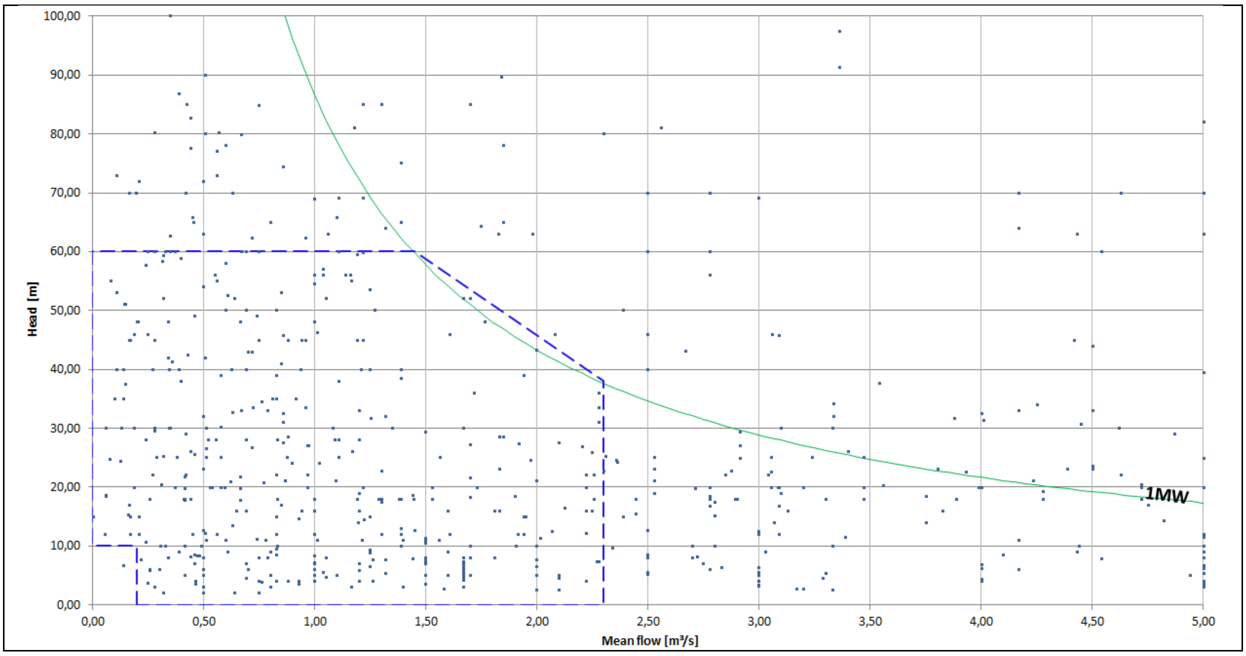


Diagram 46: Preliminary standardization range

The preliminary standardization range covers parts of all application ranges, where the VLSP is intended to use, this justifies the selected range. This is illustrated below in Diagram 47.

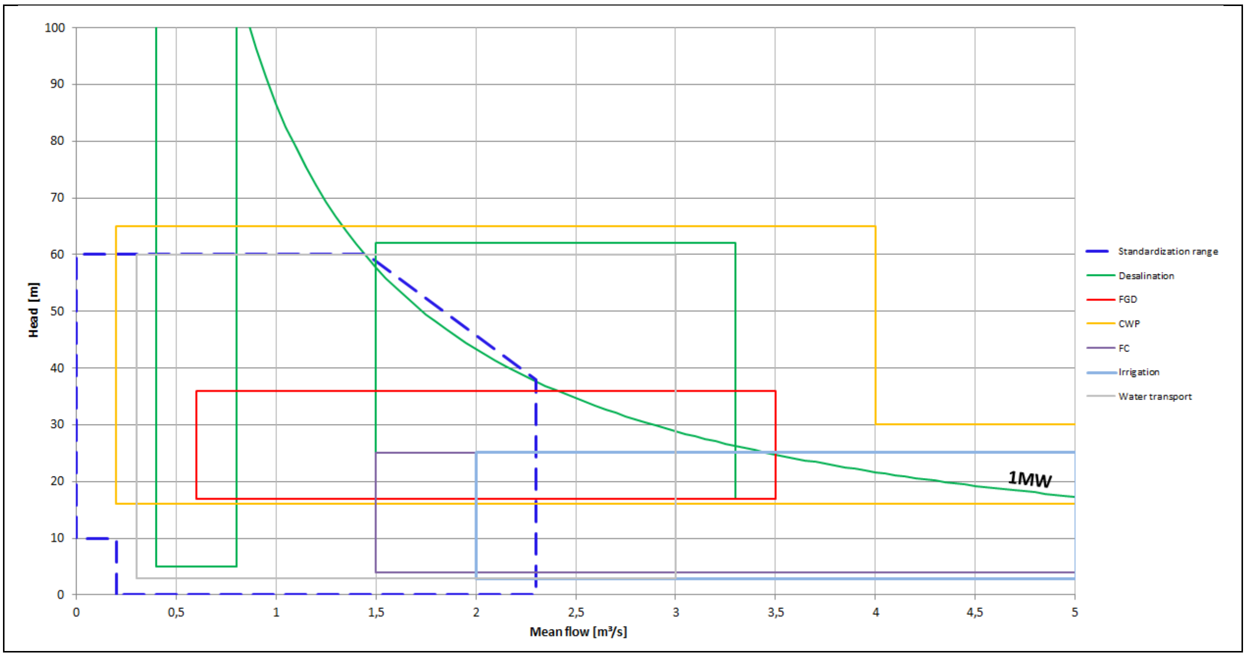


Diagram 47: Comparison standardization range – application ranges

This chosen range, where standard sizes are delivered, is possible to enlarge anytime, if other ranges turn out to be good as well.

7.3 Standard mapping

The standard mapping is indicated to realize parts of the cost saving potential as figured out in Chapter 6.1 and summarized in Table 6. With the standard mapping the pattern costs are not part of the project costs anymore, these will be investment costs. The patterns will be used multiple for different projects. Currently, the pump characteristic curves are scaled to the desired operation point. With this standard pump mapping the desired operation point is covered by fixed pump sizes. The disadvantage of using fixed hydraulic sizes is that the pump may be a bit heavier than with scaling exactly to the desired rated point and the pump efficiency at this point is probably not the highest.

To cover the preliminary standardization range, different hydraulics are used to fulfill this range with distinct pump characteristic curves. These might be low specific speed hydraulics, which may be trimmed enough and high specific speed hydraulics which are manually adjustable.

After different variations and analysis of the hydraulics, the mapping is based on the following hydraulics:

- ns145
- ns220
- ns262
- ns382
- ns812

The ns145, ns220, ns262 and ns382 were chosen because these most certainly can be trimmed to a certain extent. The ns812 is chosen as a manually adjustable hydraulic.

The trimming is limited according to Gülich to 5% of the outer diameter for guide vane pumps, when no experimental trimming studies were executed, because the pump curve becomes instable if the impeller is trimmed too much (Gülich, 2013, p. 194). To know which hydraulic is possible to trim more than 5% in diameter, an experimental test shall be conducted. The trimming of the pump characteristic curve is carried out with Equation 17. The exponent m is hydraulic- and company dependent and is distinct for H and Q. Gülich suggests an m of 2. For the VLSP mapping the exponents of hydraulics of another internal pump series are used. These are for m(Q)= 1,5 and for m(H)=2,5.

$$\frac{H'}{H} = \frac{Q'}{Q} = \left(\frac{d'_2}{d_2}\right)^m$$

Equation 17: Trimming law (Gülich, 2013, p. 190)

The operating speeds are selected to 1490rpm, 990rpm, 740rpm and once to 590rpm, hence to offer the lowest possible diameter. The highest possible speed with 2990rpm is not used due to resonant frequency of the pump which should be lower than the operating speed. These speeds are all operating with 50Hz and with an asynchrony motor. Each colour is dedicated to a certain speed. Blue is 1490rpm, green is 990rpm, yellow is 740rpm and purple is 590rpm.

The impeller diameters are determined to be used with different speeds and to save additional pattern costs.

For the dimensioning of each pump characteristic curve the cavitation criteria of zero efficiency loss was considered. The $NPSH_{av}$ is for a VLSP minimum 10m, because of the ambient pressure. Additionally, this pump is submerged to a certain degree hence the $NPSH_{av}$ is bigger than 10m. All the pump curves are scaled to maximum 10m $NPSH_{req}$. The cavitation criterion of zero efficiency loss is especially important when sizing the pump to about 10m $NPSH_{req}$ because when dimensioning with the criterion 3% head loss, there is no safety margin left. But the 3% head loss criterion is very important to be illustrated on the dimension sheet as well, since this is the most common criterion in the pump industry and customers may compare the different quotations with the 3% head loss NPSH criterion.

The zero-efficiency loss NPSH criterion was not measured for all hydraulics in the hydraulic test, hence a σ_{zul} /ns diagram is used to estimate the $\sigma(0\eta)$. This is illustrated in Diagram 48 below.

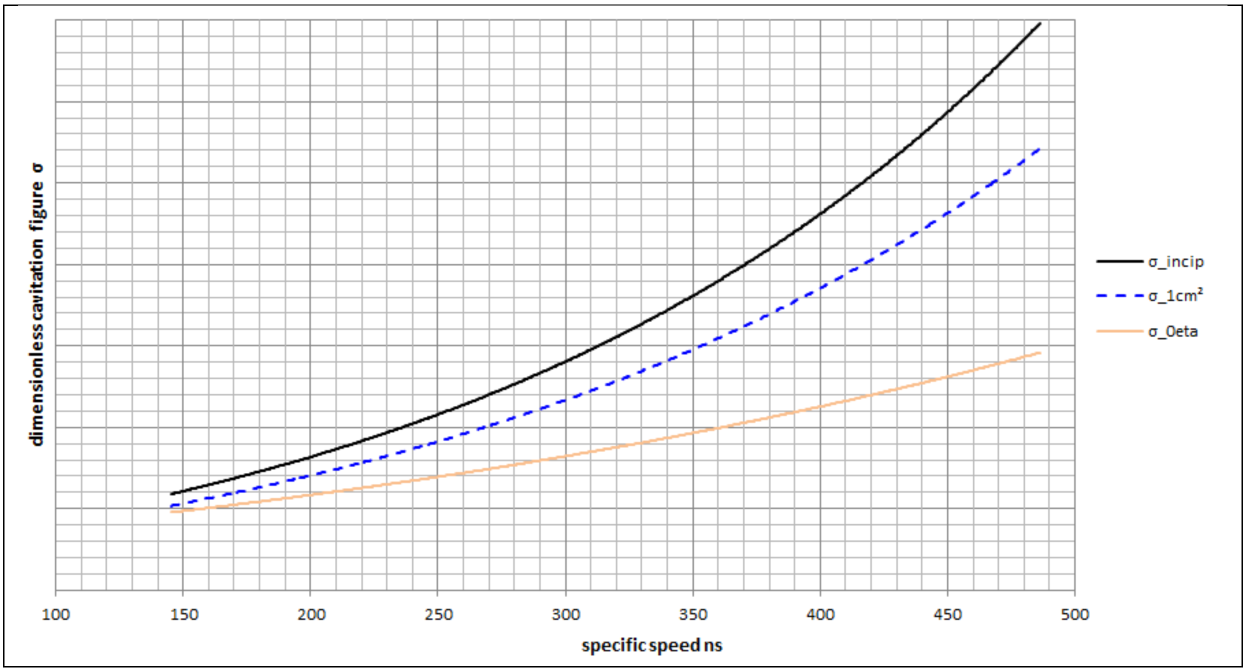


Diagram 48: σ_{zul}/ns statistic

The overview of the preliminary mapping is shown below in Diagram 49. All hydraulics and all sizes are shown in a simplified display. Each field is indicated for a hydraulic with different speed. On top of each field the used hydraulic and the speed for this field is specified. At each point a new pump size is indicated and the lines where the points are, state the optimum of this pump size. The different gradations of the colours distinguish the fields.

The biggest diameters, hence the top pump curve of each field is designed to 10m $NPSH_{req}$, as already mentioned. Then the sizes are scaled down along the ns-line with decreasing diameter to cover the range. Hence, the NPSH would not be a problem for the smaller sizes.

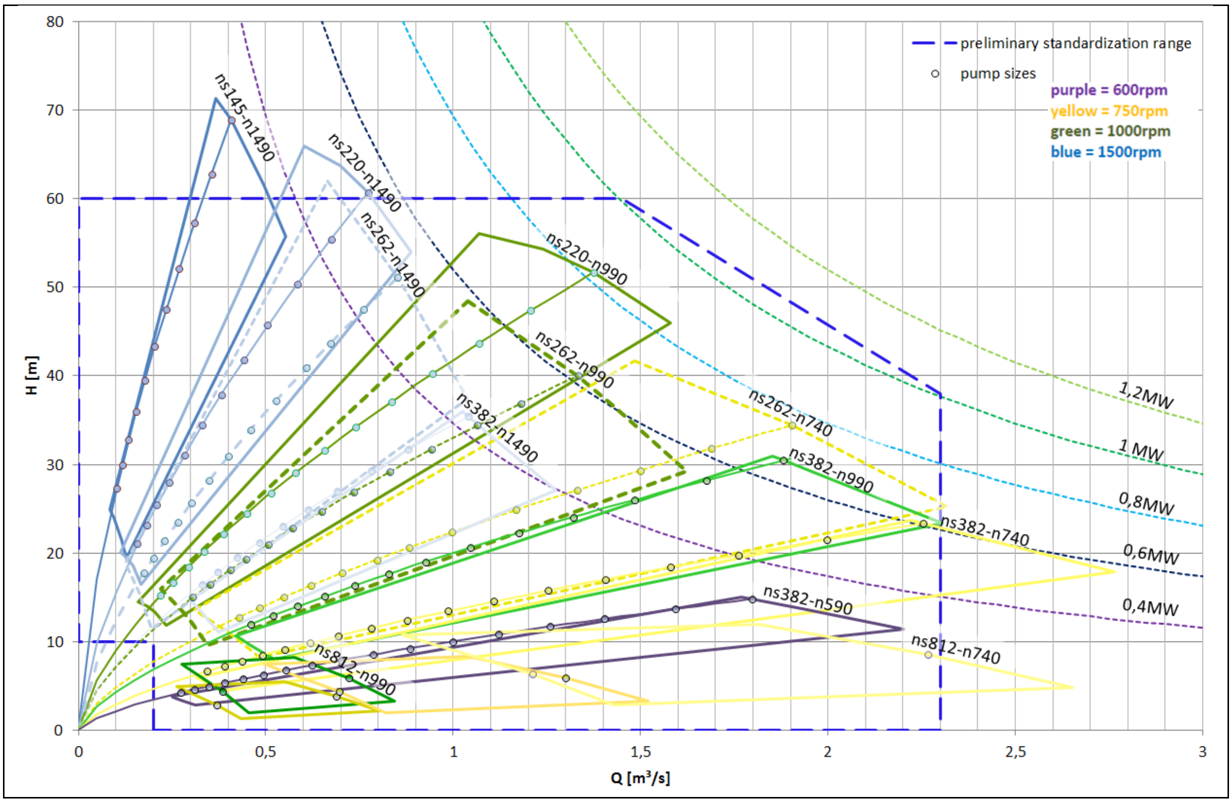


Diagram 49: Overall mapping

The values of flow rate, head, efficiency and NPSH, as well as the dedicated diameters, are listed for each pump size in the appendix 7.

