Date July 27, 2018
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Your reference n/a
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Reviewers
Wind Energy Science

Dear Reviewers,

First of all, the authors would like to thank the reviewers for their positive and constructive feedback. We believe that the comments help us to significantly improve the quality of the paper. The objective of this document is to respond to the points raised by the reviewers (blue) and to provide an overview of the actions that are taken (red). When in this response the authors refer to adjustments in a particular section, figure or table, we ask the reviewer to refer to the marked-up manuscript version to evaluate the changes.

The document consists of five sections, each addressing the comments of the reviewers separately.

Yours sincerely,

Sebastiaan Paul Mulders
Niels Frederik Boudewijn Diepeveen
Jan-Willem van Wingerden

Enclosure(s): Response to comments of Reviewer 1
Response to comments of Reviewer 2
Response to comments of Reviewer 3
Response to comments of Reviewer 4
Response to comments of Reviewer 5

Response to comments of Anonymous Referee #1

Reviewer 1 comments: This paper presents an interesting analysis of a novel hydraulic wind turbine concept. The concept consists on replacing the conventional mechanical drivetrain components with a seawater pump directly driven by the wind turbine, whose outlet flow is directed to a Pelton generator. However, the analysis presented in the paper refers to an intermediate solution, in which the seawater pump is driven by a close loop oil-based hydrostatic transmission. The paper topic is certainly relevant for the journal; the approach is rigorous and the authors appear to be very familiar and qualified for work in the field. However, the paper could be significantly improved in certain aspects. Therefore, this Reviewer recommend its publication only after major changes are implemented to the submitted manuscript.

The authors thank the reviewer for his/her thorough review, invaluable comments and remarks. The considerations are especially informative as in particular considerations are raised on the analysis and justification of the hydraulic drivetrain and its components. Processing these comments very much helped the authors in closing the gap between system design and mathematical evaluation of the employed turbine with hydraulic transmission.

1. [Major] The paper is quite long, it contains too many equations and figures. This Reviewer suggests the authors to reduce the number of equations and figures. Some suggestions are provided in the following comments. Consider also that this Reviewers is asking for some additional details, therefore some additional figures might be necessary in the revised version of the paper.

   The authors agree with the fact that this paper is quite long. In the revised version, this issue is addressed by revising the relevance of all content. We are grateful of the reviewer suggestions, and the authors will take these remarks into account.

   The most notable changes are noted:
   - Section 2.2 is extended to provide a more detailed description of the DOT500 prototype set-up;
   - Section 3.2.1 is shortened, now only showing the results. The derivation is moved to the appendix;
   - Section 4.2.2 is shortened.

2. [Minor] Not sure about the significance of fig. 2, since the concept is quite obvious. In any case, if the authors decide to keep this image, this Reviewer suggests to include labels for the different components represented

   Thank you for this comment. The authors agree that this figure is not of direct scientific relevance, however, it presents a nice comparison of the (potential) space-advantage of a wind turbine with hydraulic drivetrain.

   The authors decided to keep the figure, but included a description of the distinct components, as suggested by the reviewer.

3. [Major] The authors provide a quite exhaustive overview of the past effort, which is very appreciated. However, at pag. 2, they affirm "To date, none of the above described full hydraulic concepts made its way to a commercial product. All concepts use oil as the hydraulic medium because of the favorable fluid properties and wide component availability, but therefore also need to operate in closed-loop." This Reviewer has two problems with such sentence:

   - For the size of the components required for wind turbine applications, there are almost no available commercial products. Those chosen by the authors in their work are probably the among the very few ones available (considering also that they had to turn a motor into a pump!). This is because as the authors stated, there are no successful application for hydraulic wind turbines. Therefore, if there is no market (thus no demand), there is no offer. The message is that nowadays someone wants to design a hydraulic wind turbine, he/she necessitates to design the hydraulic components as well (or partner with a component manufacturer to get a unit specially designed).

     The authors agree with the reviewer on this point. DOT is founded with the philosophy that offshore wind can be simplified and exploited more efficiently by centralizing energy production. However, to date, a water pump with capabilities to operate under high load and low speed is not commercially available. DOT is very aware of this, and is therefore (1) working to develop a seawater pump by their selves, and (2) actively cooperating with renowned (water) pump manufacturers on the development of such a pump.

     The in-field tests with an intermediate drivetrain including a closed oil circuit, is the first step towards the final DOT concept. The goal of DOT therefore is to abandon the use of oil all together by making wind turbines to cooperate and use what is abundantly available offshore: seawater. The authors (and DOT) do realize that a lot of hurdles need to be overcome before this goal becomes reality.

- The reference to close loop (close circuit ?) hydraulic transmissions (HTs) is questionable. HTs for many mobile applications (wheel loaders, excavators, etc) are close loop, but again the components for these HTs are too small for wind turbine applications. HTs can also be open-loop. Many HTs for areal platforms, forklifts, hydraulic fan drives are open-loop. What are the requirements that determine the need of having a close loop HT for a wind turbine application? This Reviewer can have some guesses, but this should be better addressed.

