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New method for the lubrication of wind turbine pitch gears using embedded micro-nozzles

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Abstract

The increase of power generated by wind turbines has increased the stresses applied in all of its parts, which causes the appearance of premature failures. In particular, pitch and yaw gears suffer of excessive wear, mainly due to inappropriate lubrication. This paper presents a novel method to automatically lubricate the wind turbine pitch gear during operation. A micro-nozzle to continuously inject fresh grease between the teeth in contact was designed, manufactured and installed in a test bench of a 2 MW wind turbine pitch system. The test bench was used to characterize the fatigue behavior of the gear surface using conventional wind turbine greases under real cyclic loads. The measurements of wear evolution in a pitch gear with and without micro-nozzle show a decrease of 70% of the wear coefficient after $2x10^4$ cycles.

Keywords: *Abrasive wear; Corrosive wear; Gears; Grease application; Lubricating greases; MEMS devices; Open gears; Oxidative wear; Power generation*.

1. Introduction

In recent years renewable energies, and more specifically wind power generation, is a sector that has grown in market share with significant energy generation. In the wind sector, this growth has led to the design and implementation of machines with greater power generation capacities. In order to design more efficient wind turbines, manufacturers are increasing the rotor diameter to capture more kinetic energy from the wind and increase energy generation. This leads to increased stresses and deformations in all parts of the wind turbine, such as the foundations, yaw, nacelle, tower and drive train unions. Most of these unions are static except for the drive train that transmits torque, the yaw system that turns the nacelle and the pitch system that turns the blade around its axis [1]. During dynamic operation of the wind turbine, the yaw and pitch bearings suffer from sequential traction inducing compression stresses that cause millions of micro-movements per cycle between the gear teeth. Therefore the thickness of the lubricating film between the teeth is reduced and direct contact between metal surfaces occurs. Wind turbine manufacturers have observed a new phenomenon appearing in high power wind turbines in form of excessive wear in the tooth located at the zero degree position in the pitch bearing (Figure 1). Currently, to slow this phenomenon the pitch gear lubrication at the zero degree position is carried out during un-wind periods or programmed stops. One main drawback with stops for lubrication is the important losses in electricity generation.

Figure 1 shows an example of the operation graphic for an ideal power curve of a 2 MW wind turbine at different wind speeds. In a wind farm with a common wind rose distribution, more than 60% of the operation time the pitch gear of the wind turbines works in the same position as for the case of low wind speeds, i.e. the zero degree position. When the wind turbine approaches its nominal power for high wind speeds the pitch system starts to actuate, resulting in a system stable during oscillations from the continuous movement due to the thrust and variations in wind velocity.

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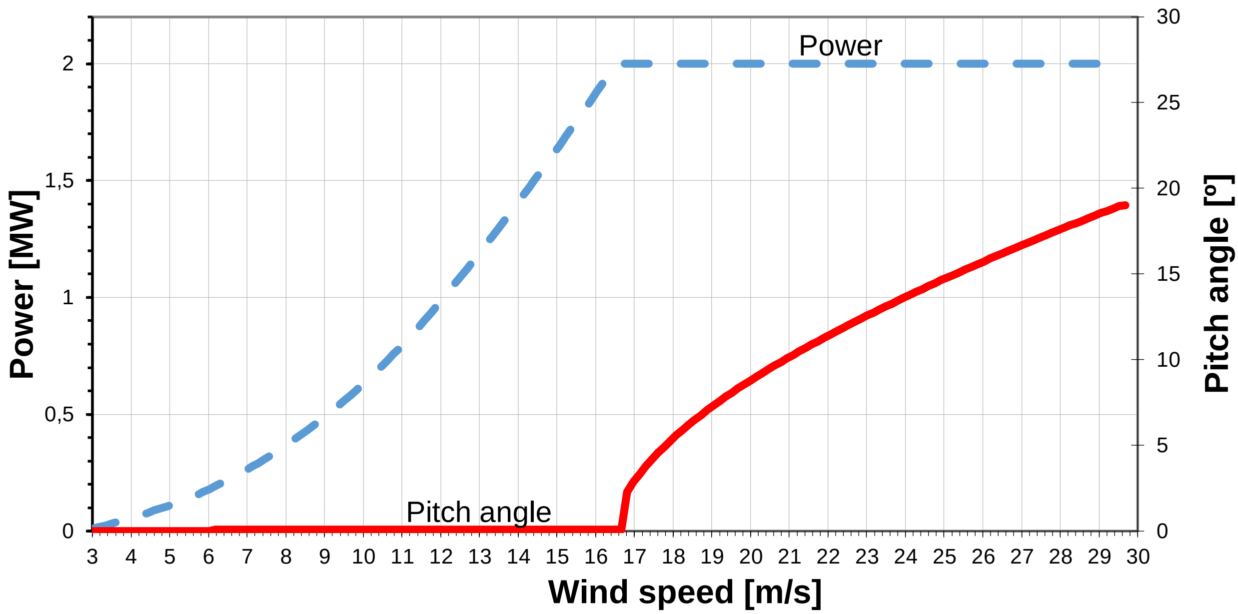

Figure 1: 2 MW wind turbine operation graphic. On the x-axis is the wind velocity and on the y-axis the wind turbine power generated: ideal power plot and pitch angle.

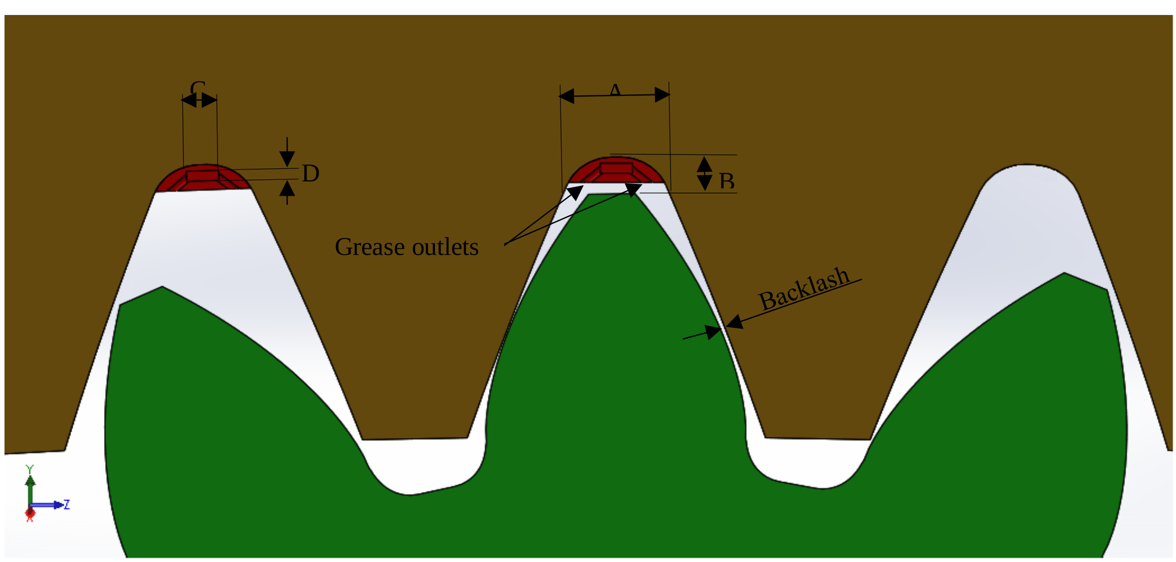

Figure 2: The gear cross-section according to DIN867; 1986 [15]. The gear geometry has a deddendum free space (dimensions A - B) where the proposed part is embedded. The dimension B is 0.15 times the deddendum space, which is 50% of the total volume.

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Focusing on the pitch systems, Gamesa Innovation [2] describes the different types of pitch systems that are currently installed in wind turbines, in form of the pinion-corona, hydraulic system and cogged belt respectively. The excessive wear in the pitch gear is only present in pinion-corona systems, and it is currently a major issue [3]. Furthermore, the tendency within wind turbine manufacturer is to increase the rotor diameter in order to capture a larger amount of the wind kinetic energy and thereby increase energy generation. This induce increased stresses and deformations in all parts of the wind turbine, even in non-static unions such as the drive train transmitting torque, the yaw system turning the nacelle, and pitch system which turns the blade around its axis. All these factors contribute to accelerate the wear at the gears.

