Model based control design for a high performant weaving machine: torsional vibration- and backlash compensation using non-linear feedforward control

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Torsional vibration- and backlash compensation using non-linear feedforward control:

Model based control design for a high performant weaving machine

October 2017

Cyrano Vaseur
Torsional vibration- and backlash compensation using non-linear feedforward control
Model based control design for a high performant weaving machine

Eindhoven University of Technology
Stan Ackermans Institute - Automotive/Mechatronic Systems Design

PDEng-Report: 2017/073

The design that is described in this report has been carried out in accordance with the rules of the TU/e Code of Scientific Conduct.

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Date
October 2017
Model based control design for a high performant weaving machine: Torsional vibration- and backlash compensation using non-linear feedforward control.


This study is concerned with relaxing the mechanics of the weaving machine shedding system, i.e., reducing shaft stiffness and increasing gearbox backlash gap, and compensating for the undesired side effects using non-linear feedforward control: i.e., torsional vibration- and backlash compensation. As preliminary research, initial implementation is done on a Small Modular Driveline (SMD). Both tracking error and torque demand are used as performance indicators for the proposed non-linear feedforward control strategies. After relaxing mechanics, the proposed control strategies do improve performance. However, not to the same performance as with tight mechanics. Nevertheless, this is not the main aim here. Instead, this study illustrates the possibility for making a trade-off between relaxed mechanics and performance (tracking and torque demand) in order to achieve improved overall equipment effectiveness.

Keywords: Weaving, torsional vibration, backlash, non-linear feedforward control

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Foreword
At the start of Cyrano’s PDEng project, we showed him the weaving loom set-up and told him: “This is the weaving loom set-up. Its nominal speed is 1200 rpm and it handles torques of 1kN. These mechanisms allow an individual control of the movement of the 5000 yarns in the loom. We would like you to develop a model-based control strategy for this machinery such that their productivity is increased.”
Cyrano kept smiling and took up this multi-disciplinary challenge, in which mechanics and control systems are key. He designed an experimental proof-of-concept set-up and managed to demonstrate that advanced control techniques can contribute significantly to enhance the performance of weaving machines. Thanks to his successful results, we are now able to start an implementation on production machines. We really appreciate all this effort he has done at Flanders Make, and above all his professional and friendly attitude.

Maarten Witters (project manager)
Albert Rosich (project mentor)
October 2017
Preface

This report captures the work conducted in the second (last) year of the Professional Doctorate in Engineering (PDEng) program in Mechatronics Systems Design (MSD) at the Eindhoven University of Technology (TU/e). This project is carried out at Flanders Make (Belgium, Leuven) and is entitled: Torsional vibration- and backlash compensation using non-linear feedforward control: Model based control design for a high performant weaving machine. The program duration is two years and aims for educating highly skilled design engineers for the high-tech system industry. In this program, working with industry is characteristic. In the first year, trainees work in teams on projects for various companies. This project, carried out in the second year, is individual. Throughout both years, the focus is on systems design, architecture and engineering in a multidisciplinary field. Trainees get the opportunity to apply and strengthen both technical and non-technical competencies e.g., project management (section 6).
Acknowledgements

Carrying out this project has been enjoyable. I would like to thank all the people who helped enable this. I would like to start by thanking Peter Heuberger and Ellen van Hoof-Rompen for arranging the initial interviews with Flanders Make. Also for arranging return days where the PDEng trainees could exchange their experiences. I would like to thank Judith Beenakker, Peter Zomer and Frank Jansbeken for their peer coaching, technical reflection and project management sessions respectively as part of the return days. I am thankful to Greet Heylen, Gie Panis and Liesbet Decuyper for making all the necessary arrangements regarding my project with Flanders Make. Gratitude goes to Maarten Witters. He gave me the opportunity for doing this project. Midway the project, he also gave me the opportunity to give a demo of my work to his clients. I also thank him for his critical feedback and steering. Bruno Depraetere, Jonathan Baake and Yvan Beckers helped me at times the test setup broke down. Thank you for that. A thank you goes to Niels Jeurgen. He helped me with the mechanical design and ordering of parts. I am grateful to Bram de Jager, one of my TU/e supervisors. He carefully read through my whole report and gave me valuable feedback. Also during the Project Steering Group Meetings (PSGMs), he gave worthwhile feedback and steering. I would like to thank Herman Bruyninckx, also a TU/e supervisor. I am thankful for his feedback during the PSGMs. I also appreciate his challenging questions and his feedback aimed at helping me improve my critical self-reflection skills. I would like to give thanks to Edward Kikken. His genuine interest in my project and his ability and willingness to think along with me made asking for help so easy. I went many times to him with problems ranging from practical implementations on the setup to debugging code to even high level brainstorming. I cannot recall even one time he was not able to help me. I would like to show my gratitude to Albert Rosich, my project mentor/ company supervisor. Despite his busy schedule and many responsibilities in the company, he showed inspiring professionalism. We rarely missed a weekly meeting. During these meetings, Albert took his time to answer questions and give feedback and steering which ultimately played a significant role in the course and evolution of the project. I also want to thank, the employees of Flanders Make for creating a wonderful atmosphere, also during lunch and outside working hours such as the monthly drinks we had. I am thanking my PDEng colleague and friend Vincent with whom I had many exchanges of experiences, both personal and professional. I learned a lot from our conversations. I thank my mother and sisters for supporting me in my career choices even when it means spending less time with them. Finally, I thank Myriam for loving me throughout both the good and bad times.

Cyrano Vaseur
October 2017
Abstract

The weaving machine shedding system with tight mechanics, i.e., a stiff drive shaft and a minimum backlash gearbox, does not intercept alignment errors with neighboring components. This causes component damage and consequently machine downtime. Relaxing mechanics, i.e., reducing stiffness and increasing gap size, can remedy this, but degrades tracking performance due to the resulting side effects: torsional vibration and backlash. For achieving high overall equipment effectiveness, the current trend is on reducing machine downtime rather than increasing operating speed for instance. This study is therefore concerned with applying relaxed mechanics and compensating for torsional vibration and backlash through improved control, i.e., non-linear feedforward control. As preliminary research, initial implementation is done on a Small Modular Driveline (SMD). Both tracking error and torque demand are used as performance indicators for the proposed non-linear feedforward control strategies. After relaxing mechanics, the proposed control strategies do improve performance. However, not to the same performance as with tight mechanics. Nevertheless, this is not the main aim here. Instead, this study illustrates the possibility for making a trade-off between relaxed mechanics and performance (tracking and torque demand) in order to achieve improved overall equipment effectiveness.
Executive Summary

Goal
This project involves improving the weaving machine, through improvement of its shedding system. This involves relaxing shedding system mechanics, i.e., reducing drive shaft stiffness and increasing backlash gap size in the gearbox. These modifications are desired for intercepting alignment errors and for improved lubrication respectively. In the long term, this can reduce machine maintenance cost. However, these modifications also make the system susceptible to torsional vibration and backlash respectively. If left untreated, this can lead to degraded tracking performance. Therefor a main aim here is to compensate for torsional vibration- and backlash through control, i.e., improving control. As the focus is on proving feasibility, improvements are not applied to the actual machine directly, but instead, to a Small Modular Driveline (SMD). Therefore, the shedding system is mimicked on the SMD.

Approach
For mimicking the shedding system camshaft mechanism, a slider-crank mechanism is designed and implemented on the SMD. The camshaft mechanism causes a disturbance load which is responsible for backlash gap opening and shaft torsional vibration excitation. The slider-crank is chosen over other concepts, e.g., a four bar mechanism because it enables mimicking the disturbance load to a greater extend. For torsional vibration- and backlash compensation, non-linear feedforward control is applied. This technique is preferred over feedback control techniques such as observer based control or adaptive mechanisms. One of the reasons for this is that, with accurate models, non-linear feedforward control surpasses the other techniques w.r.t. tracking performance.

The applied backlash compensation technique is based on quick and smooth gap traversal. At gap opening, i.e., at disturbance load torque zero-crossings, the motor position reference is adjusted with a steep s-curve (second order) correction profile. The s-curve, covering the gap size, is subtracted from, or added to the reference when the disturbance load has positive or negative sign respectively. In order to predict the zero-crossings and the sign of the disturbance load, an extensive inverse model of the slider-crank is developed. In addition to modifying the motor reference, motor inertia feedforward is applied for gap traversal at the zero-crossings, conform to the second order position correction profile. Maximum motor torque is fully available for gap traversal, because at the zero-crossings, the disturbance load torque is zero. Furthermore quick traversal is possible because only the motor inertia is accelerated at gap traversal, i.e., when motor and load are physically decoupled.

To compensate torsional vibration through feedforward, an inverse model is developed for the flexible shaft. To synchronize with the collocated feedback, collocated feedforward is applied which is used to suppress vibration excitations. These involve excitation at motor (collocated) side caused by the reference as well as excitations at the load (non-collocated) side caused by the disturbance load.

When, in the slider-crank mechanism, spring load dominates over inertial load, torsional vibration- and backlash compensation do not interfere with each other. Backlash is compensated at the zero-crossings. This occurs when the effective arm in the mechanism is zero, implying that the equivalent inertia is zero. Therefore, shaft resonance is increased and the shaft is less sensitive to impacts and torsional vibration.

Results
The applied backlash compensation technique proved to work well, i.e., significant error reduction is achieved without significant extra torque demand. The applied technique has its limitations as well. Compensating for a greater gap size is at the expense of proportional increase in torque demand or proportional decrease in the squared speed. The feedforward torsional vibration compensation technique, applied to the collocated side, reduces the tracking error at the respective side significantly. Nevertheless, on the other hand, significant torque demand is requested to keep tracking accurately when the shaft is excited at the anti-resonance. Although the error at the collocated side can be reduced, the relative error (w.r.t. the non-collocated side) is inevitable.
Conclusion
The obtained results show that with improved control, applied to the system with relaxed mechanics, performance is improved. I.e., by applying torsional vibration- and backlash nonlinear feedforward compensation, both the tracking error and the torque demand can be reduced. However w.r.t. the benchmark system (with tight mechanics) performance is still inferior. Nevertheless, the results do illustrate a possible trade-off between on the one hand relaxed mechanics (for reduced maintenance/cost) and on the other hand high performance. This trade-off allows for improvement of the weaving machine in terms of cost and overall equipment effectiveness.

Room for improvement
One main challenge with the applied backlash compensation technique is to obtain accurate prediction of the load zero-crossings. For this, complete and correct models (correct parameter estimation) are required. Also, model pre-conditions need to be met. I.e., the applied models are based on perfect tracking. Especially when moving towards implementation on the actual system, model extension is required. This involves: (i) dealing with sudden parameter changes e.g., heddle spring stiffness, (ii) dealing with gradual parameter changes, e.g., backlash gap increase due to wear and bearing oil viscosity change due to system temperature increase, and (iii) dealing with overlooked phenomena such as heddle friction. Additionally, there is also room for improvement in finding an optimal, smooth and quick, gap traversal profile. In this study, an s-curve, second order position profile, is selected more or less arbitrarily. The main challenge with the applied torsional vibration compensation technique is dealing with (reducing or eliminating) the relative error, i.e., error between collocated and non-collocated side.