The standardization range cannot be covered to the maximum extent of one megawatt, because the NPSH limits this. These ranges should be covered with multistage and lower speeds. The gap on the left might be covered with a lower specific speed hydraulic or multistage design.

The elaborate display is shown in appendix 7 with Diagram 58. Overall, 138 numbers of pump sizes are used to cover the standardization range. By using the same diameters of the same hydraulic with different speeds, for 138 numbers of sizes only 85 different patterns are used. In appendix 7 in Table 18-24 all pump sizes are listed, according to its' specific speed. There is a pump number dedicated to each pump size. The pump number one starts on top of each field and each dot symbolizes a new pump size. The tables include the flow rate, head, $NPSH_{req}$ and the efficiency at the optimum of each pump size. This is indicated for the full and the trimmed diameter. The suction- and the pressure diameter is listed in the two most right columns per pump size.

Each hydraulic is shown in Diagrams 50-54 to examine the detailed differences.

In Diagram 50 the ns145 is shown. The ns145 is only used with 1490rpm because with lower speeds the diameter increases and other hydraulics with higher speed, hence it is possible to use lower diameters for the same range. The ns145 is used with 11 pump sizes, these are listed in Table 18. The pressure diameter of the first pump size, hence the biggest, is 470mm.

In the next Diagram 51, the ns220 is illustrated. This hydraulic is used for the mapping with the speeds 1490rpm and 990rpm. Overall, the ns220 is applied with 27 sizes. These are listed in Table 19. The pressure diameters of the first pump size are 510mm for 1490rpm and 710mm for 990rpm. The gap between these two fields will be covered with the next size, the ns262. This is also the case to offer the best efficiency at almost any point of the range. The range of the ns262 is shown in Diagram 52. This hydraulic is used with 1490rpm, 990rpm and 740rpm. The different pump sizes for this hydraulic are listed in Table 20 and 21. The pressure diameters of the first pump sizes are 510mm for 1490rpm, 680mm for 990rpm and 850mm for 740rpm.

The sizes of the ns382 are illustrated in Diagram 53. This hydraulic is intended to operate with speeds from 1490rpm to 590rpm. The 590rpm sizes are the same as with 710rpm and hence, no additional patterns are used. Overall, 57 pump sizes are listed in Table 22 and 23. The pressure diameters of the first pump sizes are 480mm for 1490rpm, 670mm for 990rpm and 790mm for 740/590rpm.

The last applied hydraulic is the ns812, which is an automatically adjustable hydraulic, but is used for this standardized range as a manually adjustable hydraulic, as described in Chapter 3.5.3.3.1.1 and is designed as shown in Figure 17. The ns812 range is illustrated in diagram 54. This hydraulic is used with 990rpm and 740rpm speed, otherwise the $NPSH_{req}$ would extend the 10m with higher speeds. All the sizes are listed in Table 24.

