Model-scale podded propellers for maritime research

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Model-scale Podded Propellers for Maritime Research

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Podded propellers constitute a major new development in ship design, especially for cruise liners but also increasingly for freight ships. A podded propeller (pod) consists of an electric motor which directly drives the propeller. This motor unit is located inside a gondola that is hanging from a strut. This strut has a rotating connection to the ship. Advantages with respect to conventional propulsion are an increase in fuel efficiency (up to 10%), better manoeuvrability and low vibrations. However, recent service breakdowns have slowed down the market. The availability of highly accurate model-scale measurements may help to discover the causes of failure in these 23 MW devices.

The Maritime Research Institute Netherlands (MARIN) provides the maritime industry with performance predictions and design consultancy. Part of this is based on model-scale tests performed in large water basins, so-called ‘towing tanks’. The ability to perform highly accurate model-scale experiments with pods is a major challenge for MARIN.

As yet, all model-scale experiments with pods have been performed with electric motors external to the pod, using bevel gears to drive the propeller axis. This set-up limits the geometric modelling flexibility. Also, it introduces vibrations, which make it difficult to measure dynamic loads which are essential for analysing manoeuvring tests. Further, the six-component load balance that measures the propulsive force of the pod-unit does not deliver the <1% accuracy that is required. The goal of the project was to solve the existing technology problems and to design a new pod model. With this new pod model, new testing opportunities will be created. A successful new design will generate a new impulse to pod research as well as new market opportunities for MARIN.

The key to reaching this goal is to design a pod model with the powering motor housed inside the gondola. Hydrodynamic scaling laws in combination with flexibility requirements demand a very high power density from the drive motor; up to 1.6MW/m³, which is about 2-3 times larger than the full scale power density. This makes the search for and the design of the drive motor a challenging task.

Of the available choices – a hydraulic or an electric motor – a hydraulic motor has the highest power density. However, hydraulic solutions have many negative side effects like: low user-friendliness and disturbance of the pod-unit force measurements. Most electric motors lack power density when used according to supplier specifications, which is the reason that currently no solutions are available in the field. The key step is to utilise the towing tank water as efficient cooling medium while accepting a reduced lifetime. With these boundary conditions, the supplier specifications can ‘thoughtfully’ be overruled. For an electric motor, this approach leads to an increase in power density by a factor of 2-4, so that sufficient shaft power can be achieved.

In hydrodynamic research on pods, several standard types of experiment on model scale exist which all imply different requirements for the pod model. As a result, no single generic solution exists Therefore, a toolbox has been defined which provides direct propeller drive
solutions, with the propeller drive motor located inside the pod gondola. The toolbox covers over 80% of all model-scale experiments with pods at MARIN. A direct propeller drive is the core of solving problems with the existing devices.

In this project, the two most relevant tools for MARIN have been designed and tested successfully. First, a small scale (1:30) basic pod model has been realised, which is especially suited for manoeuvring and sea-keeping tests. Here, the drive motor is an electric motor with a gear box, which was optimised to deliver 3.5 times its supplier-rated power. No measurements at the propeller shaft are possible. However, the measured motor current can be used to provide a measure for the motor torque with an accuracy of 1% up to a measurement frequency of 10 Hz. This measurement frequency is sufficient for manoeuvring and sea-keeping tests.

Second, a direct propeller drive has been realised for large-scale (1:20) powering experiments. This design path yielded a more complex, large-scale pod model which is equipped for most regular powering optimisation tests. The drive motor was purchased as a separate rotor-stator package and optimised through water cooling to deliver a torque up to 17 Nm, which is about 3 times the supplier-rated limit and 50% higher than required. Using separate motor parts allowed full integration of the direct-drive motor and the force sensor in the propeller shaft.

The six-component force balance that is used to determine the propulsive forces of the pod-unit was found to lack accuracy, with typical errors of 2-5%, depending on the loading case. The existing balance is also not suitable for dynamic measurements as of its low resonance frequency of about 10 Hz, where 80 Hz is required. Therefore, first the correct implementation of underlying theory in calibration routines and data processing has been reviewed and updated. Also, a new six-component, pod-unit force balance has been designed, to fit the state-of-the-art accuracy requirements. Measurement errors could be reduced to 1.5% for complex combined loads. Finally, dynamic pod-unit force measurements during manoeuvring have been proven feasible up to a measurement frequency of 40 Hz, which equals a typical first blade harmonic.

The current results also form the starting point for further design efforts. First, the developed concepts have to be implemented in the operational towing tank processes. Further, the next major challenge is to further improve the data quality of dynamic pod-unit force measurements during manoeuvring.

The design of an innovated pod model has removed a major obstacle on the way to new model-scale testing opportunities. Existing test services will improve when the current concepts are implemented. This will consolidate MARIN’s leading position in model-scale research on pods. Furthermore, with the developed tools MARIN is prepared for future developments, thus capable to stay ahead of global competitors.

Finally, the innovated pod model provides the maritime community with a range of opportunities to gain deeper knowledge on podded propulsion. This knowledge will help to strengthen the position of pods in the maritime market. In this way, podded propellers can continue to grow, offering their benefits to the maritime industry and the global society.
Samenvatting

Podded propellers (pods) zijn een belangrijke recente ontwikkeling op het gebied van scheepsontwerp. Pods worden vooral veel toegepast in cruiseschepen, maar ook steeds vaker in vrachtschepen en veerboten. Een pod bestaat uit een elektromotor die direct op de schroefas bevestigd is. Deze motor is geplaatst in een gondel die aan een zgn. strut hangt, deze strut heeft een draaibare verbinding met het schip. De voordelen van pods zijn een verhoogde brandstof efficiëntie (tot 10%), betere manoeuvreerbaarheid en minder trillingen. Recent hebben operationele problemen en uitval tijdens service ervoor gezorgd dat de markt voor pods is verzwakt. De beschikbaarheid van nauwkeurige metingen op modellschaal zou kunnen helpen om de oorzaken van het falen van deze 23MW grote machines te achterhalen.

Het Maritime Research Institute Netherlands (MARIN) voorziet de maritieme industrie van prestatie voorspellingen en ontwerp adviezen. Een belangrijk deel van dit werk is gebaseerd op modellschaal testen in zogenaamde 'sleptanks'. Het is voor MARIN een cruciale uitdaging om zeer nauwkeurige modellschaal testen met pods uit te kunnen voeren.

Tot nu toe, worden alle modellschaal experimenten met pods uitgevoerd met een elektromotor buiten de pod, waarbij rechtehoek tandwieloverbrengingen het vermogen naar de schroef overbrengen. Deze modelpod opstelling legt sterke geometrische beperkingen op. Verder veroorzaken de tandwielen trillingen, die het moeilijk maken om de dynamische metingen uit te voeren die essentieel zijn voor het analyseren van manoeuvreerproeven. Verder levert de zescomponenten balans voor het meten van de pod-unit voortstuwingskrachten niet de vereiste nauwkeurigheid van <1%. Het huidige project had tot doel om de bestaande technische problemen op modellschaal op te lossen en een nieuwe modelpod te ontwikkelen. Met deze nieuwe modelpod zullen nieuwe testmogelijkheden worden gecreëerd. Een succesvol nieuw ontwerp zal een nieuwe impuls generen in het onderzoek naar pods en derhalve nieuwe commerciële mogelijkheden voor MARIN.

De sleutel tot het bereiken van dit doel is het ontwerp van een modelpod waarbij de aandrijfmotor zich in de gondel van de pod bevindt. Door de gebruikte hydrodynamische schaalwetten en de vereiste flexibiliteit wordt er een hoge vermogensdichtheid van 1.6MW/m³ gevraagd, dit is 2 tot 3 keer zo hoog als de vermogensdichtheid op ware grootte. Hierdoor is het zoeken en ontwerpen van een aandrijfmotor een lastige uitdaging.

Van de twee mogelijke opties, − een hydraulische of een elektrische motor − heeft een hydraulische motor de hoogste vermogensdichtheid. Echter, vergeleken met een elektromotor heeft hydrauliek veel negatieve consequenties zoals een lage gebruiksvriendelijkheid en verstoring van de pod-unit metingen. De meeste elektromotoren lijken te weinig vermogensdichtheid te hebben als ze volgens de specificaties van de leverancier gebruikt worden. Dit laatste is dan ook de reden waarom er op dit moment nog geen oplossing beschikbaar zijn met een elektrische motor in de gondel. De cruciale stap richting een oplossing, is het gebruik van het sleeptankwater als een effectieve koeling, gecombineerd met het weloverwogen overschrijden van de specificaties van de leverancier. Dit leidt tot een verkorte levensduur, die
echter voor een sleeptank omgeving geheel acceptabel is. Op deze manier kan het vermogen aan de schroefas worden vergroot met een factor 2 tot 4.

Bij het hydrodynamisch onderzoek naar schepen worden verschillende typen modellschaal experimenten uitgevoerd die allemaal hun eigen eisen stellen aan de testinstrumentatie. Als gevolg daarvan is het niet mogelijk een algemeen toepasbare oplossing te realiseren. Daarom is een ‘toolbox’ gedefinieerd met drie verschillende oplossingen voor een directe schroefaandrijving in de gondel. Met deze gereedschappen kan meer dan 80% van alle modellschaal experimenten met pods worden uitgevoerd. Een directe aandrijving in de gondel is de kern van de oplossing voor de problemen met bestaande modelpods.

De twee, voor MARIN, meest relevante tools zijn ook daadwerkelijk ontworpen en succesvol getest. Eerst is een eenvoudige modelpod gerealiseerd voor zeegang en manoeuvreer experimenten op kleine schaal (1:30). Hierbij is de aandrijfmotor een elektromotor met een tandwielkast die zo is geoptimaliseerd dat de motor 3.5 keer het vermogen kan leveren dat door de leverancier is gespecificeerd. Bij deze modelpod zijn geen directe metingen in de schroefas mogelijk. Uit de gemeten motorstroom kan echter wel het geleverde schroefaskoppel worden afgeleid met aan nauwkeurigheid van 1% tot een meetfrequentie van 10 Hz. Deze meetfrequentie is hoog genoeg voor zeegang en manoeuvreer experimenten op kleine schaal (1:30).

Daarnaast, is een tweede, meer complexe, directe aandrijving gerealiseerd met een geïntegreerde schroefassensor. Dit ontwerp is geschikt voor grote schaal (1:20) voortstuwingsexperimenten. De elektrische aandrijfmotor is aangeschaft als een los rotor-stator pakket. Het motorvermogen is geoptimaliseerd door waterkoeling tot een maximaal koppel van 17 Nm, wat ongeveer drie keer zo hoog is als opgegeven door de leverancier en wat 50% hoger is dan vereist voor dit type toepassingen. Door het gebruik van een los rotor-stator pakket is maximale integratie van de assensor en de motor mogelijk gemaakt.

De bestaande zes componenten balans die wordt gebruikt om de voortstuwingskrachten van de pod te meten is onnauwkeurig gebleken, wat leidde tot fouten in de orde van 2-5%, afhankelijk van de specifieke krachtencombinatie. Verder is de bestaande balans ook niet geschikt om dynamisch mee te meten doordat de resonaanse frequentie rond de 10 Hz ligt, terwijl 80 Hz vereist is. Daarom is eerst een belangrijke randvoorwaarde voor nauwkeurige zes componenten pod-unit metingen verbeterd, namelijk de correcte implementatie van de onderliggende theorie in calibratie routines en de dataverwerking. Daarnaast is een nieuwe pod-unit balans ontworpen die het mogelijk maakt om met < 1.5% nauwkeurig te meten. Tenslotte is aangetoond dat dynamische pod-unit metingen tijdens manoeuvres mogelijk zijn met een meetfrequentie van 40 Hz wat overeenkomt met een typische eerste bladharmonische.

De huidige resultaten vormen ook het startpunt voor verdere ontwikkelingen op het gebied van modelpods. In de eerste plaats moeten de ontwikkelde concepten geïmplementeerd worden in de dagelijkse sleeptank processen. Verder is de eerstvolgende belangrijke uitdaging het verbeteren van de datakwaliteit van dynamische pod-unit metingen tijdens manoeuvres.

Het ontwerp van een innovatieve modelpod heeft een belangrijk obstakel uit de weg geruimd op weg naar nieuwe experimentele mogelijkheden op modelschaal. Bestaande test services zullen verbeteren als de huidige concepten in de praktijk worden toegepast. Dit zal de leidende marktpositie bevestigen die MARIN heeft op het gebied van modellschaal onderzoek naar pods. Daarnaast zijn met het ontwerp van deze innovatieve modelpod mogelijkheden gecreëerd voor nieuwe test producten die anticiperen op toekomstige marktvragen. Op deze manier kan MARIN zijn voorsprong op de concurrentie wereldwijd blijven behouden.

Tenslotte biedt de innovatieve modelpod een scala aan mogelijkheden aan de inter-
nationale maritieme industrie om een dieper inzicht te verwerven in pods. Deze kennis zal helpen om de positie van pods in de maritieme markt te versterken. Op die manier kan het marktaandeel van pods verder blijven groeien en kunnen zowel de maritieme industrie als de internationale gemeenschap blijven profiteren van de voordelen van pods.
List of symbols

\begin{align*}
V_a & \quad [\text{m/s}] \quad \text{forward speed} \\
n & \quad [\text{rpm}] \quad \text{propeller rotation rate} \\
J & \quad [-] \quad \text{dimensionless forward speed} \\
\rho & \quad [\text{kg/m}^3] \quad \text{density} \\
D & \quad [\text{m}] \quad \text{propeller diameter} \\
\alpha & \quad [\text{°}] \quad \text{steering angle} \\
(F_x)_{\text{prop}} & \quad [\text{N}] \quad \text{propeller thrust} \\
(M_x)_{\text{prop}} & \quad [\text{Nm}] \quad \text{propeller torque} \\
(F_x)_{\text{unit}} & \quad [\text{N}] \quad \text{pod-unit thrust} \\
(F_y)_{\text{unit}} & \quad [\text{N}] \quad \text{pod-unit lateral force} \\
(F_z)_{\text{unit}} & \quad [\text{N}] \quad \text{pod-unit lifting force} \\
(M_x)_{\text{unit}} & \quad [\text{Nm}] \quad \text{pod-unit bending moment} \\
(M_y)_{\text{unit}} & \quad [\text{Nm}] \quad \text{pod-unit lateral bending moment} \\
(M_z)_{\text{unit}} & \quad [\text{Nm}] \quad \text{pod-unit steering torque} \\
C_P & \quad [\text{MW/m}^3] \quad \text{power density of the propeller drive motor} \\
K_F & \quad [-] \quad \text{dimensionless force } F \\
K_M & \quad [-] \quad \text{dimensionless moment } M \\
Re & \quad [-] \quad \text{Reynolds number} \\
F_n & \quad [-] \quad \text{Froude number} \\
I_{\text{motor}} & \quad [\text{A}] \quad \text{motor current} \\
t & \quad [-] \quad \text{coverage factor} \\
U_i & \quad [%] \quad \text{combined measurement uncertainty of quantity } i \\
\theta_i & \quad [-] \quad \text{dimensionless sensitivity of the measurement result to quantity } i \\
R & \quad [-] \quad \text{result of the measurement} \\
\%_{\text{FSO}} & \quad [-] \quad \text{percentage of the full scale output of a sensor} \\
\lambda & \quad [-] \quad \text{scaling factor} \\
x_s & \quad [-] \quad \text{quantity } x \text{ on full scale, subscript } s \text{ denotes } \textit{ship} \\
x_m & \quad [-] \quad \text{quantity } x \text{ on model scale, subscript } m \text{ denotes } \textit{model}
\end{align*}
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1 Introduction

Abstract
Podded propulsion has become a major technology in ship propulsion. Model-scale testing generally forms an essential part of the design procedure of ships equipped with podded propellers. Globally, a limited number of maritime research institutes (about 7) govern the global shipbuilding research market. For maritime research institutes, it is of major importance to be able to perform reliable model-scale tests of ships with podded propellers. A crucial step forward in model-scale testing of pods is the new innovated design of model-scale podded propellers described in this thesis. It describes this innovation that facilitates the current and future model-scale testing needs of the maritime industry.

1.1 Maritime research in the Netherlands

Ever since man populated the Netherlands, the Dutch have challenged wind and waves. During history, it became part of the Dutch identity to investigate ways to conquer the oceans and to deal with the caprices of wind and waves. Over time, the focus gradually shifted from the bare struggle to survive towards profitable shipping in adverse conditions. In the latter era of hydrodynamic research, the Maritime Research Institute Netherlands MARIN has played a major role.

MARIN has been dedicated to extending maritime knowledge in its broadest sense since 1932. Today, it has become an internationally recognised authority on hydrodynamics which is involved in many frontier-breaking research programs. Also, MARIN provides the maritime and offshore industry with state-of-the-art performance predictions, design consultancy and testing services [34]. Globally, MARIN is one of the leading institutes together with about 7 other institutes of similar size and importance.

A major part of these research and consultancy activities is based on model-scale experiments in large water basins, so-called towing tanks (Figure 1.1 and Appendix A). In a towing-tank experiment, a model-scale replica of some hydrodynamic structure - generally a ship - is tested.¹ This ship model is equipped with sensors that measure relevant important hydrodynamic quantities like resistance, propulsion forces, course stability and safety in waves. The ship model is exposed to a range of varying conditions during which the ship’s response is measured (Figure 1.2). These conditions may include various sailing speeds and propeller rotation rates, different wave conditions, steering patterns or even varying wind speed and direction.

Model-scale measurements in towing tanks are used to predict full-scale quantities like propulsion forces, fuel efficiency and manoeuvrability. Scaling laws determine how these full-

¹The focus of this thesis is on ships, therefore in the following, only ship models will be considered.
1. INTRODUCTION

Figure 1.1: One of MARIN’s most advanced towing tank facilities; The Seakeeping and Manoeuvring Basin.

Figure 1.2: Model-scale testing of a motor yacht in the Seakeeping and Manoeuvring Basin.

scale quantities can be deduced from model-scale measurements. Which scaling law is to be used depends on the type of experiment. Through scaling, small deviations on model scale can be magnified to large full-scale errors. Therefore, to achieve reliable predictions of full-scale quantities, the towing tank conditions need to be controlled carefully and measurements need to be very reliable, generally with an accuracy of 0.5-1.0%. Chapter 2 discusses model-scale research and its consequences for the current design in more detail.

1.2 Podded propellers

One of the latest innovations to make shipping more profitable and European shipbuilding more competitive is the podded propeller. During the last 25 years, podded propellers (or in short: pods) were a major development in ship design. Pods have been applied in many cruise liners and are more and more applied in other ship types like ferries and freight carriers [7].

A podded propeller (pod) consists of an electric motor which directly drives the propeller. This motor unit is located inside a gondola that is hanging from a strut. This strut has a rotatable connection to the ship (Figure 1.3). The propeller can be pulling (in front of the pod) or pushing (behind the pod). Also, pods exist with two propellers which either co- or contra-rotate. The majority of all single propeller pods have a pulling propeller which has a nearly undisturbed inflow. Figure 1.4 shows examples of pods produced by different manufacturers.

Originally, pods were developed to enhance the manoeuvrability and flexibility of icebreakers [59]. As a consequence of the free propeller inflow, the interaction between the propeller and the ship body is minimal. As a result, the vibration levels are lower than in ships with a regular propeller in combination with a rudder. Therefore, pods also offer a considerable
1.2 Podded Propellers

Figure 1.3: Schematic representation of podded propellers with main parts indicated.

An advantage for cruise ships, where comfort is of major importance. An advantage that is important for almost any ship type is an increased fuel efficiency which is expected to reach up to 10% [59, 62]. Given the finite conversion efficiency of a diesel combustion engine to electrical power, this figure demonstrates the large gain in hydrodynamic efficiency. Finally, the main diesel engine does not need to be in line with the propeller shaft. Therefore, pods offer more flexibility in the location of the engines that drive the electrical generators (general arrangements).

Every new ship with innovative technology like pods needs to be tested on model scale before it is actually built. Therefore, for a maritime research institute like MARIN, it is of great commercial importance to be able to perform reliable model-scale measurements on ships with pods. Until now, MARIN has been market leader in performing model-scale experiments with pods for commercial shipbuilders. Also, the model-scale experiments for several international research projects on pods were carried out at MARIN.

Until recently, the existing model-scale units were sufficient to provide the testing services required by the industry. As podded propulsion technology grew more mature, more complex and detailed questions arose and more sophisticated (full scale) pods have been developed. As a result, the inherent drawbacks of the existing model-scale pod units (pod models) at MARIN have turned into intolerable limitations in model-scale testing. To keep its leading position, MARIN has to keep ahead of the market. Hence, model-scale testing capabilities should match the dominating research issues in the podded propulsion industry.

The necessity to gain a better understanding of the forces and moments in pods becomes apparent from several serious service breakdowns that have occurred recently with existing ships equipped with pods (Appendix B). Majority of these breakdowns were caused by unexpected high loads on the bearing and suspension of the pod and the propeller. Such pod problems are partly caused by the fast development of the pod; in 10 years time a scale enlargement took place from approximately 1MW to 23MW for a single pod. This rapid
1. INTRODUCTION

(a) The Azipod, produced by ABB, Finland. [1]
(b) The Compact Azipod, produced by ABB, Finland. [1]
(c) The SSP produced by the Siemens-Schottel consortium, Germany. [55]
(d) The SEP produced by Schottel, Germany. [55]
(e) The Mermaid, produced by the Rolls-Royce-Kamewa consortium, Sweden. [30]
(f) The Dolphin, produced by STN Wärtsila, Netherlands [64]

Figure 1.4: Full-scale pictures of existing pods from different manufacturers.
development together with a quick introduction into the market seems to have outpaced the full understanding of pods.

The service problems with existing pods have caused the pod market to stagnate. However, pods have many benefits to offer, to many areas of the maritime industry. Therefore, ongoing research is projected to improve full understanding of the pods bearing and suspension loads and to further optimise fuel efficiency. This will allow for improved designs of new pods and will prevent existing units from future service breakdowns. This future model-scale research is expected to have a strong focus on dynamic measurements of the instantaneous pod forces and structural loads during transient conditions. The current thesis describes the design of an innovated pod model which suits the demands of the current podded propulsion industry.

1.3 Why innovation of model-scale pods?

The following describes why the existing pod models need to be redesigned. Model-scale testing at MARIN is often part of industrial consultancy. For this purpose, the towing tanks and the model production facilities should be considered as non-stop production facilities. Further, profitability margins on model-scale tests are generally small. This implies that only limited time can be spent on building ship models. So, to be able to build model-scale versions of all pod shapes that exist on the market (Figure 1.4), very flexible testing tools are needed.

At MARIN, this flexibility is realised through the use of standardised units containing all important drive and measurement components. Around this standard unit, a shell is placed, which on the outside is an exact geometric scale model of the full-scale pod to be tested. This shell is made of synthetic material which is cheap and easy to manufacture, generally polyurethane or PVC.

Until now, the usual approach to build the standardised drive units has been the use of a so-called Z-drive: an external electric motor, placed in the ship, drives a mechanical unit with bevel gears which transmits power to the propeller shaft (Figure 1.5). The propeller shaft contains a sensor to measure propeller thrust and torque. A slip-ring unit transfers the data from the shaft to the fixed world. Inherent to the use of right-angle bevel-gears are the following drawbacks that seriously restrain model-scale testing on pods:

- New innovative pod shapes like ABB’s compact Azipod (Figure 1.4(b)) cannot be modelled correctly through the geometric restrictions of the right-angle bevel gears.
- The required angle between the propeller shaft and rotational axis often cannot be realised as this angle is fixed to $90^\circ$ by the gear transmission.
- Dynamic-force measurements are seriously disturbed through vibrations introduced by the gear train.
- Accurate measurement of the pod-unit forces is difficult and error-prone due to the complex mechanical interactions in the current set-up.
- Active steering in combination with accurate six-component pod-unit measurements is not possible.

Hence, the existing equipment should be improved in flexibility, reliability and accuracy, to meet the demands of the podded propulsor industry. In this way, a new design will help
1. INTRODUCTION

(a)

Drive motor
Pod unit forces
Propeller shaft measurements
Right-angle bevel gears
Data transfer

(b)

Figure 1.5: Picture (a) and schematic overview (b) of a ship model with four pods built according to the existing set-up with an external drive motor and bevel gears to transmit the power to the propeller shaft.

to solve the urgent questions that currently restrain the pod market. A complete redesign is required to realise a conceptual breakthrough that leads to a pod model suited for modern-day maritime research. Section 1.4 further elaborates on the goal of the current design project and the approach towards an innovative solution. A more detailed analysis of the problems with the existing set-up is given in Chapter 3.

1.4 Removing obstacles

The full potential of podded propulsion cannot be utilised as long as crucial questions on the dynamic loads stay unanswered. Certain answers can only be found through detailed model-scale testing. Currently, the limited model-scale testing capabilities prevent a better understanding of pods. To keep ahead of the market and to maintain its leading market position in model-scale testing on pods, MARIN has to innovate its pod models. Therefore, the goal of the project is to be able to serve the maritime industry with model-scale testing equipment that meets the demands of the current pod market. To achieve this goal, an innovated pod model has been designed.

The propeller drive method is predominantly responsible for the limitations of the test set-up. Hence, the most straightforward way to solve the existing problems is by eliminating all right-angle transmissions and to switch to a direct-drive motor inside the pod gondola, similar to the full-scale reality. However, at the start of the project, none of the maritime testing institutes was known to have a set-up other than an indirect transmission with a motor external to the pod.

The reason for this conceptual deadlock in pod model testing is that the required shaft power is high. Through scaling laws and model-scale test conditions that are more demanding than the full-scale design conditions, the required power from a motor that fits into many different model-scale pod shapes is higher than what can be achieved in a straightforward way with regular electric motors. Hence, designing a propeller drive solution which can supply sufficient power is a challenging task.

Once such a unique conceptual breakthrough can be realised, other problems that limit model-scale testing capabilities can be solved and new opportunities will be created. These
include accurate measurement of pod-unit forces in combination with active steering, dynamic measurement of pod forces and moments, improved flexibility, and reduced vibrations. A detailed specification of the project result is given in Chapter 3.

1.5 Relevance

The advantages of podded propulsion can be beneficial for a much wider range of ships than cruise liners and ice-breakers only. Pods already found their way to yachts, ferries and navy ships. However, very few pods have been applied in freight carriers and tankers. An increased market share of pods in the propulsion industry will lead to a significant decrease in fuel consumption (Table 1.1), an increase of efficiency and improved safety. However, with the current reliability issues mentioned in the previous section, for many shipbuilders, the risks are too high to make the switch from conventional propulsion to pods.

Further research is needed to both regain the confidence of the market in pods and to further exploit the advantages of pods. The result of the current project is a key step towards a wider and more profitable application of pods. With the knowledge from this project, MARIN will have a strong advantage on its global competitors. If limited testing capabilities no longer restrain new developments and research, podded propellers can continue to grow, offering their benefits to the maritime industry and the global society.

Table 1.1: Facts (based on internet resources [3, 4, 5]) on the potential global impact of pods, assuming 10% fuel reduction in 50% of all freight carriers. The 10% fuel reduction is an assumption of the eventual gain in efficiency based on [59] and [62]. These figures should be considered as illustrative only.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
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</thead>
<tbody>
<tr>
<td>Globally, total maritime transportation:</td>
<td>$2 \times 10^{10}$ ton-miles/year</td>
</tr>
<tr>
<td>Average fuel efficiency:</td>
<td>500 ton-miles/gallon</td>
</tr>
<tr>
<td>Total fuel use by maritime transportation:</td>
<td>$4.6 \times 10^9$ gallon/year</td>
</tr>
<tr>
<td>Typical fuel price:</td>
<td>1 US$/gallon</td>
</tr>
<tr>
<td>Global yearly fuel reduction:</td>
<td>$2.3 \times 10^9$ gallon/year</td>
</tr>
<tr>
<td>Global yearly cost reduction:</td>
<td>2.3 billion US$/year</td>
</tr>
</tbody>
</table>

1.6 Contents of thesis

For readers without a background in maritime research, Chapter 2 provides the essential background information on model-scale maritime research, thus providing the framework for the rest of the thesis. Chapter 3 continues with a system analysis of the existing pod model set-up. In a more detailed elaboration of the problems with the current set-up, the main design areas are indicated and the requirements for the project result are determined.

The first design area is the propeller drive method, of which the design is described in Chapter 4. Prototypes for two different testing applications were built and tested both in the laboratory and in towing-tank conditions. The accurate measurement of six-component forces requires correct implementation of the underlying theory into calibration routines and data processing. Chapter 5 provides an update of the relevant theory on calibration and application of six-component force sensors. Within the current project, the accuracy and the
dynamic measurement of pod forces has been investigated in detail, with a focus on the pod-unit forces. This led to the redesign of the pod-unit force balance sensor, as described in Chapter 6. The design result is ultimately tested in a towing-tank test, where the old and the new set-up are both built into the same ship model. Chapter 7 reports on these tests. The thesis concludes with an evaluation of the project result (Chapter 8), including a full uncertainty analysis of measurements with the designed pod model. Chapter 8 further summarises the recommendations for future development. Finally, concluding remarks are given in Chapter 9.
2 Model-scale research in maritime technology

Abstract

To predict the performance of ships, model tests are performed. A scaled replica – scale ratio of 1:20 to 1:40 – of the ship is placed in a towing tank and exposed to relevant seakeeping, manoeuvring and powering conditions. The test conditions are prescribed by scaling laws. To predict full-scale quantities from model-scale measurements, scaling laws are used again in combination with appropriate correction methods that take into account the finite validity of scaling laws. With pods, different types of experiment are performed. Each type of experiment focuses on a specific area of ship hydrodynamics.

2.1 Basics

About 50% of the activities at MARIN is related to model-scale tests in large water basins, so-called towing tanks (figures 1.1 and 1.2 in Chapter 1). The concept of model testing is based on the assumption that similarity exists between physical processes (geometry and flow characteristics) on model scale and on full scale. In a towing-tank test, an accurate, geometrically scaled replica of a ship (ship model) is equipped with sensors that measure relevant hydrodynamic quantities like resistance, propulsion forces or fluctuating hull pressures on the aft body of the hull\(^1\). In the towing tank, the ship model is exposed to a range of varying conditions during which the ship’s response is measured.

To investigate the performance and the operational window of the ship design, parameters such as propeller rotation rate, steering angle and wave conditions are varied over a wide range. Sometimes even wind speed and direction are incorporated in the model tests. Also, multiple design alternatives can be compared under identical conditions. Possible design alternatives are: different propeller designs, different orientations of stabiliser fins or different ship after-bodies. So, model-scale measurements are used, either to determine the optimum from different design alternatives in a relative sense or to predict full-scale quantities like propulsion forces, fuel efficiency and manoeuvrability in an absolute sense. Many projects comprise a combination of both types of experiment.