Recommendations
System (tracking) performance can be improved through improvements on hardware and control (possibly combined). Recommendations for hardware improvement involve: (i) Reducing shaft flexibility even further to use the relative shaft error to the advantage of shed opening. Then for maximum shed opening this involves operating at system resonance, implying minimum energy input. (ii) Applying a second motor, i.e., at the non-collocated side, in order to reduce the error at non-collocated side. (iii) Applying a torque sensor for monitoring for instance gradual changes in the parameter which can then be updated online. Recommendations for feedforward control include: (i) Increasing model complexity to take into account the sudden parameter changes and the remaining overlooked phenomena. (ii) Optimizing model parameters to improve model correctness. Finally recommendations for feedback control include: (i) Applying a homing procedure at machine startup to reset the system to a predefined initial condition. This improves parameter estimation. (ii) Applying observers to e.g., cover for non-modeled disturbances. (iii) Applying optimal control for finding the smoothest and quickest gap traversal profile. (iv) Applying torque feedback to prevent high contact impact at gap traversal.
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1. Introduction

Flanders Make is the strategic research center for the manufacturing industry to which leading weaving machine manufacturers belong to. For these manufacturers, Flanders Make is concerned with improving the weaving machines (Figure 1) in terms of main key-drivers: i. Cost, ii. Versatility and iii. Effectiveness. This is pursued through the improvement of the weaving machine shedding sub-system (Figure 2). Therefore, this project is concerned with relaxing the mechanics and simultaneously improving the control of this sub-system. In this project, relaxed mechanics and improved control are implemented on the Small Modular Driveline (SMD: Figure 3) instead of directly to the actual shedding system.

1.1 Problem statement

**Main problem:** The weaving machine shedding sub-system with tight mechanics, i.e., a stiff drive shaft and a minimum backlash gearbox, achieves high tracking performance, but does not intercept alignment errors with neighboring components such as support bearings. This causes component damage which increases machine cost. Another undesired effect is the resulting machine downtime for maintenance and repair. This degrades machine effectiveness.

**Project goal:** This project aims at relaxing mechanics, i.e., reducing stiffness and increasing gap size, to remedy machine downtime and cost. Tackling machine downtime is in harmony with the current trend. With the aim at improving equipment effectiveness of weaving machines, nowadays the focus is not on increasing operation speed, but rather on reducing machine downtime which is the main cause for poor effectiveness.

**Resulting problem:** Relaxing mechanics however, degrades tracking performance due to the resulting side effects: torsional vibration and backlash.

**Project goal:** Therefore this project aims at applying torsional vibration and backlash compensation using advanced control techniques to recover performance.

**Resulting problem:** To be expected is that, compared to the system with tight mechanics, performance of the system with relaxed mechanics cannot be fully recovered.

**Main project goal:** Therefore, this study mainly aims at illustrating a possible trade-off between on the one hand relaxed mechanics and on the other hand good performance by applying advanced control, in order to achieve reduced cost and improved overall equipment effectiveness of the weaving machine.

1.2 The weaving machine

The weaving process, a process to produce fabric from yarn, is typically built up out of five sub-processes as numbered in Figure 1. In sub-process one, shedding, (1) warp yarn is pulled open. During the two next processes, (2) a weft yarn is inserted in the opening and (3) beaten up with a reed after the warp is closed and at the same time re-opened for a subsequent weft insertion. Parallel to these processes, (4) warp yarn is let-off while (5) fabric is taken-up.
The weaving process built up out of five sub processes: 1. shedding, 2. insertion, 3. beating, 4. letting-off and 5. Taking-up.

1.3 The shedding system

As the focus is on the shedding system, its current architecture is presented in Figure 2. This project concerns a shedding process executed by a Jacquard shedding system. This process is synchronized with the other sub processes. For this, good tracking control, i.e., accurate tracking, is required. In the bottom part of Figure 2, the hardware configuration is presented. In the top half, the control configuration is shown. These interact through information flow, sensing of the physical motor (angular) position \( \theta_m \) used for feedback control and requesting torque from the motor \( T \) for moving it to the desired position \( \theta_m \):

- **Hardware configuration**: For opening the shed, the motor moves the knives and displaces the harness/heddle springs. The motor is connected to a rigid drive shaft through a gearbox with minimum backlash. The drive shaft is connected to the camshaft mechanisms. Motor rotation is therefore converted to oscillation (translation) of the loads (knives masses and heddle springs).
- **Control configuration**: The control objective here is to track an (angular) position reference \( \theta_m \) of the motor (reduce tracking error \( e_m \)). With a rigid shaft, this is directly related to tracking of the shed opening. For tracking, currently cascaded position-velocity PID feedback control is applied combined with feedforward \( T_{FF} \) for moving the motor inertia \( I_m \), shaft inertias \( I_{rs} \), knives mass \( m_k \) and heddle spring \( k_h \).
1.4 The SMD (Small Modular Driveline)

As mentioned earlier, in this project, improvements are implemented on the SMD. The SMD, Figure 3, is one of the test beds in the lab of Flanders Make. As the name implies the driveline is provided with modular components: inertia discs, rigid and flexible shafts, bearings, connectors, coupling, gears and backlash elements. Furthermore, the set-up consists of a motor, encoders and torque sensors. The setup is typically used to conduct test before implementation on the actual machines, as is also the case in this project.
1.5 Report Outline

The next chapter (2) in this report elaborates on the system requirements. This involves tracing functional requirements regarding hardware and control improvement of the shedding system back to the main key-drivers for improving the weaving machine. In chapter 3, the system architecture for the improved shedding system, to be mimicked on the SMD, is developed based on both functional and non-functional requirements (process requirements). Chapter 4 dives into the development, realization and verification of the selected hardware and control subsystems. Chapter 5 is concerned with the integration of these subsystems and validation of the complete system. Chapter 6 is dedicated to discussing the management of this project. In the next chapter (7) the final conclusions and recommendations are given. Finally, the project retrospective is given in chapter 8.
2. System Requirements

In this chapter the functional and non-functional requirements are discussed, see Figure 4:

- **Functional requirements:** From the key-drivers for improvement of the weaving machine, requirements are derived for the improvement of the shedding system. From this, requirements are derived for the improvement of the mechanics and control of the shedding system.

- **Non-functional requirements:** These include design limitations and process requirements:
  - Design limitations: Improved control and mechanics are to be implemented on the SMD (with limited hardware). This requires mimicking shedding system dynamics, the oscillating load among others, whilst meeting the specifications of the SMD.
  - Process requirements: In order for the design process to run smoothly, several design methodologies are followed: Bottom-up approach, V-model approach and Co-design of hardware and control. This leads to modular and tunable (sub) systems.

2.1 Improving the weaving machine

As the requirement diagram (Figure 4) illustrates, improving the weaving machine in terms of cost, versatility and effectiveness, branches out into more specific requirements. In the following, this decomposition is briefly discussed:

**R1.** Machine operational and acquisition costs can be reduced by increasing energy and resource efficiency. One way is by reducing motor power consumption.

**R2.** Machine versatility can be improved through configurable flexibility. This implies allowing different machine settings for different end products, e.g., fabric or carpet.

**R3.** Machine effectiveness is dependent on machine performance, machine availability and product quality, see e.g., [1]:
  a. Product quality is high when production and startup rejects are low. To reduce production rejects, process defects must be reduced. To reduce startup rejects, startup acceleration needs to be increased.
  b. Machine availability losses are caused by both un-planned and planned maintenance stops. In this project unplanned stops due to repair of damaged hardware in the weaving machine shedding system, e.g., of the gear-box and bearings, are considered non-negligible.
  c. Machine performance is higher when cycle speed is higher and short-stop down-times are lower:
     i. Cycle speed is limited by the weft insertion process, see [2]. However, current developments are not towards improving system speed, but rather reducing short stoppages, caused by yarn breakage. Speed should therefore be optimized according to yarn characteristics, see [3].
     ii. Short stoppages cause for the bulk of total down time, sometimes up to 90% see e.g., [4]. Yarn breakage contribute to the majority of short-stoppages, see [5]. Compared to weft yarn, warp break repairs (walking to machine, finding broken warp, tying broken warp) cause for longer stoppages. For example, in some air jet weaving application, for 13 warp breaks (with a repair time of 1.9 [min] per break) per 120 [min] (13x 1.9 [min] =24.7 [min]) an efficiency loss of (24.7/120)*100%≈20.6% is experienced. This is almost twice as much for 12 weft breaks (with a repair time of 1.3 [min] per break) per 120 [min], see [6]. Many causes exist for warp breakage: poor yarn quality, uncontrolled temperature and humidity, uncontrolled weaving tension and loom speed, see [6]. However according to [7], deleterious fatigue such as repeated extension and relaxation and abrasion (inter rubbing of yarn due to shed change) cause highest load on the yarn.
Figure 4 Requirement diagram
2.2 Improving the shedding system
In this project, the aim is to meet a relevant sub-set of the abovementioned requirement through improvement of the shedding system:

- **R4.** Reduce shedding system hardware maintenance: This supports reduction in overall system hardware maintenance.
- **R5.** Reduce tracking error: Minimum tracking error is desired in order to achieve smooth shedding to prevent warp yarn breakage.
- **R6.** Reduce motor torque demand: This is with the advantage of reducing power consumption.

2.3 Improving the shedding system mechanics and control
The above mentioned requirements are pursued through improvement of mechanics (relaxing mechanics) and control of the shedding system:

- **R7.** Relax mechanics:
  - a. Reduce driveshaft stiffness: With reducing shaft stiffness, the aim is to reduce hardware maintenance. A more flexible shaft, resulting from stiffness reduction, can intercept alignment errors due to its additional flexible degree of freedom. Alignment error interception leads to less damage on neighboring components, such as bearings, and therefore less hardware maintenance.
  - b. Increase gearbox backlash gap: Also here the aim is to reduce hardware maintenance. In general, backlash is considered as an undesired effect, degrading tracking performance. Nevertheless, in some cases a backlash gap is desired, i.e., to allow the gears to mesh without binding and to provide room for a film of lubrication oil between the teeth. This prevents overheating and tooth damage and leads to better preservation of the equipment and therefore less maintenance.

- **R8.** Improve control: The aim here is to reduce tracking error. Modifications in the mechanics impose additional challenges and require control compensation:
  - a. Shaft stiffness reduction can induce undesired torsional vibrations. The aim here is to compensate for this undesired effect through control.
  - b. Backlash gap increase can deteriorate tracking as well. Also here the aim is to compensate for this undesired effect through control.

2.4 Requirement trade-off
With these three main objectives: hardware maintenance (R4), tracking error (R5) and torque demand (R6), a trade-off is to be made. Relaxing mechanics, i.e., reducing shaft stiffness and increasing backlash gap size, reduces maintenance but on the other hand degrades tracking. Shaft flexibility causes vulnerability for torsional vibration and higher backlash degrades tracking as well. To deal with these undesired effects, torsional vibration and backlash, control can be applied for compensation. On the other hand, this is at the expense of motor torque demand. This trade-off is illustrated in Figure 5.
2.5 Design limitation: Implementing improved mechanics and control on Small Modular Driveline (SMD)

This project is concerned with the intermediate step of implementing the improvements on the SMD instead of directly on the shedding system. This has consequences for the hardware implementation:

R9. Meet the SMD specifications: Testing on the SMD requires taking into account its limitations:
   a. Power limitations: For actuation, the SMD is provided with a 3000 [Watt] motor. The motor has a maximum speed of 3000 [rpm], a continuous torque of 10 [Nm] and a peak torque of 30 [Nm].
   b. Control limitations: The motor torque sampling rate is 750 [Hz].
   c. Measurement limitations: The SMD is provided with rotational encoders with a resolution of $0.175\times10^{-4}$ [rad]. Measurements with these encoders are possible for speeds up to 2637 [rpm].
   d. Footprint limitation: the setup has a usable footprint of roughly 1[m] x 2[m] and a height limitation of roughly 0.4[m].
   e. Limited standard components available: The SMD is provided with standard hardware components such as bearings, shafts, inertias, backlash elements and couplings. Standard interfacing diameters are used in-between the components: 5[mm], 24[mm] and 25[mm].