  The point raised by the reviewer is valid. A closed circuit drivetrain for a wind turbine utilizing oil as the hydraulic medium is needed because:
  (1) Operating a turbine with oil at remote offshore locations might pollute the environment in the event of a calamity;
  (2) Not having to provide a continuous fresh oil supply to the circuit;

  A consideration for operating in closed circuit is cooling of the hydraulic medium. When losses in hydraulic components are significant and the natural convection of heat to the surroundings is insufficient for cooling, an additional cooling circuit needs to be incorporated.

The authors recognize that the statement is posed too strong and lacks further explanation. Therefore, the comments of the reviewer and our considerations are processed in the revised version of the introduction. Furthermore, we made sure that we reserved the term "circuit" for hydraulic matters, and "loop" for control purposes.

4. [Minor] Was the concept of the paper presented also at the IFK2018 conference? A better reference to that paper, and the novel contents of this paper, should be provided.

   Yes, this is correct. We did not yet include a reference to the conference paper, as it was not published at time of submission of the WES manuscript.

   The authors included a reference to the conference article in the introduction, and stated clearly what the contribution of this paper is as opposed to the conference article.

5. [Minor] figure 3 might not be necessary, can be removed.

   Thank you for this comment. Because of the length and increased complexity of this paper we included a paper organization flow chart. However, providing both a textual and graphical outline might indeed be redundant.

   The figure is removed from the manuscript.

6. [Major] Section 2.2. The hydraulic circuit needs to be better detailed. A more realistic ISO schematic with respect to the one provided in fig. 6 is needed. The authors give the impression that a pump can be simply be coupled with a motor to form a close loop HT. However, other components are needed to guarantee the operation of the system:

   The authors agree with the reviewer that the hydraulic diagram should be better detailed. The included simplified hydraulic diagram was a trade-off between complexity and relevance for the drivetrain modeling provided in the manuscript. In the real-world set-up, numerous additional components were in place for turbine operation.

   Reconsidering the performed trade-off, the authors revisited the hydraulic diagram by including vital components. The components that are not included/considered for modeling and analysis are presented in gray. In this way, the authors think that the updated diagram provides a middle course from a system design and theoretical modeling point of view.

- Is a charge pump present? How was that sized? Can be neglected in the analysis? Why? What is the pressure level of the low pressure line? Is a flushing valve / cooling of the hydraulic motor present / necessary for the long operation of the system? A HTs for continuous operation usually necessitates for a significant oversizing of the charge unit for cooling purposes.

  Correct, charge pumps are presents in both the oil and water circuits. The oil charge pump was sized in such a way that a sufficient flow with a constant (controlled) feed-pressure of 21 bar to the oil pump could be delivered. Additionally when cooling was required, the charge pump supplied more flow to be directed through the parallel cooling circuit connected by a pressure relief valve (see updated hydraulic diagram). For the water circuit, the centrifugal charge pump provided a lower charge pressure of 2.6 bar. This difference in charge pressure is due to the different pump types used: the radial piston oil pump (motor) requires a higher charge pressure, as this is used to actively push the piston bearings to the cam ring; whereas the water plunger pump largely alleviates this requirement.

  As for modeling purposes of the closed oil circuit only pressure differences over the hydraulic components (taking into account mechanical/volumetric efficiency losses) are considered, the feed pressure is left out from the analysis. For the open water circuit however, the feed pressure is neglected because of its low value and for convenient derivation of the passive torque control strategy.

  Cooling equipment was indeed necessary for the closed oil circuit, and a flush oil cooling circuit was in place to ensure long-term operation and prevent the working fluid from overheating.

  The updated manuscript now provides a more detailed description of the aspects discussed above in Section 2.2.

- Circuit of the water pump. The authors say that there is an external centrifugal pump, which seems to be connected in series with the fixed displacement pump. How is the schematic? Is there a relief valve in between to provide a reference pressure level? Why this part can be neglected in the subsequent analysis?

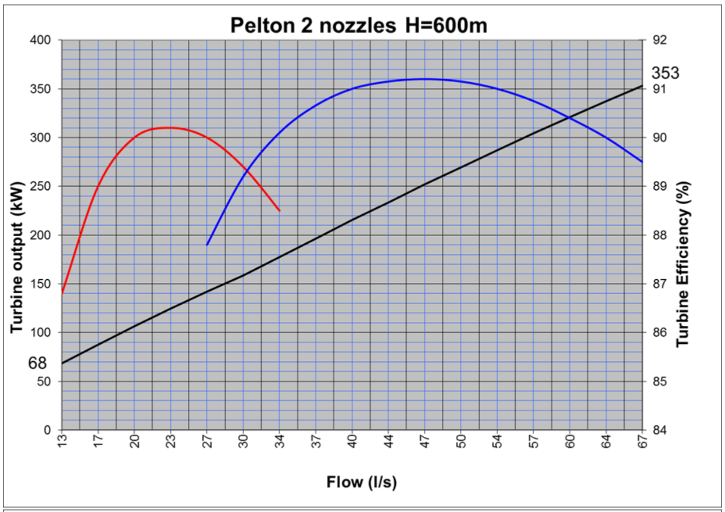
  The reviewer is correct, the centrifugal pump is connected in series to the water pump, and the updated hydraulic diagram presents the working principle. To prevent a disturbed flow entering the water pump, a two-reservoir set-up is used. The speed of the centrifugal pump is controlled to maintain feed pressure of approximately 2.6 bars. The charge pump is enabled before the water pump starts speeding up to ensure feed-pressure and thus to avoid cavitation. The low-pressure side of the water circuit does not contain a pressure-relief valve, as the water plunger pump allows for a direct feed-though of the flow; the high-pressure side however does include a pressure relief valve.

