STRUCTURAL ANALYSIS AND OPTIMIZATION OF STIFFENERS IN VERTICAL AGITATORS TO ENHANCE WELDING PRODUCTION

Lappeenranta–Lahti University of Technology LUT

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Jere Serell

Examiners: Associate Professor Tuomas Skriko

Professor Timo Björk

ABSTRACT

Lappeenranta–Lahti University of Technology LUT

LUT School of Energy Systems

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Jere Serell

Structural analysis and optimization of stiffeners in vertical agitators to enhance welding production

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Professor Timo Björk

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This study analysed the vertical agitators of Mamec Oy, focusing on components with various stiffeners, namely shafts, paddles, and hubs. The structures were initially examined without stiffeners to understand their behaviour and later analysed with the currently used stiffeners. The focus was on analyzing the structures under static loads, but fatigue was also discussed. Based on the results, proposed improvements to the stiffeners were evaluated based on their effectiveness and cost-efficiency of the manufacturing and then compared to the current solutions. Finite element analysis was mainly utilized, but experimental tests were included for the analysis of the hubs.

The shafts' current stiffeners were found to be suboptimal and alternate designs offered comparable solutions. Removing the stiffeners altogether proved beneficial but relies on the critical flange weld's quality. For paddles, alternative solutions did not provide clear enhancements to the load bearing capacity but increasing the thickness of the paddles proved to be a feasible alternative to the stiffeners without excessive added costs.

Finite element analyses of the hubs yielded inconclusive results due to difficulties in determining the friction connection's quality with the shafts. Experimental tests validated the proposed design, which reduced welding and manufacturing steps significantly.

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Tässä työssä analysoitiin Mamec Oy:n pystysekoittimia, keskittyen jäykistettyihin rakenteisiin, joihin kuuluivat niiden akselit, lavat sekä navat. Aluksi rakenteita tutkittiin ilman vahvikkeita, jonka jälkeen nykyisten vahvikkeiden vaikutuksia tarkasteltiin. Rakenteita tutkittiin staattisen kuormituksen perusteella, mutta väsymistä myös pohdittu. Näiden tulosten perusteella rakenteille kehitettiin vaihtoehtoisia ratkaisuja, joita verrattiin nykyisiin rakenteisiin niiden toimivuuden sekä valmistuksen kustannustehokkuuden kannalta. Elementtimenetelmä toimi pääasiallisena tutkimusmenetelmänä, mutta kokeellisia testejä sisällytettiin napojen tarkasteluun.

Akseleiden nykyiset vahvikkeet osoittautuivat epäedullisiksi sekä vaihtoehtoiset ratkaisut olivat niihin verrattavia. Vahvikkeiden poisto osoittautui suotuisaksi vaihtoehdoksi, mutta tällöin kestävyys riippuu laipan ja putken välisestä hitsistä. Lavoille ei löydetty selvää parempaa vaihtoehtoista ratkaisua, mutta lavan paksuuden kasvattaminen osoittautui varteenotettavaksi vaihtoehdoksi, joka ei lisää kustannuksia huomattavasti.

Napojen laskennalliset analyysit eivät tuottaneet selkeitä tuloksia, joiden perusteella napojen ja akselien välistä kitkaliitosta olisi voitu arvioida. Kokeelliset testit kuitenkin tukivat tuloksia, joiden avulla päästiin kustannustehokkaampaan ratkaisuun.

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Jere Serell

Lohja 13.7.2023

SYMBOLS AND ABBREVIATIONS

Roman characters

F_s	stalling side load
I_{sh}	length of shaft
M	torque
M_b	bending moment

Greek characters

ε	elongation
σ	stress

Abbreviations

CFD	Computational Fluid Dynamics
DFM	Design For Manufacturing
EEUA	Engineering Equipment Users Association
FEM	Finite Element Method
FEA	Finite Element Analysis
NDT	Non-destructive Testing
RBE2	Rigid Body Element 2

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Appendix 1. Cross section of the larger shaft

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1 Introduction

Vertical agitators are widely used in many industrial applications to agitate various types of liquids and substances. The design and construction of these machines are critical to achieving the desired mixing performance while ensuring safety, reliability, and durability. One important aspect of the agitator's design is the use of stiffening components in the shafts and paddles to prevent excessive deformation and failure under different loading conditions. In this thesis, the stiffening components of vertical agitators are analysed, and their effectiveness is evaluated in improving the agitator's performance and load bearing capacity. A vertical agitator manufactured by Mamec is shown in Figure 1.

Figure 1. A top-mounted vertical agitator manufactured by Mamec.

The results of the analysis will be presented and discussed, and recommendations for improving the stiffeners will be proposed. By understanding the effects and performance of the stiffening components, the aim is to contribute to the development of more cost-efficient production while maintaining the reliability of the products.

1.1 Background and motivation

This master's thesis is done to optimise the welding production of vertical agitator at Mamec Oy. The optimisation is achieved through analysing and further developing the various stiffeners and braces used in the products. For the most part, the company has relied on previous knowledge and practical testing when adding stiffeners to the structures, which has resulted in functioning solutions, but the deeper understanding of their effects is lacking. This hinders the ability to analyse their effects, which makes any further development difficult. Although the structures experience variable loading, fatigue has not been a problem so far and static loading is considered critical. By analysing the structures and used stiffeners, it is possible to propose realistic solutions in which the structures and production can be optimised further.

1.2 Objectives

The goal of the thesis is to study the current structures, the used stiffeners and propose improvements to achieve more cost-efficient solutions. Optimization of the structures can lead to increased performance, less manufacturing steps, minimize welding and allow less material to be used overall.

This research attempts to answer these questions:

- What are the most critical parts of the studied products?
- How effective are the currently used stiffeners?
- How can the stiffeners be improved to increase cost-effectiveness of the production?

The analyses will be mainly based on the finite element method (FEM) but also some experimental tests will be conducted, to increase the validity of the results.

1.3 Limitations

The research focuses solely on the shafts and paddles of the vertical agitators and excludes any other component, as the focus is on components which use stiffeners commonly. Various austenitic stainless steels are utilized by the company, but EN 1.4404 material grade is most regularly used and will be used as the base material in the analyses. Although most of the used materials outside of EN 1.4404 have similar or better material properties, it must be noted that the results should not be blindly applied to all the other materials as they behave differently and can have significant differences in weldability and other fabrication related aspects. Automation of the welding production will not be considered in this study due to large variances in products with small manufacturing quantities.

Assessing the loading of agitators is a challenging task due to its reliance on the use case of the agitator and the rheological aspects of the agitated substance. Computational fluid dynamics (CFD) is commonly utilized, but even then, the results are limited to the specific process and substance studied, which cannot be applied broadly to other processes with different substances. To mitigate this issue, simplified but conservative loadings are used which exaggerate the results. Pre-existing CFD data is utilized to gain a deeper understanding of the realistic loading and to evaluate the accuracy of the simplified approach.

In the design of agitators, resonance and vibrations must be considered. The assessment needs to be considered on a case-by-case basis, which requires an understanding of the specific application details. However, due to the broad scope of this study which does not focus on specific applications, vibrations will not be considered. The same limitations apply to the assessment of fatigue capacity, but fatigue will still be considered to some extent in the design of the stiffeners. Although fatigue has not been problematic yet, future proofing the structures to withstand fatigue better, can prove to be beneficial, especially as the company grows and takes on larger and more demanding projects.

The analyses primarily rely on FEM, which can provide realistic results but should not be blindly applied to the real-world products without deeper inspection and testing. The analyses tend to be idealistic and do not account for the variance and imperfections that are often experienced in reality.

2 Theory

This chapter introduces the general design of vertical agitators with specific focus on the studied parts. The realistic loading of the structures is defined after which commonly used design methodologies and simplistic loading types are examined. Finally, the design for manufacturing (DFM) and welding related aspects are considered.

2.1 Vertical agitators

Various names are used for mixing equipment such as agitator, mixer, blender or stirrer. The used terminology varies between industries and is often affected by the mixing processes they are used in. This thesis focuses on industrial scale mixers which are often referred to as agitators in the mixing industry. Agitation as a term is also broader, as it includes processes that do not fall strictly under the terminology of mixing, but nonetheless these products can be used for. Some of these processes are solids suspension, liquid-liquid dispersion, gas dispersion and mass or heat transfer. (Abatzoglou, Cullen, Rielly & Romanach 2015, p. 312.)

Vertical agitators are mechanical devices used for mixing or agitation of process fluids and other substances in variety of applications. These agitators are typically top-mounted to vessels, providing easy installation. While less common, it is also possible to mount the agitators at the bottom of vessels, which can be particularly advantageous in taller tanks where a shorter shaft length is required. When bottom mounting, magnetic drives can be utilised, although the torque capacity is significantly limited compared to other drive types (Atiemo-Obeng, Kresta & Paul 2004, p. 1286). Vertical agitators are most commonly employed in wastewater treatment, biogas plants and pulp and paper industry but are also used in paint industry, resin mixing, adhesive blending processes and many other industries (Mamec 2023, p. 31).

In Figure 2, a generic structure of a top-mounted agitator installed on a mixing vessel is presented, displaying the various components. The main drive unit utilised to produce torque is typically an electric motor. However, direct attachment of the motor to the shaft is generally avoided due to inappropriate rotational speeds or torque, so a reducer, such as a gearbox or belt transmission, is commonly employed to obtain the suitable speed and torque

combination. Additionally, variable-frequency drives can be used to control the output of the motor which can also be adjusted during use. Couplings that are supported by bearings connect the main shaft to the power drive. Various types of seals are employed to pressurise the vessel and prevent leaks when necessary. (Atiemo-Obeng, Kresta & Paul 2004, p. 1248.)

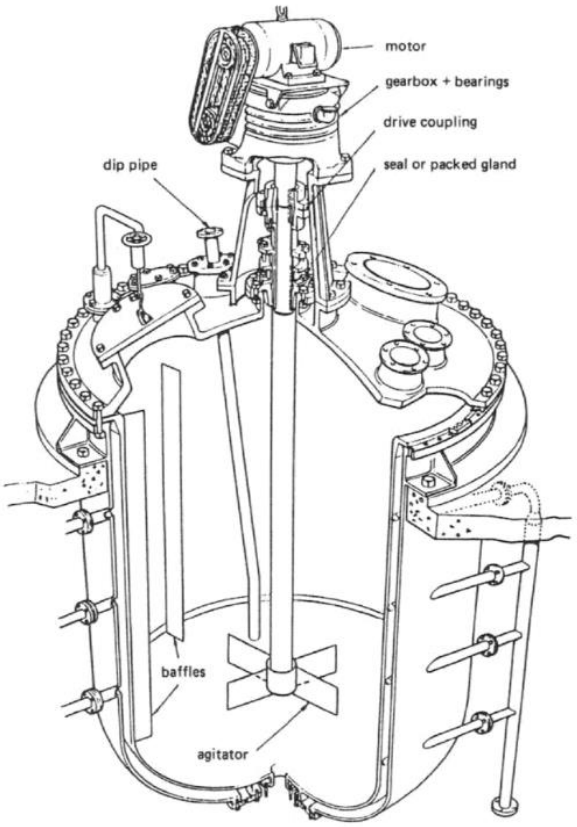

Figure 2. A vessel with a top entering agitator which displays the components commonly found in agitators (Edwards, Harnby & Nienow 1997, p. 252).

Despite the relative simplicity of the agitator design, it is essential to tailor it to the specific process and application it will be used in, resulting in significant design variations. The required mixing process and the rheological properties of the process substance often dictate the selection of impeller type, diameter of the impeller, and rotational speed, which in turn sets the requirements for many of the other mechanical aspects.

Various types of impellers are utilised in vertical agitators depending on the specific application, with the key selection criteria being the viscosity and density of the substance

alonsgside the desired mixing process. Some substances, such as non-Newtonian fluids, present additional challenges in defining viscosity, requiring an understanding of their unique behaviour. Some of the most used impeller types in vertical agitators include blades, paddles, anchors, turbines, and propellers (Edwards et al. 1997, p.120-121). Various impellers are presented in Figure 3.

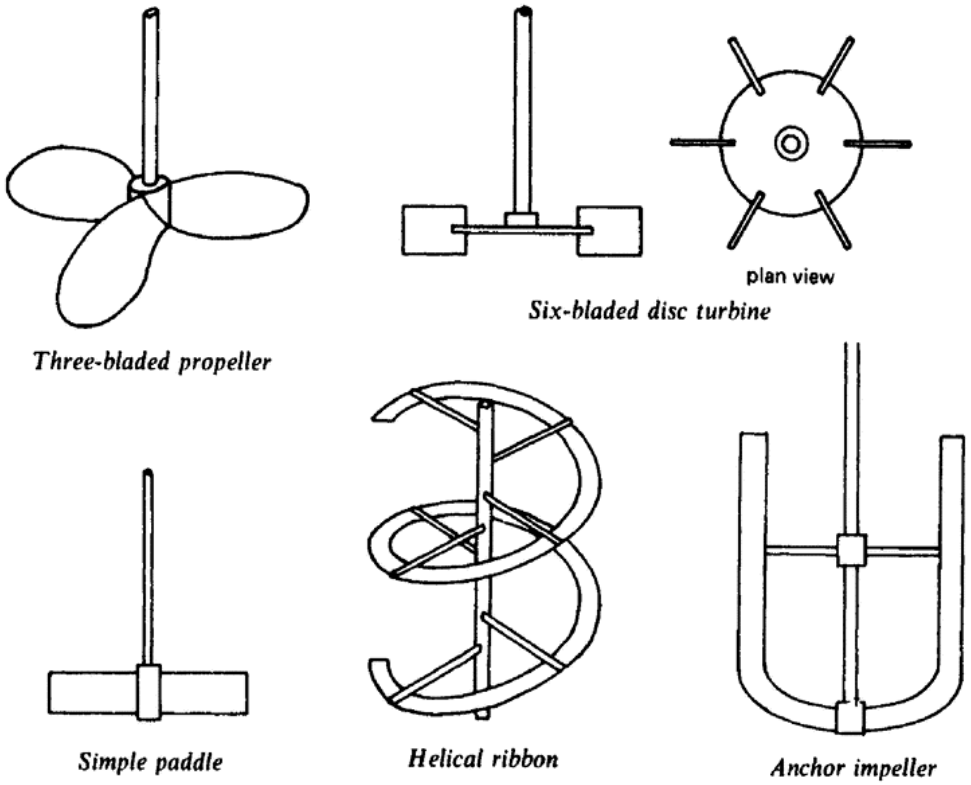

Figure 3. Several impellers used in vertical agitators (Edwards et al. 1997, p. 121).