During the dynamic operation of the wind turbines, the pitch bearing system repeatedly suffers from sequential traction and consequently compression stresses causes millions of micro-movements per cycle. This phenomenon together with long periods where the pitch is operated in same position produces friction between the teeth at the zero degree position due to the lack of a proper grease layer [3], [4].

Numerous investigations have established strategies to minimize the gear wear at specific positions like zero degree position in the pitch system. The proposed solutions essentially tune and modify the system parameters such as surface state [5], [6], stress, mechanical contact [7] and tribology [8]-[10] - parameters that Takadoum [11] defined as critical to minimize the effects of wear. There are methods to measure the excessive tooth wear to predict when the failure will occur were presented by Mashue et al. [12] and Nordex Energy [13]. There are also attempts to predict dynamic life of pitch and yaw bearing to help in the design of the maintenance plans [14]. Therefore, it is generally accepted that excessive wear in wind turbines gears is an unsolved challenge. Few researchers have addressed the problem from the mechanical perspective. The solutions presented in Refs. [5], [6] cannot however eliminate the preventive maintenance, and the solutions presented in Refs. [7], [8] require expensive parts in form another pinion and compressor. In Ref. [9] lubrication is done while the wind turbine operates causing generation loses and the solution presented in Ref. [10] is difficult to integrate in ongoing wind turbines due to the necessity of pinion replacement.

This paper proposes a novel pitch lubrication system which injects fresh lubricant directly to the contact tooth flank, and hence decreases the wear of the teeth by increasing the lubrication layer. The main advantage of the new system is that it extends the tooth lifetime and prevents stops for lubrication, which leads to an increased energy generation while decreasing maintenance costs. The approach we have used in this study aims to proof the decrease of wear in a pitch wear working under real loads, with a novel injector allocated in the gear deddendum, as shown in Figure 2 which injects fresh grease directly to the contact tooth flank. In order to develop this new concept of injector, micro-fluidics technology and the micro-manufacturing technology are used and a pitch test bench is used for validation.

2. Material and methods

2.1 Design Principle

Figure 2 shows a gear cross-section geometry according to DIN867; 1986 [15], highlighting its maximum dimensions A and B. According to the standard, B should be between 0.1 and 0.3 times the gear module and in specials cases up to 0.4 times the gear module. For the wind turbine field with pinion-corona pitch systems a module-12 gear is used for 2 MW turbines. The common free space (dimension B in Figure 2) is 0.3 times the module, i.e. 3.6 mm for a module-12 gear.

The new micro-nozzle was designed to fit in only 0.15 times the module space (50% of the total volume) to prevent contact between the teeth and the part during elastic deformation. The final dimension of the micro-channel is shown in Figure 2, with dimension D in the range of 500-1000 µm and C in the range of 1000-2000 µm. The length of the injector depends on the gear width.

2.2 Grease Fluency study

The grease fluency was investigated using micro Particle Image Velocimetry (µPIV) to analyze the flow behavior of a NLGI grade 2 grease in a micro-channel fitted in a 12-module gear. The experimental setup is presented in Figure 3a, while the principle of µPIV is shown in Figure 3b. The µPIV system is comprised by a pulsed laser which is used to illuminate a volume (micro-channel in this case) with a tracer particle seeded material/fluid. Images of the seeded flow were taken with a specified time step, and through a correlation algorithm the motion of the particles were calculated whereby the flow could be visualized through e.g. a vector field. Compared to 'macro' PIV where the flow was measured in a single plane set by the laser light sheet as shown in Figure 3b, the whole domain is illuminated using µPIV. Then, the focal depth of the microscope set the plane of measurement. The µPIV system used in this study is made by Dantec Dynamics A/S and the software used was DynamicStudio version 3.40.82.

In the present study, a curved 250x1000 µm^2 micro-channel manufactured according to the process presented in Farré-Lladós et. al [16] was used; see Figure 4a. The channels had a minimum width equal to the elbow that the micro-nozzles part had near the outlets (Figure 4b). The flow velocity field and velocity profile in the cross-section marked were obtained; see

Figure 4c-d. The velocity profile changes with time and in order to capture- and investigate these temporal changes, and its effect on the flow, µPIV measurements were done approximately every 5-6 minutes. The greases considered are Lithium complex greases with density of 880-1070 kg/m^3, service temperature -40ºC to 150ºC, and drop point < 220ºC. They are similar to the greases studied in Westerberg et al. [17] where the flow properties and its relation to the rheology of the grease was investigated, showing a complex semi-solid behavior. The actual grease properties allowed the grease to remain in the position where it is injected; in this case the tooth flank, but it's fluency through the proposed micro-nozzle should be verified. For more details on the grease flow dynamics in an elbow channel, the study done by Westerberg et al. [18] provides more details.

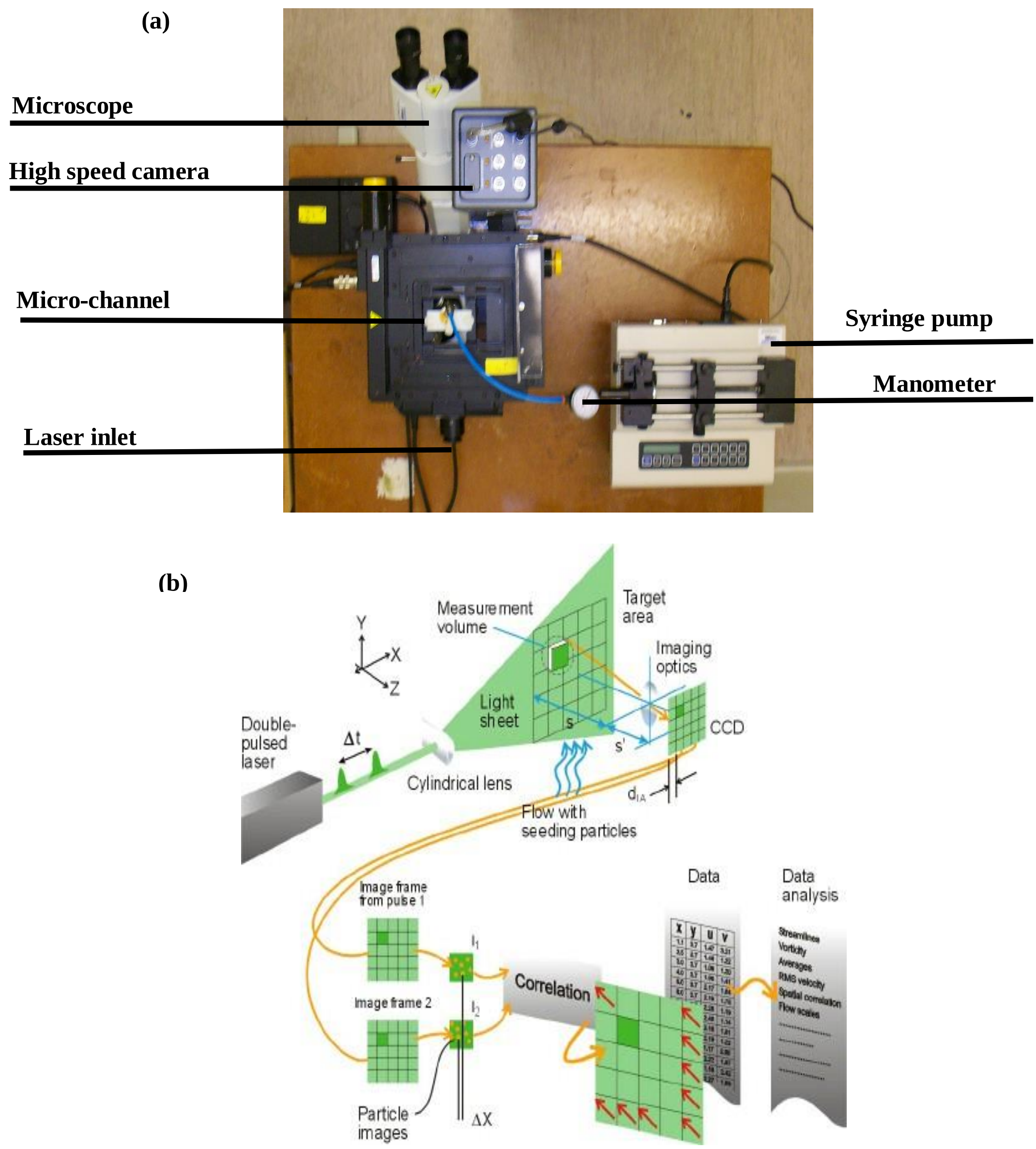

Figure 3: µPIV experimental setup. (a) Top view of the test set up. (b) Main principle of PIV.