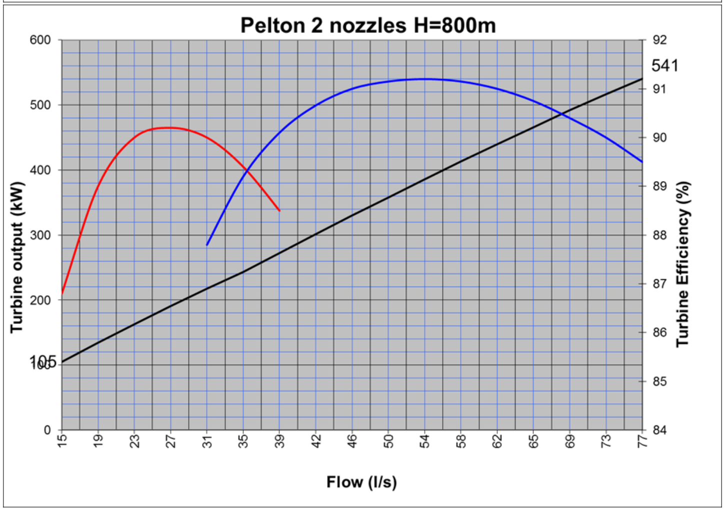
  Section 2.2 is updated with the details provided in the response to the reviewer.

- Pelton Turbine. The concept of using a Pelton Turbine is very interesting. However, it seems that the Head [m] of this turbine is way above to the existing Pelton turbines, so that it might be impossible to borrow an existing design. What is the specific speed of the Pelton Turbine of this paper? Is a commercial Pelton wheel available? Is a two-jet turbine such as the one of fig. 5 sufficient? This is not the scope of this paper, however, the authors could be more clear on this part, perhaps using more references.

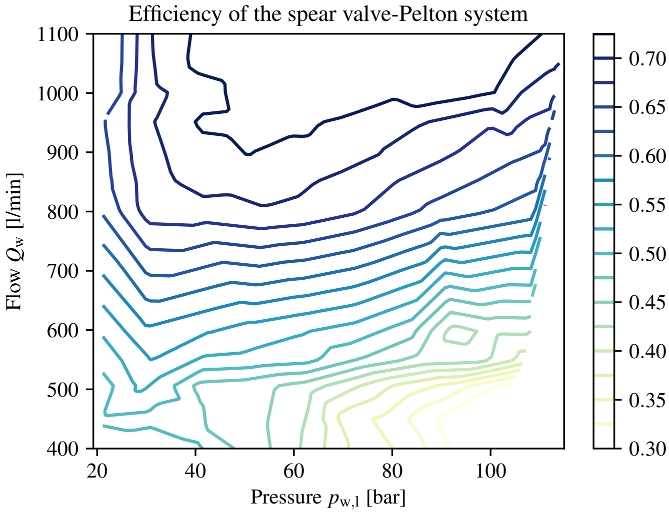
  Thank you for raising this comment, we could have been more clear on this aspect. Pelton turbines are highly specialized pieces of equipment and need to be design for specific condition requirements [1]. The Sy Sima 315 MW turbine in Norway, for 88.5 bar of head pressure is to date the largest known [2]. The employed custom manufactured Pelton turbine for the DOT500 is designed to match the nominal pressure and the speed conditions of the connected electrical generator.

  A custom-made Pelton turbine is designed such that the efficiency is optimal under the expected operational conditions. For this, the turbine is designed for optimal operation using 2 spear valves, subject to a nominal flow of 58 l/min. Graphs of the turbine manufacturer (given below) show that the efficiency is primarily a function of the supplied flow, and to a lesser extent of the head. The red line indicates operation with 1 spear valve, the blue line 2 spear valves.





The efficiency aspect is confirmed later by experiments executed by DOT, of which an efficiency evaluation figure is also given below. In the figure an evaluation of the combined spear valve-Pelton efficiency from hydrostatic fluid to mechanical power at the generator axis is given as a function of flow and pressure. Whereas during the experiment the flow was not sufficient to explore the overall characteristics, the results clearly show that the steepest partial efficiency gradient goes with flow; at higher flow rates the gradient with respect to pressure becomes negligible.



In the updated manuscript, more information on the custom-design Pelton turbine is given. Also the nominal operating conditions are discussed, and relevant references are included.