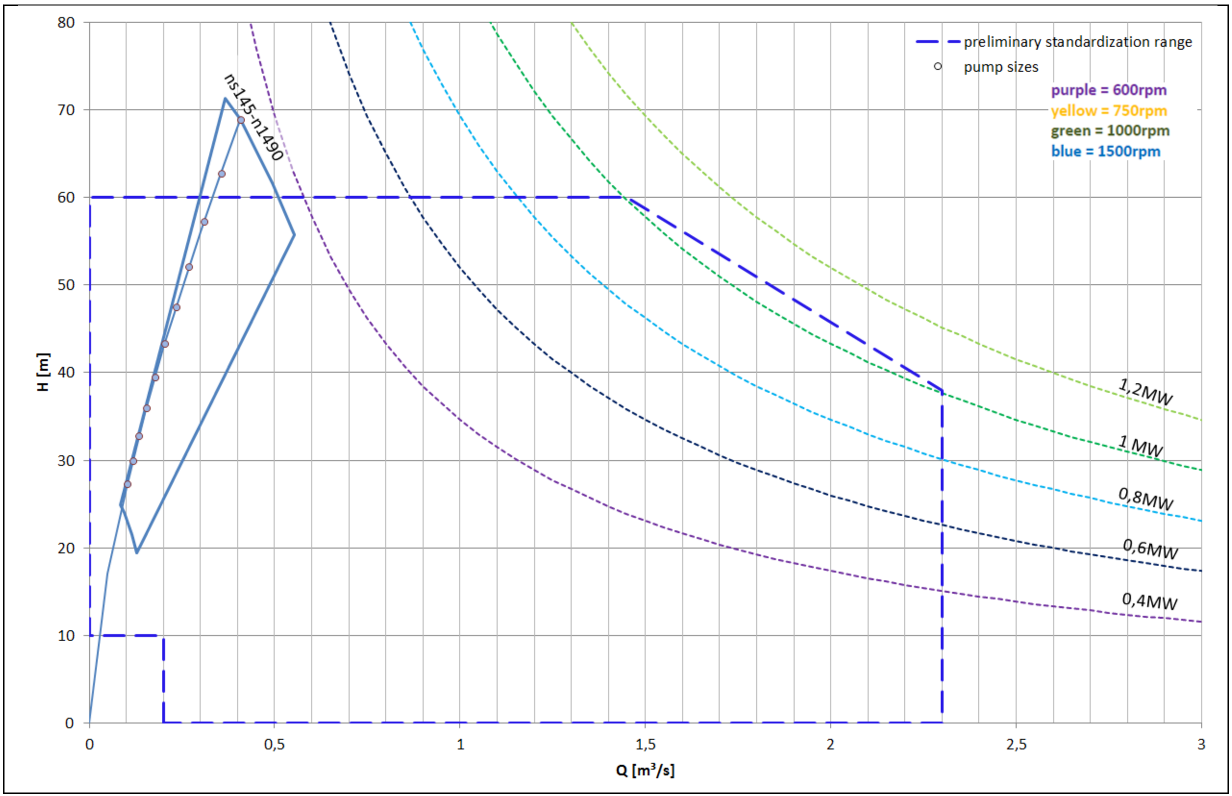


Diagram 50: ns145 hydraulic

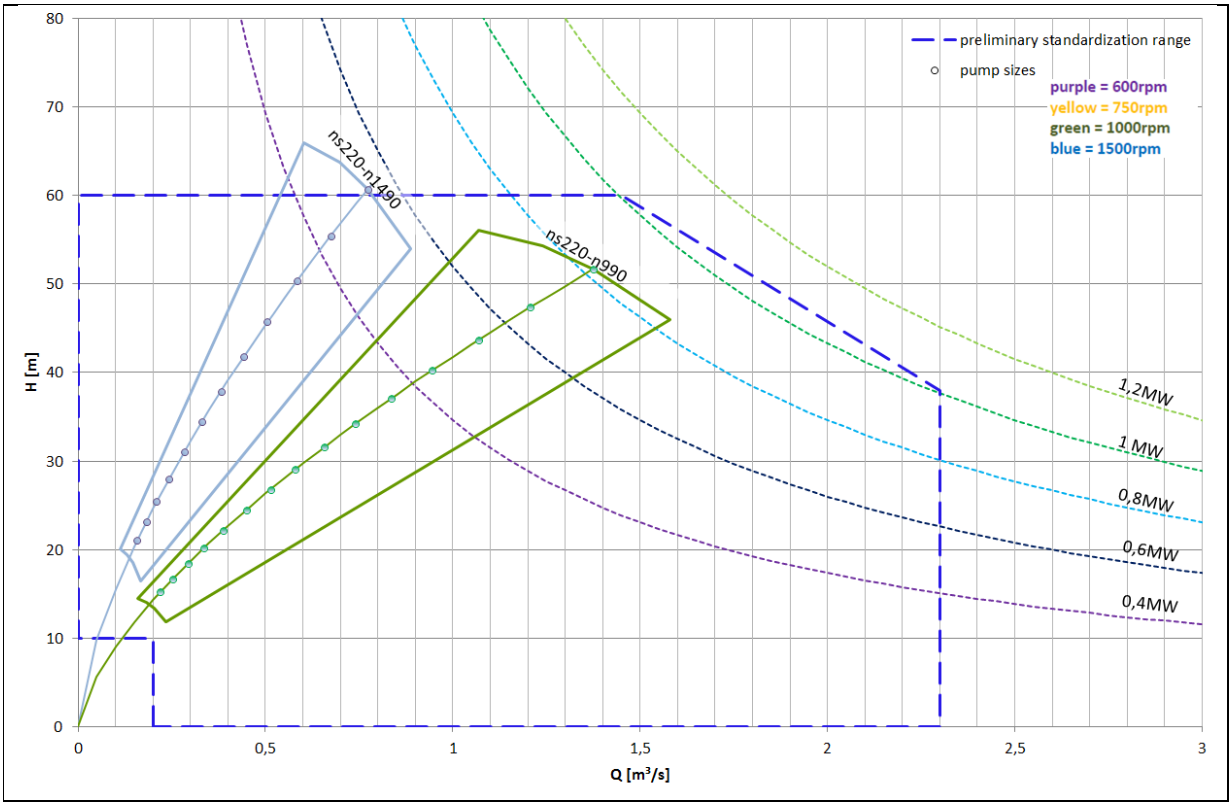


Diagram 51: ns220 hydraulic

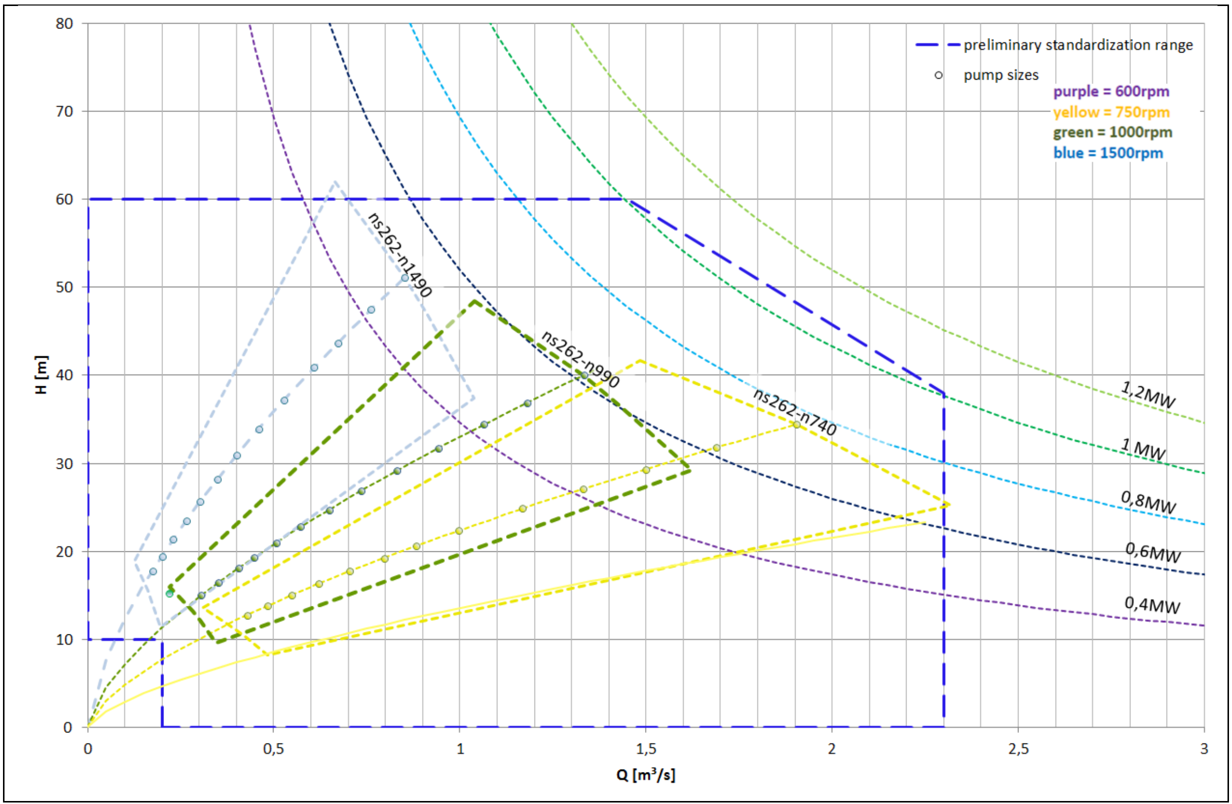


Diagram 52: ns262 hydraulic

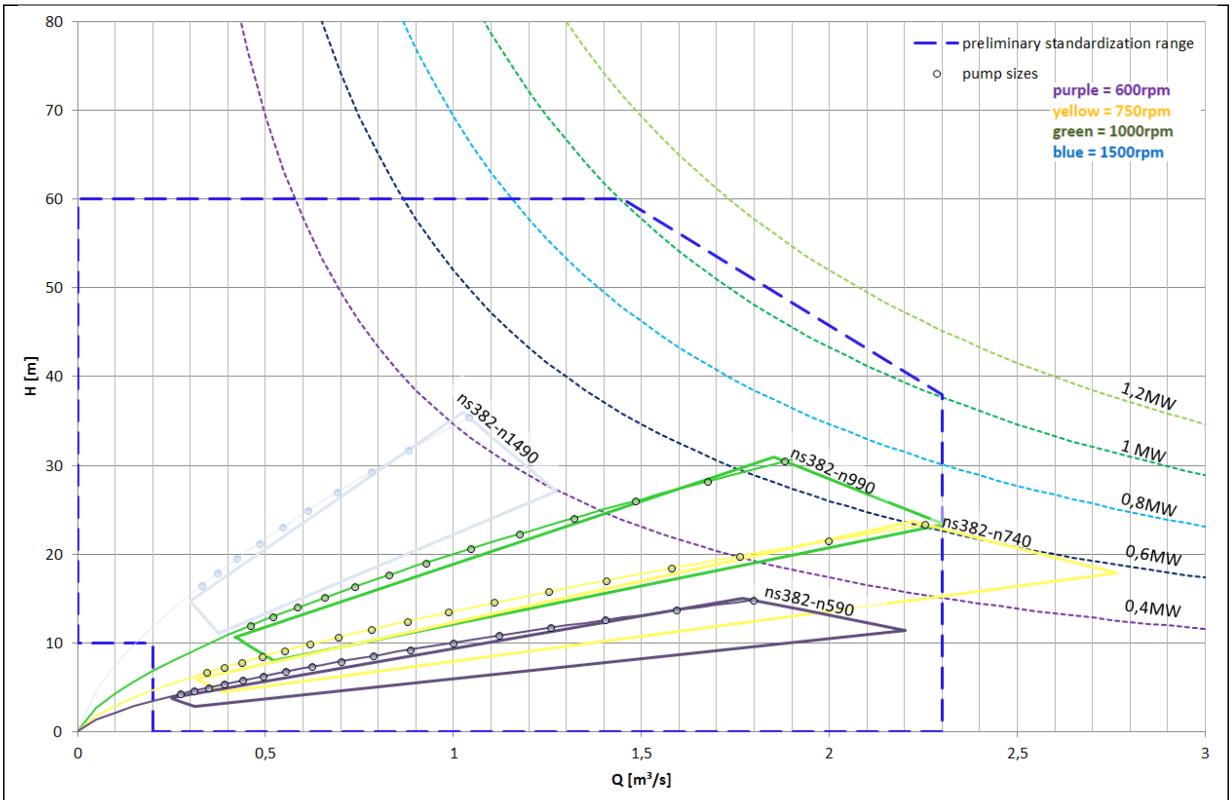


Diagram 53: ns382 hydraulic

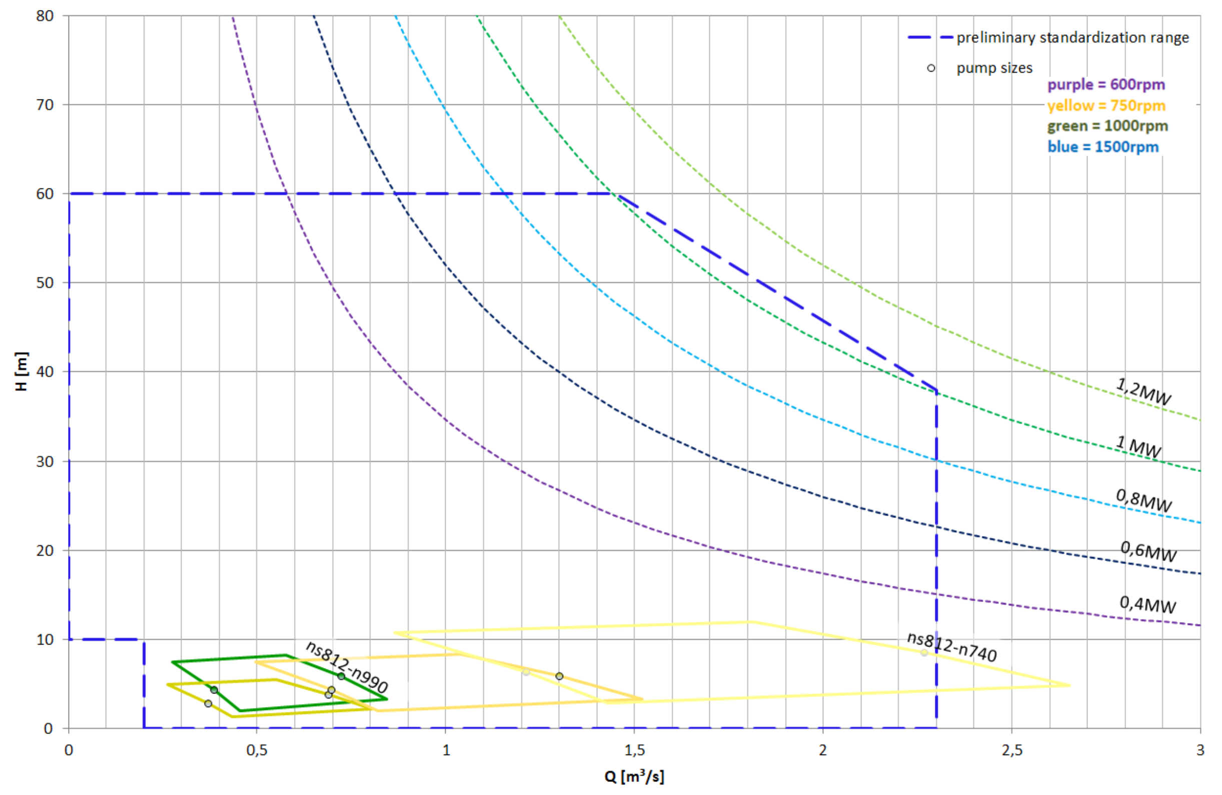


Diagram 54: ns812 hydraulic

With this standard mapping, in future it is possible to implement the standard sizes into a standard hydraulic selection tool.

8 Standardization concept

To enhance the cost structure, the engineering costs must be reduced, according to the cost analysis. This is realized with a standard calculation tool and a standard design.

The planned standard calculation tool is described in detail in Chapter 8.1 and the standard design in Chapter 8.2.

8.1 Standard calculation concept

Establishing a standard calculation will help to reduce additional calculation efforts. The consistent calculation steps and documenting the results will help gain know-how. For the beginning, additionally a finite element simulation is executed to adjust the calculation. Later this is only done for multistage and high flow rate pumps, extending the standardization range.

The calculation concept for the standardization is illustrated in Figure 73. The mapping with all the different sizes and the assigned data are implemented in the hydraulic selection programme. By inserting the desired flow rate and head, the selection programme will suggest different possibilities to choose from. After selecting the proper size and hydraulic, this programme will export the required data to the calculation tool for further calculations. Additionally, the data of the project specification, such as site conditions, the discharge option and the water levels, should be inserted to the calculation tool.

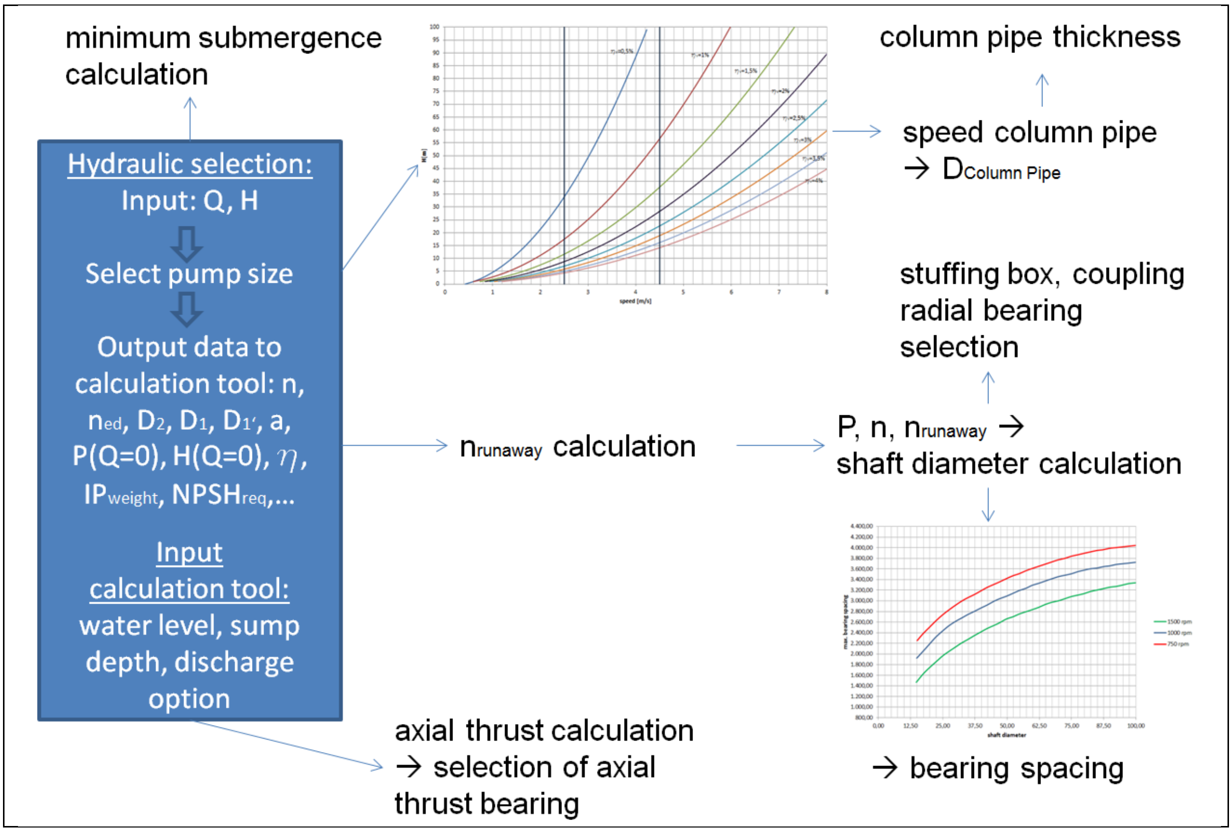


Figure 73: Calculation concept

8.1.1 Minimum submergence

As a first step, the minimum submergence is determined. For this, the diameter of the suction bell is the dominant factor. The suction bell diameter can be decided with Diagram 7 and 8. The speed at the suction bell is recommended with 1,7m/s according to HI. Afterwards the minimum submergence may be determined with the Diagram 9, or with Equations 2 and 3. If the minimum submergence is higher than the $NPSH_{req}$, the minimum submergence value is the determinant factor.

8.1.2 Length of the pump

The length of the VLSP is calculated with the depth of the sump, minus minimum water level, plus minimum submergence. If the floor clearance with 0,3 x D_{Bell}-0,5 x D_{Bell} is not sufficient with this length, installations in the sump must be placed (Figure 25) or the sump should be covered, as these arrangements lower the required submergence. The calculation of the length is executed with Equation 18. Each dimension is illustrated in Figure 74.

$$length = depth - min.waterlevel + \frac{min.submergence}{NPSH_{req}}$$

Equation 18: Length of the pump

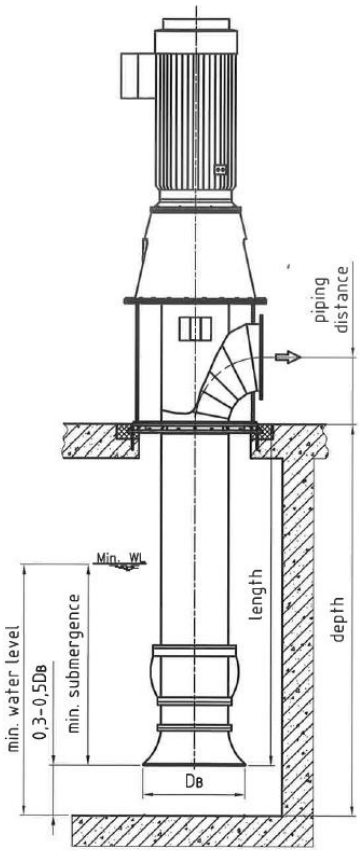


Figure 74: VLSP dimensions

8.1.3 Column pipe diameter

Next the column pipe diameter is selected. The decisive factor for the column pipe diameter is the resulting velocity in it. Diagram 10 gives a small hint which diameter to choose. To have a precise decision, the selection is done according to Figure 75. In the upper diagram only the bowl pump curve and bowl efficiency is illustrated. By selecting the diameter in a drop-down menu, the velocity in the column pipe and the efficiency loss to the discharge is calculated. The losses are calculated according to losses inside the pipe and the discharge elbow. The elbow head loss is calculated, as illustrated in Equation 6, where c is the velocity inside the column pipe and ζ is according to the mentioned ζ in Chapter 3.5.10.

The pipe loss is calculated with Equation 4. λ is representing the pipe friction coefficient, L the length of the pipe and D the inner diameter of the pipe. For the selected flow rate and the restricted velocities inside the column pipe, the flow is turbulent and always above Re>10^5. Hence, an empirical equation is used for calculating the loss coefficient, this is calculated with Equation 19. This equation is only applicable, if Re>10^5 and the pipe is smooth. If the Reynolds number is lower than this value and the pipe is rough, the Moody diagram, as described in Chapter 3.5.7, must be used. With Equation 19 the loss coefficient λ is calculated iterative starting with the value one. The velocity should not exceed the constraints of 4,5m/s or 2m/s as the lower limit.

$$\frac{1}{\lambda} = 2 \times \log\left(Re \times \sqrt{\lambda}\right) - 0{,}8$$

Equation 19: Prandtl's resistance law

λ	Loss coefficient	[-]
Re	Reynolds number	[-]

Simultaneously the corrected pump- and efficiency curve is illustrated below in Figure 75, according to the head and efficiency loss calculations.

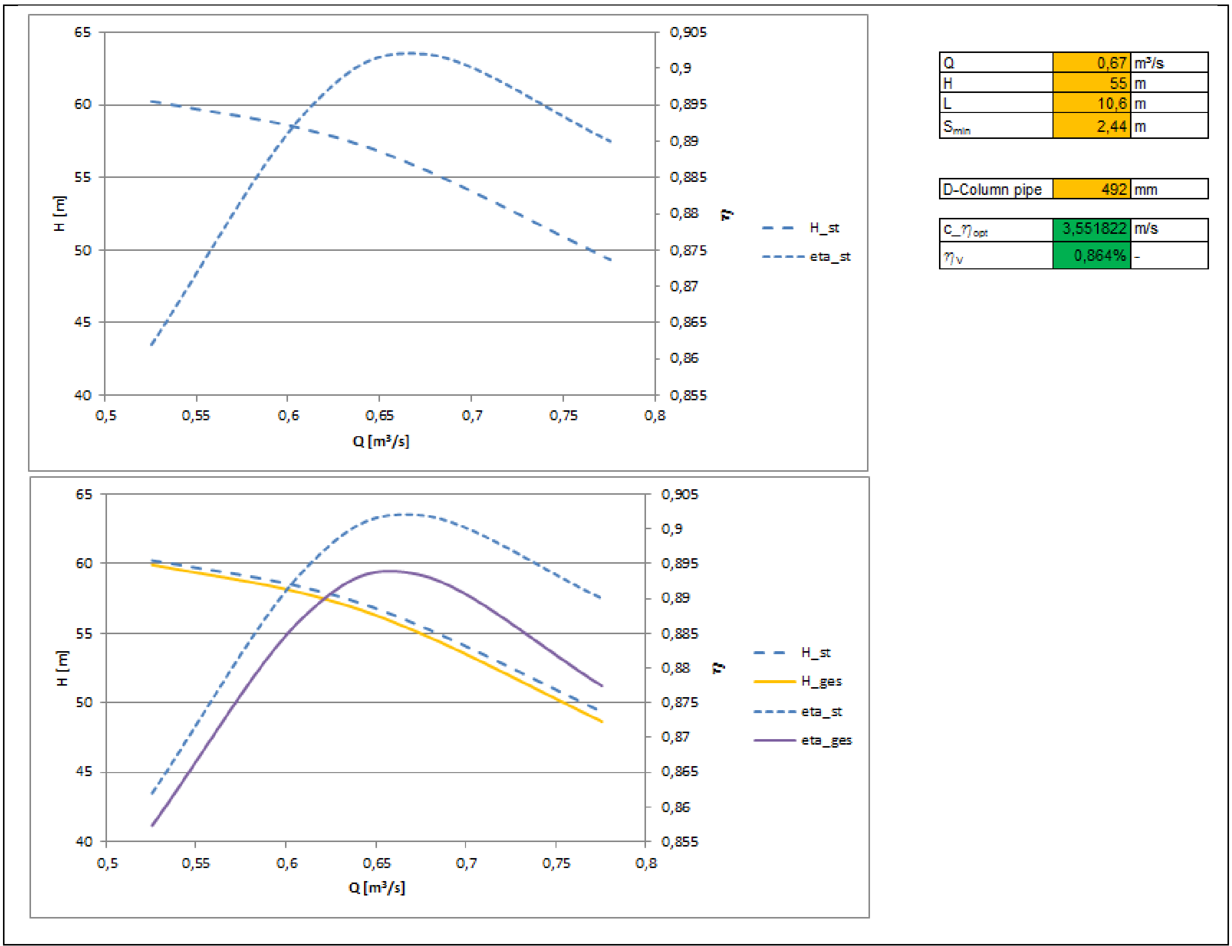


Figure 75: Column pipe diameter selection

The column pipe diameter is determinant for the wall-thickness selection. Due to experience the wall-thickness is selected according the systematic illustrated below in Table 9.