Impellers are predominantly designed to produce either axial or radial flow, which affects the flow patterns that are formed inside the tank. The differences between the two flow patterns are depicted in Figure 4. Axial flow is often preferred for simple mixing processes as it causes the substance to mix between different layers more efficiently. (Atiemo-Obeng et al. 2004, p. 347.) The paddles which are studied in this thesis are designed to produce axial flow.

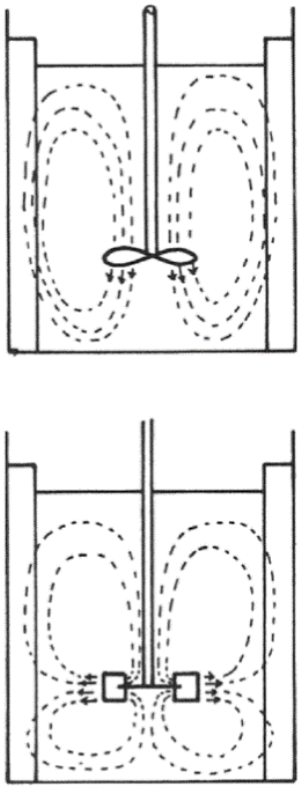

Figure 4. Axial and radial flow patterns (Edwards et al. 1997, p. 146).

Baffles can be added to the vessels as they cause disturbances to the flow of the substance, which deters solid body rotation and increases the mixing between the different layers. Baffles are not usually needed if the setup is asymmetric or if the vessel has sharp corners as they naturally introduce disturbances to the flow. (Atiemo-Obeng et al. 2004, p. 347.)

2.1.1 Shafts, paddles and hubs

This thesis centres specifically on the paddles, hubs which the paddles are welded to, and shafts found in the company's vertical agitators, thus warranting a more detailed introduction. In total, two different shaft sizes and four different hub sizes for paddles are examined. A comprehensive list outlining the dimensions of each shaft and paddle is provided in Table 1. For shafts, the diameter is measured as the outer diameter of the pipe, whereas for paddles, the diameter is defined as the size of the shaft to which the hub is attached to. The minimum thickness is determined as the smallest thickness used for the shaft pipes and paddle plates. Additionally, the maximum length for shafts is defined as the

total vertical length of the shaft, while for paddles, it is calculated as the diameter of two combined paddles which represents the entire mixing element. Some of the dimensions have been rounded to the closest integer for clarity.

Table 1. Defined dimensions for the shafts and paddles.

	Shafts (mm)		Paddles (mm)			
Diameter of shaft	219	273	114	168	219	273
Minimum thickness of pipe or plate	4	4	8	10	12	12
Maximum length	10 000	10 000	2 500	3 000	4 000	5 000

A paddle manufactured by Mamec is displayed in Figure 5, highlighting the different types of stiffeners employed. Typically, two types of stiffeners are incorporated on the hubs, including small ribs welded alongside the bends of the hub plate and larger flanges welded at the ends of the hub. Based on the size of the paddle, a third larger rib may be added to the middle of the hub as well. In addition, a larger box stiffener can be added to the suction side of the paddle. For these types of paddles, two hubs form a mixing element, and multiple mixing elements can be installed along the shaft. The two hubs are fastened together with a bolt connection and tightened with sufficient torque to secure them in place on the shaft via friction.

Figure 5. Paddles which display the different types of stiffeners used: 1) box, 2) small rib, 3) large end flange and 4) large middle rib.

Less stiffeners are used for shafts, but for the larger shaft sizes, ribs are added to the end of the shaft which attaches to the drive coupling. Two types of configurations are used, one with four thicker ribs and the other with ten thinner ribs. The two different configurations are displayed in a Figure 6.

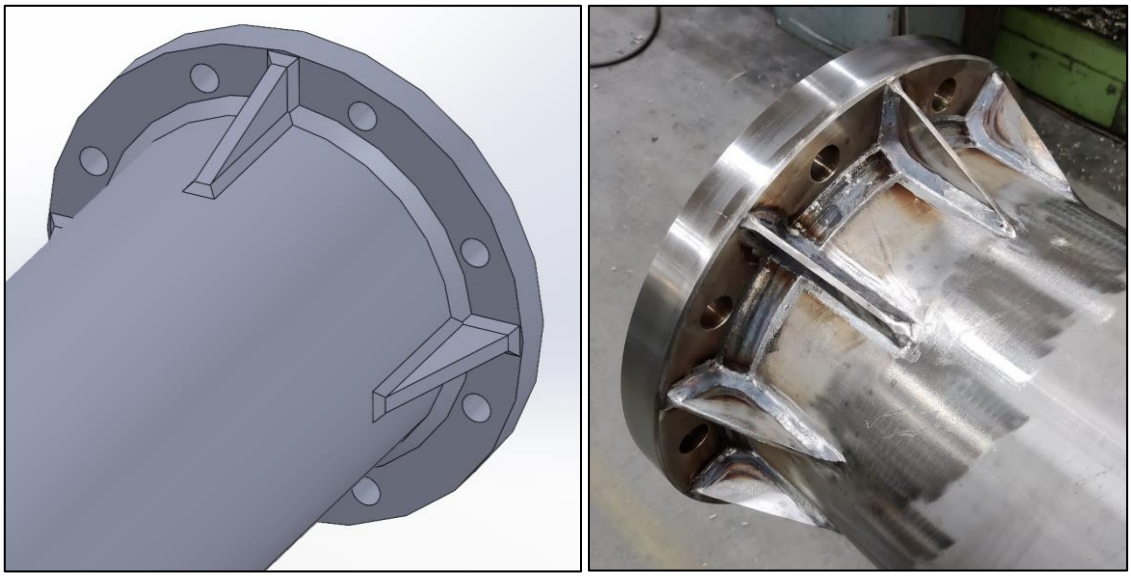

Figure 6. Two types of stiffener configurations used for shafts.

The configuration with four stiffeners is strictly used with the smaller 219 mm diameter shaft and the other with the larger 273 mm diameter shafts. The two shafts also have different flange designs, the smaller shaft has a ring welded onto the pipe and the inner part of the pipe is separately welded shut, whereas the larger shaft flange consists of a single large disc onto which the pipe itself is welded on to.

2.1.2 Loading types of the structure

Realistic loading of the structure is presented to provide a foundation for the analysis and to allow for comparison with the simplified loading that is used in the analysis. Figure 7 illustrates the various loads acting on a top-mounted vertical agitator. The primary load on the shaft is generated by the power drive as torque, but bending moments also occur, particularly when forces acting on the impellers are momentarily significant. The criticality of the bending moment depends heavily on the length of the shaft alongside the hydraulic forces exerted to the impellers. (Atiemo-Obeng et al. 2004, p. 1269.) High momentary bending moments can for example occur if a larger debris hits the impellers but is heavily dependent on the application. Frictional forces are also present in the drive gears, seal, and bearings (Uhl & Gray 1967, p. 289).

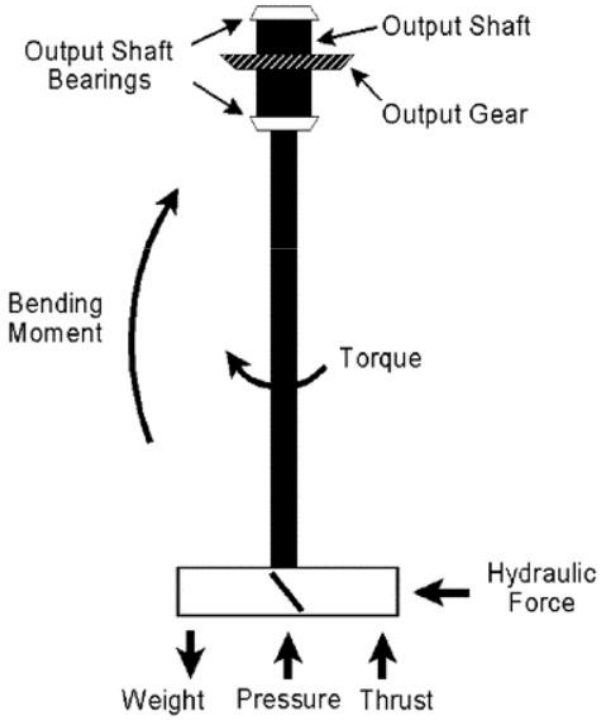

Figure 7. Depiction of all the loads experienced by a vertical agitator (Atiemo-Obeng et al. 2004, p. 1269).

The impellers can experience various loads which heavily depend on the type and design of the impeller. These loads are generated by the hydraulic forces present during the agitation process and are largely affected by the rheological aspects of the process and the substance. Moreover, the pressure generated by the fluid can cause an upward force on the structure if the tank is fully sealed and pressurised. (Atiemo-Obeng et al. 2004, p. 1269.) Other axial loads are generated by the thrust created by the axial flow, and the weight of the impeller and shaft (Edwards et al. 1997, p. 275). The axial and pressure forces are often insignificant enough to not be considered in basic calculations, but if the paddles are situated near the bottom of the mixing vessel, these loads can become more significant. In this case the hubs are attached to the shaft with a friction connection which requires a significant bolt preload to be applied, introducing considerable stresses to the shaft and hubs as well.

For specific impellers, CFD analysis or in-depth physical experiments are required if realistic loading of a paddle is to be determined. In this case however, only previously conducted analyses are utilised to demonstrate how the paddles are generally loaded. Figure 8 displays

pressure distributions that are experienced by two different paddles with similar geometries to the paddles examined in this study.

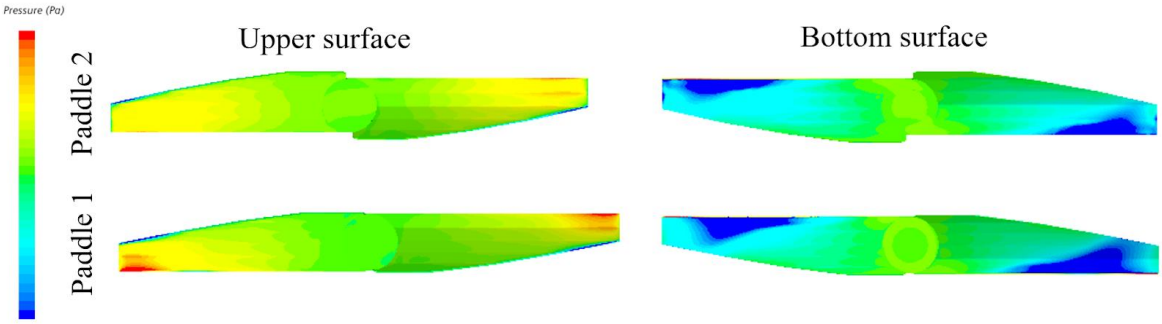

Figure 8. CFD analysis conducted displaying the pressures on both top and bottom surfaces of two different paddles. Data provided by Mamec Oy.

On the top surfaces of the paddles, the pressure concentrates around the tips of the paddles on the leading edge, but on the top and bottom surfaces, larger surface areas are also visibly loaded. Water was used as the substance and a typical tip speed of 3,5 m/s for the paddles was utilised, which is not representative of every application but provides a good general understanding of the loading. Any changes in the flow cause constant variation to the loading, which depends heavily on the process itself, and if solid objects are included in the process, higher temporary loading can be expected when they hit the paddles.

A lot of variances can also be found in the loading of the shaft. The highest torque amount is most likely to be experienced during the startup of the agitator, as it is accelerated to its desired rotational speed. Assuming that a constant speed is maintained, no significant changes in the torque should be experienced and a relatively constant torque is sustained. The peak torque can be 2,5 times the rated torque value, but ultimately depends on the specific motor (Edward, Harnby & Nienow 1997, p. 254). A simple descriptive graph of the experienced torque is displayed in Figure 9.

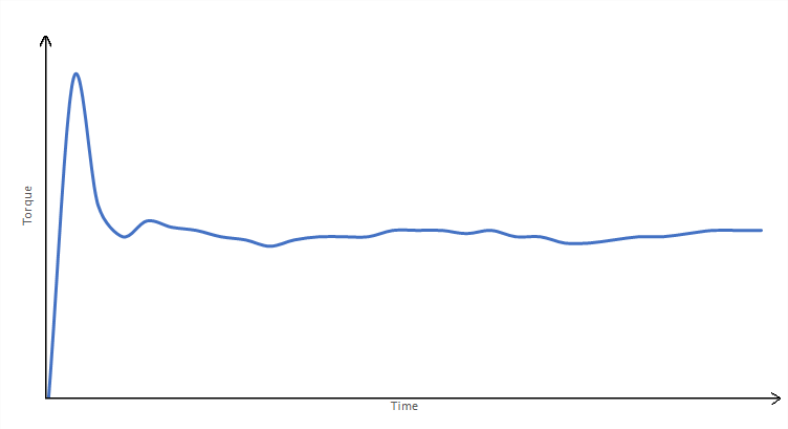

Figure 9. A simplistic depiction of torque loading that a shaft could experience.

In ideal conditions, the bending moment experienced by the shaft would be minimal as both paddles would be loaded equally. But as described with the loading of the paddles, a lot of variance is experienced during the process, which causes imbalances between the paddles. This exerts itself as momentary bending moment on the shaft until a relative balance is reached again. Realistically, small amounts of bending moment is experienced constantly but any significant peaks would be caused by solid objects hitting a blade or the process itself causing a notable brief imbalance between the paddles. A simple depiction of the variance of the bending moments experienced by a shaft is shown in Figure 10.

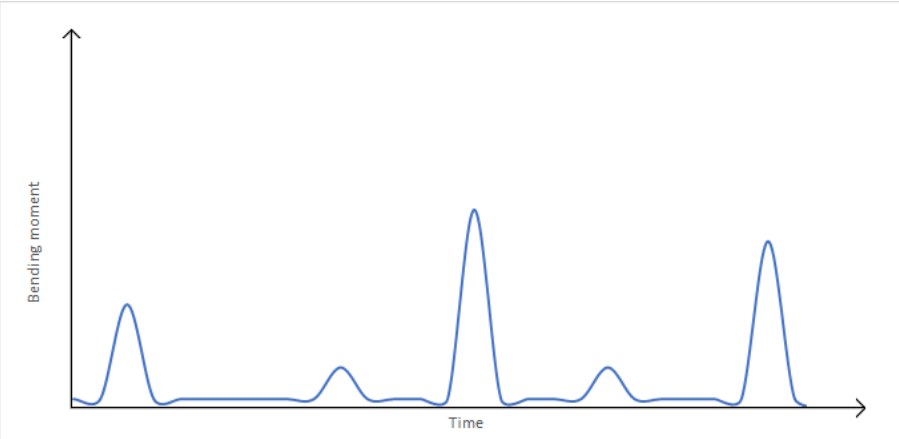

Figure 10. A simplistic depiction of bending moment loading that a shaft could experience.