7. [Major] At page 7, the authors say After the water flow exits the spear valve, the aim to operate the Pelton turbine generator combination at maximum efficiency is a decoupled control objective from the rest of the drivetrain, and is outside the scope of this paper. Actually, this sentence is at the basis of many assumptions made in the development of the model and the controller design. This Reviewer, although without specific experience in designing HTs for wind turbine applications, has some conceptual doubts on this choice made by the authors. A HT has to be designed according to the features of both the load and the prime mover; this also drives several choices of constant torque (variable displacement pump) or constant power (variable displacement motor) HTs. In this case, the authors decide to neglect the features of the user (the Pelton wheel). Is that correct? To this Reviewer, it is like affirming that all the points that satisfies Eq. 12 (relation nozzle area and HT pressure) are indifferent for the Pelton turbine. This is quite hard to believe. The Pelton turbine should have preferred operating points that the HTs should be able to handle. This is a very basic question that the authors should address properly. Otherwise their proposed controller might not be beneficial on a real application.

   The reviewer correctly points out that the system design, as well as the applied control strategy should go hand-in-hand. Changing the operational strategy on the wind turbine side affects the operating point of the Pelton turbine and thereby for example the maximum amount of energy it can extract from the given flow. This is completely understood by the authors.

   The features of the Pelton wheel are not neglected. It is known from literature [3][4] that the ratio of tangential Pelton and water jet speed needs to be maintained at approximately $1/2$. As the Pelton wheel is mechanically coupled to an asynchronous generator, which can change its operational speed, a pressure measurement is used to determine the most favorable (speed) operating point to be as efficient as possible, given the conditions it is subjected to. However, the Pelton wheel will in the given set-up always be subjected to varying conditions, and thus suboptimal operation in the considered drivetrain using fixed-displacement components. This is for now a design choice, and further research needs to be conducted to elaborate on Pelton design and efficiency maximization given the varying operational conditions.

   The point of which operational path is most efficient, given varying Pelton conditions, remains. Operation at $C_{\tau,\max}$ will result in higher pressures for equal flows when compared to $C_{\mathrm{p},\max}$ operation. As was concluded in the previous question, the main driver determining the Pelton efficiency is the flow it is subjected to, whereas the head has negligible influence.

   The above given considerations are included in the manuscript in Section 2.2.

8. [Major] Section 3.1.2. The authors here affirm "the volumetric efficiency of a pump or motor is generally high and fairly constant over the entire operating range". For a simplified model the assumption of constant efficiency could be a fair starting point. But the statement that hydraulic pumps and motors have a constant volumetric efficiency for any pressure and shaft speed is clearly wrong. Otherwise, all the literature on empirical efficiency models (starting from Merritt in the 60s), standards for measuring volumetric efficiency (ISO, etc), tribological models for studying the lubricating gap flows, would not be justified. Particularly at low speed, the volumetric efficiency can be particularly low for both pumps and motors. Please revise this statement and better justify the assumption of constant efficiency, which can be very limited.

   The reviewer makes a very valid point, and we agree with it. The reason we have chosen to assume a constant volumetric efficiency factor is (1) the fact that for most of the given components, no volumetric efficiency data is available, and (2) the aim is to provide a simplified model of the hydraulic drivetrain.

   The assumption of a volumetric efficiency is revised in the updated manuscript.

9. [Minor] Section 3.2.1. here there are several equations that are well known. This section could be reduced.

   Thank you for this comment. The authors recognize that Section 3.2.1 is lengthy and reduced it. Now, only the major results are presented. However, we would like to be as complete as possible, seen the journal we are publishing in is not primarily focussed on hydraulics. Therefore, we moved the derivation of the results to the appendix.

   Section 3.2.1 is shortened and the derivation of the results has been moved to the appendix.

10. [Major] Section 3.2.2. In the list of assumptions it is stated that the inertia of the hydraulic components is neglected. While it is true that hydraulic components have fast dynamic, in comparison with other technologies for transmitting power, it has to be proven that within a hydraulic system the hydraulic line is the element with slowest dynamic. This statement, in general is not true, and Merritt never affirmed that. Moreover, the authors consider infinitely rigid lines, therefore the fastest lines possible (is this realistic?). Please justify this statement.

    Thank you for this comment, the authors agree with the reviewer that the (slow) line dynamics cannot be discarded. As exact specifications of the line bulk modulus are unknown (not publicly available), we decided to take a reasonable value of $K_\mathrm{l} = 0.8$ GPa from [5], which is twice as low as the bulk modulus value taken for the oil column. The equivalent bulk modulus is calculated by the relation $K_\mathrm{e} = (1/K_\mathrm{f} + 1/K_\mathrm{l})^{-1} = 0.52$ GPa. The equivalent value is used subsequently in the remainder of the paper.

    The assumption of infinitely rigid lines is removed from the analysis in Section 3.2.2, and Appendix A is updated to include relation for calculating the equivalent bulk modulus. The resulting equivalent modulus is subsequently used in the analyses throughout the different sections of the paper. Furthermore, Section 3.1.1 of the manuscript is updated, and now includes a more elaborate justification on why the drivetrain component dynamics are neglected and assumed as analytic expressions.

11. [Major] Section 4.1.1. The authors say "hydraulic components are known to be more efficient in high-load operating conditions, it might be advantageous for a hydraulic drivetrain to operate the rotor at a lower tip-speed ratio". This statement can be arguable. first, shaft speed has a major effect, and not all units have a clear trend with load. Can the authors provide the overall efficiency plot for the commercial units they utilize (even in normalized form?). This is very important, because all the controllers of case 1 and case 2 are based on this assumption!