column pipe diameter [mm]	wall-thickness [mm]
< 1000	8
1000-2000	10
>2000	12

Table 9: Wall thickness systematic

8.1.4 Shaft diameter and spacing

In the next step, the shaft diameter and the bearing spacing are calculated. The determinant factors for the shaft diameter and bearing spacing calculation are the torsional stress and the critical bending speed. First the diameter of the motor coupling is calculated according to torsional stress. This is executed with the maximum hydraulic power consumption and the operating speed. The allowed torque tension is limited to 40MPa. This limit is according to the experience of the company. First the torque with the maximum power consumption and the rotational velocity is calculated (Equation 20). Afterwards the shaft diameter d_{mc} at the motor coupling is estimated with the torque and the polar section modulus according to Equation 21.

$$M = \frac{P}{\omega} = \frac{P}{\frac{n * \pi}{30}}$$

Equation 20: Torque

$$\boxed{d_{mc} = \left(\frac{M \times 16}{\pi \times \tau_{li}}\right)^{\frac{1}{3}}}$$

Equation 21: Calculation of the diameter at the motor coupling

M	Torque at the motor coupling	[Nm]
P	Motor power	[W]
ω	Rotational velocity	[1/s]
d_{mc}	Diameter at the motor coupling	[m]
τ_{li}	Limited torque tension	[MPa]

The spacing between the slide bearings is calculated according to Equation 22. This equation is converted from a reference value for a globular supported shaft according to Roloff Matek. The diameter is taken from Equation 21 and for the speed the runaway speed n_R with a safety margin of 50% is considered because this represents the worst case scenario.

$$\boxed{bearing\ spacing\ bsp = \sqrt{\frac{122{,}5 \times 10^6 \times d_{mc}}{n_R \times 1{,}5}}}$$

Equation 22: Bearing spacing calculation (Wittel , et al., 2009, p. 363)

To consider the shaft critical speed and the bearing spacing, the shaft diameter at the bearing is calculated with the shaft bending. The bending w of a cantilever shaft according to the weight force is calculated with Equation 23. In Equation 23, the modulus of resistance I is calculated for a shaft with the following equation $I = \frac{d^4 \times \pi}{64}$. The law of Dunkerley, mentioned with Equation 24, describes the critical rotational speed with the bending of the shaft.

$$w = \frac{(m_{ip} + m_W) \times g \times (bsp + oh) \times oh^2}{3 \times E \times I}$$

Equation 23: Bending of the cantilever shaft

$$\frac{1}{\omega^2{}_{kr}} = \frac{w}{g}$$

Equation 24: Dunkerleys law (Wittel , et al., 2009, p. 363)

bsp	Bearing spacing	[m]
w	Bending	[m]
m_{IP}	Impeller weight	[kg]
m_W	Water weight	[kg]
g	Gravitational constant	[m/s^2]
oh	Overhang of the impeller	[m]
E	Elasticity modulus	[MPa]
I	Modulus of resistance	[kg m^2]
ω_{kr}	Critical rotational speed	[rad/s]

Figure 76: Cantilever shaft

The bending Equation 23 is inserted in Equation 24 and results in Equation 25, the critical rotational speed of the shaft.

$$\omega_{kr} = \sqrt{\frac{3 \times E \times I}{(m_{ip} + m_W) \times (bsp + oh) \times oh^2}}$$

Equation 25: Critical shaft speed

$$\omega = \frac{n_R \times \pi}{30}$$

Equation 26: Rotational velocity

Inserting Equation 25 in Equation 26 and considering the speed with the runaway speed as a worst case scenario, will result in Equation 27. This calculation is considered with an empirical factor of 3 to include the rigidity of the slide bearings. This empirical factor is determined by checking the Equation 27 with several realized projects.

$$\boxed{d_B = \sqrt[4]{\left(3 \times \frac{n_R \times \pi}{30}\right)^2 \frac{64 \times (m_{ip} + m_W) \times (bsp + oh) \times oh^2}{\pi \times 3 \times E}}}$$

Equation 27: Calculation of the diameter at the slide bearing

The runaway speed is different for each hydraulic and is measured experimentally with a 4-quadrant test in turbine operation. At the tests a dimensionless factor n_{ed} is calculated. This is illustrated in Diagram 55 as a function of the specific speed.

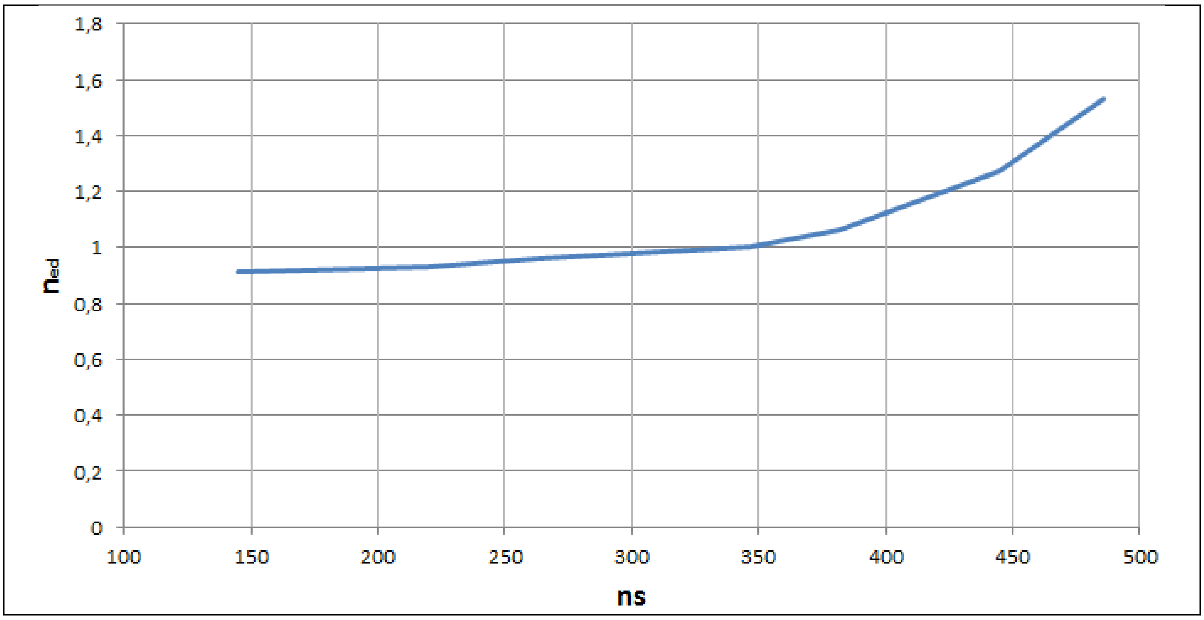


Diagram 55: Runaway speed coefficient as a function of the ns

The actual runaway speed is calculated according to Equation 28. Where the head H is the static head, which is relevant for turbine operation and D_2 is the inlet diameter at the impeller in pump mode.

$$n_R = n_{ed} \times 60 \times \frac{\sqrt{g \times H_{static}}}{D_2}$$

Equation 28: Runaway speed calculation

n_R	Runaway speed	[rpm]
n_{ed}	Factor for the runaway speed	[-]
H_{static}	Static head	[m]
D_2	Impeller inlet diameter in pump mode	[m]

8.1.5 Axial thrust calculation

The axial thrust calculation is executed according to the general description in Chapter 3.5.13. For one hydraulic the axial thrust calculation is exemplified.

Equation 7 is the base of the calculation. For the calculation of the weight force, Equation 8 is used.

The general momentum force equation is mentioned with Equation 11. This is applied for a certain impeller (Figure 77) and the result is described in detail with Equation 29. The momentum force is the sum of all occurring at the impeller- at the inlet and the outlet. The dimensions and the occurring momentum forces at the inlet and outlet are illustrated in Figure 77.

$$F_I = \rho \times \frac{Q^2}{\left(\frac{D_2{}^2 \times \pi}{4}\right)} + \rho \times \frac{Q^2 \times \sin\alpha}{2 \times \pi \times D_1 \times (D_1 - D_1')}$$

Equation 29: Momentum force

Calculating the hydraulic axial thrust force for the standardized range is executed with Equation 9 for the hydraulic force and Equation 12 for the suction force. This simplified calculation assumes that the pressure distribution is constant. The calculation depicts the worst case of the axial thrust. For sizes extending the standardization range, the axial thrust force calculation is executed with the company’s elaborate axial thrust calculation tool.

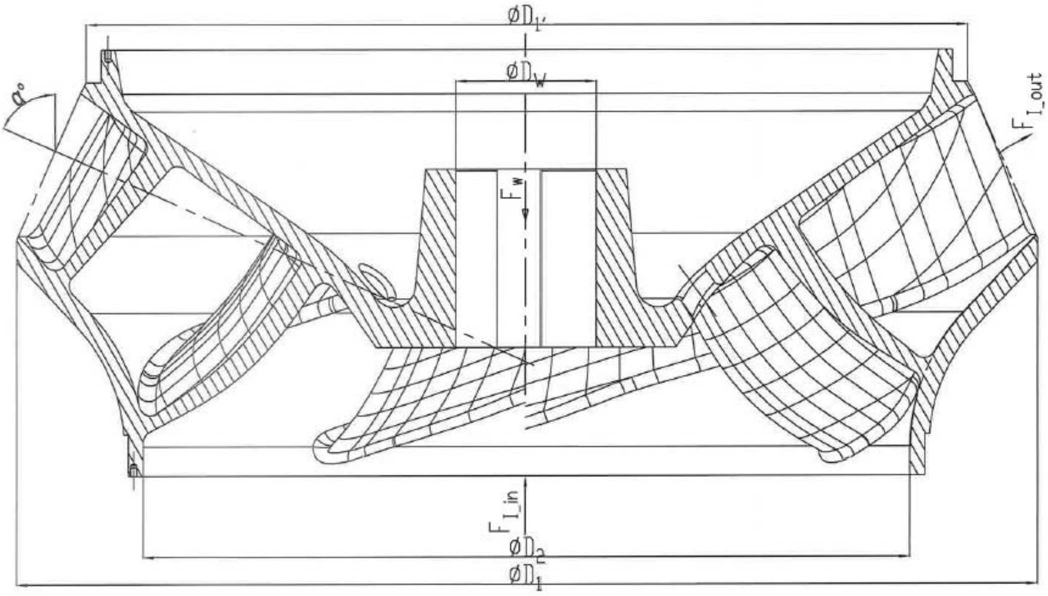


Figure 77: Pressure distribution

8.2 Standard design

8.2.1 Structure for standard pumps

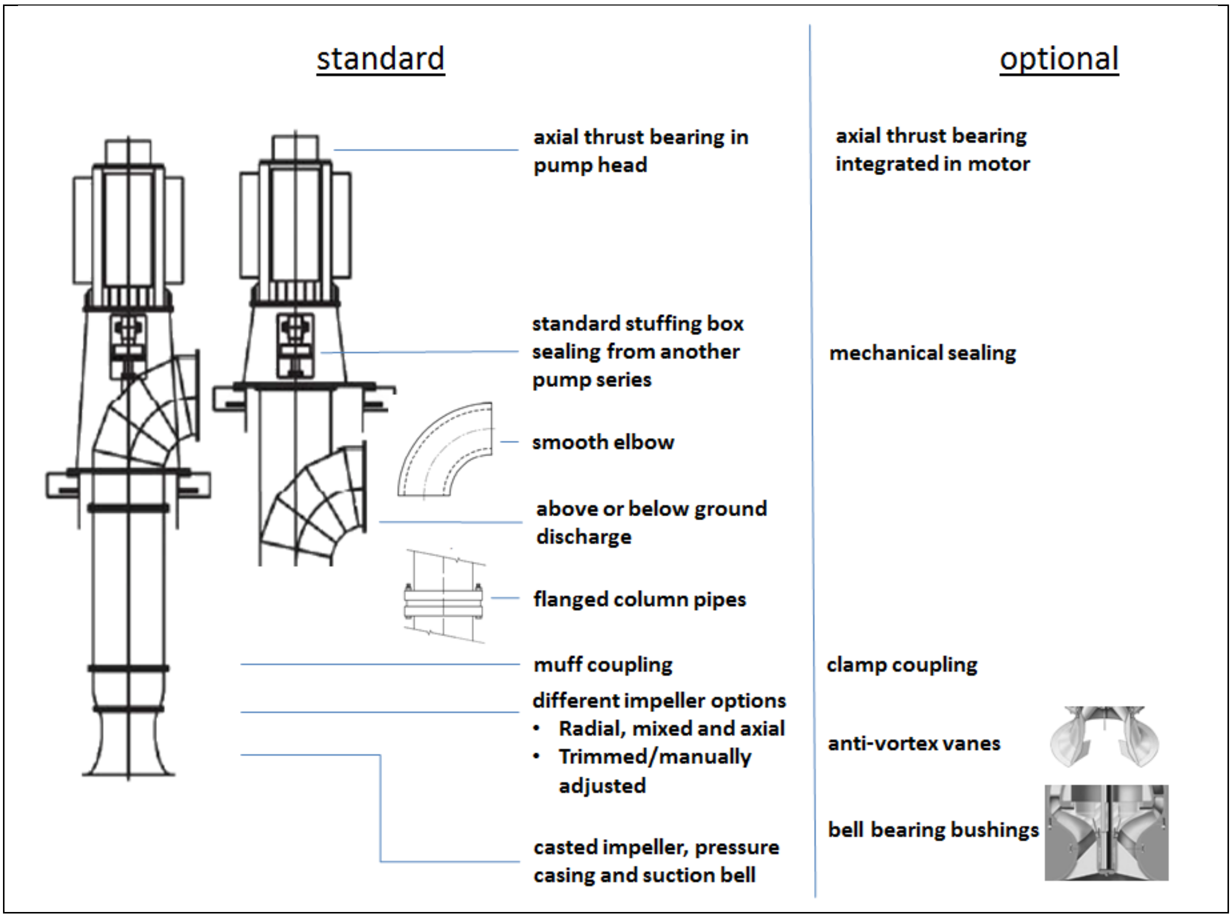


Figure 78: Standard design

For standardization of the VLSP it is important to define which parts are standard and which are optionally available. The standard and optional design is illustrated and mentioned in Figure 78. The discharge configuration is available above and below floor as a standard because these are the most common designs. The impeller, pressure casing and the suction bell are casted because for the standard sizes higher amounts are needed, hence, casting is cheaper. For the suction bell, standard sizes are available. These are mentioned next in Chapter 8.2.2. The impellers are offered in different designs depending on the specific speed. These are trimmed or manually adjustable to reduce the needed patterns as mentioned in Chapter 7. The suction bell may accommodate optionally bell bearing bushings to reduce the shaft deflection and anti-vortex vanes, which are important, especially for high specific speed pumps. The column pipes are flanged as a standard and in between the radial bearing stars are pressed.

