Received: 6 April 2023 | Revised: 19 September 2023 | Accepted: 30 October 2023

DOI: 10.1002/we.2883

RESEARCH ARTICLE

WILEY

Wind turbine main bearing rating lives as determined by IEC 61400-1 and ISO 281: A critical review and exploratory case study

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Funding information
EPSRC, Grant/Award Number: EP/S023801/1; Royal Commission for the Exhibition of 1851; National Renewable Energy Laboratory, Grant/Award Number: DE-A#36-08GO28308; US Department of Energy Office of Energy Efficiency and Renewable Energy Wind Energy Technologies Office

Abstract

This paper studies the rating lives of wind turbine main bearings, as determined by the IEC 61400-1 and ISO 281 standards. A critical review of relevant bearing life theory and turbine design requirements is provided, including discussion on possible shortcomings such as the existence (or not) of the bearing fatigue load limit and the validity of assuming linear damage accumulation. A detailed exploratory case study is then undertaken to determine rating lives for two models of main bearing in a 1.5 MW wind turbine. Rating life assessment is carried out under different conditions, including various combinations of main bearing temperature, wind field characteristics, lubricant viscosity, and contamination levels. Rating lives are found to be sufficiently above the desired 20-year design life for both bearing models under expected operating conditions. For the larger bearing, operational loads are shown to be below or close to the bearing fatigue load limit a vast majority of the time. Key sensitivities for rating life values are temperature and contamination. Overall, the results of this study suggest that an ISO 281 rating life assessment does not account for reported rates of main bearing failures in 1 to 3 MW wind turbines. It is recommended that a similar analysis be undertaken for ISO/TS 16281 rating lives, along with further efforts to identify principal root causes of main bearing failures in future work, possibly leading to a new application standard specific to this component. It is also recommended that the impacts of partial wake impingement on main bearing rating lives are investigated.

KEYWORDS

design standards, main bearing, rating life, rolling contact fatigue, wind turbine

1 | INTRODUCTION

Wind turbine main bearing life is an important driver of system reliability, availability, and operations and maintenance costs. However, a number of studies have indicated that these bearings are requiring replacement at a higher rate and sooner than expected based on design specifications and rating lives.[1–6] Across a 20-year lifetime, previously reported field failure data suggests replacement may be required for up to 30% of three-point-mounted main bearings in 1.5 to 2.5 MW wind turbines[5] (the same study showed replacements at 15% for four-point-mount systems). A more recent study,[6] utilizing field data from 15.3 GW of wind energy capacity and consisting predominantly of 1 to 3 MW three-point-mounted spherical roller main bearings, predicts that by year 20, replacement will be required for 22%–25% of a main bearing population. Concerns regarding main bearing reliability have led to increased research efforts in recent years focused on this component,[7–22] which seek to address premature failures in current and future systems.

While rolling contact fatigue remains a principal design driver, a general consensus is emerging from the industry that rolling contact fatigue is perhaps not to blame for the high numbers of premature failures observed in the field. Yet few studies exist in the scientific literature in which detailed and transparent assessments of main bearing rating life (under realistic conditions) have been undertaken. Without detailed studies and data on this question available in the scientific literature, the consensus must be treated as anecdotal. Main bearing rating life assessment is generally performed within industry, and as such, the details of assessment processes and resulting life predictions are not available in the public domain. Importantly, one published study predicted main bearing rating lives as low as 8 years under some conditions.[7] If rolling contact fatigue modelling indicates that these failures may occur on such timescales, then it must remain a candidate contributor to the observed rates of premature failure. This particular problem has further nuances, however, since wind turbine main bearings are typically large-diameter, flexible, grease-lubricated, low-speed, high-load bearings that experience large-scale continuous variations in load throughout their operational lives.[9,17,23] As such, it is not clear a priori to what extent the rating life calculations are accurate in this case. This work therefore seeks to consider the insight provided by IEC 61400-1[24] and ISO 281[25] design standards when applied to the problem of rating life assessment for wind turbine main bearings. More specifically, we consider the following research question:

Can the rating life assessment, as codified in the IEC 61400-1 and ISO 281 standards, account for the reported rates of premature failure for 1–3 MW wind turbine main bearings?