R10. Capture relevant dynamics of the shedding system:
   a. In the shedding system, the driveshaft is amongst the stiffest hardware components.
   b. Main inertia lies with the knives.
   c. Main compliance comes from the heddle springs.
d. Furthermore, translating the knives and heddle springs through the camshaft mechanism causes a non-linear disturbance load on the shaft. This disturbance load has a more or less sinusoidal profile.

2.6 **Design oscillating load mechanism**

R11. To meet the abovementioned design limitations an oscillating load mechanism is designed. As the SMD is limited in hardware components, the non-linear load disturbance cannot be mimicked with the existing hardware. Therefore an oscillating load mechanism is to be designed.

2.7 **Follow specific design methodologies**

In this project certain design methodologies are followed (for both hardware and control design):

R12. Bottom-up approach: The bottom-up approach is preferred in order to gain experience and to detect infeasibility early on.
R13. V-model approach: The bottom-up approach is combined with the V-model approach to ensure system verification and validation, i.e., to make sure the (sub) systems meet the requirements.
R14. Hardware and control co-design: For optimal design, hardware and control co-design is favored.

2.8 **Design modular and tunable (sub-systems)**

System modularity and tunability are desired:

R15. Modularity supports independent sub-system verification as proposed by the V-model approach.
R16. System tunability supports hardware and control co-design. Tuning both hardware and control simultaneously supports achieving a more optimal design.
3. System Architecture

This section treats the architecture of the improved mechanics and control to be implemented on the SMD. The architecture is shown in Figure 6. First a closer look is taken on the hardware configuration. And finally, the control configuration is discussed.

![System Architecture Diagram]

**Figure 6 System Architecture: Bottom: Hardware configuration (mimicked shedding system); Top: Control configuration (PID feedback and model based feedforward)**

### 3.1 Hardware configuration

The improved configuration includes relaxed mechanics (R7), i.e., some backlash $b_g$ and shaft flexibility $k_s$, see Figure 6. As the trade-off analysis in Figure 5 shows, relaxing mechanics (R7) degrades tracking performance, especially at load side. A direct consequence of this is that tracking performance cannot be evaluated by the measured motor position $\theta_{m,s}$ only. Therefore additional measurement at the load side is applied $\theta_{ls}$, see Figure 6. This is purely used for evaluation of the tracking performance and not for control. For evaluating system performance, both the tracking error $e$ at the
load side (non-collocated) and the torque demand $T$ are evaluated for a given set of hardware parameters (backlash gap $b_g$ and shaft flexibility $k_s$).

The hardware configuration to be implemented on the SMD includes a mechanism to mimic the oscillating load by the camshaft of the actual system (R11). For this mechanism, a slider-crank is selected. Shown in Table 1, the slider-crank has been compared to other concepts: camshaft-translating, camshaft-oscillating and four-bar mechanism. Comparison is done w.r.t R9, R10, R14, and R15. The slider-crank exceeds the other concepts w.r.t. this set of requirements. The camshaft solutions have the drawback of less modularity and tunability (R15 & R16) due to their relatively more complex design. The four-bar mechanism is also inferior. This w.r.t mimicking dynamics (R10). Note: Selecting the slider-crank is not based on the requirement of relaxing mechanics (R7) and as such does not imply that.

<table>
<thead>
<tr>
<th>Oscillating load concepts</th>
<th>Mimic dynamics (R10)</th>
<th>Fit in SMD (R9)</th>
<th>Modular &amp; Tunable (R15 &amp; R16)</th>
<th>Selected</th>
</tr>
</thead>
<tbody>
<tr>
<td>Slider-crank</td>
<td>H</td>
<td>H</td>
<td>H</td>
<td>Y</td>
</tr>
<tr>
<td>Camshaft-translating</td>
<td>H</td>
<td>H</td>
<td>L</td>
<td>N</td>
</tr>
<tr>
<td>Camshaft-oscillating</td>
<td>H</td>
<td>H</td>
<td>L</td>
<td>N</td>
</tr>
<tr>
<td>Four-bar</td>
<td>L</td>
<td>H</td>
<td>H</td>
<td>N</td>
</tr>
</tbody>
</table>

Table 1 Oscillating load concepts for mimicking the dynamics of the camshaft in the shedding system and to be implemented on the SMD. In the table, H implies high/positive and L implies low/negative.

### 3.2 Control configuration

Taking a closer look at the improved control configuration (Figure 6), conventional PID collocated position ($\theta_{m,s}$) feedback control is applied similar (but not identical) to the structure in the actual shedding system (Figure 2). Improvement in the control structure involves torsional vibration compensation and backlash compensation, R8.a and R8.b resp. Non-linear active inverting/feedforward control is selected as the compensation technique. As shown in Table 2 the selection of this technique is based on comparison with other techniques w.r.t. modularity (R15) tunability (R16) and requirements derived from R8, improving control:

- R17. Improve control performance
- R18. Improve control robustness
- R19. Reduce modeling complexity
With respect to R15-R19, non-linear feedforward control is chosen over (non-)linear feedback control techniques such as observer based control or adaptive control mechanisms. For compensation of non-linear phenomena (backlash belonging to this group), adaptive control strategies are preferred over robust control techniques. The non-linearities can cause significant model uncertainties which scale up the required robust control gain. This can lead to instabilities. However, adaptive control techniques which switch between inversions of piecewise linearly parameterized nonlinearities are restricted by the actuator bandwidth. Compared to the non-linear feedforward control technique both of these feedback control techniques have better robustness properties due to less sensitivity to noisy signals and model errors. In [8] brief literature review is done regarding this. Nevertheless, with accurate models, non-linear feedforward control strategies have better performance. Next to that, applying this technique allows for modular and tunable control design. Non-linear model based feedforward control is therefore preferred.

<table>
<thead>
<tr>
<th>Compensation technique</th>
<th>Performance (R17)</th>
<th>Robustness (R18)</th>
<th>Modeling Simplicity (R19)</th>
<th>Process requirements (R15 &amp; R16)</th>
<th>Selected</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linear feedback control</td>
<td>L</td>
<td>H</td>
<td>H</td>
<td>L</td>
<td>N</td>
</tr>
<tr>
<td>Non-linear feedback control</td>
<td>L</td>
<td>H</td>
<td>H</td>
<td>L</td>
<td>N</td>
</tr>
<tr>
<td>Non-linear feedforward control</td>
<td>H</td>
<td>L</td>
<td>L</td>
<td>H</td>
<td>Y</td>
</tr>
</tbody>
</table>

Table 2 Compensation techniques considered for torsional vibration compensation and backlash compensation

3.2.1. Backlash compensation

Dither, a non-linear feedback control technique, is often proposed as a simple compensation technique for nonlinearities such as backlash and friction. However, for hydraulic and pneumatic systems and not for electromechanical systems. Although injecting an additional high frequency signal can partially alleviates these non-linear phenomena it has several negative effects. The additional high frequency signal excites higher order dynamics such as motor-load-coupling resonances. Next to that, it causes fatigue and wear and extra power consumption [9]. Therefore, non-linear feedforward control/ inverting control is preferred. To by-pass numerical issues during model inversion, smoothening the inverted non-linear models can be applied. Such applications can be found in studies on backlash compensation for control of cable actuated devices in [10] and [11]. The backlash compensation technique applied in this project is inspired from those studies.

As indicated in Figure 8, backlash compensation applied here is based on quick/instant gap traversal at backlash gap opening. This is a combination of motor reference $\theta_m$ construction and motor inertia feedforward $T_{FF,m}$. The motor reference $\theta_m$ is constructed from the load reference $\theta_l$ by adding a correction profile $\theta_b$, see equation 1. The correction profile $\theta_b$ ensures instant gap traversal based on expected gap opening caused by the known load disturbance $T_{d}$. With derived physical models of the oscillating load, the torque disturbance $T_{d}$ on the shaft can be calculated from the given acceleration reference $\alpha_l$. Motor inertia feedforward $T_{FF,m}$ is used to support gap traversal by accelerating the physically decoupled motor to close the gap. The acceleration profile $\alpha_b$ corresponds with the position backlash correction profile $\theta_b$.

$$\theta_m = \theta_l + \theta_b$$  (1)
3.2.2. Torsional vibration compensation

Amongst the various feedforward compensation techniques such as inertia and snap feedforward, for higher performance, model inversion feedforward is considered the better option. Both inertia and flexible modes are considered, e.g., as applied in [12]. As also shown in Figure 8, torsional vibration compensation is achieved by applying flexible shaft feedforward torque $T_{FF,fs}$ calculated to counteract torsional vibration of the shaft caused by accelerations in the reference $\alpha_l$ and load disturbances $T_d$ on the shaft.

![Figure 8 Approach for backlash- and torsional vibration compensation](image_url)
4.(Sub)System design, realization and verification

This chapter elaborates on the individual blocks/subsystems of the architecture for the system implemented on the SMD, Figure 6. First the design of the hardware subsystems is discussed. Thereafter the design of the blocks in the control configuration is covered.

4.1 Hardware subsystems

In this section, the design, realization and verification of the flexible shaft, the backlash element and the slider crank is discussed. The design is based on the requirements as presented in Figure 4.

4.1.1. Flexible shaft selection

Looking at the shedding system dynamics, currently the driveshaft is stiff. The improved system suggest introducing some flexibility in the shaft, R7.a. Nevertheless, even after introducing the flexibility, the driveshaft will remain among the stiffest hardware component, R10.a. The selected shaft flexibility will serve as a reference for other compliances in the system. Shaft selection is based on available components in the SMD. Hence, a solid stainless steel shaft ($G = 77$[GPa], density $\rho = 8 \cdot 10^3$[kg/m$^3$]) with radius $r = 0.005$[m] and length $l = 0.3$[m] is selected. Then, for the shaft torsional stiffness applies:

\[
{k_s} = \frac{GJ}{l} \quad \text{with} \quad J = 0.5Ar^2 \quad \text{and} \quad A = \pi r^2
\]

From this follows that $k_s = 250$ [Nm/rad].

4.1.2. Backlash element selection

This section is concerned with the selection of the backlash gap size, R7.b. This is determined by the SMD limitations. Gap size is to be large enough so it is well measurable given the encoder resolutions ($res = 0.001$[°], R9.c). Therefore, the backlash gap size is chosen large enough:

\[
bg > 0.1[°]
\]

On the other hand, gap size is limited based on the available motor power (R9.a) for gap traversal. The larger the gap size, the more torque is required for moving the motor inertia ($I_m = 0.0034$[kg/m$^2$]) in order to achieve instant/quick gap traversal. This is explained in full detail in section 4.2.2, Eq.(27):

\[
bg \leq \frac{\pi^2T_{max}}{450\theta^2l_m}
\]

4.1.3. Slider-crank design

This section treats the derivation of the slider-crank specifications (see Figure 9) from the requirements. These involve capturing the dynamics of the oscillating load in the actual shedding system (R10.d) and following the limitations set by the SMD (R9). After the specifications have been set, the CAD design and realization are presented, followed by a verification of the design.
Mimic dynamics
To mimic the dynamics of the oscillating load, the heddle springs are captured through the tension spring with stiffness $k_t$ and the knives mass is captured through the slider mass $m_s$, see Figure 9. As this is the dominant mass (R10.b), the remaining link-masses $m_c$ and $m_r$ are specified well below $m_s$:

\[ m_c, m_r < 0.05 m_s \]

The geometry of the slider-crank, link lengths $l_c$ and $l_r$ and height offset $h$, is specified based on obtaining a more-or-less sinusoidal motion of the slider (see equation 13) to mimic the motion of the knives. To mimic the dynamics of the complete shedding system, with the main compliance in the heddle springs R10.c, the oscillating load resonance $\sqrt{\frac{k_t}{m_s}}$ is selected well below the shaft resonance $\sqrt{\frac{k_s}{I_s}}$ (with $I_s$ the equivalent slider inertia: equation 6):

\[ \sqrt{\frac{k_t}{m_s}} < 0.1 \sqrt{\frac{k_s}{I_s}} \]

Figure 10 represent the frequency response for the shedding system, to be mimicked with the slider-crank.

