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IFAC PapersOnLine 56-2 (2023) 11255–11260

Control of Large Wind Energy Systems for Acoustic Noise Reduction by Using Multi-Objective Optimal Control

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Work financed by the Federal Ministry of Economic Affairs and Climate Action (BMWK)

Abstract: An important restriction to the social acceptance of the wind energy systems is the acoustic noise that they introduce into the environment, particularly during the night hours in settlements close to the wind farms. This problem is solved by reducing the rotational speed of the machines with a corresponding power loss. The usual way to do this is to switch the set point of the rotor speed between day and night operations.

The present contribution studies the problem and proposes two control system configurations that try to minimize the power losses by tracking and adjusting the rotational speed. The concept is based on two controllers working cooperatively. The controller tuning is carried out by using a game-theoretic approach solved by multi-objective optimization. The simulation results show an improvement with respect to the common procedure, such that it looks promising for the application to real machines.

Keywords: Large wind turbines, acoustic noise, pitch control, noise mitigation

1. INTRODUCTION

A major constraint on society's acceptance of wind energy is the concrete fact that wind turbines are acoustically noisy. This is a clear impediment in the case of onshore systems, but this is also slowly becoming important in offshore installations.

This is one reason why studies on noise emission, propagation, measurement, modelling, estimation and reduction, which some time ago were of rather peripheral concern, have taken on essential relevance. All these aspects are commonly studied separately. However, they should be jointly considered.

The acoustic noise emitted by a wind turbine has several sources, of which the most important are the aerodynamic noise and the mechanical noise (Hubbard and Shepherd, 1991). Other less important noise contributions are, for instance, ground reflection and sound refraction. The mechanical noise can be acoustically damped in the nacelle, (Sørensen, 2012) by using isolating panels, (Barone, 2011) combined with active damping control of the drive train (Gambier, 2022). Hence, the treatment is concentrated at present on the acoustic noise generated by the machine that reaches a receiver located several meters away.

Large wind turbines are not available for experimentation. Thus, dynamic models for the noise emission coupled with the aeroservoelastic models are important for simulation purposes. On the other hand, the noise is actually significant not at the emission point but at the receiver's place. Hence, noise propagation models should be considered and included. A stable and reliable noise measurement is necessary for the correct assessment, as well as a real-time control system. The noise measurement can be improved by using estimation and prediction, which in turn depend on the model accuracy.

Although all abovementioned aspects are relevant, integrated, and constitute an extended research field, which cannot be completely treated here because of space limitations, the present contribution is only devoted to the acoustic noise mitigation by using the control system. Topics on noise emission, prediction and radiation models can be found in, e.g., (Amiet, 1975; Brooks et al., 1989; Howe, 1978) and (Rozenberg et al., 2010) (see (Deshmukh et al., 2019) for a review).

The dominant sources of the aeroacoustic noise in modern large wind turbines are the noise caused by the turbulent inflow as well as the sound induced by the air flowing at the trailing edges of the blades (Howe, 1978; Wagner et al., 1996; Rogers et al., 2006). Moreover, the aerodynamic noise produced by the blade of a wind turbine grows roughly to the fifth power of the relative wind speed (Sørensen, 2012). Aeroacoustic noise can also be classified into *tonal noise* and *broadband noise*. The tonal noise is a discrete pulsing low frequency signal (20–100 Hz), which is caused by the unsteady air velocity as a consequence of the rotating blades.

The broadband noise has a spectrum over 100 Hz with relevant frequencies under 200 Hz. It consists of several non-periodic signals, which together constitute a cyclical envelope. It is produced by the moving blades and the interaction with the boundary layer (Bhargava and Samala, 2019; Rogers et al., 2006). Hence, lowering the rotor speed reduces noise at the expense of producing less power. This is the traditional mechanism used to switch between night and day operations.

On the other hand, the induced sound is sensitive to the angle of attack of the blade aerofoils (Oerlemans et al., 2007), which depends on the pitch angles, i.e., the angle of attack decreases with the increase of the pitch angle, which leads to a contraction in the turbulent boundary layer of the suction side of the airfoils, causing noise reduction but also power. Contrarily, reducing the pitch angle increases the noise and the power extraction. Thus, pitch-controlled wind turbines will cause changes in the induced sound all the time since the pitching activity is an essential characteristic of such machines. Hence, the pitch control system introduces disturbances into the sound source but is also a means to mitigate them.