To achieve similarity, the starting point is geometric scaling of the full-scale dimensions of the ship design. The problem of scaling – how to determine experimental conditions that yield similar physical processes on model scale as in full-scale conditions – will be discussed in Section 2.2. How model-scale measurements are used to predict full-scale quantities is described in section 2.3, followed by a description of the most common types of experiment.

---

\(^1\) aft = rear part, hull = ship body
with model-scale podded propellers in Section 2.4.

2.2 Scaling

Geometric similarity can be achieved in a straightforward way through scaling of the full-scale ship dimensions: all dimensions are divided by a scaling factor \( \lambda \) which is defined as:

\[
\lambda = \frac{L_s}{L_m}
\]  

(2.1)

where \( L_s \) is the ship length on full scale and \( L_m \) the ship length on model scale. Scaling factors can range from \( \lambda = 10 \) to \( \lambda = 40 \), depending on the goal of the experiment. The level of detail up to which geometric similarity can be achieved is limited. For instance, as a result of the material and the machining process that are used to produce ship models, the surface roughness cannot be scaled exactly. As a compromise, the ship models are made very smooth and the turbulent boundary layer is artificially enhanced by turbulence stimulatets on the surface of the ship hull.

When further considering the boundary layer of the flow over a surface, the main problem of scaling becomes clear. Whereas geometric properties can straightforwardly be scaled, properties of the tank water like density, viscosity and vapour pressure are approximately the same as on full scale. Obviously, these material properties of the tank water cannot be influenced, except by changing to another fluid. Considering the size of a towing tank (250m × 10.5m × 5.5m) and the costs of replacing its contents, changing to another fluid is not an option. This becomes even more clear when on considers that the scaling factor differs between experiments, which leads to different fluid properties for each experiment. Therefore, other experimental conditions have to be controlled in such a way as to achieve similar physical behaviour as on full scale.

Each group of physical properties requires its own specific scaling law to obtain similarity. A scaling law is generally expressed as a non-dimensional parameter which is the ratio between two physical properties of a fluid. The main scaling laws that play a role at MARIN will be discussed below.

Froude

The Froude number \( F_n \) is named after Froude (1846-1924), one of the first scientists that used model-scale tests in a towing tank to predict full-scale properties of ships. It expresses the ratio between inertia forces and gravitational forces which are proportional to:

\[
\text{inertia forces} \sim \frac{\rho L^3 V}{t}
\]

\[
\text{gravitational forces} \sim \rho L^3 g
\]  

(2.2)

(2.3)

where \( \rho \) [kg/m\(^3\)] is the water mass density, \( g \) [m/s\(^2\)] the gravitational constant, \( L \) [m] a typical length, \( V \) [m/s] a typical speed and \( t \) [s] time. The relation between inertia forces and gravitational forces can then be expressed by the Froude number according to:

\[
F_n^2 = \frac{\rho L^3 V}{t} \cdot \frac{1}{\rho L^3 g} = \frac{V^2}{Lg} \Rightarrow F_n = \frac{V}{\sqrt{gL}}
\]  

(2.4)
2.2 Scaling

When Froude scaling is applied to ships, $V$ is the forward (advance) speed of the ship and $L$ is the ship length. When inertia and gravity play a major role, $F_n$ should be the same at model scale and full scale conditions. This means that $F_n$ has to be used as a scaling law:

$$\frac{V_m}{\sqrt{L_m}} = \frac{V_s}{\sqrt{L_s}} \quad (2.5)$$

where the subscripts $m$ and $s$ denote model and ship respectively. In this way, $F_n$ also determines the ratio between model speed $V_m$ and full-scale speed $V_s$ according to:

$$V_m = V_s \left(\frac{T_m}{T_s}\right)^{\frac{1}{2}} = V_s \lambda^{-1/2} \quad (2.6a)$$

From the scaling relations for speed and length, the scaling rules for forces, moments and the propeller rotation rate can be deduced:

$$F \sim L^3 \frac{L}{t^2} = L^2 V^2 \quad \Rightarrow F_m = F_s \lambda^{-3} \quad (2.6b)$$

$$M \sim L^3 \frac{L}{t^2} L = L^3 V^2 \quad \Rightarrow M_m = M_s \lambda^{-4} \quad (2.6c)$$

$$n \sim \frac{V}{L} \quad \Rightarrow n_m = n_s \lambda^{1/2} \quad (2.6d)$$

The resistance and propulsion properties as well as the manoeuvring and stability characteristics of a ship are dominated by inertia forces, thus requiring Froude scaling. Table 2.1 gives an example of a typical scaling calculation. Although inertia effects dominate, viscous friction also plays a significant role in the resistance of a ship. When considering the propeller only, viscous effects become even more important.

Reynolds

To determine the viscous effects on model scale, Reynolds scaling should be used. The Reynolds number $Re$ expresses the ratio between inertia forces and viscous forces. Viscous forces are proportional to:

$$\text{viscous forces} \sim \nu \rho V L \quad (2.7)$$

where $\nu$ is the kinematic viscosity in $[m^2/s]$. The ratio between inertia forces and viscous forces can now be expressed as:

$$Re = \frac{\rho L^3 V}{\frac{1}{\nu} \rho V L} = \frac{VL}{\nu} \quad (2.8)$$

When applying Froude scaling, we obtain the scaling for the Reynolds number as:

$$(Re)_m = (Re)_s \lambda^{-3/2} \quad (2.9)$$

It is clear, that Froude and Reynolds scaling laws cannot both be satisfied at the same time. Inertia effects are predominant in ship design, while viscous effects can be more easily estimated from flat-plate measurements. Furthermore, Reynolds scaling would require a very exact representation of the surface roughness. Also, Reynolds scaling yields extremely high forward speeds. When, for instance, the values in Table 2.1 are assumed, Reynolds scaling
would lead to a forward speed of 200m/s. These are both very impractical (thus costly) implications of Reynolds scaling.

Therefore, in ship building, Froude scaling is generally applied, which is also the assumption for the rest of the thesis, unless stated otherwise. This automatically means that $R_n$ is too low by a factor 90 to 250 for $\lambda = 20$ and $\lambda = 40$ respectively, which leads to an underestimation of the viscous friction. Hence, measured model-scale forces need to be corrected for this simplification, mostly referred to as scaling effect.

Scaling effects are a serious drawback of model-scale measurements. Experience, as well as a long measurement history, including full-scale verifications, are MARIN’s main assets that make it possible to calculate reliable corrections for these scaling effects.

**Cavitation**

The Froude number and the Reynolds number both concern forces. Cavitation is an aspect of ship design in which pressure is involved. Cavitation is the phenomenon that liquid water changes locally to the vapour phase in regions with very low pressures. These low pressures are caused by high local velocities in the flow. Roughly, it can be stated that cavitation occurs when the local pressure becomes equal or lower than the vapour pressure. Besides the local pressure, other aspects, like the presence of nuclei, play a role in the development of cavitation.

Cavitation causes several adverse effects. Once the cavity (bubble filled with vapour) leaves the low pressure area, it collapses violently (implosion). These implosions cause noise and vibrations in the ship body, which is a problem for passenger vessels as well as for navy ships. When the implosion of the cavity takes place near the surface area of a component, that surface may be damaged by the implosion of the cavity, which is called erosion. Erosion caused by cavitation can have devastating effects on rudders or propeller blades. Finally, large extend of cavitation may cause a decreased propeller thrust, so-called thrust breakdown.

To achieve similarity of cavitation on model scale and to correctly predict the full-scale occurrence of cavitation, the cavitation number $\sigma$ should be maintained the same on model and on full scale. Scaling of cavitation is subject to many scaling effects. So, besides maintaining the cavitation number, other aspects have to be taken into account like similarity of the turbulent boundary layer and surface roughness. These additional measures have no influence on the current design and will therefore not be discussed further. The cavitation number $\sigma$ is defined as:

$$\sigma = \frac{p_0 - p_v}{0.5 \rho V^2}$$

(2.10)

where $p_0$ is the undisturbed pressure in the flow, $p_v$ the vapour pressure and $\rho$ the water density. Equation 2.10 determines the pressure in the test area for a given forward speed. For the current design, this test area is MARIN’s Depressurised Towing Tank, which is a towing tank in which the pressure can be controlled, down to 2.5 kPa (Appendix A). Table 2.1 gives a scaling example of a typical test in the Depressurised Towing Tank.

**Consequences for the current design**

The scaling laws that are used to determine the model-scale testing conditions also determine the boundary conditions as well as the requirements for the design of an innovated pod model. The requirements will be discussed in detail in Section 3.4.
Table 2.1: Typical scaling example of a test in the Depressurised Towing Tank. The model speed and rotation rate are calculated according to Froude scaling; the air pressure in the towing tank is calculated using the cavitation number.

<table>
<thead>
<tr>
<th>Full-scale conditions</th>
<th>Model-scale conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_s$ = 10 m/s</td>
<td>$V_m$ = 2.23 m/s</td>
</tr>
<tr>
<td>$n_s$ = 135 rpm</td>
<td>$n_m$ = 603.7 rpm</td>
</tr>
<tr>
<td>$(p_0)_s$ = 10$^5$ Pa</td>
<td>$(p_0)_m$ = 6.87 * 10$^3$ Pa</td>
</tr>
</tbody>
</table>

Froude scaling determines the required power density of the propeller drive motor. So a propeller drive motor has to fit into the scaled pod dimensions, while at the same time delivering the shaft power required according to Froude scaling. According to Froude scaling, the required motor power scales according to:

$$n_m M_m = n_s M_s \lambda^{7/2}$$

(2.11)

The power density of the propeller drive motor $C_P$ [MW/m$^3$], which equals the available motor power in [MW] divided by the volume of the gondola in [m$^3$], scales according to:

$$(C_P)_m = (C_P)_s \lambda^{-1/2} \quad (Froude)$$

(2.12)

For model scale we thus have decreasing power density requirements. However, electrical motor power density does not scale according to Froude. No straightforward scaling laws for electric motors can be given which are valid for the entire range from 30 MW at full scale to 1kW at model scale. However, based on supplier information of motors in the range of 0.3-10kW, it was found that within a certain electric motor type, the power density approximately scales according to:

$$(C_P)_m = (C_P)_s \lambda^{-1} \quad (Electric\ motors)$$

(2.13)

Hence, the available power seems to decrease faster by a factor $\lambda^{1/2}$ compared the required power according to Froude. For this reason, finding or designing such a motor is a challenging task. The scaling relation for the power density of electric motors (Equation 2.13) is very crude and should be considered as illustrative only. Scaling of electric motor power is complex and it is beyond the scope of this thesis to derive such scaling relationships in detail. A thorough discussion on the scaling of electric motors can be found in [33].

Furthermore, not only the Froude scaled design conditions need to be tested, also a wide range of conditions around the design point are generally investigated. Hence, the propeller drive motor has to fit into many different pod shapes while at the same supplying sufficient power for a wide range of test conditions. This makes the search for a suitable electric motor even more challenging. Section 4.1 further describes how this problem can be solved.

Cavitation scaling determines that the design also has to operate under vacuum conditions down to 2.5 kPa. Not only the vacuum conditions itself, but also the frequent transitions to and from atmospheric pressure make the sealing of the pod model particularly

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$^2$The design conditions of a ship are the conditions, under which the ship is designed to operate, this is generally the point with the optimum combination of speed and fuel efficiency.
challenging. Further, many electronic components cannot resist vacuum conditions and cooling of the electronics under vacuum conditions is less efficient than under atmospheric pressure. Hence cavitation scaling also has strong implications for the pod model design, including the drive electronics.

2.3 Model-scale measuring and full-scale prediction

Once model-scale measurements have been performed, scaling laws are used again to predict the corresponding full-scale quantities. Also, non-dimensional parameters may be calculated to generalise the measured data and to make it scale-independent. As a result of scaling effects, the scaled-up data has to be corrected.

The corrections for scale effects generally add considerable uncertainty to the final result. Also, the repeatability of a towing-tank test is limited. However, the prediction accuracy demanded by customers is high; generally an uncertainty of about 1-2% for absolute full-scale predictions is required. For instance, a few percent change in fuel efficiency can determine whether a ship can operate profitably or not. Furthermore, in research projects where absolute full-scale values are important, small deviations on model scale can be magnified – through scaling – to large full-scale errors.

Therefore, to achieve reliable predictions of full-scale quantities, the towing tank conditions need to be controlled carefully and the basis of the predictions – the measurements – should be very reliable. Required measurement uncertainties of the sensors are typically on the order of 0.5-1%. However, when different alternatives are compared in an optimisation, an even lower uncertainty is sometimes demanded. Required measurement uncertainties are specified further in Section 3.4.

2.4 Tests with podded propellers at MARIN

The majority of pod experiments can be grouped into several standardised tests. A qualitative description of these standard tests is given below. The numerical elaboration of the specifications is given in Section 3.4. The different tests are grouped by the towing tank in which they are carried out. For a description of the towing tanks, see Appendix A, for more information on towing-tank experiments see also [32, 43].

Deep Water Towing tank

All standard tests in the deep water towing tank (DT) consist of a series of steady-state runs, during which all conditions (forward speed, propeller rotation rate and steering angle) are kept constant. The ship model is always towed or guided by the carriage (Figure 2.1). Scale factors typically are in the range $15 \leq \lambda \leq 25$.

Open water test Only the pod model, without being mounted in a ship model (Figure 2.1), is towed through the towing tank at a constant forward speed and a constant propeller rotation rate. The rotation rate is equal for all runs, while the forward speed is increased every run, from zero until the point where the produced propeller torque becomes negative. To reduce the scaling effects, rotation rates are generally higher than in the Froude scaled design conditions.
2.4 Tests with Podded Propellers at Marin

**Bollard Pull** At zero speed, the rotation rate is increased each run, from zero until some maximum, generally determined by the motor that drives the propeller, or the limits of the used sensors. Bollard pull tests can be done with just the pod or with the pod mounted in the ship model.

**Propulsion test** The ship including the pod models is towed through the basin at different speeds and at rotation rates around its design conditions.

### Seakeeping and Manoeuvring Basin

In the seakeeping and manoeuvring basin (SMB), two operation modes are distinguished: free-sailing and captive. During the free-sailing mode, the ship model propels itself and can move freely. The model is only connected to the towing tank carriage by the power and sensor cables. The carriage follows the ship model through the use of an optical target. Scale factors for free-sailing test typically are in the range $25 \leq \lambda \leq 40$. During captive tests, the carriage guides the ship model and the ship model is generally connected to the carriage by a force sensor, where in many cases the same ship model is used as in propulsion tests.

**Free-sailing manoeuvring test** The ship model follows a prescribed steering angle pattern, under different wave conditions, during which the ship’s response is measured. The propeller rotation rate is either kept constant or controlled by a computer model that represents the full-scale behaviour of the ship’s engine.

**Captive manoeuvring test** Steering forces are measured under different sailing and steering conditions, while the ship model is attached to the towing tank carriage.

**Free-sailing seakeeping test** The ship model follows a straight course under different wave conditions, during which the ship’s response is measured. During seakeeping tests, the propeller rotation rate is either kept constant or controlled by a computer model of the ship’s engine.

### Depressurised Towing tank

The main goal of the depressurised towing tank (or Vacuum Tank, VT) is to facilitate investigations related to cavitation. When under atmospheric pressure, the VT can also be used for the same experiments as the Deep water towing Tank (DT). Tests with pods are done with ship models as well in the open water set-up. The forward speed and propeller rotation rate are generally similar to DT conditions. However, to approach cavitation similarity, it may be necessary to use propeller rotation rates higher than according to Froude scaling. Scale factors typically are in the range $15 \leq \lambda \leq 25$.

There are many ways in which cavitation can be observed, which will not be discussed here. All these observation methods make use of photographic or video images. Recently, also high-speed video has been introduced as an observation technique. The synchronisation of high-speed video images with pressure and force measurements requires very accurate measurement and control of the propeller rotation rate. Besides the visual observation of cavitation, the vibration effects can also be measured. This is achieved through the measurement of pressure fluctuations at the ship hull close to the propeller. These pressure fluctuations generally have low amplitudes and are prone to disturbance by non-hydrodynamic vibrations of the ship body.
Figure 2.1: Schematic overview of the pod model mounted in the open water set-up (a) and in a ship model (b).
2.4 Tests with podded propellers at MARIN

Unsteady tests

A rather new research area is the area of dynamic or unsteady model-scale testing of pods. During unsteady or dynamic tests, not only the average value over a test run is important, but also the behaviour of hydrodynamic quantities during transient conditions. Unsteady testing can refer to dynamic measurements under steady-state conditions. For instance, measuring the variation of the propeller torque as a function of time under constant rotation rate and forward speed, measuring broadband cavitation noise or measuring blade passage effects. Unsteady testing may further comprise dynamic measurements during varying conditions. For instance, measuring the variation of pod-unit bearing loads during manoeuvring at high speed or finding the peak loads during a crash-stop manoeuvre.

Until now, with the existing pod models, most standard tests with pods do not comprise dynamic measurements; no time traces or spectral analyses are part of the end result. This means that for the majority of the experiments, only the averaged data needs to be correct. Exception to this are pressure fluctuation measurements in the VT, which are analysed and presented in the spectral domain. Occasionally, unsteady shaft forces and moments are required.

For pods, it is expected that the dynamic loads on bearings and seals play an important role in the operational problems and breakdowns that have been observed recently (Appendix B). Model-scale dynamic measurement of pod forces during manoeuvring is assumed to reveal crucial information on the design of pod bearings and seals. Therefore, this type of measurements will be required more and more in the near future for projects dedicated to investigating these phenomena.
3 Solving pod model problems

Abstract

The existing way to build pod models has several drawbacks which limit modelling flexibility as well as measurement accuracy. However, until 2001, none of the maritime test institutes worldwide had a solution other than using an external propeller drive. Crucial for a major step forward in model-scale testing of pods, and indirectly also for full-scale pod developments, is to place the propeller drive motor inside the pod gondola. In fact similar as is done on full scale. Innovating the drive method this way creates new opportunities to further improve pod model measurements.

Highly accurate measurement of the full 3D force and moment vector between pod-unit and ship body is crucial for reliable model-scale experimental results on which the full-scale predictions are based. Such measurements require a profound theoretical understanding to correctly calibrate sensors and process measured data. An update of the underlying theory and a redesign of the pod-unit balance are major steps towards reliable pod-unit force measurements that meet the modern day accuracy demands.

In the current chapter, the pod model problems are elaborated further, the requirements for the project result are determined and the main design areas are indicated.

3.1 Current pod model set-up

To build model-scale versions of many different pod geometries at an acceptable cost level, the pod model should be highly flexible. At MARIN, this flexibility is realised through the use of standardised units. Surrounding this standardised drive unit, a shell is placed, which on the outside is an exact model-scale replica of the full-scale pod to be tested (Figure 3.1). This shell is made of a material which is cheap and easy to manufacture, generally polyurethane or PVC. The drive unit is re-used many times, while the pod shape is machined every time a new type of pod is tested. Several sizes of the mechanical unit are available for different scales.

Currently, MARIN, just as all other maritime test facilities worldwide, builds pod models with the (electric) drive motor placed external to the pod. A mechanical drive train transmits the power to the propeller shaft. The drive train comprises a flexible shaft coupling and two right-angle gears as shown in Figure 3.2. The standard unit contains the right-angle gears as well as the shaft sensor that measures propeller thrust and torque. The position of the motor inside the ship body depends on the specific configuration of the ship model.

The fixed unit is coupled to the electric motor by a flexible coupling. When pod-unit measurements are needed, the complete set-up is placed upon the measuring plate of a pod-unit six-component (6C) balance that measures the forces between the pod and the ship body (Figure 3.2). In cases where active steering, in combination with pod-unit measurements is
3. SOLVING POD MODEL PROBLEMS

Figure 3.1: Picture of the existing method to build pod models. The standard mechanical drive unit contained in the polyurethane shell is made visible.

required, a steering servo should also be placed on the measuring plate of the balance. This is a rather complicated construction which compromises the pod-unit force measurements. Therefore, active steering in combination with six-component pod-unit load measurements has rarely been applied until now.

3.2 Pod model problems

The current way of modelling pod propulsors has several inherent drawbacks:

A - Limited flexibility
The right-angle gears contained in the fixed units determine the orientation of the steering shaft and the propeller shaft with respect to each other to be $90^\circ$. The actual angle at full scale is rarely $90^\circ$. Further, in many cases, the fixed angles cause the rotation point and the axis of the steering shaft to be dislocated. Also, for innovative pod designs like ABB’s compact azipod, the vertical shaft may even stick out of the pod body. Finally, the length of the propeller shaft has to be adjusted to the specific pod shape. These geometric aspects are illustrated in Figure 3.3.

B - Vibrations
It is inherent to gear transmissions that they cause vibrations. These vibrations can only be minimised through careful fine-tuning of the construction. However, when preparing ship models with pods, the gears often need to be re-assembled for small modifications. Thus, fine-tuning of the gears is required each time a pod model is built into a ship model. In practice, such repeated fine-tuning is impractical, thus too expensive.

Furthermore, the helical gears used are designed for a single direction of rotation only, in which they run smoothly. In practice this preferred direction of rotation is not taken into account. Both aspects of incorrect usage lead to increased vibrations of the pod model. These vibrations appear as noise in measured quantities like force, torque, pressure and acceleration. In Figure 3.4, this is illustrated by an example of noise in the measured propulsion force of the
3.2 Pod Model Problems

Figure 3.2: Schematic view of the traditional way of modelling podded propulsors, with the main parts indicated. See Chapter 6 and [47], for a description of the operating principle of the pod-unit force balance.
3. SOLVING POD MODEL PROBLEMS

Figure 3.3: Geometric limitation of the current pod models, illustrated for an ABB compact azipod (Figure 1.14). The dislocation of the rotation point and the axis of the steering shaft is illustrated. Also, the locations are indicated where the strut becomes too thin for the vertical shaft, as a result of its streamlined shape. Further, it is shown that the length of the propeller shaft has to be adjusted for each type of pod.

pod-unit. The frequencies of this noise generally lie in the same range as (and often coincide with) the hydrodynamic frequencies of interest. This makes it hard to distinguish between propeller drive artifacts and hydrodynamic phenomena.

C - Limited accuracy

For correct measurement of pod-unit forces, there should be no rigid connection between the measuring plate of the balance and the fixed world (ship body). Therefore, the propeller drive motor has to be integrated with the six-component (6C) pod-unit balance. As a result, the drive train is involved in transferring the pod forces to the measurement plate. The drive train contains rotating parts and several shaft couplings, which increases the risk on non-reproducible and non-linear behaviour of the force balance, which cannot be taken into account through regular calibration procedures.

As yet, the pod-unit force balance is typically calibrated without the drive train mounted. Once the drive train is mounted, it will introduce tension in the frame, which is not taken into account in the calibration, which leads to measurement (bias) errors. This has also been shown experimentally and is reported in [53]. The conclusion was that it is necessary to calibrate the frame with an identical construction as used during the measurements, i.e. with the motor and the drive train mounted on the frame.
D - Limited steering
If besides the drive motor, the steering servo motor also has to be mounted on top of the measuring plate, the construction as a whole becomes large and unbalanced. Not only will the construction be complicated and impractical, it also introduces additional tension in the balance, which in most cases leads to measurement errors. However, the maritime market does currently ask for simultaneous steering and six-component pod-unit force measurements. Hence, a solution needs to be found where steering can be combined with highly accurate pod-unit force measurements.
Figure 3.4: Example of noise in the measured steady-state propulsion force of a pod-unit, measured with the existing set-up with external motor and gear transmissions. Shown are measured time series and their corresponding Fourier amplitude spectrum. Indicated in the spectrum is a significance level which equals the single-point uncertainty limit of the signal conditioners. The upper plot shows a typical bollard pull run: zero speed and constant rotation rate. The lower figure shows a typical open water run: constant forward speed and rotation rate. The upper limit of the vertical axis of the time series equals the maximum sensor output. The upper plot gives an impression of the noise caused by the propeller drive only. The lower graph illustrates the noise caused by the propeller drive together with the vibrations of the moving carriage. It is clear that with such noise levels, it is hard to determine any dynamic property from the signal in the range 0-100 Hz.
Besides these inherent drawbacks of the drive method, the following problems with the current pod model exist that limit the measurement and control accuracy:

I - 6C measurement routines
Just after strain gauges became commercially available, MARIN has performed a great deal of research on designing force transducers. Several high-quality sensor designs have been developed by MARIN engineers [21, 65], amongst which the propeller-shaft sensor and the highly accurate single-component (1C) sensor, which are both used in the current design (Chapter 4 and 6). However, the internal documentation on this topic ceases after about 1980. From this moment on, the internal documentation on force measurement becomes limited and scattered. Recently, as a result of improved signal processing techniques and higher accuracy demands by customers, force measurement has got renewed attention. However, until the current project, little effort has been put into six-component force measurements. As a result, MARIN data processing routines need to be updated and the measurement uncertainty of such sensors is not exactly known, as has been reported in [53].

II - Calibration
The current approach to calibrating six-component force balances introduces many uncertainties since the calibration routine is rather basic as was shown in [53] and [48]. The simplifications in the calibration may have been introduced in the past to save computing time in the data processing. Further, the accuracy demands have increased over time. The accuracies required nowadays lead to a necessary review of these routines (Chapter 5), to facilitate high quality measurements with the innovated pod model.

III - Integration of steering servo
Ideally, the steering servo of the pod model should be integrated with the pod-unit balance. In this way, the influence of the steering servo can be taken into account in the calibration, and the unit is easy to place in the ship-model. The steering servo motor designs currently available at MARIN are not suited for such integration [60].

IV - Control steering servo
The control accuracy of the currently existing steering servos is not as accurate as required [60].

The problems described above should all be solved to be able to improve the reliability and efficiency of model-scale testing and to achieve the modern day customer demands. The key to solving these problems will be given in the Section 3.3.

3.3 Key to solving pod model problems
The heart of the problem is the propeller drive method. First, the fundamental difference between full scale and model scale inevitably leads to geometrical problems. Further, the choice for a gear transmission inherently implies vibration noise. Also, the required flexibility demands a modular approach. With a gear transmission, this modularity implies frequent re-assembly of the vital parts of the drive train making the current concept error-prone and sub-optimal by definition. The approach that solves the core of the problem is to drive the propeller the same way as on full scale. Major improvements can only be expected when the
drive motor is located inside the gondola [43]. A motor inside the gondola means:

- Elimination of the vertical shaft that goes through the strut of the pod.
- No vibration from the right-angle bevel gears.
- The opportunity to design a fixed unit containing all the vital parts, which does not need to be re-assembled in between tests.
- No propeller drive motor on top of the pod-unit force balance, which yields:
  - a less complicated construction of the force balance
  - the opportunity for an integrated steering servo
  - less disturbance of the pod-unit force measurements
- The opportunity to combine 6C pod-unit measurements with active steering

Therefore, the result of the current project is defined as: an innovated pod model with its propeller drive motor placed inside the pod gondola. This novel approach will be a major breakthrough in model-scale testing of pods. A further description of the envisaged result, including detailed specifications is given in Section 3.4.

### 3.4 Project result: An innovated pod model

The result (deliverable) of the current project is defined by MARIN – the customer of the current project – as:

> A widely applicable, innovative concept to build pod models with the propeller drive motor located inside the gondola [40].

The pod model has to suit the demands of modern-day model-scale research on pods. Also, it should fit into MARIN’s production process of model-scale testing. The requirements for the design are elaborated in further detail in the subsections below.

The requirements given below have been determined in discussion with MARIN. Not only the commercial consulting departments that are responsible for the given advice and marketing of towing-tank tests were involved in formulating the requirements. Also, the operational departments that perform the testing have been consulted. The requirements are discussed in detail in the reports [40], [41] and [43]. The following subsection gives an overview of the essential steps.

#### Requirements

In the following, the requirements are discussed in more detail. First, the general features are discussed. After that, the required power density is given, followed by the requirements for controls and measurements. Finally, a discussion on the costs is given. The specifications vary between different types of experiment. In the following, the description of the requirements is coupled to the experiment types described in Section 2.4. The experiments can be grouped into three main categories, which together cover over 80% of the model-scale experiments with pods. These categories are presented in Table 3.1.
### 3.4 Project Result: An Innovated Pod Model

#### Table 3.1: The three categories of experiment that together cover over 80% of the model-scale experiments with pods.

<table>
<thead>
<tr>
<th>Category</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Category 1</td>
<td>Small-scale, free-sailing manoeuvring and seakeeping tests that require low torque and no shaft measurements.</td>
</tr>
<tr>
<td>Category 2</td>
<td>Large-scale, propulsion and open-water tests as well as captive manoeuvring tests. These tests require medium torque as well as measurements at the shaft.</td>
</tr>
<tr>
<td>Category 3</td>
<td>Large-scale cavitation observations, which require a high torque in combination with shaft measurements.</td>
</tr>
</tbody>
</table>

#### General features

First, the pod model should suit the demands of modern day model-scale testing. For the new design this means that:

- new innovative pod shapes like ABB's compact azipod may no longer cause geometric or constructive problems.

- it should be possible to combine reliable pod-unit force measurements with active steering.

- the full six-component load should be measured at the propeller shaft, instead of the propeller thrust and torque currently measured.

- Dynamic measurements up to the first blade harmonic\(^1\) have to be possible for the pod-unit forces as well as the propeller-shaft forces.

Further, the innovated pod model has to fit into the production process of MARIN, which means that:

- The design should not disturb any of the other measurements performed in the same ship model.

- The direct drive should be a fixed unit, around which a wide variety of pod shapes can be built.

- The measurement and control interfaces should fit into the MARIN infrastructure.

- The designed concept should be suited for the production-oriented environment of the towing tanks, which requires the design to be user-friendly in terms of MARIN operational procedures.

#### Power and geometry

The required power at the propeller shaft is given in Table 3.2. The gondola dimensions indicated are valid for a certain range of scales. A measure for the scale is the propeller

\[^1\text{Blade harmonics, or blade passage frequencies, are multiples of the product of the propeller-shaft frequency and the number of propeller blades. For example, a typical rotation rate is 600\text{rpm} and a propeller has typically 4 blades, which makes the first blade harmonic }\frac{600\text{rpm}}{\text{60s/min}} \times 4\text{blades} = 40\text{Hz.}\]
diameter which is also given for comparison. Dimensions other than those presented in Table 3.2 may occur, requiring different shaft power. However, the values indicated are the most commonly occurring combinations of required shaft power and gondola dimensions.

Table 3.2: Power and geometry requirements for the different experiment categories. For each type of experiment; a range of typical propeller diameter is also given.