An answer in the affirmative would corroborate previous findings[7] and indicate that rolling contact fatigue may in fact be leading to premature failures in wind turbine main bearings. Note that the previously outlined questions regarding applicability of rolling contact fatigue models to the main bearing would still hold in this case; further work would therefore be required before any concrete conclusion might be reached. An answer in the negative (assuming the same is also true for ISO/TS 16281) means one of two scenarios must hold: (1) Rolling contact fatigue is not a dominant driver for premature main bearing failures (meaning observed failures are principally caused by other damage mechanisms) or (2) rolling contact fatigue does contribute to the observed failures in wind turbine main bearings, but current rating life assessment methods do not capture it sufficiently and hence are unable to provide realistic rating life predictions. Again, without further work, it would not be possible to determine which scenario holds. Regardless, a negative answer to the posed question indicates that the applied methodology does not provide the required insight at the design stage. A truly comprehensive answer to the research question will require significant and sustained research effort across multiple projects. The current paper seeks to initiate investigation of the broad research question by undertaking an exploratory analysis of predicted rating lives obtained for two models of main bearing in a 1.5 MW wind turbine. This power rating is well represented in the available field failure data.[3,4,6] Moreover, the selected models of main bearing represent upper and lower limits for bearing sizes as utilized in such machines and so approximately span the design space. IEC- and ISO-derived predictions can then be compared to the main bearing replacement rates and rating lives reported in the literature.

In order to facilitate a careful consideration of main bearing rating life assessment, a detailed summary and critical discussion of key aspects of rolling bearing life theory and the ISO 281 standard are presented in Section 2. Details of IEC 61400-1 design requirements are also provided. Rating lives for the two models of main bearing in the 1.5 MW wind turbine are then assessed using operational loads extracted from aeroelastic simulations. The methodology for this analysis is detailed in Section 3, with results presented in Section 4 and then discussed in Section 5.

2 | BACKGROUND

2.1 | Rolling bearing rating life

ISO 281[25] specifies methods for estimating the basic and modified rating life across a population of identical rolling bearings. In this context, the *life* of an individual bearing is the operational duration (calculated in revolutions, but convertible to time) before the first evidence of fatigue damage is observed, examples of which are shown in ISO 15243.[26] The *rating life* is the predicted value of bearing life based on its basic dynamic load rating. Rating life has two subcategories. The *basic rating life* (for a population of bearings manufactured with commonly used high-quality material, of good manufacturing quality, and operating under conventional operating conditions) is the value of bearing life which 90% of the population are expected to attain or exceed. The ISO 281 standard also provides methods to calculate the *modified rating life*, which incorporates additional effects, including lubricant viscosity, contamination, and the bearing fatigue load limit. As such, ISO 281 seeks to account for effects on bearing life from both subsurface- and surface-initiated rolling contact fatigue (although not necessarily exhaustively in the latter case). A seminal treatment of rolling bearing life assessment and its historical development is given by Zaretsky.[27] Only a brief summary of the aspects most directly relevant to the current paper will be provided. The technical specification ISO/TS 16281(E)[28] also accounts for other detailed factors such as the effects of misalignment, tilt, and bearing clearance on rating life. However, application of ISO/TS 16281 methods require access to additional technical and/or *in situ* data, which are not readily available. For this reason, the current work is specifically focused on the rating life predicted by ISO 281, as indicated by the proposed research question. For the sake of completeness, it is pertinent to mention that formulations of rating life equations which differ from those adopted in ISO 281 have also been proposed.[27,29,30]