The equivalent inertia is determined from the slider mass $m_s$ over an effective arm $r_{eff}$:
17

In the following, the effective arm \( r_{\text{eff}} \) is defined. To start, the corresponding equivalent torque \( T_s \) is determined from moving \( I_s \) with an effective angular acceleration \( \ddot{\theta}_{\text{eff}} \):

\[ \ddot{\theta}_{\text{eff}} = \frac{\ddot{x}_s}{r_{\text{eff}}} \]  

(7)

\[ T_s = I_s \cdot \ddot{\theta}_{\text{eff}} \]  

(8)

Substituting equations 6 and 7 into equation 8, \( T_s \) can be expressed as:

\[ T_s = r_{\text{eff}} m_s \ddot{x}_s \]  

(9)

Comparing the right hand side of equation 9 with the second term on the right hand side of equation 18, the following can be derived for the effective arm \( r_{\text{eff}} \):

\[ r_{\text{eff}} = \left( \frac{C_s C_r}{C_r} + S_c \right) \]  

(10)

For definition of the terms in equation 10, see equations 13 and 14.

**Follow SMD limitations**

For successful control compensation of torsional vibration, shaft resonance \( \sqrt{k_s/I_s} \) is selected well below motor torque sampling rate \( f_{s,m} \) (750[Hz]: R9.b), see Eq. 11. This overview is also given in Figure 10.

\[ \sqrt{\frac{k_s}{I_s}} < 0.1 f_{s,m} \]  

(11)

The motor torque \( T_d \) required to move the slider mass \( m_s \) connected to the spring with stiffness \( k_t \) at a speed \( \omega \) is kept below the maximum supply torque \( T_{\text{max}} \) (15[Nm]) i.e.:

\[ T_d (m_s, k_t, \omega) < T_{\text{max}} \text{ with } \omega < \omega_{\text{max}} (3600[rpm]) \]  

(12)

For elaboration on \( T_d \), see section 4.2.1., equation 18.

Finally, the SMD footprint also puts limitations to the specification of the slider-crank geometry, R9.d. Figure 3 gives an impression of the standard components provided in the SMD, R9.e: Motor, sensors (encoders and torque sensors), bearings, shafts, inertias. It is also clear that a limited footprint is available.

With the aid of simulation, the specifications for the slider-crank are set, as shown in Table 3 according to Figure 9, in order to meet the above requirements. Bearing specifications correspond to the operating conditions \((T, \omega)\) of the slider-crank and the reaction forces \( F_{\text{react}} \) in the nodes.
### Table 3 Slider-crank specifications

<table>
<thead>
<tr>
<th>Geometry</th>
<th>Links</th>
<th>Slider</th>
<th>Springs</th>
<th>Bearings</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h = 135$[mm]</td>
<td>$l_c = 50$[mm], $100$[mm]</td>
<td>$l_r = 300$[mm]</td>
<td>$k_z = 0.15$[N/mm]</td>
<td>$k_c = 0.78$[N/mm]</td>
</tr>
<tr>
<td>$X_1 = 15$[mm]</td>
<td>$m_c = 0.75$[kg/m]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$X_2 = 285$[mm]</td>
<td>$m_r = 0.75$[kg/m]</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

- $0.2$[kg] < $m_s$ < 2[kg]
- $85$[mm] < $l_{kz} < 327$[mm]
- $27$[mm] < $l_{kc} < 82$[mm]
- $3.28$[N] < $F_{kc} < 39.31$[N]
- $F_{kc} < 42.6$[N]
- $T = 30$[Nm]
- $F_{react} = 200$[N]

**Cad design, realization and verification**

With the determined specifications, the slider-crank is designed (Figure 11) and manufactured (Figure 12). Finally, the design is verified. Table 4 is a copy of Table 3 except for the specifications that are not met. For these, the deviating measured or weighted values are presented in bold. The link masses $m_c$ and $m_r$ came out higher than expected and the geometrical parameter $h$ came out lower. This was required to ensure structural integrity and to follow the SMD limitations. Another deviation is that the center of mass of the slider is lower than expected, therefor changing the effective rod length $l_r$. To cope with these deviations, their effects are included in the slider-crank model development used for feedforward control.

![Figure 11 Slider-crank CAD design](image1)

![Figure 12 Slider-crank realization](image2)
4.2 Control subsystems

In this section, the design, realization and verification of the control blocks are discussed: The oscillating load/slider-crank, backlash and motor and finally the flexible shaft. Block output(s) $U$ are mathematically expressed in terms of the input(s) $X$ and the parameter(s) $\Theta$: $U = F(X, \Theta)$. In Figure 6, the parameters are represented in grey and enter the blocks through grey arrows.

4.2.1. Oscillating load

This part of the report concerns expressing the oscillating load disturbance torque $T_d$ in terms of the (angular) acceleration, velocity and position (load) reference $\dot{\theta}_t$, $\dot{\theta}_r$, $\theta$, and the slider-crank physical parameters. This is done by conducting a kinematic and dynamic analysis on the slider-crank. Results from the derived equations are compared with multibody simulations and finally verified with experiments on the realized slider-crank.

**Kinematic analysis**

The motion of the joints and links (centers of masses) is expressed in terms of the degree of freedom, i.e., motion of the crank $\dot{\theta}_t$, $\dot{\theta}_r$, $\theta$, using kinematic analysis. Starting from some basic geometric equations (Eq.13) slider and rod motion can be expressed in crank motion, see Eq.13 and Eq.14 resp.

![Figure 13 Geometry slider-crank](image)
\[ x_s = C_c + W - x_s(0) \]
\[ \ddot{x}_s = \left( -\dot{\theta}_t s_c - (\dot{\theta}_t)^2 C_c + \frac{(\dot{\theta}_c - \dot{\theta}_t)^2 s_c (h - s_0) - (\dot{\theta}_t)^2 c_c^2}{w} \right) - \frac{(\dot{\theta}_t)^2 c_c^2 (h - s_0)^2}{w^3} \]

\[ C_c = l_c \cos \left( \dot{\theta}_t t + \theta_t (0) \right) \]
\[ S_c = l_c \sin \left( \dot{\theta}_t t + \theta_t (0) \right) \]
\[ W = \sqrt{l_t^2 - (h - S_c)^2} \]
\[ \ddot{x}_r = \left( -\dot{\theta}_l S_c - (\dot{\theta}_l)^2 C_c + \frac{(\dot{\theta}_c - \dot{\theta}_l)^2 s_c (h - s_0) - (\dot{\theta}_l)^2 c_c^2}{2w} \right) - \frac{(\dot{\theta}_l)^2 c_c^2 (h - s_0)^2}{2w^3} \]
\[ \dddot{y}_r = -\frac{1}{2} C_r \dddot{\theta}_r + \frac{1}{2} S_r \dddot{\theta}_r \]
\[ \ddot{\theta}_r = \frac{C_l \dot{\theta}_l - S_c \dddot{\theta}_r}{C_r} \]
\[ S_r = l_r \left( \frac{s_c - h}{l_r} \right) \]
\[ C_r = l_r \sqrt{1 - \left( \frac{s_c - h}{l_r} \right)^2} \]

**Dynamic analysis using free body diagrams**
The torque \( T_d \) required for moving the link masses and inertias and compensating for external forces such as gravity \((g)\) and spring forces \((F_{k_t} \text{ and } F_{k_c})\) is determined using free body diagrams (FBDs). Starting from the equations of motion for the crank (Eq. 15), rod (Eq. 16) and slider (Eq. 17) derived from the FBD, the torque \( T_d \) required for moving the slider-crank can be determined (Eq. 18).

![Figure 14 FBD slider-crank](image-url)
\begin{align}
- F_{k_t} + F_{XSR} &= m_s \ddot{x}_s \\
- F_{YGS} + F_{YSR} &= 0
\end{align}

\begin{align}
T_d &= -(S_c \ddot{x}_r + C_c \ddot{y}_r)m_r - \left( \frac{C_c S_c}{\epsilon_r} + S_c \right) \left( F_{k_t} + F_{k_c} + m_s \ddot{x}_s \right) - 0.5C_c(F_{Gc} + F_{Gr}) - \frac{C_c}{\epsilon_r} l_\theta \ddot{\theta}_r + l_c \ddot{\theta}_l \\
F_{k_c} &= \begin{cases} 
  k_c x_{im}, & x_{im} > 0 \\
  0, & x_{im} \leq 0 
\end{cases} \\
x_{im} &= x_s - (1 - 0.01p)x_{s,range} \\
x_{s,range} &= \sqrt{(l_r + l_c)^2 - (h)^2} - \sqrt{(l_r - l_c)^2 - (h)^2} \\
F_{k_t} &= k_t x_s \\
F_{Gi} &= m_l g \\
l_i &= \frac{1}{3} m_l l_i^2
\end{align}

Model comparison with multibody simulations

Given a specific motion profile (4[Hz]), the calculated torque $T_d$ is compared with multibody simulations in Simulink, see Figure 15. Although there is a small discrepancy with the multi-body simulation, for this control task, the equations are considered sufficient.

![Comparison calculated torque $T_d$ with multi-body simulations](image)

Model verification with realized setup

The calculated torque $T_d$ is compared with experimental results from the slider-crank on the SMD. The torque is measured with a sensor as displayed in Figure 12. The results in Figure 16, Figure 17 and Figure 18 correspond to moving the slider-crank (with specifications presented in Table 4) at 3[Hz], 4[Hz] and 5[Hz] respectively. More fluctuations appear in the results for 3 [Hz] as this is closer to the resonance of the slider mass, $m_s$, connected to the tension spring, $k_t$, see Figure 9 and Figure 10.
10. Using the specifications from Table 4, this resonance amounts to $\sqrt{\frac{k_t}{m_s}} = 1.86$ [Hz]. For the task at hand, the match between calculations and measurements is sufficient. I.e., with this model, feedforward control is feasible as presented in section 5 with the results plotted in Figure 42. Nevertheless, for a better match, model parameter optimization can be executed. An example for this is given in the appendix.

![Figure 16 Comparison calculated torque $T_d$ with experimental results (3 [Hz])](image1)

![Figure 17 Comparison calculated torque $T_d$ with experimental results (4 [Hz])](image2)
4.2.2. Backlash and motor

This section deals with determining the backlash correction profiles $\theta_b, \alpha_b$ from the load disturbance torque $T_d$ (responsible for gap opening). Next, the motor feedforward torque $T_{FF,m}$ is calculated from $\alpha_b$ and the motor inertia (parameter) $I_m$. Finally, the developed blocks are verified through simulation and experiments with a backlash element, vulnerable to gap opening caused by a load disturbance. Here, an unbalanced inertia is used instead of the slider-crank (to be used in final test).