Overall, reasonably static loading can be assumed with the torque except for the startup and any possible jamming of the agitator. Bending moment on the other hand is usually experienced only momentary in higher peaks which makes it harder to be accounted for if fatigue is considered. When considering the loading, the significance of the torque and bending moment is affected by the construct of the agitator. With a long agitator shaft, the effects of the forces causing bending moment are significantly greater than for shorter shaft, whereas the torque is more affected by the paddle size and required tip speed.

2.1.3 Simplified loading and design methodologies

As previously noted, a difficult aspect in this thesis is to analyse the structures in a way where the results can be broadly applied to the products regardless of its application. The loading and structural characteristics of agitators vary considerably from one application to another, making it difficult to widely apply design methods that rely on specific details of the agitator and its application, including its rheological and mechanical aspects. Furthermore, many design methods that do not employ CFD or FEM tend to provide overly conservative results with large safety margins. The specific methods used tend to differ from company to company as no large-scale industry standard exists.

The Engineering Equipment Users Association's (EEUA) method, a long-standing technique for sizing shafts in vertical agitators, relies on the assumption that the paddle will become overloaded or jammed to the point until its motor stalls. The shear stresses present in this situation are used to calculate the appropriate shaft size. (Edwards et al. 1997, p. 256.)

This leads to a design philosophy where the motor should always be the weakest link of the structure, which out of all the components, is one of the easiest to replace. The failure of the motor also has significantly smaller consequences as the damage is contained to the motor itself whereas if a shaft breaks during use, it can lead to much larger damages to the other components and longer downtimes.

As a design method, the EEUA tends to oversize the shaft considerably unless it is expected that the impeller can in fact jam during use. The method assumes ideal conditions in which only a combination of torsion and bending forces are applied to the shaft, as shown in Figure 11. (Edwards et al. 1997, p. 256.)

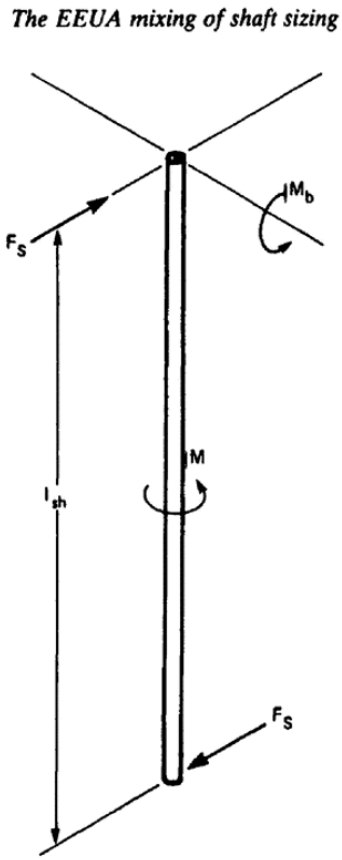

Figure 11. The loading of a shaft according to the EEUA method (Edwards et al. 1997, p. 257).

Although the EEUA method is old and less commonly used, similar principles of sizing shafts in vertical agitators are still regularly used. The general sizing methodology for paddles can rely on a similar ideology, they need to be able to withstand a concentrated load applied at the free end of the paddle until the motor stalls. This leads to the same design approach where the motor is made to be the weakest link, assuming that the shaft's capacity is not exceeded. For a flat plate paddle, one source recommends that the point load on the paddle should be set near the tip of the paddle but at 3/4th of the paddle's length, which is considered to reflect the true loading more effectively (Uhl & Gray 1967, p. 293).

Similar simplistic methods, as described here, will be used in the analysis of the shafts and paddles as results which include significant safety margins are desired and the methods to acquire more realistic results have been deemed to be outside of the scope of this study. For the shafts, the combined loading of torsion and bending, as presented in the EEUA method,

will be used to analyse the effectiveness of the stiffeners where other loads are assumed to be negligible. For the paddles, a point load will be set at the free end of the paddle. Based on the previously conducted CFD analyses, a point load is very simplified representation of the actual loading but should give results on the safer side. Additionally, the hub stiffeners will be examined by themselves where only the bolt preload is applied. These configurations are illustrated and described in more detail in the finite element analysis chapter.

2.2 DFM and welding production

The main goal of DFM is to create high quality products that meet the set requirements while also maintaining low prices and good manufacturability, which leads to more efficient production. (Poli 2001, p. 2.) When considering any design changes to the stiffeners and the structures in general, the effects on the manufacturing must be considered. Changes in the manufacturing can also affect the quality of the production, which needs to be considered, although simpler designs with less manufacturing steps tend to simplify the quality control as well.

In fatigue loaded structures, the presence of notches and stress concentrations have a significant effect on fatigue life. In vertical agitators, all the primary components can experience some degree of variable loading, but shafts are considered the most susceptible to fatigue failure. As with the loading in general, fatigue must be considered based on the specific application and process, but even without specific analysis the fatigue life of the structures can still be improved by mitigating notch effects and other sources of stress concentrations around critical areas. As an example, some improved alternatives for sharp corners on a shaft are presented in Figure 12.

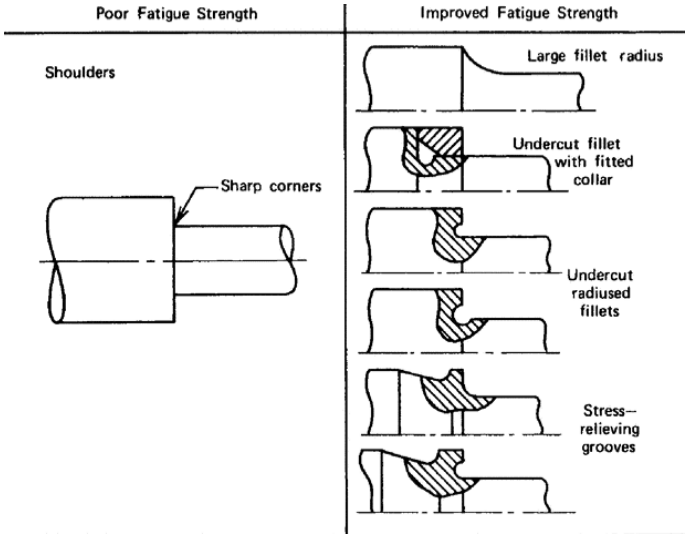

Figure 12. Alternative designs to improve the notch effect (Fatemi, Fuchs, Stephens R.I & Stephens R.R 2001, p. 194).

When considering the fabrication of a welded design, welding positions and the easiness of the welding become factors. Good designs result in easier welding which require less preparation time. The less time that is used fabricating the product while maintaining consistent quality also leads to a more cost-efficient solution. Avoiding welding altogether could often be the best choice but it cannot always be avoided.

The most common welded materials, such as EN 1.4404 and EN 1.4401 stainless steels, have low carbon contents which offers good weldability. With lower thermal conductivity and higher thermal expansion compared to mild carbon steel, less heat is allowed to be conducted away from the weld and the material expands more, which can lead to increased warping and distortions in the austenitic steels. (American Iron and Steel Institute 2020, p. 4-6.)

3 Applying the finite element method

FEM serves as the main method of analysing the structures and evaluating the effectiveness of the stiffeners. Simcenter Femap 2021 software is utilised to conduct the analyses. Quadratic tetrahedral solid elements are used as they can be applied to complex geometries more easily compared to many other element types (Taylor, Zhu & Zienkiewicz 2005, p. 265). As elements, they are more prone to issues due to bad shapes and often require higher density meshes than other element types but with the quantity of models examined, this approach allows a faster workflow, albeit at the expense of longer analysis times.

Both linear and nonlinear analysis types are utilised based on the studied part and loading. Nonlinear analysis will be conducted when it is expected to produce more realistic results such as when determining the buckling capacity of the paddles. Otherwise, linear analyses will be used to expedite the analysis times. Alongside geometric nonlinearity, material nonlinearity will be utilised if it is expected that the materials yield stress is exceeded. All the stresses will be examined as von Mises from the analyses.

For the analysis of the paddles, an element size of ten millimetres is used. As for the shafts, element sizes ranging from 5 to 20 millimetres are used, with the highest mesh density located around the stiffeners. Respectively, element size of five millimetres was used for the hubs. The same element sizes are used for each structure to maintain comparability between the results. These mesh sizes proved to be suitable for the analysis with static loads, as no significant differences in the results were noticed with higher mesh densities. Reasonable analysis times were also achieved.

3.1 Material model and geometry

The used material in the analysis is austenitic stainless steel EN 1.4404. The necessary mechanical properties for the finite element analysis (FEA) are displayed in Table 2. For ultimate tensile strength, the smallest provided value was used and Poisson’s ratio of 0,3 was chosen.

Table 2. Material properties for cold-formed EN 1.4404 steel (SFS-EN 10088-2 2014, p. 43).

Yield strength (MPa)	Tensile strength (MPa)	Elongation, A80 (%)	Modulus of elasticity (GPa)	Poisson's ratio
240	530-680	40	200	0.3

A nonlinear elastic material model is employed by creating a bilinear stress-strain curve, which is shown in Figure *13* alongside a typical engineering stress-strain curve of a stainless-steel alloy. A bi-linear material model does not fully depict the material's true stress-strain behaviour, but it allows plastic behaviour to be considered by defining two linear lines to represent the material's behaviour before and after yield stress is reached. The connecting point between the lines is defined at the yield stress of the material where the strain is calculated by dividing the yield stress with the material's modulus of elasticity. The last point is defined at the ultimate tensile strength of the material where the strain is determined by ultimate elongation.

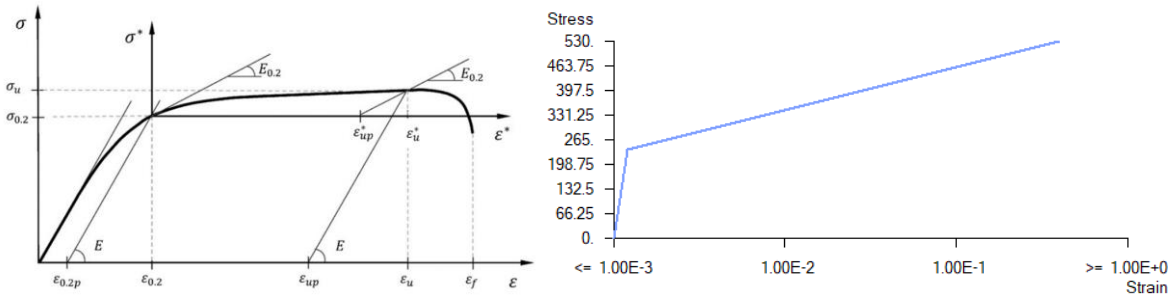

Figure 13. A typical engineering stress-strain curve is depicted on the left, next to the used bilinear material model function (Arrayago, Gardner & Real 2015, p. 541).

All the geometries of the studied parts are modelled fully due to asymmetrical geometries present, and no initial imperfections are included. Three types of models are created which are presented in Figure *14*.

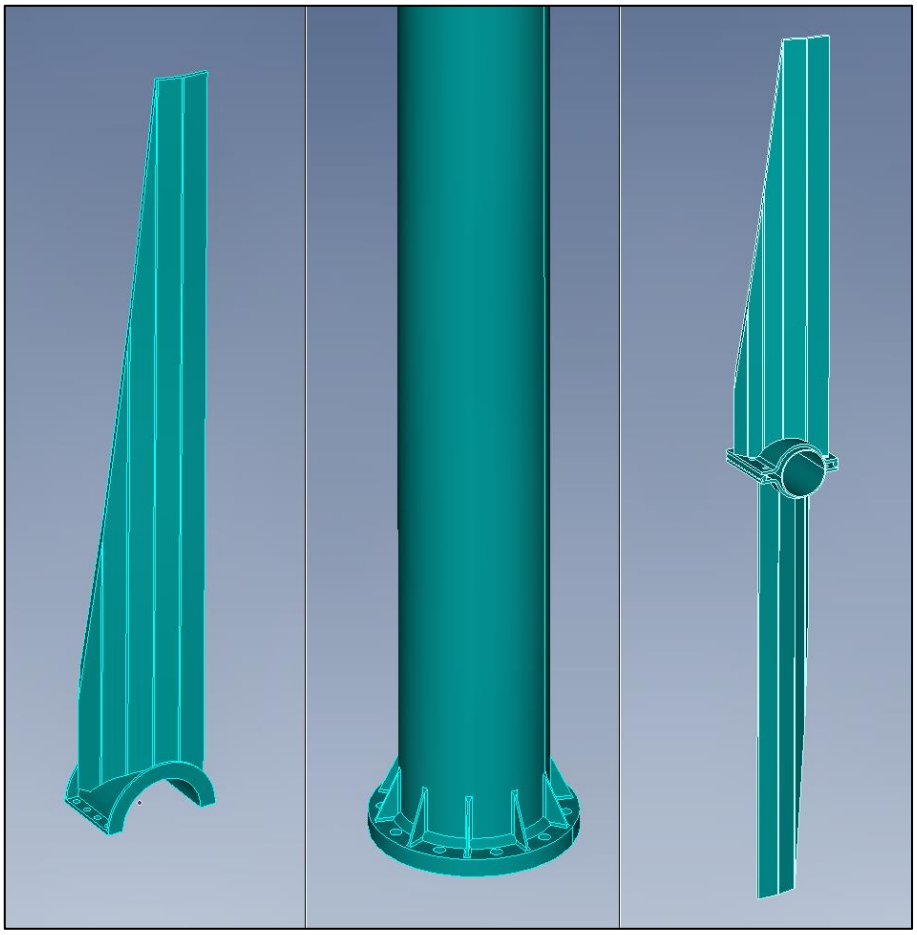

Figure 14. A model of a paddle is depicted on the left, a shaft in the middle and a hub connection on a shaft on the right.

The paddle's critical failure mode is assumed to be buckling, and therefore nonlinear buckling analysis is conducted. For the shafts, only the local effects of the stiffeners are examined, and similar results were achieved with nonlinear analysis, so linear analysis is conducted. For the hub connection, the effects of the preloaded bolt connection are examined with a nonlinear contact analysis.

3.2 Constraints and loading

Shafts are fixed from the end which attaches to the coupling. Rigid body element 2's (RBE2) are added to the free end, which are utilised to apply and distribute the combined loading of

torsion and bending moment to the shaft. The RBE2 elements are presented in Figure *15* which depict a load applied at the end of a paddle.