2.1.1 | Basic rating life

In a 1924 paper,[31] Arvid Palmgren presented the foundational ideas for rolling bearing fatigue life theory. Exercising a great deal of insight, Palmgren recognized that such a theory would need to account for combined (axial and radial) loading and variable conditions,[*] as well as the natural variability present in time-to-failure data (even for bearings of the same type under equal conditions). The former considerations led Palmgren to propose that a rule be established for the conversion of combined loads to a purely radial "equivalent load" on which life prediction might be based. In order to account for variations in applied loads and speeds, Palmgren also devised the earliest example of a "linear damage accumulation" rule. The previously mentioned consideration of failure-time variability prompted Palmgren to advocate a probabilistic approach to rolling bearing reliability. This took the form of the L_{10} rating life, which is the operational life that 90% of a population of identical bearings are expected to equal or exceed without failing due to rolling contact fatigue. As discussed in Zaretsky,[27] the L_{10} rating life encapsulated the concepts of designing for finite life and reliability at an acceptable level of risk, an idea that was very much ahead of its time. Combining Hertzian contact theory with empirical test data in the same 1924 paper, Palmgren presented the first life equation of the form:

$$L_{10} = \left(\frac{C_D}{P_{eq}}\right)^p, \tag{1}$$

where L_{10} was given in millions of revolutions, p is the load-life exponent, P_{eq} is the equivalent applied bearing load, and C_D is the dynamic load capacity/rating (the load at which $L_{10} = 1$ million revolutions). The dynamic load rating accounts for bearing geometry and material, among other things, and is semiempirical. Formulas for calculating C_D based on bearing material and geometry are provided in ISO 281.[25,32] Bearing manufacturers will also generally supply a C_D value for each of their bearings; the values are likely altered to incorporate improved modelling and/or additional effects. Despite much of this early work being heavily empirical and lacking a strong theoretical basis, the core ideas presented by Palmgren in 1924 remain central to rolling bearing life prediction. In 1939, Waloddi Weibull developed a framework for statistically evaluating the fracture strength of materials.[33,34] An important output of this work was a detailed description of what is now known as the Weibull distribution—a continuous parametric probability distribution that is particularly well suited for describing life dispersion for a population of an engineering component. The Weibull distribution has since been widely applied across multiple fields.[35] In the context of component survival, $S(L)$ denotes the probability of an individual member of the population surviving for a duration of L or more.[†] The two-parameter Weibull survivability distribution then takes the form

$$S(L) = \exp\left(-\left(\frac{L}{L_\beta}\right)^e\right), 0 < S \leq 1. \tag{2}$$

where e is referred to as the Weibull slope, for reasons which will be outlined, and L_β is the characteristic life (the number of revolutions for which there is a 36.8% survival probability). Equation (2) may be reexpressed as

$$\ln \ln \frac{1}{S} = e \ln \frac{L}{L_\beta} \tag{3}$$

$$= e \ln L - e \ln L_\beta. \tag{4}$$

The latter expression is recognizable as a straight-line equation, in variables $\ln L$ and $\ln \ln \frac{1}{S}$, with gradient e (the "Weibull slope") and y-intercept $-e \ln L_\beta$. The Weibull parameters e and L_β may therefore be obtained from the line of best fit between $\ln L$ and $\ln \ln \frac{1}{S}$ values obtained experimentally, highlighting an additionally important characteristic of the Weibull distribution in the mid 20th century—the ability to readily estimate parameter values from measured data without the need for involved numerical procedures. Weibull's work focused on the fracture strength of materials. A life model based on the Weibull distribution was later applied to the problem of rolling contact fatigue, possibly at the suggestion of Weibull himself.[27] The first such model related the probability of survival to the number of stress cycles (NL, where N is the number of stress

*Due to variations in the loads and speed applied to the bearing itself.