10.1016/j.ifacol.2023.10.317

The prevention of aeroacoustic noise by using control has been studied, e.g., in (Cardenas-Dobson and Asher, 1996) and in (Møller and Pedersen, 2011). Moreover, several contributions are devoted to mitigate the aeroacoustic noise by using the pitch control system. For example, some implementations of individual pitch control dedicated to attenuate noise can be found in (Bertagnolio et al., 2014; Maizi et al., 2017) and (Mackowski and Carolus, 2021).

The above-mentioned contributions to reducing noise do not take into account the general case that individual pitch control is not always available in real wind turbines. Moreover, they also do not consider the trade-off between maximum power extraction and minimal noise emission. Thus, these two aspects are considered in the present work, where noise mitigation and power extraction are compromised by using multi-objective optimization in a similar way as used in (Gambier, 2017), for collective pitch control and tower damping control. Hence, a control loop for the attenuation of noise is optimally combined with the collective pitch control.

The rest of the paper is structured as follows: Section 2 is devoted to present the necessary fundamentals of acoustic noise from wind turbines, and in Section 3, the control issues are discussed. The control system topology and the design are described in Section 4, whilst a numerical study on a 5 MW reference wind turbine, including the simulation results, is the subject of Section 5. Finally, conclusions are drawn in Section 6.

2. WIND TURBINE ACOUSTIC NOISE FUNDAMENTALS

As it was already mentioned, the field of wind turbine acoustic noise is vast and complex. Therefore, the interest in the current section is to describe the basic aspects in order to understand the problem, so that the needs can be satisfied in this work. The interest is to obtain a prediction model to estimate the noise and the formulation of an objective function as a performance index, which can be used in the optimization problem.

2.1 Prediction Models

Many noise prediction models have been proposed in the specialized literature. These models have been classified by (Lowson, 1992), where three classes are distinguished: Class I for very simple stationary models, Class II for models of middle complexity, which include some parameters of the wind turbine, and Class III for models with full information about the noise process associated with the wind turbine. A comparison of the different models is provided by (Zidan et al., 2014). The interest here lies primarily in the simple Class I models. The most important ones are:

Lowson's Model

The model of (Lowson, 1992) is given by

$$L_{wA} = 10\log_{10} P_{wT} + 50\,, \tag{1}$$

where L_{wA} is the A-weighted sound power level of the source and P_{wT} is the rated power of the wind turbine in Watts.

Hau's Model

In (Hau et al., 1993), a simple model depending on rotor diameter D is proposed. The formula is given by

$$L_{wA} = 20\log_{10} D + 72\,. \tag{2}$$

Hagg's Model

The model suggested by (Hagg et al., 1992) includes the tip speed, i.e.,

$$L_{wA} = 50\log_{10} v_{tip} + 10\log_{10} D - 4\,. \tag{3}$$

Since that for the tip speed applies, $v_{tip} = 0.5\,D\,\omega_r$, with ω_r as the rotor speed, (3) can be reformulated in terms of the rotor speed, namely

$$\begin{aligned} L_{wA} &= 50\log_{10}(0.5D\,\omega_r) + 10\log_{10} D - 4 \\ &= 50\log_{10}(\omega_r) + 60\log_{10} D - 19.0515\,. \end{aligned} \tag{4}$$

The three formulas are evaluated for a 5 MW reference wind turbine with a rotor radius of 63 m. As real data, the simulation data provided by the high-resolution model implemented in OpenFAST is used (Bortolotti et al., 2020). The results are shown in Figure 1.

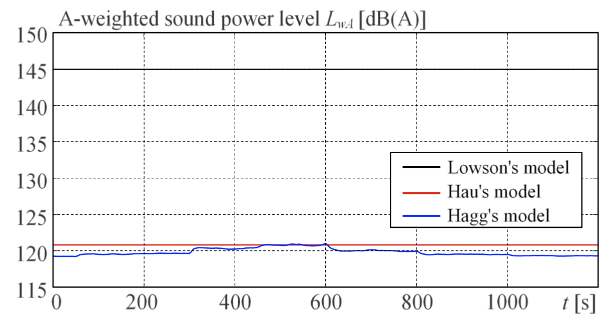

Figure 1. Results for the prediction of the sound power level

As it is pointed out in (Wagner et al., 1996), these formulas are very inaccurately and cannot be used if exactitude is required. However, they can be readjusted for a particular case if measurement data is available. In the case of real-time control, the prediction can be complemented by sensor data.