    The authors agree with the comment made by the reviewer, and the statement the reviewer refers to is changed. However, the manuscript already includes (mechanical) efficiency data for the oil pump and motor, given in Figure 13. The figure includes the proposed operational strategies (case 1 and case 2), and a steady-state analysis of the total drivetrain efficiency for both strategies is given in Figure 14.

    The titles of the plots in Figure 13 and the legend in Figure 14 are now updated, to make their purpose more clear. Also, the statement the reviewer refers to is adjusted, and an additional consideration on the efficiency aspect is made in Section 4.1.2.

12. [Minor] Pag. 21. The reference to fig 14 might be wrong, since the figure refers to mechanical efficiency.

    Thanks for pointing out this mistake. The reference should be pointing to Fig. 13.

    The reference is corrected in the revised manuscript.

13. [Major] 4.2.1. L=50m... are the pump and motor connected by a 50 m straight line? If there are line discontinuities, some terms, particularly the inductance terms, can be entirely wrong

    Indeed, apart from a swivel which enables continuous yaw motion located below the nacelle, the high pressure lines have no discontinuities, and run from the nacelle all the way to the oil motor located in the monopile. Furthermore, the rotor inertia, which is expressed in terms of hydraulic induction, is predominant in the lumped induction term. The contribution of the hydraulic inductance term is thus negligible, and discrepancies would have a negligible effect on the analysis.

Response to comments of Anonymous Referee #2

Reviewer 2 comments: Thank you for your submission. In general I found the paper to be interesting and well-presented. Background literature was complete and informative, and the introductory material explains well what this paper adds to the growing literature. The figures and illustrations are particularly well done and helpful. The writing is clear, with only very few grammatical errors. The paper is well structured, such that one new to hydraulic drivetrain wind can follow. Finally, the inclusion of field results and comparing to the theoretical work is informative. Excellent work.

Thank you, the authors are pleased to read this!

Therefore, only a few comments to be given, overall:

1. Is there provided, or could the authors provide, a quick impression of how the efficiency of the proposed system, in total, would compare to a similar conventional system? For example, given the same rotors, what would a standard efficiency to final electrical power be (90%?) and what would it be for this system?

   For the described DOT prototype, the total power transmission efficiency was predictably low, as a result of the double hydraulic circuit. In the below-rated region an efficiency is attained of $30 - 45\,\%$ depending on the operating conditions, whereas in the above-rated region a consistent drivetrain efficiency of $45\,\%$ is attained. As described in the manuscript, off-the-shelf components are used, of which the optimal efficiency operational envelopes do not match. The drivetrain has a fixed-volumetric displacement, which means that the pressure and flow changes according to the turbine operating conditions. Figure 13 shows that drivetrain components have a specific region in which they yield maximum efficiency.

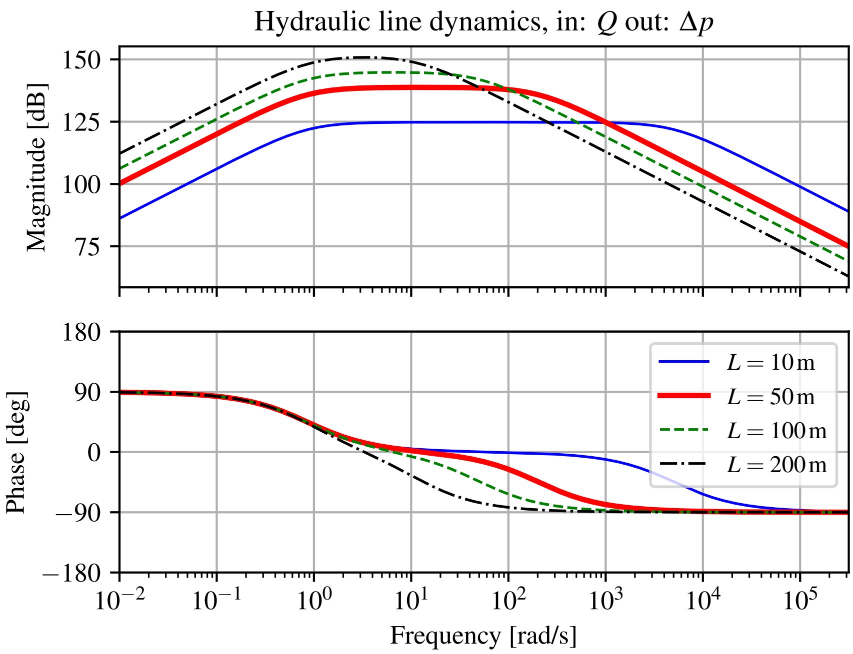
   The above given reasoning holds for the described prototype. It is however yet unclear what the drivetrain efficiency of the final DOT concept will be, as the seawater pump is still under development. An earlier PhD thesis on hydraulic wind turbine networks [7] provides an estimate on the overall conversion efficiency of conventional and hydraulic wind turbines of $82 - 84\,\%$ and $70 - 80\,\%$, respectively.

   The efficiency numbers attained with the intermediate DOT500 prototype are added to Section 2.1.