†An equivalent definition for $S(L)$ is the proportion of the population surviving a duration of L or more.

cycles due to roller passage per revolution, and L in revolutions), the critical shearing stress (τ), and the stressed volume[‡] (V) occurring below the contact as follows:

$$\ln\frac{1}{S}=\left(\frac{L}{L_\beta}\right)^e \text{ (from Equation2)} \tag{5}$$

$$\propto \tau^c (NL)^e V. \tag{6}$$

This is equivalent to defining

$$L_\beta \propto \left(\frac{1}{\tau}\right)^{c/e}\left(\frac{1}{V}\right)^{1/e}\frac{1}{N}, \tag{7}$$

where c remains a free parameter for fitting the distribution to measured data. Setting $S=0.9$, which corresponds to $L=L_{10}$, it follows that

$$L_{10} \propto \left(\frac{1}{\tau}\right)^{c/e}\left(\frac{1}{V}\right)^{1/e}. \tag{8}$$

From the Hertzian theory of point contact, this may be reexpressed as

$$L_{10} \propto \left(\frac{1}{s_{o,\max}}\right)^{c/e}\left(\frac{1}{(s_{o,\max})^2}\right)^{1/e} \propto \left(\frac{1}{s_{o,\max}}\right)^{n_o} \propto \left(\frac{1}{P_{eq}}\right)^{p_o}, \tag{9}$$

and for line contact as

$$L_{10} \propto \left(\frac{1}{s_{l,\max}}\right)^{c/e}\left(\frac{1}{s_{l,\max}}\right)^{1/e} \propto \left(\frac{1}{s_{l,\max}}\right)^{n_l} \propto \left(\frac{1}{P_{eq}}\right)^{p_l}, \tag{10}$$

where $s_{o,\max}$ and $s_{l,\max}$ denote the maximum surface contact stress in point and line contact, respectively. For the above Weibull formulation,

$$n_o=\frac{c+2}{e}, p_o=\frac{n_o}{3}, n_l=\frac{c+1}{e}, p_l=\frac{n_l}{2}. \tag{11}$$

These exponent values linking bearing life to the applied load, P_{eq}, did not provide a good fit to the bearing life data available at the time. In 1947, Lundberg and Palmgren attempted to improve the fit between predicted fatigue life and experimental data by including an additional variable, the depth (z) at which τ occurs, such that

$$L_{10} \propto \left(\frac{1}{\tau}\right)^{c/e}\left(\frac{1}{V}\right)^{1/e} z^{h/e}, \tag{12}$$

where h was a new parameter to be fitted as part of the Weibull analysis, along with c and e. Following a similar process, the Lundberg-Palmgren model again resulted in

$$L_{10} \propto \left(\frac{1}{s_{o,\max}}\right)^{n_o} \propto \left(\frac{1}{P_{eq}}\right)^{p_o} \text{ (point contact)} \tag{13}$$

$$L_{10} \propto \left(\frac{1}{s_{l,\max}}\right)^{n_l} \propto \left(\frac{1}{P_{eq}}\right)^{p_l} \text{ (line contact)}, \tag{14}$$

but this time with

[‡]Both τ and V are those occurring below the point of maximum Hertzian contact stress.

$$n_o = \frac{c+2-h}{e}, n_l = \frac{c+1-h}{e}, \tag{15}$$

and p_o and p_l as per Equation (11). If the constants of proportionality are expressed in the form C_D^p, Equations (13) and (14) are of the same form as Equation (1). Fitting to experimental data resulted in exponent values of $c = 10.33, h = 2.33, e = 1.11$ in point contacts, and $e = 1.125$ in line contacts; hence, $p_o = 3$ and $p_l = 4$. These exponent values generated life predictions that correlated well with the experimental data available at the time. Lundberg and Palmgren subsequently advocated using a value of $p = 10/3$ for all roller bearings.[27] Identical exponent values (including $p = 10/3$ for all roller bearings) to those listed above remain the basis for ISO 281 rolling bearing rating life formulas.[32]