The last equation is useful because ω_r is a measurable state variable, and therefore, the sound power level L_{wA} is variable, calculable, and dependent on the pitch control system. Hence, this model has been selected for this first study.

The above-presented models correspond to the emission places. Since the important aspect is the sound power level at the receiver position, the model has to be complemented by a propagation model. This is presented in the following subsection.

2.2 Propagation Models

The choice of a propagation model follows the same criterion as for the prediction model, i.e., simplicity and dependence of useful variables from the control point of view. A detailed treatment of the topic can be found in (Wagner et al., 1996).

The ISO 9613 (ISO-9613, 1993; ISO-9613, 1996) serves as the basis for modelling the propagation of sound waves. The sound pressure level at the receiver location is calculated for each source by

$$L_{pA} = L_{wA} + L_{cf} - A\,, \tag{5}$$

where L_{wA} is already defined in the previous subsection, L_{cf} is a correction factor in dB (zero for the radiation into the free space), and A is the attenuation in dB. Moreover, the attenuation A is composed of several factors, such as, for instance,

$$A = A_{gd} + A_{atm} + A_{gr} + A_{bar} + A_o\,. \tag{6}$$

The factor A_{gd} is the attenuation caused by geometric divergence. It can be defined for hemispherical spreading, spherical spreading, or cylindrical spreading, i.e.,

$$A_{gd,h} = 10\log_{10}(2\pi(d/d_0)^2) = 20\log_{10}(d/d_0) + 8\text{dB}$$
$$A_{gd,s} = 10\log_{10}(4\pi(d/d_0)^2) = 20\log_{10}(d/d_0) + 11\text{dB} \quad (7)$$
$$A_{gd,c} = 10\log_{10}(2\pi(d/d_0)) = 10\log_{10}(d/d_0) + 8\text{dB}$$

with d as the distance to the receiver in meters. d_0 is a reference distance (normally 1 m), 10 $\log_{10}(2\pi) \approx 8$ dB and 10 $\log_{10}(4\pi) \approx 11$ dB. The distance d can be computed by considering the hub height h_h of the wind turbine and the horizontal distance l_d to the receiver as

$$d = \sqrt{l_d^2 + h_h^2}\ . \quad (8)$$

The atmospheric attenuation A_{atm} is given by

$$A_{atm} = \alpha\, d \quad (9)$$

with α as atmospheric absorption in dB/m, which depends on the frequency, temperature, humidity, and pressure. d is the distance in meters. Other factors that can be considered are, for instance, ground absorption (A_{gr}), screening (A_{bar}), as well as other possible factors (A_o). According to (Lovtidende, 2017), another factor is, for example, the sound insulation A_σ.

Factors A_{gr}, A_σ, and α are frequency-dependent parameters, which are normally expressed for each central frequency of the 1/3-octave bands (see (Lovtidende, 2017)). For each band, applies (5), and the total sound pressure level is obtained from

$$L_{pA,tot} = 10\log_{10}\left(\sum_{i=1}^{n} 10^{L_{pA}(i)/10}\right). \quad (10)$$

In the sense of formulating simple models, it is commonly assumed that the propagation takes place in a spherical spread, i.e., the sound pressure level suffers an attenuation of 6 dB per distance doubling (Møller and Pedersen, 2011). For the spherical spreading, the A-weighted sound pressure level (SPL) L_{pA} is formulated by

$$L_{pA} = L_{wA} - 20\log_{10}(d/d_0) - \alpha\, d - 11\,\text{dB}\,. \quad (11)$$

A typical value for α is 0.005 dB/m (Rogers et al., 2006).

In the downwind direction, the spherical assumption is valid for distances closer to the machine. For distances larger than 200 m, the propagation presents a cylindrical spread, which means that the decay rate is about 3 dB per distance doubling (Hubbard and Shepherd, 1991). The cylindrical propagation for distances greater than 200 m can be modeled by using

$$L_{pA} = L_{wA} - 20\log_{10}(200\text{m}/1\text{m}) - 10\log_{10}(d/200\text{m}) - \alpha\, d - 11\text{dB} + A_g\ , (12)$$

(Møller and Pedersen, 2011). A_g is a ground effect correction (1.5 dB for onshore and 3 dB for offshore machines). The Danish standard (Lovtidende, 2017) proposes a propagation equation for low-frequency noise given by

$$L_{pA} = L_{wA} - 10\log_{10}(h_h^2 + d^2) - \alpha\sqrt{h_h^2 + d^2} - A_\sigma - A_{gLF} - 11\text{dB} + A_g\ . (13)$$

ΔL_{gLF}, and ΔL_σ are ground effect and sound isolation at low frequencies, respectively. Propagation model results for the same example are presented in Figure 2.