Position backlash correction profile $\theta_b$

As shown in Figure 20 and Eq.19, $\theta_b$ is a pulse train profile with smooth transitions. Smooth transitions are chosen in order to accelerate with acceptable torques during gap traversal. For achieving smooth transition, s-curves are used with equal widths $w_{\alpha_b}$ for acceleration during $\theta_{b1}(t)$, constant velocity during $\theta_{b2}(t)$ and deceleration during $\theta_{b3}(t)$. Greater $w_{\alpha_b}$ implies more time for acceleration and therefore less torque demand $T$. On the other hand, the gap traversal is slower, resulting to higher tracking errors $e$. The transitions in the pulse train are synchronized with the instances of expected gap opening. Gap opening is expected when the load disturbance torque $T_d$ changes direction, i.e., at the zero-crossings $t_{zc}$ of $T_d$, see Figure 19.

Figure 18 Comparison calculated torque $T_d$ with experimental results (5 [Hz])
After having obtained a mathematical expression for $\theta_b$ (Eq. 19), $\alpha_b$ (see Eq. 20 and Figure 21) is simply constructed by analytically taking the double derivative of $\theta_b$:
Figure 21 Backlash correction profile $\alpha_b$

$$\alpha_b(t) = \text{sgn}(T_d) \cdot \begin{cases} 
\alpha_{b1}(t) & t < t_{zc} + w_{ab} \\
\alpha_{b2}(t) & t_{zc} + w_{ab} < t < t_{zc} + 2w_{ab}, \text{else: } \alpha_b(t) = 0 \\
\alpha_{b3}(t) & t_{zc} + 2w_{ab} < t < t_{zc} + 3w_{ab}
\end{cases}$$

(20) $t_{zc} = \text{zero-crossing}(T_d)$

$\alpha_{b1} = \frac{bg}{2w_{ab}^2}$
$\alpha_{b2} = 0$
$\alpha_{b3} = -\frac{bg}{2w_{ab}^2}$

**Motor**

In the motor block, the feedforward torque $T_{FF,m}$ is calculated from $\alpha_b$ and $I_m$ by simply applying the equation of motion to the rotating motor shaft (assuming negligible friction and flexibility):

(21) $T_{FF,m} = I_m \alpha_b$

**Assets**

As shown in Figure 22, the feedforward torque $T_{FF}$ is used to compensate for the predictable load disturbance $T_d$ and to traverse the gap $T_{FF,m}$. At low operating speed, full motor torque $T_{max}$ is available for gap traversal $T_{FF,m}$. At low speed, the zero-crossings $t_{zc}$ are further apart from each other and $T_{FF,m}$ is effectively closer to the $t_{zc}$. At $t_{zc}$, $T_d$ is zero and $T_{max}$ is fully available for $T_{FF,m}$. Consequently, at high speed, the available motor torque $T_{max}$ is distributed over $T_{FF,m}$ and $-T_d$. Furthermore quick traversal is possible because only the motor inertia $I_m$ is accelerated at gap traversal, i.e., when motor and load are physically decoupled.
Figure 22 Feedforward torque $T_{FF}$ to compensate for the load disturbance $T_d$ and to traverse the backlash gap $T_{FF,m}$

**Limitations**

Thus, at low speed, the torque applied for backlash compensation $T_{FF,m}$ is limited by the maximum motor torque $T_{max}$ in agreement with R6:

\[
T_{FF,m} \leq T_{max}
\]

Equations 19 and 20 show that an s-curve reference with equal widths $w_{\alpha_b}$ of acceleration $a_b$ is used to traverse the gap $b$. Equation 21 shows the torque $T_{FF,m}$ required from the motor ($I_m$) to accelerate with $a_b$. Combining these equations yields:

\[
T_{FF,m} = \frac{I_m b g}{2 w_{\alpha_b}^2}
\]

Combining equations 22 and 23 reveals that due to the limited motor torque $T_{max}$, the width $w_{\alpha_b}$ for acceleration $a_b$ is lower bounded:

\[
w_{\alpha_b} \geq \sqrt{\frac{b g I_m}{2 T_{max}}}
\]

This limitation on $w_{\alpha_b}$ puts limitations on the operating speed $\theta_i$. From Figure 20 it can be derived that in-between zero-crossings at least a time distance of $3w_{\alpha_b}$ is required. From Figure 23, a basis load profile, zero-crossings nearest to each other appear at a time distance 5% of the period $P$. This means that:

\[
3w_{\alpha_b} \leq 0.05P
\]

The speed (defined as $\frac{2\pi}{P}$), is therefore upper bounded:

\[
\hat{\theta}_i \leq \frac{\pi}{30w_{\alpha_b}}
\]
Rearranging equations 25 and 26 yields equation 27 which shows that the gap $bg$ is proportional to the maximum torque $T_{\text{max}}$ and the inverse squared speed $\frac{1}{\dot{\theta}_l}$.

\begin{equation}
bg \leq \frac{\pi^2 T_{\text{max}}}{450 \dot{\theta}_l I_m}
\end{equation}

Figure 23 Basis for profile oscillating load

Figure 24 illustrates the resulting trade-off between on the one hand relaxing mechanics, i.e., increasing backlash gap and on the other hand maintaining performance, i.e., high operating speed and low torque demand.

Figure 24 Trade-off between on the one hand relaxed mechanics (increased backlash gap) and on the other hand performance (high speed and low torque demand)
Backlash block verification

To see if the developed backlash feedforward technique is able to compensate for backlash, the tracking error and torque demand are evaluated before and after compensation. As the focus here is on verifying the developed feedforward backlash block (Figure 26), gap opening is provoked by an unbalanced inertia load disturbance (Figure 25, Figure 29), i.e., a rather much simpler disturbance load compared to the slider-crank. The disturbance torque $T_d$, expressed in Eq.28, is caused by the perpendicular gravity component $g \sin \theta_l$ acting on the decentered disc mass $m_u$ over arm $y_u$. The unbalanced inertia $I_u$, used for feedforward, is expressed in Eq.29.

\[
T_d = y_u \cdot F_u \sin \theta : \text{Sinusoidal torque disturbance}
\]
\[
y_u = \frac{4r_u}{3\pi} : \text{Center of mass for half disc}
\]

(28)

\[
F_{Gu} = m_u g : \text{Gravitational force with } g = 10 \left[ \frac{m}{s^2} \right]
\]
\[
m_u = \rho_u A_u h_u : \text{Mass half disc with density } \rho_u \text{ and thickness } h_u
\]
\[
A_u = 0.5\pi r_u^2 : \text{Area half disc}
\]

(29)

\[
I_u = I_{u,0} + m_u y_u^2 : \text{Moment of inertia half disc w.r.t. central axis of rotation}
\]
\[
I_{u,0} = \int r_u^2 dm_u, \quad dm_u = \rho_u h_u \pi r_u dr_u \Leftrightarrow I_{u,0} = 0.25\rho_u h_u \pi r_u^4
\]

For a half stainless steel disc (solid), $\rho_u = 7480 \text{ [kg/m}^3\text{]}$, radius $r_u = 0.1 \text{ [m]}$ and thickness $h_u = 0.005 \text{ [m]}$, the following values are obtained from the abovementioned equations: $m_u = 0.5875 \text{[kg]}$, $y_u = 0.0424 \text{ [m]}$ and $I_u = 0.004 \text{ [kg} \cdot \text{m}^2\text{]}$.

![Figure 25 Unbalanced Inertia – half disc](image-url)
Previous to verifying the control block, backlash gap, $bg = 0.14[\text{rad}]$, is identified as shown in Figure 27. For this, a sinusoidal load position reference is applied to the system shown in Figure 28. The motor position is plotted against the load position. This results in a typical backlash plot from which the gap size can be read. The friction module (see Figure 28) is used to eliminate bouncing during impact, which allows for better identification.
When applying the proposed backlash compensation technique, significant improvement is achieved. Tracking error is reduced at the expense of little increase in torque demand due to the smooth transitions during gap-traversal. Good results are obtained, both in simulation and implementation, Figure 30 and Figure 31 respectively. The left plots show the slight increase in torque demand while the right plots illustrate significant reduction in tracking error when applying backlash compensation.
Figure 30 Simulation results: with and without backlash compensation; Left: Torque demand; Right: Tracking error
Figure 31 Implementation results: with and without backlash compensation; Left: Torque demand; Right: Tracking error
4.2.3. Flexible shaft

In this section, the flexible shaft feedforward torque $T_{FF,fs}$ (required for torsional vibration compensation: see Figure 32) is calculated. Thereafter system identification is conducted to validate the developed model (equations). Finally, to verify if the developed flexible feedforward technique is able to compensate for torsional vibration, the tracking error and torque demand are evaluated before and after compensation. This is done on the flexible shaft two-inertia setup shown in Figure 33.

![Figure 32 System Architecture for torsional vibration compensation (without backlash compensation)](image)

![Figure 33 flexible shaft two-inertia setup](image)

**Model development flexible two-inertia system**

For the development of the flexible feedforward block, the equations of motion of a flexible shaft $k_s$ – two inertia $I_m, I_l$ system are used. The system is vulnerable to torsional vibration due to acceleration in the reference $\ddot{\theta}_i$ as well as the external torque disturbance $T_d$. 

\begin{align}
   T_{FF,fs} &= \left( I_m + d_s \frac{1}{s} + k_s \frac{1}{s^2} \right) \ddot{\theta}_m - \left( d_s \frac{1}{s} + k_s \frac{1}{s^2} \right) \ddot{\theta}_l \\
   \ddot{\theta}_i &= \frac{T_s}{t_i} + \frac{1}{s} \left( \ddot{\theta}_m - \ddot{\theta}_i \right) + \frac{1}{s^2} \left( \ddot{\theta}_m - \ddot{\theta}_l \right)
\end{align}

(30)
System identification and model validation

The developed model, Eq. 30, is validated with the identified plant. This is presented in Figure 34 (Yellow: model; Green: identification). For identification, the open-loop system is excited with a Schroeder – phased multi-sine according to equation 31.

\[ \sum_{k=1}^{N} A_m \cos(2\pi \omega_k t + \phi_k) \]
\[ \phi_k = -\frac{k(k-1)\pi}{N} \]

The excitation frequency \( \omega_k \) is taken up to five times the highest expected resonance \( \omega_R \) and the sampling frequency \( f_s \) is taken ten times higher than \( \omega_R \):

\[ 0 < \omega_k < 5\omega_R \]
\[ f_s = \frac{50\omega_R}{2\pi} \]

The highest expected resonance \( \omega_R \) is expressed in Eq. 34 which is derived from the transfer function in Eq. 33 based on the equations of motion, Eq. 30.

\[ \omega_R = \sqrt{\frac{k_s}{I_m + I_l}} \]
\[ \omega_{AR} = \sqrt{\frac{k_s}{I_l}} \]

With \( I_l = 0.0025 \text{ [kg/m}^2\text{]} \), \( I_m = 0.0053 \text{ [kg/m}^2\text{]} \) and \( k_s = 250 \text{[Nm/rad]} \), \( \omega_R \) is expected to be around 385 [rad/s].
Flexible feedforward block verification
To see if the developed flexible feedforward technique is able to compensate for torsional vibration, the tracking error and torque demand are evaluated before and after compensation. As shown in Figure 35, a ramp reference with a chirp added to it is selected for $\theta_1$ to excite the system resonance $\omega_R$ and anti-resonance $\omega_{AR}$. Results without and with torsional vibration compensation are first shown in simulation, Figure 36, and then in implementation, Figure 37. The results show that control with flexible feedforward allows better tracking at resonance and anti-resonance. However, torque demand at anti-resonance is significantly increased. Furthermore, it is noticeable in the experimental results, Figure 37, that the applied technique does perform not perform as well as in the simulations. This is most likely caused by a model mismatch and requires further investigation in future work.