Since a bearing consists of multiple components, the expected life for the whole bearing is dependent on (and strictly less than) the expected life of each subcomponent. Excluding ball/roller failures and cage fatigue (because raceways tend to be the dominant locations for fatigue failures), the bearing rating life is dependent on those of the inner and outer rings. Assuming identical Weibull slopes, e, the resulting bearing rating life may be obtained via the application of "strict series reliability"[27]:

$$L_{10_{sys}} = \left(\frac{1}{L_{10_{in}}^e} + \frac{1}{L_{10_{out}}^e} \right)^{-1/e}. \tag{16}$$

In this case, the full bearing rating life also follows Equation 1. Assume that

$$L_{10_{in}} = \left(\frac{C_{D_{in}}}{P_{eq}} \right)^p \text{ and } L_{10_{out}} = \left(\frac{C_{D_{out}}}{P_{eq}} \right)^p. \tag{17}$$

From Equation (16), the system life is then

$$L_{10_{sys}} = \left(\frac{C_{D_{sys}}}{P_{eq}} \right)^p, \tag{18}$$

with

$$C_{D_{sys}} = \left(\frac{1}{C_{D_{in}}^{pe}} + \frac{1}{C_{D_{out}}^{pe}} \right)^{-1/pe}. \tag{19}$$

The basic rating life, L_{10}, of a complete bearing therefore also follows Equation (1), which is the form it takes in ISO 281. The ISO formula provides the bearing basic rating life, L_{10}, in units of millions of revolutions. It is the whole-system C_D value ($C_{D_{sys}}$), which is provided by a bearing manufacturer or obtained from the semiempirical formulas provided in the standard. While excluded in the above formulation, the life of ball/roller sets is implicitly accounted for through the empirical component of C_D equations. ISO 281 also prescribes formulas for the dynamic equivalent radial load, P_{eq}, as a function of applied radial (F_r) and axial (F_a) loads. These take the form

$$P_{eq} = XF_r + YF_a, \tag{20}$$

with dynamic load factors (X and Y) dependent on the type of bearing, number of rows, nominal contact angle and the resulting limiting value, and the ratio F_a/F_r. The term "limiting value" can sometimes be misconstrued. This value simply indicates the ratio of F_a/F_r at which the X and Y dynamic load factors change in value. At a load ratio above the limiting value, the contribution to fatigue from the axial load increases by 50%, while the contribution from the radial load decreases by 33% for a double-row spherical roller bearing. Relatively minor changes in contact angle of a few degrees can have a relatively large effect on the limiting value and the dynamic load factors. The standard indicates that the presented life equations do not necessarily apply in cases of severely truncated contact areas or if $P_{eq} > C_D/2$. Conversely, the standard also indicates that very light loads may cause different failure modes to occur.

2.1.2 | Modified rating life

There are applications where it is desirable to consider the rating life corresponding to different levels of survivability (i.e., other than 90%), as well as accounting for the effects on rating life of lubrication, contamination levels, and (somewhat controversially, as will be discussed) the bearing fatigue load limit. These effects are handled in ISO 281 via modification factors a_1 and a_{ISO} from which the modified rating life, corresponding to survivability $S=(1-n)/100$ (for n in %), is obtained as

$$L_{nm}=a_1 a_{\mathrm{ISO}} L_{10}. \tag{21}$$

The modification factor a_1 adjusts the considered level of survivability; hence, $a_1=a_1(S)=a_1(n)$. It may be derived as follows: Considering a Weibull distribution in the case where L_{10} (at which point the survivability is 0.9) is known, the rating life (L) associated with other levels of survivability (S) are obtainable through the relationship[§]:

$$\frac{\ln 0.9}{\ln S}=\left(\frac{L_{10}}{L}\right)^{e}, \tag{22}$$

from which the following two expressions may be derived:

$$L=\left(\frac{\ln S}{\ln 0.9}\right)^{1/e} L_{10} \tag{23}$$

$$S=0.9^{\left(\frac{L}{L_{10}}\right)^{e}}. \tag{24}$$

Analysis of experimental bearing life data has shown that in the region where $S>0.9$ (which corresponds to $L/L_{10}<1$), a three-parameter Weibull distribution provides a superior fit.[36] In order to account for this, Equations (23) and (24) are adjusted as follows[36]:

$$L=\begin{cases}\left[(1-C_\gamma)\left(\frac{\ln S}{\ln 0.9}\right)^{1/e}+C_\gamma\right]L_{10} & \text{if} S>0.9\\ \left(\frac{\ln S}{\ln 0.9}\right)^{1/e}L_{10} & \text{if} S\le 0.9,\end{cases} \tag{25}$$

$$S=\begin{cases}0.9^{\left(\frac{L}{L_{10}}-C_\gamma\right)^{e}(1-C_\gamma)^{-e}} & \text{if} L/L_{10}<1\\ 0.9^{\left(\frac{L}{L_{10}}\right)^{e}} & \text{if} L/L_{10}\ge 1.\end{cases} \tag{26}$$

Equation (25) allows for the calculation of a_1 coefficients at any level of survivability once the 90% duration, L_{10}, is known. Equation (26) allows one to calculate the proportion of the population expected to survive other durations, L, once the 90% duration, L_{10}, is known. Since a_1 and a_{ISO} are proportional coefficients, the above formulations remain valid for use with the adjusted values $L_{10m}=a_{\mathrm{ISO}}L_{10}$. A single "safe" value of $C_\gamma=0.05$ is recommended and implemented in ISO 281,[36] along with the Weibull slope $e=1.5$. This value of Weibull slope appears to be an odd choice, since the basic rating life is based on e values of 1.11 and 1.125 (refer to Section 2.1.1). While a Weibull slope of 1.5 may well have been chosen to reflect the fact that current experimental data indicate e values vary between 1 and 2,[27] it remains the case that $e=1.5$ is inconsistent with the basic rating life formulations. Furthermore, the main application of Equation (25) in ISO 281 is the calculation of a_1 coefficients corresponding to survivabilities greater than 90%. In that region, the larger value of $e=1.5$ actually provides a less conservative characterization of the increases in survivability realized as L decreases. For example, Figure 1 shows S versus L/L_{10} values obtained from Equation (26)[¶] when using two different Weibull slopes of $e=1.5$ and $e=(1.11+1.125)/2=1.118$. Note, $L/L_{10}=a_1$.

The factor a_{ISO} seeks to capture the influence that lubrication, contamination, and the bearing fatigue load limit, C_{u}, have on the bearing rating life. As presented in ISO 281, a_{ISO} takes the form:

$$a_{\mathrm{ISO}}=f\left(\frac{e_{\mathrm{C}}C_{\mathrm{u}}}{P_{\mathrm{eq}}},\frac{\nu}{\nu_1}\right), \tag{27}$$

where e_{C} is the contamination factor, ν is the kinematic viscosity of the bearing lubricant (or that of the base oil for grease), and ν_1 is a reference kinematic viscosity, below which the rating life is reduced and above which the rating life is extended. The bearing fatigue load limit, C_{u}, is conceptually the applied load below which the bearing will not fatigue; therefore, $L\to\infty$. The fatigue load limit is specified in the bearing manufacturer's

§This relationship may be derived from the standard expression for the gradient of a straight line given two points (x_1,y_1) and (x_2,y_2) on that line.[27] The straight line in question is that which exists between $\ln L$ and $\ln\ln 1/S$ Weibull distribution values (Equation 4).

¶This is equivalent to plotting based on Equation (25), since the two expressions are equivalent. One is simply a rearranged version of the other.