The micro-channel was filled using a syringe pump (KD scientific, model KDS410) until the grease leaked through the outlet and the pressure was stable (denoted the static pressure value). The actual experimental flow rate was then set to the pump without removing the circuit pressure, meaning grease with a constant flow rate is fed into the channel. In Figure 5 and Figure 6 the evolution of the maximum velocity value in the center of the channel is shown for a low- and high flow rate respectively. The theoretical velocity was calculated from the cross-section area of the channel and the applied flow rate. The time until stationary conditions occur is denoted the transitory flow period (the yellow zone in Figure 5 and Figure 6). When the grease velocity stabilizes around the theoretical velocity the stationary flow period starts (the red zone in Figure 5 and Figure 6). A high- and low flow rate were considered, where the high flow rate was 0.5 ml/min and simulates an injection of grease when the wind turbine starts to work at the zero-degree position. The low flow rate was 0.002 ml/min and simulates the continuous lubrication mode of the zero-degree position. Grease samples were checked for any physical changes such as separation of base oil and thickening agent in accordance to DIN 51817 [19]. Further, chemical changes such as additives and components loss were analyzed using an infrared spectrum analysis and a consistency analysis under the ASTM standard D1403-02: 2007 [20] to verify that the grease had not suffered any properties degradation. The infrared spectrum determined all compounds in the mixture to be analyzed recording the absorption curve for each hydrocarbon, and superimposing it on the radiant energy background [21].

The amount of grease needed per micro-nozzle (i.e. each tooth) was determined from the manufacturer's recommendations in respective gear datasheet. The recommendation is 2 kg per year in a 139 teeth crown operating a 90º-degree section. Considering that only 35 teeth are in operation for the control of the pitch angle, each tooth consumes 57 grams per year. In this study a common grease pump for the wind industry (Lincoln, model QLS 401) is considered. For this pump the minimum grease injection volume is 0.2 cm^3 per stroke. In the present study NLGI 2 greases with density 0.99 gr/cm^3 is considered, therefore 0.198 gr were required every 30 hours. In general, grease fluidity is highly dependent on the temperature as the grease viscosity is dependent on the temperature; see e.g. Lugt [22]. The lower the temperature is, the more difficult it is to reach correct lubrication and a higher pressure is required. In order to measure the required injection pressures at different temperatures a climatic test in the working range of wind turbines was performed. Considering the worst case, the offshore wind turbines as reported in Ref. [23], the extreme temperature is -20ºC. The data from this report was used to select the range of temperatures to measure the required injection pressure for the proposed device using a high consistency grease (NLGI 2), since this grease bounds to the gear surface and requires higher injection pressures. To determine the micro-channel pressure losses during cold climate conditions, the micro-channel and the grease pump were placed into the climatic chamber. After 6 hours stabilizing the components and the fluid temperature, 0.4 cm^3 of grease was injected to produce a high pressure at the micro-channel inlet whereafter the evolution of the pressure is monitored until it reach stationary conditions.

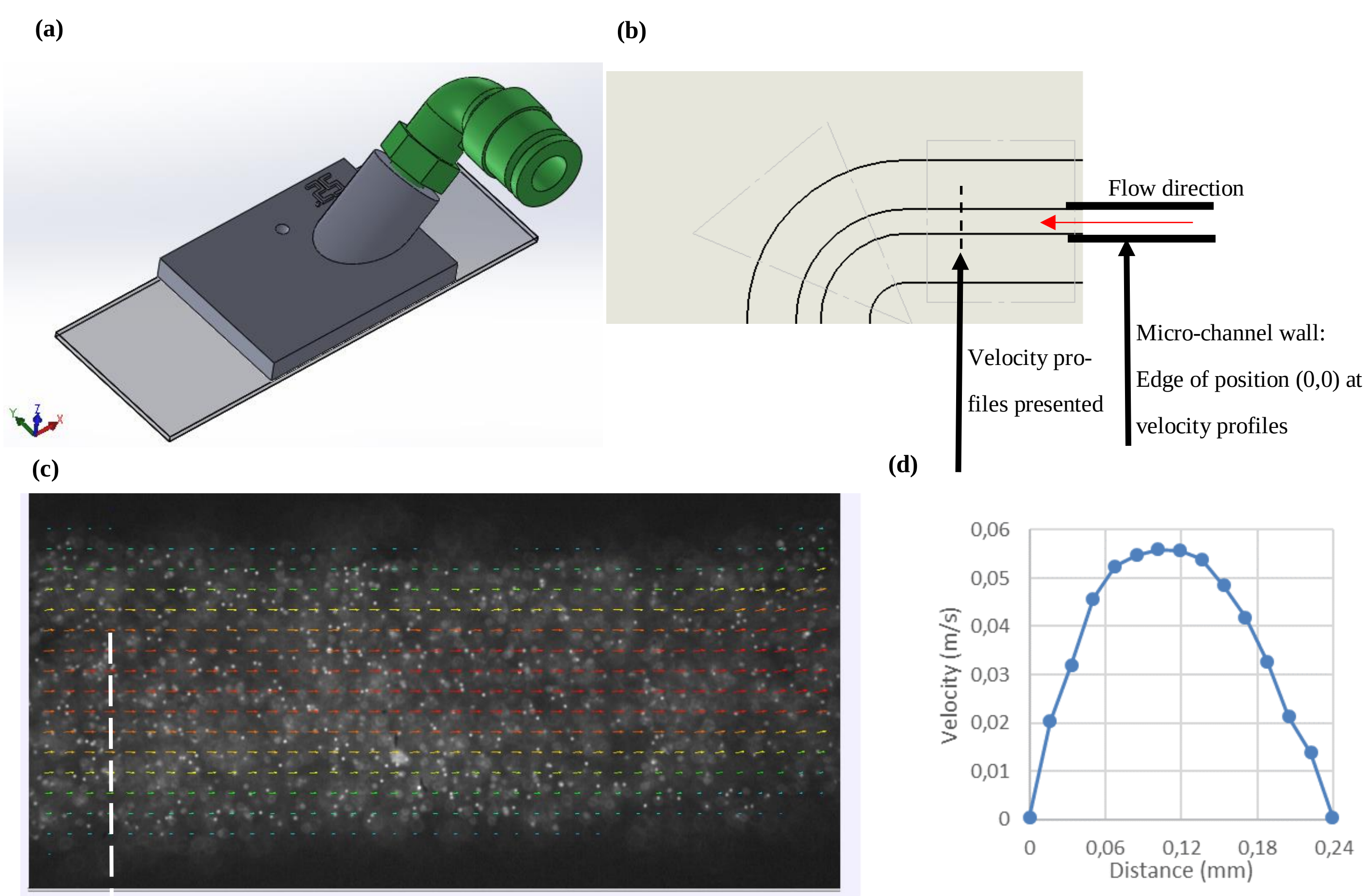

Figure 4: Micro-channel design and velocity profiles in the elbow. (a) The micro-channel setup (top view). (b) The micro-channel setup: bottom view with the channel and its elbow. (c) Microscope picture with a 20x objective superposed with the average vectors velocity calculated from the µPIV measurements. (d) Velocity profile at the dotted line in (c).