<table>
<thead>
<tr>
<th>Experiment type</th>
<th>Typical prop. diameters $D$ [mm]</th>
<th>Gondola length $L$ [mm]</th>
<th>Gondola Cross section $O$ [mm]</th>
<th>Rotation rate $n$ [rpm]</th>
<th>Torque $M$ [Nm]</th>
<th>Power Density $C_P$ [MW/m$^3$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Category 1</td>
<td>100-150</td>
<td>140</td>
<td>45</td>
<td>1500</td>
<td>2</td>
<td>1.4</td>
</tr>
<tr>
<td>Category 2</td>
<td>230-250</td>
<td>200</td>
<td>85</td>
<td>1200</td>
<td>10</td>
<td>1.1</td>
</tr>
<tr>
<td>Category 3</td>
<td>230-250</td>
<td>250</td>
<td>100</td>
<td>1200</td>
<td>25</td>
<td>1.6</td>
</tr>
</tbody>
</table>

The required power density is larger than what can be expected according to Froude scaling. This is caused by the fact that many different pods with different power distributions need to be covered with a single device. This motor should thus fit into a wide variety of pod shapes, which yields a smaller available space for the motor than on full scale. Further, many experiments like manoeuvring and open water tests require off-design test conditions which require a much higher torque and rotation rate than on full scale. As a result, the required motor power density $C_P$ [MW/m$^3$] (Section 2.2) on model scale ($1:1$ to $1:6$ MW/m$^3$) is about 1.5-2 times larger than on full scale ($0.7$ to $1.0$ MW/m$^3$).

Also, the scaling considerations in Section 2.2 emphasise that it is challenging to achieve sufficient power with an electric motor. Hence, the model-scale requirements cannot be fulfilled with a straightforward electric solution. Therefore, to achieve a direct propeller drive solutions on model scale, all possible optimisation measures need be taken to enhance the motor power. Further, it should investigated to what extent alternative working principles like hydraulics can provide a solution.

Controls

The two quantities to be controlled are the propeller rotation rate $n$ and the pod steering angle $\alpha$. For the steering angle, the difference between the set-point and the true angle has to be limited to $\Delta \alpha \leq 0.1^\circ$. In practice, the maximum angular acceleration has to be limited to typically $2000$ $^\circ$/s$^2$ [60] to protect the other sensors in the ship model. As a result, the typical required reaction time of $22$ ms for a step change of $1^\circ$ is well within the limits of modern-day motion control technology. Therefore, no reaction time for the control of the steering angle has been determined.

For the propeller rotation rate, the requirements vary between types of experiment. A distinction has been made between the average rotation rate over a steady state run and the instantaneous rotation rate. Table 3.3 presents the specifications for the rotation rate control. The instantaneous control accuracy requirements are particularly challenging as electric motor always exhibit so-called torque and speed ripple. These are oscillations of torque and rotation rate oscillations on top of the average value. Although these oscillations are very small, they yield significant deviations of the rotation rate. Torque and speed ripple are hard to control as the frequencies of oscillation are typically beyond the control bandwidth of the servo system.
Table 3.3: Required control accuracy for the propeller rotation rate \( n \), for a typical average rotation rate of \( n = 600 \text{rpm} \).

<table>
<thead>
<tr>
<th>Experiment type</th>
<th>Rotation rate control accuracy ( \Delta n ) [rpm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Category 1</td>
<td>Average: 1 Instantaneous: 10</td>
</tr>
<tr>
<td>Category 2</td>
<td>Average: 0.1 Instantaneous: 3</td>
</tr>
<tr>
<td>Category 3</td>
<td>Average: 0.1 Instantaneous: 3</td>
</tr>
</tbody>
</table>

Measurements

The required types of measurement, including the required measurement uncertainty and the dynamic range (bandwidth) are given in Table 3.4 for each type of experiment. In Figure 3.5, the different quantities to be measured are illustrated. \( F[N] \) stands for a force, \( M[Nm] \) stands for a moment. The six-component forces and moments of the propeller shaft and the pod-unit can also be grouped into vectors according to:

\[
\vec{F}_{\text{prop}} = [F_x, F_y, F_z, M_x, M_y, M_z]^T \quad (3.1a)
\]

\[
\vec{F}_{\text{unit}} = [F_x, F_y, F_z, M_x, M_y, M_z]^T \quad (3.1b)
\]

Not all of the components of these vectors need to be measured in every experiment. Therefore, \( \vec{F}_{\text{prop}} \) and \( \vec{F}_{\text{unit}} \) may also contain a selection of the components given in equations 3.1a and 3.1b.

The required measurement frequency is the effective measured frequency (Section 6.4) up to which the data is undisturbed by filtering of the data acquisition system. The dynamic range of the measurements should extend up to the first blade harmonic. Typically, for 600 rpm and a four-blade propeller, this yields \( \frac{600 \text{rpm}}{60 \text{s/min}} \times 4 \text{blades} = 40 \text{Hz} \). A worst-case scenario for the first blade harmonic is a rotation rate of 1200 rpm and six propeller blades, which yields a dynamic range of \( \frac{1200 \text{rpm}}{60 \text{s/min}} \times 6 \text{blades} = 120 \text{Hz} \). For dynamic measurements, the measurement uncertainty is allowed to increase with frequency. As yet, no clear specifications for this decrease with frequency exist. In Section 6.4 the specifications for dynamic measurements are discussed further.

Indicated in Table 3.4 is that for Category 2 and 3, the full six components need to be measured at the propeller shaft. Currently, just \( F_x \) and \( M_x \) are sufficient for the majority of the standard experiments. However, since bearing forces appear to determine to a large extent maintenance intervals, the six-component loads at the propeller shaft are important forces to measure. Therefore, to prepare for future research questions, this has been set as an optional demand for the pod model.

Costs

The initial as well as the operational costs of the pod model unit should be acceptable. Whether costs are acceptable depends on the lifetime and applicability of the unit. A typical project budget for a towing tank is 50-100 k€. Assuming a lifetime of at least 20 towing-tank tests, an initial costs limit of about 10-15 k€ and operational costs of about €250-500 per project can be considered as reasonable guidelines.
It was agreed with MARIN, that the main focus of the project is on technological feasibility, rather than on costs [40]. Once a technological solution has been found, there are many ways to reduce the costs or to spread the costs of the pod model over multiple projects. Further, a strongly enhanced quality of pod model testing through a new design creates the opportunity of a higher selling price of the test, thus allowing higher costs.

3.5 Roadmap towards an innovated pod model

The first step in the design process towards an innovated pod model is a system analysis, as presented below. This system analysis indicates the areas that are most crucial for a successful project result. For each of these design areas, a general outline of the most promising solutions is given. In Figure 3.6, a schematic drawing is given of the pod model system. Indicated are the main parts as well as the system boundaries. As indicated, the system contains all components involved in driving the propeller, steering the pod and measuring the pod forces.

Exclusions

The following aspects are not included in the current project, thus forming the boundary conditions of the system.
Table 3.4: Requirements for measurements on the pod model grouped by type of experiment. The given uncertainties are required for steady state tests. In general, for dynamic measurements, the uncertainty is allowed to increase with frequency. For the dynamic range of Category 2 & 3 experiments, a typical value of the first blade harmonic is given. Between brackets, a worst-case first blade frequency is given.

<table>
<thead>
<tr>
<th>Category 1</th>
<th>Quantity</th>
<th>Measurement uncertainty (±)</th>
<th>Dynamic range</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>[0 – 10] %FSO [10 – 100] %FSO</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>$M_{x\text{-prop}}$</td>
<td>0.1Nm</td>
<td>1%</td>
<td>10Hz</td>
</tr>
<tr>
<td></td>
<td>$F_{x\text{-unit}}, F_{y\text{-unit}}$</td>
<td>1N</td>
<td>1%</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$M_{z\text{-unit}}$</td>
<td>0.1Nm</td>
<td>1%</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$\alpha$</td>
<td>0.1°</td>
<td>n.d. **)</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$n$</td>
<td>0.1rpm</td>
<td>0.1rpm</td>
<td>&quot;</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Category 2 &amp; 3</th>
<th>Quantity</th>
<th>Measurement uncertainty (±)</th>
<th>Dynamic range</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>[0 – 10] %FSO [10 – 100] %FSO</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>$F_{x\text{-prop}}$</td>
<td>1N</td>
<td>0.5%</td>
<td>40(120)Hz</td>
</tr>
<tr>
<td></td>
<td>$F_{y\text{-prop}}, F_{z\text{-prop}}$</td>
<td>1N</td>
<td>1%</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$M_{x\text{-prop}}$</td>
<td>0.1Nm</td>
<td>0.5%</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$M_{y\text{-prop}}, M_{z\text{-prop}}$</td>
<td>0.1Nm</td>
<td>1%</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$F_{x\text{-unit}}$</td>
<td>1N</td>
<td>0.5%</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$F_{y\text{-unit}}$</td>
<td>1N</td>
<td>1%</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$F_{z\text{-unit}}$</td>
<td>5N</td>
<td>5%</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$M_{x\text{-unit}} \ldots M_{z\text{-unit}}$</td>
<td>0.1Nm</td>
<td>1%</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$\alpha$</td>
<td>0.1°</td>
<td>n.d. **)</td>
<td>&quot;</td>
</tr>
<tr>
<td></td>
<td>$n$</td>
<td>0.1rpm</td>
<td>0.1rpm</td>
<td>&quot;</td>
</tr>
</tbody>
</table>

* %FSO = Percentage of the full-scale sensor Output; It is used to indicate the sensor range for which a certain specification is valid.
** n.d. = not defined

Control infrastructure The design should match the standardised interfaces at MARIN. These are defined by BSS\(^2\), which is the control infrastructure used at MARIN to control all testing variables involved in a towing-tank test. BSS contains a standard interface unit which matches most industrial motion control standards. The communication with these units goes through a CAN fieldbus system. Therefore, the standard interface unit is referred to as: CAN-module.

Towing-tank carriage The towing-tank carriage, including its connections, sub-frames and controls cannot be influenced. Therefore, any influence of the carriage on the pod model

\(^2\)BSS stands for Basis Stuur Systeem, which is Dutch for: Basic Control System
system should be considered as a boundary conditions to be dealt with, rather than a problem to be solved.

**External error sources** Not all sources of error in pod model tests can be controlled. Error sources outside the scope of the project are referred to as external error sources. For example, vibrations of the carriage may disturb force measurements. These vibrations itself cannot be eliminated. However, its adverse effects on the quality of the measured data should be minimised.

**Signal conditioning and data recording** All signal conditioning and data recording is done within a standardised system, the MARIN Measurement System (MMS). The pod model should fit into MARIN’s production process of model-scale testing. As a result, all sensors have to be conditioned by MMS. The quality of the signal conditioning stage and the data recording thus are a boundary conditions of the pod model system.

**Design space**

The pod model system can be subdivided into design areas according to the different functions of the pod model. For each of the design areas, it can be determined how it relates to the problems indicated in Section 3.2. The functions of the pod model are:

- Representing the full-scale pod geometry on model scale.
- Driving the propeller and measuring rotation rate.
- Measuring pod-unit forces.
- Measuring propeller-shaft forces.
- Steering the pod and measuring steering angle.

The relationship of each of these functions with the pod model problems is discussed in the following subsection. Also, the design opportunities and the most promising approach towards a solution is given.

**Options for design**

**Representing the full-scale pod geometry on model scale**

With the excellent machining equipment available at MARIN, the scaled geometry of pod and propeller can be realised within linear tolerances of $\pm 10 \mu m$. So, the limiting factor in representing the full-scale geometry is the construction that has to be built inside the pod, rather than the machining process. Hence, to correctly perform this function, the geometric problems summarised in Figure 3.3 should be solved. The best way to avoid these problems is to design a pod model concept with the motor inside the gondola. The machining process of the outside pod housing should be kept the same.

---

3MMS stands for Marin Measurement System, which indicates the whole of signal conditioning and data recording stages at MARIN.
3.5 Roadmap towards an innovated pod model

Figure 3.6: System analysis of the pod model. The thick dashed line indicates the system boundaries. MMS stands for Marin Measurement System, which indicates the whole of signal conditioning and data recording stages at MARIN. BSS stands for Basis Stuur Systeem (Dutch for: Basic Control System) which indicates the whole control infrastructure at MARIN.

Driving the propeller and measuring rotation rate

Core of the problems shown in Section 3.2, is the propeller drive method. Designing a drive method with the motor located inside the pod gondola will solve many of the problems. Further, this approach creates the opportunity for additional improvements. Apart from allowing for a correct model-scale pod geometry, the drive motor has to deliver the required power as specified in Section 3.4. Several challenges arise when designing the drive motor.

- The scale of the pod determines both the required power and the available space. Thus, in most cases, Froude scaling determines the required power density. The available motor power and the distribution of the power over torque and rotation rate do not necessarily scale according to Froude. This may lead to model-scale power requirements that are difficult to meet.

- At full scale, the drive motor is highly optimised for that specific geometry and power distribution required by the propeller. At model scale, the motor has to fit inside a wide range of pod geometries. The motor has to drive many propellers with very different power requirements. As a result, the available space for the motor is a much smaller
fraction of the pod volume than is the case in a full-scale pod, where pod geometry and drive motor are well matched in the design phase.

- Another challenge is that the propeller drive motor has to be commercially available for an acceptable price. Whether the price of a motor is reasonable, depends on many factors like quality, required in-house engineering, application area, necessary auxiliary equipment and delivery times. A reasonable cost limit for the complete drive motor with an integrated shaft sensor is 10k€.

- In general, many different experiments are done with the same ship model. As a result, the pod model needs to be suited for all these experiments. Thus, the available power should be sufficient for the most demanding experiment that can be expected.

For the propeller powering there are two options available: an electric drive or a hydraulic drive. Apart from choosing the motor type, a choice has to be made whether a planetary reduction gear can be used at the propeller shaft, to reduce speed and increase available torque. The pros and cons for each of the options and the result of this design area will be elaborated further in Chapter 4.

### Measuring pod-unit forces

The most crucial measured parameter on a pod model, is the total force the pod-unit exerts on the ship body, the so-called pod-unit force $F_{\text{unit}}$. The pod-unit force can be measured in two ways. First, by a sensor that is integrated with the steering shaft of the pod. The second way to measure pod-unit forces is by an external sensor, placed inside the ship body, like it is done in the existing approach.

The latter way, using an external sensor that consists of two plates connected by six single component force transducers, is a widely applicable concept. Within MARIN, most six-component force sensors are constructed this way. It is a very flexible concept, since the sensor does not need to be fully integrated with the construction to be tested. Further, the single component force transducers can be replaced easily, to adjust the load range and to achieve the required sensitivity.

Because of the versatility of the concept together with the wide application of similar measurement frames within MARIN, it was decided to follow the existing concept to measure pod-unit forces. However, the sensor will be redesigned and a thorough review of the theory on calibration and data processing of six-component force sensors will be carried out. Regardless of the sensor that is used, six-component force measurements should be carried out based on the correct underlying theory. First the sensor should be calibrated correctly and also, the data processing should be carried out correctly. The underlying theory on calibration and data processing will be reviewed in Chapter 5. The redesign of the pod-unit balance will be reported in Chapter 6.

### Measuring propeller-shaft forces

For the experiments in Category 2 and 3, propeller-shaft measurements are also crucial. For the majority of the current experiments, just $F_{x,\text{prop}}$ and $M_{x,\text{prop}}$ are required. However, for the current design, six-component force measurements at the propeller shaft are an optional demand, to prepare for future research.
Apart from the vibration noise introduced by the current drive train, no serious problems have surfaced with the existing shaft sensor. The existing shaft sensor measures $F_{x\text{-prop}}$ and $M_{x\text{-prop}}$ only. Therefore, as long as only these components are needed, the existing shaft sensor can be used. The existing sensor should then be integrated in the drive motor. For six-component force measurements at the propeller shaft, no existing sensor design is available. For these experiments a new sensor needs to be designed.

Directly related to the propeller-shaft sensor is the transfer of power and data, to and from the fixed world to the rotating shaft. For a small number of components, this can be done through slip rings, as is done in the current set-up. For the data transfer of a six-component sensor, many channels are required, for which a slip ring unit may become too large. Also, using slip rings implies remote signal conditioning and long wires, which makes this method prone to EMI (see also Chapter 5). So, slip rings are less than ideal and the data transfer should be redesigned done when a new propeller-shaft sensor is designed.

Six-component force measurements at the shaft are an optional demand. As a result of limited time and budget, no complete design for a six-component shaft sensor has been included in this work. The existing shaft sensor that measures $F_{x\text{-prop}}$ and $M_{x\text{-prop}}$ will be integrated in the design. For a six-component shaft sensor, only the recommended approach towards a solution will be discussed in Section 6.5.

**Steering the pod and measuring the steering angle**

Ideally, the steering servo should be integrated with the new pod-unit balance. To allow for such integration, the design of the design of the steering servo should be carried out in the final stage, when the pod-unit balance design is finished. Further, the requirements for control speed and accuracy, as well as available power can be met by commercially available parts. Hence, in the current project, only the predesign stage has been carried out by Van der Meulen [60]. The result of this study is summarised in Section 6.4.
4 Driving the propeller

Abstract

Providing sufficient shaft power while maintaining the correct pod geometry is the main challenge in designing a direct propeller drive method. From the market research in the predesign phase of the project, it could be concluded that no generic solution exists. Therefore, a toolbox has been defined to provide solutions for the three main categories of experiments, which together cover over 80% of all model-scale experiments with pods. Three design paths lead to three different tools for the main types of experiment. For two of these tools, prototypes have been built and tested successfully: a small-scale low-tech electric solution and a large-scale high-tech electric solution. After the realisation of a propeller drive motor inside the pod gondola, the additional challenge is to facilitate dynamic measurements. For the large-scale high-tech prototype, it has been investigated to what extent dynamic measurements are disturbed by the propeller drive motor.

4.1 Pod modelling toolbox

Locating the driving motor inside the pod gondola implies that the propeller shaft is driven directly, without any transmission in between. Therefore, in the following, this solution is also called a direct propeller drive. For the propeller drive motor, two main options exist: hydraulic or electric. An electric motor is the most straightforward and convenient choice. However, according to supplier information, electric motors lack the power density required to drive the propeller on model scale. Therefore, because of the high power density of hydraulic motors, these should be considered as an option as well.

A choice has to be made based on the available power as well as on the consequences for the measurements to be performed with the pod model. As for the available power, the torque is generally the limiting factor. Maximum rotation rates are generally sufficient and for electric motor mostly much higher than required. Therefore, a choice has to be made whether a reduction gearhead can be used to reduce the rotation rate while increasing the torque. In addition, the suitability of integrating a shaft sensor with the direct-drive motor has to be taken into account.

During the predesign phase of the project as reported in [41] and [43], a thorough market search has been performed which will be summarised here. The following subsections discuss the different options and lead to the selection of the most promising approach towards a successful design of a direct propeller drive.
Hydraulic motors

When considering high power density, and especially high torque, hydraulics is one of the primary options. In [41], [43] and [9], hydraulics have been investigated as an option for the direct propeller drive. Hydraulics have the promise of high power density (Table 4.1), in combination with good control opportunities.

A hydraulic motor is powered by a so-called power pack, which is connected to the motor by pipes or tubes that carry the powering fluid. The power pack contains the driving pump and the fluid reservoir as well as the control and safety valves. The produced torque is roughly proportional to the pressure while the rotation rate is determined by the fluid velocity. Pressures inside a hydraulic system can go up to 150-300 bars, dependent on the motor type and the required torque.

Table 4.1: Available versus required power of commercially available hydraulic motors [43]. Of the required power, only the torque is shown since the available torque most critical.

<table>
<thead>
<tr>
<th>Size</th>
<th>Available</th>
<th>Required</th>
</tr>
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<tbody>
<tr>
<td></td>
<td>Rotation rate</td>
<td>Torque</td>
</tr>
<tr>
<td>Cat. 1</td>
<td>Cat. 2</td>
<td>Cat. 3</td>
</tr>
<tr>
<td>[mm]</td>
<td>[rpm]</td>
<td>[Nm]</td>
</tr>
<tr>
<td>45</td>
<td>1500</td>
<td>8</td>
</tr>
<tr>
<td>60</td>
<td>1500</td>
<td>25</td>
</tr>
<tr>
<td>80</td>
<td>1500</td>
<td>40</td>
</tr>
</tbody>
</table>

The crucial advantage of hydraulics is the high power density of the motor; it can provide sufficient power at the right rotation rates, even for the most demanding experiments in Category 3. Further, as a result of internal pressures, the motor does not need to be made watertight. Also, with respect to the high internal pressures, the conditions in the depressurised towing tank are negligibly different from atmospheric conditions. So, no specific measures are necessary to operate the motor under vacuum conditions. In addition to these clear advantages, hydraulics also imply several serious drawbacks:

- No motors with a diameter smaller than \( \varnothing = 45 \text{mm} \) could be found
- The pipes or tubes between the power pack inside the ship model and the motor inside the pod gondola will have to pass through the six-component pod-unit balance. In this way, the pipes (or tubes under high pressure) form a rigid connection between the measuring plate of the force balance and the fixed world, thus disturbing the measurements. Figures 4.1 - 4.3 illustrate several possible solutions to this problem, together with the main implications [43]. None of these solutions completely eliminates the problem and each solution also introduces new problems. In all cases, hydraulics can be expected to adversely influence the accuracy of the pod-unit measurements.
- The risk of leakage of hydraulic fluid into the tank water cannot be eliminated completely. Contaminations of the tank water with hydraulic fluids may disturb the towing tank measurements through, for instance, altered skin friction of the ship model. This problem can be solved by using a hydraulic fluid with minimal consequences for the towing-tank test results. Possible fluids have been reported by Boink et al. (2003) in [9].
No rigid connections between the ship and the towing-tank carriage are allowed. So, no tubes or pipes can run from the carriage to the ship body. Therefore, the power pack has to be placed inside the ship body. The power pack is a large and heavy object, which will cause weight and space problems in smaller models.

At MARIN, hydraulics is an unknown technology to drive ship model propellers. Regarding the required user friendliness of the designed solutions, this is a drawback of hydraulics. This problem can partly be tackled by putting much emphasis on the user-friendliness in the further design and by educating the tank operators in the use of hydraulics. However, that will only partly eliminate the risks involved in introducing a new technology. On the other hand, introducing a new technology enhances the range of available tools to solve future drive technology issues, which can be regarded as an advantage of hydraulics.

Steering the pod requires an additional complicated construction on top of the pod-unit balance: a so-called rotary joint for hydraulic fluids.

Summarising, the high power density of hydraulics can be utilised at the expense of several major drawbacks. Some of these drawbacks can be solved. The expected adverse effects on the pod-unit measurements may not be acceptable for many experiments. Therefore, hydraulics will not be a solution for the main types of experiments. However, for experiments with extreme power requirements, a hydraulic motor seems to be the only solution available. Therefore, a predesign study has been performed, which is reported in Section 4.4.
4. Driving the Propeller

Figure 4.1: First possible solution to pass the hydraulic pipes or tubes through the pod-unit balance. The complete power pack is placed on top of the pod-unit force balance, similar to the current solution. Main drawback is the large size of the power pack, which makes this solution very impractical. Also, a large weight on top of the balance will cause errors when the ship is oriented non-horizontally, like in waves. Also, vibrations of the power pack can be expected to disturb the force measurements.

Figure 4.2: Second possible solution to pass the hydraulic pipes or tubes through the pod-unit balance. The base plate and the measuring plate are connected by hydraulic tubes with a stiffness that is either known or that has a negligible effect on the force measurements. If the stiffness is known, it can be taken into account during the data processing. Otherwise, the connecting tubes or pipes can be incorporated in the calibration of the balance. The main problem here is that the high pressures in the piping make it complicated to design a connection with negligible effect on the force measurements. Further, the high and variable fluid pressure inside the tubes causes a varying tension, thus a varying effects on the force measurements. As a result, incorporating the stiffness of the piping into the calibration will be very complicated, if not impossible. Thus, the accuracy of the pod-unit force measurements is likely to be adversely influenced when this solution is chosen.
The hydraulic pipes can also be designed into a smart shape and then be instrumented with strain gauges. In this way, the hydraulic piping functions as a force transducer. Although this solution is space saving and robust, the varying pressure inside the tubes causes strain at the pipe surface that is of similar order of magnitude as the strains caused by the pod-unit forces. This is an error source that cannot be ignored and extensive testing will be required to determine its effects and to find a way to incorporate it into the calibration and data-processing routines.
4. DRIVING THE PROPELLER

Electric motors

Compared to hydraulics, an electric motor has lower power density and generally too high rotation rates, especially at small scales (Table 4.2). When the required shaft power is compared with the supplier specifications, no electric motor can be found with sufficient power density. However, using an electric motor introduces fewer problems than a hydraulic solution.

An electric motor requires drive and control electronics, which occupy substantial weight and space. However, only wires run from the motor to the drive electronics. Therefore, the drive electronics can be placed outside the ship body, on the towing tank carriage.

The motor has to be made watertight as well as resistant to the conditions in the depressurised towing tank. These are however technical aspects on which broad experience is available at MARIN. Therefore, making the motor watertight and vacuum resistant is not expected to yield crucial bottlenecks. With an electric motor, only cables run from the pod-unit balance to the ship body. The cables can be arranged in such a way that pod-unit measurements are not disturbed significantly.

It can be concluded that an electric direct propeller drive is the preferred option as it solves existing problems without creating new ones. Therefore, it should be investigated to what extent the available motor power can be optimised.

For the given size ranges, the type of electric motor with the highest power density was found to be a brushless, permanent-magnet, AC, synchronous servo motor (BLAC motor). In a BLAC motor, the torque is mainly determined by the mean current (RMS) through the motor coils (windings), the rotation rate by the mean voltage (RMS) over the windings. Several opportunities exist to increase the available shaft power of a BLAC motor. However, the key to achieve sufficient power is to realise that supplier specifications can be ‘thoughtfully overruled’. Supplier specifications are valid for very general conditions. These general supplier conditions do not necessarily match the conditions at MARIN. This means that, with the different conditions in mind, certain specifications can be ‘overruled’.

Elaborating this further, it can be assumed that a towing-tank test typically takes 1-2 minutes, followed by a rest period of about 10-15 minutes. As a result, the motor is operated only for 2 hours, during a testing session of 16 hours. However, most electric motors are designed for industrial applications where the motor should run fail-safe for thousands of running hours. So, further assume that overruling the specifications reduces the lifetime of a motor with a factor of 20. For a motor with a nominal lifetime of 5000 hours, this means that still will last for over 125 testing sessions of 16 hours. In conclusion, it can be said that lifetime reductions are generally negligible for MARIN applications. Especially when the costs of replacement parts are low compared to a typical towing tank budget. Considering this, the following measures will increase the available torque.

- For BLAC motors, the maximum rotation rate can often be exceeded as the maximum rotation rate is mainly determined by the lifetime of the bearings. A higher rotation rate implies quicker wear of the bearing and thus a reduced lifetime. However even if exceeding the rotation rate with a factor 2 would reduce the lifetime with a factor 20-50, this is still an acceptable consequence.

- If the available rotation rate is higher than required, this excess rotation rate can be reduced by the use of a reduction gearhead to increase the available torque.

- Utilise the tank water for cooling. The coils inside the motor (windings) produce heat as a result of their 'ohmic' resistance. If the windings reach a temperature higher than the
maximum winding temperature (typically 125-140°C), the insulation will be destroyed. If the windings are effectively cooled, a higher maximum current (torque) can be allowed. When operated under safe, supplier specified conditions, an electric motor can produce twice its regular torque when water-cooling is applied. If a reduced lifetime can be accepted and when duty cycles are short, an increase of about 2.5-3 times the specified torque may be possible. However, it should be kept in mind that the break down of the insulation occurs rather abrupt once the maximum winding temperature is exceeded. Monitoring the winding temperature during operation is a way to avoid exceeding of the maximum temperature.

Exceeding the rotation rate specifications in combination with a gearhead will further increase the available torque. Applying water cooling in combination with a gearhead, while exceeding the rotation rate with a factor 2, yields the values given in Table 4.2. The required rotation rate is not shown as it is sufficient for all applications. For the frameless\(^1\) motor, only water cooling is assumed.

Table 4.2: Available versus required power of commercially available BLAC electric motors [43] which are optimised through water cooling and ‘thoughtfully overruling’ the specifications. Between brackets, the available torque without optimisation is given. Of the required power, only the torque is shown since the available torque is most critical. For the frameless motor, only water cooling is assumed.

<table>
<thead>
<tr>
<th>Motor type</th>
<th>Size</th>
<th>Available</th>
<th>Required</th>
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<tbody>
<tr>
<td></td>
<td>Diameter</td>
<td>Rotation</td>
<td>Torque</td>
</tr>
<tr>
<td></td>
<td>[mm]</td>
<td>rate [rpm]</td>
<td>[Nm]</td>
</tr>
<tr>
<td>Regular + gearhead</td>
<td>32 100</td>
<td>1500 1.32</td>
<td>0.33 2.58</td>
</tr>
<tr>
<td>Frameless</td>
<td>70 160</td>
<td>1500 10</td>
<td>5 1.28</td>
</tr>
<tr>
<td></td>
<td>80 120</td>
<td>1500 10</td>
<td>5 1.30</td>
</tr>
</tbody>
</table>

Suitability for a shaft sensor

In a comparison of the different motor options, it should be kept in mind that for experiments of Category 2 & 3, force measurements at the shaft are required. This implies that space should be available to mount a propeller-shaft sensor, including the data transfer to the fixed world. Further, a shaft sensor implies that vibrations excited by the motor, so-called torque and speed ripple, should not disturb the shaft measurements.

A hollow-shaft or frameless motor provides much flexibility in integrating the shaft sensor with the drive motor, which yields the most compact solution. With a frameless or hollow-shaft motor, it is possible to use the existing shaft sensor design to measure propeller torque \(M_{\text{prop}}\) and thrust \(F_{\text{prop}}\). If the motor has a regular shaft, the propeller-shaft sensor has to be redesigned.

In the market search reported in [41, 43], no hydraulic motors could be found which are frameless or hollow-shaft. This is mainly caused by the principle of design of most relevant motor types, which implies a blind shaft i.e. the shaft does not run all the way through the

\(^{1}\)A frameless motor is purchased as a separate rotor-stator package, without a housing, shaft and bearings. Thus, a frameless motor offers the opportunity of maximum integration of the different functions.
motor. This means that the rotation rate sensor, necessary for measurements as well as for feedback, also has to be placed at the front end of the motor. Thus, with respect to the available space for a shaft sensor, hydraulics are not ideal. However, this drawback is not expected to form a crucial bottleneck as it is largely counteracted by the compactness of hydraulic motors.

Most electric motors have a through shaft, which makes it possible to mount a rotation rate sensor at the rear end of the motor. Also, frameless or hollow-shaft electric motors are commercially available in the required size ranges. Therefore, with respect to the suitability for a shaft sensor, an electric motor is a better option than a hydraulic motor.

Reduction gearhead

Especially small size electric motors suitable for Category 1 experiments have a maximum possible rotation rate that is much higher than required. If this excess power can be utilised through a reduction gear, the available power density can be greatly enhanced.

A reduction gearhead is mounted at the front end of the motor, in line with the shaft i.e. the shaft centerlines of the motor and reduction gear coincide. Most relevant gearheads are of the planetary type, which is a compact way of building gear trains with high reduction ratio and high torque. When using a reduction gearhead, the rotation rate is reduced by a certain ratio, while the torque increases with the same ratio. No hollow-shaft gearheads in the relevant size ranges were found.