   Specific:

2. 3.2.1: "and vice versa for the latter.." I could not fully understand what is meant by this. Could a Bode plot of the transfer functions be included to visualize the inverted notch functions?

The authors did separate the theory from the results. For this reason, in Section 3.2.1, only theory is provided, whereas in Section 4.2.1 an illustrative example is given, which considers the system and hydraulic properties of the DOT500 system. In the latter mentioned section, a Bode plot of the of the transfer function $G_{\mathrm{Q}/\Delta\mathrm{p}}(s)$ is given. For clarity reasons (and considering the length of the paper), a visualization of $G_{\Delta\mathrm{p}/\mathrm{Q}}(s)$ is omitted in the manuscript, but given in the figure below. It is shown that the inverted notch characteristics is still present. However, exciting the flow (instead of pressure) results in amplification/transmission to pressure in a wider frequency region for shorter line lengths. For longer line lengths, the amplification magnitude increases, but at a more specific interval. This effect is a result of the inverse proportionality between the damping coefficients ζ_{Q} and ζ_{p}.



In Section 3.2.1 the phrase "and vice versa for the latter.." is removed and replaced with a more convenient description. The paragraph now also references to the illustrative example in Figure 15.

3. Eq 38: The B-matrix in this version includes the inputs? There is a dot following the matrix but it is not clear what the dot product will be with? In eq 54 there are only 3 columns total for the 3 B matrices, but are there 4 variables provided in this version?

   The representation given by Eq. 38 is the rewritten form of the dynamic system derived in in Eqs. (33)-(37). There is no dot-product after the input vector, this is just punctuation to indicate the end of the sentence. The pressures Δp_{h} and Δp_{b} cannot be controlled directly. For this reason, the rotor torque and spear valve pressure characteristics are evaluated and linearized at different operating points. By doing so, a linear state space system defined in Eq. (54) is obtained. By substitution of the linearized characteristics, defined in Eqs. (46) and (51), the terms redistribute in the A, B and B_{U} matrices: some are defined in the state vectors, others can be regarded as control inputs or wind disturbance inputs in B and B_{U}.

   The authors hope to have clarified the unclarities, and slightly updated the section to improve readability.

4. Section 4.1.1: "advantageous for ... operate a lower tip-speed ratio", this is counterintuitive, but do I understand correct that although the rotor power will be reduced, the improved hydraulic efficiency will lead to higher final electrical power? Is this demonstrated conclusively?

   The reviewer is correct. Normal wind turbines operate the rotor at the maximum power coefficient, maximizing the efficiency in the below-rated region. For the DOT500, however, the wind turbine drivetrain is retrofitted, while retaining the original turbine rotor. As hydraulic components are in general more efficient in high-load operating conditions, we additionally perform an analysis for operating the turbine at the maximum possible torque coefficient. The maximum torque coefficient is located at a lower tip-speed ratio (lower rotor speeds, higher torques for equal wind speeds), and corresponds with a lower power coefficient.

   So indeed, from a aerodynamic efficiency perspective this is unfavorable, but from a hydraulic drivetrain perspective this might result in an overall efficiency advantage. In the subsequent section, an efficiency analysis is given on a component level for both operating cases in Figure 13, and an overall evaluation of the drivetrain efficiency is presented by Figure 14. The analysis takes into account the reduced rotor power coefficient for operation at maximum torque.

   The titles of the plots in Figure 13 and the legend in Figure 14 are now updated, to make their purpose more clear. Also the introductory paragraph of Section 4.1.1 is updated, and a concluding remark referring to the next section where the actual efficiency analysis is performed is added.

5. Section 4.1.2: Does the lower TSR also risk increased occurrence of dynamic stall?

   Thank you, this is a very good question. We cite the following phrase from [8]:

   "Stall on lifting surfaces is commonly encountered, mostly undesired, and occurs when a critical angle of attack is exceeded. Depending on the unsteady rate of change of the airfoil's angle of attack, static and dynamic stall are distinguished. (...) During dynamic stall, the shear layer rolls up into a large scale dynamic stall vortex which grows locally and temporally until vortex induced separation occurs. During static stall on the other hand, the shear layer rolls up continuously into large-scale structures that grow spatially."

   So indeed, there is an increased occurrence of dynamic stall, especially in turbulent wind conditions when the angle of attack continuously varies.

   Furthermore, from a discussion with a professor in aerodynamics from our faculty, it became clear that (dynamic) stall could indeed occur in the region of the blade root. Stalling of a larger blade would result in increased loading with a reduced power capture, however, as we are not stalling during normal operation, the effects on loads should be minor. He also clarified that dynamic stalling could even be slightly beneficial, as it introduces a dissipating/damping effect.

   We have to admit that we did not perform a detailed analysis on this aspect. The aim of the in-field test was to show the feasibility of the hydraulic drivetrain, and while we ensured safe operation of the turbine, effects such as dynamic stall were disregarded. The authors have noted the comment, and the effects of (dynamic) stall will be considered in later stages of the project.