2.3 Ambient Acoustic Noise

Ambient acoustic noise lebels ranged from about 30 dB(A) in rural and suburban zones to 120 or more dB(A) in urban and commercial

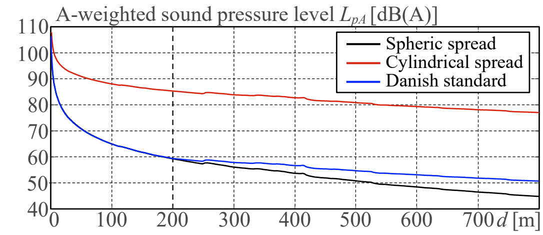

Figure 2. Results for the models of sound propagation

areas (Fitzell and Phil, 2019). Zones closer to wind farms are normally characterized by suburban settlements. Thus, it is possible to assume a noise level fluctuating between 30 and 48 dB(A) (Hansen and Hansen, 2020).

Moreover, the wind interaction with the foliage in rural settings produces an A-weighted broadband sound pressure, which is almost proportional to the base 10 logarithm of the wind speed (Fégeant, 1999), namely,

$$L_{pAwind} = K_1 \log_{10}(v_w) + K_2\ . \quad (14)$$

According to (Rogers et al., 2006), the wind produces ambient noise varying between 25 dB(A) (calm conditions) to 42 dB(A).

3. CONTROL OF WIND TURBINES

As previously mentioned, the main noise sources are the trailing edges of the blades and the rotational speed, which both change according to the pitch control system. Thus, noise generation and the control of the rotational speed are closely related. Therefore, in order to understand the control problem associated with the noise mitigation of a wind turbine, wind turbine control in general has to be clarified first. These aspects are briefly undertaken in this section.

3.1 General Aspects of Wind Turbine Control

Since the operation of an upstream horizontal-axis variable-speed variable-pitch wind turbine is widely known (see, for example, (Burton et al., 2011; Manwell et al., 2009; Bianchi et al., 2007; Gambier, 2022), it is just briefly discussed in the sequel. Based on wind speed, the operation of the machine can be divided into four adjacent zones. The wind turbine cannot convert energy in the first region because the wind speed is lower than the *cut-in* value for which the machine was designed. When the wind speed goes above the cut-in value, the operation moves to the second region, where the wind speed is appropriate for energy conversion without reaching the rated values. The control objective is to maximize power by tracking the optimal characteristic curve of the generator.

If the wind speed increases over the rated value, the machine enters Region III, where it stays as long as the wind speed does not reach the cut-out threshold. In this region, the control objective is to maintain constant the rotational speed (and indirectly, power) by pitching the rotor blades to the feather. Over the cut-out value, the wind turbine goes into Region IV, where it must be shut down in order to protect its integrity.

On the other hand, the transition zones between Regions I and II as well as between Regions II and III are often called Region I½ and Region II½, respectively. The difference between both transition regions is given by the corresponding set-points. All regions are described in Figure 3.

3.2 Speed Control in Region I and II

In Regions I and II, the wind speed is low enough that the rotational speed is also low, and therefore, no control is generally needed for it. However, sometimes it is required to control the

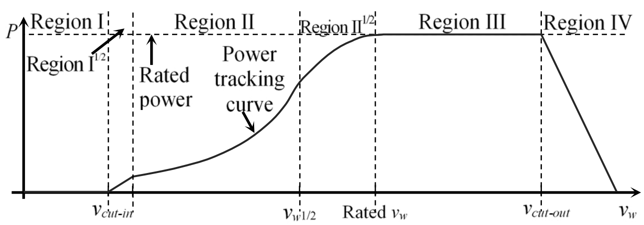

Figure 3. Operational regions of large wind turbines, (Gambier, 2022)

rotational speed. In such a case, this can be carried out by using the generator torque, as is illustrated in Figure 4.