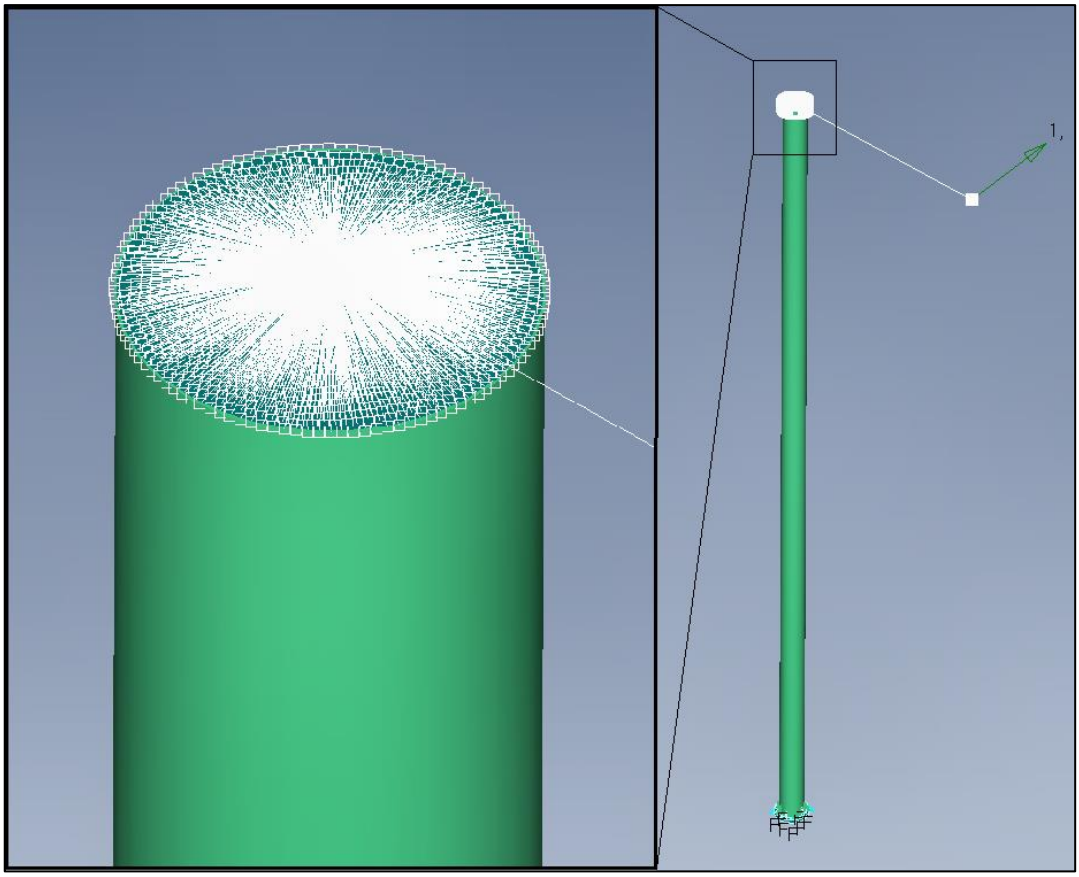

Figure 15. The rigid body elements of a shaft which are colored in white.

The dependent nodes of the rigid element are set to the end surface of the shaft and translations are restricted in all three directions which makes the rigid body element infinitely stiff. The independent node is connected to another RBE2 element which is used to illustrate a paddle as the moment arm. A singular nodal force is then applied to the end which simulates a point load that is applied at the free end of the paddle. The length of the moment arm is defined by the maximum length of the paddles for the respective shaft.

By applying the RBE2 to the end of the shaft, some unnecessary stiffness is added but it does not have a significant effect on the result as the main deformation occurs along the shaft and not where the RBE2 is applied. The rigid element allows torsion and bending force to be applied to the end of the shaft with the single nodal force. This does not inherently represent

the torque loading applied by the motor, but when analysing the stiffeners, they barely affect the torque capacity of the shaft and mainly take on the load caused by the bending moment.

The paddles are fixed from the bottom surfaces of the hub and the load is applied at the end of the paddle on the leading edge as a point load. As mentioned, the loading is not representative of a realistic loading, but it is used due to the limitations of the study. A paddle model with the applied constraint and load is shown in Figure *16*.

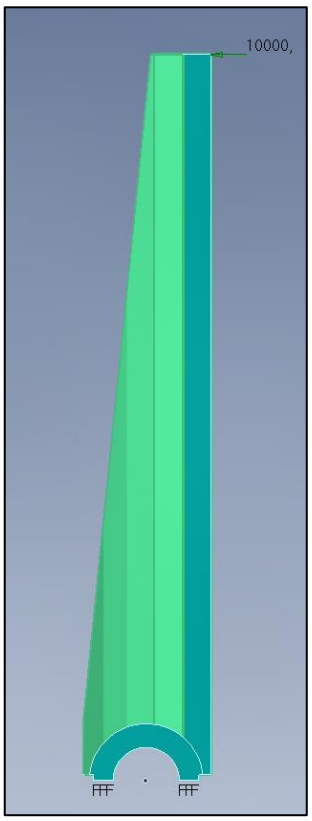

Figure 16. Loading and constraints of a paddle.

To analyse the bolted connection, a model with two hubs with paddles and a piece of a shaft was created with solid elements. Additionally, the bolts are modelled as beam elements and attached to the hubs with rigid body elements. Contact surfaces are created between the necessary surfaces and the connectors are created. Bolt preloads based on the specific tightening torques are applied to all the bolts. The model is displayed in Figure 17, where solid elements are coloured in grey, rigid body elements in red and beam elements in green.

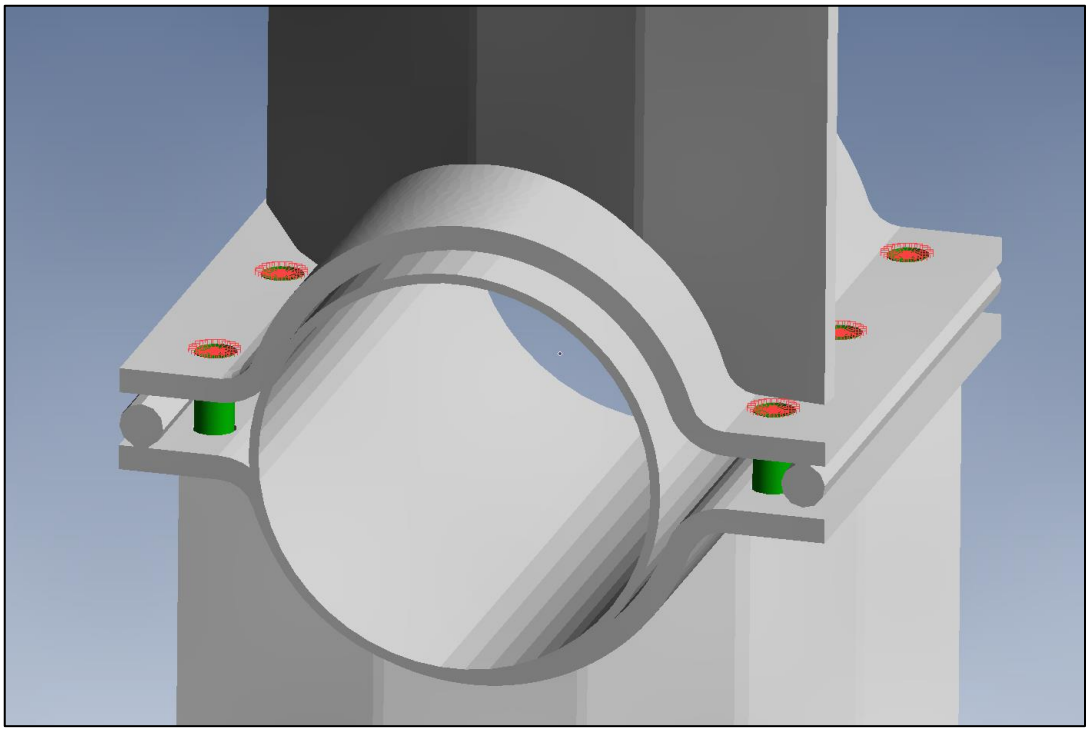

Figure 17. The bolted hub connection to the shaft.

Due to the manufacturing tolerances, the shafts and hubs tend to have slight variances in their respective dimensions, which can have effects on the quality of the connection. These possible imperfections are not considered in the model and a good fit is assumed in the ideal model.

3.3 Computational fluid dynamics

CFD is a numerical approach for studying fluid flows, heat flows, mass transfer problems and chemical reactions (Sharma 2021, p. 3). In this thesis, CFD is utilized in small amounts to analyze the flow patterns of the alternative solutions of the paddle stiffeners. The stiffeners contribute to the formation of vortices and add turbulence, each of which is to be avoided as much as possible. The conducted CFD analysis is simple, and the results will be treated as a supportive element in the inspection of the paddles. Solidworks fluid simulation is used by generating a small cubic chamber around the paddle as is shown in Figure 18. An inlet velocity of two meters per second is set on to the side in front of the paddle and a static pressure outlet is set on the opposite side.

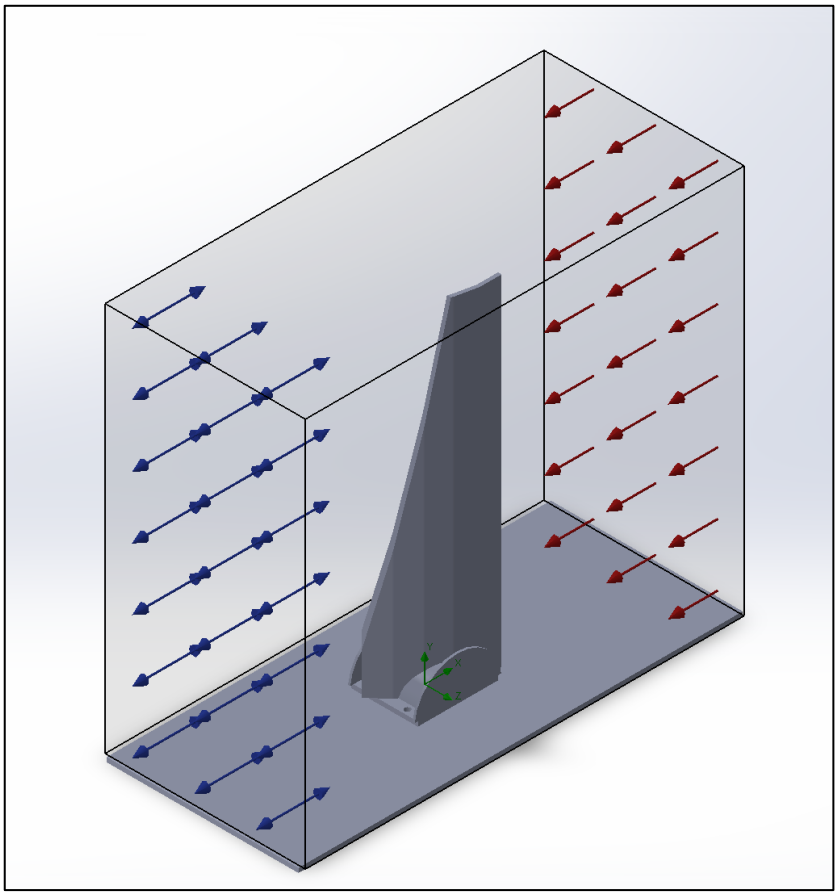

Figure 18. A simple CFD model for a paddle without any stiffeners.

All the analyses are done with water as the fluid while assuming turbulent flow type. As this analysis does not rotate the paddle around a shaft, it does not account for the differences in flow velocity that the paddle experiences along its length, where the fastest speeds occur near the tip and gradually become slower near the shaft. Only the local effects of the stiffeners are desired, which should be able to be achieved even with the simplified approach.

4 Experimental test

Ideally experimental tests would be conducted alongside all finite element analyses to ensure their reliability and accuracy but in this case, only one type of experiment is performed to examine one of the alternative hub solutions. A simple test is conducted where a hub with the alternate design is manufactured and attached to a shaft where the only applied load is the tightening torque of the bolted connection, similarly as in the calculations. To ensure that a suitable connection is achieved, the specified tightening torque will be applied, and the structure will be monitored to see if excessive deformation occurs, or if the hubs do not compress on to the shaft well enough. An inadequate connection would allow the hub to slip on the shaft which must be avoided.

The tested structure is manufactured in a similar way as the actual structure would be manufactured and the connection is tested on a cut section of a shaft, where the ends are welded shut. The paddle length is limited to 300 mm which is assumed to represent the added rigidity that a full-sized paddle would provide. The tested structure is displayed in Figure *19* alongside a sketch of a similar design.

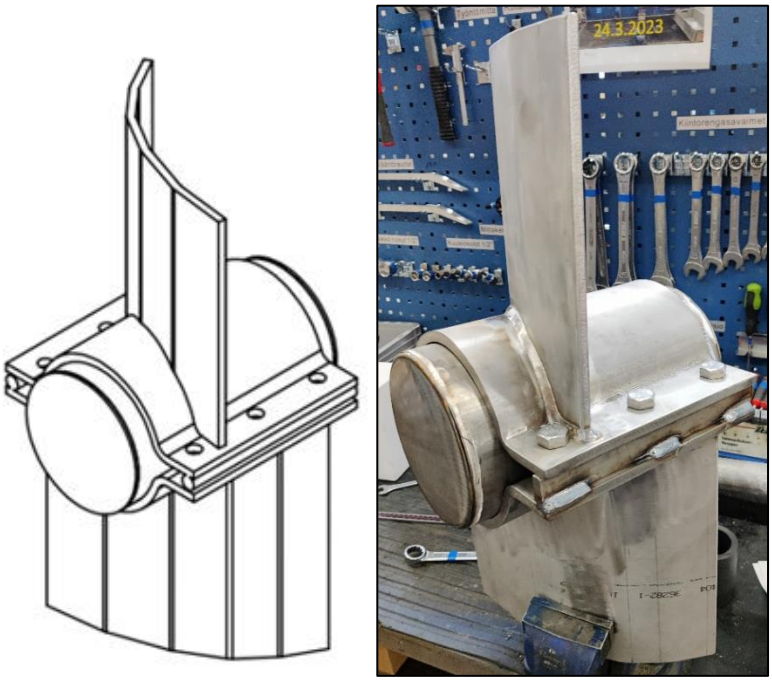

Figure 19. The experimental structure as a sketch and as a tested structure.

To test the friction connection between the hubs and the shaft, a press will be used to determine how large a load can be applied to the shaft before it starts to slip from between the hubs. This is depicted in Figure *20*.

Figure 20. The press is used to push the shaft through the connection.

The pressing is not representative of the slipping where the hub rotates on the shaft, but it allows the normal force of the friction connection to be calculated. If the friction coefficient is assumed to be the same in all directions, this allows the defining of torque capacity for the connection. It is assumed that if the connection can withstand a pressing load of 5 kN, it is deemed sufficient.

5 Results

In this chapter, the results of the finite elements analyses of the shafts and paddles are presented alongside the observations from the experimental tests. The proposed alternate designs for stiffeners are also presented, evaluated, and compared based on their effectiveness and cost efficiency.