The advantage of a gearhead is that the available motor power can be fully utilised. However, the gearhead increases the motor length and no hollow-shaft options exist. Furthermore, the gears will again introduce vibrations at the propeller shaft, whereas reducing the gear vibrations of the existing design was one of the goals of the design.

As a result of the reduction ratio (10-15), the vibration noise caused by a gearhead can be expected to appear at frequencies higher than the second or third blade harmonic. Therefore, this vibration noise may be removed by low-pass filtering. Also, a gearhead will be tuned much better than the currently used gear train, as it is purchased as a fixed unit. So, a direct-drive solution with a gearhead can still be considered as an improvement with respect to the vibration noise in the existing set-up. Nevertheless, the consequences given above make a gearhead unsuitable for applications where shaft measurements are required.

Towards a direct propeller drive

From the market research and the discussion given above, the following conclusions can be drawn:

- For the small scale experiments in Category 1, no suitable hydraulic motor can be found.
- The power demands for experiments in Category 1 can only be achieved through the use of a water-cooled electric motor with a gearhead. The maximum rotation rate specified by the supplier needs to be overruled with a factor $\approx 1.4$, in combination with water-cooling.
- A water-cooled electric motor without a gearhead can supply sufficient power for Category 2 experiments.
- Hydraulic motors are the only way to meet the power demands for experiments in Category 3.
When a propeller-shaft sensor is required, a frameless or hollow-shaft motor is to be preferred. A frameless or hollow-shaft motor allows full integration of the shaft sensor with the motor, which yields the most compact solution. If the motor has a ‘blind shaft’, it is only possible to apply a shaft sensor, if the motor is very compact, like a hydraulic motor. However, this requires the design of a shaft sensor first.

A motor with gearhead is not suitable for combination with a propeller-shaft sensor.

To apply hydraulics in combination with pod-unit measurements, several crucial bottlenecks need to be solved first.

These conclusions lead to the decision scheme given in Figure 4.4. Based on a few criteria, the different design areas are defined. Indicated in the scheme are the design areas which lead to three design paths that match with the three main experiment Categories given in Section 3.4. With the results of these three design paths, over 80% of the model-scale experiments with pods are covered. The three design paths are defined as:

**Path A** Small-scale, low-tech, electric drive method, suited for Category 1 experiments. A housed motor with gearhead will be used. No shaft sensor will be applied, the propeller torque $M_{z\text{-prop}}$ will be determined from the measured motor current. For the pod-unit measurements, a three-component sensor of an existing design will be placed in the steering shaft. This pod-unit sensor measures $F_{x\text{-unit}}$, $F_{y\text{-unit}}$, and $M_{z\text{-unit}}$.

**Path B** Large-scale, high-tech, electric drive method, suited for Category 2 experiments. A frameless motor without reduction gear will be used. A propeller-shaft sensor will be integrated with the drive motor. The pod-unit forces will be measured by a redesigned six-component, pod-unit balance as described in Chapter 6.

**Path C** Hydraulic drive method, suited for Category 3 experiments. A propeller-shaft sensor needs to be designed which fits a ‘blind shaft’ construction. Also, solutions need to be found for the problems described above.

The experiments in Category 1 and 2 both are of higher commercial relevance than Category 3. Further, Path C contains several bottlenecks which imply an extensive and risky design effort. Therefore, within the current project, Path A has been elaborated until the first commercial application in the SMB towing tank (Section 4.2). Path B has been elaborated until a prototype, with which all relevant functionality could be tested under towing tank conditions (Chapter 4.3). For Path C, only a predesign study has been carried out by a student group under supervision of the author (Section 4.4). In the following, the results with respect to the propeller drive method are presented. The design of the sensors measuring the pod-unit and the propeller-shaft forces is presented in Chapter 6.
Figure 4.4: Decision scheme defining the three design paths which together lead to a pod modelling toolbox which covers at least 80% of all model-scale experiments with pods.
4.2 Path A: Small-scale, low-tech drive method

The result of Path A will be suited for Category 1 experiments. The following subsection describes the design and testing of two subsequent prototypes, as well as the first commercial application of the design result. The presented results are also reported in [43] and [42].

Prototype A1: Testing available motor power and using motor current as torque measurement

Sufficient power for Category 1 experiments can only be achieved when water cooling is combined with the use of a gearhead while exceeding the maximum rotation specified by the supplier. This was concluded from the information gathered in the predesign stage of the project[41, 43], as presented in the previous Section. The first prototype of Path A has been built to verify this conclusion.

As a result of using a gearhead, no space is left for a shaft sensor. However, measurement of the propeller torque $M_{x-prop}$ is required for part of the experiments in Category 1. For BLAC motors, the produced torque is proportional to the RMS current through one of the motor windings: $I_{motor}$. The drive and control electronics of the motor generate a signal proportional to $I_{motor}$. In the testing of Prototype A1, it was investigated whether this signal can be used as a measure for $M_{x-prop}$ [42].

In the market research, two suppliers were found that produce powerful small-scale BLAC motors; Maxon [36] and Faulhaber [15]. The power and size combinations of the Maxon EC-range matched best with the requirements. Therefore, a prototype was built with a motor from this range. Figure 4.5 shows an overview drawing of the prototype. The motor was made watertight using seals, o-rings and coating. The motor was placed inside a tight fit aluminium housing, which provides cooling through the contact with the tank water. A temperature probe was integrated with the motor housing to monitor the winding temperature. The rotation rate signal from the encoder is used for feedback as well as for measurement.

Figure 4.5: Overview drawing of first prototype of Path A: the small-scale, low-tech electric drive method. The main parts are indicated: 1 sealing cap; 2 pod housing; 3 tightening bolt; 4 Maxon gearhead GP42 ratio (15 : 1); 5 Maxon motor EC40; 6 encoder housing; 7 Scancon encoder.
4. DRIVING THE PROPELLER

The prototype pod was mounted on a frame which could be placed under water (Figure 4.6). The shaft of the prototype pod model was connected to a torque sensor, which was connected to an induction brake through right-angle gears. The induction brake was placed above the water. With the induction brake, it was possible to apply a controlled constant load torque. During these tests, the delivered torque $M_{\text{prop}}$ was measured together with the rotation rate $n$, the motor current $I_{\text{motor}}$ and the winding temperature $T_{\text{winding}}$.

Runs at different rotation rates were made to determine the temperature as a function of motor torque. The temperature time trace was determined at increasing torque. Thermal equilibrium was mostly reached within 60 seconds, which is the typical duration of a towing-tank test. Hence, the thermal equilibrium will always be reached within a single run. Therefore, only the steady-state temperature has been taken into account as a design parameter.

At fixed rotation rate, the load torque was increased until the steady-state temperature was a few degrees below below the maximum winding temperature given by the supplier. The corresponding delivered motor torque was recorded as the maximum torque. These values are given in Figure 4.7. Also shown is the required torque, as well as a typical propeller load curve of a highly loaded propeller.

It can be seen that the found maximum torque ($\approx 2.3$ Nm) is well above the required level and independent of the rotation rate. The maximum torque shown in Figure 4.7 corresponds to the maximum current (10A) of the drive electronics. In [42] it was extrapolated that, with drive electronics that can supply a higher current (11A), the maximum torque can be increased to

Figure 4.6: Overview of the 'test bench' set-up to test Prototype A1. The main parts are indicated.
Figure 4.7: Measured available torque (diamonds) of Prototype A1, plotted against rotation rate. The measured torque is compared with the required torque (dashed line). Also, the load curve of a typical heavily loaded propeller is shown (solid curve). It can be seen that the required torque can well be achieved and that sufficient power is available to drive a heavily loaded propeller up to about 1300 rpm.

about 2.7 Nm. It can be seen in Figure 4.7 that sufficient power is available to drive a heavily loaded propeller up to about 1300 rpm. The full 1500 rpm cannot be reached. However, this is not a problem as heavily loaded propellers generally do not require very high rotation rates.

The maximum rotation rate specified by the supplier is exceeded. No duration tests have been performed. So, it remains to be determined, to what extent the lifetime of the parts is reduced. As yet it can be concluded that the motor lasts at least 4 towing-tank tests. Further, from the tests with Prototype A3 (see below) it was found that the encoder seems to be the most critical part with respect to lifetime. However, the costs of the motor parts are low (< €750,-) compared to a typical project budget (50-100 k€). Therefore, even if the parts would last only a few commercial projects, this is not a problem.

The results on water cooling have been extrapolated towards a one step smaller and larger motor in the same Maxon EC-range. These results were compared to the scaled power demands of a worst-case pod [42]. It was found that if the same design principles are applied to a smaller or a larger motor, the available power will still be sufficient.

Figure 4.8 shows the measured torque $M_{x-prop}$ plotted against the measured motor current $I_{motor}$. Also shown are linear curve fits, together with an estimate of the uncertainty $U$ of the fitted curves. It can be seen that:

- the relationship is exactly linear
- the slope of the curve is independent of the rotation rate
- the intercept of the curve varies with rotation rate, which is caused by the friction of the seal
4. DRIVING THE PROPELLER

Figure 4.8: Measured motor torque versus the motor current signal from the drive electronics. Calculated linear fits are shown, together with an estimate of the uncertainty $U$.

In [42] it was reported that the intercept is approximately linearly proportional with the rotation rate $n$. The found relationship can be expressed as:

$$M_{x\text{-prop}} = C_1 I_{motor} - C_2 n - C_3$$

where, for the current prototype,

$$C_1 = 0.305 \frac{Nm}{A}$$
$$C_2 = 0.00013 \frac{Nm}{rpm}$$
$$C_3 = 0.195 Nm$$

The parameters $C_1 - C_3$ are highly dependent on shaft friction and should be determined each time the motor or the shaft seal are re-assembled. With the given relationship, for measurement frequencies up to 10 Hz, the uncertainty was estimated in [42] to be $\leq 1\%$, which is sufficient for Category 1 experiments.

With these tests, it has been shown that the approach of Path A is feasible. Therefore, a second prototype was built, which is suitable for application in the towing tank. The design and testing of the second prototype of Path A (Prototype A2) is reported in the following subsection.
Prototype A2: Testing an implementation-ready prototype in towing tank

The successfully tested concept of Prototype A1 has been developed further into a prototype that is ready for implementation in the towing tank; Prototype A2. The main goal of these prototype tests was to verify the designed concept under operational towing-tank conditions. Also, the feasibility of applying an existing three-component (3C) force sensor as pod-unit sensor has been tested.

For this prototype, the seals and the end cap of the motor have been made as compact as possible. Further, no adaptations to the motor have been made. A typical pod shape (the Mermaid from Figure 1.4) has been machined from aluminium. It is necessary to machine the pod housing from aluminium, to facilitate the water cooling. It was found that the costs of an aluminium housing are negligibly higher than for polyurethane. The machining of the aluminium was well within the capabilities of the design tools and milling machines available at MARIN.

Inside the aluminium housing, a space was machined which fits tightly around the prototype motor. The pod shape was built around the prototype motor (Figure 4.9). In this way, the water cooling by the tank water is identical to the set-up in which Prototype A1 was tested. Between the steering shaft and the aluminium pod housing, an existing 3C pod-unit force sensor was placed which measures $F_x$, $F_y$, and $M_z$.

The prototype pod model was built inside a typical ship model (Figure 4.10). A regular propulsion test was performed in the Deep Water towing tank (DT). Further, runs have been performed where the pod was placed under an angle, to test the 3C pod-unit force sensor. As reported in [43], no operational problems were encountered and the 3C force sensor worked well to measure the pod-unit forces. It was concluded that the designed concept was ready for implementation.

![Figure 4.9: Overview of Prototype A2, built inside a model-scale version of the Mermaid (Figure 1.4), with an arbitrary propeller. The outside pod housing is machined from aluminium.](image)

Prototype A3: Applying the design in a commercial towing-tank test

In January 2006, the result of Path A was applied for the first time in a commercial towing-tank test. Two new versions of Prototype A2 were built into the pod shape to be tested: the ABB Compact Azipod (Figure 4.12). The pods were mounted inside the ship model. A standard test series of seakeeping and manoeuvring tests has been performed.
Figure 4.10: Direct-drive Prototype A2:
(a) compared with the existing set-up with external drive motor, where the geometric advantages are obvious.
(b) built inside a ship model (front), together with an identical pod-unit (back) according to the existing set-up with external drive motor.
Due to an unexpected difference between the motor in prototype A and the newly supplied motors of equal type number, water leaked into the motor during the first series of tests. The tests had to be abandoned and the customer was put on hold for several weeks. To solve the problems quickly and to avoid any further leakage problems, a third prototype (Prototype A3, Figure 4.11) has been designed and built. From this experience, it can be concluded that the sealing is very sensitive to minor geometric changes. And further that the watertightness should be thoroughly checked before the start of an experiment.

In the Prototype A3, a brass sleeve around the complete motor unit was designed, with fewer sealing points. The brass sleeve can be put under an air pressure slightly higher than atmospheric. It was necessary to use brass for the sleeve, as soldering connections were used to decrease the number of seals. As time was limited, soldering was applied as the quickest solution. As a result of these adaptations, no leakage occurred and the test could be completed successfully.

With prototype A3, the operational reliability has been improved as the slight overpressure inside the brass sleeve provides a high guarantee against leakage. The brass sleeve enlarges the required space for the motor. Also, the heat conductivity of brass is about half the heat conductivity of aluminium and the sleeve has no tight fit around the motor parts. As a result, the cooling efficiency is reduced.

However, during the tests with prototype A3, the motor was fully loaded. Therefore, the reduced cooling efficiency does not seem to be significant. This can be explained by the fact that the cooling efficiency is only partly determined by the heat conductivity and much more by surface effects which may be comparable for aluminium and brass. For future implementation, it is recommended to design a final prototype which is a trade off between prototypes A2 and A3. Hence, the compactness of Prototype A2 should be combined with the anti-leakage measures in Prototype A3.

Figure 4.11: Overview drawing of the third prototype of Path A (Prototype A3). 1 Front cap which contains the shaft seal and is soldered to the brass sleeve (2). The pod model can be put under pressure, through a silicon hose which is connected to the hose nipple (3). The hose nipple is soldered to the watertight end cap (4). The hose also contains the cables.
Conclusions on Path A

The concept developed through Path A is a successful approach to solve the problems stated in Section 3.2 for Category 1 experiments. Summarising, the following conclusions can be given:

- A small-scale electric motor with a gearhead can supply sufficient power for Category 1 experiments.
- No propeller-shaft sensor can be applied. However, $M_{x-prop}$ can be reliably be determined from the measured motor current $I_{motor}$ and rotation rate $n$.
- The pod-unit forces $F_{x-unit}$, $F_{y-unit}$ and $M_{z-unit}$ can be measured using an existing three-component sensor.

Thus, all specifications for Category 1 experiments have been met and Path A has been completed successfully. An additional design effort is recommended to combine the high compactness of prototype with the reliability of Prototype A3.

Within the current project, no destructive duration tests have been performed. During the prototype tests, the maximum winding temperature specified by the supplier has not been exceeded. However, supplier limits are generally very safe. Therefore, it will be very useful to perform a destructive duration test i.e. increasing the load torque until the motor burns within the duration of a typical towing-tank test. Assuming that supplier specifications are about 10-25% too safe, a further increase in available torque of 10-25% can be expected.

From the experience with Prototype A3, it seems that the encoder (used for position feedback) is the most vulnerable part of the motor\(^2\), which thus determines its lifetime. One encoder broke down after Hence, eliminating the encoder will make the motor more robust. Additionally, eliminating the encoder decreases the length of the motor. Elimination of the

---

\(^2\)One encoder broke down at the end of the tests with Prototype A3. This encoder had already survived several prototype tests of about 30 hours in total. Since the encoder was not mounted fully correct, it is expected that typically, when mounted correctly, encoders will last longer. However, this should be investigated in a duration test.
encoder may be possible when the control strategy is changed in such a way that only the built-in Hall-elements can be used for feedback. It should be investigated whether this is a feasible option for improvement.

4.3 Path B: Large-scale, high-tech drive method

From path A it was concluded that using the tank water in combination with an aluminium pod housing as an effective water cooling, doubles the available torque of an BLAC motor. This knowledge on water cooling creates the opportunity to design a solution for Path B. Therefore, the next challenge is to design a large-scale high-tech solution for Category 2 experiments. The following subsections describe the prototype design and test results for Path B as also reported in [44], [45] and [49].

Prototype B1: Testing available power and control accuracy

As found during the market research, the best way to solve problems for Category 2 experiments is to:

- use a frameless electric motor,
- use the tank water for cooling,
- integrate the shaft sensor with the drive motor.

The first prototype of path B was designed to test whether this approach is feasible [44]. The available power, control accuracy and disturbance of shaft measurements by the motor were investigated.

Based on a visit to the Hannover Messe, a selection was made of possible frameless motor suppliers [43]. From this selection the suppliers Moog [52] and Parvex [37] produce the most powerful BLAC motors in the size indicated in Table 3.2. The motors from both suppliers were highly comparable on technical aspects as well as on price. Therefore, based on a quicker delivery time, the Parvex motor was chosen for the first prototype.

Figure 4.13 shows an overview drawing of the prototype, with the main parts indicated. The motor as well as the resolver (position feedback device) was purchased as a separate rotor-stator package, without any housing, shaft or bearings. For the shaft sensor, an existing design was used, which measures $F_{x\text{-prop}}$ and $M_{x\text{-prop}}$.

Also, the slip ring unit, used for data transfer of the shaft sensor is similar to the ones used in the existing pod model units. Slip rings are a straightforward approach which can be copied directly from the existing units. Therefore, this is a suitable solution for the prototyping phase of the project. However, regarding Electro Magnetic Interference and noise, slip rings are not optimal. In Section 6.5 the different data transfer options will be discussed.

The permanent magnet rotor was press-fitted on a hollow stainless steel shaft with a flange at the front end to mount the shaft sensor. The shaft sensor was mounted at the front end of the motor, with the wires running through the hollow shaft to the slip ring unit at the rear end of the motor. The stator was shrink fitted into an aluminium housing and a frameless resolver was mounted at the shaft between the motor and the slip ring unit.

At the rear end of the motor, a watertight cap with cable glands was mounted. At the front end of the motor an oil seal was placed between the shaft and the aluminium housing.
The drive electronics, also from Parvex, were mounted inside a closed cabinet, together with all the input and output filters required to minimise the EMI effects. During the prototype tests, the winding temperature was measured using a thermocouple attached to the stator.

The first tests with the motor were performed on a low-cost, straightforward test bench, which was specifically built for the current project (Figure 4.14). On the test bench, the pod model could be placed under water in a water container. The pod-model shaft was connected to a torque sensor which was connected to another BLAC motor which was operated as an induction brake with controllable torque. In this way, a variable load could be applied to the pod model shaft. All connections were made using torsionally stiff, flexible shaft couplings. On the shaft, an encoder was mounted to measure the rotation rate.
Figure 4.13: Overview drawing of Prototype B1
Figure 4.14: Overview drawing of the test-bench on which prototype B1 was initially tested. 1 Water container 2 Adjustable aluminum frame 3 Prototype B1 4 Flexible couplings 5 Torque sensor 6 Slip rings 7 Encoder 8 Servo motor, used as an induction brake
First, the heat balance of the motor was determined. As the size of the pod model is larger, its heat capacity is larger, which leads to a larger time constant of the thermal system. As a result, reaching thermal equilibrium takes more time (about 600 s) than the duration of a typical towing tank run (about 60 s). Therefore, in Path B, it is essential to take the thermal dynamics into account, instead of only the steady-state thermal equilibrium.

To determine the thermal time constant as well as the overall heat resistance, the temperature as a function of time was measured at different torques. The winding temperature as a function of motor current can be modelled as:

\[ T_w = T_a + R_{th} \left[ 1 - e^{-\frac{t}{t_{tc}}} \right] (I^2 R_{act}) \]  

(4.2)

with \( T_w \) the winding temperature, \( T_a \) the ambient temperature, \( I \) the motor current and \( R_{act} \) the terminal ohmic resistance of the windings \(^3\). The thermal parameters are the thermal resistance \( R_{th} [K/W] \) and the heat capacity, \( c[J/K] \). Figure 4.15 shows an example of the curve fitting. It can be seen that the thermal behaviour of the motor can be predicted well with the model given in Equation 4.2, table 4.3 shows the parameters found in [44]. Also, the relationship between the measured torque and the motor current has roughly been determined to be:

\[ M_{x-prop} = 1.36 \text{ [Nm/A]} I_{motor} - 0.71 \text{ [Nm]} \]  

(4.3)

Here, no exact relationship between \( M_{x-prop} \) and \( I_{motor} \) was required, as the propeller torque is also measured by the integrated propeller-shaft sensor.

Once the parameters \( R_{th} \) and \( c \) are known, Equation 4.2 can be rewritten to determine the maximum on-time \( t_{max} \) for a given motor current:

\[ t_{max} = -cR_{th} \log \left[ 1 - \frac{T_{max} - T_a}{T^2 R_{act} R_{th}} \right] \]  

(4.4)

where the maximum on-time \( t_{max} \) is the time at which the winding temperature \( T_w \) reaches its maximum value \( T_{max} \). With the values in Table 4.3 together with Equation 4.3 this yields maximum on-time as a function of torque as plotted in Figure 4.16. It can be seen that for a period of 60 seconds, being the typical duration of a towing-tank test, the motor can provide a torque of 17Nm, which is even more than required for Category 2 experiments. This means that the given motor can also be used for experiments in Category 3 that require a torque of 17Nm, or even 24Nm if the test duration is less than 30 s.

Table 4.3: Thermal parameters from Equation 4.2 found in [44].

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( R_{th} )</td>
<td>0.14 K/W</td>
</tr>
<tr>
<td>( c )</td>
<td>1100 J/K</td>
</tr>
</tbody>
</table>

Due to resonance problems with the test bench and the induction brake, the rotation rate control accuracy of the pod model could only be determined at a rather low servo stiffness. It was found that even with the low servo stiffness, the average rotation rate over a steady state run was controlled sufficiently accurate (not shown) [44].

\(^3\)The ohmic resistance of the copper in the windings increases with temperature, \( R_{act} \) is the resistance at the expected maximum value of \( T_w \).
Figure 4.15: Temperature as a function of time, measured at 660 rpm and at 10 Nm. Also shown is the curve fitted according to Equation 4.2.

Figure 4.16: Operation limits of prototype B3; shown is the maximum on-time as a function of motor torque. A duty cycle is assumed where the off-time is long enough for $T_w$ to reach $T_a$, about 10-15 min.
The dynamic control accuracy could not be determined correctly due to insufficient control of the variable load, in combination with resonance of the set-up. However, the results found in [44] were promising enough to conclude that sufficient control accuracy can be expected, once the servo stiffness is optimised for a towing-tank situation. Also, it was found that the dynamic behaviour of the motor influences the propeller-shaft measurements. This will be discussed further in Section 4.3.

From the tests with prototype B1, it could be concluded that the motor can supply sufficient torque for Category 2 experiments. Taking into account Equation 4.4, even the lower power range of Category 3 experiments may be possible with the current pod model. Therefore, the next step in the design could be made; testing the prototype in a practical towing tank situation.

Prototype B2: Practical application in towing tank

To test the pod model design under towing tank conditions, Prototype B2 was built [45]. The successfully tested pod model Prototype B1 was placed inside an aluminium pod housing which yields Prototype B2 (Figure 4.17). The pod and propeller shape were copied from a previously performed commercial project. The pod model was mounted in the open-water set-up of the Deep Water Towing Tank (Figure 4.18). In the open-water set-up, the pod model can be tested by itself without a ship model.

The pod model steering shaft was connected to an existing six-component pod-unit balance, on top of which a standard MARIN steering servo motor was placed. The pod-unit balance was mounted in the open-water sub-frame, which was mounted on the towing-tank carriage. Test runs were performed to determine the following aspects of the pod model design:

- operational feasibility of the drive motor,
- comparison of steady state measurements with identical test runs from a the previously performed project with the existing set-up and with identical pod and propeller geometry,
- dynamic quality of the measured data.

Figure 4.17: Prototype B1 built inside an aluminium pod housing.
4. Driving the Propeller

(a) Picture

(b) Overview drawing:
1. Prototype pod model (see also Figure 4.17)
2. Propeller shaft sensor that measures $F_{x\text{-prop}}$ and $M_{x\text{-prop}}$
3. Propeller
4. Strut of the aluminum pod housing
5. Flow profile
6. 6C-frame
7. Transmission wheel of steering servo
8. Steering shaft
9. Steering servo

Figure 4.18: Picture and drawing of Prototype B2 mounted in the open water sub-frame.
It was found that the designed prototype worked well in a practical towing-tank situation. The steady-state propeller-shaft measurements with prototype B2 were compared with identical runs, with identical pod and propeller, performed with the existing set-up with an external drive motor. The differences found are within the known uncertainty limits of towing-tank tests with pods. Also, the rotation rate control accuracy was found to be within the given specifications for the steady state runs.

Also, during runs with dynamic steering, the rotation rate could be controlled with sufficient accuracy. However, the measured rotation-rate signal contains EMI noise which makes it difficult to determine the instantaneous deviations from the set-point rotation rate.

When comparing the pod-unit measurements, we found significant differences with the identical test, performed previously. These differences can be explained by the existing problems with the six-component force balances as will be discussed further in Chapter 6. Again, it was found that the propeller-shaft measurements are disturbed by the motor as will be discussed further in the next section.

Prototype B1 has primarily been designed for prototyping purposes only. As result, it is too large in size for many existing pod shapes. The pod and propeller shape that have been used for Prototype B2 are relatively large ($\lambda = 13$).

Therefore, it was possible to fit Prototype B1 inside the given pod shape. However, it is necessary to design a more compact version of Prototype B1. This is possible since the watertight cap at the rear end of the motor can be made much more compact. Also, the shaft sensor can be integrated further with the motor shaft. The expected dimensions of a more compact prototype (Figure 4.19) have been compared with a range of existing pod shapes (not shown). It was found that, if the dimensions can be reduced to those given in Figure 4.19, the motor will fit into the majority of all large scale pod shapes.

Furthermore, the application area of the design can be greatly enhanced when the same concept is applied to a smaller motor. This is possible, because it was found that the available torque exceeded the requirements for Category 2 experiments. Therefore, it is recommended to design a second version of the pod model with a somewhat smaller motor, which can still...
deliver sufficient torque.

**Influence of the motor on the propeller-shaft measurements**

The design of a direct-drive method provides the interesting opportunity to perform dynamic measurements, which potentially opens a whole new area of hydrodynamic research on pods. The direct-drive motor eliminates the vibrations caused by the existing gear transmissions. When there is less noise in the recorded data it becomes possible to retrieve time traces or spectral information from the data.

However, electric motors do not run completely smoothly and exhibit dynamic variations of the torque and rotation rate (torque and speed ripple). In [44], [45] and [49], it has been investigated how these speed and torque ripples influence the shaft measurements. Speed and torque ripples appear as multiples of the shaft rotation rate, called shaft harmonics. In [44] and [45] it was found that the 1\textsuperscript{st}, 5\textsuperscript{th}, 6\textsuperscript{th}, 10\textsuperscript{th} and 15\textsuperscript{th} shaft harmonics are induced by the motor. These harmonics appear in the measured torque ($M_{x-prop}$), rotation rate ($n$) and propeller thrust ($F_{x-prop}$). Which shaft harmonics are induced by the motor is related to the number of motor poles (5) and the number of windings (3).

Figure 4.20 shows a typical Fourier amplitude spectrum for a bollard pull (zero carriage speed) run in the Deep Water Towing tank, with prototype B2. The motor-induced shaft harmonics are clearly observed as distinct sharp peaks. The test was performed with a 4-bladed propeller. Therefore, any variation of the propeller would appear as multiples of the number of blades times the rotation rate (blade harmonics). However, because of the uniform propeller inflow, no blade harmonics are present in the measured signals. Hence, it can be concluded that the found variations are all caused by the motor.
Motor-induced shaft harmonics can clearly be distinguished as sharp peaks in the Fourier domain. Further, these harmonics do not contain hydrodynamic information and can be considered as noise in the measured signal. Therefore, a filtering routine was defined which removes these frequencies from the measured signal by band-stop filtering in the Fourier domain.

Figure 4.21 shows the results of this filtering routine for a measured signal of $F_{x_{\text{prop}}}$ during a zigzag run with Prototype B2 in the Deep Water Towing tank. It can be seen that removing the motor-induced shaft harmonics yields a much smoother signal. Dynamic effects, like propeller thrust variations during manoeuvring or peak loads during a crash-stop manoeuvre, can well be determined with such signal quality. Similar results are found for $M_{x_{\text{prop}}}$, also measured in the propeller shaft. See also Section 6.3 for an example of dynamic effects during manoeuvring.

Removing motor-induced shaft harmonics is not always possible. Propeller induced harmonics contain valuable hydrodynamic information. So, when propeller induced shaft harmonics coincide with motor-induced harmonics, the motor disturbances cannot be removed. Therefore, it can be concluded that with Prototype B2, when applying the proposed filtering routine, it is feasible to perform dynamic shaft measurements, as long as motor induced harmonics do not coincide with hydrodynamic frequencies of interest (e.g. blade harmonics).

It would be even better if motor disturbances are not present at all. In [49] motor-induced shaft harmonics have been investigated further. The following conclusions have been drawn.

- For the combination of motor and drive electronics as used for Prototype B2, stiffening the servo loop does not reduce the speed and torque ripple.
- Motor-induced shaft harmonics in the measured signals are not always mechanically present. It was shown that EMI and mechanical vibrations due to resonance may enhance the measured amplitude of the motor-induced shaft harmonics.
- Motor-induced shaft harmonics can be reduced by:
  - finding servo drive electronics and a feedback device which can form a stable servo loop with sufficient bandwidth to capture the motor-induced shaft frequencies at high rotation rates,
  - finding a motor with a more ideal winding layout and more ideal magnetic structure on the rotor,
  - finding drive electronics that allows detailed tuning of all relevant parameters and not just the servo loop,
  - having the drive electronics tuned to the specific motor by a specialist from the supplier,
  - improved EMI reduction.

Speed and torque ripple can never be completely eliminated. Therefore, another option to avoid this problem is to search motors with different torque and speed ripple characteristics. When designing an experiment, a motor can then be chosen which only induces shaft harmonics that do not coincide with the blade harmonics.
Figure 4.21: Example of the proposed filtering routine applied to a time series of $F_{x\text{-prop}}$, measured during a zigzag run in the deep water towing tank. The steering angle alternates between $\pm 90^\circ$ with $\approx 3s$ stops in between. The upper figure shows the raw data measured at 1 kHz, the middle figure shows the data after low-pass filtering at 100 Hz, the lower figure shows the combined effect of low-pass filtering and band stop filtering all the shaft harmonics that are induced by the motor: the 1$^{st}$, 5$^{th}$, 6$^{th}$, 10$^{th}$ and 15$^{th}$ shaft harmonic.
4.4 Path C: Hydraulic direct drive

For Path C, only a predesign study has been carried out by a student group supervised by the author. In [9] the following conclusions and recommendations are given on the use of hydraulics.