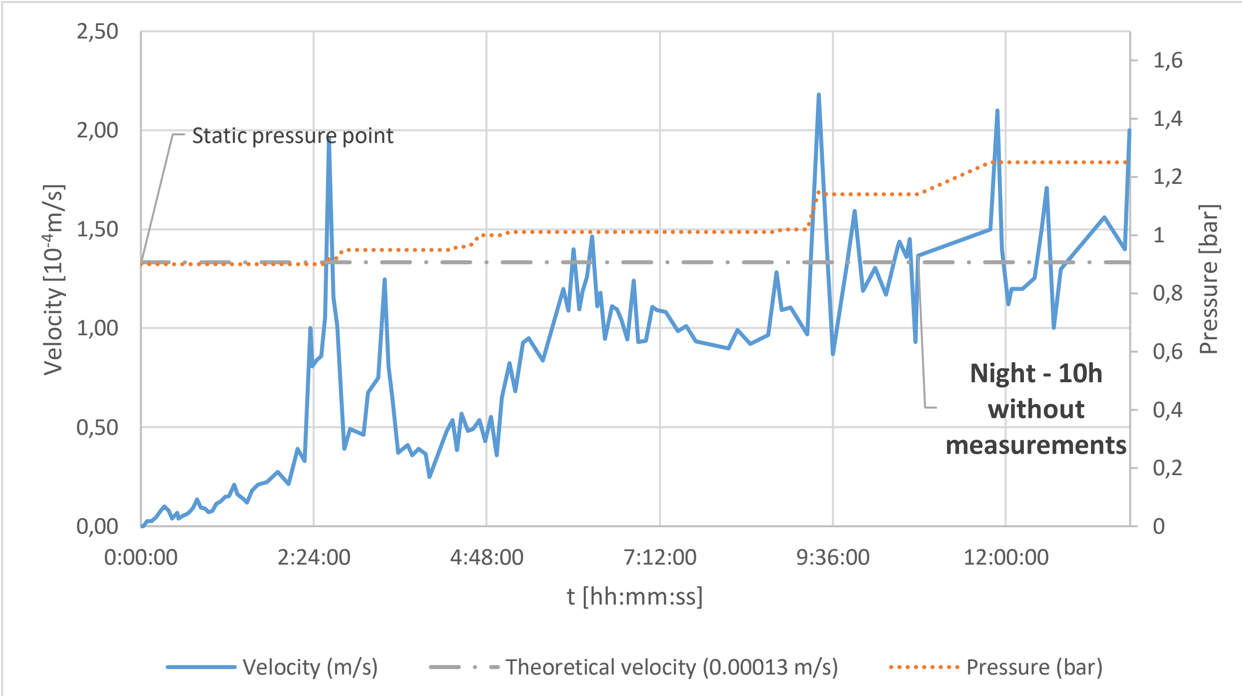

Figure 5: Velocity evolution for the low flow rate case.

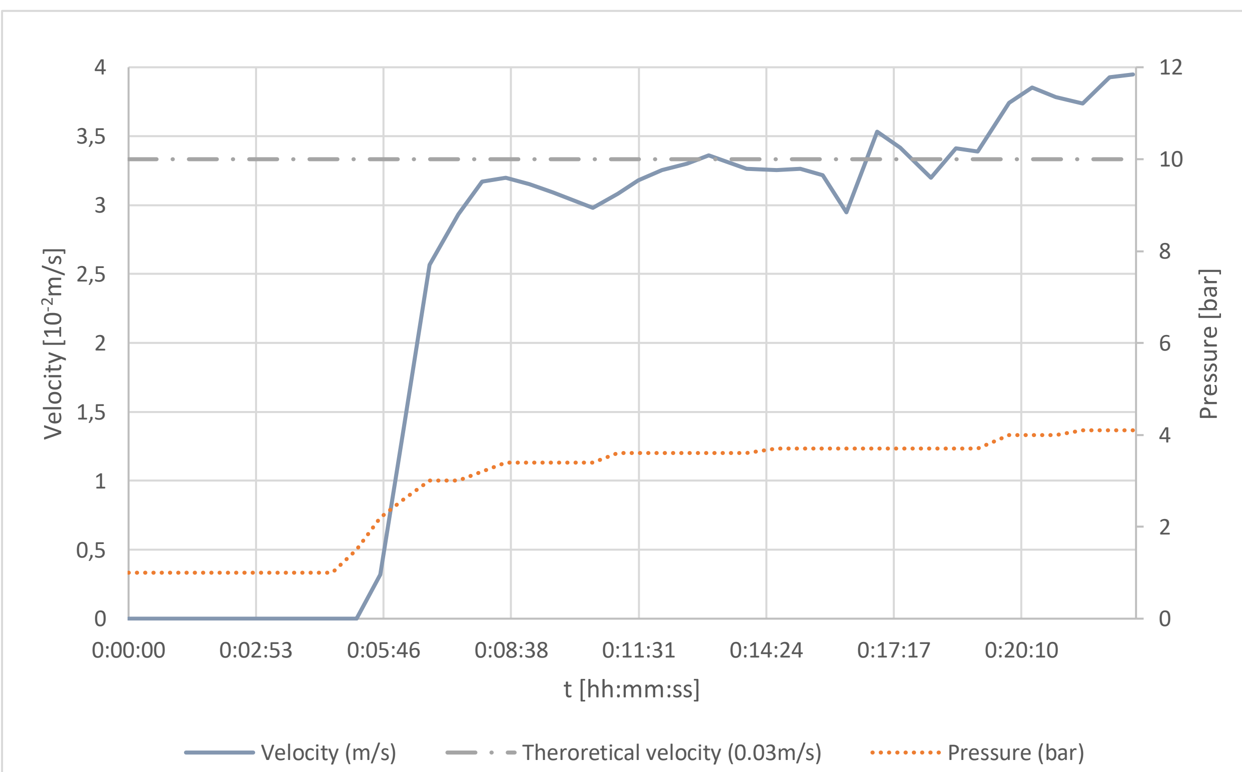

Figure 6: Velocity evolution for the high flow rate case.

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2.3 Micro-nozzle integration in a Wind Turbine Gear Pitch

To analyze the effect of lubrication through the micro-nozzle part on the gear teeth of the pitch system, the micro-nozzle part was mounted in a test bench capable to reproduce the phenomena of the excessive wear at the zero degree position. Figure 7a shows the schematics of its integration in a common wind turbine lubrication system. The grease pump was connected to an additional selective valve that through a grease distributor turn aside the grease flow when the wind turbine works at the gear tooth through the proposed micro-nozzle. This schematic is compatible with the common lubrication system to lubricate the rest of the crown through the progressive grease distributor. Figure 7b represents a blade detail to show where the grease outlets were connected to each lubrication micro-nozzle (lubrication of the crown and proposed micro-nozzle). The blade is assembled to the pitch bearing and is actuated by the motor through the gearbox. The wind turbine control activates the system through the electric control. Figure 8 shows a 3D view of the integration of the micro-nozzle in a 12-module pitch bearing. This integration can be carried out using a supplementary plate with two thread holes or with two thread holes at the crown to fix the new system.