Figure 35 Ramp reference with a chirp added to excite the flexible shaft – two-inertia resonance and anti-resonance
Figure 36 Simulation: Torque demand (left) and tracking performance (right); Comparison between Rigid two mass, Flexible two-mass with Rigid FF and Flexible two-mass/inertia with Flexible FF
Figure 37 Implementation: Torque demand (left) and tracking performance (right); Comparison between Rigid two mass, Flexible two-mass with Rigid FF and Flexible two-mass/inertia with Flexible FF.
5. System integration, implementation and validation

This chapter starts with system integration and implementation. The complete control and hardware configurations are modeled in Matlab-Simulink. The chapter concludes with system validation. The validation involves the evaluation of the performance of the integrated system. The performance requirements are finally traced back to the main key-drivers and the project goal.

5.1 System integration and implementation

The complete hardware and control configuration as presented in Figure 6 is developed in Matlab-Simulink depicted in Figure 38. This is used for simulation. The developed control configuration is also used for code generation in order to implement the controllers on both the benchmark- and relaxed mechanics setup, Figure 39 and Figure 40 resp.

Figure 38 Hardware and control implemented in Simulink: Left: Control configuration (PID feedback and model based feedforward); Right: Hardware configuration (mimicked shedding system)
5.2 **System validation**

This section deals with validation of the complete system. For this, the results for backlash and torsional vibration compensation for the complete system are considered. Simulation and implementation results (for moving the slider crank at 5[Hz]) are presented in Figure 41 and Figure 42 resp. Here, tracking performance (torque demand and tracking error) of the various systems are presented: (i) the benchmark system (tight mechanics), (ii) the system with relaxed mechanics and (iii) the system with relaxed mechanics and improved control. The results are discussed in detail in section 5.2.2. after defining the performance indicators and metrics in section 5.2.1. Finally in section 5.2.3. room for improvement is discussed.
Figure 41 Simulation: Torque demand (left) and tracking performance (right)
Figure 42 Implementation: Torque demand (left) and tracking performance (right)
Figure 43 Feedforward (FF) torque- and feedback (FB) torque contribution; Left: Simulation; Right: Implementation
5.2.1. Performance metrics

For validating the improved/modified system, comparison is done with the benchmark (tight mechanics: rigid shaft and zero-backlash) w.r.t. the tracking error and the torque demand, R6 and R5. Both absolute and relative tracking errors are considered:

**R20.** Absolute tracking errors: Good tracking at motor ($\theta_m$) and load side ($\theta_l$) is desired to ensure sufficient shed opening to reduce process defects, R3.a. However, imagining passing through a single weft yarn through a shed opening of 7 [cm], allows for quite some room tracking error. More desirable is smooth shed opening rather than highly accurate shed opening. Less fluctuations imply less abrasion (R3.c.ii).

**R21.** Relative tracking error: Also here, restricting the error between $\theta_m$ and $\theta_l$ implies less abrasion (inter-rubbing of the warp yarn; R3.c.ii).

To make sure that significant contribution is made by the applied feedforward control, feedback control is applied with a low bandwidth. A bandwidth of 100 [rad/s] is selected to make sure that shaft resonances (between 300 and 400 [rad/s]; see section 4.2.3.) are truly compensated by feedforward. Both feedback and feedforward torques are measured. This is done to evaluate if the applied feedforward has significant contribution as required.

5.2.2. Discussion - Result interpretation

This trade-off supports/allows improvement of the weaving machine in terms of reduced cost, R1, and equipment effectiveness, R2.

**Results - Torsional vibration compensation**

As shown in simulation and implementation, Figure 36 and Figure 37 resp., when applying torsional vibration compensation for the system with reduced shaft stiffness, tracking is improved. This requires higher torque demand when hitting the anti-resonance and lower demand when hitting the resonance. Performance is however still less compared to the benchmark system with the rigid shaft. Nevertheless, as intended, these results illustrate the possible trade-off between low hardware maintenance/cost on the one hand and high performance (i.e., good tracking and low power consumption) on the other hand. This is illustrated in Figure 44. Trade-off: Reducing shaft stiffness (R7.a), increases tracking error and torque demand. Applying torsional vibration compensation (R8.a) partially recovers tracking (R5) but at some instances (i.e., at anti-resonance) further increases torque demand (R6).

Torsional vibration is compensated for the collocated side. This is done to synchronize with feedback control so that no counteracting takes place between these. As a result, torsional vibration is not compensated at the non-collocated side. For that, non-collated control, both feedback and feedforward is required. With one actuation degree of freedom, i.e., one motor, it is not feasible to compensate torsional vibration at both collocated and non-collocated side. This results in an inevitable relative tracking error and therefore not meeting R3.c.ii. This requires improvement in future work.

**Results - Backlash compensation**

The applied backlash compensation technique proved to work well, i.e., when applying improved control for the system with backlash gap, significant error reduction and reduced torque demand are achieved, see the simulation and implementation results, Figure 41 and Figure 42 resp. Figure 43 distinguishes between the contribution of the feedforward- and feedback torque. The results show that the feedforward torque is the main responsible for gap traversal while feedback torque is used to compensate for the impact(s) after traversal. Note that these results represent compensation for the system disturbed by: the slider-crank without the impact spring $k_c$, i.e., with a load profile shown in Figure 52, rather than the more challenging load profile shown in Figure 18. Leaving out the impact spring $k_c$, is due to time constraint. Extension with $k_c$ is left for future work.
A major asset of the applied backlash compensation technique is that, as explained in section 4.2.2, at low speed, maximum motor torque is available for gap traversal. Furthermore, quick gap traversal is possible because this requires acceleration of only the motor inertia. Despite of that, the torque demand and tracking error are still higher when compared to the benchmark system (with minimum/zero gap size). Nevertheless, these results illustrate the possible trade-off between low hardware maintenance/cost on the one hand and high performance (i.e., good tracking and low power consumption) on the other hand. This is illustrated in Figure 45. Trade-off: Increasing gap size (R7.b), increases tracking error and torque demand. Applying backlash compensation (R8.b) partially recovers both tracking (R5) and torque demand (R6).

A major condition for the technique to work is that the feedforward model is able to predict the load disturbance zero-crossings. As the system load complexity increases, also the prediction becomes more challenging. This causes the applied compensation technique to be more vulnerable to failure. This is evident when comparing the relatively simple unbalanced inertia load (Eq. 28) with the more complex slider-crank load mechanism (Eq. 18). Furthermore, as also explained in section 4.2.2, the limited motor torque $T_{max}$ (Eq. 22) enforces longer accelerations, i.e., greater values for $w_{ab}$ (Eq. 24), which consequently limits operating speed $\dot{\theta}_l$ (Eq. 26).

It is also worth mentioning that, at lower speeds, i.e., when spring load dominates over inertial load, see Figure 54, torsional vibration- and backlash compensation do not interfere with each other. Backlash is compensated at the zero-crossings, i.e. when disturbance load is zero. This occurs when the effective arm is zero, implying that the equivalent inertia is zero. Therefore, shaft resonance is increased and the shaft is less sensitive to impacts. Section 4.1.3. is complementary to this inference. At higher speeds, i.e., when inertial load dominates over spring load, see Figure 55, not every load zero-crossing is correlated with zero equivalent inertia.

**Figure 44. Trade-off:** Reducing shaft stiffness (R7.a), increases tracking error and torque demand. Applying torsional vibration compensation (R8.a) partially recovers tracking (R5) but at some instances (i.e., at anti-resonance) further increases torque demand (R6).
Figure 45. Trade-off: Increasing gap size (R7.b), increases tracking error and torque demand. Applying backlash compensation (R8.b) partially recovers both tracking (R5) and torque demand (R6).

5.2.3. Room for improvement

As overviewed in Figure 46, for both techniques, torsional vibration- and backlash compensation, there is room for improvement:

**Torsional vibration compensation**

With the applied collocated feedback and feedforward torsional vibration compensation technique, a relative tracking error is inevitable. Unless beneficially used, this is not desirable, R21, and therefore requires attention in future work.

**Backlash compensation**

The major challenge here is the timing/ prediction of the load zero-crossings, see Figure 8. Good prediction depends on model correctness, completeness and preconditions:

- Improve model correctness: Among other things, better model parameter estimation improves the model predictions. E.g., a wrongly estimated initial condition ($\theta_i(0)$) leads to wrongly predicted (/shifted) zero crossings.
- Improve model completeness: Especially when applying the proposed control technique to the actual shedding system, model extension will be required for sufficient predictions:
  - Deal with sudden changes in parameters: For weaving a pattern on the fabric, the load on the shedding system varies, i.e., the number of heddles (-springs) being picked up varies. As a result, the effective spring stiffness and heddle friction are subject to sudden changes.
  - Deal with gradual changes in parameters: Over time, the system heats up, as does the lube-oil in e.g., the bearings. This changes the viscosity parameters of the oil and therefore also the friction parameters of the bearings. Over longer periods of time, equipment wear
may take place. As such, the backlash gap in the gearbox is vulnerable to increasing, causing model versus plant mismatch.

- Deal with overlooked phenomena: Phenomena not taken into consideration in this study but which may play an important role in the actual machine are e.g., heddle friction and other nonlinearities.

- Improve preconditions: Model predictions are based on perfect tracking, see section 4.2. Therefore, in the presence of speed fluctuation, model prediction will not match reality.

A secondary challenge with the applied backlash compensation technique is traversing the gap both quickly and smoothly. In this study a second order s-curve correction profile is selected, see Figure 20. Improvements on this may for instance involve increasing the order of the correction profile.

**Figure 46 Overview: Room for improvement for torsional vibration- and backlash compensation and corresponding recommendations for hardware and control**
6. Project Management

In this chapter, the management of this project is discussed. First, the approach that is followed is elaborated upon. This involves decomposing the project in tasks to ensure successful delivery. Tools and strategies used for executing the tasks are discussed as well. Next, the milestones set in this project are discussed. Both the approach and the milestones are discussed using Figure 47, the project planning. There they are represented as the tasks and the Projects Steering Group Meetings (PSGMs) respectively. The PSGMs are synchronized with the milestones. This chapter continues with the conducted risk analysis and the mitigations taken as measures. Finally, stakeholder management is discussed.

6.1 Project Approach

The approach followed in this project is according to five task as displayed on the utmost left of the project planning (Figure 47):

1. Project Initiation: The aim with project initiation is for the stakeholders (both company and university supervisors) to come to an agreement on the deliverables of the project. Formulating the deliverables of the project involves setting the goal and scope. For setting these, proper orientation is conducted, i.e., literature study as well as familiarization with the SMD (Figure 3).

2. Requirements: After setting the goal, scope and deliverables, the system requirements are derived. To make sure requirement traceability is transparent, a requirement/key-drivers diagram is used (Figure 4). The key-driver diagram stems from the CAFCR methodology of taking five different views for gaining more understanding of the project and tackling it in a better way. These views correspond to the letters CAFCR: Customer objectives view, Application view, Functional View, Conceptual view and Realization view. The key-drivers diagram captures the first three views and shows traceability from the customer objectives (key-drivers) to the functional requirements.