5.1 FEA results of the current solutions

In this subsection, the main results from the finite elements analyses are presented where the focus is on the basic structures without any stiffening components and the currently used solutions. The most critical areas of the structures and the effects of the current stiffeners are presented.

5.1.1 Shafts

These analyses are conducted with relatively long shaft lengths where the results are not representative of all possible shaft lengths. The length of the shaft does not have an effect on the shearing forces generated by the torque but it impacts the forces generated by the bending moment substantially. The used stiffeners are always the same size no matter how long the shaft is, and therefore the effects are always disproportional to the shaft length. Due to the length of the shaft, the bending moment created by the load has a significantly larger effect than the torsion and therefore the most critical area is found near the weld toe on the shaft. Due to the ideal shape of the weld, there are no clear nucleation points for the stresses to concentrate on. Figure *21* displays the stress distributions that form near the fixed end of the shafts when no stiffeners are used.

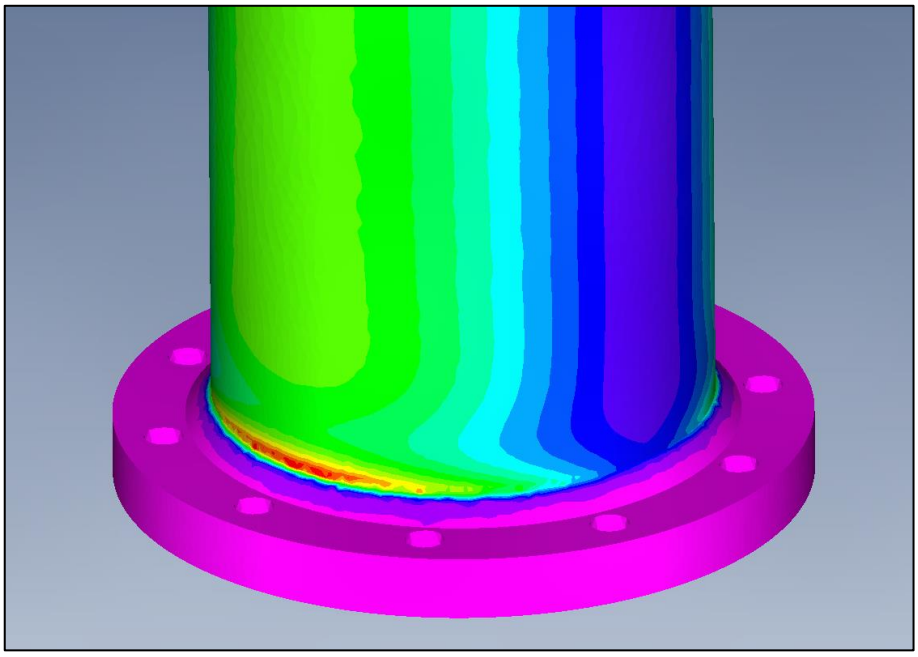

Figure 21. Stress distributions on an unstiffened shaft.

Cross sections of both unstiffened shafts are included in (Appendix 1) and (Appendix 2), which display the stresses throughout the welds. If fatigue was to be considered, by IIW recommendations, the root would be critical for fatigue failures for these types of connections (Hobbacher 2008, p. 72). The weld roots do not experience as high stresses as the weld toes, especially for the smaller shaft where the inserted plate acts as a backing plate. The welds do require good penetration, as is assumed in these analyses, or otherwise the roots will experience much higher stresses.

In Figure *22*, both shaft sizes are displayed with the currently used stiffeners. The highest stress concentrations are moved from the flange weld to the tip of the stiffeners. In both stiffened solutions, the stresses formed near the flange weld are lesser than in the unstiffened shafts, but overall higher peak stresses are formed at the tips of the stiffeners.

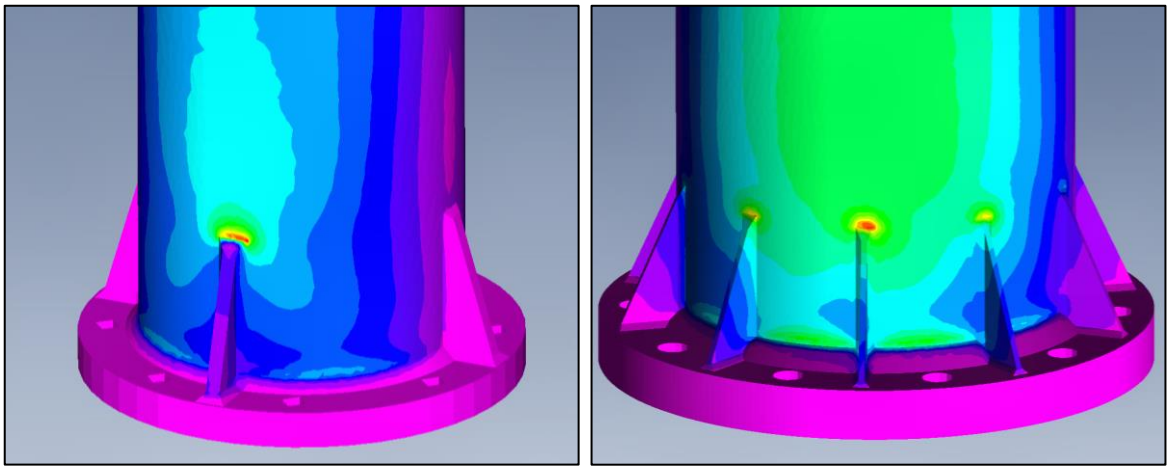

Figure 22. Stress distributions of the currently used solutions for shaft stiffeners.

In static loading, the higher stress peaks are not inherently problematic as the stress concentrations are local and the surrounding structure can take on the excess load. In fatigue loading, it can be problematic, although so far with these solutions, fatigue failures have not occurred, which proposes that the current structures do not experience significant fatigue loads, or the loading is not significant enough to cause failure. One benefit that the current solutions offer is that as the critical area is moved from the flange weld to the stiffener welds, where a lower quality can be accepted for the more laborious flange weld. Without the stiffeners, a sufficient weld quality must be achieved for the entire flange weld, if no possibility of premature failures is desired.

Compared to the unstiffened shafts, when four stiffeners were added, a 108% higher stress peak was formed, and respectively 16% higher when ten stiffeners were added. Based on these results, the currently used solutions, and especially when four stiffeners are used, do not benefit the structures loading capacity in a significant manner. If the flange weld quality is good enough and the throat thickness is sized accordingly, the stiffeners should not be needed from a structural standpoint.

The stiffeners could also serve to prevent deformations in the flanges, where a good connection to the coupling needs to be achieved, but in these cases the flanges are at least 30 millimeters thick and should not deform to the point where the stiffeners would be needed.

5.1.2 Paddles

When nonlinear buckling analysis of the paddles is examined, the paddle starts to lose its stability near the middle of the paddle. As the load carrying capacity is lost, nearby sections start to take on the load until the cross section of the paddle loses its stability and the paddle finally buckles. Five timesteps from a single paddle are shown in Figure *23* which depicts the changes in stress distributions as more load is applied until failure. The paddle buckles near the middle but slightly closer to the hub and a large area of the paddle seems to be utilised.

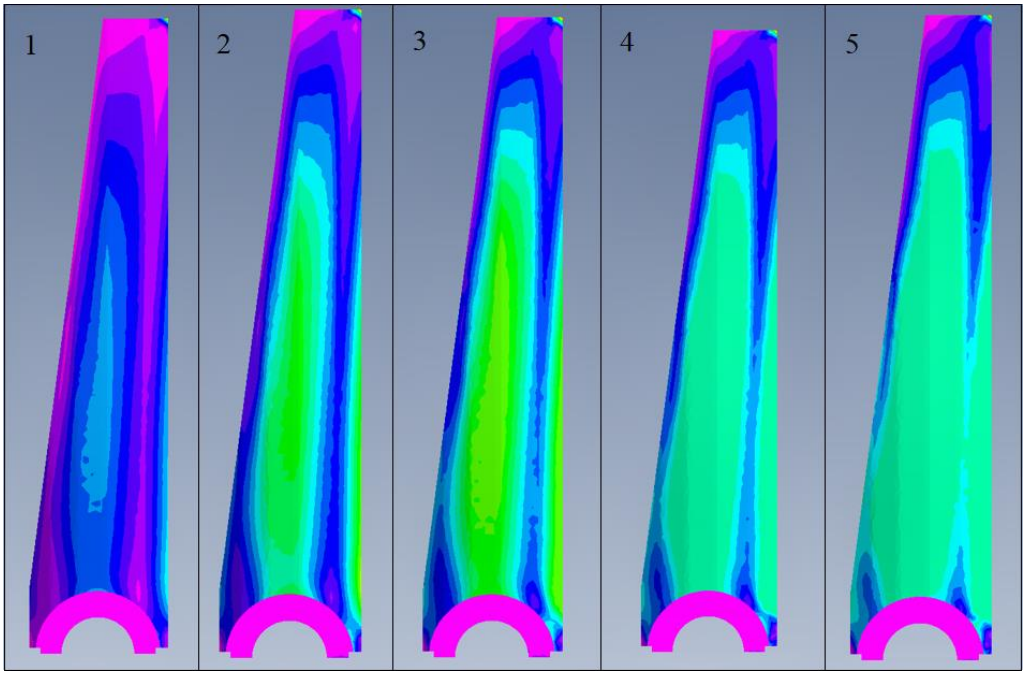

Figure 23. Five timesteps in order as more load is applied at the tip of the paddle.

The currently used box stiffener increases the paddle's buckling capacity by roughly six percent but the paddle still buckles in a similar mode. The stiffener adds rigidity especially to the base of the paddle and transfers the critical buckling area further away from the hub of the paddle. The final timestep of a paddle with the box stiffener is shown in Figure *24* which displays notably different stress distributions than when no stiffener is used.

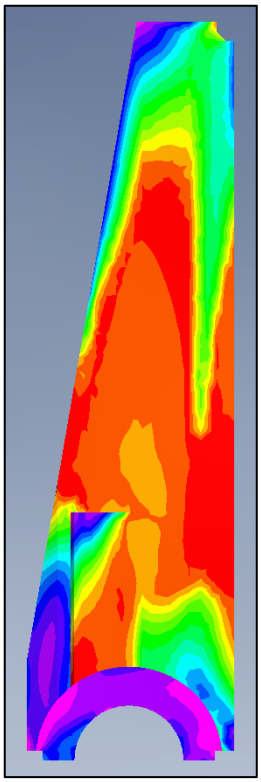

Figure 24. Final timestep of a stiffened paddle.

Better effect could be achieved by increasing the stiffener size, but as will be discussed in the later sections, the possible geometry of the stiffener is limited. The current solution does still prevent buckling and is relatively simple in design which is easy to fabricate.

5.1.3 Hubs

The friction connection is harder to estimate compared to stresses and deformations, but from previous knowledge at the company, it is known that excess flange deformation often leads to insufficient connection. This is used as the main criteria to evaluate the results, but experimental tests are conducted to ensure the viability of the chosen designs. A deformed hub without stiffeners is displayed in Figure 25.

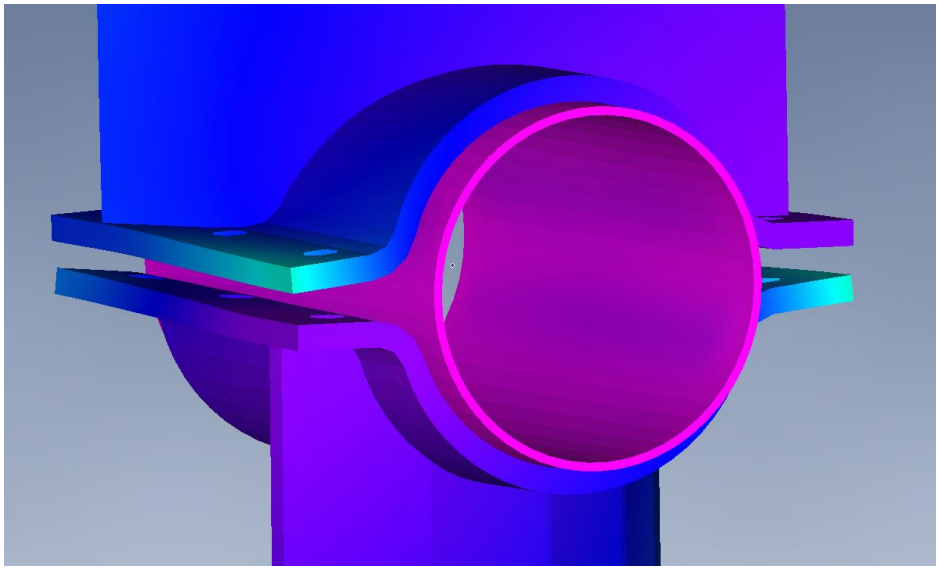

Figure 25. A deformed hub when a bolt preload has been applied.

This previous knowledge was backed up by the FEA results as when no stiffeners were used with the current plate thickness, the hub flanges visibly deformed beyond an acceptable amount. When the currently used stiffeners were added, the hubs barely deformed which reflect their known real behaviour.

5.2 Alternate designs and cost-efficiency comparisons

In this subsection all the considered alternate designs are presented. At the end of each subchapter, a comparison table is presented which compares the different solutions based on structural and cost related aspects. When comparing the stresses and deformations, the basic structure is set as the baseline which every other solution is compared to, and the difference is given as a percentage. Rheological aspects are also considered with paddles through simple CFD analysis.

The costs of each design are considered by determining baseline costs for the material, cutting, and bending plates and lastly the cost of labour all of which are presented in Table *3*. The cut and bent plates are outsourced by the company which is why they are combined and presented as price per kilogram. Many other sources for costs exist but are difficult to predict, such as any downtime during fabrication and preparation times, which is why only directly numerable types of costs are included.

Table 3. Baseline costs.

	Price
Material	10 €/kg
Precut and bent plates	12 €/kg
Labour	47.5 €/h

Due to similar limitations in determining the other factors, only welding is considered as direct labour costs. An average welding speed of 40 cm/min is assumed, which is used to calculate the time it takes to weld each individual design to determine the costs.

5.2.1 Shafts

As the two different shaft sizes are similar in design, the alternate designs are considered to have similar effects on both shafts and therefore same stiffener designs are not analysed on both shafts. The designs one and two are slight variations of the original stiffeners, which were considered initially, with slight variations. They focused on alleviating the stress concentrations formed at the tip of the stiffeners by adding curvature to the stiffener and rounding the weld at the tip of the stiffeners. In the third design, stiffeners that were adjacent to the surface of the pipe rather than tangentially like in the first two designs, were considered. The first three designs are presented in Figure 26.