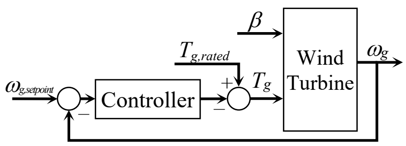

Figure 4. Rotational speed control diagram for Region I and II

3.2 Speed Control in Region III

In Region III, the main control objective is to regulate the rotational speed and the power conversion to the rated values. This is accomplished by including a second control loop that keeps the rotational speed constant by pitching the blades collectively. The collective pitch controller and the corresponding control loop are shown in Figure 5 (see (Gambier, 2021)).

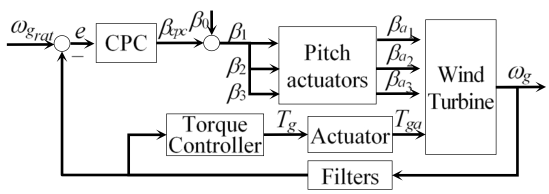

Figure 5. Rotational speed control diagram for Region III

When the system reaches the steady state, the controller output is equal to zero, but the pitch angle has to be equal to the necessary angle to maintain the rotational speed at the rated value for the current wind speed. This is obtained by including β_0. In other words, β_0 is the value of the pitch angle, which yields a rated rotor speed $\omega_{r,rat}$, for the corresponding wind speed.

4. WIND TURBINE CONTROL FOR NOISE MITIGATION

4.1 General Concepts

Regulation in Germany for onshore installations requires a minimum distance of 400 m and thresholds of 55 dB(A) and 40 dB(A) for day and night, respectively (Nieuwenhuizen and Köhl, 2015). In addition, these threshold values refers to the total noise. This means that if the ambient noise is louder for a while, then the wind turbines have to be quieter in order to maintain the noise limit.

From the practical point of view, the regulation is satisfied by including in the supervisory control system two fixed operational modes: "*normal operation*" for the day and "*power limited*" for the night. Thus, the operation is switched between these two modes according to the time of day indicated by the clock in the supervisor. In the following, another control concepts for the noise mitigation are proposed.

4.2 Control in Region III

First, the concept has been developed for Region III because this is the zone of high wind speed and, consequently, the noisiest. The idea is to implement a tracking control system such that the power conversion takes place adaptively in order to maximize power while maintaining the sound threshold below the limit.

Two control system topologies can satisfy this purpose. Both use two controllers: the collective pitch controller (CPC) and the active sound damping controller (ASDC). The first topology connects the controllers in a cascade configuration. The second does it in the parallel configuration. Both configurations are shown in Figure 6.

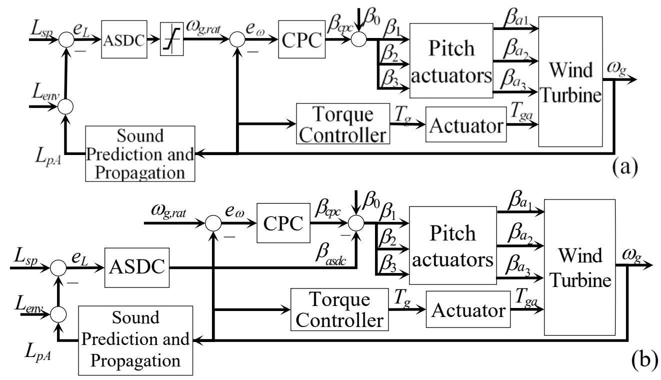

Figure 6. Control system for pitch control and noise mitigation. (a) Cascade configuration. (b) Parallel configuration

The cascade configuration subordinates the speed control loop to the noise level control loop. Thus, the power reduction is forced until the maximum allowable noise limit is reached. The maximum value in the saturation block is set to the rated rotational speed. This ensures that the speed does not exceed its rated value if the noise level is very low. Hence, this yields a hard maintenance of the allowable noise level.

In the parallel configuration, both controllers compete without, a priori, one prevailing over the other. However, if the noise level increases, its control error and control signal decrease, and then the pitch angle increases, reducing the speed and consequently the noise level. In this case, it is about a soft maintenance of the allowable noise level because no limit is imposed. The relative importance of the control loops is managed by assigning priorities.

In all cases, controllers must work coordinately to achieve the best possible result. One way to achieve this is to tune the parameters of both controllers together as players of a cooperative game using MOO (multi-objective optimization, see, e.g., (Gambier, 2017)).