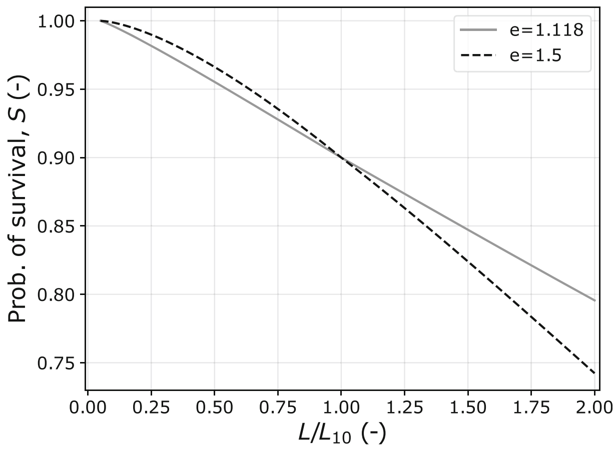

FIGURE 1 S versus L/L_{10} (obtained from Equation (26)) using two different Weibull slopes of $e = 1.5$ and $e = 1.118$.

catalogue or, alternatively, semiempirical equations given in ISO 281, and ν_1 is specified by different formulas depending on the shaft speed, Ω. For the system considered in this paper, the manufacturer's specification of fatigue load limit was used and

$$\nu_1 = 45000\Omega^{-0.83}D_p^{-0.5} \text{ for } \Omega < 1000\text{rpm and } D_p \text{ is the bearing pitch diameter in mm.} \tag{28}$$

The kinematic viscosity, ν, may be calculated using the prescribed ASTM method[37] (among other approaches), which accounts for the operating temperature of the bearing. For further information on the derivation and limitations of Equation (28), refer to Baalmann[38] and Heemskerk.[39] Contamination factor (e_C) equations all have the same form, but contain coefficients which vary with the level of contamination. For grease-lubricated bearings, there are five categories of contamination ranging from "high cleanliness" to "very severe contamination." Contamination levels are affected by assembly environment cleanliness, material ingress through bearing seals, and regreasing intervals. In reality, contamination may also occur due to the shedding or wear of metallic particles from internal bearing surfaces during operation. However, current standards do not prescribe a methodology to account for the evolution of this form of contamination. For grease-lubricated bearings, grease flushing (if used) and regreasing intervals impact the category of contamination which should be selected.[25] As an example, under conditions of "normal cleanliness" the e_C equation for grease lubrication is[25]

$$e_C = \left(1 - \frac{1.141}{D_p^{1/3}}\right) \cdot \min\left[0.0432\left(\frac{\nu}{\nu_1}\right)^{0.68} D_p^{0.55}, 1\right]. \tag{29}$$

The factor a_{ISO} is then calculated using equations of the same form as each other, but with differing coefficients based on the value of ν/ν_1 (which itself is capped at 4). For the case $1 \le \nu/\nu_1 \le 4$ in a radial roller bearing,

$$a_{ISO} = \min\left[0.1\left(1 - \left(1.5859 - \frac{1.2348}{(\nu/\nu_1)^{0.071739}}\right)\left(\frac{e_C C_u}{P_{eq}}\right)^{0.4}\right)^{-9.185}, 50\right]. \tag{30}$$

In any case where $e_C C_u / P_{eq} > 5$, a_{ISO} is also set equal to 50. The effect of grease contamination on modified rating life can be independently assessed by varying the value of e_C; however, the effect of the fatigue load limit on modified rating life is inherent in the formulation and cannot be removed. The formulas for a_{ISO}, C_u, e_C, and ν_1 were all developed based on a combination of computer-supported theory, empirical testing, and practical experience.[25] Extensive details on their development are provided in ISO 1281-2:2008.[36]

2.1.3 | Variable operating conditions

The need to accommodate variable operating conditions when characterizing bearing life was also considered by Palmgren. Over a decade before the same was independently proposed by Langer and then Miner, Palmgren outlined the first example of a linear damage accumulation rule.[27] This rule states:

1. A bearing operating for m_1 revolutions[#] under conditions for which the rating life is L_1 has (on average) consumed a proportion m_1/L_1 of its overall rating life.
2. The quantity m_1/L_1 is interpreted as the "proportional damage" associated with the time spent in this operating state.
3. Proportional damage from operating under different conditions is assumed additive, with failure (more specifically, failures in the population reaching 10%) occurring when the summation equals 1 (i.e., when the entire rating life has been proportionately consumed).