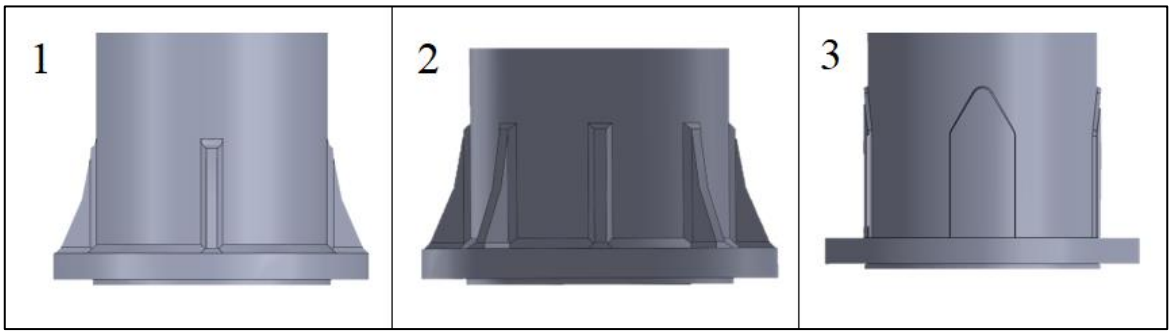

Figure 26. The first three alternate designs for the shafts.

As the first two designs stay loyal to the original design, there are minimal changes to the costs and fabrication methods. To achieve the rounded weld, post weld treatment such as

grinding would be needed. Although an additional cost, it could increase the quality by forcing the most critical spot to be inspected and treated every single time, compared to leaving it as welded, where mistakes would be less likely to be found. These two solutions are more fatigue friendly than the original designs and could be easily employed if fatigue becomes a concern. They do not offer significant benefits in static loading. The third design could prove to have a more difficult fabrication process as the plates need to be cut and then bent to the radius of the respective shaft but with a press and the correct tooling, a viable workflow could be achieved. Alternatively the plates could also be cut from a bent plate or from a cylinder.

Other three designs where much different compared to the previous designs, where designs four and five increased the thickness of the shaft at various lengths and the sixth design added considerably longer stiffeners of the original design. In the fourth design the thickness of the pipe is doubled at one percent of the length of the shaft and five percent for the fifth design respectively. The last design was considered as to whether there would be considerable benefits to increasing the stiffener sizes overall. These three designs are displayed in Figure 27.

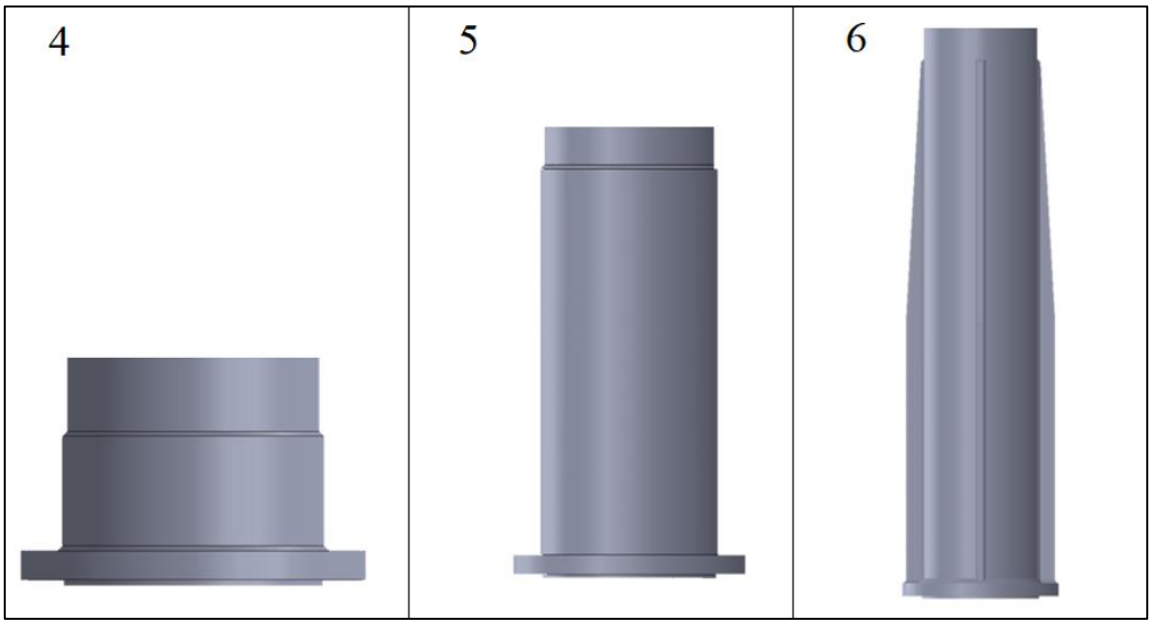

Figure 27. The last three shaft stiffener designs.

These three designs use more material and require more welding than the other proposed solutions, but it is also predicted that they will perform better. The two designs which simply add thickness are the only ones which remove the geometric sources of nonlinear peak stresses. Alongside the various stiffeners, the shaft was examined when one millimetre was added to the thickness.

For both shafts alternate flange designs were also considered. The flanges are highlighted in red and the connecting welds as green. For the first flange design, presented in Figure *28*, a bevel edge was added to move the connecting weld between the flange and the shaft further from the root.

Figure 28. The alternate flange design for the smaller shaft.

For the second design, a similar effect was attempted to be achieved by moving the weld further away from the most critical spot. From a fatigue standpoint, this design offers the better solution, but it needs to be ensured that sufficient penetration for the weld is achieved. The second flange design is presented in Figure *29*.

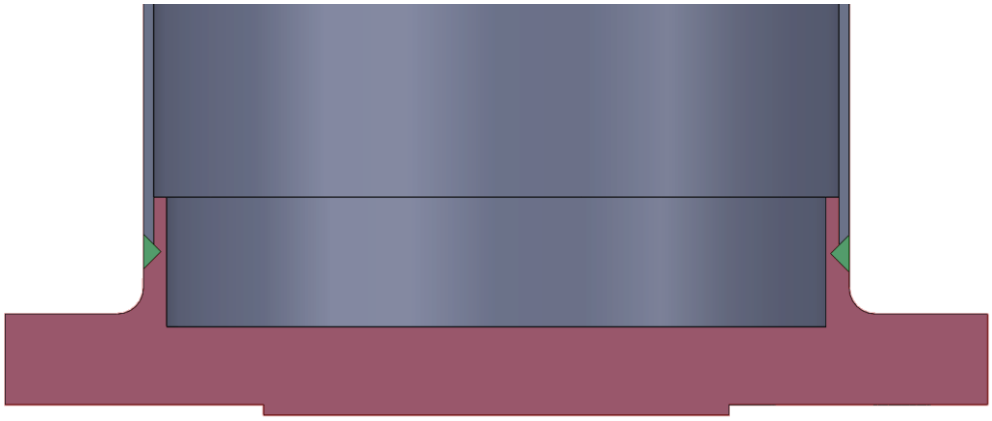

Figure 29. The alternate flange design for the larger shaft.

The flanges are currently machined from large blocks of steel and this design would almost double the required block size where majority of the material goes to waste. With this design, an alternate fabrication method could prove to be more cost efficient. Casting would not incur as large material losses but would require a mould to be made. Some of the surfaces would also require machining to achieve good connections to other parts but overall casting could offer a realistic solution. Table *4*. displays results of the various solutions where the highest stress peaks, deformations are presented alongside the added costs of each design. The stress peaks were always located near the weld toe of the most critical weld except for the flange design depicted in Figure *29* where the stress peaks were located further away from the weld where the flange stops supporting the pipe.

Table 4. All the alternate solutions compared on a ten-meter-long shaft.

			Costs (€)			Total added costs (€)
	Peak Von Mises (%)	Max deformation (%)	Material	Plates	Labour	
Without stiffeners	0	0	0	0	0	0
219 mm diameter shaft with original stiffeners	+108	−1	10	12	2	24
273 mm diameter shaft with original stiffeners	+16	0	15	18	6	39

Table 4 continues. All the alternate solutions compared on a ten-meter-long shaft.

			Costs (€)			Total added costs (€)
	Peak Von Mises (%)	Max deformation (%)	Material	Plates	Labour	
Design 1	+63	−2	10	12	2	24
Design 2	+49	−3	20	24	4	48
Design 3	+61	−2	20	24	6	50
Design 4	+8	-2	50	0	1	51
Design 5	0	−8	200	0	1	201
Design 6	+58	−10	110	132	16	258
+1 mm to thickness	−15	−20	530	0	0	530
Alternate flange design 1	+8	-1	12	0	0	120
Alternate flange design 2	-6	0	34	0	0	340

From the results it can be seen that every design which implements notable sources for notches, such as ribs, leads to higher stress peaks than when no such stiffeners are added, although all of the designs lessen the peak compared to the original stiffener design.

When the shaft length varies, the added costs of almost every solution remain unchanged except for when the overall thickness of the shaft is increased. In this case the costs correlate with the length of the shaft and as the shafts tend to be shorter than ten meters, the table does not represent the costs of the solution realistically. The results should not be extrapolated to different shaft sizes, but for many solutions they are comparable.

When evaluating the six stiffener designs, designs four and five demonstrate the lowest stress peaks as due to the lack of notable stress concentrations. Deformations are not considered critical for the structures, but they decreased the most with fifth and sixth designs which is mostly caused due to the designs being large and impacting a larger area of the shafts. These two designs are also more costly to implement than other solutions.

If the lowest possible stress peaks and deformations are desired, the best results were achieved when the shaft thickness was increased by one millimeter. This resulted in the costliest implementation which alludes to the reason why increasing the shaft thickness is not always the most optimal solution. With a shorter shaft more optimal results can be achieved. The length of the shaft could be divided by the cost which would give a price of 53 €/m that is more indicative of the costs for any shaft length.

The first alternate flange design did not decrease the stress peaks compared to the current flange design, but the second design decreased the stress peaks. This design does not only decrease the possible fatigue stresses, but it also moves the weld away from the base of the shaft, decreasing the possible problems caused by poor weld quality and the heat input from the welding.

If fatigue is a concern, the second flange design could significantly alleviate the possible fatigue failure but implementing the design would create costs that are not considered in the table. If the current fabrication method is kept, a larger metal block is needed to machine the part fully which, although possible, would lead to increased waste material. An alternative form of fabrication could be casting, which would involve initial costs to create the molds. The company does not have the capability to cast the flanges themselves, so a subcontractor is needed. Relying on subcontractors adds a layer of uncertainty to the manufacturing process, but implementing the fabrication process at the company would prove to be too costly. If cast, larger quantities would be produced periodically and stored, deviating from the current production that is based more on demand. Additionally, some of the surfaces of the cast flanges would need to be machined to meet specified tolerances which would incur additional costs.

5.2.2 Paddles

As the paddles significantly affect the mixing process, any changes to the paddle geometries can easily impact the effectiveness of the paddle in the mixing process. One proposed solution was to weld multiple small stiffeners longitudinally on the paddle as is shown in Figure *30*.

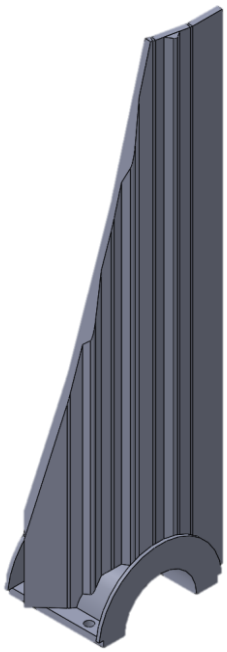

Figure 30. A stiffened paddle when no geometric limitations were set.

This resulted in significantly better buckling capacity and less deformations, but the results of a simple CFD analysis showed that the solution might not be ideal. The flow patterns of the paddle with and without the stiffeners are displayed in Figure 31, where turbulence and vortices that are formed on the suction side are stronger when the stiffeners are added.

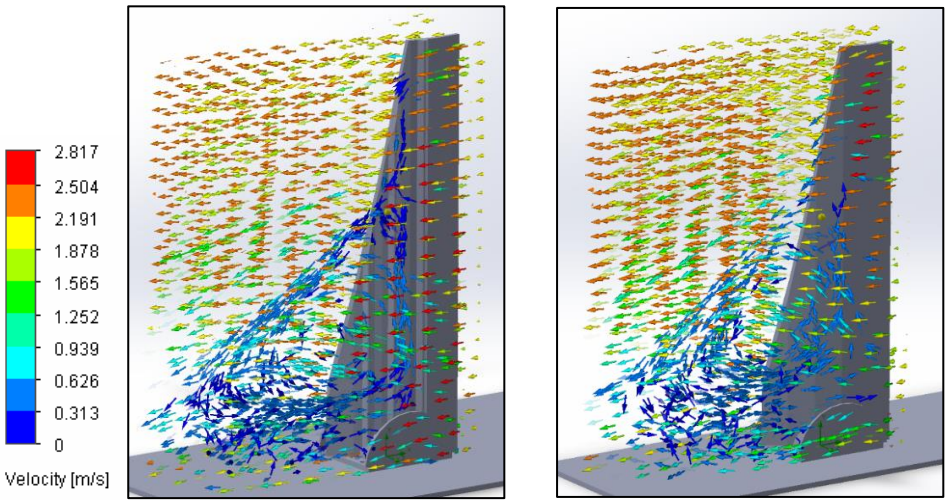

Figure 31. CFD analyses for a paddle with no stiffeners on the right and the same paddle with three longitudinal stiffeners on the left. Arrows display the fluid velocity.

This behaviour should be strictly avoided anywhere near the tip of the blade which makes this specific solution not suitable. Based on this study and guidance from the company, other stiffening solutions were limited to 30% of the paddle's length as any geometrical changes to the rest of the paddle could severely affect their performance. Many solutions, which are not presented here, were tested which proved to be unideal due to either not increasing the buckling capacity or by affecting the flow too significantly. Solutions which were considered are shown in Figure 32.