The shaft assembly and transmitting of the torque is done with a muff coupling as a standard for the smaller sizes, optionally, the clamp coupling is also available.

The elbow is a smooth version as a standard and the standard sizes are supplied by a vendor, only the flanges are welded onto. This helps to reduce the costs and the losses in the elbow are reduced as well.

The shaft sealing is placed inside the elbow, where the shaft is exceeding through. The standard shaft sealing is a stuffing box. To take advantage of other companies pump series, parts of the stuffing box assembly are common. Optionally, a mechanical seal is possible as well, also with the same sizes of the congruent pump series.

The axial thrust bearing is either placed inside the pump head or the motor but as a standard, it is assembled in the pump head. For the American and Asian market the integrated axial thrust is preferred. The bearing is an angular contact roller bearing as a standard. The cooling of the axial thrust bearing oil is realized as standard with a cooling coil and optionally with a casing cooling system.

The slide bearings are rubber, plastic or composite bearings as a standard. This is due to good sliding properties and good dry running conditions during start-up. For the slide bearings, standard sizes across vendors are used, to have the possibility to use different suppliers.

The standardized VLSP would not be a pull-out version because the design with a pull-out unit increases the complexity and the costs.

The pump flexibility increases with a hydraulic adjustable impeller and is especially benefiting at the higher flow rates, hence, for the standardized range this is not used. The flexibility possibility can be additionally achieved with a frequency converter for the standard sizes but also not as a standard.

The standard VLSP is not designed as a multistage pump but the parametric design is executed in such a way that with some additional design effort it can be realized.

The standard materials for each part are dedicated in Table 10.

materials		
impeller	stainless steel	duplex
pressure casing	stainless steel	duplex
suction bell	cast iron	stainless steel
shaft	stainless steel	duplex
other parts	carbon steel	stainless steel

Table 10: Standard materials

For the rotating parts the material is at least stainless steel or better, e.g. duplex steel. The operating speeds are between 590rpm and 1490rpm, hence, the circumference speed is, depending on the impeller diameter, high. Certain materials, such as grey cast, are not sustained enough to withstand this high circumference speed according to abrasion. (Gülich, 2013, p. 1024)

The steel construction parts are selected according to the pumped fluid and customer requirements.

8.2.2 Parametrization

Currently, the VLSP is engineered customer specific, with the highest costs and time expenses for this process. As already mentioned in the cost saving potential chapter the parametric design, according to Figure 1, is selected for the VLSP standardization. This enhances the design process and fastens up the engineering but also allows remaining flexible enough for customer requirements. The parametrization helps to reduce the design time significantly, hence, also reduces the engineering costs.

All the calculated and selected sizes are handed over to the design programme. With the parametric design these variables feed the programme and design the pump up to a certain extent.

In Table 11, the standard sizes for column pipes and elbows are noted. These are determined according to supplier's information.

	Column pipe- /elbow outer diameter [mm]	wall thickness [mm]	Column pipe- / elbow inner-diameter [mm]	R/D
DN300	323,9	8	307,9	1
DN400	406,4	8	390,4	1
DN500	508	8	492	1
DN600	610	8	594	1
DN700	711	8	695	1
DN800	813	8	797	1
DN900	914	8	898	1
DN1000	1016	8	1000	1

Table 11: Standard column pipe and elbow dimensions

For each elbow diameter two different types of discharge heads will be needed, below and above floor, which results in 16 different discharge heads. To reduce the total number of discharge heads, two elbow diameters share one discharge head size (e.g. DN300/400,...) Thus, 8 different discharge heads are available, 4 above and 4 below floor.

The abbreviation BS represents the bearing support size. Different dimensions are assigned to each BS, as can be seen in Table 12 and 13.

The shaft is available with standard dimensions. The length of the shaft is individual to fulfil the customer site conditions. The standard dimensions for the drive shaft are illustrated in Figure 79 and noted in Table 12. The dimension “A” represents the diameter of the shaft coupling. The shaft coupling is manufactured in-house. “B” is the diameter for the stuffing box. “C” is the inner diameter dedicated for the angular contact roller bearing. “D” indicates the diameter of the motor coupling used. These standard dimensions are selected according to supplier’s data sheets.

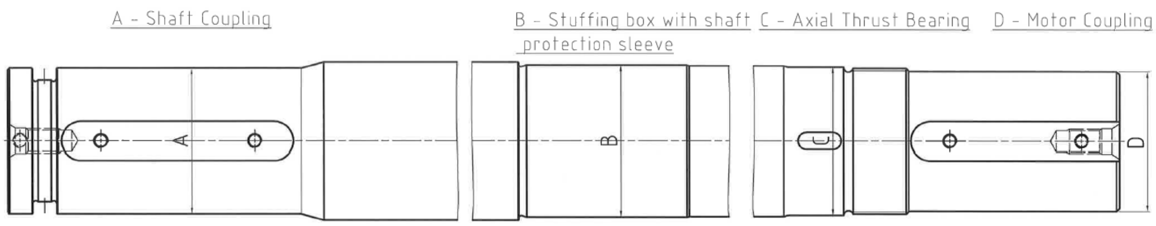


Figure 79: Drive shaft

	ØA (shaft coupling)	ØB (stuffing box)	ØC (axial thrust bearing)	ØD (motor coupling)	Max. Ø
BS80	85	90	85	80	95
BS95	100	115	100	95	120
BS110	115	130	120	110	135
BS125	130	155	130	125	160
BS140	145	170	150	140	180
BS160	165	180	170	160	190
BS180	190	200	190	180	210

Table 12: Drive shaft standard dimensions

The standard dimensions of the pump shaft are illustrated in Figure 80 and the different sizes are mentioned in Table 13. “A” represents the shaft coupling diameter and the impeller fit. “B” indicates the diameter of the slide bearing, which is selected according to supplier’s information.

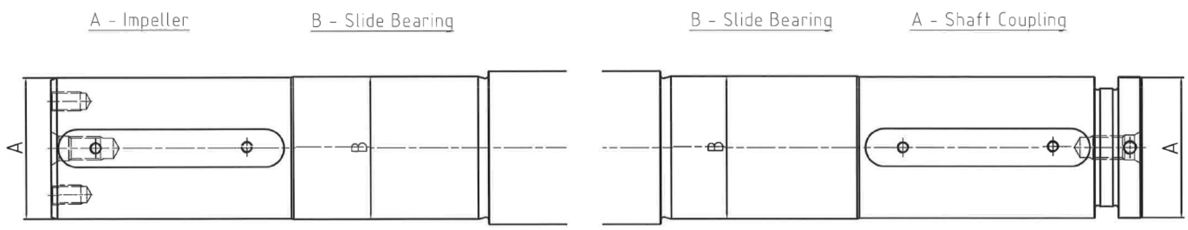


Figure 80: Pump shaft

	ØA (shaft coupling/ impeller)	ØB (slide bearing)
BS80	85	90
BS95	100	105
BS110	115	120
BS125	130	135
BS140	145	150
BS160	165	170
BS180	190	200

Table 13: Pump shaft standard dimensions

The recommended suction bell inlet diameter is according to HI calculated with the velocity at the inlet with 1,7m/s. To standardize different sizes the lower and upper limits are used. These are, as illustrated in Diagram 7, with 0,9-2,4m/s for flow rates between $0m^3/s$ and $1{,}4m^3/s$ and for the flow rate range $1{,}4^3/s$ to $20m^3/s$ with 1,2-2,1m/s. The determined standard suction bell inlet diameters are illustrated with Table 14.

Q [m³/s]	Suction bell inlet-Ø [m]
0 - 0,22	0,35
0,23 - 0,55	0,55
0,56 - 0,9	0,7
0,9 - 1,2	0,8
1,21 - 1,6	1
1,61 - 2,5	1,2

Table 14: Standard suction bell inlet diameters

First the hydraulic selection is done and the mechanic calculation is executed. After calculating the main dimensions, the available standard sizes are chosen. These calculated and selected dimensions are used for the parametric design and are handed over to the design application. All the dedicated dimensions to each part are transferred as well e.g. flange dimensions. This process is illustrated below in Figure 81.

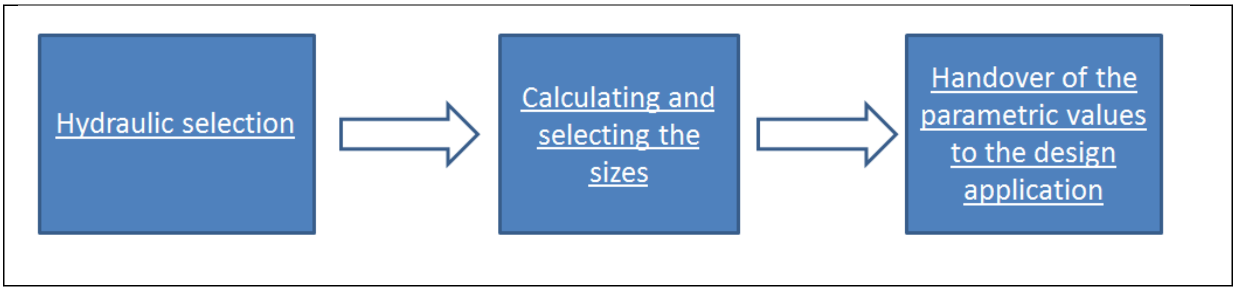


Figure 81: Parametric process

9 Conclusion

The VLSP is a promising pump for the future. It is the right product for applications that are on the rise, such as flood control, irrigation and water intake for desalination.

For the standard range the company faces a lot of small and bigger competitors, hence, the competition is fierce. Mostly the price and the lead time is the determinant factor for sales decisions, even less than the efficiency. With standardizing the highest demanded range, the product is competitive again. The evidence for that are the calculated potential cost savings, which account for approximately 14,5%. This is mainly achieved by decreased pattern- and engineering costs.

To gain this saving potential the pump mapping and the design should be standardized. With the standard pump mapping the patterns are used multiple times. For the standard design the parametric structure was selected, to stay flexible enough for customer requirements. The parameters are delivered with a standard calculation tool. The engineering costs and time are reduced with this standard calculation tool and the parametric design. These factors will contribute to the lead time reduction as well. This standard engineering process is lean shaped and digitalized.

Having a globally unified product strategy helps to align the supply chain and the cost benefits of a low-cost country are possible to exploit. Due to standard sizes, and with a clear product structure, the ease of the variant management may be implemented and the automatic Bill of Material creation is possible.

The gained know-how of the standard sizes will help to execute the bigger sizes off the standardization range. The companies' high level hydraulic- and technological knowledge will contribute for the demanded bigger sizes. As the competitor analysis depicted that the elaborate designs with adjustable blades, pull-out- and multistage design is highly valued.

The parametric pump design will be conducted as a follow up to this thesis.

10 Appendix

10.1 Appendix chapter 3

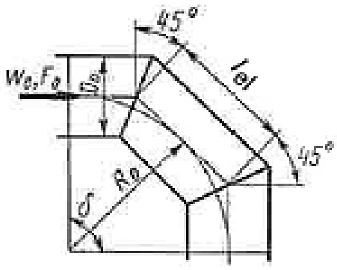


Figure 82: 3 elements welded elbow (Idelchik, 1986, p. 302)

ζ for a 3 elements welded elbow										
l_{el} /D	0	0,2	0,4	0,6	0,8	1,0	2,0	3,0	4,0	5,0	6,0
R/D	0	0,24	0,48	0,7	0,97	1,2	2,4	3,6	4,8	6,0	7,25
ζ	1,10	0,95	0,72	0,6	0,42	0,38	0,32	0,38	0,41	0,4	0,41

Table 15: Loss coefficients for a 3 elements welded elbow (Idelchik, 1986, p. 302)

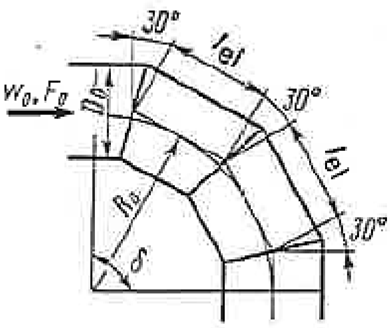


Figure 83: 4 elements welded elbow (Idelchik, 1986, p. 301)

ζ for a 4 elements welded elbow											
l_{el} /D	0	0,2	0,4	0,6	0,8	1,0	2,0	3,0	4,0	5,0	6,0
R/D	0	0,37	0,75	1,12	1,5	1,85	3,70	5,55	7,40	9,25	11,0
ζ	1,1	0,92	0,70	0,58	0,4	0,3	0,16	0,19	0,2	0,2	0,2

Table 16: Loss coefficient for a 4 elements welded elbow (Idelchik, 1986, p. 301)

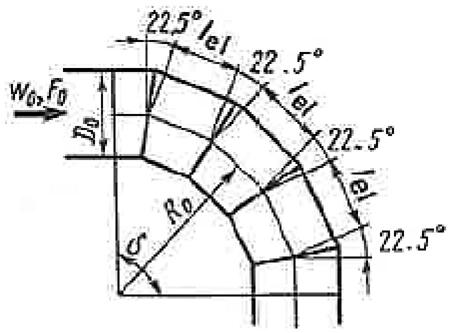


Figure 84: 5 elements welded elbow (Idelchik, 1986, p. 301)

ζ for a 5 elements welded elbow										
l_{el} /D	0,2	0,4	0,6	0,8	1,0	2,0	3,0	4,0	5,0	6,0
R/D	0,5	0,98	1,47	1,9	2,5	5,0	7,50	10,0	12,5	15,0
ζ	0,75	0,45	0,34	0,15	0,12	0,16	0,15	0,2	0,2	0,2

Table 17: Loss coefficient for a 5 elements welded elbow (Idelchik, 1986, p. 301)