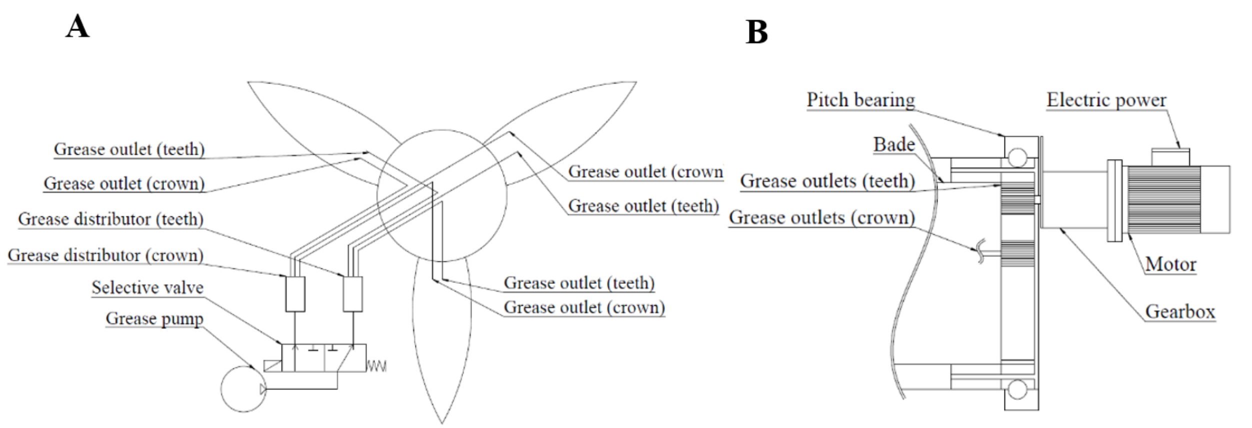

Figure 7: A: Schematic drawing of the micro-nozzles integrated in the wind turbine lubrication system. B: Integration per each blade.

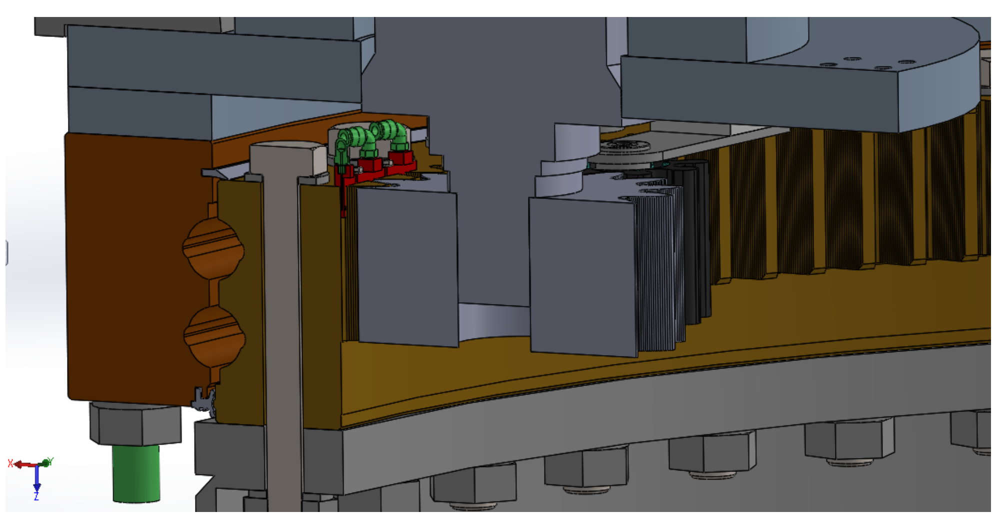

Figure 8: Pitch cross-section at zero degree position.

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The test bench simulates operating conditions where the trailing edge of blade 1 (Figure 9), is under compression stresses while traction stresses occur in the blade leading edge. When the blade is located in position 2 the situation is just the opposite as the blade has initiated the descent. The force direction affecting each blade edge has been interchanged, generating a torque M_x. Moreover, a torque M_y is generated due to the wind turbine regulation with the wind velocity that induces similar traction-compression cycles. Finally, each blade suffers a torque in the z-direction (M_z) caused by the aerodynamic forces making it turn around its axis; a load that is balanced by the gearbox through the pinion. The loads playing an important role in the appearance of the excessive wear during wind turbine dynamic operation in the test bench are:

- The blade micro-movements caused by the gearbox backlash due to M_z. Figure 9 shows the aerodynamic forces acting on the blade. At the zero degree position the gearbox is braked. M_z corresponds to the torsion torque ($M_{torsion}$) used by Holierhoek et al. [3].
- The load cycle tractions causing compression on the blade. These loads are due to torques M_x and M_y in Figure 9 when the pitch works at the 0º position, or M_{edge} and M_{flap} according to the definition in Ref. [3].

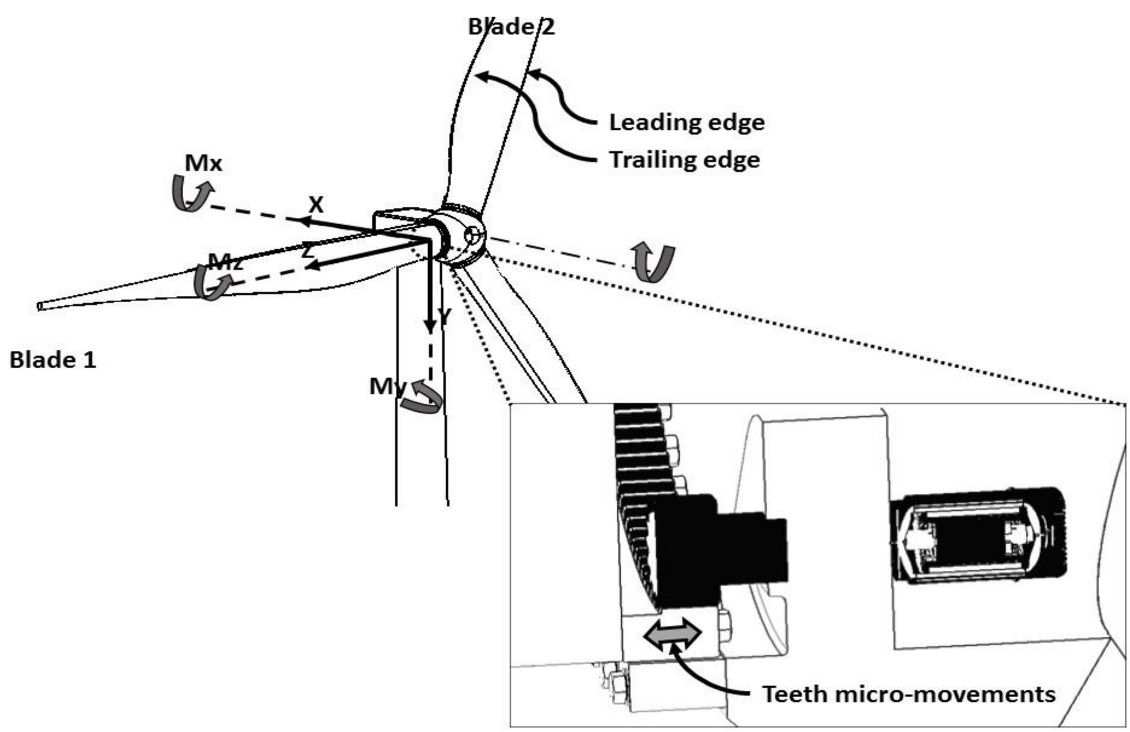

Figure 9: Wind Turbine hub with the blade coordinate system from the Germanischer Lloyd Industrial Services [1]. The arrow represents the micro-movements in the pitch teeth system due to the dynamic wind turbine operation. More details of the cross section in Figure 8.

The load operation scenario reproduced in the used test bench is summarized in Table 1. The crown hardness was 58±3 HRC and the pinion hardness was 24±3 HRC. The lower pinion hardness was chosen in accordance to allow a faster appearance of the wear for the test bench measurements. The gearbox model 700T/W from Bonfiglioly showed a 0.5 mm of backlash measured in the test bench drive train.

Backlash	Mz/ $M_{torsion}$	Mx/ M_{edge} and My/ M_{flap}
0.5mm at 0.2Hz	Constant ±40KNm	2000 kNm at 0.3Hz

Table 1: Test bench load situation according to the manufacturer Laualgun Bearings

Using this setup the grease distribution on the tooth flank was analyzed for different micro-nozzle lengths and positions in a 12-module gear 0.1 m wide to maximize the tooth flank area covered with grease. The micro-nozzle cross-section used is represented in Figure 2 (D = 250 μm and C =1000 μm). Table 2 summarizes the different micro-nozzle lengths tested. Each micro-nozzle injected the grease in different positions along the gear width and the test bench performed one complete cycle; see Figure 10 showing the positions for micro-nozzle A-C and the grease distribution.