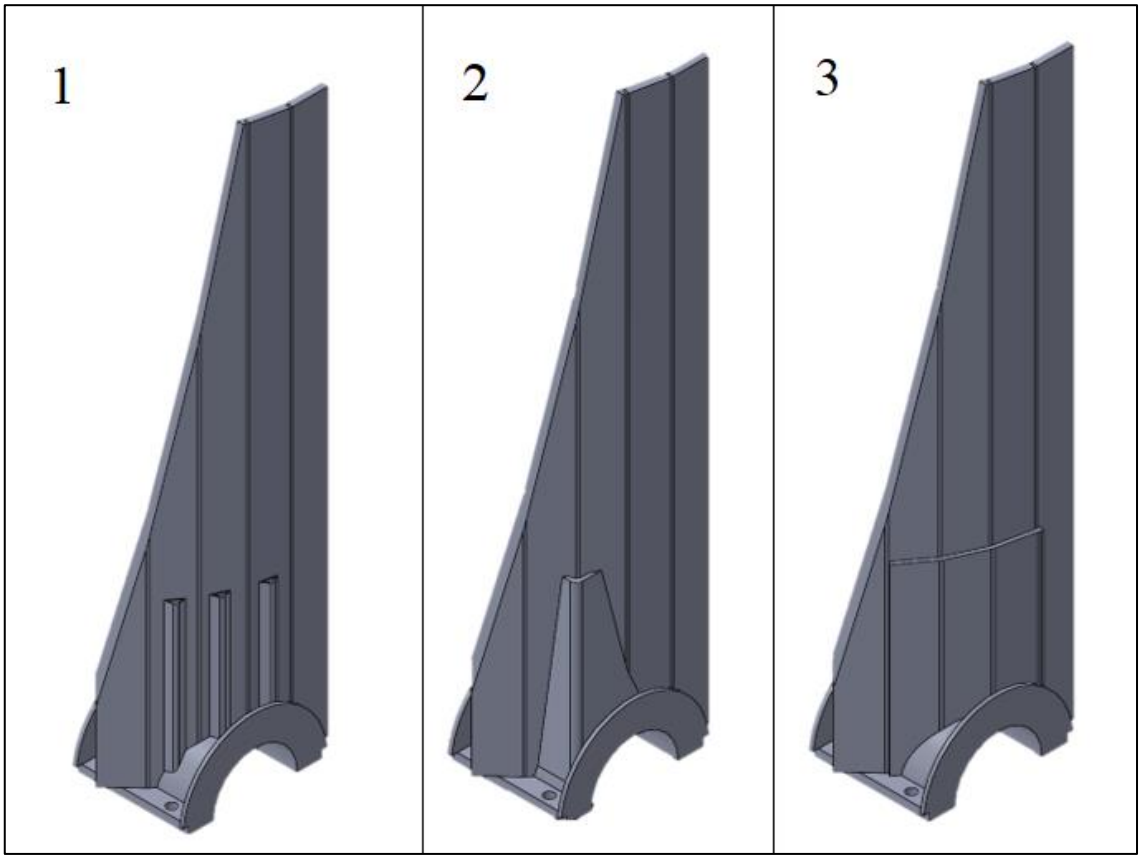

Figure 32. The three paddle stiffener designs.

The first design is based on the initially proposed design but adjusted to be only 30% of the paddle's length. The three stiffeners are halves of square hollow section profile that could simply be sawed from the full profile or simply be bent from a plate. The second design is like the currently used box stiffener, but with the sides narrowing towards the end. The third

design simply increases the thickness of the plate near the hub. All the results of the analysed structures are shown in Table 5.

Table 5. Comparison of the alternate paddle solutions.

		Costs (€)			Total added costs (€)
	Buckling capacity (%)	Material	Plates	Labour	
Without stiffeners	0	0	0	0	0
With original stiffeners	+6	10	12	2	24
Design 1	+4	3	3	2	8
Design 2	+4	15	18	1	34
Design 3	+10	16	19	2	37
+2 mm to paddle thickness	+32	20	0	0	20

The stiffeners are determined to be 30% of the length of the paddle itself so the results should be comparable between different paddle sizes as they share same design. When increasing the paddle thickness entirely, the results will differ more between different paddle sizes and the costs are different as the larger paddles can be significantly longer. These are presented in Table *6*, where the paddles are set at their maximum length and compared between the basic structure and when two millimetres are added to the thickness. As the only changing parameter is the thickness, only material costs are considered. As previously, the results of the basic structures are set at 0% to which the other results are compared to.

Table 6. Comparisons between the different paddles and the relative material costs. The longest paddle lengths per hub are examined.

Paddle	Buckling capacity (%)	Added material cost (€)
Hub size 114	0	-
+2 mm to thickness	+32	20
Hub size 168	0	-
+2 mm to thickness	+32	60
Hub size 219	0	-
+2 mm to thickness	+29	90
Hub size 273	0	-
+2 mm to thickness	+27	110

Between the paddles from the smallest hub and the largest hub, roughly five percentage point difference is found between the buckling capacity, when 2 millimetres is added to the paddle thickness, but the costs become more significant for the larger hub paddles.

5.2.3 Hubs

Proposing alternate designs to the hubs was more difficult as the quality of the friction connection cannot be easily estimated from FEA. Due to this, the study was limited to designs that could be manufactured and physically tested with relative ease. The proposed designs are based on removing the current flanges and adding rigidity to the parts that deform the most which, if successful, results in significantly reduced manufacturing steps and time. The two hub designs are depicted in Figure 33, where the only difference is found in the added element. One design utilises a round bar whereas the other uses a flat bar.

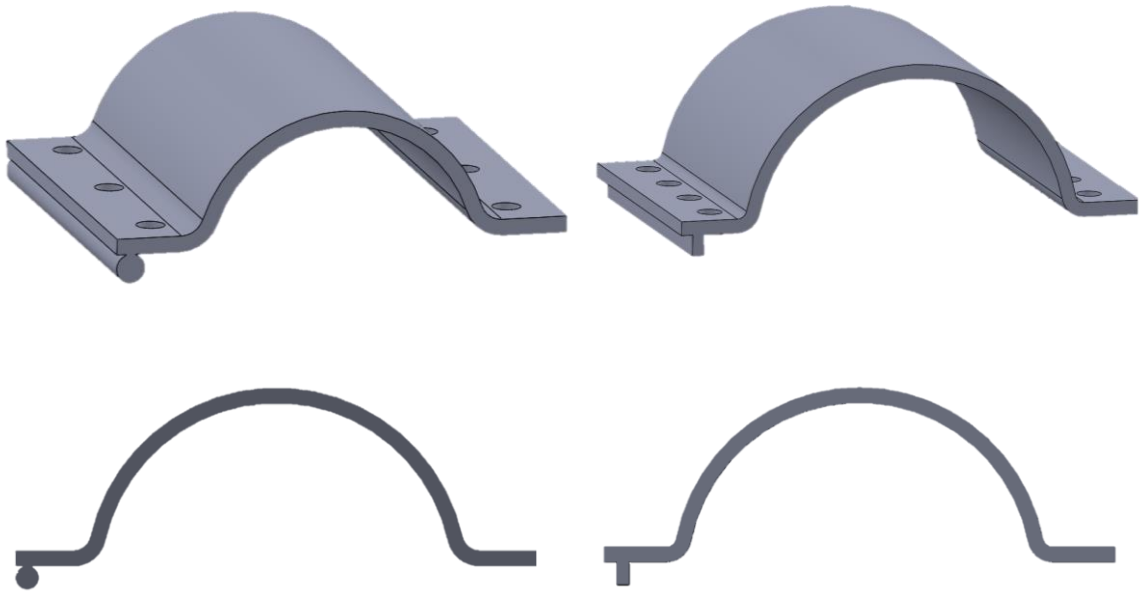

Figure 33. Alternate hub designs displaying the round and flat bar variations.

The round bar design allows for simpler fabrication process as it only needs to be cut to correct length, and less space is required for the weld beads. The flat bar allows the gap to be adjusted more easily between the hubs in the fabrication process, but without bevelled edges, it will take more space on the flange when welded, which requires the hub flanges to be wider. Two different sizes of flat bars were inspected, with thicknesses of five and ten millimetres. The friction connection of the results is graded on a scale where a bad connection is very likely to fail, neutral connection is on the verge and the quality of the connection is hard to estimate and a good connection is likely to hold. The results are displayed in Table *7*.

Table 7. Comparison of the alternate hub solutions.

		Costs (€)			Total added costs (€)
	Friction connection	Material	Plates	Labour	
No stiffeners	BAD	0	0	0	0
Original stiffeners	GOOD	60	72	2	134
Alternate design, round bar	GOOD	5	0	1	6
Alternate design, 10 mm flat bar	GOOD	5	6	1	12
Alternate design, 5 mm flat bar	GOOD	3	3	1	7
+2mm to plate thickness	NEUTRAL	10	0	0	10

From the results, any alternate design compared to the current design would incur significantly less costs. Between all the alternate designs, the costs are comparable. Good friction connection was achieved with every solution except when two millimeters were added to the flat bar thickness.

5.3 Experimental tests

Three different variations of the hub structure were tested featuring different sizes of gaps between the hubs. These variations included a large gap, a small gap, and no gap, all of which are displayed in Figure 34.

Figure 34. Three tested gaps from left to right: large gap, small gap and no gap.

An adequate connection was achieved with the large gap variation but in this case no contact was achieved between the hubs, which lead to the added flat bars simply contributing to the rigidity of the hub. The added rigidity is still beneficial as the edge flanges of the hubs are the most susceptible to deformation. When the gap is not closed after the bolt preload is applied, the hubs are able to deform more easily.

To achieve no gap, three small tabs were welded to the end of the flat bar. With the bolt preload applied, this resulted in an insufficient friction connection. The bolt connection is located close to the added flange which presumably causes the compressive force from the bolt preload to be exerted to the connection between the hub flanges rather than compressing the hubs to the shaft, ultimately causing the hubs to slip on the shaft.

Lastly, the added tabs were machined to achieve a middle ground between the initial gap and no gap at all. This resulted in a good connection, where a compressive connection was created between the hubs but also a similarly good friction connection was achieved between the hubs and the shaft. It was noted that the hubs started to connect to each other when roughly 2/3rd of the tightening torque was applied. This proposes a good solution but requires strict tolerances for the dimensions of the individual parts. A larger gap would not be problematic as seen from the previous results, but an even smaller gap could prove to be problematic, and an insufficient friction connection could be the result. Choosing a solution which has an inherent risk to create a faulty product could lead to significant quality issues if not addressed.

Although the finite element analyses provided promising results, the initial results from the experimental tests did not correspond with the expected results. The heat input from the welding of the paddle significantly warped the hub plate which is something that did not occur with the original design as the large end plates added a lot of rigidity to the hub. The distortions are not by themselves problematic and can be dealt with but if the distortions are allowed it hurts the consistency and therefore the quality of the production.

All the variations suffered from similar issues with visible gaps between the surface of the shaft and the hubs, as shown in Figure 35, which does not inherently limit the friction connection but creates higher peak stresses at the connected points.

Figure 35. The visible gap between the hub and the shaft after the preload has been applied.

This resulted in a crimp like connection, where the hubs only compress to the shaft near the bends of the hubs. The contact points between the hubs and the shaft can be determined visually but also by the scratch marks left when the shaft is pressed through the hub. The marks are shown in Figure 36 where visible scratches can be seen on the shaft around the location of the bend of the hub which gives a clearer understanding of the contact area. The marks appeared roughly in the same location for every test where a good friction connection was achieved.

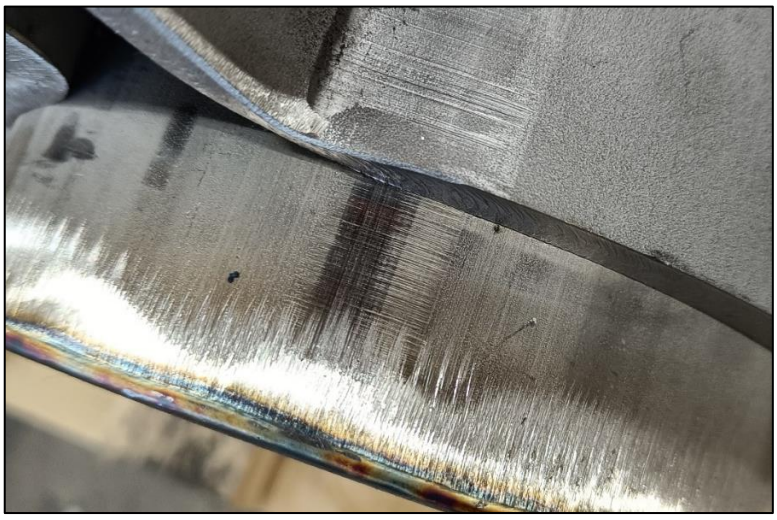

Figure 36. The scratches left on the shaft after the shaft was pressed through the hub displaying the contact area.

To achieve a more even contact area, additional rigidity could be added to the hub or a slightly larger radius for the hubs could be tested which could allow for better initial contact to be achieved and the bolt preload would then pull the sides more evenly on to the shaft. A jig could also be made that holds the hub in its intended shape during the welding of the paddle that could lessen the effects of the distortions. Currently no such jig is used. The shafts are welded circular profiles and suffer from distortions as well which can contribute to the fitting problems and cannot be fully avoided unless also individually addressed.

From a fatigue standpoint, a rigid connection is better for bolted connections. The bolted connection should not experience significant fatigue load but without deeper examination it cannot be assumed to be entirely negligible. Fatigue could prove to be an issue, but it is not examined in this thesis.

6 Discussion

In this section, the used methods, their efficacy, and validity are discussed. The achieved results are examined and their relatability to real use cases is inspected. Lastly, possible further research is addressed based on the questions and observations which rose during this study.

6.1 Used methods

Simplified loading was used throughout the analysis, which differs greatly from the described realistic loading. The limits were set to keep the scope of the study reasonable and to achieve more widely applicable results. For shafts, the main loading types of torque and bending moments were able to be applied in the analysis, and other loadings were excluded. The combination of torque and bending moment differ from one application to another, which limits the applicability of the results, but from the point of view of the stiffeners, only the bending moment is critical.

In-depth CFD analysis was deemed to be too extensive for this thesis which heavily limited the possibilities for the analysis. It was also in the company's interest to apply methods that are more likely to give results on the safer side which resulted in the simplistic loading. When inspecting the company's previously conducted CFD of the paddles, the hydraulic forces concentrated mainly on the leading edge but also on the top and bottom surfaces of the paddle. In the used loading, loading was only applied onto the leading edge which does not account for the loading in the axial direction relative to the shaft. When introducing loading to the surfaces, it was deemed that the stiffeners have a better effect on the buckling capacity. It would have been beneficial to include some of the loading in the analyses, but difficulties similarly rise as to how much loading should be applied to the surfaces compared to the leading edge. This would be affected by the process itself, the amount of thrust generated and how close the paddles are to the bottom of the tank among other factors.

No initial imperfections were assumed in the welds or the geometries in general, which is not representative of the real products. Although, with ductile materials in static examinations, their effects are not as significant. When examining the peak stresses of the

products, it could be assumed that solutions which do not introduce significant notches would experience higher peak stresses than what the analysis shows due to small notches or other variations in the geometries caused by the fabrication process. Sufficient penetration is also assumed, which might not always be achieved. Non-destructive testing (NDT) is not conducted for every weld which leaves a possibility for welding mistakes and poor penetration to occur.