10.2 Appendix chapter 4

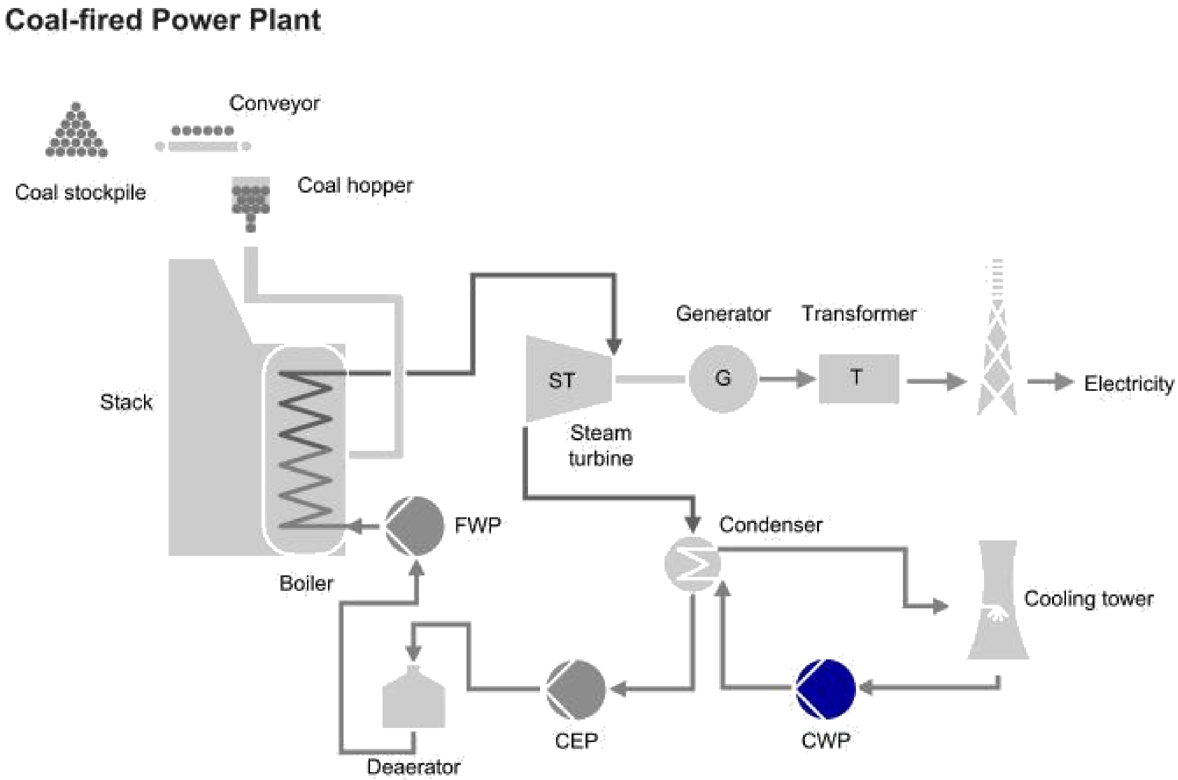


Figure 85: Coal/oil fired power plant (Sulzer – CWP for coal- and oil-fired power plants, 2017)

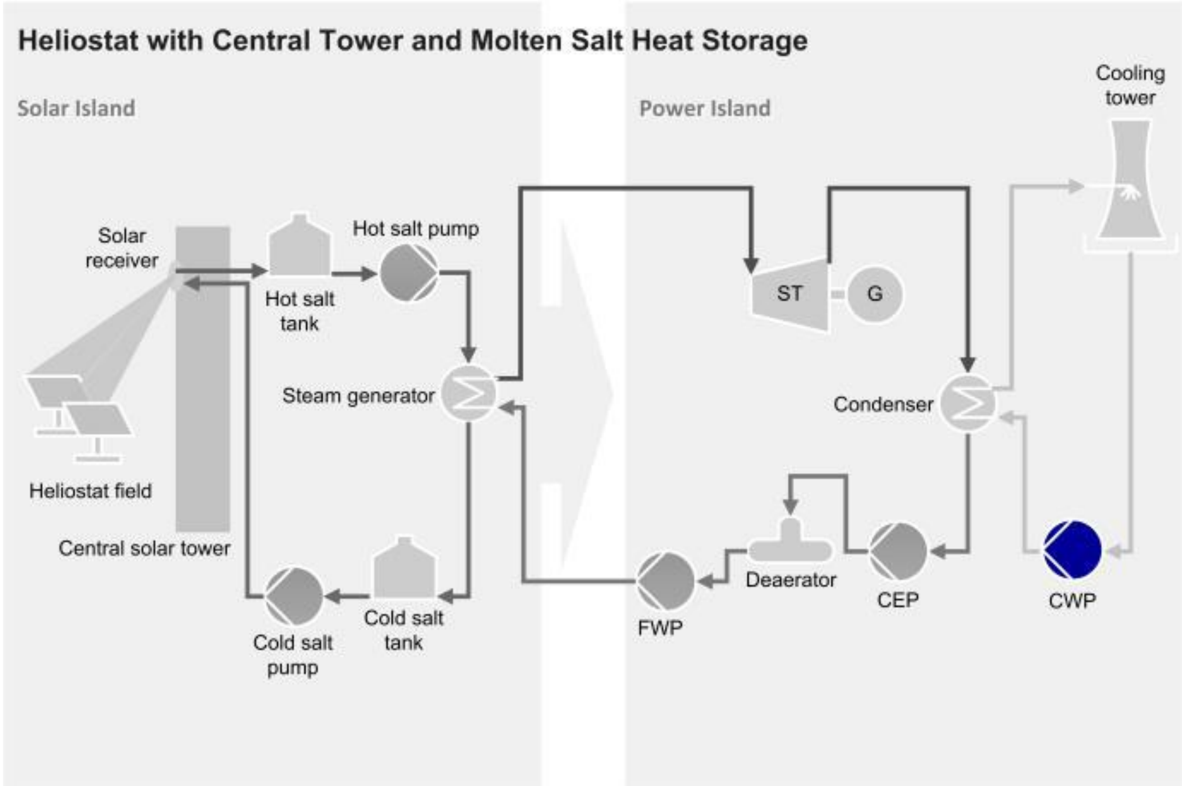


Figure 86: Heliostat with central tower and molten salt heat storage (Sulzer – Solar power generation, 2017)

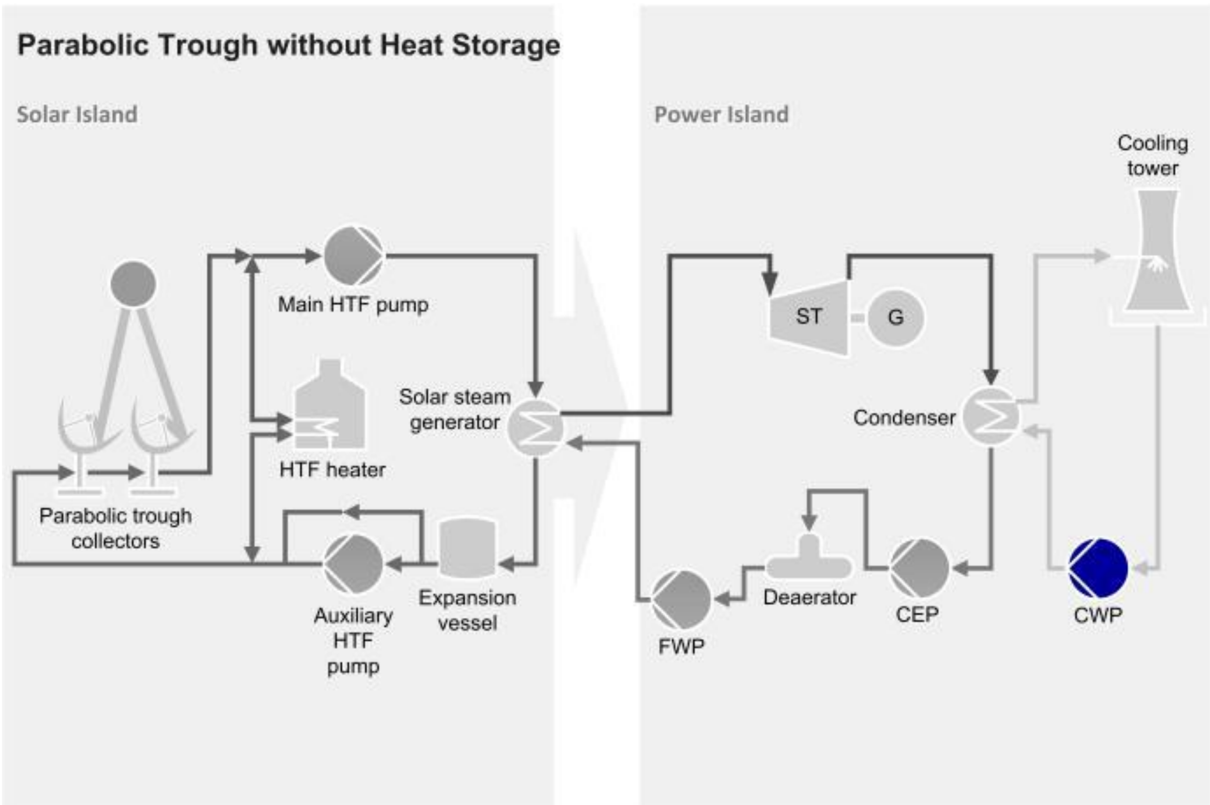


Figure 87: Parabolic trough (Sulzer – Solar power generation, 2017)

10.3 Appendix chapter 5

10.3.1 Comparison of the design options

This matrix is annexed inside the back book cover.