- The J2 series from hydraulic motor supplier Eaton [24] meets the power requirements for Category 3 experiments (Table 4.4). Except that at the maximum torque, the maximum rotation rate is just below the required value. Figure 4.23 gives a preliminary sketch of the selected motor built inside a typical pod model.

- For the piping, copper pipes with an external diameter of 10 mm and an inside diameter of 8 mm are recommended.

- It is recommended to use spiral-shaped tubes, mounted in the \( z \)-direction between the pod-unit balance and a clamping point on the fixed world (ship body). This concept is illustrated in Figure 4.24. The stiffness of these tubes is then relatively low and can then be taken into account during calibration. However, it should be investigated whether the disturbance of the pod-unit measurements due to the varying hydraulic pressures can be kept at an acceptable level for Category 3 experiments.

- A water-glycol mixture is recommended as a replacement for traditional hydraulic fluids. These mixtures are not expected to disturb the measurements in case of leakage.

- Several alternatives for a control system are proposed. However, the control accuracy of the system could not be predicted and has to be determined experimentally.
Figure 4.22: Example illustrating the data quality of pod-unit force signals measured with Prototype B2 during the same run as in Figure 4.21. The data is low-pass filtered at 100 Hz and the motor-induced shaft harmonics are removed.
Figure 4.23: Preliminary outline of the hydraulic motor, selected in [9] inside a typical pod shape. (From [9])

Figure 4.24: Set-up to connect the hydraulic piping to the pod-unit balance, suggested by Boink et al. in [9].
4. Driving the Propeller

Table 4.4: Properties of the recommended hydraulic propeller drive motor from the J2 series of Eaton [24]. For the length, the propeller-shaft sensor is estimated to require 50 mm additional length.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter</td>
<td>60 mm</td>
</tr>
<tr>
<td>Length</td>
<td>180 mm</td>
</tr>
<tr>
<td>Torque</td>
<td>25 Nm</td>
</tr>
<tr>
<td>Rotation Rate</td>
<td>1450 rpm</td>
</tr>
<tr>
<td>Power Density</td>
<td>7.5 MW/m³</td>
</tr>
</tbody>
</table>

This study confirms the feasibility of hydraulics as a solution for Category 3 experiments. Also, it confirms that the effect of the pipes on the pod-unit measurements needs to be investigated. As pod-unit measurements are crucial in most Category 3 experiments, this aspect should be investigated before implementing hydraulics.

4.5 Conclusions

The toolbox defined in the current chapter provides a direct-drive motor inside the pod gondola, for over 80% of all model-scale experiments with pods. The approach for experiments of Category 1 and 2 has been proven successful. With these design results, a crucial breakthrough in pod model testing has been achieved for Category 1 and 2 experiments, which are commercially most relevant.

The direct propeller drive solutions create the promising opportunity to further improve pod-model experiments. Having eliminated the right-angle gears and the external drive motor greatly simplifies the construction of the pod-unit balance. As a result, the pod-unit balance can be redesigned to improve the accuracy and signal quality of the pod-unit force measurements. The design efforts in the current project on force measurement in pod models is described further in Chapter 6. First, Chapter 5 provides some essential background on six-component force measurement theory.

Another crucial benefit of a direct propeller drive is the opportunity to combine steering with accurate six-component force measurements. The tests with Prototype B2 show that a steering servo can straightforwardly be integrated with an existing pod-unit balance. Section 6.4 further elaborates steering of the pod and integration of a steering servo with the pod-unit balance.
5 Six-component force measurements

Abstract

From tests with existing six-component pod-unit balances, it was found that besides a balance redesign, also a review of the calibration and data processing routines was necessary. A six-component force balance measures forces in three perpendicular directions and the moments about these three axes. These six load components together determine the magnitude, direction and point of application of the load vector.

To take cross coupling (Sections 5.3 and 5.4) between force components into account, a calibration model is needed. Such a model can be of first or second order. Using a first-order calibration model is cheaper and more convenient. However, a second-order model will yield higher accuracies and thus potentially a better product for MARIN. Implementing a second order model implies many modifications in the work routines for the towing tank as well as the calibration department.

In the specification of the quality of a force balance, the measurement uncertainty should be used as the quantified doubt about the quality of the measurements. When specifying the measurement uncertainty of a six-component force balance, one should be aware that the uncertainty increases with an increasing number of loaded components.

5.1 Introduction

The design of a direct propeller drive has created the promising opportunity to improve the pod-unit measurements. From recent experience with problematic Category 2 experiments it was concluded that “something is wrong with those pod-unit balances” [61]. Therefore, two existing balances were investigated by Van Rijsbergen et al. [53]. It was concluded that the calibration method as well as the data processing needed to be revised. Therefore, the current chapter provides the essential basic theory on six-component force measurement and the application to pod models.

Van Rijsbergen et al. [53] also concluded that the balance itself should be redesigned to increase the reliability and accuracy of pod-unit force measurements, as it was lacking stiffness and reproducibility. The redesign of the balance is presented in Chapter 6, together with other aspects of measuring pod forces.

5.2 Why six components?

In general, the load vector at a certain point in a construction can be determined through measurement of forces in three perpendicular directions and the moments about these three axes. This is called six-component force or load measurement. These six force or load
components together determine the magnitude, direction and point of application of the load vector.

In ship research, measured loads at model scale are input for further design or research calculations. These calculations do not always require all six components. In many cases, only one or just a few components are sufficient. For these cases, a sensor measuring just the components of interest can be used. However, measuring only the components of interest may not always be sufficient. When the direction or point of application of the force vector is unknown, it is necessary to measure all six components. From a fundamental point of view, there are only two cases in which it is sufficient to measure only the components of interest.

**Sensor is insensitive to 'parasitic' forces.** Generally, a force sensor also responds slightly to force components other than the component it is designed to measure (cross coupling). The magnitude of this parasitic response depends on the design of the sensor and the loading situation. Whether the magnitude of this error is significant depends on the specific experiment it is used for.

**No 'parasitic' forces are present.** In some cases, when the construction to be tested is well-known, it can be determined beforehand that, during the test, the sensor will only be loaded by the components of interest. In this case, even if the sensor would be sensitive to 'parasitic' forces, the measurement error is negligible.

In all other cases, it is necessary to measure all six components to be able to calculate the load components of interest with the required accuracy. Recent developments in the pod industry have shown that many questions exist about the loads on pods (See also Appendix B). Especially the bearing loads i.e. lateral forces and moments on the propeller shaft as well as dynamic loads of the pod-unit on the ship hull seem to be important. Therefore, future research will require measurement of all components at the pod unit, as well as at the propeller shaft.

### 5.3 Sensor basics

The basic elements of a force transducer are illustrated in Figure 5.1. In all load sensors, forces and moments are measured through deformation or displacement [22]. The current chapter focusses on deformation sensors only.

The basis of a force sensor is its elastic element which is either part of the structure that is being investigated or is placed between the structure and a fixed reference system. Forces and moments on the elastic element cause deformations that can be measured by the primary transducer. This primary transducer is bonded to the elastic element where it generates a signal proportional to the local deformation (strain). The primary transducer yields an output proportional to the local deformation or strain, generally expressed in relative units \( \text{nm/nm} \) per unit of force or torque [23]. In the following, only the most commonly applied primary transducers are considered, which are being electrical resistance strain gauges.

Electrical resistance strain gauges are small foil resistors that are bonded to the surface of the elastic element. Strain at the surface of the elastic element causes strain in the gauge, which yields a proportional change of the resistance [23]. The changes in resistance are very small \( \Delta R / R = 1 \mu \Omega / \Omega \). Therefore, the change in resistance has to be measured in a Wheatstone bridge with a very stable supply voltage. Even when the bridge supply is optimal, output voltages are low, which makes strain gauge measurements sensitive to electromagnetic interference (EMI).
5.4 Calibration theory

General

In the measurement of six force components (6C), the output of the 6C-sensor is a vector of six or more strains ($\bar{\epsilon}$) which is related to the applied forces and moments through:

$$\bar{\epsilon} = f(\bar{F})$$  \hspace{1cm} (5.1)

with

$$\bar{\epsilon} = [\epsilon_1, \epsilon_2, \ldots, \epsilon_k]^T ; k \geq 6$$  \hspace{1cm} (5.2)

and

$$\bar{F} = [F_x, F_y, F_z, M_x, M_y, M_z]^T$$  \hspace{1cm} (5.3)

where $\bar{F}$ is the load vector defined in the coordinate system of the sensor. In the following, it is assumed that $k = 6$, which is the case for the majority of all six-component load sensors. The function $f$ in Eq. 5.1 has to be determined experimentally through calibration.

Ideally, each element of $\bar{\epsilon}$ responds to one force component only, which results in a diagonal matrix for the function $f$. However, in practice, this is never the case. Each element of $\bar{\epsilon}$ responds also slightly to other force components than the one it should measure. For instance if $\epsilon_1$ should measure $F_x$, it will also slightly respond to $F_y...M_z$; this is called cross coupling. To account for this cross coupling, a calibration model is needed to transfer measured strains into forces and moments.

Figure 5.1: The basic elements of a force sensor, illustrated for standard strain gauges.

To achieve accurate force measurements, the sensor should exhibit a reproducible, mostly linear behaviour which can be described by a generic model. Further, the sensor should be designed to respond to its specified load component only. However, in practice, a force sensor also slightly responds to other components. The following section describes how this cross coupling can be taken into account.
The higher the order of the calibration model, the more complicated and time consuming the calibration process will be. Until now, a first-order model has been used at MARIN. However, to achieve the maximum accuracy that is theoretically possible, a second-order model may be necessary for certain applications. Third-order (and higher) calibration terms are generally found not to be significant [10, 13, 16, 56]. In the following, the first and the second-order models will be explained further together with the corresponding calibration routines and data processing.

First order model

If we assume a first-order model, Eq. 5.1 becomes:

\[
\bar{\varepsilon} = T \bar{F} + \bar{A}
\]

(5.4)

where \( T \) is the transfer matrix of \( 6 \times 6 \) elements. The vector \( \bar{A} \) is the sensor offset at zero load. At MARIN it is common practice at the beginning of an experiment, to adjust the offset of the signal conditioners to make the output of all force sensors zero. Thus, the intercept can be ignored, which means that \( \bar{A} = 0 \). To calculate forces from measured strains by:

\[
\bar{F} = C \bar{\varepsilon}
\]

(5.5)

the calibration matrix \( C \) has to be calculated:

\[
C = T^{-1}
\]

(5.6)

When \( k = 6 \) in Equation 5.2, normal matrix inversion techniques can be used. When \( k > 6 \), the Moore-Penrose algorithm has to be used to calculate the inverse matrix [19].

Calibration

Strains are measured as bridge voltages, so in calibrations, strain is expressed in the unit \([\text{mV/V}]\), which is common in force measurement technology. The vector \( \bar{F} \) consists of forces and moment with SI-units \( \text{N} \) and \( \text{Nm} \) respectively. Therefore, the units of \( T \) are \([\text{mV/V N}, \text{mV/V Nm}]\).

Under the assumption of linear elastic behaviour of the sensor material, the superposition principle is valid. Therefore \( T \) can be determined through straightforward calibration. Each component of \( \bar{F} \) can be applied separately. The resulting strain vector \( \bar{\varepsilon} \) for each component is then stored in the corresponding column. By measuring the sensor response for a number of values for each separate component, the elements of \( T \) can be determined through regression methods.

Data processing

Matrix inversion techniques can be easily performed in any data-processing environment through built-in functions. However, when \( T \) is ill-conditioned, using its inverted form can cause small errors in the measured strains to blow up to large errors in the calculated forces and moments. This is called error magnification. Physically, an ill-conditioned transfer matrix means that there is no unambiguous relationship between the measured strain and the applied forces and moments. Therefore, an ill-conditioned transfer matrix implies poor sensor design.
Second order model

When a second-order model is assumed, Eq. 5.1 becomes:

\[
\vec{e} = T_1 \vec{F} + T_2 \vec{F}_{\text{quad}} + T_3 \vec{F}_{\text{cross}} + \vec{A}
\]  

(5.7)

\(\vec{F}_{\text{quad}}\) is a 6 \(\times\) 1 vector containing the quadratic terms of \(\vec{F}\) according to:

\[
\vec{F}_{\text{quad}} = [F_x^2, F_y^2, F_z^2, M_x^2, M_y^2, M_z^2]^T
\]  

(5.8)

and \(\vec{F}_{\text{cross}}\) is a 15 \(\times\) 1 vector containing the cross product terms of \(\vec{F}\) according to:

\[
\vec{F}_{\text{cross}} = \begin{bmatrix} F_x F_y, F_x F_z, F_x M_x, F_x M_y, F_x M_z, & \ldots & F_y M_x, F_y M_y, F_y M_z, & \ldots & M_x M_y, M_x M_z, & \ldots & M_y M_z \end{bmatrix}^T
\]  

(5.9)

Hence, \(T_2\) is another 6 \(\times\) 6 matrix and \(T_3\) is a 6 \(\times\) 15 matrix. It may be clear that determining every term of these matrices is very labour intensive. Each row of \(T_1, T_2\) and \(T_3\) requires a range of at least 3 values for \(\vec{F}_{\text{quad}} \) or \(\vec{F}_{\text{cross}}\). Each calibration run with a single force component yields a row of \(T_1\) as well as a row of \(T_2\), so a full basic calibration according to Equation 5.7 model requires \((6 + 15) \times 3 = 63\) calibration runs.

Calibration

The strain \(\vec{e}\) has the unit \(\frac{mV}{N}\) and \(\vec{F}\) has units \([N]\) and \([Nm]\). Therefore, the units of \(T_1, T_2\) and \(T_3\) are \([mV/N]\), \([mV/N]\) and \([mV/N]\), respectively.

The full determination of all coefficients of the second-order model as described in [13] and [16] is given below. Experience can reveal that a reduced calibration is sufficient to achieve the required accuracy. However, for a new balance design the full calibration should always be performed. This way, good insight in the interactions of the balance is gathered and no significant terms are ignored.

To calibrate the balance, first the same superposition calibration is performed as in the case of the first-order model. However, a second-order polynomial is fitted instead of a linear function which yields the matrices \(T_1\) and \(T_2\). This is illustrated for a simulated calibration of the channel that measures \(F_x\) in Figure 5.2-A. To illustrate visually whether any significant quadratic effects are present, the differences between a first-order fit and the measured data \((\Delta \epsilon_{1st})\) can be plotted as in Figure 5.2-B.

To determine the cross terms, the calibration for one component is repeated with one of the cross components set to its maximum value as well as its minimum value. This is illustrated in Figure 5.2-C, where \(F_y\) is applied as a cross component of the main component \(F_x\) to determine the cross product term \(F_x F_y\). The force \(F_y\) is set to \((F_y)_{\text{max}}\) and \(F_x\) is varied in several steps from \((F_x)_{\text{min}}\) to \((F_x)_{\text{max}}\). This is repeated with \(F_y\) set to \((F_y)_{\text{min}}\). The deviation of these curves from the calibration curves found with single loads is plotted as a function of the cross product \(F_x F_y\). A linear fit then gives the according terms of \(T_3\) (Figure 5.2-D).

Whether all of the terms in \(T_2\) and \(T_3\) are significant depends on the specific balance design. Experience with a certain sensor design can lead to the conclusion that only some of the cross terms are significant and that others can be ignored.

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Figure 5.2: Simulated second-order calibration of the channel that measures $F_x$. A) A first and a second-order polynomial are fitted to the measured points. B) From the difference between the linear fit and the measured points, the significance of the quadratic effects is determined. C) The calibration is repeated with a cross component (here: $F_y$) set to its maximum and its minimum value. D) The deviations caused by the cross component are plotted as a function of the cross product (here: $F_x F_y$). A linear fit has been calculated which yields the value for the matrix $T_3$. 
Data processing

To calculate forces from measured voltages, the calibration equation needs to be inverted. An iterative method, is a safe and accurate way to perform this inversion [10]. The basis for the data processing is Equation 5.7:

\[
\vec{e} = T_1 \vec{F} + T_2 \vec{F}_{\text{quad}} + T_3 \vec{F}_{\text{cross}} + \vec{A}
\]

The first step is to calculate the inverse \( T_1^{-1} \) of the linear effect. When multiplied by \( T_1^{-1} \), Equation 5.7 is rearranged to:

\[
\vec{F} = T_1^{-1} \left( \vec{e} - \vec{A} \right) - T_1^{-1} \left[ T_2 \vec{F}_{\text{quad}} + T_3 \vec{F}_{\text{cross}} \right] \tag{5.10}
\]

If the terms in \( T_2 \) and \( T_3 \) are small compared to \( T_1 \), then

\[
\vec{F}_1 = T_1^{-1} \left( \vec{e} - \vec{A} \right) \tag{5.11}
\]

is a good first-order estimate of \( \vec{F} \). This first estimate is used to get a second, more accurate estimation:

\[
\vec{F}_2 = T_1^{-1} \left( \vec{e} - \vec{A} \right) - T_1^{-1} \left[ T_2 (\vec{F}_1)_{\text{quad}} + T_3 (\vec{F}_1)_{\text{cross}} \right] \tag{5.12}
\]

where, similar to equations 5.9 and 5.8

\[
(\vec{F}_1)_{\text{quad}} = [F_{1,x}^2, F_{1,y}^2, \ldots M_{1,z}^2]^T \tag{5.13}
\]

and

\[
(\vec{F}_1)_{\text{cross}} = [F_{1,x}F_{1,y}, F_{1,x}F_{1,z}, \ldots M_{1,y}M_{1,z}]^T \tag{5.14}
\]

This iterative process then should be repeated according to:

\[
\vec{F}_n = T_1^{-1} \left( \vec{e} - \vec{A} \right) - T_1^{-1} \left[ T_2 (\vec{F}_{n-1})_{\text{quad}} + T_3 (\vec{F}_{n-1})_{\text{cross}} \right] \tag{5.15}
\]

until the difference between consecutive outcomes is smaller than a certain predefined value. Depending on the magnitude of the second-order terms, 2-5 iterations are generally sufficient.

5.5 Application at MARIN

The following sections discuss how the presented theory should be applied within MARIN.

First or second-order model?

Calibration of force sensors is time consuming and has to be repeated on a regular basis. Sensor calibration is an expense on the budget of commercial projects. Therefore, the time that can be spent on calibration is typically limited to 4 hours. The standard calibration procedures at MARIN are based on a linear model. Using a higher-order calibration model implies a longer – on the order of 12-16 hours – and thus more expensive calibration routine.
On the other hand, an extended calibration generally yields a lower uncertainty. This can be expected to reduce the time spent on data analysis and repeated measurements. Furthermore, a more thorough calibration yields better understanding of the sensor behaviour, which may lead to less or simpler re-calibrations over time. Finally, with higher-accuracy measurements, MARIN potentially has a better product to sell, which is an opportunity for increased revenues. So, depending on the case, higher initial costs through extended and thorough calibration are likely to be justified through decreased operational costs and increased revenues.

For each newly designed sensor, it should be determined whether it shows significant second-order behaviour. Once the second-order effects are determined, five options exist:

1. only sensor designs that behave sufficiently linear are used;
2. sensors that show second-order effects are calibrated according to the described second-order calibration process;
3. sensors that show second-order effects are calibrated with a linear model and the resulting reduced accuracy is accepted;
4. sensors that show second-order effects are initially calibrated according to the described second-order calibration process. During periodical recalibrations, only $T_1$ is determined. The second-order effects measured in the initial calibration are assumed to be constant.
5. sensors that show second-order effects are calibrated with a linear model for a small range around a certain expected working point. This implies that a calibration is only valid for a certain type of experiment where the working point is approximately the same.

The best alternative would be option (1). However, due to geometrical constraints as well as complex interactions in multi-component sensor, it is not always possible to design fully linear multi-component sensors. For sensors that show significant second-order effects, a trade off then has to be made between accuracy (option (2)) and cost effectiveness (option (3)). This is a management decision which highly depends on the specific application. For the current project, accuracy (option (2)) is used as a guideline for further design efforts.

Options (4) and (5) are both trade offs, where a slightly reduced accuracy is accepted. Both options are only possible if the second-order effects are sufficiently small compared to the linear terms. For each new sensor design, it should be tested beforehand to which extent a linear treatment is acceptable.

**Consequences of a second-order model**

When a linear calibration model is used, the sensor offset can be ignored; at the start of an experiment, the signal conditioners are adjusted to make the conditioner output zero (zeroing). Only the slope of the calibration curves is used in the data processing. The sensor has no absolute zero.

When a second-order model is used, the zero situation is fully determined by the zero-load condition in the calibration set-up. This is the absolute zero of the sensor. The measured sensor offset at zero-load is included in the calibration curve. The sensor readings at the start of an experiment cannot be ignored when a second-order calibration is used. Instead of zeroing, the sensor outputs should be recorded at the beginning of an experiment during
a zero-load measurement. After processing the data, these zero-load values can be used to correct the measured values.

The field of wind tunnel testing for aerospace research has published many papers on dealing with second-order calibrations. Many of the references in this chapter are from that area. How to determine the absolute zero of a force balance and how to deal with measured offsets at the start of an experiment is described in detail in [6, 51, 58, 54, 18, 20].

The assumption that zeroing of force sensors is possible is fully incorporated in the design of the measurement software and in the work routine for the MARIN operators. So, if zeroing of a force sensor is no longer possible, this will imply radical modifications in data acquisition software, calibration system and operational routines. These modifications are expensive and time consuming.

Finally, when zeroing is not possible, the sensor becomes sensitive to offsets caused by, e.g., temperature changes. Therefore, temperature and other external factors that can cause zero shifts should be compensated for in the sensor design or in the data processing afterwards.

**Trade off: semi first-order model**

As stated in the previous paragraph, a second-order calibration model causes many problems and should thus be avoided. One way to avoid a second-order calibration model is to use the semi first-order approach proposed in the following.

The second-order effects of most force sensors are symmetrical around zero. Therefore, a trade off between a first- and a second-order model is possible, when it is assumed that $T_3 = 0$ in Equation 5.7. The quadratic curve is approached by two linear curves, one for $\bar{F} > 0$ and one for $\bar{F} < 0$. Thus, $T_2 = 0$ and $T_1$ becomes either $T_1^+ = T_1^+$ or $T_1^- = T_1^-$. Hence, a discontinuity at zero load is created. This linear approach makes zeroing possible again, so $\tilde{A} = 0$. With these modifications, 5.7 becomes:

\[
\tilde{c} = T_1^+ \bar{F} \quad \text{for} \quad \bar{F} > 0
\]
\[
\tilde{c} = T_1^- \bar{F} \quad \text{for} \quad \bar{F} < 0
\]

(5.16)

To invert this model, the sign of $\bar{F}$ has to be known beforehand. Further, the elements of $\bar{F}$ do not necessarily have the same sign. Therefore, a direct matrix inversion is not possible and an iterative method has to be used, hence the following approach is proposed:

First, the cross coupling is ignored and only positive forces are assumed. This leads to a first-order approximation according to: $T_1^+$ and $T_1^-$ are small:

\[
\bar{F}_1 = (e \cdot T_1^+)^{-1} \tilde{c}
\]

(5.17)

where $e$ is the $6 \times 6$ identity matrix. Then, a new calibration matrix $T_1'$ is compiled according to:

\[
T_1'(k, n) = T_1^+(k, n) \quad \text{for} \quad k = 1 \ldots 6, \quad \bar{F}_1(n) > 0
\]
\[
T_1'(k, n) = T_1^-(k, n) \quad \text{for} \quad k = 1 \ldots 6, \quad \bar{F}_1(n) < 0
\]

(5.18)

Now, $T_1'$ can be inverted and $\bar{F}$ can be calculated according to 5.5 and 5.6.
5. SIX-COMPONENT FORCE MEASUREMENTS

This semi first-order approach fits well into the operational routines of MARIN. Therefore, for each balance design where second-order effects are significant and cross terms can be neglected, this is a useful approach. It should be kept in mind that the discontinuity at zero load, may yield problems when small loads occur which vary around zero. However, it is necessary to check for each balance design, through a full second-order calibration, whether the proposed semi first-order approach is a sufficiently accurate approximation.

2-5 component load sensors

To what extent is the above theory also valid for multi-component sensors which measure 2, 3, 4 and 5 components (2-5C)? The same calibration procedure as described above should be carried out for 2-5C balances. For the measured components, the cross coupling has to be determined. For the components that are not to be measured by the sensor, it should be determined whether these ‘parasitic’ components cause a significant error in the measured components.

If ‘parasitic’ forces cause errors, the sensor should be perfectly mounted in such a position that these ‘parasitic’ components are not present. If ‘parasitic’ forces do not cause errors in the measured components, the ‘parasitic’ forces can be omitted in further calibrations.

5.6 Expressing sensor quality

The general term to indicate the quality of a sensor is its accuracy. The accuracy is a qualitative term to indicate the closeness of agreement between measurement result and true value [8]. What is generally meant when using the term accuracy while discussing sensor quality, is its measurement uncertainty.

The measurement uncertainty \( U_x \) is a measure for the spread in the measured result \( x \), generally expressed as: \( x \pm U_x \). The uncertainty \( U_x \) can be expressed in absolute or relative units. In this way, the measurement uncertainty indicates the boundaries which contain the true value with a certain level of confidence, for instance 95%. So, accuracy is a qualitative term while the uncertainty is the quantified doubt about the result of a measurement [8].

When uncertainties are to be compared, relative units are more convenient. The relative uncertainty can be expressed with respect to maximum sensor output (\%\text{FSO}) or to the actual value (%). Where, \%\text{FSO} stands for a percentage of the full-scale output. Most sensor suppliers will express the uncertainty in \%\text{FSO}, as this yields lower uncertainties which suggest a better product. However, from a users point of view, expressing the uncertainty relative to the actual value (%) is more informative as it shows the importance of the error with respect to the actual measurement.

Therefore, in the remainder of the current document, the uncertainty will be expressed relative to the actual measured value. When using relative units, actual values close to zero cause problems. Therefore, for the lower range of a sensor (0-10 \%\text{FSO}), the uncertainty is expressed in absolute units.

In the specification of the measurement uncertainty of a multi-component force sensor, it should be taken into account that the uncertainty increases with the number of loaded components. The more components are loaded, the more calibration parameters are used to calculate the end result. Therefore, in [48] it is proposed to express the measurement uncertainty of a multi-component force sensor similar to Ewald in [17], which yields:
where $i$ and $n$ are the number of the force component ($1 \ldots 6 \equiv F_x \ldots M_z$). The right-hand part of Equation 5.19 expresses the effect of combined loads on the measurement uncertainty. The dimensionless factor $a$ represents the order of magnitude of the uncertainty, which for high-quality force sensors generally can be set to $a = 0.001$. The dimensionless parameter $b_i$ expresses the uncertainty for a single load component $i$.

The second term between the brackets represents the adverse effect of combined loads on the uncertainty. The more components, the higher this term and the higher the uncertainty. The importance of this effect is expressed by $c_i$ which is generally set to unity ([17]). It is recommended to use Equation 5.19 to express the balance uncertainty, as this expression represents the the physical properties of a six-component load balance. Therefore, the uncertainty specifications given in Section 3.4 should be valid for the most commonly occurring combination of force components.
6 Measuring propulsion forces

Abstract

Based on the recommendations given in [53], a prototype pod-unit balance has been designed. The redesigned balance has been tested statically in MARIN's calibration rig. The measurement uncertainty of the balance could be determined up to the limits of the calibration rig, being about 1.5% for most components. Through the limitations of the calibration rig, it was not possible to achieve the required uncertainties of 0.5% for $F_{x\text{-unit}}$ and 1% for the other components. For applying the balance in a ship model, its resonance frequency of about 80 Hz is a promising improvement with respect to the old set-up with resonance frequencies of about 10-15 Hz.

The redesigned pod-unit balance has also been applied in the towing tank in combination with the prototype B2 propeller drive. No operational problems occurred and it was found feasible to perform accurate pod-unit measurements in combination with dynamic steering. However, the output signal of the pod-unit force measurements is dominated by vibration noise caused by resonance in combination with carriage vibrations.

It seems feasible to achieve dynamic measurements in a ship model, at the first propeller blade harmonic (40 Hz). Further, it can be concluded that the open-water set-up is inherently unsuitable for dynamic measurements. Therefore, it is recommend to perform dynamic measurements in a ship model with minimal connections to the towing tank carriage.

6.1 Problems with six-component load measurements

Recent experience with six-component pod-unit force measurements led to the conclusion that the design and the application of the existing pod-unit balance had to be investigated.

Many publications exist on the design and calibration of six-component load balances (Chapter 5). Also, many multi-component force sensors are commercially available. Therefore, this part of the design is not strictly innovative or fundamentally new. However, locally at MARIN, this problem needed to be solved. Furthermore, no commercially available sensors were found to fit the specific needs of model-scale testing with pods. Therefore, this design problem was included in the project to provide a complete pod model, including the required pod-unit force measurements.

Van Rijsbergen et al. investigated two existing pod-unit balances. Also, the existing routines for calibration and data processing have been reviewed. The following conclusion and recommendations were reported in [53]:

- The stiffness of the balance plates and the steering shaft connection is dominated by the bending stiffness of an aluminium plate of 20 mm. Also, the mounting connections are different between the calibration set-up and the towing tank set-up. Therefore,
the existing balance is highly sensitive to the mounting surface and its orientation with respect to gravity. As a result, errors up of 2-5% were found for combined forces.

- The existing calibration method at MARIN is based on the semi-linear approach presented in Section 5.5. However, linearity is often not checked during calibration and typically very few points are applied to determine the calibration matrix. This calibration method introduces errors up to 3%, hence not being sufficient to achieve the currently demanded accuracies of 0.5-1%.

- The existing data processing routines are based on the iterative approach presented in Section 5.4. However, the single iteration which is currently performed is insufficient to achieve the required accuracies of 0.5-1%.

- The following recommendations were given.
  - The balance should be redesigned with stiffer plates as well as a stiffer clamping of the steering shaft.
  - The connections of the pod-unit balance to the fixed world as well as the clamping of the steering shaft should be made identical between towing tank and calibration set-up.
  - The data processing should be done with direct matrix inversion, for a linear model. Or, for a semi-linear model, with the full iterative approach presented in Section 5.5.

These conclusions stress the need to redesign the unit force balance and to thoroughly review the calibration theory. It should be noted here, that in many pod experiments, only $F_{x\text{-unit}}$ is part of the final measurement result. For the most commonly occurring load combination with pods $(F_{x\text{-unit}} + M_{y\text{-unit}})$, the errors in $F_{x\text{-unit}}$ were generally below 2%, which was the reason that many of the above stated aspects stayed undiscovered for several years.

Additionally, from the test results of prototype B2 [45], it was concluded that the resonance frequency of the existing balance is too low ($<10-15$ Hz). It was recommended to increase the resonance frequency, to facilitate dynamic measurements. The theory review on six-component force measurement has been presented in Chapter 5. The following section describes the redesign and prototype testing of a new six-component pod-unit force balance.