Micro-nozzle (250x 1000μm^2)	Length (m)
A	0.02
B	0.04
C	0.03

Table 2: Micro-nozzle sizes tested

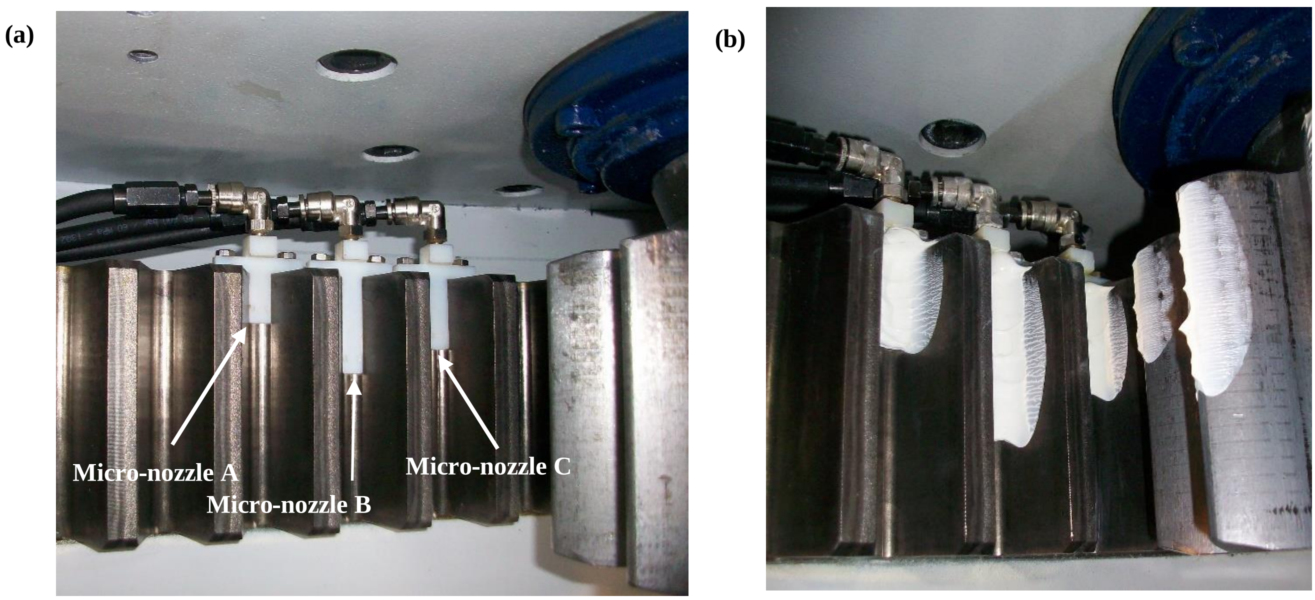

Figure 10: Micro-nozzle integration in a pitch gear. (a) 2-cm, 4-cm and 3-cm length micro-nozzle tested. (b) Grease distribution on the tooth flank.

2.4 Wear analysis

A methodology to quantify wear is not defined by the standard ANSI/AGMA 1010-E95 [24], but a comparison of the teeth profile over the load cycle can help to quantify the performance of the micro-nozzles. In this study a polyurethane resin model F1-F12 from Axson molds was used as shown in Figure 11a to reproduce the teeth profile (Figure 11b). The profiles were obtained through scanning using a perfilometer PGI 1220 from Talysurf as shown in Figure 11c. The scanned profiles were superposed in Solidworks 2013 and the wear affectation was measured.

To compare the wear affectation measured, Archard [25] wear equation was used. This reads

$$Q = \frac{K \cdot W}{H}, \tag{1}$$

which yields that the wear volume is proportional to the work done by the friction forces. Here Q is the wear volume, K the wear coefficient, W the load and H the hardness of the material.

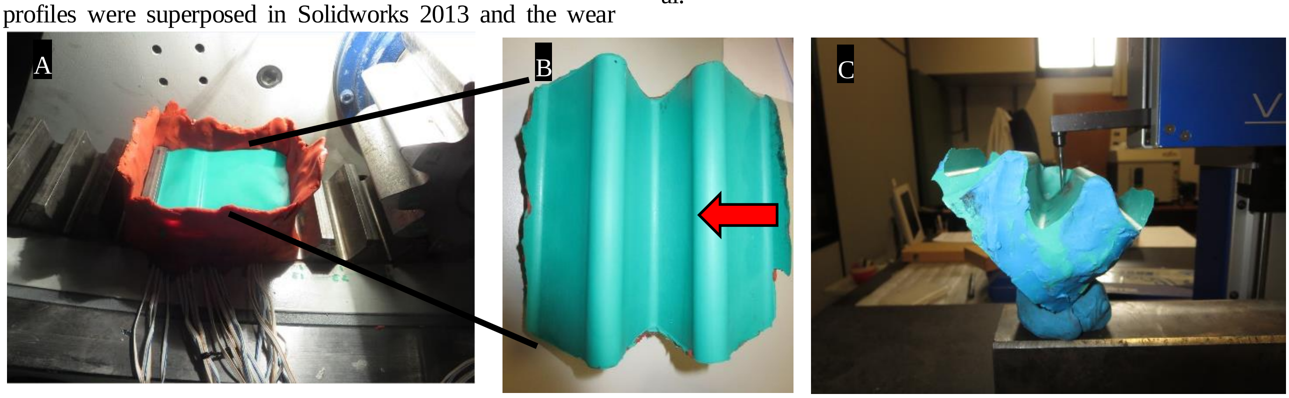

Figure 11: Tooth mold using polyurethane resin model F1-F12 from Axson. The arrow points at the position where the tooth profile is measured and evaluated to get the wear effect.

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3. Results

3.1 Grease flow test

Figure 5 and Figure 6 show the time evolution of the maximum velocity in the velocity profile and the circuit pressure at 0.002 ml/min and 0.5 ml/min respectively. The grease is injected into the circuit and the experiment is initiated by applying specified flow rate, inducing a static pressure until the flow is stationary. The µPIV measurements showed that the grease could flow in a (width x height) 250x1000 µm channel. Figure 5 shows that for low flow rates, the flow is more continuous but it is difficult to control the volume of the injected grease. For high flow rates on the other hand, the required volume of grease is injected almost instantaneously. Figure 6 shows that in only six minutes the grease flows at the theoretical velocity in the outlet.

Lubricating greases have a characteristic yield stress behaviour, meaning the grease behaves as a solid (or more correctly - a visco-elastic) material for values of the shear rate below a certain threshold (yield) value. The grease yield behaviour generally increases with the NLGI grade, but the yield stress for a given grease also varies with temperature such that the yield stress value increase with a decreasing temperature – and vice versa decrease with increasing temperature [22], [26], which also is the case for the grease viscosity. This means that for a sufficiently low temperature the yield stress is so large that the grease is unable to flow with the shear rates induced through the applied pressure by the grease pump. As the pump is set to deliver a specified flow rate rather than a specified pressure, the pump will continuously deliver the actual amount of grease per time unit.

For a flow medium with zero yield stress (such as e.g. an NLGI 00 grease, or an oil) the flow pressure would once the initial transitory effects have ceased stabilise around a constant value determined by the pressure needed to apply the actual flow rate. Due to the pressure drop in the channel (which in turn is due to the difference in pressure between inlet and outlet) the grease is subjected to larger shear rates close to the inlet compared to further downstream. Hence, for the case of higher yield stress value in the grease - due to lower temperature or higher thickener concentration (higher NLGI grade) – the grease will remain stationary at some point downstream until a sufficient pressure (and consequently sufficient shear) has been built by the continuous feeding rate by the grease pump. Once sufficient pressure is built the flow is released and the pressure decrease and stabilize. This effect is illustrated in Figure 12 by the build-up and peak during the first 15 minutes. The trend is clear for the thinner NLGI 1 grease where the pressure decrease with decreasing temperature. For the thicker NLGI 2 the trend is analogous except for the case with lowest temperature (-15˚C).