10.3.2 Flowserve mappings

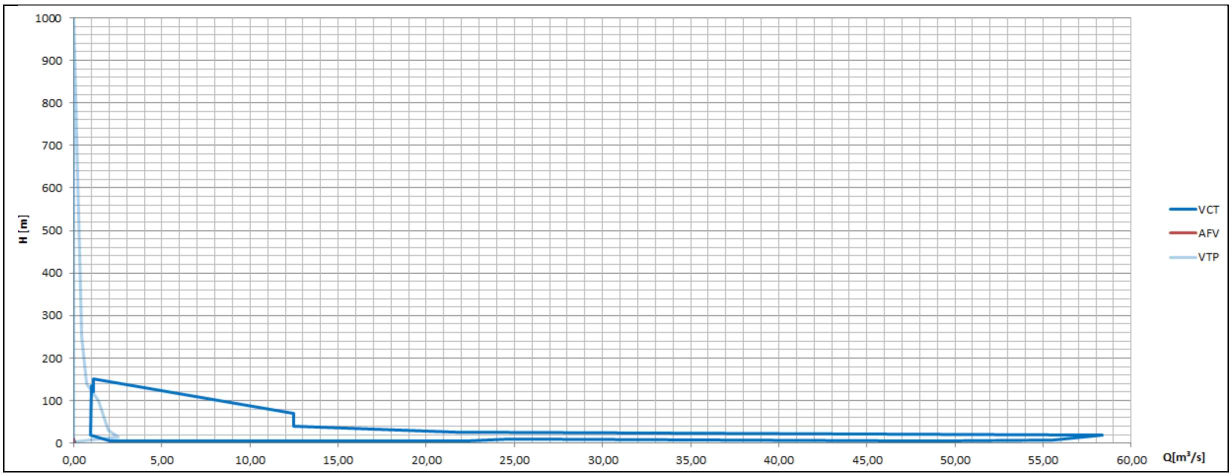


Diagram 56: Flowserve Q-H mapping 1

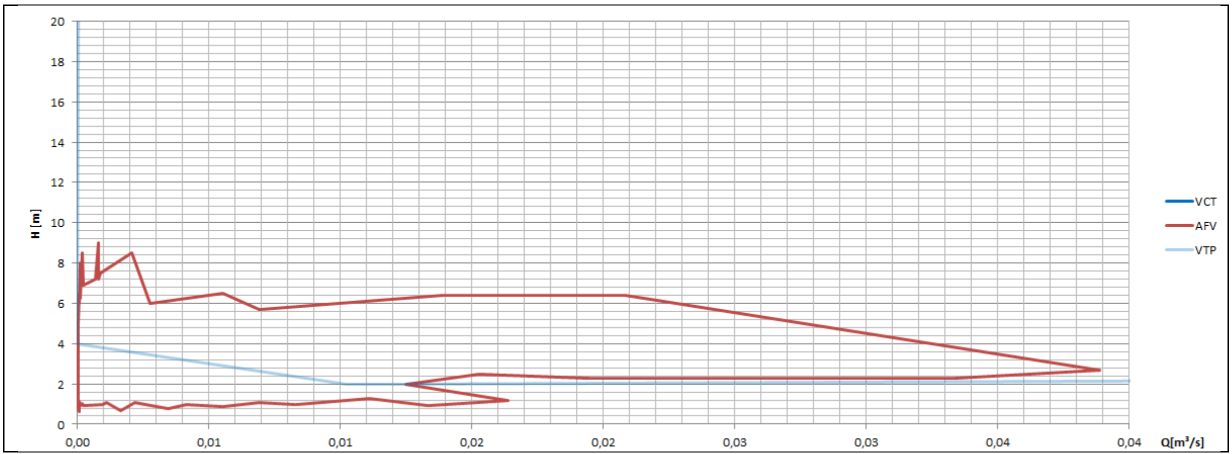


Diagram 57: Flowserve Q-H mapping 2

10.4 Appendix chapter 7

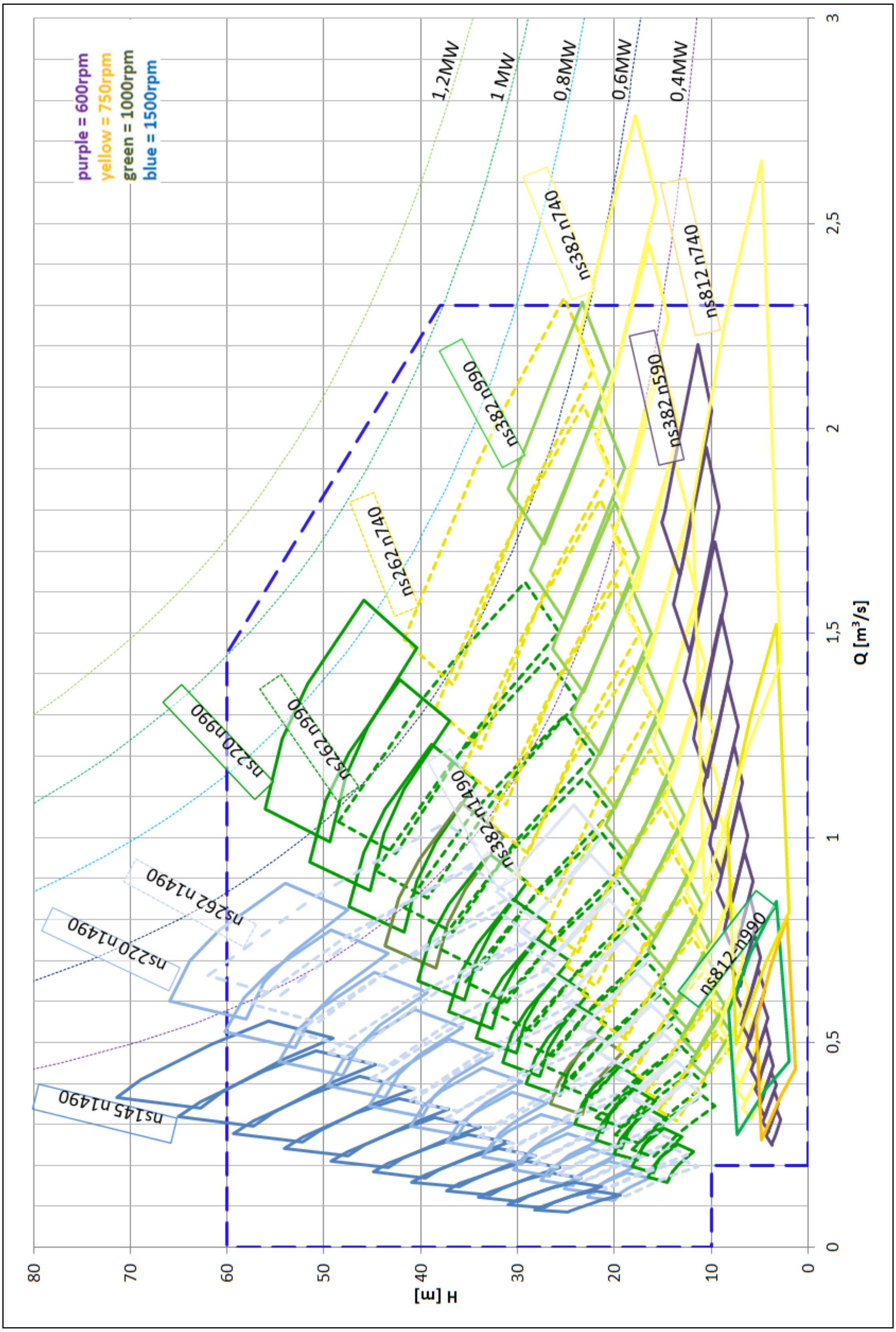


Diagram 58: Overall mapping with distinct sizes

ns145

n [rpm]	pump-#	pump definition	Q_{opt} [m³/s]	H_{opt} [m]	η	NPSH [m]	d [m]
1490	1	AVLSP0,305-145-1490_1	0,41	68,90	0,90	9,80	0,47
	1_trimmed		0,38	60,60			0,45
1490	2	AVLSP0,29-145-1490_2	0,35	62,80	0,90	8,94	0,45
	2_trimmed		0,33	55,24			0,43
1490	3	AVLSP0,28-145-1490_3	0,31	57,25	0,90	8,15	0,43
	3_trimmed		0,29	50,36			0,41
1490	4	AVLSP0,27-145-1490_4	0,27	52,18	0,90	7,43	0,41
	4_trimmed		0,25	45,90			0,39
1490	5	AVLSP0,25-145-1490_5	0,23	47,57	0,90	6,77	0,39
	5_trimmed		0,22	41,84			0,37
1490	6	AVLSP0,24-145-1490_6	0,20	43,36	0,90	6,17	0,37
	6_trimmed		0,19	38,14			0,35
1490	7	AVLSP0,23-145-1490_7	0,18	39,52	0,90	5,62	0,35
	7_trimmed		0,16	34,77			0,34
1490	8	AVLSP0,22-145-1490_8	0,15	36,03	0,90	5,13	0,34
	8_trimmed		0,14	31,69			0,32
1490	9	AVLSP0,21-145-1490_9	0,13	32,84	0,90	4,67	0,32
	9_trimmed		0,12	28,89			0,31
1490	10	AVLSP0,2-145-1490_10	0,12	29,94	0,90	4,26	0,31
	10_trimmed		0,11	26,33			0,29
1490	11	AVLSP0,19-145-1490_11	0,10	27,29	0,90	3,88	0,29
	11_trimmed		0,09	24,00			0,28

Table 18: ns145 pump sizes

ns220

n [rpm]	pump-#	pump definition	Q_{opt} [m^3/s]	H_{opt} [m]	η	NPSH [m]	d [m]
1490	1	AVLSP0,34-220-1490_1	0,77	60,7	0,90	10,33	0,51
	1_trimmed		0,72	53,4			0,49
1490	2	AVLSP0,32-220-1490_2	0,67	55,5	0,90	9,43	0,49
	2_trimmed		0,62	48,8			0,47
1490	3	AVLSP0,31-220-1490_3	0,58	50,3	0,90	8,56	0,47
	3_trimmed		0,54	44,3			0,44
1490	4	AVLSP0,3-220-1490_4	0,50	45,7	0,90	7,77	0,45
	4_trimmed		0,47	40,2			0,42
1490	5	AVLSP0,28-220-1490_5	0,44	41,8	0,90	7,10	0,43
	5_trimmed		0,41	36,8			0,40
1490	6	AVLSP0,27-220-1490_6	0,38	37,9	0,90	6,44	0,41
	6_trimmed		0,35	33,3			0,39
1490	7	AVLSP0,26-220-1490_7	0,33	34,4	0,90	5,85	0,39
	7_trimmed		0,31	30,3			0,37
1490	8	AVLSP0,24-220-1490_8	0,28	31,0	0,90	5,27	0,37
	8_trimmed		0,26	27,3			0,35
1490	9	AVLSP0,23-220-1490_9	0,24	28,0	0,90	4,76	0,35
	9_trimmed		0,22	24,6			0,33
1490	10	AVLSP0,22-220-1490_10	0,21	25,4	0,90	4,32	0,33
	10_trimmed		0,19	22,4			0,32
1490	11	AVLSP0,21-220-1490_11	0,18	23,2	0,90	3,94	0,32
	11_trimmed		0,17	20,4			0,30
1490	12	AVLSP0,2-220-1490_12	0,16	21,0	0,90	3,57	0,30
	12_trimmed		0,15	18,5			0,29
990	1	AVLSP0,47-220-990_1	1,37	51,7	0,90	8,78	0,71
	1_trimmed		1,27	45,5			0,68
990	2	AVLSP0,45-220-990_2	1,21	47,4	0,90	8,06	0,68
	2_trimmed		1,12	41,7			0,65
990	3	AVLSP0,43-220-990_3	1,07	43,7	0,90	7,43	0,66
	3_trimmed		0,99	38,4			0,62
990	4	AVLSP0,42-220-990_4	0,94	40,2	0,90	6,84	0,63
	4_trimmed		0,87	35,4			0,60
990	5	AVLSP0,4-220-990_5	0,84	37,1	0,90	6,31	0,60
	5_trimmed		0,77	32,6			0,57
990	6	AVLSP0,38-220-990_6	0,74	34,2	0,90	5,81	0,58
	6_trimmed		0,69	30,1			0,55
990	7	AVLSP0,37-220-990_7	0,66	31,6	0,90	5,37	0,56
	7_trimmed		0,61	27,8			0,53
990	8	AVLSP0,35-220-990_8	0,58	29,1	0,90	4,94	0,53
	8_trimmed		0,54	25,6			0,51
990	9	AVLSP0,34-220-990_9	0,51	26,8	0,90	4,56	0,51
	9_trimmed		0,48	23,6			0,49
990	10	AVLSP0,32-220-990_10	0,45	24,5	0,90	4,16	0,49
	10_trimmed		0,42	21,5			0,47
990	11	AVLSP0,31-220-990_11	0,39	22,2	0,90	3,78	0,47
	11_trimmed		0,36	19,5			0,44
990	12	AVLSP0,3-220-990_12	0,34	20,2	0,90	3,43	0,45
	12_trimmed		0,31	17,8			0,42
990	13	AVLSP0,28-220-990_13	0,29	18,4	0,90	3,14	0,43
	13_trimmed		0,27	16,2			0,40
990	14	AVLSP0,27-220-990_14	0,25	16,7	0,90	2,84	0,41
	14_trimmed		0,23	14,7			0,39
990	15	AVLSP0,26-220-990_15	0,22	15,2	0,90	2,58	0,39
	15_trimmed		0,20	13,4			0,37

Table 19: ns220 pump sizes

ns262

n [rpm]	pump-#	pump definition	Q_{opt} [m³/s]	H_{opt} [m]	η	NPSH [m]	d [m]
1490	1	AVLSP0,39-262-1490_1	0,91	46,9	0,89	9,73	0,51
	1_trimmed		0,84	41,2			0,49
1490	2	AVLSP0,37-262-1490_2	0,81	43,5	0,89	9,02	0,49
	2_trimmed		0,75	38,2			0,47
1490	3	AVLSP0,36-262-1490_3	0,72	40,0	0,89	8,31	0,47
	3_trimmed		0,67	35,2			0,45
1490	4	AVLSP0,35-262-1490_4	0,65	37,5	0,89	7,78	0,46
	4_trimmed		0,60	33,0			0,44
1490	5	AVLSP0,33-262-1490_5	0,57	34,1	0,89	7,07	0,44
	5_trimmed		0,52	30,0			0,42
1490	6	AVLSP0,32-262-1490_6	0,49	31,0	0,89	6,44	0,42
	6_trimmed		0,46	27,3			0,40
1490	7	AVLSP0,3-262-1490_7	0,43	28,3	0,89	5,88	0,40
	7_trimmed		0,40	24,9			0,38
1490	8	AVLSP0,29-262-1490_8	0,37	25,8	0,89	5,35	0,38
	8_trimmed		0,34	22,7			0,36
1490	9	AVLSP0,27-262-1490_9	0,32	23,5	0,89	4,88	0,36
	9_trimmed		0,30	20,7			0,35
1490	10	AVLSP0,26-262-1490_10	0,28	21,5	0,89	4,46	0,35
	10_trimmed		0,26	18,9			0,33
1490	11	AVLSP0,25-262-1490_11	0,25	19,6	0,89	4,06	0,33
	11_trimmed		0,23	17,2			0,32
1490	12	AVLSP0,24-262-1490_12	0,21	17,8	0,89	3,70	0,32
	12_trimmed		0,20	15,7			0,30
1490	13	AVLSP0,23-262-1490_13	0,19	16,2	0,89	3,37	0,30
	13_trimmed		0,17	14,3			0,29
990	1	AVLSP0,52-262-990_1	1,33	40,0	0,89	7,60	0,68
	1_trimmed		1,23	35,2			0,65
990	2	AVLSP0,49-262-990_2	1,18	36,9	0,89	7,01	0,66
	2_trimmed		1,09	32,4			0,62
990	3	AVLSP0,48-262-990_3	1,06	34,5	0,89	6,55	0,63
	3_trimmed		0,99	30,3			0,60
990	4	AVLSP0,46-262-990_4	0,94	31,7	0,89	6,03	0,61
	4_trimmed		0,87	27,9			0,58
990	5	AVLSP0,44-262-990_5	0,83	29,2	0,89	5,55	0,58
	5_trimmed		0,77	25,7			0,55
990	6	AVLSP0,42-262-990_6	0,73	26,9	0,89	5,11	0,56
	6_trimmed		0,68	23,6			0,53
990	7	AVLSP0,4-262-990_7	0,65	24,7	0,89	4,70	0,54
	7_trimmed		0,60	21,8			0,51
990	8	AVLSP0,39-262-990_8	0,57	22,8	0,89	4,33	0,52
	8_trimmed		0,53	20,0			0,49
990	9	AVLSP0,37-262-990_9	0,51	21,0	0,89	3,98	0,49
	9_trimmed		0,47	18,4			0,47
990	10	AVLSP0,36-262-990_10	0,45	19,3	0,89	3,67	0,47
	10_trimmed		0,41	17,0			0,45
990	11	AVLSP0,35-262-990_11	0,45	19,3	0,89	3,67	0,46
	11_trimmed		0,41	17,0			0,44
990	12	AVLSP0,33-262-990_12	0,40	18,1	0,89	3,43	0,44
	12_trimmed		0,37	15,9			0,42
990	13	AVLSP0,32-262-990_13	0,31	15,0	0,89	2,84	0,42
	13_trimmed		0,28	13,2			0,40

Table 20: ns262 pump sizes 1

ns262

n [rpm]	pump-#	pump definition	Q_{opt} [m³/s]	H_{opt} [m]	η	NPSH [m]	d [m]
740	1	AVLSP0,64-262-740_1	1,90	34,4	0,89	6,54	0,85
	1_trimmed		1,76	30,3			0,81
740	2	AVLSP0,61-262-740_2	1,69	31,8	0,89	6,04	0,81
	2_trimmed		1,56	28,0			0,77
740	3	AVLSP0,59-262-740_3	1,50	29,3	0,89	5,58	0,78
	3_trimmed		1,39	25,8			0,74
740	4	AVLSP0,57-262-740_4	1,33	27,1	0,89	5,15	0,75
	4_trimmed		1,23	23,8			0,71
740	5	AVLSP0,54-262-740_5	1,17	24,9	0,89	4,72	0,72
	5_trimmed		1,08	21,9			0,68
740	6	AVLSP0,52-262-740_6	1,00	22,4	0,89	4,25	0,68
	6_trimmed		0,92	19,7			0,65
740	7	AVLSP0,49-262-740_7	0,88	20,6	0,89	3,92	0,66
	7_trimmed		0,82	18,1			0,62
740	8	AVLSP0,48-262-740_8	0,80	19,2	0,89	3,66	0,63
	8_trimmed		0,74	16,9			0,60
740	9	AVLSP0,46-262-740_9	0,70	17,7	0,89	3,37	0,61
	9_trimmed		0,65	15,6			0,58
740	10	AVLSP0,44-262-740_10	0,62	16,3	0,89	3,10	0,58
	10_trimmed		0,58	14,4			0,55
740	11	AVLSP0,42-262-740_11	0,55	15,0	0,89	2,85	0,56
	11_trimmed		0,51	13,2			0,53
740	12	AVLSP0,4-262-740_12	0,48	13,8	0,89	2,63	0,54
	12_trimmed		0,45	12,2			0,51
740	13	AVLSP0,39-262-740_13	0,43	12,7	0,89	2,42	0,52
	13_trimmed		0,40	11,2			0,49

Table 21: ns262 pump sizes 2

ns382

n [rpm]	pump-#	pump definition	Q_{opt} [m³/s]	H_{opt} [m]	η	NPSH [m]	d [m]
1490	1	AVLSP0,47-382-1490_1	1,04	35,4	0,92	10,04	0,48
	1_trimmed		0,96	31,1			0,46
1490	2	AVLSP0,44-382-1490_2	0,88	31,7	0,92	8,99	0,48
	2_trimmed		0,96	31,1			0,46
1490	3	AVLSP0,42-382-1490_3	0,78	29,3	0,92	8,31	0,44
	3_trimmed		0,72	25,8			0,42
1490	4	AVLSP0,41-382-1490_4	0,69	27,0	0,92	7,66	0,42
	4_trimmed		0,64	23,7			0,40
1490	5	AVLSP0,39-382-1490_5	0,61	24,9	0,92	7,06	0,41
	5_trimmed		0,57	21,9			0,39
1490	6	AVLSP0,38-382-1490_6	0,55	23,0	0,92	6,53	0,39
	6_trimmed		0,50	20,3			0,37
1490	7	AVLSP0,36-382-1490_7	0,48	21,2	0,92	6,02	0,37
	7_trimmed		0,45	18,7			0,36
1490	8	AVLSP0,35-382-1490_8	0,42	19,5	0,92	5,53	0,36
	8_trimmed		0,39	17,1			0,34
1490	9	AVLSP0,33-382-1490_9	0,37	17,8	0,92	5,06	0,34
	9_trimmed		0,34	15,7			0,33
1490	10	AVLSP0,32-382-1490_10	0,33	16,5	0,92	4,67	0,33
	10_trimmed		0,30	14,5			0,31
990	1	AVLSP0,65-382-990_1	1,88	30,5	0,92	8,65	0,67
	1_trimmed		1,74	26,8			0,64
990	2	AVLSP0,63-382-990_2	1,68	28,2	0,92	8,01	0,65
	2_trimmed		1,55	24,8			0,62
990	3	AVLSP0,6-382-990_3	1,48	26,0	0,92	7,38	0,62
	3_trimmed		1,37	22,9			0,59
990	4	AVLSP0,58-382-990_4	1,32	24,1	0,92	6,83	0,60
	4_trimmed		1,22	21,2			0,57
990	5	AVLSP0,56-382-990_5	1,17	22,3	0,92	6,32	0,58
	5_trimmed		1,09	19,6			0,55
990	6	AVLSP0,53-382-990_6	1,05	20,6	0,92	5,85	0,56
	6_trimmed		0,97	18,1			0,53
990	7	AVLSP0,51-382-990_7	0,93	19,0	0,92	5,40	0,53
	7_trimmed		0,86	16,7			0,51
990	8	AVLSP0,49-382-990_8	0,83	17,6	0,92	5,00	0,51
	8_trimmed		0,77	15,5			0,49
990	9	AVLSP0,48-382-990_9	0,74	16,3	0,92	4,63	0,49
	9_trimmed		0,68	14,3			0,47
990	10	AVLSP0,46-382-990_10	0,66	15,1	0,92	4,28	0,48
	10_trimmed		0,61	13,3			0,45
990	11	AVLSP0,44-382-990_11	0,59	14,0	0,92	3,97	0,46
	11_trimmed		0,54	12,3			0,43
990	12	AVLSP0,42-382-990_12	0,52	12,9	0,92	3,67	0,44
	12_trimmed		0,48	11,4			0,42
990	13	AVLSP0,41-382-990_13	0,46	11,9	0,92	3,38	0,42
	13_trimmed		0,43	10,5			0,40