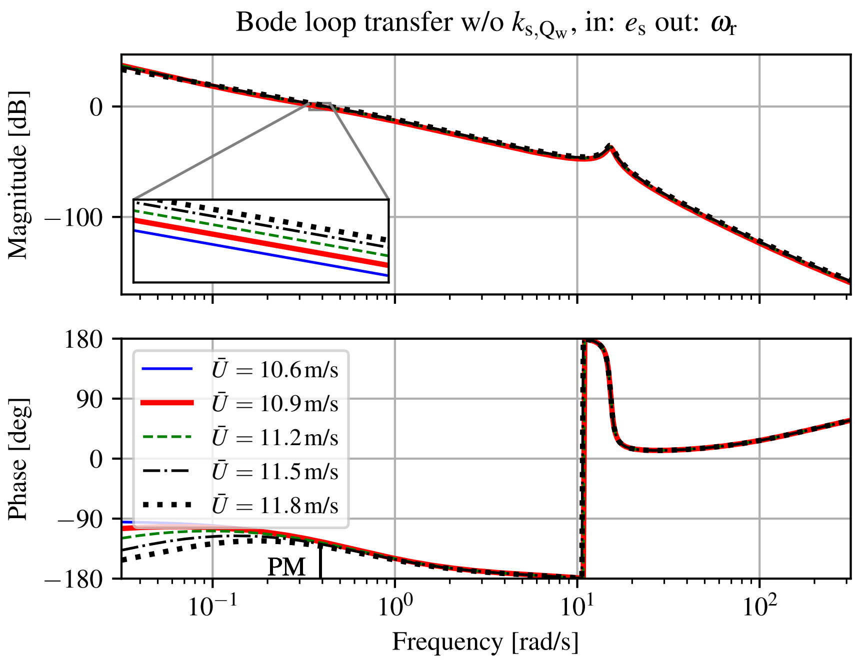
6. Fig 16: Title is incomplete

   Correct, thank you for pointing out. We referred back to the submitted manuscript, but there the title is correct. Somehow, processing of the manuscript during upload must have changed the title by accident.

7. Fig 18: The legend is hard to understand, why the lower-case bold "without" following the period? Since the phase margin is discussed later, could it be indicated in this figure?

   Thank you for pointing out this mistake, the point should be a comma. The suggestion of indicating the phase margin is taken, and both bode plots are updated.

Also, again, the upload process changed the figure, by omitting some symbols. The correct figure is shown below for reference.



The mistake in the caption is fixed, and the Bode plots include an indication of the phase margin (PM).

8. Fig 22: Color legend missing label

The point raised by the reviewer is not entirely clear for the authors. Figure 21 is a representation of the tip-speed ratio for a range of turbine operating points. This is also stated title and the caption of the figure. Adding an additional label to the color legend would be redundant in our opinion.

9. Fig 24: Good figure, just wish to confirm, is rotor speed scaled correctly? Does it always stay so close to its maximum? Or is this period special in that there is only a brief excursion into region 2 (which makes sense, youve selected a period covering 2,2.5,3) just want to be sure.

   Thank you for this comment. Yes, the rotor speed is scaled correctly, however, we took a part of the time-series where the environmental conditions were such that the turbine operated around region 2.5. What we want to show in this figure is how the spear valve torque controller (only active in region 2.5) works as expected and switches nicely to region 2 (no control), and region 3 (pitch control).

   The text is adjusted slightly such that the purpose of the figure is more clear.

   An interesting, and well-presented paper.

   Thanks again!

Response to comments of Anonymous Referee #3

Paper is very well written and subject matter thoroughly presented.

Thank you, the authors are grateful to hear this.

Some general comments: towards the beginning of the paper it is stated that "To date, none of the above described full hydraulic concepts made its way to a commercial product", this merits some justification as to why the concept of hydraulic transmission for wind turbines has not been commercially viable so far, and whether the presented work can potentially overcome these barriers to market.

The authors agree with the reviewer, and a similar points has been posed by Referee #1. We would like to refer the reviewer to the answer given in Question 3-2 in our response to the first referee.

We further elaborated on this point in the introduction of the manuscript according to the comments of both referees.

One minor comment would be to add some labelling of the components in figure 2.

Thank you for this comment. We agree that labelling of the components improves the quality and relevance of the figure.

The figure is updated accordingly, now including labels indicating the components.

Response to comments of Anonymous Referee #4

The research in the paper is original, well conceived, and of interest to the readers of this journal. The other reviewers have done a good job in providing a detailed review.

Thank you, we are grateful to read this positive comment.

I only have one comment. The reviews advocate maximizing the torque coefficient rather than maximizing the power coefficient for this particular case. They further demonstrate that this provide more energy for the system considered since hydraulic components in this particular case are more efficient at higher torque and lower speed.

It is important to point out that while the torque coefficient optimizing approach is advantageous in this particular case, it is not true in general. The overall efficiency of a hydraulic pump or motor is the product of the mechanical efficiency and the volumetric efficiency. Depending on the particulars of the unit and its operating conditions, either of these might be dominant. A truly rigorous approach would be to optimize the system power coefficient with all losses included. This would work for any case.

In their final version the authors must clearly state that the torque coefficient optimization approach they advocate is true in this case, but is not true in general.