3. Design and verification: From the requirements, concepts are generated for the system architecture as well as for the sub-systems. To make design manageable, the work is spread over four stages, presented on the time-line in the project planning. Figure 47. Each stage involves design
and verification of two subsystems (packages A & B), see Figure 48. Packages C are dedicated to integration. The four stages are:

1. Stage 1: Packages 1A and 1B involve hardware and control design and verification for torsional vibration compensation and backlash compensation respectively.
2. Stage 2: Packages 2A and 2B are similar to 1A and 1B but for more challenging test scenarios, i.e., more challenging references and loads. 2C is dedicated to integration of 2A and 2B.
3. Stage 3: This stage is dedicated to implementation of the slider-crank mechanism on the SMD. Packages 3A and 3B correspond to the inertia load and spring loads respectively, while 3C involves their integration.
4. Stage 4: Stage 4 involves integration of the previous stages. 4A combines 2A with 3C. 4B combines 2B and 3C. 4C combines 2C and 3C: This represents implementation of the complete system.

Subsystem design and verification is conducted according to the V-model approach consistent with R13. Furthermore, control design is done using Matlab/Simulink.

4. Integration and validation: In this task, the developed (complete) system is validated. I.e., there is checked whether the system requirements are met, as proposed by the V-model approach (R13).
5. Documentation – Dropbox, Word, PowerPoint: From the project planning, Figure 47, it is clear that documentation is spread over the complete time-line. The time-line also shows the milestones for handing in the draft and final reports. The contents of the report correspond to Figure 49.

Figure 48 Work break-down/ Work packages
6.2 **Meetings**

To keep the project going in the right direction, regular meetings are organized. With the company supervisors weekly progress meetings are planned. And with the Project Steering Group (both company and university supervisors) regular meetings (Project Steering Group Meetings: PSGMs) are scheduled as well. These meetings are scheduled every four to six weeks. They are planned well ahead of time (one month) using Doodle. Meeting invitations are sent using Outlook. Meetings are prepared by making PowerPoint presentation of the progress made, the planning and the next steps. Especially for the PSGMs, meeting minutes are taken to record the comments, feedback and steering from the PSG. Just as the time-line in Figure 47 (the project planning) shows, the PSGMs are synchronized with the milestones for achieving the project tasks. In addition to the weekly progress meetings and the PSGMs, several return days are planned for the trainees to present their work to each other. Here the emphasis is on project management. Finally, a final presentation is given at both the company and the university.

6.3 **Risks Analysis**

Risk analysis is conducted in order to mitigate project risks. Table 5 presents the identified risks and discusses their probability, impact and mitigation plan. To keep track of the risks, monthly updates/checkups are done. The last column of Table 5 shows when the risk started and when the mitigation was completed.
<table>
<thead>
<tr>
<th>#</th>
<th>Risk</th>
<th>Probability</th>
<th>Impact</th>
<th>Mitigation</th>
<th>Status</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Solving the wrong problem by applying the bottom up approach (R12).</td>
<td>This (R12) is a process requirement set by the company/client because they prefer to get tangible results early on.</td>
<td>Will not necessarily lead to solving the right problem. Nevertheless, the client is aware of the consequences of applying the bottom-up approach.</td>
<td>Combine bottom-up approach with V-model approach (R13).</td>
<td>Started (16/12/2016)  Completed (22/08/2017)</td>
</tr>
<tr>
<td>2</td>
<td>Missing useful feedback from company project manager due to limited visits to weekly meetings.</td>
<td>The company project manager has many tasks and possibly overlapping meetings with higher priority, disabling him to visit all weekly meetings.</td>
<td>Undetected requirements leading to non-optimal solution.</td>
<td>Schedule PSGMs (see timeline in Figure 47) with company project manager.</td>
<td>Started (16/12/2016)  Completed (22/08/2017)</td>
</tr>
<tr>
<td>3</td>
<td>Difficulty interpreting results from implementation on SMD (Figure 3) w.r.t real size system (Figure 2).</td>
<td>Dynamics of the real system are mimicked in the SMD on a smaller scale (R10). Therefore, interpreting results becomes easier.</td>
<td>No clear advice for client from the obtained results.</td>
<td>Using the requirement diagram (Figure 4), ensure good traceability from system requirements to requirements for implementation on SMD, such as mimicking system dynamics.</td>
<td>Started (16/12/2017)  Completed (22/08/2017)</td>
</tr>
<tr>
<td>4</td>
<td>Difficulty scheduling PSGM with 6 members.</td>
<td>Both company and university supervisors have busy schedules. On top of that, traveling between company and university is at least two hours.</td>
<td>Absence of supervisors/stakeholders imply less feedback and therefore less desirable solutions.</td>
<td>Schedule meetings ahead (one month) using Doodle. Also schedule conference calls.</td>
<td>Started (16/12/2017)  Completed (22/08/2017)</td>
</tr>
<tr>
<td>5</td>
<td>Difficulty fulfilling PDEEng requirements with reduced project scope.</td>
<td>With some high level design already done by the company, the project scope reduces leaving less room for high level design choices.</td>
<td>A PDEng project comprises of both high level and low level system design. Making less high level design choices can create imbalance.</td>
<td>High Level design choices are made for mechanical and control design (R11&amp;R8).</td>
<td>Started (26/01/2017)  Completed (23/02/2017)</td>
</tr>
<tr>
<td></td>
<td>Having hardware ordered and delivered late.</td>
<td>In the company site at Leuven, there is one person mainly responsible for mechanical design and ordering the parts.</td>
<td>At some point, project progress and the deliverables heavily depend on the ordered parts, mainly stage 3 in Figure 48.</td>
<td>Put high priority to finish hardware design and all related to it: Offer help to mechanical designer.</td>
<td>Started (23/02/2017) Completed (31/07/2017)</td>
</tr>
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</tr>
<tr>
<td>7</td>
<td>Staying too long on a conceptual/research level.</td>
<td>Following the V-model/ Top down approach requires spending time on high-level system design.</td>
<td>High level system design results are less tangible.</td>
<td>Combine the V-model with the bottom-up approach to have some tangible results early on.</td>
<td>Started (23/02/2017) Completed (19/04/2017)</td>
</tr>
<tr>
<td>8</td>
<td>Loosing time on redefining requirements because of new inquired confidential data.</td>
<td>As an employee of the TU/e and not of the company. The company spends some time deciding what info/data to give me.</td>
<td>Receiving new data/insights later than preferred can cause the necessity to redefine requirements and therefore delaying project progress.</td>
<td>Discuss changes in planning early on.</td>
<td>Started (23/02/2017) Completed (19/04/2017)</td>
</tr>
<tr>
<td>9</td>
<td>The SMD being occupied and or hardware being maintained.</td>
<td>The SMD is used in various projects and equipment does get damaged every now and then.</td>
<td>Repairing damaged equipment implies less time for testing on the SMD. As a result project progress is stagnated.</td>
<td>Be flexible: Shift to other tasks when setup occupied being repaired or adjust planning: e.g., shift vacation earlier or later.</td>
<td>Started (23/02/2017) Completed (22/08/2017)</td>
</tr>
<tr>
<td>10</td>
<td>Not meeting different stakeholder wishes.</td>
<td>The company project manager prefers SMART (Specific, Measurable, Achievable (Attainable), Relevant (Realistic), Time phased (Timely)) requirements while the company supervisor prefers a more proof of concept approach.</td>
<td>When differences in stakeholder wishes are not discussed a non-optimal solution is delivered.</td>
<td>Discuss differences in opinions to come to a compromise.</td>
<td>Started (19/04/2017) Completed (22/08/2017)</td>
</tr>
</tbody>
</table>

**Table 5 Risk Analysis**
### 6.4 Stakeholder Management

In this section, stakeholder management is briefly discussed. Figure 50 shows the various stakeholders in this project. The figure illustrates the level of power the stakeholder contains for influencing the project direction. Next to that, the stakeholder’s level of interest in the project is shown. Four categories can be distinguished:

- **High power – high interest:** To this category belong the company supervisors. As the project is executed for them, they have most interest and power in the direction the project takes. They are managed closely by involving them in both the PSGMs and the weekly company meetings.
- **High power – low interest:** The TU/e (Eindhoven University of Technology) supervisors belong to this category. To ensure that the project evolves according to academic standards, the TU/e supervisors are allowed to practice decent power through steering. To keep them satisfied, they are involved in the PSGMs.
- **Low power – high interest:** The company client is in this category. As they are the company’s client, naturally they have considerable interest. They are kept informed through the company supervisors and occasionally through demos.
- **Low power – low interest:** Finally, the program coordinator is monitored and kept informed by including him regularly in emails and meetings.

<table>
<thead>
<tr>
<th>Power</th>
<th>Interest</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Keep satisfied:</strong></td>
<td><strong>Manage closely:</strong></td>
</tr>
<tr>
<td>TU/e supervisors:</td>
<td>Company supervisors</td>
</tr>
<tr>
<td><strong>Monitor:</strong></td>
<td><strong>Keep informed:</strong></td>
</tr>
<tr>
<td>Program coordinator</td>
<td>Company clients</td>
</tr>
</tbody>
</table>

*Figure 50 Overview stakeholders*
Conclusions
In this study, the weaving machine shedding system with relaxed mechanics, R7, is mimicked on the SMD setup in Flanders Make. The relaxed mechanics involve reduced shaft stiffness (R7.a) and increased gearbox backlash gap (R7.b). Although these can reduce hardware maintenance cost R4, they do provoke torsional vibration and backlash which deteriorate tracking. To compensate for these effects, improved control is applied (R8), i.e., feedforward torsional vibration compensation (R8.a) and feedforward backlash compensation (R8.b) respectively:

- The feedforward torsional vibration compensation technique, applied to the collocated side, reduces the tracking error at the respective side significantly, see Figure 37. Nevertheless, on the other hand, significant torque demand is requested to keep tracking when the shaft is excited at the anti-resonance. Although the error at the collocated side can be reduced, the relative error (w.r.t. the non-collocated side) is inevitable.
- The applied backlash compensation technique proved to work well, i.e., significant error reduction is achieved without extra torque demand, see Figure 42. The applied technique has its limitations as well. Compensating for a greater gap size is at the expense of proportional increase in torque demand or proportional decrease in the squared speed, see equation 27.

The applied improved control strategies do improve performance (tracking and power consumption) of the relaxed system. However, w.r.t. the benchmark system with tight mechanics, performance is still inferior, see e.g., Figure 42. Nevertheless, as intended, these results illustrate the possible trade-off between low hardware maintenance/cost on the one hand and high performance (i.e., good tracking and low power consumption/torque demand) on the other hand:

- For torsional vibration compensation this is illustrated in Figure 44. Trade-off: Reducing shaft stiffness (R7.a), increases tracking error and torque demand. Applying torsional vibration compensation (R8.a) partially recovers tracking (R5) but at some instances (i.e., at anti-resonance) further increases torque demand (R6).
- For backlash compensation this is illustrated in Figure 45. Trade-off: Increasing gap size (R7.b), increases tracking error and torque demand. Applying backlash compensation (R8.b) partially recovers both tracking (R5) and torque demand (R6).

These trade-offs support/allow improvement of the weaving machine in terms of reduced cost, R1, and equipment effectiveness, R2.