### 6.2 New pod-unit force balance

The conclusions and recommendations from [45] and [48] were the starting point for the redesign of the pod-unit force balance, which is reported in [47]. The main aspects of the design as well as the main test results are presented in the following subsections.

#### Design process

The redesign of the pod-unit balance was aimed at the following goals.

- Increase the resonance frequency of the balance to at least 80 Hz. The demanded measurement frequency is 40 Hz which equals a typical first blade harmonic. When assuming an acceptable amplitude distortion of $\pm 25\%$, which occurs at half the resonance frequency (Section 6.4), a resonance frequency of 80 Hz will be sufficient for most experiments.
6.2 NEW POD-UNIT FORCE BALANCE

- Measurement errors caused by varying orientation and mounting surface should not exceed 0.5%.
- Reduce the error related to the reproducibility of the connections to less than 0.1%. Further, make the connections identical between calibration set-up and towing tank set-up.

The basic layout for the design of the balance is according to the well-known concept as presented in [21] and [47]. This is the same concept as is used for the existing balances. Figure 6.1 shows a schematic overview of the basic principle (see also Figure 2.1). The balance consists of a base plate which is attached to the fixed world and a measuring plate which is connected to the pod by a vertical steering shaft. The relative position of the two plates is determined by six transducers with elastic spherical joints. The transducers are arranged in such a way that they give a unique set of readings for each load combination.

There are three transducers in the $z$-direction, which together measure $F_z$. The moments $M_x$ and $M_y$ can be deduced from the differences in measured force between these three transducers and their relative positions. Similarly, two transducers in the $y$-direction together measure $F_y$ and $M_z$. The force $F_x$ is measured by one transducer in the $x$-direction. This is the basic concept, according to which the new six-component (6C) pod-unit balance has been designed.

The following boundary conditions had to be taken into account:

- for the six transducers, existing high-quality MARIN-made single-component (1C) transducers are used,
- the maximum height of the balance is 120mm,
- the steering shaft diameter should not exceed 63mm,
- the steering shaft length should be at least 500mm for open water tests and 200mm for ship model tests.

During the predesign phase of the pod-unit balance, two straightforward theoretical models have been used. The goal of these modelling efforts was to gain insight in the dynamics of the 6C balance concept. A mass-spring model indicated the main points of improvement. Subsequently, a finite-element model was used to optimise the geometry of the most crucial parts indicated by the mass-spring model [47]. Also, experts on six-component force measurement at the Dutch Aerospace Laboratory [38] have been consulted on the design and application of six-component force balances. From the modelling, the following conclusions could be drawn.

- Most crucial to increase the resonance frequency is to increase the bending stiffness of the steering shaft, the stiffness of the clamping of the steering shaft and the stiffness of the measuring plate.
- For the current basic lay-out of the balance, the maximum resonance frequency that can be expected is about 70 Hz for the open-water set-up and 150 Hz for application inside the ship model.
Figure 6.1: Schematic layout of the basic concept according to the article by Gommers (1976) on force sensor designs [21]. All sensors are single component sensors according to a well-known MARIN design [35]. The three transducers in the z-direction together measure $F_z$. The moments $M_x$ and $M_y$ can be deduced from the differences in measured force between these three transducers and their relative positions. Similarly, two transducers in the y-direction together measure $F_y$ and $M_z$. The force $F_x$ is measured by one transducer in the x-direction.
6.2 New pod-unit force balance

Table 6.1: Measuring ranges of the Prototype F1 of pod-unit force balance.

| \( F_x \) | 800  N |
| \( F_y \) | 1600 N |
| \( F_z \) | 2400 N |
| \( M_x \) | 160  Nm |
| \( M_y \) | 320  Nm |
| \( M_z \) | 400  Nm |

- The choice of material has little influence on the resonance frequency, except for using a carbon fibre steering shaft which typically yields an increase in resonance frequency of a factor 2 [47]. Carbon fibre parts are relatively expensive and cannot be machined within MARIN. Therefore, carbon fibre is not very suitable for the first prototyping phase. However, it can well be applied in a later stage of the design, when the final dimensions of the balance are known.

The modelling results, combined with the boundary conditions led to the design of the first pod-unit balance prototype (Prototype F1) which has been built and tested [47]. Figures 6.2 and 6.3 give an overview drawing of Prototype F1 which has the following important features:

- massive aluminium plates,
- outer dimensions: 250x500x120 mm,
- plate thickness: 42.5 mm,
- local plate thickness around clamping point: 60 mm.

The steering shaft is clamped inside the measuring plate by a stiff clamping unit, which is connected to the frame by a flange that can be either fixed or rotating. The base plate is fixed with three tension-free spherical washers combined with conical seats (DIN6319). This yields a statically determined and reproducible connection to the fixed world [39]. The steering shaft has a thickness of 63x34mm and, at the clamping point, the shaft is machined to an outer diameter of 60mm.

According to the modelling results, a plate thickness of 60 mm would be ideal. However, the maximum total height is 120 mm and some space between the plates is needed for the transducers as well as for fixing tools. Therefore, the maximum possible plate thickness is 42.5 mm. This is lower than the optimum thickness. Therefore, at the place where the thickness is most crucial - around the clamping of the steering shaft - the thickness is locally made 60 mm.

According to the modelling results, for a thickness of 40 mm, a beam structure would perform better than a plate. However, to save cost and time the plates were built from massive aluminium. Standard 1C transducers were used with a full-scale output of 800N. This yields the measuring ranges given in Table 6.1.

Balance calibration

With Prototype F1, an extensive testing program on MARIN’s calibration rig has been performed (Figure 6.4). Forces in the \( x \)- and \( y \)-direction of the balance cannot be applied in the centre of the balance. Therefore, compensation forces are required as shown schematically in
Figure 6.2: 3D overview of pod-unit force balance Prototype F1. The grey plate is the fixed plate, the green plate is the measuring plate. The balance consists of massive aluminium plates with outer dimensions 250x500x120 mm and plate thicknesses of 42.5 mm. Around the clamping point, the local plate thickness is 60 mm. The steering shaft is clamped inside the measuring plate by a stiff clamping unit, which is connected to the frame by a flange that can be either fixed or rotating. The base plate is fixed with three tension-free spherical washers combined with conical seats (DIN6319). This yields a statically determined and reproducible connection to the fixed world [39]. The steering shaft has a thickness of 63x34 mm and at the clamping point, the shaft is machined to an outer diameter of 60 mm.
Figure 6.5. The test series that led to the final measurement uncertainty yielded the following general conclusions on balance calibration [47].

- A linear or semi-linear approach using a single calibration matrix does not describe the balance behaviour accurately and yields uncertainties > 2%.

- The cross-terms in $F_{\text{cross}}$ from Equation 5.7 (Section 5.4) play a significant role in the calibration and should be incorporated achieve uncertainties ≤ 0.5%.

- The balance should be either calibrated under the same pre-tension as in the experimental conditions, or with a tare weight applied.\(^1\)

- If a tare weight is applied, errors introduced by rotating the balance during calibration are < 0.5%.

\(^1\)With a tare weight, the ‘internal’ weight of the balance and calibration body is compensated.
Figure 6.4: The balance prototype during static testing in MARIN’s calibration rig. The calibration body is attached to the frame by the ETP clamp bushing that was replaced later in the design process.

Figure 6.5: Application of pure forces using a compensation force $F_1$, to compensate for the moment $M_y$ caused by $F_2$. If $a_1 = 2a_2$ and $F_2 = 2F_1$, then $F_x = F_1$ and $M_y = 0$.

Typical alignment errors, on the order of $\pm 0.2^\circ$ of the coordinate system of the calibration rig with respect to gravity, cause errors in the calibration matrices which yield measurement errors $\geq 1.1\%$.

Based on these conclusions, the final calibration matrices from Equation 5.7 could be determined as given in [47]. With these calibration matrices, a range of independent check-loads has been applied to determine the measurement uncertainty of the balance prototype.
Table 6.2: Steady-state measurement uncertainty of pod-unit balance Prototype F1, determined on MARIN’s calibration rig. Also shown is the demanded value as well as the uncertainty of the applied forces in the calibration rig.

<table>
<thead>
<tr>
<th>Balance uncertainty</th>
<th>Calibration rig uncertainty</th>
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<tbody>
<tr>
<td></td>
<td>Range: $0 - 10%_{\text{FSO}}$</td>
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<tr>
<td></td>
<td>demanded [N, Nm]</td>
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<tr>
<td>$F_x$-unit</td>
<td>1.0</td>
</tr>
<tr>
<td>$F_y$-unit</td>
<td>1.0</td>
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<tr>
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<tr>
<td>$M_z$-unit</td>
<td>0.1</td>
</tr>
</tbody>
</table>

*\%_{\text{FSO}} = Percentage of the Full-Scale sensor Output; It is used here to indicate the sensor range for which a certain specification is valid.

Measurement uncertainty

A series of check-loads was applied to determine the (static) measurement uncertainty. The measured forces are calculated from the measured strain, using the calibration matrices. From these check-loads, the measurement error was calculated. With the measurement error, the measurement uncertainty $U_F$ was determined for each component.

From the check-loads, $U_F$ was found to contain a bias (systematic) error. However, this bias error was not consistent between subsets of measurements. Therefore, this bias has been included in the measurement uncertainty. Table 6.2 presents the measurement uncertainties that were found for Prototype F1. Also shown is the uncertainty of the applied forces in the calibration rig as reported in [47].

In Table 6.2, it can be seen that the required measurement uncertainty could be achieved for a few components only. However, it can also be seen that the uncertainty of the applied forces in the calibration rig is the same order of magnitude as the required uncertainties as well as the balance uncertainties. Therefore, it could be concluded that with the current calibration rig, no better results can be expected than those presented in Table 6.2. Further, it was found that the uncertainty introduced by the reproducibility of the mechanical connections is an order of magnitude smaller ($<0.1\%$) than the measurement uncertainty found.

It was found that the effect of the cross terms in $\vec{F}_{\text{cross}}$ (Equation 5.7) is significant. However, due to the uncertainties in the applied forces, no reliable and consistent terms of $T_3$ could be determined. As a result, using a second-order model only slightly reduces the uncertainty with respect to a linear model. For $F_{y\text{-unit}}$ and $F_{z\text{-unit}}$, using a second-order model even increases the uncertainty due to incorrect second order terms which are expected to be caused by alignment errors. The decrease in measurement uncertainty is most likely not enough to justify the increased costs related to a second order calibration model.

Resonance frequency

With impact tests, the resonance frequencies of the balance have been determined as given in Table 6.3. The tests yielded two vibration modes, bending in the x- and in the y-direction,
indicated in the table as x-mode and y-mode. Three different steering shaft lengths have been tested, which are typical for the open-water set-up. The results have been extrapolated to a steering shaft length of 200 mm by approximating the steering shaft as a cantilever beam. A steering shaft length of 200 mm is typical for application in a ship model.

Table 6.3: Resonance frequencies found with the impact tests. Different steering shaft lengths were applied. Also, a test has been performed with the balance only, without the pod mounted.

<table>
<thead>
<tr>
<th>Set-up</th>
<th>x-mode [Hz]</th>
<th>y-mode [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steering shaft 600mm</td>
<td>25</td>
<td>18</td>
</tr>
<tr>
<td>Steering shaft 520mm</td>
<td>28</td>
<td>21</td>
</tr>
<tr>
<td>Steering shaft 440mm</td>
<td>33</td>
<td>23</td>
</tr>
<tr>
<td>Steering shaft 200mm (extrapolated)</td>
<td>80 ± 20</td>
<td>65 ± 15</td>
</tr>
<tr>
<td>Without pod</td>
<td>150</td>
<td>300</td>
</tr>
</tbody>
</table>

It can be seen that the lowest resonance frequency is higher than the old set-up (≈10 Hz) but lower than the intended 80 Hz. The extrapolated result for a steering shaft of 200 mm yields a resonance frequency that is about the demanded 80 Hz for x-mode and somewhat lower for y-mode. However, the current prototype balance has not been fully optimised with respect to weight reduction of the steering shaft and the measuring plate. Therefore, it is concluded feasible to achieve the first blade harmonic with the current design, when the pod is mounted inside a ship model.

The resonance frequency of the balance can be improved further, up to up to about 250 Hz for a ship model and 100 Hz for the open water set-up, if the following design steps are made [47]:

- further minimise the weight of the measuring plate and the aluminium pod housing,
- minimise the length of the steering shaft,
- apply a carbon-fibre steering shaft.

**Conclusions**

Although the specifications from Chapter 3.4 are not yet fully satisfied, Prototype F1 is regarded as a successful improvement with respect to the existing situation. The measurement uncertainty of the balance could reliably be determined, up to the limits of the existing calibrations set-up. The connections are reproducible and also identical between calibration set-up and towing tank. The resonance frequency is significantly higher and shows promising results towards the first blade harmonic.

Based on the presented results, it is recommended to discuss internally at MARIN, whether the calibration rig should be improved to achieve lower measurement uncertainties. Or, that the specifications need to be adjusted to the uncertainty of the calibration rig. A third option is to let an external institute like NLR [38], perform the second order calibration for those sensors that require such high accuracy.
6.3 Towing-tank test

Once the balance prototype has been tested statically, the next step on path B is to combine the pod-unit balance prototype with the new drive motor. To this end, the pod model Prototype B2 was combined with the pod-unit balance Prototype F1, which yielded Prototype B3 [46] (Figure 6.6). The test set-up was identical to the set-up shown in Figure 4.17, except that the existing pod-unit balance is replaced by Prototype F1. Further, based on steering servo problems with Prototype B2, a stronger steering servo was chosen with a maximum torque of 25Nm instead of the 5Nm in Prototype B2. The tests were performed in the Depressurised Towing tank, under atmospheric conditions [46].

The goal of the tests was to test operational feasibility and to investigate the data quality and vibration characteristics of the prototype pod model. Therefore, the pod housing, the pod-unit balance and the towing tank carriage were equipped with acceleration sensors, to determine the influence of carriage vibrations on the measured forces.

In practice, active steering during runs could be performed well with Prototype B3. The right-angle transmission of the steering servo exhibited a few degrees of play. During changes of the steering direction, this caused a temporarily inaccurate control of the steering angle, yielding position errors of about $0.5^\circ$. Apart from these instantaneous play-induced inaccuracies, the existing servo motor seemed to control the steering angle with an accuracy of $\leq 0.1^\circ$.

However, the size ($\phi = 70\text{mm}$, $L = 300\text{mm}$) and weight (10 kg) of the steering servo are similar to that of the old external propeller drive motor. So, when using an existing steering servo, nothing has been gained with respect to the weight and space on top of the pod-unit balance. Furthermore, based on a market research is was found that a weight and
measuring propulsion forces

space reduction of a factor 3 is possible when designing a dedicated solution for pod models. Therefore, the options for steering the pod have been investigated and will be discussed further in Section 6.4.

Vibration noise

During the steady-state runs, it was found that at frequencies above 50 Hz, the measured vibration noise levels were lower than with Prototype B2. However, it is uncertain whether this is caused by the differences between the carriages of the Depressurised and the Deep Water towing tank, or that this decrease in noise levels can be attributed to the new pod-unit balance. The vibration noise levels below 50 Hz were still high, especially in $F_{x\text{-unit}}$ and $M_{y\text{-unit}}$, as can be seen in figures 6.7 and 6.8.

In Figure 6.8, the Fourier amplitude spectra of $F_{x\text{-unit}}$, $F_{y\text{-unit}}$, $M_{x\text{-unit}}$ and $M_{y\text{-unit}}$, under the same conditions, but without steering are shown. Besides the motor-induced shaft harmonics, considerable low-frequency white noise is visible between 10 and 50 Hz. Therefore it cannot be removed by straightforward filtering. This noise is caused by a combination of carriage-induced vibrations and resonance of the set-up. The resonance frequency of the complete towing tank set-up was determined with an impact test, which yielded $10-15$ Hz. The resonance frequency of the balance was found to be 28 Hz and 21 Hz for the x-mode and the y-mode respectively (Section 6.2). Hence, resonance of both the open water set-up and the pod-unit balance contribute to the high noise levels in the 10-50 Hz range.

One way to decrease the noise levels in the measured signals is to increase the resonance frequency of the set-up. However, besides resonance of the pod model set-up, also resonances of the complete open-water set-up occur. How to deal with resonance and vibrations in dynamic measurements is discussed further in Section 6.4.

Dynamic effects

With Prototype B3, the opportunity to perform dynamic measurements has been investigated. Examples of dynamic experiments are: finding the peak loads during a crash-stop manoeuvre, measuring broadband cavitation noise or measuring blade passage effects. Figure 6.9 shows an example of dynamic effects that can be investigated with the new set-up. With the old set-up, it was only possible to measure pod-unit forces at fixed steering angles. The current example shows that when these loads are measured dynamically, during steering, significantly higher values are found measured than during steady state.

Shown in Figure 6.9 is the same measured $M_{x\text{-unit}}$ as in Figure 6.7 plotted against steering angle. During the run, the steering angle $\alpha$ varies sinusoidally according to: $\alpha = 20.6[^\circ] + 10[^\circ] \sin(5t)$. It can be seen that there is a significant effect of the steering direction. At identical angle, values for $M_{x\text{-unit}}$ show differences up to 25% between different steering directions. This effect was found to increase with the frequency of the sine-shaped steering pattern [45, 46]. These effects are similar to results found with airfoils in a wind tunnel [11]. The asymmetry in the figure is probably caused by the direction of rotation of the propeller.

Figure 6.9 shows a clear example of the added value of the innovated pod model. The dynamic values can be up to 25% different from the steady state values. These significant dynamic effects could not be measured with the old set-up.

Figure 6.9 looks rather smooth, this is however achieved through ensemble averaging over many instances (about 20). Without ensemble averaging, the figure will be dominated by vibration noise [45, 46]. Ensemble averaging is not always possible. Therefore, to make these
Figure 6.7: Typical time series of pod forces measured during a run with a sinusoidal steering pattern. The steering angle $\alpha$ varies according to: $\alpha = 20.6[^\circ] + 10[^\circ]\sin(5[^\circ]s^{-1}t)$. The forward speed is $V_a = 1.5\text{m/s}$ and the rotation rate is $n = 595\text{rpm}$. The data has been low-pass filtered at 40 Hz and the motor-induced shaft harmonics have been removed by band-stop filtering in the Fourier domain.
6. MEASURING PROPULSION FORCES

Figure 6.8: Fourier amplitude spectra of the same time series as shown in Figure 6.7. No low-pass filtering is applied, nor are the motor-induced shaft harmonics removed. Diamonds with a number indicate shaft harmonics. The dashed line indicates the significance level which is the uncertainty limit of the signal conditioners.
Conclusions

In practice it is feasible to apply the redesigned pod-unit balance Prototype F1 in combination with the direct-drive method of Prototype B2. Active steering in combination with accurate pod-unit measurements is also feasible. However, the signals of the pod-unit force measurements are dominated by vibration noise which is caused by resonance in combination with carriage vibrations.

To achieve high-quality dynamic signals, the resonance frequency of the balance should be maximised. Carriage vibrations can never be avoided. Further, it seems feasible to remove vibration noise from the measured signal through a correction model based on measured accelerations of the set-up. It is recommended to further investigate the feasibility of such a correction method.

6.4 Feasibility of dynamic force measurements

In the process of gaining further insight into the behaviour of podded propulsion, it is expected that MARIN customers will more and more demand dynamic force measurements during manoeuvring and cavitation tests. This area of research is rather new and little experience with
6. Measuring Propulsion Forces

measuring dynamic propulsion forces exists. As a result, the specifications given in Section 3.4 (a effective measurement frequency of 40-120 Hz) should be considered as indicative rather than strictly required. Further, the results with Prototype B3 indicate that even with an improved drive method, considerable vibration noise is present in the signal.

Therefore, in the following, the practical and fundamental limits of dynamic measurements are discussed. Further, noise reduction measures are discussed together with the consequences of dynamic experiments for the experimental set-up. To avoid ambiguity about sampling of dynamic signals, some definitions and assumptions will be given first.

The internal sampling frequency is the speed at which the ADC digitises the signal, at MARIN this is 40 kHz. Before the ADC, the signal passes through an analogue anti-aliasing (AA) filter with a cut-off frequency of 9 kHz.

The external sampling frequency is the frequency at which the data is stored onto disk. After the ADC, the data is digitally AA-filtered at one third of the external sampling frequency and saved to disk at the external sampling frequency.

The effective measured frequency is the frequency up to which the data is undisturbed by filtering, which for the currently used AA-filters is approximately one tenth of the external sampling frequency. Data analysis below this frequency is safe, above this frequency, care should be taken to account for artifacts of the AA-filter. The measurement frequencies specified in Section 3.4 are all effective measured frequencies.

Steering the pod

Crucial for dynamic force measurements during manoeuvring is an accurate steering servo that can be integrated with the pod-unit balance. The existing steering servos at MARIN are strong enough and capable of controlling the steering angle with an accuracy of \( \pm 0.3^\circ \) [60].

However, a control accuracy of \( \pm 0.1^\circ \) is required for accurate manoeuvring predictions. Also, the existing servos at MARIN are relatively large (\( \Omega = 70 \text{mm}, L = 300 \text{ mm} \)) and heavy (10 kg). Furthermore, a chain or geared belt transmission has to be used, which may influence the balance accuracy through the inherent pre-tension of such transmissions. Therefore, a predesign study has done by Van der Meulen under supervision of the author [60].

The result of this study is the recommendation for a hollow shaft steering servo with a harmonic drive transmission as given in Figure 6.10. A harmonic drive is an innovative transmission where high reduction ratios of 1:50-1:100 can be realised with only three moving parts [2]. According to supplier data, the recommended servo can control the steering angle with an accuracy of \( < 0.1^\circ \). It can provide 50 Nm torque at a rotation rate of 240 \( \text{°} / \text{s} \), which is sufficient for Category 2 experiments. Further, it is three times more compact and its mass is 3.2 kg [60, 2].

Further, the steering servo offers the opportunity of a straightforward and tension-free integration with the pod-unit balance. Hence, the pod-unit force measurements will not be adversely influenced when this motor is applied. Finally, if the control accuracy is \( < \pm 0.1^\circ \), this implies that a position feedback signal with an accuracy of at least \( < \pm 0.1^\circ \) is generated by the servo. The servo drives the steering shaft without any transmission. Therefore, this signal can directly be used to measure the steering angle.

Concluding, the recommended steering servo is a great improvement with respect to the
6.4 Feasibility of Dynamic Force Measurements

The first limitation to dynamic force measurements is the resonance frequency of the force sensor with its attached mass. To estimate the effect of resonance, a force sensor with an attached mass can be considered as a Mass-Spring-Damper System (MSDS). Based on a critically damped MSDS, it can be calculated that the amplitude and phase of the dynamic information in the signal is distorted when the frequency approaches the resonance frequency of the sensor. Above the resonance frequency, the sensor response strongly decreases with frequency and measured results cannot be trusted. Table 6.4 shows calculated values [48] for the amplitude and phase distortion of a critically damped MSDS.

In general, the uncertainty of the measured amplitude is allowed to decrease with increasing frequency. At MARIN, no clear specification of this decreasing accuracy could

Table 6.4: Amplitude error and phase shift for different effective measured frequencies. The effective measured frequency is expressed as a fraction of the resonance frequency $f_r$.

<table>
<thead>
<tr>
<th>effective measured frequency $f / f_r$ [-]</th>
<th>amplitude error [%]</th>
<th>phase shift [$^\circ$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.12</td>
<td>0.5</td>
<td>-1.4</td>
</tr>
<tr>
<td>0.14</td>
<td>1</td>
<td>-1.6</td>
</tr>
<tr>
<td>0.17</td>
<td>2</td>
<td>-2.0</td>
</tr>
<tr>
<td>0.24</td>
<td>5</td>
<td>-2.9</td>
</tr>
<tr>
<td>0.32</td>
<td>10</td>
<td>-4.0</td>
</tr>
<tr>
<td>0.46</td>
<td>25</td>
<td>-6.7</td>
</tr>
</tbody>
</table>

existing steering servos at MARIN. Applying this servo will further improve the suitability of the innovated pod model for the dynamic experiments demanded by MARIN customers.

Resonance

Figure 6.10: The pod model steering servo recommended by Van der Meulen [60]
Table 6.5: Theoretical maximum resonance frequencies of pod model force sensors, together with the according effective measured frequency for different amplitude distortions.

<table>
<thead>
<tr>
<th>Sensor type</th>
<th>Set-up</th>
<th>Steering shaft material</th>
<th>Maximum resonance frequency [Hz]</th>
<th>Effective measured frequency [Amplitude distortion 1% [Hz] 5% [Hz] 25% [Hz]]</th>
</tr>
</thead>
<tbody>
<tr>
<td>pod-unit</td>
<td>open-water</td>
<td>steel</td>
<td>70</td>
<td>10 17 32</td>
</tr>
<tr>
<td></td>
<td></td>
<td>carbon fibre</td>
<td>140</td>
<td>20 34 64</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ship model</td>
<td></td>
<td>steel</td>
<td>150</td>
<td>21 36 69</td>
</tr>
<tr>
<td></td>
<td></td>
<td>carbon fibre</td>
<td>300</td>
<td>42 72 138</td>
</tr>
<tr>
<td>propeller-shaft</td>
<td>-</td>
<td>steel</td>
<td>375</td>
<td>53 90 173</td>
</tr>
</tbody>
</table>

be given and the demanded frequencies are based on hydrodynamic questions, no technical feasibility of the measuring set-up was taken into account. Therefore, it should be investigated, to what extent dynamic force measurements on pod models are feasible.

Table 6.5 summarises the theoretic maximum possible resonance frequencies for the pod-unit and the propeller-shaft sensor. The results for the pod-unit balance are based on the mass-spring model and the assumption that using a carbon steering shaft doubles the resonance frequency. The result for the propeller-shaft sensor is based on a straightforward beam calculation assuming a maximum shaft diameter of 20x18 mm, a length of 50 mm and a propeller mass of 4 kg. In practice, the propeller-shaft sensor will be less stiff to allow sufficient sensitivity to $F_{x-\text{prop}}$. Hence, this value is rather hypothetical and mainly indicates the boundary conditions.

The calculated resonance frequencies can be combined with Table 6.4 to determine amplitude distortion at different effective measured frequencies. This has been carried out for different values of amplitude distortion as is also given in Table 6.5. From Table 6.5, the following can be concluded.

- Whether measurements can be performed at the first blade harmonic frequency depends on the allowed amplitude distortion.
- With the open-water pod-unit set-up and a steel steering shaft, the typical first blade harmonic of 40 Hz or higher cannot be achieved.
- If an amplitude distortion of only 1% is allowed, 40 Hz can only be achieved for the propeller-shaft sensor and for the pod-unit with a carbon fibre shaft.
- If an amplitude distortion of 25% is allowed, 40 Hz can be achieved for all options except for the open-water pod-unit sensor with steel steering shaft. For the pod-unit sensor inside the ship body and the propeller shaft even the third blade harmonic (120 Hz) can be achieved when 25% amplitude distortion is allowed.

In judging these estimations, it should be kept in mind that the maximum resonance frequencies presented are theoretically calculated values, which in practice may not be achieved. However, these figures indicate the fundamental resonance limits. For the shaft sensor, optimising the resonance frequency will also mean the sacrifice of sensitivity to gain stiffness, which increase the measurement uncertainty.
6.4 Feasibility of Dynamic Force Measurements

As yet, it is unknown to what extent these resonance frequencies can be approached in the future. Further, it is yet unclear to what extent amplitude distortions are acceptable at higher frequencies. This will only become clear in the future, when more experience with dynamic measurements has been gained. It can be concluded that measuring the higher-order (4th-6th) propeller blade harmonics related to cavitation, as well as broadband noise will be particularly challenging, if not impossible on regular model scales.

Carriage and model vibrations

Dynamic measurements of propulsion forces are rather new at MARIN. Until now, most experiments where propulsion forces are measured require only measurements at steady-state or at low frequencies (< 10 Hz). As a result, the towing tanks and the standard techniques to build ship models do not necessarily provide the optimal boundary conditions for dynamic measurements.

For the open-water set-up this is most apparent. The combination of a long steering shaft and a rigid connection to the towing tank carriage yields strong vibrations in combination with a low resonance frequency. As a result, open-water force measurements inherently contain high vibration noise levels.

Assuming that the towing tank carriage cannot be adjusted, the only way to avoid measured vibrations is to dynamically decouple the open pod-unit balance and the towing tank carriage. This can be realised through a connection with very low stiffness and high damping. However, such a connection will yield geometric errors when a steady force component is present on top of the vibrations. In all towing-tank tests, steady forces are present. Therefore, this is not likely to be a feasible solution.

The remaining option is to correct the measured data afterwards. A possible approach is to measure accelerations of the test set-up as is done during the prototype tests of Prototype B3. A correction model may then be developed that calculates the carriage-induced part of the measured vibrations from the measured accelerations. This carriage-induced part can then be subtracted from the measured signal. With the measurements performed with Prototype B3, it was found that there was a distinct similarity between the measured accelerations of the test set-up and the vibration noise in the force measurements. Therefore, it was concluded that such a correction model is a possible approach to improve the data quality of the measured pod forces. Therefore, it is recommended to further investigate the feasibility of such a correction method.

In view of its inherent limitations, it can be concluded that the open-water set-up is not suitable for dynamic experiments. Therefore, it is recommended to perform dynamic measurements at high frequencies in a ship model with minimal connections to the towing tank carriage. When the ship model is fabricated, special care should be taken to make the aft body sufficiently massive to prevent vibrations of the ship model. Internal vibrations of the ship model are not representative for the hydrodynamics of the ship and should thus be avoided. Chapter 7 describes the final prototype tests with the pod model Prototype B3 mounted inside a ship model.
6.5 Towards measurement of six-component forces at the propeller shaft

Besides the pod-unit forces, propeller-shaft force measurements are also required for Category B and C experiments. In Prototype B1-B3, a standard propeller-shaft sensor is used, which measures $F_{x\text{-prop}}$ and $M_{x\text{-prop}}$. This is a high-quality sensor design with a measurement uncertainty $U_{F_{x\text{-prop}}} = U_{M_{x\text{-prop}}} \leq 0.5\%$ [35]. Based on the result in [49], the resonance frequency of this sensor is estimated to be about 200 Hz.

This sensor design thus meets the requirements and has proven to be reliable. Therefore, for those experiments that only require $M_{x\text{-prop}}$ and $F_{x\text{-prop}}$ the existing sensor can be used and no new design is necessary. However, for those experiments that require measurement of the full six components, no existing shaft sensor is available. Hence, a six-component propeller-shaft force sensor should be developed.

Sensor design

Within the current project, no six-component shaft-sensor has been developed. In [48], several conceptual alternatives have been compared. These alternatives have been discussed internally at MARIN and it has been decided to design a prototype which is an expansion of the existing two-component shaft sensor. As result of limited capacity and expertise and to minimise risks, it has been decided to contract the National Aerospace Laboratory NLR to design a six-component shaft sensor.