Realistic results can be achieved with FEA, but as shown with the hub experiments, all the aspects that can affect the results such as welding distortions, might not be considered. They are idealistic results in nature and should always be treated as approximate. Multiple similar analyses were done in total which eliminates most of the probability of singular mistakes in the analyses, and a good understanding of what should be expected from the results was achieved which made identifying any clear mistakes easier.

6.2 Achieved results

Consistent results were achieved with FEA, but the validity of the results have not been proven in practice. Generally, the results of the current structures were within expectations which provides some validity to the results. The experimental test done to the hubs proved, that the FEA results of the hubs were predictive of the results but not indicative of the true behaviour. Many of the sources that cause inconsistencies between FEA results and true behaviour stem from the idealistic nature of the analyses which could be accounted for but requires significantly more in-depth analysis.

When considering the structural integrity of the used products, often hefty safety factors are included, or extreme cases are used as the basis for the design. Although the agitators themselves are not expensive enough to warrant such safety factors, the effects of an agitator malfunction can lead to a plant-wide process halt where the costs exceed the agitator’s price multiple times. Obviously, faulty products hurt the company’s reputation significantly and for a smaller company competing among larger companies, it can have severe consequences if they are deemed unreliable. Based on this the safer nature of the results does not invalidate them but a deeper understanding of the behavior of the structures would still be beneficial from a cost-efficiency standpoint.

The underlying understanding of the necessity of all the stiffeners is still unclear, and as it relies heavily on the application, it must be determined on a case-by-case basis. The question becomes, whether to design the structures to have excess capacity, keeping the design process simple but bear the unnecessary costs in the fabrication process, or whether to spend more time and optimize the design to its application which leads to more cost-efficient production.

6.3 Further development

Generally, all proposed changes should be tested to validate the results and to ensure their safety. The shaft stiffeners are mainly deemed unnecessary, if the braces are removed, the quality of the flange weld and especially penetration needs to be sufficient. With poor penetration the root of the weld could prove to be critical in fatigue loading and an ensuring quality system might need to be set in place. Any further development could focus on the flanges, as one of the designs decreased the possible peak stresses experienced which could alleviate any of the problems that could arise from fatigue loading. The fabrication process of the proposed flange would have to addressed as the current machining process would lead to undesirably higher costs.

Although fatigue has not been considered a critical failure mode, it was considered in the design of the stiffeners from the premise of future proofing the structures by reducing stress concentrations and by designing them to be able to withstand higher loads while maintaining good cost-efficiency of the production. For the shafts, the root side of the flange weld was determined to be critical for fatigue, but the root side was not examined as the focus was only on inspecting under static loading. The fatigue capacity of the structures could be examined further, which would provide further clarity of whether fatigue should be considered in the future design of the agitators.

The paddle analyses suffered severely from the set limitations to the geometries. In-depth CFD analysis would need to be integrated into the analysis to achieve more realistic results. Similar issues would still need to be addressed, as in will the paddles be inspected based around their most critical applications, where any changes to the designs would lead to excess load bearing capacity in their more usual applications, or would they be inspected by

their most typical application, leading to less widely applicable results. Ultimately, some of the design choices will always have to be done based on the specific application.

Based on current knowledge, it is assumed that the tested hub designs do not experience significant fatigue loading. The current and the tested hubs are still similar in design, where the tested hub allows for more deformations compared to the current more rigid solution, which is less ideal from the fatigue standpoint. It could also be assumed that most of the loading is directed more towards the hub and shaft connection rather than to the bolted connection. Testing could be conducted to ensure that the hub bolts do not experience much fatigue loading, to not only rely on assumptions. Based on deeper analysis, some additional braces could be added to lessen the deformations in the hubs or a completely different designs could be considered.

The tested hub design could be bent from a single plate which would completely remove welding from the hub fabrication process, further reducing costs. A press is used to achieve the main shape of the hub and the additional bend could be included in the pressing process. At the time of the study, it was found plausible that the entire hub could be pressed into shape with the existing machinery, but further analysis still needs to be conducted beyond this study to determine whether it is applicable. Some possibilities can arise from the predrilled holes which can deform into an oval shape during the pressing process if they are left too close to the bends. This either adds length to the flange, or the holes need to be drilled or adjusted after pressing.

7 Conclusions

The aim was to analyse the vertical agitators of Mamec Oy, with the focus being on components which employ various stiffeners. The studied components are shafts, paddles, and the hubs that connect the paddles to the shafts. The structures were first analysed without any stiffeners to achieve a general understanding of their behaviour, after which they were analysed with the currently used stiffeners. Based on these results, improvements to the stiffeners were proposed, analysed and compared to the previous results to measure the effectiveness and production cost-efficiency of the different solutions. The analyses were mainly conducted with FEA, but for the hubs, due to the difficulties in determining their realistic functionality, experimental tests were also conducted.

For the shafts, the currently used stiffeners were found to be unideal from a structural point of view, causing higher peak stresses which could be problematic if considerable fatigue loading is experienced. If sufficient penetration and quality is achieved for the flange weld, the stiffeners are not needed, but if these are not achieved, the current stiffeners do shift the critical point from the flange weld to the tips of the stiffeners, allowing lower quality requirements for the flange weld. Alternate designs provided comparable solutions to the current stiffeners, mostly with lower stress peaks and slightly higher manufacturing costs. The most optimal design most likely can be achieved without any stiffeners if sufficient weld quality can be ensured. Alternate flange designs were also considered which offered better solutions, especially from a fatigue durability standpoint, but proved to be too expensive to be manufactured with the current fabrication methods, requiring alternative methods to be researched.

The paddles were studied with many limitations, which resulted in alternative solutions which did not offer clear improvements to the paddles load bearing capacity. The currently used stiffener is simple in its design and offers some reinforcement. All the stiffeners, including the current solution, suffer from size limitations, which are set in place to not induce additional turbulence or vortices to the agitation process. This heavily limits their effectiveness to restrict buckling, as the critical areas cannot be reached by the stiffeners. Increasing the thickness of the paddle proved to be a feasible alternative to the stiffeners,

which lead to much higher buckling capacities, especially with smaller paddle sizes where the added material costs stay adequate.

Analyses of the hubs and varying solutions were conducted but the results were not conclusive, as the quality of the friction connection between the hubs and the shafts was difficult to determine. The quality of the connection is also affected by many variables, such as distortions caused by welding, which cannot be accounted for in idealistic FEA. The experimental tests allowed the FEA results to be validated. The new proposed design decreased the amount of welding and manufacturing steps significantly which led to a more cost-effective solution.

For further studies, fatigue emerged as a possible failure mode for the structures but was not addressed thoroughly in this study. The inherent loading of the structures is variable but is most likely not significant enough as no fatigue related failures have been experienced so far. A deeper understanding of the fatigue loading and its effects could still be beneficial and give more confidence to the designed lifespan of the products. Comprehensive CFD analysis would also provide a better foundation for the analysis of the structures, as it would allow a more realistic loading to be applied.

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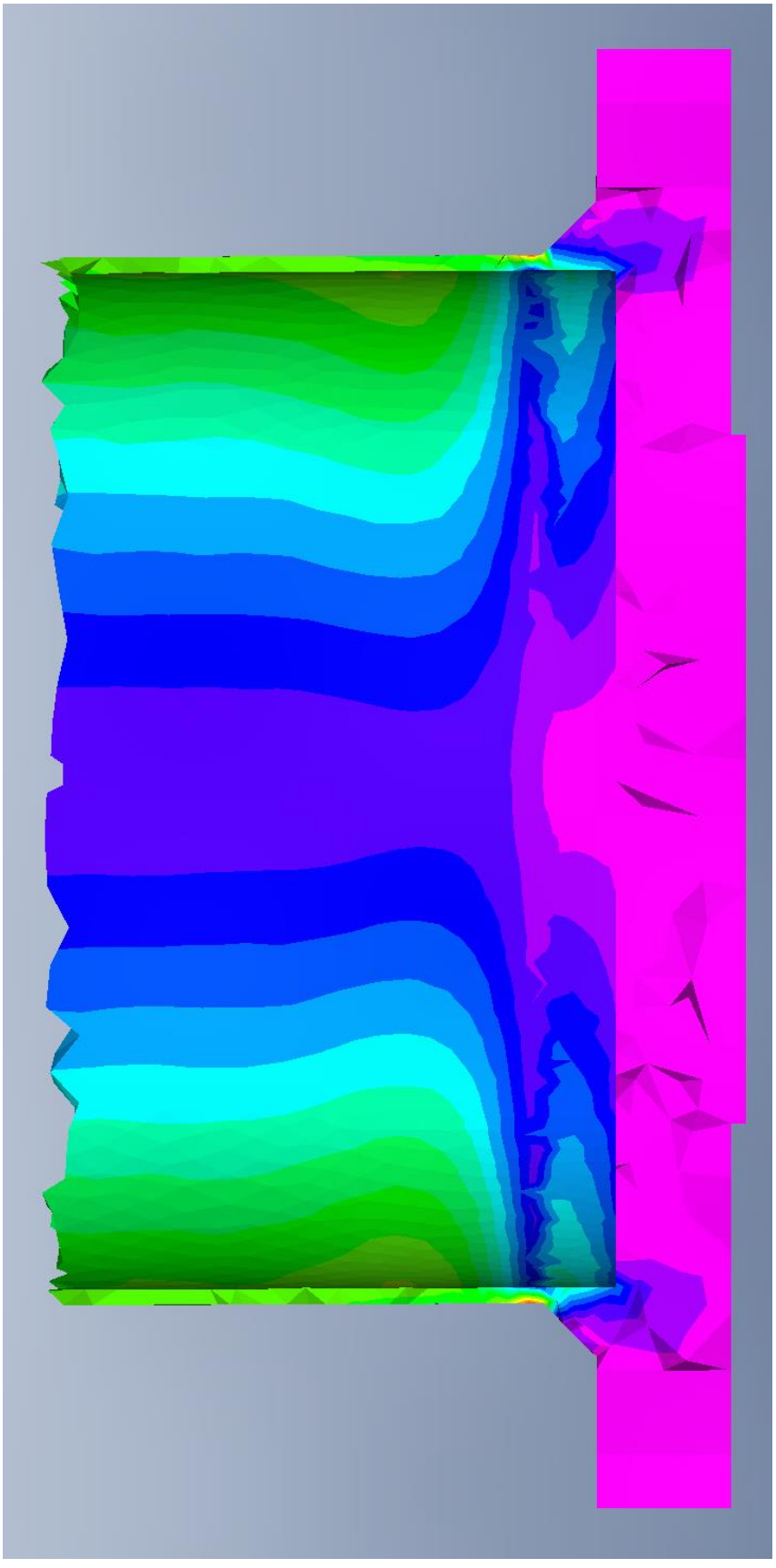
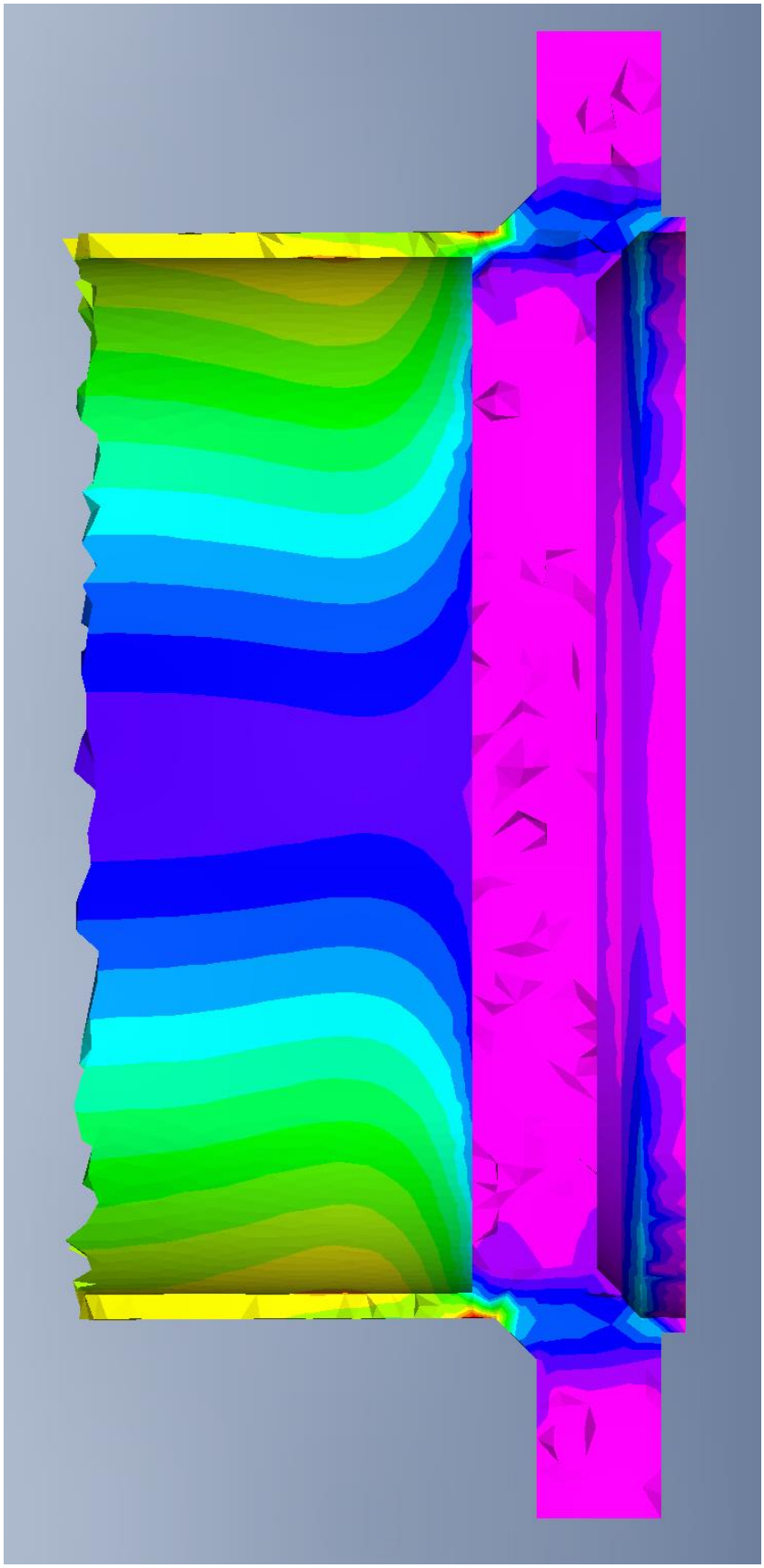
Appendix 1. Cross section of the larger shaft.



Appendix 2. Cross section of the smaller shaft.