Recommendations
Figure 46 gives an overview for recommendations on how to improve the developed system. System performance can be improved through improvements on hardware and control (possibly combined):

Hardware design
Recommendations on hardware involve reducing the effective arm of the load mechanism, reducing shaft flexibility, alternatively further reducing shaft stiffness and finally applying additional hardware (torque sensors, motors):

- Reduce effective arm (see equation 10): With a small effective arm of the non-linear load mechanism, the effective inertia reduces (see equation 6) and consequently the resonance increases (equation 5). This can eliminate the relative shaft error caused by torsion.
- Reduce shaft flexibility: Increasing shaft rigidity to prevent exciting the resonances can help eliminating the relative error as well.
• Reduce shaft stiffness further: Here, instead of trying to reduce the relative error, as according to R21, the intention is to utilize the shaft relative error to the advantage of shed opening.

• Apply torque sensor: This is related to monitoring the gradual changes in model parameters so that they can be updated online.

• Apply second motor: Applying a second motor, i.e., a motor at non-collocated side, can resolve the issues with shaft relative error.

Control design:
Recommendations on control involve improvements on both feedforward and feedback control.

• Improve Feedforward: This involves operating at system resonance, increasing modeling complexity and model parameter optimization:
  o Operate at system resonance: This is complementary to the recommendation on further reduction of the shaft stiffness. Operating on system resonance for shed opening through shaft relative error is consequently at the cost of minimum energy input.
  o Increase model complexity: This is desirable, e.g., when dealing with the sudden parameter changes and the overlooked phenomena such as heddle friction.
  o Conduct parameter optimization: This is done for retrieving a model with improved capabilities for zero-crossings prediction. It is therefore desirable to first conduct a sensitivity analysis to see which parameters influence the prediction of the zero-crossings the most. Furthermore it is meaningful to gain insight on how much the prediction can deviate from reality before the applied control strategy fails.

• Improve feedback: This involves applying homing procedures, observers and optimal controllers:
  o Apply a homing procedure: This procedure is performed at machine startup to reset the system to a predefined initial condition (e.g. \( \theta_1 \)). The known initial condition enhances model prediction.
  o Apply observers: The application of observers has many advantages, some of which are: Gaining insight of the machine initial condition as well as compensating for effects not covered in feedforward such as non-modeled disturbances. Additionally, a very meaningful application of an observer is to, e.g., use encoder data to evaluate whether gap traversal was timed well. If timed poorly, high impact between backlash slider and case may occur. Such impact could be traced back from the encoder data.
  o Apply optimal control: In this study, s-curves are used for gap traversal, see Figure 20. Whether this is the optimal profile, i.e., smoothest and quickest, is subject to discussion.
  o Apply torque feedback: Compared to position feedback, applying torque feedback for gap traversal can prevent high contact impact.
8. Project Retrospective

This chapter involves a reflection on the approach and tools used in this project, as discussed in section 6.1. The first part of this chapter mentions some noteworthy gains for Flanders Make from the applied approach and tools. The second part discusses some shortcomings of the applied approach and tools and what Flanders Make missed consequently.

Gains
The gains worth mentioning, regarding torsional vibration- and backlash compensation, involve: (1) improved understanding, (2) a good feasibility analysis and (3) a good (control) system architecture:

1. The applied backlash compensation technique is inspired from literature as mentioned in section 3.2.1. Further improved understanding is acquired by extensive simulation in Matlab/ Simulink/ Simscape and experiments on the SMD.
2. A good feasibility analysis is conducted by applying the bottom-up approach. Doing early simulations and experiments helped detect redundancies early on. An initial concern with backlash gap opening or traversal has been: multiple unpredictable impacts between slider and case. However early simulations and experiments revealed that when (i) motor and load travel in the same direction and (ii) the system is not undamped, no unpredictable impacts and gap openings occur. This allowed to focus only on compensating predictable gap openings which consequently reduced time spent on researching the other cases. Similarly, concerns with model inversion were eliminated early on by initial simulations and experiments. Also here, consequently no time was spent (“wasted”) on researching these issues.
3. A good (control) system architecture is obtained by applying the system engineering/ Top-down/ V-model approaches. Process requirements, which form the basis for the architecture, are identified using CAFCR approach/ key-drivers diagram. As long as the identified process requirements hold, the developed architecture will remain valid.

Shortcomings
Three main shortcomings are identified for the applied approaches:

1. The first one involves the low pace of delivery of tangible technical results at the project beginning phase. Applying the system engineering approach, the first project tasks involve Project Initiation and Requirement Design, see Figure 47. The output of these stages is not focused on obtaining tangible technical results. That is left for the next stage, Design. The approach in Flanders Make (bottom-up) is different from this one. Therefore, from their perspective, they experienced low income of tangible technical results at the project beginning phase.
2. The second shortcoming involves non-optimal solutions as a consequence of following the work break-down structure, Figure 48, too strictly. Following the structure too strictly, restricted from zooming in and out often enough and coming up with alternative, optimal or out of the box solutions.
3. And thirdly, full deliverables have not been met. As discussed in section 5.2.2., implementation of the impact spring is left for future work. Again, this is a consequence of following the work break-down structure too strictly. By including the impact spring, backlash compensation becomes more challenging. For good results, a homing procedure is required as discussed in section 7. As the development of a homing procedure is not included in the work break-down structure, temporary deviation from/ adjustment of the work-down structure in an early stage could have been favorable.
List of Abbreviations & Glossary

SMD: Small modular driveline
FBD: Free body diagram
PSGM: Project Steering Group Meeting
TU/e: Eindhoven University Technology

\( h_u \): Thickness half disc
\( A_u \): Area half disc
\( F_{k_t} \): Tension spring force
\( F_{gl} \): Gravitational force on mass \( i \)
\( F_{kca} \): Preload impact spring
\( F_{stl} \): Preload spring
\( F_{react} \): Reaction forces slider-crank nodes
\( I_k \): Equivalent inertia knives
\( I_i \): Inertia load
\( I_m \): Inertia motor
\( I_{rs} \): Inertia rigid shaft
\( I_s \): Equivalent inertia slider
\( I_u \): Moment of inertia half disc
\( P \): Period corresponding to speed
\( T_{FB} \): Torque feedback motor
\( T_{FF,fs} \): Torque feedforward flexible shaft
\( T_{FF,l} \): Torque feedforward load
\( T_{FF,fs} \): Torque feedforward rigid shaft
\( T_d \): Torque oscillating load disturbance
\( d_s \): Torsional damping shaft
\( e_m \): Tracking error motor
\( e_{ms} \): Tracking error motor speed
\( k_h \): Stiffness harness heddles
\( k_c \): Stiffness constant compression spring
\( k_s \): Torsional stiffness shaft
\( k_t \): Stiffness constant tension spring
\( l_c \): Length crank
\( l_r \): Length connecting rod
\( m_{u} \): Mass half disc
\( m_c \): Mass crank
\( m_k \): Mass knives
\( m_r \): Mass connecting rod
\( m_s \): Mass slider
\( r_u \): Radius half disc
\( os \): Torque measurement offset
\( p \): Percentage of slider stroke
\( p \): Model parameters
\( t_{zc} \): Zero-crossing instance
\( w_{a_b} \): Width (angular) acceleration backlash correction
\( w_{at} \): Width (angular) acceleration load

\( y_u \): Center of mass for half disc
\( \alpha_b \): (Angular) Acceleration backlash correction profile
\( \alpha_l \): (Angular) Acceleration load reference
\( \theta_b \): (Angular) Position backlash correction profile
\( \theta_{lo} \): (Angular) Position load initially
\( \theta_{ls} \): (Angular) Position load measured
\( \theta_l \): (Angular) Position load reference
\( \theta_{io} \): (Angular) Position load initially
\( \theta_{ms} \): Measured (Angular) Position motor measured
\( \theta_m \): (Angular) Position reference motor
\( \rho_u \): Density half disc
\( \omega_l \): (Angular) Velocity load desired
\( \omega_{ms} \): (Angular) Velocity motor measured
\( \omega_m \): (Angular) Velocity motor reference
\( h \): Height difference between connection of crank and connection of slider
\( G \): Shear modulus
\( T \): Torque demand motor
\( bg \): Backlash gap
\( e \): Tracking error (load)
\( l \): Length
\( r \): Radius
\( res \): Encoder resolution

\( \frac{1}{s^2} \): Double integral
\( \frac{d}{dt} \): Derivative
\( G^{-1}(\theta_m) \): Plant inverse
Bibliography


Appendix

Model parameter optimization – slider-crank

In this part of the appendix, the slider-crank parameters $p$ are optimized such that the model $\hat{T}_d$ matches the experiments $T_{d,meas}$ as close as possible. Model output $\hat{T}_d$ is a function of the measured angular position $\theta_{l,meas}$, velocity $\dot{\theta}_{l,meas}$ and acceleration $\ddot{\theta}_{l,meas}$ and the parameters $p$. The optimal parameters $p$ correspond to the least square solution between $\hat{T}_d$ and $T_{d,meas}$, see equation (35):

$$
\begin{align*}
\min_p |\hat{T}_d - T_{d,meas}|^2 & \text{ such that } \{Aeq \cdot p = beq \\
lb \leq p \leq ub \}
\end{align*}
$$

(35)

$$
\hat{T}_d = f(\theta_{l,meas}, \dot{\theta}_{l,meas}, \ddot{\theta}_{l,meas}, p)
$$

$$
p = [\theta_l(0) \ l_c \ l_r \ h \ k_t \ m_s \ m_r \ m_c \ F_{k,0} \ os]
$$

This problem is solved numerically using the Matlab function “fmincon”. The verified slider crank design specifications (Table 4) are taken as the starting point $p_0$:

$$p_0 = [3.1416 \ 0.05 \ 0.3 \ 0.07 \ 150 \ 1.1 \ 0.25 \ 0.175 \ 10.5 \ 0]$$

Searching for the optimal parameters is done within lower and upper bounds on the specifications:

$$lb = l p_0 \text{ with } l = 0.75$$

$$ub = u p_0 \text{ with } u = 1.25$$

Additionally, a linear equality constraint is set for system total mass:

$$Aeq \cdot p = beq \iff m_c + m_r + m_s = 1.6$$

The model parameters are optimized for moving the slider crank at a speed slightly fluctuating from 5[Hz], see Figure 51. Figure 52 shows the plant (measured torque), the nominal model (based on design specifications) and the optimized model (optimal parameters):

$$p = [3.1208 \ 0.0476 \ 0.2620 \ 0.0525 \ 150 \ 1.2375 \ 0.1875 \ 0.175 \ 12.7683 \ -0.0323]$$
Figure 51 Measured angle, velocity and acceleration

Figure 52 Comparison: Measured plant, nominal model & model with optimized parameters
Simulation - Complete system validation
Validation of the complete system involves moving the slider-crank mechanism (including impact spring) while compensating for backlash and torsional vibration. Simulation results for moving the slider-crank at 3 [Hz] are shown in Figure 53. Consistent with Figure 41, the applied techniques work well.

It is also worth mentioning that, at lower speeds, i.e., when spring load dominates over inertial load, see Figure 54, torsional vibration- and backlash compensation do not interfere with each other. Backlash is compensated at the zero-crossings, i.e. when disturbance load is zero. This occurs when the effective arm is zero, implying that the equivalent inertia is zero. Therefore, shaft resonance is increased and the shaft is less sensitive to impacts. Section 4.1.3. is complementary to this inference. At higher speeds, i.e., when inertial load dominates over spring load, see Figure 55, not every load zero-crossing is correlated with zero equivalent inertia.
Figure 53 Simulation: Torque demand (left) and tracking performance (right)
Figure 54 Normalized equivalent inertia and slider-crank load at 1[Hz]

Figure 55 Normalized equivalent inertia and slider-crank load at 5[Hz]
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