Data transfer

The shaft sensor requires bridge excitation as well as transfer of the measured signals to the fixed world. The different options for data transfer from the rotating shaft are reviewed in [48]. This review yielded three main options:

1. slip rings & off-board conditioning,
2. slip rings & on-board conditioning,
3. fiber-optic & on-board conditioning.

‘On-board’ conditioning means that on the sensor body, on small printed circuit boards, the bridge excitation voltage is generated while the bridge output voltage is amplified and converted to a multiplexed digital signal. In this way, less EMI noise will be picked-up and signals from multiple components can be transmitted through a single channel.

Option 1 is a conventional method which is well known, compact and robust. However, EMI noise is picked up easily by the long cables signals. Further, expansion of the number of channels requires replacement of the slip-ring unit which then may become too large.

Option 2 is a new method with respect to the ‘on-board’ data processing. It has the advantages of Option 1 while being less sensitive to noise. Also, it is easier to expand the number channels than with Option 1.

In Option 3 new optical data transfer technology is applied, which is inherently insensitive
to EMI. Also, it is expected to be compact. The main disadvantage is that the technology of fiber optic data transfer is rather new at MARIN which implies considerable design efforts before it is ready to use in pod models.

Space at the shaft is generally very limited. Therefore, the lay-out, connection and placement of the ‘on board’ electronics is critical and should be specialised for a specific sensor design. So, ‘on board’ conditioning is a solution that is in general only possible, when the design of the sensor body is finished.

To decide which data transfer option is best, the stage of the design process as well as the expected long-term developments should be taken into account. During the prototyping phase of the design process, it should be possible to test several alternatives within reasonable time. So, the primary transducer and data transfer should be sufficiently versatile. Finally, the expected future plans for MARIN need to be known to avoid a design that, once finished, does not fit into the MARIN environment. Considering the above, the following conclusions can be drawn.

- For the prototyping phase of the elastic element, Option 1 is the best choice as this option requires the least development time, thus being the most versatile. Furthermore, the technology is well known, which speeds up the design process.

- Once the design of the elastic element with strain gauges is finished, there are two options for the final data transfer:
  - when optical data transport of digital data becomes customary at MARIN, Option 3 should be chosen,
  - when optical data transport will not become the future standard, Option 2 should be chosen.
7 Ship model trial

To ultimately demonstrate the advantages of the innovated pod model over the existing set-up, a final towing-tank test has been conducted with Prototype B3. In this final prototype test, the old and new set-up could be compared under identical test conditions. The main goal of the test was to evaluate the operational feasibility and to compare the vibration characteristics of the old and new set-up.

All previous towing-tank tests with Prototype B3 were performed in the open-water set-up, which has a rigid connection to the towing-tank carriage. In that way, all carriage vibrations were directly measured by the pod model. As a result, the improved vibration characteristics of the new design were difficult to determine. Therefore, the prototype was finally tested inside a ship model with minimal connections to the towing-tank carriage. The experimental set-up will be presented first, followed by a discussion of the initial results and the conclusion.

7.1 Test set-up

Prototype B3 (Chapter 6) was mounted on a ship model (Figure 7.2), together with a pod model that was built using the existing set-up with external drive motor. The ship model was a generic cruise liner model. For the pod shape, a model-scale version of the ABB compact azimuth pod has been used (Figure 7.1). A pair of 4-bladed propellers from MARIN’s stock were used. The steering shaft of the pod was connected to the pod-unit force balance (Prototype F1) as reported in Chapter 6. On top of the force balance, an off-the-shelf steering servo from MARIN was mounted.

Close to each pod model, two acceleration sensors were mounted. The location of the acceleration sensors resembled the location in a typical cavitation experiment. In cavitation experiments, ship model accelerations in the vertical $z$-direction may disturb pressure fluctuation measurements at the ship hull. Therefore, MARIN is developing a method which uses measured vertical accelerations on the aft of the ship body to calculate corrections for the pressure fluctuation measurements. By measuring these vertical accelerations at both pod model set-ups, the disturbance of pressure fluctuation measurements by drive-motor induced vibrations can be compared.

The tests were performed in the Deep Water towing tank. A standard propulsion test was performed first (Figure 7.3), to determine the ideal rotation rate for the combination of the ship model (scale factor $\lambda = 25.33$) and the podded propeller model, which yielded $n = 869\text{rpm}$. The forward speed was derived from the full scale speed of 25 knots, which equals $V_a = 2.55\text{m/s}$ on model scale. Performing the propulsion test exactly according to standard test routines, provided good insight in the operational feasibility.

In a propulsion test, the ship model is propelled by its model-scale propellers which provide the required thrust to balance the resistance of the ship at a certain speed. A very
7. Ship model trial

Figure 7.1: Picture of Prototype B1 built inside an aluminium pod housing with the outside geometry of an ABB compact azipod

small additional towing force is exerted by the carriage through a steel rod which contains a force sensor. This additional towing force is applied to correct for scale effects in the resistance. The ship model can move freely, with minimal connections to the towing tank carriage. Only rotation around the x-axis (roll) is prevented. Also in the propulsion set-up, the resonance characteristics of both pod models have been determined. Finally, in the propulsion set-up, several steady-state runs were carried out while operating only one pod at a time.

To demonstrate the dynamic capabilities of the new set-up, the model was also towed in a captive set-up i.e., fixed to the towing tank carriage. In this set-up, several runs were performed while steering the pod. Because of the fixed connection between the carriage and the ship model, these results contain more vibration noise from the carriage than those obtained with the propulsion set-up. As a result, the measurements can be expected to be very similar to those reported in Section 6.3. With the old set-up, it was not possible to steer the pod. Hence, for these runs no comparison can be made between the old and the new set-up since steering is a completely new functionality of the large-scale pod models. Two types of dynamic runs were performed:

- runs with sinusoidal steering patterns with a small steering-angle amplitude of 10°. These runs simulate for instance autopilot steering in heavy waves;
- runs where zigzag steering patterns are applied with large amplitudes up to 180°. These runs simulate for instance emergency manoeuvres and crash-stops.

7.2 Discussion of the results

Operational feasibility

Regarding the operational feasibility, only minor startup problems were experienced. In the Prototype B3, no seal between the ship body and the steering shaft has been incorporated. The current tests showed that air is entrained (i.e. drawn into the water by the flow) through
7.2 Discussion of the Results

Figure 7.2: Pictures of the pod models mounted inside the ship model ($\lambda = 25.33$). On the left side (Port Side, front in picture), the innovated pod model is mounted, on the right side (Starboard Side, back in picture) the existing set-up with external drive motor is mounted. (a) bottom view (b) top view (c) side view

Figure 7.3: Pictures of the ship model ($\lambda = 25.33$) of a generic cruise liner used for the Deep Water towing tank trials of the old and new pod.
the space between steering shaft and ship hull (Figure 7.8, picture 1). Therefore, for future applications, a seal between the steering shaft and the ship body has to be designed, which prevents air entrainment without disturbing the pod-unit measurements. This is a rather trivial problem which is expected to require only little design effort to be solved.

**Vibrations in force measurements**

Figure 7.4 compares vibrations in the measured loads $M_{x\text{-prop}}$, $F_{x\text{-unit}}$ and $F_{y\text{-unit}}$ between the old and new set-up. It can be seen that for all three forces, the vibrations are much lower in the new set-up – on the order of a factor 5 to 10 – which is a strong improvement with respect to the old set-up. Further, it can be seen that the many of the important vibrations in the new set-up are motor induced shaft harmonics (1st, 5th, 6th, 10th, 15th and 20th), which can be removed from the signal through filtering (Section 4.3 and [49]).

For the propeller torque $M_{x\text{-prop}}$, the only vibrations present in the new set-up are motor induced shaft harmonics (torque ripple). When these motor harmonics are removed by filtering, the resulting spectrum will contain less vibrations than that of the old set-up. The propeller thrust $F_{x\text{-prop}}$ shows similar results. For the pod-unit forces $F_{x\text{-unit}}$ and $F_{y\text{-unit}}$ a strong decrease in vibrations is visible and especially below 100 Hz the advantage of the new set-up is evident. Below the first blade harmonic of about 60 Hz, the standard deviation of the signal is reduced approximately by a factor 15 for $F_{x\text{-unit}}$ and a factor 6 for $F_{y\text{-unit}}$. The results for the other pod-unit force components are comparable to the results found for $F_{x\text{-unit}}$ and $F_{y\text{-unit}}$.

It can also be seen that in $F_{x\text{-unit}}$, measured with the old set-up, the first and second blade harmonic (the 4th and 8th shaft harmonic) are well above the significance level, while in the new set-up, these are clearly below the significance level. Since both set-ups are exposed to identical conditions, it can be concluded that this apparent amplification of the blade harmonics is caused by the propeller-drive system, instead of being solely of hydrodynamic origin. Thus, the old system also introduces frequencies that seem to be of hydrodynamic origin while being caused by the drive system. Obviously, such apparent hydrodynamic results can be very deceptive.

The 'hump' between 20 and 50 Hz in $F_{x\text{-unit}}$ and $F_{y\text{-unit}}$ measured with the new set-up is caused by resonance frequencies of the ship model (see below). Hence these results also indicate that, to further reduce vibration noise, not only the drive motor induced vibrations should be reduced, but also the ship model vibrations have to be reduced.

It should be noted here that the Fourier spectra were calculated in a straightforward way, using a single squared sine window. Such a simple and straightforward calculation is sufficient for the qualitative comparison as performed with the current results. As a result, the calculated amplitudes should be considered as indicative, rather than being solid estimates of absolute values. Also, the indicated significance level is only a reasonable estimate. However, it allows comparison of the calculated amplitudes with the uncertainty of a single-point measurement.

**Disturbance of pressure fluctuation measurements**

Figure 7.5 compares the ship model vibrations up to 75 Hz, using the old and the new set-up. The two vertical acceleration sensors are called $Acc_1$ and $Acc_2$. The location of the

---

1Motor induced harmonics can only be removed by filtering if they do not coincide with hydrodynamic frequencies of interest, which would be the case with a five bladed propeller. (Section 4.3)
Figure 7.4: Fourier amplitude spectra of the measured pod forces $M_{x\text{-prop}}$, $F_{x\text{-unit}}$ and $F_{y\text{-unit}}$, during a steady state run at $V_a = 2.55 \text{ m/s}$ and $n = 869 \text{ rpm}$. The figures on the left are measured with the old set-up, the figures on the right are measured with the new set-up. Diamonds with a number indicate shaft harmonics with amplitudes higher than the significance level. The dashed horizontal line indicates the significance level which is the single point uncertainty limit of the signal conditioners.
acceleration sensors relative to the pod model is identical on both sides of the ship model.

For both acceleration sensors, it can be seen that below 75 Hz, the vibrations are much lower when the new set-up is used. For frequencies above 75 Hz, the differences between both spectra are less clear. Therefore, the total spectral energy has been calculated for frequency bands of 75 Hz, which is compared to Table 7.1.

It can be seen that for 0-75 Hz and 150-225 Hz the advantage of the new set-up is clear. For 75-150 Hz, the new set-up still yields a small improvement. For the 225-300 Hz band, the new set-up shows a somewhat higher spectral energy content than the old set-up. The measured vibrations are not all necessarily caused by the propeller drive system. Resonance of the ship model also play an important role, as was found from the resonance tests.

Concluding, it can said that the new set-up offers distinct advantages with respect to the old set-up. A further decrease in vibrations will also require adaptations to the ship model to improve its resonance characteristics.

Figure 7.5: Fourier amplitude spectra of the measured vertical accelerations $Acc_1$ and $Acc_2$ during a steady state run at $V_a = 2.55 \text{m/s}$ and $n = 869 \text{rpm}$. The figures on the left are measured with the old set-up, the figures on the right are measured with the new set-up. Diamonds with a number indicate shaft harmonics. The dashed horizontal line indicates the significance level which is the uncertainty limit of the signal conditioners.
Table 7.1: Spectral energy content in bands of 75 Hz. The accelerations are measured at two typical locations ($Acc_1$ and $Acc_2$), using the old and the new pod model.

<table>
<thead>
<tr>
<th>Frequency band [Hz]</th>
<th>$Acc_1$ $10^{-3}$[m$^2$/s$^3$] Old set-up</th>
<th>$Acc_1$ $10^{-3}$[m$^2$/s$^3$] New set-up</th>
<th>$Acc_2$ $10^{-3}$[m$^2$/s$^3$] Old set-up</th>
<th>$Acc_2$ $10^{-3}$[m$^2$/s$^3$] New set-up</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-75</td>
<td>1.25</td>
<td>0.05</td>
<td>1.33</td>
<td>0.09</td>
</tr>
<tr>
<td>75-150</td>
<td>0.47</td>
<td>0.34</td>
<td>1.09</td>
<td>0.28</td>
</tr>
<tr>
<td>150-225</td>
<td>0.30</td>
<td>0.04</td>
<td>0.41</td>
<td>0.04</td>
</tr>
<tr>
<td>225-300</td>
<td>0.32</td>
<td>0.53</td>
<td>0.20</td>
<td>0.36</td>
</tr>
</tbody>
</table>

**Resonance frequencies**

To determine the resonance frequencies of the both the old and new test set-up, impact tests were performed, while measuring pod forces and accelerations. With the ship floating freely in the water, each pod pod was hit in the side direction as well as in the longitudinal direction, which yielded two different vibration modes. Also, the ship model has been hit in different directions. Because of the complexity of the vibrations of the ship model with the two pod models, the measured results were not very clear. However, by comparing different measurements, the results presented in Table 7.2 could be derived.

Table 7.2: Resonance frequencies of the old and new pod model set-up and of the ship model. The resonance frequencies have been determined from impact tests.

<table>
<thead>
<tr>
<th>Resonance Frequencies [Hz]</th>
<th>x-direction</th>
<th>y-direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Old set-up</td>
<td>15-18</td>
<td>15</td>
</tr>
<tr>
<td>New set-up</td>
<td>88</td>
<td>47</td>
</tr>
<tr>
<td>Ship model</td>
<td>20-50</td>
<td>20-50</td>
</tr>
<tr>
<td></td>
<td>&gt; 250</td>
<td>&gt; 250</td>
</tr>
</tbody>
</table>

It can be seen that the resonance frequency of the new set-up is much higher and roughly corresponds to the extrapolated resonance frequencies discussed in Section 6.2. The resonance frequency in the x-direction is sufficient to measure at a typical first blade harmonic of 40 Hz. However, for the first blade harmonic frequency of the current set-up, which is $\approx 60$ Hz, the resonance frequency in the x-direction should be increased with a factor 1.5. In the y-direction, an increase with a factor 2.5 is required to achieve reliable measurements at a first blade harmonic of 60 Hz.

Resonance of the ship model is very complex as has also been reported in [31]. Hence, the current impact tests only give an impression in which ranges ship model resonance plays an important role. The results indicate that reducing measured vibrations requires an integral approach where the drive method, the force sensors and the ship model are all improved simultaneously.
Examples of dynamic measurements

Finally two types of runs were performed to demonstrate the dynamic capabilities of the new set-up. Figure 7.6 shows the measured results of a run with sinusoidal steering, which are low-pass filtered at 20 Hz, to remove the ship model vibrations. The time trace of the forces can clearly be distinguished. Some remaining vibrations can be seen, which may originate from model resonance, or from vibration of the old pod model set-up. The signals contain less noise than those in Figure 6.7, measured with the open water set-up, This is however mainly caused by the lower cut-off frequency. Through the use of a captive set-up, the vibration noise levels were comparable to the open water set-up. Further, with the measured data, figures similar to Figure 6.9 could be made. These test runs illustrate the capability to measure dynamic pod-unit loads during, for instance, sailing in waves while operating the autopilot.

During the current test it became clear that the chain transmission between the servo and the steering shaft caused small fluctuations of the steering angle. These fluctuations are not measured by the steering-angle sensor since this sensor is located before the transmission. Further, angular play in the transmission caused errors up 0.7°. These results emphasise the need for a dedicated steering servo as proposed in Section 6.4.

Finally, a crash stop manoeuvre has been simulated by applying a 180° zigzag steering pattern. Figures 7.7 and 7.8 present measured forces and underwater pictures of part of this run. Just after 90° steering angle, thrust breakdown occurs through ventilation. It can be seen that the thrust breakdown is clearly visible in the measured forces. Furthermore, these measurements can be matched with underwater video frames.

Concluding, time traces of the pod forces can be monitored in detail during complex manoeuvres. Furthermore, these measurements can be matched with other observations like underwater video. Although it is doubtful whether this crash-stop manoeuvre – 180 degree in 5 s at half the design speed – represents a realistic ship operation, these results clearly demonstrate the new testing opportunities of the innovated pod model.

7.3 Conclusion

The first preliminary analysis – only a couple of days – of the experimental data, obtained during the trial of the new pod model mounted in a ship model, fully confirms the results found (or predicted) in the previous chapters. In the propulsion set-up, the advantages of the innovated pod model with respect to vibrations could clearly be demonstrated. It can be concluded that the design is successful in reducing the vibration caused by the drive method and by resonance of the pod-unit balance. Finally, the enhanced functionality of the innovated pod model with respect to dynamic measurements is clearly illustrated.
Figure 7.6: Measured steering angle and pod forces during a run with sinusoidal steering pattern. The steering angle $\alpha$ varies according to: $\alpha = 15[^\circ] + 10[^\circ] \sin(4.0[^\circ]/\text{s} \cdot t)$. Further, $V_o = 2.55$ m/s and $n = 869$ rpm. The data has been low-pass filtered at 20 Hz and the motor-induced shaft harmonics have been removed by band-stop filtering in the Fourier domain.
Figure 7.7: Measured steering angle and pod forces during an imaginary crash-stop manoeuvre. The steering angle is varied between $0^\circ$ and $180^\circ$. $V_a = 1.275 \text{m/s}$ and $n = 434 \text{rpm}$. The data is low pass filtered at the first blade harmonic of 60 Hz. Indicated in the upper figure are the instants at which the pictures are taken that are presented in Figure 7.8.

![Picture 1; $0^\circ$](image1) ![Picture 2; $90^\circ$](image2) ![Picture 3; $180^\circ$](image3)

Figure 7.8: Frames from the under water video, taken during the same manoeuvre as shown in Figure 7.7. Approximate steering angles are indicated below each sub-figure. At $90^\circ$ the thrust breakdown through ventilation is just starting. At $180^\circ$ full thrust breakdown is clearly visible.
8 Evaluation of the project results

The value of the current design project lies not only in the concrete asset of the new hardware. The knowledge gained in the project also provides an excellent starting point for many further improvements. The following section evaluates the design and indicates these points of improvement. First, the delivered results are compared to the original requirements. After that, Section 8.2 elaborates on the overall uncertainty in a typical Category 2 case, measured with the innovated pod model. Finally, the dynamic measurement uncertainty is discussed briefly.

8.1 Result versus requirements

To assess the quality of the design, the delivered project results are compared against the specifications given in Section 3.4. For each function from Section 3.5, the results are evaluated and the most relevant points of improvement are discussed.

Representing full-scale pod geometry

To be able to correctly represent the full-scale pod geometry, the direct propeller drive motor should be small enough to be built inside a wide variety of pod shapes. To this end, maximum motor dimensions have been defined for each experiment type. Table 8.1 presents a comparison of the delivered and required motor dimensions.

As can be seen from Table 8.1, the required geometry could not be achieved exactly. For both Category 1 and 2 experiments, the delivered motor is about 15% longer than the specified length. For Category 1 experiments, the length of the motor cannot be reduced further for the current motor type. However, for the two completely different pod shapes of prototypes A2 and A3, the increased motor length did not yield any problems. It can thus be concluded that, with

<table>
<thead>
<tr>
<th>Experiment</th>
<th>Parameter</th>
<th>Required [mm]</th>
<th>Delivered [mm]</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Category 1</td>
<td>Length</td>
<td>140</td>
<td>170</td>
<td>Prototype A3</td>
</tr>
<tr>
<td></td>
<td>Diameter</td>
<td>45</td>
<td>45</td>
<td></td>
</tr>
<tr>
<td>Category 2</td>
<td>Length</td>
<td>200</td>
<td>237</td>
<td>Prototype B1, not optimised on</td>
</tr>
<tr>
<td></td>
<td>Diameter</td>
<td>85</td>
<td>87</td>
<td>geometry</td>
</tr>
<tr>
<td>Category 3</td>
<td>Length</td>
<td>250</td>
<td>180</td>
<td>predesign estimate</td>
</tr>
<tr>
<td></td>
<td>Diameter</td>
<td>100</td>
<td>60</td>
<td></td>
</tr>
</tbody>
</table>

Table 8.1: Comparison of delivered and required motor dimensions.
the concept developed in path A, the pod geometry of a wide range of pod shapes can well be represented.

For pod shapes other than those tested, a possibility exists that the enhanced length causes geometric problems. To avoid such problems, a careful match between motor power, propeller type and scale factor has to be made during the preparation phase of a towing-tank experiment. Further, the following improvements of the design for Category 1 experiments will further minimise the risk on geometric problems.

- Design a watertight end cap with a conical shape rather than a straight cylindrical shape. A conical shape matches better with the typical hydrodynamic shapes of a pod.

- Design a control strategy which only requires signals from the built-in Hall elements. In this way, the encoder can be eliminated, which yields a decrease in motor length of at least 25 mm. Currently, the standard interface with the MARIN control infrastructure requires an encoder signal for feedback. Hence, this improvement requires an additional effort to design a dedicated interface between the pod model controls and the MARIN control infrastructure. The corresponding decrease in length is very crucial. Also, the set-up becomes more robust as the encoder is the most vulnerable part of the motor. Therefore, changing the control strategy in this way seems to be worth the effort.

- Find a similar motor type with smaller length: The delivered torque is 15% higher than required. Further, the ultimate torque limit has not yet been determined and is expected to be 25% higher than currently found. Within a certain range of motors of the same type, available torque scales with length. Hence, a motor that is 25% shorter can still be expected to deliver sufficient torque.

The prototype for Category 2 experiments has not yet been fully optimised for compactness. Further, the motor can provide more power than required (Section 4.3). So, a smaller motor with a diameter of about 60 mm can be applied according to the same concept and still deliver the required power. A smaller motor will significantly increase the applicability of the direct-drive concept. Therefore, the size of the prototype for Category 2 experiments is not expected to yield problems in the future. It can thus be concluded that the approach for Category 2 experiments allows a correct representation of the full-scale pod geometry. The main future tasks are:

- design a more compact prototype similar to Figure 4.19,
- apply the concept to a smaller motor with a diameter of about 60 mm.

The predesign results for Category 3 experiments look very promising regarding the size and power. No shaft sensor for such applications has been designed yet. Therefore, it should be kept in mind that the space required for a propeller-shaft sensor is a rough estimate. The main future task for Category 3 experiments is to design a conceptual prototype to test the feasibility of hydraulics in practice.

**Driving the propeller**

The required power could well be achieved for all experiment types, except for a slightly lower maximum rotation rate for Category 3 experiments as is presented in Table 8.2. For Category 2 experiments, the available power can even be made 50% higher than required if the thermal
Table 8.2: Comparison of delivered and required motor power.

<table>
<thead>
<tr>
<th>Experiment</th>
<th>Parameter</th>
<th>Required</th>
<th>Delivered</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Category 1</td>
<td>Torque [Nm]</td>
<td>2</td>
<td>2.3</td>
<td>experimentally tested</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>2.7</td>
<td>extrapolated torque limit</td>
</tr>
<tr>
<td></td>
<td>Rot. rate [rpm]</td>
<td>1500</td>
<td>1500</td>
<td></td>
</tr>
<tr>
<td>Category 2</td>
<td>Torque [Nm]</td>
<td>10</td>
<td>10</td>
<td>continuous</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>17</td>
<td>test run of 60 seconds</td>
</tr>
<tr>
<td></td>
<td>Rot. rate [rpm]</td>
<td>1200</td>
<td>1500</td>
<td></td>
</tr>
<tr>
<td>Category 3</td>
<td>Torque [Nm]</td>
<td>25</td>
<td>25</td>
<td>predesign estimate</td>
</tr>
<tr>
<td></td>
<td>Rot. rate [rpm]</td>
<td>1200</td>
<td>1450</td>
<td>&quot;</td>
</tr>
</tbody>
</table>

Table 8.3: Comparison of delivered and required rotation rate measurement uncertainty.

<table>
<thead>
<tr>
<th>Experiment</th>
<th>Required</th>
<th>Delivered</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Category 1</td>
<td>Steady state [rpm]</td>
<td>1</td>
<td>&lt;1 extrapolated from uncertainty analysis in [50] estimate</td>
</tr>
<tr>
<td></td>
<td>Dynamic [rpm]</td>
<td>10</td>
<td>≈ 10 estimate</td>
</tr>
<tr>
<td>Category 2</td>
<td>Steady state [rpm]</td>
<td>0.1</td>
<td>0.5 based on uncertainty analysis in [50] estimate based on [49]</td>
</tr>
<tr>
<td></td>
<td>Dynamic [rpm]</td>
<td>3</td>
<td>&lt;3</td>
</tr>
</tbody>
</table>

dynamics are taken into account. Given the typical duration of a typical towing-tank test, this is a feasible approach to enhance the available torque.

Based on the results in [42] and [49], it can be concluded that the control of the rotation rate is sufficient for experiments in Category 1 and 2. According to [9], the control accuracy cannot be estimated beforehand, when hydraulics is used for Category 3 experiments.

Measuring rotation rate

Experimentally, the uncertainty of the rotation rate measurements has not been tested very thoroughly. However, in [50] an uncertainty analysis has been performed for steady-state experiments in Category 2. This analysis can be extrapolated towards Category 1 experiments since the measurement principle – a digital encoder signal – is identical. The results are given in Table 8.3. The dynamic measurement uncertainty for Category 2 experiments is derived from the results presented in [49]. For Category 1 experiments, the dynamic uncertainty has been derived from the steady-state uncertainty, also according to [50].

It can be seen that according to the current estimates, the demands are all satisfied except for the steady state uncertainty of Category 2 experiments. In [50], it was shown that this uncertainty can be straightforwardly be decreased by increasing the encoder resolution to about 3000 pulses per revolution. Therefore, it can be concluded that the required uncertainty of the rotation rate measurement can be achieved with the current approach for Category 1 and 2 experiments.
Table 8.4: Comparison of delivered and required measurement uncertainty of the steady state pod-unit forces for Category 2 & 3 experiments.

<table>
<thead>
<tr>
<th>Component</th>
<th>Required</th>
<th>Delivered</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{x\text{-unit}}$ [%]</td>
<td>0.5</td>
<td>1.64</td>
</tr>
<tr>
<td>$F_{y\text{-unit}}$ [%]</td>
<td>1</td>
<td>1.66</td>
</tr>
<tr>
<td>$F_{z\text{-unit}}$ [%]</td>
<td>5</td>
<td>0.83</td>
</tr>
<tr>
<td>$M_{x\text{-unit}}$ [%]</td>
<td>1</td>
<td>1.18</td>
</tr>
<tr>
<td>$M_{y\text{-unit}}$ [%]</td>
<td>1</td>
<td>1.42</td>
</tr>
<tr>
<td>$M_{z\text{-unit}}$ [%]</td>
<td>1</td>
<td>&lt;0.65</td>
</tr>
</tbody>
</table>

Measuring pod-unit forces

In [50], the uncertainty of pod-unit measurements has been analysed. It was found that an additional uncertainty as a result of temperature effects should be combined with the results in Table 6.2. Also, the uncertainty introduced by the calibration rig should be included. This yields the values in Table 8.4.

It can be seen that the only components for which the required accuracy has been achieved are $F_{z\text{-unit}}$ and $M_{z\text{-unit}}$. In [47], it is concluded that the uncertainties in Table 6.2 correspond to the maximum possible accuracy that can be achieved when the balance is calibrated and tested in the existing calibration rig. Regarding the errors of 2-5% with the existing balances, the results in Table 6.2 are a great improvement with respect to the existing situation. Lower uncertainties can only be achieved if the calibration rig is improved to allow better alignment of the applied forces. Therefore, based on these results, it should be discussed whether the calibration rig should be improved, or whether the requirements should be adjusted to the limitations of the calibration set-up.

The resonance frequency of the balance with steering shaft and pod model mounted in the open water set-up is 30 Hz which is not high enough to measure at a typical blade harmonic frequency of 40 Hz. For application in a ship model, the resonance frequency will approximately be 80 Hz, which is almost twice the typical first blade harmonic of 40 Hz. Hence, with the current balance prototype applied in a ship model, the first blade harmonic can be measured with an amplitude distortion of $\leq 25\%$.

By using carbon fibre instead of aluminium for the steering shaft, the resonance frequency of the balance can be increased dramatically up to about 100 Hz for the open water set-up and about 250 Hz for mounting inside the ship model. Hence, investigating the feasibility of carbon fibre as material for the steering shaft is a key step towards reliable dynamic measurements at frequencies well above the first blade harmonic.

Measuring propeller-shaft forces

For those Category 1 experiments where measurement of $M_{x\text{-prop}}$ is required, the motor current can reliably be used to determine the propeller torque, up to 10 Hz, with an uncertainty of 1%.

The existing propeller-shaft sensor was known to be of good quality and for most Category 2 experiments, only $M_{x\text{-prop}}$ and $F_{x\text{-prop}}$ are required. Therefore, for Category 2 experiments, the existing MARIN shaft sensor design has been incorporated (Prototype B1). No separate testing has been done. However, the sensor used is of well-known
design and has successfully been applied for many years, to measure steady-state torque and thrust. Therefore, according to the expert opinion from [35] it can be stated that $U_{F_{\text{prop}}} = U_{M_{\text{prop}}} \leq 0.5\%$. In [50], it was found that the additional uncertainty introduced by temperature effects should be incorporated, which yields an uncertainty increase of 0.64\% for $F_{x \text{-prop}}$ and $M_{x \text{-prop}}$.

It was concluded in [50] that the temperature effects are the main cause for not achieving the required uncertainty. Incorporating these effects through corrections afterwards or using temperature compensation in the sensor design [63] will sufficiently decrease the uncertainty to meet the required 0.5\%.

In the current project, the resonance frequency of the shaft sensor has not been determined. However, based on the measurements in [49], it is estimated that the resonance frequency is about 200 Hz, which is sufficient to measure at a typical first blade harmonic frequency of 40 Hz.

**Steering the pod and measuring steering angle**

To steer the pod and to measure the steering angle, a servo motor has been proposed which is accurate up to 0.1° and three times more compact than the existing servo’s. Furthermore, the proposed servo motor is very well suited for integration with the pod-unit balance. The expected measurement uncertainty is estimated in [50] to be 0.02°, which is much better than the required 0.1°. Hence, the proposed solution for a steering servo is a great improvement with respect to the existing steering servos at MARIN.