Table 22: ns382 pump sizes 1

ns382

n [rpm]	pump-#	pump definition	Q_{opt} [m³/s]	H_{opt} [m]	η	NPSH [m]	d [m]
740	1	AVLSP0,76-382-740_1	2,25	23,3	0,92	6,62	0,79
	1_trimmed		2,09	20,5			0,75
740	2	AVLSP0,73-382-740_2	2,00	21,5	0,92	6,11	0,76
	2_trimmed		1,85	18,9			0,72
740	3	AVLSP0,7-382-740_3	1,76	19,8	0,92	5,61	0,73
	3_trimmed		1,63	17,4			0,69
740	4	AVLSP0,68-382-740_4	1,58	18,4	0,92	5,22	0,70
	4_trimmed		1,46	16,2			0,67
740	5	AVLSP0,65-382-740_5	1,41	17,0	0,92	4,83	0,67
	5_trimmed		1,30	15,0			0,64
740	6	AVLSP0,63-382-740_6	1,25	15,8	0,92	4,48	0,65
	6_trimmed		1,16	13,9			0,62
740	7	AVLSP0,6-382-740_7	1,11	14,5	0,92	4,12	0,62
	7_trimmed		1,03	12,8			0,59
740	8	AVLSP0,58-382-740_8	0,99	13,4	0,92	3,81	0,60
	8_trimmed		0,91	11,8			0,57
740	9	AVLSP0,56-382-740_9	0,88	12,4	0,92	3,53	0,58
	9_trimmed		0,81	10,9			0,55
740	10	AVLSP0,53-382-740_10	0,78	11,5	0,92	3,27	0,56
	10_trimmed		0,72	10,1			0,53
740	11	AVLSP0,51-382-740_11	0,69	10,6	0,92	3,02	0,53
	11_trimmed		0,64	9,4			0,51
740	12	AVLSP0,49-382-740_12	0,62	9,9	0,92	2,80	0,51
	12_trimmed		0,57	8,7			0,49
740	13	AVLSP0,48-382-740_13	0,55	9,1	0,92	2,58	0,49
	13_trimmed		0,51	8,0			0,47
740	14	AVLSP0,46-382-740_14	0,49	8,4	0,92	2,39	0,48
	14_trimmed		0,45	7,4			0,45
740	15	AVLSP0,44-382-740_15	0,44	7,8	0,92	2,22	0,46
	15_trimmed		0,40	6,9			0,43
740	16	AVLSP0,42-382-740_16	0,39	7,2	0,92	2,05	0,44
	16_trimmed		0,36	6,4			0,42
740	17	AVLSP0,41-382-740_17	0,34	6,7	0,92	1,89	0,42
	17_trimmed		0,32	5,9			0,40
590	1	AVLSP0,76-382-740_1	2,25	23,3	0,92	6,62	0,79
	1_trimmed		2,09	20,5			0,75
590	2	AVLSP0,73-382-740_2	2,00	21,5	0,92	6,11	0,76
	2_trimmed		1,85	18,9			0,72
590	3	AVLSP0,7-382-740_3	1,76	19,8	0,92	5,61	0,73
	3_trimmed		1,63	17,4			0,69
590	4	AVLSP0,68-382-740_4	1,58	18,4	0,92	5,22	0,70
	4_trimmed		1,46	16,2			0,67
590	5	AVLSP0,65-382-740_5	1,41	17,0	0,92	4,83	0,67
	5_trimmed		1,30	15,0			0,64
590	6	AVLSP0,63-382-740_6	1,25	15,8	0,92	4,48	0,65
	6_trimmed		1,16	13,9			0,62
590	7	AVLSP0,6-382-740_7	1,11	14,5	0,92	4,12	0,62
	7_trimmed		1,03	12,8			0,59
590	8	AVLSP0,58-382-740_8	0,99	13,4	0,92	3,81	0,60
	8_trimmed		0,91	11,8			0,57
590	9	AVLSP0,56-382-740_9	0,88	12,4	0,92	3,53	0,58
	9_trimmed		0,81	10,9			0,55
590	10	AVLSP0,53-382-740_10	0,78	11,5	0,92	3,27	0,56
	10_trimmed		0,72	10,1			0,53
590	11	AVLSP0,51-382-740_11	0,69	10,6	0,92	3,02	0,53
	11_trimmed		0,64	9,4			0,51
590	12	AVLSP0,49-382-740_12	0,62	9,9	0,92	2,80	0,51
	12_trimmed		0,57	8,7			0,49
590	13	AVLSP0,48-382-740_13	0,55	9,1	0,92	2,58	0,49
	13_trimmed		0,51	8,0			0,47
590	14	AVLSP0,46-382-740_14	0,49	8,4	0,92	2,39	0,48
	14_trimmed		0,45	7,4			0,45
590	15	AVLSP0,44-382-740_15	0,44	7,8	0,92	2,22	0,46
	15_trimmed		0,40	6,9			0,43
590	16	AVLSP0,42-382-740_16	0,39	7,2	0,92	2,05	0,44
	16_trimmed		0,36	6,4			0,42
590	17	AVLSP0,41-382-740_17	0,34	6,7	0,92	1,89	0,42
	17_trimmed		0,32	5,9			0,40

Table 23: ns382 pump sizes 2

ns812

n [rpm]	pump-#	pump definition	Q_{opt} [m³/s]	H_{opt} [m]	η	NPSH [m]	d [m]
990	1_min angle	AVLSP0,41-812-990_1	0,72	5,88	0,88	7,39	0,41
	1_max angle		0,39	4,35	0,85	4,05	
740	1_min angle	AVLSP0,45-812-740_1	0,69	3,87	0,88	4,86	0,45
	1_max angle		0,37	2,87	0,85	2,66	
740	2_min angle	AVLSP0,55-812-740_2	1,30	5,91	0,88	7,43	0,55
	2_max angle		0,70	4,38	0,85	4,07	
740	3_min angle	AVLSP0,66-812-740_3	2,27	8,57	0,88	8,70	0,66
	3_max angle		1,21	6,34	0,85	5,90	

Table 24: ns812 pump sizes

Bibliography

American Petroleum Institute, 2010. *Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries.* ANSI/API Standard 610, 11th ed. Washington, API Publishing Services.

Andritz – Applications, 2017. *Applications.* [Online] Available at: https://www.andritz.com/pumps-en/industries/waste-water-management [Accessed 09 10 2017].

Andritz – Vertical Line Shaft Pump, *Vertical Line Shaft Pump.* [Online] Available at: https://www.andritz.com/products-en/group/pumps/vertical-line-shaft [Accessed 21 10 2017].

Andritz – Vertikale Rohrgehäusepumpen, *Vertikale Rohrgehäusepumpen.* Graz, Andritz.

Andritz, internal source.

ANSI/HI 1.3, 2013. *American national standard for rotodynamic centrifugal pumps for design and application ANSI/HI 1.3-2013.* 1.3-2013 ed. New Jersey, Hydraulic Institute.

Bloch, H. P. & Budris, A. R., 2010. *Pump – User's Handbook: Life Extension.* 3rd ed. Lilburn, The Fairmont Press, Inc..

Brenn, U.-P. D.-I. h. & Meile, A.-P. D. D. W., 2013. *Strömungslehre und Wärmeübertragung 1.,* Graz, Institut für Strömungslehre und Wärmeübertragung.

DIN EN 10253-2, 2008. *DIN EN 10253-2 Formstücke zum Einschweißen.* Berlin, Deutsches Institut für Normung e.V..

Flowserve corporation – VTP, 2014. *VTP – Vertical turbine pump productfolder,* Texas, Flowserve corporation.

Flowserve VCT, 2008. *VCT-Vertical Mixed Flow Circulation Pumps Productfolder,* Texas, Flowserve Corporation.

Flowserve, 2008. *AFV - Vertical Axial Flow Pumps Productfolder,* Texas: Flowserve Corporation.

Gülich, J. F., 2013. *Kreiselpumpen – Handbuch für Entwicklung, Anlagenplanung und Betrieb.* 4th ed., Springer Vieweg.

Hebenstreit, D. J., 2009. *Interne Produktstandardisierung und Plattformen.* [Online] Available at: http://faim.tf.uni-freiburg.de/bbl-hebenstreit.pdf [Accessed 13 08 2017].

Hellmann, P. D.-I. D.-H., 2011. *KSB-Kreiselpumpen-Lexikon.* Frankenthal, KSB Aktiengesellschaft.

Hofbauer, G. & Hellwig, C., 2009. *Professionelles Vertriebsmanagement.* 2nd ed. Erlangen, Publicis Publishing.

Hydraulic Institute – 9.8, 1998. *American national standard for pump intake design.* ANSI/HI 9.8–1998 ed. New Jersey, Hydraulic Institute.

Idelchik, I., 1986. *Handbook of Hydraulic Resistance.* 2nd ed. New York, Hemispehre Publishing Corporation.

Infiniti Research Ltd., 2017. *Market Research Centrifugal Pumps.* London.

Jaberg, H., 2012. *Strömungsmaschinen Grundlagen,* TU Graz.

Johnson, G., Scholes, K. & Whittington, R., 2009. *Fundamentals of strategy.* Essex, Pearson Education Limited.

KSB – PNW, 1999. *KSB PNW product folder,* KSB.

KSB – Rohrgehäusepumpen von KSB, 2017. *Rohrgehäusepumpen von KSB.* [Online] Available at: https://shop.ksb.com/ims_docs/16/169A6A5232C7F718E10000000AD5062A.pdf [Accessed 16 10 2017].

KSB – SNW, 2000. *KSB SNW product folder,* KSB.

KSB, 2005. *Auslegung von Kreiselpumpen.* 5th ed. Frankenthal: KSB Aktiengesellschaft.

Sigloch, H., 2009. *Strömungsmaschinen - Grundlagen und Anwendung,* Hanser.

Stepanoff, A. J., 1957. *Radial- und Axialpumpen - Theorie, Entwurd, Anwendung.* 2nd ed. New York, John Wiley & Sons, Inc..

Sterling SIHI, 2000. *Grundlagen für die Planung von Kreiselpumpenanlagen.* 7th ed. Itzhoe, Sterling SIHI GmbH.

Sulzer – Biomass power plant, 2017. *Biomass power plant.* [Online] Available at: https://www.sulzer.com/en/Industries/Power-Generation/Biomass-Waste [Accessed 16 10 2017].

Sulzer – CWP for coal- and oil-fired power plants, 2017. *CWP for coal- and oil-fired.* [Online]
Available at: https://www.sulzer.com/en/Industries/Power-Generation/Coal-fired/Cooling-Water-Pump
[Accessed 16 10 2017].

Sulzer – CWP for gas-fired power plants, 2017. *CWP for gas-fired power plants.* [Online]
Available at: https://www.sulzer.com/en/Industries/Power-Generation/Gas-fired/Cooling-Water-Pump
[Accessed 16 10 2017].

Sulzer – CWP for geothermal, 2017. *CWP for geothermal.* [Online]
Available at: https://www.sulzer.com/en/Industries/Power-Generation/Geothermal
[Accessed 16 10 2017].

Sulzer – Desalination, 2017. *Desalination.* [Online]
Available at: https://www.sulzer.com/en/Industries/Water/Desalination
[Accessed 16 10 2017].

Sulzer – sales presentation, n.d. *Sulzer – SJT/SJM/SJP large vertical pumps.* s.l.:Sulzer.

Sulzer – Solar power generation, 2017. *Solar power generation.* [Online]
Available at: https://www.sulzer.com/en/Industries/Power-Generation/Solar
[Accessed 16 10 2017].

Sulzer – Vertical sump pumps, 2017. *Sulzer – Vertical sump pumps.* [Online]
Available at: https://www.sulzer.com/en/Products-and-Services/Pumps-and-Systems/Vertical-Pumps/Vertical-Sump-Pumps
[Accessed 13 09 2017].

Sulzer – Water intake, -transportation and Irrigation, 2017. *Water intake, -transportation and Irrigation.* [Online]
Available at: http://www.sulzer.com/en/Industries/Water/Water-Intake-and-Transportation
[Accessed 16 10 2017].

Sulzer SJM, 2015. *Sulzer SJM product folder,* Sulzer Ltd..

Sulzer SJT, 2008. *Sulzer SJT product folder,* Sulzer Pumps Ltd..

Voith – Schrift 2269, *Ausführungsbeispiele von Turbinen und Absperrorganen,* Voith.

Vorbach, S., 2015. *Unternehmensführung und Organisation.* Graz, Facultas Verlags- und Buchhandels AG.

weltkarte.at, 2017. *weltkarte.at.* [Online] Available at: https://www.weltkarte.com/typo3temp/images/topographie-china.png [Accessed 09 10 2017].

Wittel , H., Muhs , D., Jannasch, D. & Voßiek, J., 2009. *Roloff/Matek Maschinenelemente.* 19th ed. Wiesbaden, Vieweg + Teubner.

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List of abbreviations

VLSP	Vertical Line Shaft Pump
VTP	Vertical Turbine Pump
HPU	Hydraulic Pump Unit
$NPSH_{av}$	Net Positive Suction Head available
$NPSH_{req}$	Net Positive Suction Head required
HVACR	Heating Ventilation Air Conditioning and Refrigeration
SWRO	Seawater Reverse Osmosis
CWP	Cooling Water Pump
FWP	Feed Water Pump
CEP	Condensate Extraction Pump
BFP	Boiler Feed Pump
FC	Flood Control
FGD	Flue Gas Desulphurization
API	American Petroleum Institute
NFPA	National Fire Protection Association
IP	Impeller
HI	Hydraulic Institute
BS	Bearing Support