The yield stress concept is a good engineering model of a main physical property which affects the grease flow properties. Fundamentally it is though not so straightforward as modelled mathematically with a discontinuous threshold shear stress value. Physically there is a transition region by means of shear rate in where the grease goes from being non-yielded to fully yielded (continuously deforming, i.e. flowing). This transition is coupled to the visco-elastic properties of the grease which in turn is dependent on the thickener concentration and type. Other properties such as phase separation may also influence the onset of flow by means of base oil separation forming a less viscous slip layer close to the channel boundaries. The dynamics of the flow behaviour in the limit of yielded/non-yielded is illustrated also in Figure 5 and Figure 6. For higher flow rates the shear rate is higher meaning the transition from non-yielded to fully yielded grease is much smoother compared to the low flow rate case. For this case the shear in the flow is increased at such a low rate that the visco-elastic properties of the grease heavily affect the flow in possible combination with effects due to e.g. phase separation. Another condition which may affect the dynamics at such low changes in flow/shear rates is the local distribution of the thickener in the grease which may lead to phenomena like shear banding where the shear forces is locally varying due to local differences in thickener concentration. The results from the infrared spectrum analysis and cone penetration test [20] however indicate no phase separation.

Based on what is being presented above, a likely explanation for the constant pressure for the NLGI 2 grease at -15˚C is that the grease loses much of its elasticity compared to higher temperatures. In combination with the increased viscosity the applied flow rate by the pump leads to grease motion at the channel outlet after as long as 6 h, which should be compared to the expected maximum 20 min it should take to achieve a good lubrication as presented in the paper. The build-up-and-release effect is hence not present at such low temperatures. And as concluded in the paper it is not suitable to use the NLGI 2 grease at such low temperatures.

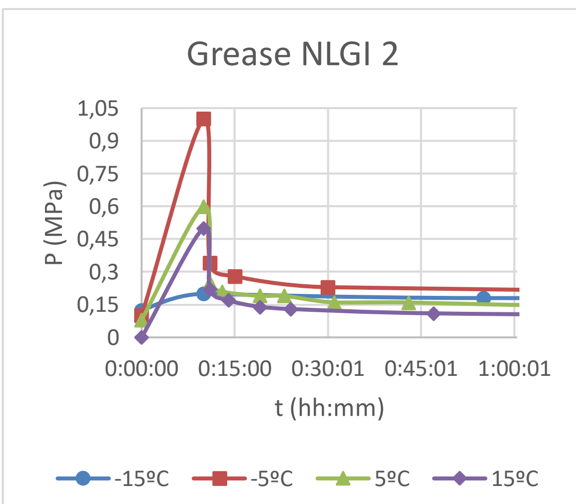
It was found that the NLGI 2 grease can operate over 0º C without the pressure overcoming 1 MPa and injecting 40% of the initial volume in less than 20 minutes. However, when the temperature is less than 0ºC grease with NLGI grade 1 should be used as the NLGI 1 grease was found to flow at -20ºC for the same amount of grease and requires a maximum pressure around 0.17 MPa as shown in Figure 12.

3.2 Test bench validation

Three different positions of the injection point has been tested (Figure 10a) showing that when the injection point is at the center of the gear tooth, the grease distribution is more uniform (Figure 10b). Furthermore, the test bench operation shows that there was no interference between the micro-nozzle and the teeth, even with the loads applied. The test bench was used to determine the number of fatigue load cycles (Table 1) needed to appreciate wear in the tooth flank without lubrication. During the first 10^4 cycles the wear appeared as shown in Figure 13a due to the bearing deformation and the friction, as Holierhoek [27] reported. However, when the micro nozzles were used to lubricate, wear was not visually observable for the same number of cycles (10^4 cycles); see Figure 13b. The test conditions without lubrication and with the micro-nozzles share the same load and material hardness conditions meaning Eq. 1 yields

$$\frac{Q_{without\ lubrication}}{K_{without\ lubrication}} = \frac{Q_{with\ micro-nozzles}}{K_{with\ micro-nozzles}}. \quad (2)$$

If the teeth profiles are compared (Figure 14), the wear coefficient up to 10^4 cycles $K_{with\ micro\text{-}nozzles}$ is 0.3 times the $K_{without\ lubrication}$. Therefore, the results of this study indicate that a reduction of the wear effect of more than 70% can already be appreciated in the first 10^4 cycles; see Figure 14. Figure 13c-d shows the tooth flank without- and with lubrication respectively when the test bench has reached $2\text{x}10^4$ cycles. The molds obtained to quantify the teeth wear shows that up to 10^4 cycles the wear increases at a constant rate in both cases (with and without lubrication); see Figure 14. For the case when the teeth are working without lubrication however, it was found that after 10^4 cycles wear is initiated adjacent to the tooth and consequently the pressure per tooth is reduced and the wear rate is reduced.



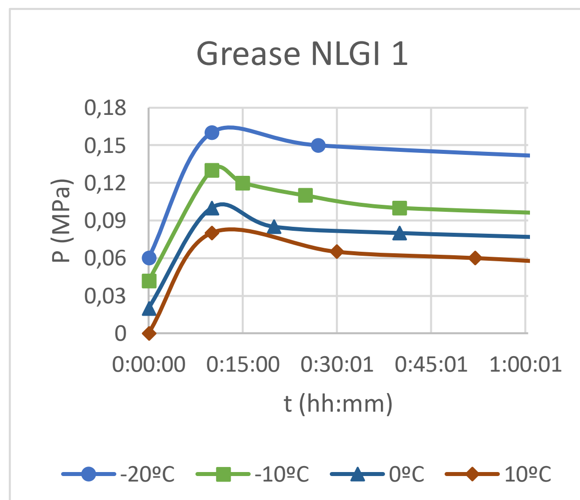


Figure 12: Temperature influence on the flowing pressure through the micro-nozzle B shown in Figure 10.

4. Discussion

This project was undertaken to design a micro injector and evaluate the possibility to decrease excessive wear in pitch/yaw gears in wind turbines. Initially, the results of this study indicate that NLGI2 grease can flow in a 250x1000 µm channel which is useful for wind turbine pitch gears without any degradation at temperatures higher than 0ºC. The µPIV measurements concludes that for the lower flow rate of 0.002 ml/min a stationary grease flow is not obtained until several hours, meaning the grease will not flow through the micro-nozzle with the desired theoretical velocity until hours after the pump has been started. The exact time is dependent on the pipe length, distributor and valves, but mainly on the compression of the grease in the system. For the high flow rate of 0.5 ml/min however, the transitory period is shorten to a few minutes. Based on these results, using a low flow rate enables a continuous lubrication but with a high uncertainty in the amount of injected grease; see Figure 5. For high flow rates on the other hand, as shown in Figure 6, a less continuous lubrication is achieved but the volume injected is more stable. According to the µPIV results, a high flow rate is preferred since it guarantees the required amount of grease in minutes, which in turn prevent metal to metal contact and wear to be initialized. The proposed micro-nozzle approach to lubricate the gear yields a more accurate strategy to inject the lubricant at the zero degree position while the wind turbine is running.