**8.2 Uncertainty analysis**

**Steady state uncertainty**

A full uncertainty analysis of steady-state measurements with the innovated pod model has been performed in [50]. The analysis is performed according to the ITTC\(^1\) standards [27, 28, 29, 57], which are based on the ISO standards [25], [26] and on [12]. The overall measurement uncertainty, as estimated in [50] for the dimensionless final result of a typical Category 2 experiment, will be presented in the following. The goal of the uncertainty analysis is to compare the contribution of all error sources to the uncertainty in the final result. To facilitate this comparison, all error sources are expressed in relative units in the current section. The results are calculated for a typical Category 2 case (Table 8.5), which was measured in [46]. Further, the following boundary conditions are considered.

- The pod model is operated in the Deep Water Towing tank (DWT).

- The sensors are all loaded in their 10-100\% FSO range.\(^2\)

- Each measurement is the average of 5 seconds, measured at 100 Hz.

Generally, the deliverable of a model-scale experiment with pods is the response (forces) of the pod to different conditions (rotation rate, speed, steering angle). Therefore, not only the

\(^1\)ITTC stands for International Towing Tank Conference, which is a voluntary association of worldwide organisations that have responsibility for the prediction of hydrodynamic performance of ships and marine installations based on the results of physical and numerical modelling.

\(^2\)%FSO stands for percentage of the full scale output of a sensor.
uncertainty of the measured response is important, also the uncertainty in the measured conditions is essential. Both the response and the applied conditions are expressed as dimensionless (or reduced) quantities. The dimensionless forces and moments \( (K_F, K_M) \) and the dimensionless speed \( (J) \) are defined as follows:

\[
J = \frac{V_a}{Dn}
\]

\[
K_F = \frac{F}{\rho n^2 D^4}
\]

\[
K_M = \frac{M}{\rho n^2 D^5}
\]

where \( \rho \) is the water density, \( D \) the propeller diameter and \( V_a \) the forward speed of the pod model, which generally equals the carriage speed. So, besides the conditions \( n \) and \( \alpha \), the forward speed \( V_a \), the water density \( \rho \) and the propeller diameter \( D \) also play a role in the end result \( J \).

The uncertainty of measured component \( i \) is expressed as \( U_i \). When the final result \( R \) is influenced by \( k \) error sources, the total uncertainty of the final result \( U_R \) consists is calculated as:

\[
U_R = \sqrt{\sum_{i=1}^{i=k} (\theta_i U_i)^2}
\]

where \( \theta_i \) is the sensitivity of the final results for the uncertainty in component \( i \). Thus, \( \theta_i \) represents the error propagation.

For each parameter involved in the measurement uncertainty of \( J, K_{F_{\text{prop}}}, K_{F_{\text{unit}}} \), the following error sources have been considered:

- calibration,
8.2 Uncertainty analysis

- vibrations,
- Electro Magnetic Interference (EMI),
- geometry,
- signal conditioning,
- environmental influences.

The steady state uncertainty will be presented, followed by a discussion on the uncertainty for dynamic measurements. Tables 8.6 - 8.11 present the uncertainty budgets for the most important dimensionless end results: \( J, K_{F_x-prop}, K_{M_x-prop}, K_{F_y-unit}, K_{F_z-unit}, K_{M_z-unit} \).

The contribution of each measured component to the uncertainty in the dimensionless end result i.e., the term \((\theta_i U_i)^2\) from Equation 8.4, is compared in Figure 8.1. For the steering angle \( \alpha \), no uncertainty budget is shown since \( \alpha \) is already dimensionless and its uncertainty is the same as given above: \( U_\alpha = 0.044\% \). It can be seen in Table 8.6 that \( U_j \) is of similar order of magnitude as \( U_\alpha \). The uncertainty in \( J \) and \( \alpha \) is very low compared to that of the dimensionless force components. Hence, the uncertainty in \( \alpha \) and \( J \) is not a crucial bottleneck.

All dimensionless force components show similar results: even though the dimensionless sensitivity to \( n \) and especially \( D \) is high, the final result is mainly determined by the measured force component. Therefore, the uncertainty of the individual force components is crucial. Hence, improving the quality of predictions based on model-scale experiments primarily requires a reduction of the uncertainty in measured force components.

Also shown in tables 8.6 - 8.8 are the uncertainties in \( J, K_{M_x-prop}, \) and \( K_{F_x-prop} \) as published by the ITTC in [28]. This publication is a recommended procedure for uncertainty analysis of open water tests with a propeller only. As an example which accompanies this recommended procedure, the final measurement uncertainty is elaborated for a typical case. It is useful to compare the results of the ITTC example with the current case, to further indicate the main points of improvement. It should be noted that the ITTC case is to be considered as an illustrative example rather than an international standard.

The ITTC uncertainty for \( J \) is significantly higher than the value reported in [50], which is caused by the lower uncertainty in both \( n \) of the pod model and \( V_a \) of the Deep Water Towing tank. The uncertainty in \( K_{M_x-prop} \) and \( K_{F_x-prop} \) closely matches with the values found in [50].

However, for the force measurements, the ITTC report assumes that the calibration is perfectly reproducible and that the sensor exhibits no cross coupling. In general this is not the case and the influence of cross coupling, temperature effects, creep and hysteresis [23, 63] need to be taken into account. This can best be done by performing a series of independent check-loads through which these error sources can be taken into account in the uncertainty. As a result, it can be assumed that the ITTC example under predicts the uncertainty.

Furthermore, the uncertainty in \( K_{M_x-prop} \) and \( K_{F_x-prop} \) can be decreased to 0.5% if the temperature effects are taken into account. This, together with the expected underestimation of the ITTC suggests that the propeller-shaft measurements with the innovated pod model at MARIN are more reliable than the example of a single propeller in open-water conditions, as published by the ITTC.
### Table 8.6: Total uncertainty budget of the reduced forward speed $J$.

<table>
<thead>
<tr>
<th>Error source</th>
<th>Mean value</th>
<th>Sensitivity $\theta_i'$ [-]</th>
<th>Relative uncertainty $U_i'$ [%]</th>
<th>Total contribution $(\theta_i'U_i')^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n$</td>
<td>595</td>
<td>-1.00</td>
<td>0.13</td>
<td>$1.6 \times 10^{-2}$</td>
</tr>
<tr>
<td>$V_a$</td>
<td>1.5</td>
<td>1.00</td>
<td>0.0084</td>
<td>$7.1 \times 10^{-5}$</td>
</tr>
<tr>
<td>$D$</td>
<td>0.246</td>
<td>-1.00</td>
<td>0.040</td>
<td>$1.6 \times 10^{-3}$</td>
</tr>
<tr>
<td>$J$</td>
<td>0.61</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ITTC:</td>
<td></td>
<td>$U_J = 0.13% \triangleq 7.6 \times 10^{-4}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>$U_J = 0.34% \triangleq 2.1 \times 10^{-3}$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Table 8.7: Total uncertainty budget of the reduced propeller torque $K_{M_x-prop}$.

<table>
<thead>
<tr>
<th>Error source</th>
<th>Mean value</th>
<th>Sensitivity $\theta_i'$ [-]</th>
<th>Relative uncertainty $U_i'$ [%]</th>
<th>Total contribution $(\theta_i'U_i')^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n$</td>
<td>595</td>
<td>-2.00</td>
<td>0.13</td>
<td>0.063</td>
</tr>
<tr>
<td>$D$</td>
<td>0.246</td>
<td>-5.00</td>
<td>0.040</td>
<td>0.040</td>
</tr>
<tr>
<td>$\rho$</td>
<td>1000</td>
<td>-1.00</td>
<td>0.0044</td>
<td>$1.9 \times 10^{-5}$</td>
</tr>
<tr>
<td>$M_x-prop$</td>
<td>3.22</td>
<td>1.00</td>
<td>0.64</td>
<td>0.41</td>
</tr>
<tr>
<td>$K_{M_x-prop}$</td>
<td>0.036</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ITTC:</td>
<td></td>
<td>$U_{K_{M_x-prop}} = 0.72% \triangleq 2.6 \times 10^{-4}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>$U_{K_{M_x-prop}} = 0.71% \triangleq 2.6 \times 10^{-4}$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Table 8.8: Total uncertainty budget of the reduced propeller thrust $K_{F_x-prop}$.

<table>
<thead>
<tr>
<th>Error source</th>
<th>Mean value</th>
<th>Sensitivity $\theta_i'$ [-]</th>
<th>Relative uncertainty $U_i'$ [%]</th>
<th>Total contribution $(\theta_i'U_i')^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n$</td>
<td>595</td>
<td>-2.00</td>
<td>0.13</td>
<td>0.063</td>
</tr>
<tr>
<td>$D$</td>
<td>0.246</td>
<td>-4.00</td>
<td>0.040</td>
<td>0.0256</td>
</tr>
<tr>
<td>$\rho$</td>
<td>1000</td>
<td>-1.00</td>
<td>0.0044</td>
<td>$1.9 \times 10^{-5}$</td>
</tr>
<tr>
<td>$F_x-prop$</td>
<td>108.1</td>
<td>1.00</td>
<td>0.64</td>
<td>0.41</td>
</tr>
<tr>
<td>$K_{F_x-prop}$</td>
<td>0.30</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ITTC:</td>
<td></td>
<td>$U_{K_{F_x-prop}} = 0.71% \triangleq 2.1 \times 10^{-3}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>$U_{K_{F_x-prop}} = 0.69% \triangleq 2.1 \times 10^{-3}$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Table 8.9: Total uncertainty budget of the reduced pod-unit thrust $K_{F_x-unit}$.

<table>
<thead>
<tr>
<th>Error source</th>
<th>Mean value</th>
<th>Sensitivity $\theta_i'$ [-]</th>
<th>Relative uncertainty $U_i'$ [%]</th>
<th>Total contribution $(\theta_i'U_i')^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n$</td>
<td>595</td>
<td>-2.00</td>
<td>0.13</td>
<td>0.063</td>
</tr>
<tr>
<td>$D$</td>
<td>0.246</td>
<td>-4.00</td>
<td>0.040</td>
<td>0.0256</td>
</tr>
<tr>
<td>$\rho$</td>
<td>1000</td>
<td>-1.00</td>
<td>0.0044</td>
<td>$1.9 \times 10^{-5}$</td>
</tr>
<tr>
<td>$F_x-unit$</td>
<td>15.1</td>
<td>-1.00</td>
<td>1.64</td>
<td>2.69</td>
</tr>
<tr>
<td>$K_{F_x-unit}$</td>
<td>0.042</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ITTC:</td>
<td></td>
<td>$U_{K_{F_x-unit}} = 1.67% \triangleq 7.0 \times 10^{-4}$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

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### Table 8.10: Total uncertainty budget of the reduced pod-unit lateral force $K_{F_y-unit}$

<table>
<thead>
<tr>
<th>Error source</th>
<th>Mean value</th>
<th>Sensitivity relative uncertainty $U_i'$ [%]</th>
<th>Total contribution $(\theta_i'U_i)^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n$</td>
<td>595</td>
<td>-2.00</td>
<td>0.13</td>
</tr>
<tr>
<td>$D$</td>
<td>0.246</td>
<td>-4.00</td>
<td>0.040</td>
</tr>
<tr>
<td>$\rho$</td>
<td>1000</td>
<td>-1.00</td>
<td>0.0044</td>
</tr>
<tr>
<td>$F_{y-unit}$</td>
<td>131.7</td>
<td>1.00</td>
<td>1.66</td>
</tr>
<tr>
<td>$K_{F_y-unit}$</td>
<td>0.37</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

$U_{K_{F_y-unit}} = 1.69\% \approx 6.3 \times 10^{-5}$

### Table 8.11: Total uncertainty budget of the reduced pod-unit steering moment $K_{M_z-unit}$

<table>
<thead>
<tr>
<th>Error source</th>
<th>Mean value</th>
<th>Sensitivity relative uncertainty $U_i'$ [%]</th>
<th>Total contribution $(\theta_i'U_i)^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n$</td>
<td>595</td>
<td>-2.00</td>
<td>0.13</td>
</tr>
<tr>
<td>$D$</td>
<td>0.246</td>
<td>-5.00</td>
<td>0.040</td>
</tr>
<tr>
<td>$\rho$</td>
<td>1000</td>
<td>-1.00</td>
<td>0.0044</td>
</tr>
<tr>
<td>$M_{z-unit}$</td>
<td>-7.25</td>
<td>1.00</td>
<td>0.65</td>
</tr>
<tr>
<td>$K_{M_z-unit}$</td>
<td>-0.082</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

$U_{K_{M_z-unit}} = 0.72\% \approx 5.9 \times 10^{-4}$
Figure 8.1: Contribution to the uncertainty of the different parameters involved in calculating the final dimensionless quantities.
Dynamic uncertainty

In the previous subsection, the steady-state measurement uncertainty for measurements with the innovated pod model has been analysed. However, dynamic measurements should also be possible. Therefore, in the following, it will be discussed to what extent the steady-state uncertainty derived above can be applied to dynamic measurements.

First, the crucial difference between steady state and dynamic measurements is that for dynamic measurements no or very little averaging is possible. This means that for all normally distributed measurements, the uncertainty strongly increases. When the 500 sample average above is compared with a single-point measurement, the uncertainty increases with a factor of about 20. Accordingly, the required uncertainty for dynamic measurements is generally less stringent than for steady-state measurements and is in the order of 5-25%. One way to increase the number of samples in a dynamic measurement is to apply ensemble averaging of repeated runs under the same conditions as has been done in [45].

Further, when temporal variation of the signal is part of the end result, amplitude distortion through resonance is an important error source. However, as long as the measurement frequency is below 50% of the resonance frequency, the amplitude distortion is smaller than 10% (Section 6.4 and [48]).

Many dynamic experiments are analysed in the spectral domain. Frequencies and amplitudes are calculated from the time series using Fourier transforms. For such spectral results, the uncertainty in frequency and amplitude should be determined. How this uncertainty in amplitude and frequency can be calculated from the original measurement uncertainty highly depends on the type of spectral conversion that is performed. It is beyond the scope of the current project to further discuss such uncertainties.

Dynamic measurements require additional error sources to be taken into account. Vibrations and Electro Magnetic Interference (EMI) do not contribute to the steady-state error. However, these are important error sources in dynamic measurements. Errors through vibration and EMI can partly be eliminated through data correction and design improvements. However, errors through vibrations and EMI can never be avoided completely.

Most dynamic experiments focus on a specific frequency band rather than on the full spectrum up to the measurement frequency. Examples of experiment types are: finding the peak loads during a crash-stop manoeuvre, measuring broadband cavitation noise or measuring blade passage effects. Such dynamic experiments require a dedicated approach focussed on the specific type of experiment to be performed. In that case, the contribution of vibrations and EMI only needs to be determined for the frequency band of interest.

Concluding, it can be stated that the steady-state uncertainty found can be extrapolated to dynamic measurements, as long as the reduced number of samples and the effects of resonances are taken into account. An increased uncertainty has to be accepted and additional error sources need to be taken into account. Each type of dynamic experiment requires a dedicated approach.
9 Concluding remarks

9.1 Innovated pod model

At the start of the project, none of the maritime institutes was known to be capable of building pod models with the motor located inside the pod gondola. The inherent consequences of the existing, external propeller drive hampered model-scale testing on pods. The existing set-up was insufficient to gather the in-depth experimental knowledge on for instance propeller shaft forces and moments and precise powering characteristics of pods. Such knowledge is required to solve current reliability issues with pods related to, for example, early failure of bearing and seals.

At the finish of the current project, a pod model testing toolbox is available, which contains direct propeller drive solutions for the three main types of experiment:

- small-scale free-sailing seakeeping and manoeuvring tests,
- large-scale powering and captive manoeuvring tests,
- large-scale cavitation observations.

These three experiment types together cover over 80 % of all model-scale experiments with pods. A direct propeller drive solves many of the problems associated with the existing devices. The two commercially most relevant tools have been designed and tested successfully. First, for small-scale seakeeping and manoeuvring experiments the available power density of an electric motor with planetary gearhead could be increased with a factor of 4, thus yielding the required shaft power. Second, for large-scale powering experiments, a direct propeller drive with integrated shaft sensor has been constructed, which can deliver up to three times its supplier-rated power, which is 50% higher than required. For both solutions, sufficient control accuracy as well as operational feasibility have been demonstrated.

Besides the propeller drive, also the required force sensors are incorporated in the pod model designs. For pod-unit measurements, the importance of a correct implementation of the underlying theory in calibration routines and data processing has been demonstrated. Also, the six-component, pod-unit force balance has been redesigned, to fit the modern day accuracy requirements. Steady-state accuracies of the pod unit balance have been improved and the limits of the existing calibration set-up have been indicated. Further, the feasibility of dynamic pod-unit force measurements during manoeuvring have been proven. Recommendations are given to further improve the accuracy and the dynamic range of the pod-unit force balance.
9. CONCLUDING REMARKS

With the design of the innovated pod model, the inherent drawbacks of the existing pod model set-up with external drive motor have been eliminated:

- vibrations are reduced,
- innovative pod shapes like ABB’s compact azipod can be modelled correctly,
- reliable pod-unit force measurements are now possible in combination with active steering of the pod, at measurement frequencies up to the first blade harmonic.

In this way the design of an innovated pod model has removed a major obstacle on the way to new model-scale testing opportunities. Existing test services will improve when the current concepts are implemented. This will consolidate MARIN’s leading position in model-scale research on pods. Furthermore, the innovated pod model creates opportunities for new testing products, such as dynamic pod-unit force measurements during manoeuvring or cavitation tests. With the developed tools MARIN is prepared for future developments, thus capable to stay ahead of global competitors.

Furthermore, the innovated pod model provides the maritime community with a range of opportunities to gain deeper knowledge on podded propulsion. This knowledge will help to strengthen the position of pods in the maritime market. In this way, pods can continue to grow offering their benefits to the maritime industry and the global society.

Finally, the relevance of the current design is ultimately confirmed by the order, recently received by MARIN from CRS\textsuperscript{1}. The CRS research group ‘Loads’ has placed an order of over 100k€ to investigate the dynamic loads on pods [14]. The requested testing program requires the full enhanced functionality provided by the innovated pod model. The activities of the CRS can be considered as a precursor indicating the main focus of hydrodynamic ship research in the near future. Hence, this research order ultimately proves the added value of the current design for MARIN.

9.2 Future

The knowledge gained in the project provides an excellent starting point for many further improvements in pod model testing. Therefore, recommendations have been given on implementation of the result and on further improvement of pod model testing. First, the developed propeller drive concepts should be implemented in the operational towing tank processes. This implementation includes making the design suitable for the Depressurised Towing tank.

To achieve the required uncertainties for six-component force sensors, a redesign of the calibration rig is necessary. Therefore, it should be decided whether the calibration rig needs to be improved or whether the requirements should be adjusted to the limitations of the calibration rig. Further, the predesign studies on a hydraulic propeller drive, on data transfer of the propeller-shaft measurements and on a new steering servo, provide the opportunity to expand the model-scale testing tools for pod models. Finally, the major challenge in improving the current design is to further improve the data quality of dynamic force measurements during manoeuvring.

\textsuperscript{1}CRS stands for Cooperative Research Ships. In CRS MARIN has brought together a group of companies with a common interest in non-competitive research. The research tasks carried out are aimed at hydrodynamics and related issues in terms of design and operation.
As a rule, new research tools provide answers to questions that would not have been asked without these tools, which also holds for the innovated pod model. Until now, dynamic force measurements on pods have hardly been possible. As a result, little is known on the specifications for dynamic model-scale measurements on pods. Therefore, evaluating the design with respect to dynamics is rather fuzzy. However, a crucial step forward has been made through eliminating a major source of vibration; the propeller drive. And further, by clearly identifying the remaining vibrations excited by the propeller drive.

The currently shown feasibility limits of dynamic force measurements on pods (Section 6.4) are rather hypothetical. Future experience will prove what is realistically feasible and, more important, what is truly needed. From the presented results it can be concluded that measuring the higher-order propeller blade harmonics related to cavitation, as well as broadband noise will be particularly challenging, if not impossible on regular model scales. Also, ways have to be found to minimise the vibrations that are inherently present in the towing tank set-up, like carriage or ship model vibrations. Or, if vibrations cannot be reduced, correction methods should be developed to remove vibration noise from the measured signals.

Model-scale testing demands become more and more complex. The currently presented questions on dynamic load measurement are a clear example. Therefore, to create innovative model-scale instrumentation should be a continuous activity at MARIN, which is needed to keep up with modern day testing demands. Complex innovative instrumentation requires considerable development budget. However, unique, innovative testing tools also add value to MARIN’s products, leading to increased revenues. These revenues can be invested again in the development of instrumentation. Thus, developing instrumentation is potentially a highly self-supporting activity. For instance, assume that with the innovated pod model, a typical towing tank test of 60k€ can be sold at a 10% higher price. When further assuming that 20 of such tests are performed in one year, the increased revenues yield 120k€, which is sufficient to finance a full man-year of innovation.

Finally, an excellent opportunity to gain further insight in the measurement of dynamic loads on model-scale is the recently initiated cooperation with other maritime research institutes; the Hydro Testing Alliance (HTA\(^2\)). In this cooperation, institutes will together develop tools that are required by the market while at the same time requiring development costs that are too high to be funded by a single institute.

\(^2\)The Hydro Testing Alliance is an EU-funded ‘network of excellence’ (No. 031316, 6th EU-frame work), in which several maritime institutes cooperate in the development of testing tools. The measurement of dynamic loads on pods is one of the research topics agreed upon. Expected start date: September 1st 2007
References


[31] J. Kampman. Ship model vibrations; reducing the influence of vibrations on pressure measurements. Msc graduation report no. CTW.05/TM-5503, University of Twente, University of Twente, Enschede, Netherlands, 2005.


Appendix A

Towing tank descriptions

**Deep Water Towing Tank**

- **Dimensions**: 250 m × 10.5 m, 5.5 m deep
- **Carriage**: Manned, motor-driven, four drive wheels, four pairs of horizontal guide wheels
- **Type of drive system and total power**: Thyristor controlled power supply, 4 × 45 kW
- **Maximum carriage speed**: 9 m/s
- **Other capabilities**: Vertical/horizontal PMM, wind-force dynamometer set-up
- **Instrumentation**: Dynamometers with strain gauge transducers in propelled hub, wind-force dynamometer, 6-component force balance dynamometer, 5-hole pitot tube, laser doppler velocity scanner, underwater photographic and video tape systems, pressure transducers, transducer for wave cut experiments
- **Model size range**: 1.5 - 8.5 m
Test capabilities

- Resistance and self-propulsion tests in calm water
- Open water propelled/ducted propeller tests
- 3-D wake surveys
- Flow observation tests by paint or tufts
- Measurement of hydrodynamic forces and moments on submerged bodies, foils etc
- Unsteady propeller blade force measurements
- Vertical/horizontal planar motion experiments
- Yacht testing
- Longitudinal wave cut experiments
- Current force measurements

Published description


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Depressurised Towing Tank

The Depressurised Towing Tank is a unique research facility for testing cavitation of the propeller(s) operating behind the complete ship model. In addition, the facility is used as a multi purpose model basin for hydrodynamic research related to the resistance and propulsion of ships. Cavitation and hull pressure tests are carried out in depressurised conditions, with the propeller(s) in Froude scaled condition and the model in free surface conditions (free to trim and thus creating the proper propeller inflow).

**Technical data**

Tank dimensions are 240 x 18 x 8 m. The harbour (preparation) area is 26 m long and 4.2 m wide. The instrumentation allows for measuring 40 channels at 5 kHz. The noise measurement system is able to test frequencies of 2 – 80 kHz.

**Carriage**

**Main frame**

The towing carriage with a maximum speed of 8 m/s consists of a main frame spanning the full width of the tank. The carriage includes an atmospheric pressure cabin for special test equipment and personnel, a visitors platform (only to be used under atmospheric conditions) and a subframe.

**Subframe**

The subframe consists of a test frame and an observation module, and can be disconnected from the carriage. The test frame holds all measurement and testing equipment, the observation module carries all equipment related to cavitation observations (camera, stroboscopes and remote controlled positioning frame). The subframe can be transferred to the harbour area for preparation through an air lock system and is finally connected to the towing carriage, where it is ready for testing.

**Air lock system**

The newly designed and relocated air lock system allows a fast transfer of the subframe to and from the harbour area. This way the complete test set-up (subframe, observation module, ship model and instrumentation) can be prepared and tested outside the tank under atmospheric conditions.

**Ambient air pressure**

According to the laws of similarity which apply to cavitation, the ambient air pressure in the tank must be reduced to the inverse of model scale. Through three vacuum pumps it can be lowered to a minimum of 2500-4000 Pa.
Observation Systems

The observation systems (both inside and outside the model) offers more operational flexibility through the remote control of camera position, camera settings (zoom and focus), stroboscope positions and intensity. The camera and stroboscope housings allow for testing at a reduced cavitation number of $c_r 0.4$.

Model size range

Models range from 2 – 12 m in length, up to 4 m beam and a maximum draught of 10 m. Maximum propeller diameter is 0.4 m.

Test capabilities

- Cavitation observation and inception
- Pressure fluctuation measurements
- Radiated noise measurements
- Wakefield measurements
- Resistance and propulsion tests

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Seakeeping and Manoeuvring Basin

Dimensions
170 × 40 × 5 m. The basin is mainly designed for performing seakeeping, manoeuvring and still water tests with models of seagoing ships and structures.

Carriage
The carriage with a maximum speed of 6 m/s runs over the total length of the tank and consists of:
- A mainframe, spanning the full width of the basin;
- A sub frame, with a max. speed of 4 m/s along the mainframe.

The carriage can follow all movements of the model in the horizontal plane. With an extra installed turntable, the system has a rotating arm capability.

Environment
Waves
At two adjacent sides of the basin, segmented wave generators consisting of hinged flaps are installed. Each flap is controlled separately by a driving motor and has a width of 60 cm. The capacity of the wave generator is up to a significant wave height of 0.45 m at a peak period of 2 seconds.

Opposite the wave generator, passive sinkable wave absorbers are installed. The wave generator system is equipped with an active wave reflection compensation feature and higher order wave synthesis techniques.

Wind
Wind can be simulated by an adjustable 10 m wide platform with electrical fans.

Motion Control and Dynamic Tracking
Free running tests are performed such that the model follows an arbitrary pre-defined track (straight or curved) through the basin. The carriage follows the model during this task.

Deviations from the pre-defined track are minimised through a dynamic positioning feedback loop which controls the propulsion units, additional thrusters and steering within a particular control scenario.

Motion control is realised by means of a feedback loop which activates related stabilisation systems (fins, foils, rudders, etc).
Model size range

- Model length of 2 - 8 m for moving objects and up to 10 m for moored objects.
- Floating structures of any kind, size depending on water depth and wave conditions.

Test capabilities

- Seakeeping tests in waves and wind from arbitrary direction.
- Resistance and self propulsion tests in calm water and waves.
- Oscillation (PMM) and rotating arm tests in calm water and waves with a restrained model to determine hydrodynamic coefficients.
- Captive or free sailing manoeuvring tests in calm water and waves.
- Installation and sea transport tests of offshore constructions.
- Tests on moored or fixed objects to determine the motions and loads due to waves and wind.

For more information please contact the department Ships
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Appendix B

References on full-scale pod problems

General references on pods


(B) Anonymous, 2003, Electric ships: the future is electric. Motor-Ship, Nexus Media Communications, Swanley, UK


(E) Terwisga van T., Holtrop J. Kuiper, G., 1999 Propulsor developments Past, Present and Future Propulsion 2000, available at MARIN, Wageningen, Netherlands

(F) Terwisga van T., Quadvlieg, F., Valkhof, H. 2001 Steerable propulsion units: hydrodynamic issues and design consequences. 80th anniversary of Schottel GmbH, available at MARIN, Wageningen, Netherlands

(G) Valkhof H. H., 2001, Podded propulsors, it has all just started. MARIN publication 2002-002

(H) T-POD First International Conference on Technological Advantages in Podded Propulsion. 2004, T-POD Conference proceedings, University of Newcastle, Newcastle, United Kingdom
### Operational problems with existing full-scale pods

(I) Recent headlines from Lloyds List (Publisher: Informa, London, UK), a daily newspaper for the Maritime Industry:

<table>
<thead>
<tr>
<th>Event Description</th>
<th>Date</th>
</tr>
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<tbody>
<tr>
<td>Pod makers sign quality standards agreement</td>
<td>June 10 2004</td>
</tr>
<tr>
<td>Royal Caribbean considers upping pod compensation claim</td>
<td>March 15 2004</td>
</tr>
<tr>
<td>Problematical pods</td>
<td>August 13 2003</td>
</tr>
<tr>
<td>Rolls-Royce call to limit risk of pod damage</td>
<td>March 14 2003</td>
</tr>
<tr>
<td>Queen Mary 2 in latest pod scare</td>
<td>February 24 2003</td>
</tr>
<tr>
<td>Infinity hit by new pod trouble</td>
<td>February 03 2003</td>
</tr>
<tr>
<td>Celebrity cancels cruises as pod problems persist</td>
<td>January 29 2003</td>
</tr>
<tr>
<td>Rolls-Royce Marine gets to root of pod trouble</td>
<td>December 17 2002</td>
</tr>
<tr>
<td>Problem pod replaced on TT-Line ferry</td>
<td>September 03 2002</td>
</tr>
<tr>
<td>New pod problem hits fourth Celebrity ship</td>
<td>May 03 2002</td>
</tr>
<tr>
<td>Third cruise operator hit by Mermaid pod problems</td>
<td>April 12 2002</td>
</tr>
<tr>
<td>Pod delays persist for TT Line ferry</td>
<td>May 08 2001</td>
</tr>
</tbody>
</table>

(J) Recent headlines from Marinelog (www.marinelog.com), a monthly magazine for the Maritime Industry:

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<tr>
<td>Pod problems prompt Celebrity ship shuffle</td>
<td>March 15, 2004</td>
</tr>
<tr>
<td>Celebrity sues over pods</td>
<td>August 8, 2003</td>
</tr>
<tr>
<td>Celebrity cancels Millennium cruise</td>
<td>July 16, 2003</td>
</tr>
<tr>
<td>Infinity problems more complex than first thought</td>
<td>February 5, 2003</td>
</tr>
<tr>
<td>More pod problems for Celebrity</td>
<td>May 2, 2002</td>
</tr>
</tbody>
</table>


(L) Anonymous, 2003, Proliferating pods aim to strengthen appeal; Solving pod problems is part of realising their full potential. Marine Propulsion & Auxiliary Machinery, Riviera Maritime Media, Enfield, UK


(N) T.C. Kontes, C. Th. Kontes, 2004 Experience of festival cruises operating pod driven ships, T-POD Conference proceedings, University of Newcastle, Newcastle, United Kingdom


(P) Heinke, H.J. 2004, Investigations about forces and moments at podded drives. T-POD Conference proceedings, University of Newcastle, Newcastle, United Kingdom
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Finally, when dealing with major tasks in life, like writing a thesis or raising a family, I realise my human helplessness and weakness. Therefore I conclude by quoting ancient and valuable words:

*Soli Deo Gloria*

Gerrit Oosterhuis

(Wageningen, August 12, 2006)
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