The multi-objective optimization algorithm minimizes by using simulation data the following objective functions

$$J_\omega = \frac{1}{t_f - t_0}\int_{t_0}^{t_f} t\, e_\omega^2\, dt \text{ and } J_L = \frac{1}{t_f - t_0}\int_{t_0}^{t_f} t\, e_L^2\, dt . \quad (15)$$

The result is a Pareto front, as portrayed in Figure 7.

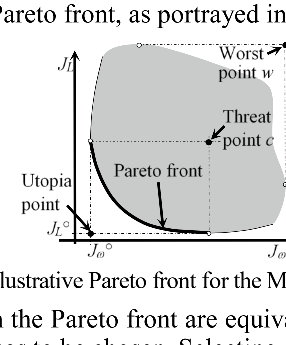

Figure 7. Illustrative Pareto front for the MOO problem

All points on the Pareto front are equivalently optimal, but only one point has to be chosen. Selecting a point closer to the vertical axis prioritizes the CPC and, contrarily, a point more distant to it emphasizes the ASDC.

4.3 Control in Region I and II

Regions I and II are characterized by a low rotational speed and, therefore, noise mitigation is not necessary. However, if the machine is likewise too noisy in Regions I and II, the control concepts and design technique proposed for Region III can be integrated with the control scheme of Figure 4 to introduce noise mitigation there.

The major distinction lies in the fact that the control variable is injected into another type of actuator and must therefore be scaled differently, requiring entirely new parameter tuning.

5. NUMERICAL STUDY

The approach for reducing the acoustic noise proposed in the previous section is now analysed by means of a numerical simulation example.

5.1 Experimental Setup for the Simulation Experiments

For the numerical study, the NREL 5MW reference wind turbine (Jonkman et al., 2009) has been chosen because it satisfies the characteristics of a typical onshore machine.

The conventional three-bladed, horizontal axis, clockwise, upwind, variable-speed and variable-pitch machine has a 126m-diameter rotor with 61.5 m-long blades. The hub height is 90 m and the generator has an efficiency of 94.4%, which corresponds to a mechanical power of 5.30 MW. The maximum power factor C_p is 0.482 and it is reached at a tip-speed ratio of 7.55. The gearbox ratio is 97:1, such that the rated rotor speed of 12.1 rpm yields a generator speed of 1173.7 rpm for a rated wind speed of 11.4 m/s. The cut-in rotational speed is $\omega_{r,ci}$ = 6.9 rpm. This leads, according to (4), to a noise emission that varies between 99.4 and 112.1 dB(A).

5.2 Scenario for the Numerical Study

Both proposed control system topologies are compared with the classic speed limitation approach. The comparison is carried out by considering the energy produced in kWh during the night operation. Moreover, three cases are considered for the classic approach: In the optimistic case, the ambient noise is supposed to be low enough that the rotational speed can be increased up to the maximum allowed value for a noise of 40 dB(A); in the pessimistic case, the ambient noise is high and the rotational speed is set to the minimum allowed value; and in the third case, the ambient noise is average.

The receiver is located 400 meters away from the wind turbine, and the aerial-ground-environmental conditions have a sound attenuation of 58.7 dB(A). The German legislation allows a total noise level of up to 55 dB(A) during the day and 40 dB(A) at night for small settlement areas.

The operation takes place in Region III with an effective wind speed changing between 11.5 and 21 m/s, including tower shadow and turbulence of 10%, for 30 minutes (see the profile as the grey line in Figure 9b).

5.2 Control System Design

All controllers are implemented as PI control laws (proportional-integral) with a back-calculation anti-windup mechanism. The controller parameters are obtained by using the method described in 4.2. The classic approach is tuned, however, by using simulation-based single objective optimization for the rotor speed control loop. Controller gains are summarized in Table 1. Notice that the parallel configuration has two values for $K_{i\omega}$. The fist is for day operation and the other one is for night operation.

Table 1. Controller Parameters

Gains	Classic Approach	Cascade Approach	Parallel Approach
K_{pL}	--	-0.5383	-0.4866
K_{iL}	--	-41.88×10^{-5}	-9.573×10^{-5}
K_{aL}	--	0.5	0.7
$K_{p\omega}$	1.51	1.64	-1.937
$K_{i\omega}$	0.653	0.5631	0.5631 0.0
$K_{a\omega}$	8.0	0.5	0.7

5.3 Simulation Results and Analysis

The simulation is started with four minutes in the day operation and then it is switched into the night operation. In total, the whole simulation lasts 30 minutes. The results for the classic approach are shown in Figure 8.