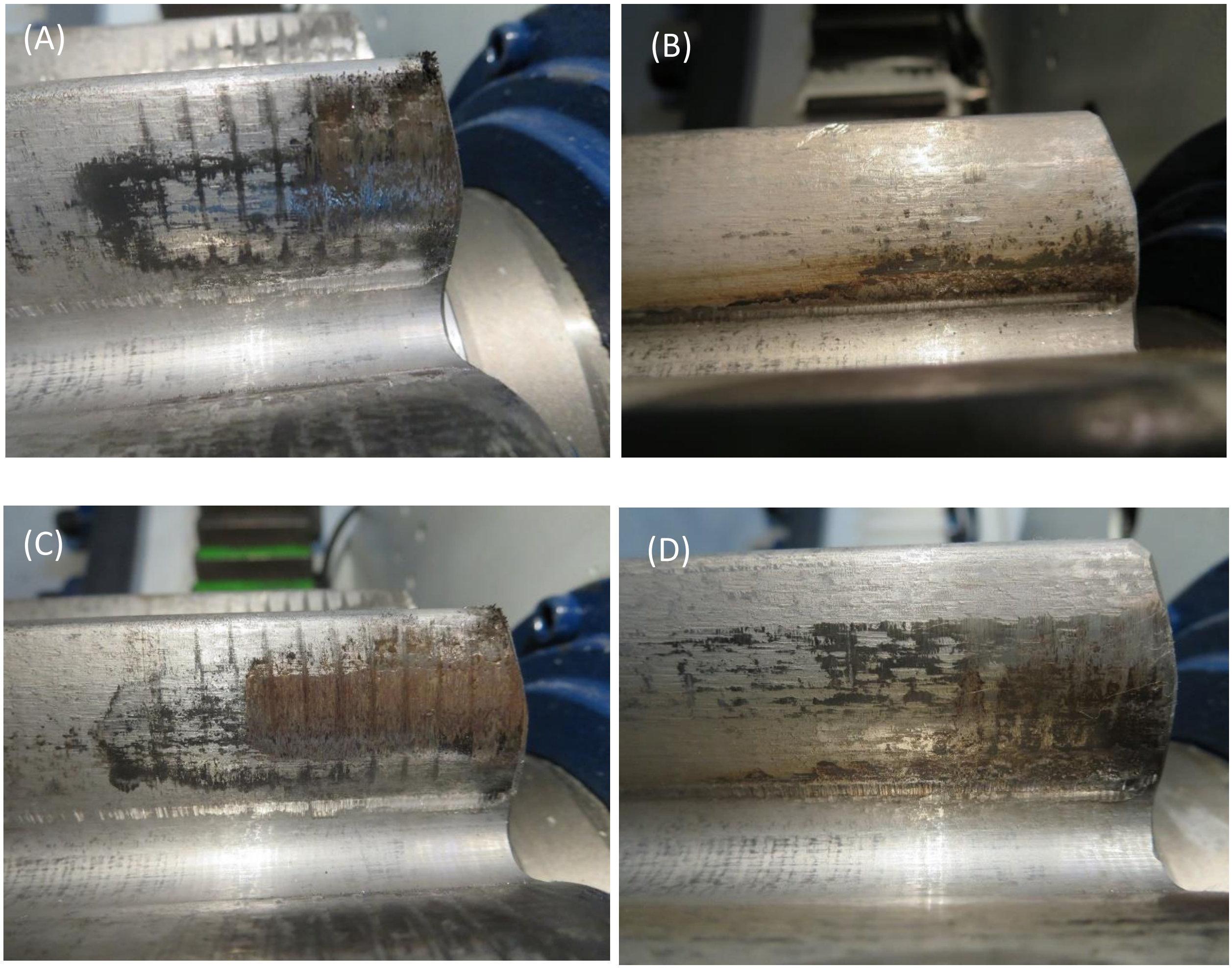


Figure 13: Wear evolution from the test bench experiments. (A) 10^4 cycles without the lubrication system. (B) 10^4 cycles with the lubrication system. (C) $2x10^4$ cycles without the lubrication system. (B) $2x10^4$ cycles with the lubrication system.

The proposed micro-nozzle part to reduce wear in the gears modifies the system tribology which Takadoum [11] defined as critical. Takadoum defined four parameters important to minimize the wear: surface state, stress in the system, mechanical contact and tribology of the system respectively. Mashue et. al [5] proposed changes of the surface state by decreasing the hardness of the pinion relative to the crown. This is however difficult to implement in running wind turbines. Dimascio et. al [6] presented a solution in the same direction to replace the damaged region of the gears in situ with a new bolt-assembled tooth. The systems in Refs. [5], [6] change the surface state but does not eliminate the wear meaning preventive maintenance is required; an operation that the technology presented in this paper would minimize. Nielsen [7] presented another solution where he changed the mechanical contact by assembling three gears; one to transmit power and the other two to be in contact with the bearing decreasing the stress per tooth. However, the wear was not eliminated and the preventive maintenance is still required. Kürzdörfer [8] designed a special injector, which sprays one side of the tooth to lubricate the joint directly. This solution cannot be easily implemented since there is no compressed air in the wind turbines and oil is difficult to be kept in the tooth flank. Paluncic and Schönfeldt [9] proposed to inject grease through the lubrication pinion, a solution that it is easy to implement in the wind turbines but require a stop of the wind turbine to lubricate the pinion at the zero degree position, which in turn leads to losses in the energy generation. The best solution so far is the one suggested by Klaus [10], which uses holes in the deddendum to inject fresh grease while the wind turbine is in operation. The main drawback of this solution is that in ongoing wind turbines the gear replacement costs are difficult to justify. Comparing the technology presented in this paper with the solutions listed above we conclude that it is easy to integrate the micro-nozzles in running wind turbines, which together with the lubricating pinion solution presented by Paluncic and Schönfeldt [9] can increase the energy generation since it eliminates the programmed wind turbine stops to lubricate.

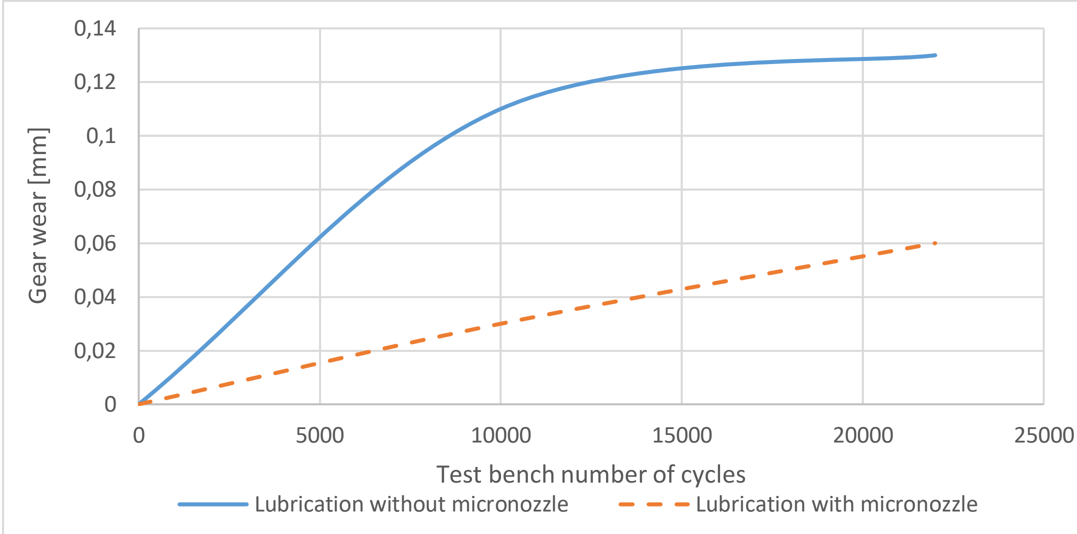


Figure 14: Average evolution of tooth wear according to the number of cycles performed with a σ-value of 0.01 mm. Dashed line is the wear with lubrication using the micro-nozzles, and solid line is the wear without lubrication.

5. Conclusions

A novel lubrication device with a micro-nozzle to provide fresh grease in the pitch gear teeth while they are in contact has been designed and fabricated. The fabricated parts were tested in a pitch gear test bench of a 2 MW wind turbine using conventional wind turbine greases under real loads working at temperatures above -20ºC. Using micro Particle Image Velocimetry (μPIV) it was found that a high flow rate is preferred in order to enable the correct amount of lubrication when required. Further, the nozzle injecting grease to the middle teeth for the zero degree position showed a more uniform grease distribution. Compared to current lubrication systems the new lubrication device:

- Extend the service life of the gear teeth by more than a factor two and reduce the appearance of wear.
- Allows the wind turbine to be lubricated during operation which is a key difference compared to previous technologies and consequently increases the wind turbine electricity generation.
- Is universally applicable to any pitch gear modules, regardless of shape, size and the lubrication system available.
- Enable the use of autonomous systems to reduce preventive and corrective maintenance such as the device presented.

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