The authors completely agree with the referee. Only two cases are considered in this paper, namely: operation at rotor maximum torque, and maximum power coefficient. For the specific case presented in the paper, it is found that the maximum torque case results in the highest overall drivetrain efficiency. However, this claim can by no means be generalized for other wind turbines with hydraulic drivetrains. A more rigorous approach would indeed be to optimize the ideal below-rated operational trajectory subject to all component characteristics. However, to perform a more concise analysis, only the two given trajectories are evaluated. For other set-ups it could indeed be the case that a different below-rated operating strategy is beneficial.

The consideration of the reviewer and our response is processed and included in Section 4.1.2.

Response to comments of Anonymous Referee #5

This is an interesting paper that deals with the use of hydraulic transmission systems to enable centralized conversion of wind power into electricity in offshore wind farms. A key advantage of this approach is the significant reduction in the nacelle weight. The paper is relevant and very timely given the present drive by industry to develop larger and heavier rotors. The following is a summary of the main comments that need to be addressed by the authors before the paper may be published:

The authors thank the reviewer for his positive comment and considerations raised.

1. Figure 5 may be deleted without affecting the quality of the paper.

   Indeed, the figure could be deleted. However, seen the journal we are publishing in is not primarily focussed on hydraulics, we think that the figure is insightful and serves readers from various disciplines.

   The authors would like to leave the figure as-is in the manuscript.

2. Table 1: It is also convenient for the reader to include the rated power of the motors

   Thank you for this comment, this is indeed a nice addition.

   Table 1 now includes the available power range of all the drivetrain components.

3. Page 6: The process of matching the pump, motor and Pelton turbine to the available wind turbine should be elaborated in further detail.

   Thank you for this comment. A description of the component matching process is not provided in detail in this paper, as the aim of the authors in this paper is to focus on the modeling and control design of the considered hydraulic drivetrain. One of the authors devoted his PhD to the system design of the presented drivetrain [9], and the authors refer the reviewer to this work for further details.

   The reference to the PhD thesis, including an elaborate description of the component matching process, is now also included in Section 2.2.

4. Page 9: specify the aerofoil data used in plotting Fig. 7.

   The presented power and torque coefficient curves are obtained from a Bladed wind turbine model which was shared confidentially with DOT. The model includes the requested airfoil data, but for reasons of confidentiality, this information cannot be shared publicly. However, the complete data set including the resulting power, torque and thrust coefficient tables (as a function of tip-speed ratio and blade pitch) is publicly available as an external asset [10].

   Section 3.1.1 now includes a reference to the externally available data set which includes power, torque and thrust coefficient tables.

5. Fig. 14: for ease of comparison, the two plots should have the same colour scale for the mechanical efficiency.

   We partly agree with the reviewer's remark. The color scales of the left and right plots range from 0.7 - 1.0 and 0.05 - 1.0, respectively. For plots with equal data (efficiency) ranges, we agree with the reviewer that equalizing them would enhance the readability. However, by using the same color bars for the presented plots would make the left plot less convenient to read by a lack of contrast (especially in grayscale). For this reason, the authors decided to leave the plots unchanged.

6. Figure 15: Possible design amendments to the system to enhance the overall conversion efficiency should be elaborated in further detail.

   A similar point is posed by reviewer #4, but focuses on the control aspect, and the authors refer the reviewer to our response in the previous section. In summary: for the considered system, only two scenarios are evaluated (maximum rotor power and maximum rotor torque trajectories) to provide a concise evaluation of both strategies on the overall drivetrain efficiency. Indeed, a more rigorous approach would be to optimize the ideal below-rated operational trajectory subject to all component characteristics.

   From a system design perspective, the presented prototype hydraulic wind turbine has the goal of showing the feasibility of a wind turbine with a hydraulic drivetrain, and is not meant to be kept as-is. In response to the first question of reviewer #2, we added the efficiency numbers of the current set-up to Section 2.1 ($30-45$ % below-rated, 45 % above-rated). Also, in Section 2.1, it is stated that the set-up allows for prototyping, and provides a proof of concept for faster development towards the ideal DOT concept. It is known that the additional components and energy conversions result in a reduced overall efficiency. For an overall increased efficiency, the amount of energy conversion steps need to be reduced. It is stated in the conclusion that by discarding the oil loop in the ideal DOT concept, only including a single water pump in the nacelle, the control design process is simplified and the overall drivetrain efficiency should be greatly improved.

   Regarding the changes already made to the manuscript, the authors think that the point raised by the reviewer is clarified.

Minor comments:

1. Figure 1 should ideally be presented on the same page where it is being referred to in the text.
2. Page 6, line 13 - remove coma after in such a way.
3. Eqt. (11) may be deleted as derived of Eqt (12) is well known.
4. Page 12, line 9 - remove coma after into the system.

Thank you for pointing out these minor remarks.

1. We agree with the reviewer, however, during typesetting the paper will be converted to a two-column format, and thus the complete mark-up will be changed again. We tried to make the figure positions as convenient as possible, but we are reticent on putting too much effort in this for now. However, we certainly keep this comment in mind for all figures during the mark-up of the final version of the manuscript.
2. Thank you, we corrected this.
3. We agree with the reviewer and deleted the equation.
4. Thank you, we corrected this.

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