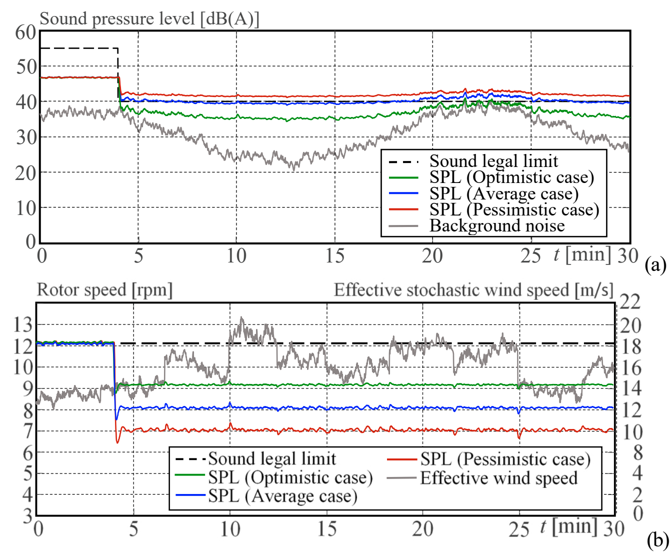

Figure 8. Results for the classic approach. (a) SPL. (b) Rotor speed.

It is appreciated that the control system can only keep the noise at the correct level in the optimistic situation, i.e., when the minimum environmental noise occurs. If some leeway is given, the average case might also be acceptable. The power restriction is not required during daytime operation since the SPL is less than 55 dB(A) for the rated rotor speed.

The results for both proposed approaches are presented in Figure 9.

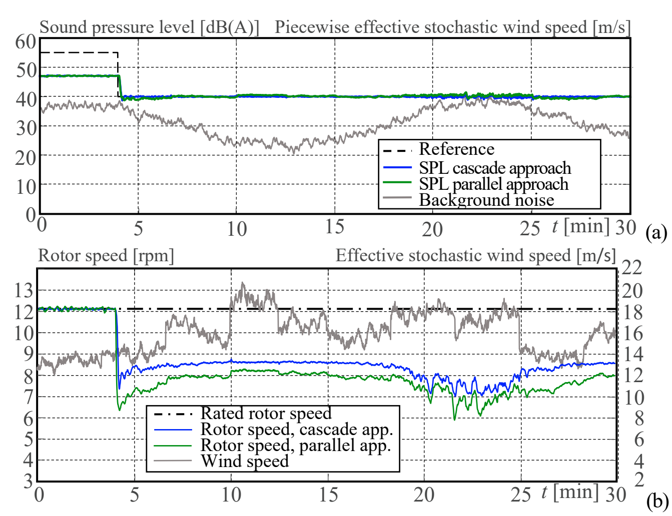

Figure 9. Results for the new approaches. (a) SPL. (b) Rotor speed.

Figure 8b also shows that, by design, the rotor speed is kept constant independently of the sound pressure level. This is the main difference with respect to the new approaches, where the rotor speed is adjusted in order to maintain the SPL at the allowed level but at the maximum possible rotor speed. This is observed in Figure 9b.

The converted energy in kWh for all cases is shown in Table 2. The optimistic case yields the maximum conversion, assuming very low ambient noise. This is not realistic and should not be used. In the pessimist case, the noise is high and the assumption leads to an anti-economic low result. The average case exceeds the permissible limit when noise is high and it should be avoided if the regulations are very strict. The new approaches guarantee threshold compliance by means of variable rotation speed, where the cascade approach performs better than the parallel approach.

Table 2. Energy converted by the different approaches in 30 minutes

	Energy in kWh		
Classic Approach	Optimistic case	Pessimistic case	Average case
	3750.4	2941.9	3559.2
Cascade Approach	3601.0		
Parallel Approach	3535.6		

7. CONCLUSIONS

In this work, two approaches for the control of wind turbines under noise restrictions with minimum losses in the energy conversion are studied. Both perform better than the classic approach. However, the cascade configuration is slightly superior. The parameter tuning is carried out by using multi-objective optimization. In the next research steps, more sophisticated models for the propagation as well as noise measurements in a Hardware-in-the-Loop system and real-time operation will be studied.

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