Considering the case where a bearing population operated under varied conditions (identical for each bearing in the population) reaches failures of 10%, it must be that

$$\frac{m_1}{L_1}+\frac{m_2}{L_2}+\ldots+\frac{m_n}{L_n}=1. \tag{31}$$

The resultant rating life for this bearing is therefore $L=m_1+m_2+\ldots+m_n$. Dividing by this term on both sides and taking reciprocals, we obtain

$$L=\frac{1}{\frac{\phi_1}{L_1}+\frac{\phi_2}{L_2}+\ldots+\frac{\phi_n}{L_n}}, \tag{32}$$

where $\phi_i=m_i/(m_1+m_2+\ldots+m_n)$ is the proportion of total operation which occurred in operating condition i. Under the assumption of linear damage accumulation, Equation (32) therefore provides a means to determine the *resultant rating life* of a bearing operating in varied conditions. In order to do this, all that is required is the rating life associated with each bearing operating condition and the proportion of total operational time/revolutions (depending on units) spent there.

The following analysis demonstrates a useful property of the resultant rating life formula. To the best of the authors' knowledge, this result has not been presented in the literature before now. Assume bearing operation proceeds under N different conditions, each containing subconditions. Let conditions k, for $k=1,\ldots,N$, occur with proportion ψ_k such that $\sum_{k=1}^{N}\psi_k=1$. Now, let there exist n_k subconditions within conditions k, each occurring with proportion φ_{k,i_k} for $i_k=1,\ldots,n_k$, such that $\sum_{i_k=1}^{n_k}\varphi_{k,i_k}=1$, and with the associated rating life L_{k,i_k}. Applying Equation (32), the equivalent life under conditions k is

$$L_k=\frac{1}{\sum_{i_k=1}^{n_k}\frac{\varphi_{k,i_k}}{L_{k,i_k}}}. \tag{33}$$

If Equation (32) is applied again across the L_k, we obtain what will be referred to as the "multistage" resultant rating life,[‖]

$$L_{\mathrm{ms}}=\frac{1}{\sum_{k=1}^{N}\frac{\psi_k}{L_k}}. \tag{34}$$

Observe that substituting Equation (33) into Equation (34) results in

$$L_{\mathrm{ms}}=\frac{1}{\sum_{k=1}^{N}\sum_{i_k=1}^{n_k}\frac{\psi_k\varphi_{k,i_k}}{L_{k,i_k}}}. \tag{35}$$

Because $\psi_k\varphi_{k,i_k}$ is the overall proportion at which subcondition (k,i_k) occurs, from Equation (35), it follows that L_{ms} is equal to the equivalent rating life obtained by combining rating lives from across all conditions and subconditions simultaneously in a "single-stage" application of Equation (32); that is, it has been demonstrated that $L_{\mathrm{ms}}=L_{\mathrm{ss}}$, where L_{ss} is the single-stage resultant rating life. This equivalence proves useful when undertaking fatigue analyses under variable conditions, as will be described in Section 3. Note that the equivalence of resultant rating lives obtained through single-stage or multi-stage combining has been demonstrated here across two "levels." This immediately generalizes to any number of levels through repeated application of the two-level result.

[#]Or m_1 units of time, if L_1 has been converted to units of time.

[‖]So-called due to the repeated application of Equation (32).

Where variable operating conditions include changes in the shaft rotational speed, it is recommended that rating lives for each set of conditions are calculated and immediately converted to units of time prior to considerations of damage accumulation and the calculation of a resultant rating life. This avoids further approximations that become necessary if a (combined) resultant rating life in revolutions must then be converted to a time duration for a bearing which is operated at various speeds.

ISO 281 does not include guidance on how to handle variable operating conditions with the load P_{eq} assumed constant. Linear damage accumulation, as presented above, is the most common approach to handling variable conditions in rolling bearing life assessment.[27] Other approaches also exist wherein loads themselves are combined, rather than rating lives, into a single value prior to application of the ISO rating life equations. While both sensible and intuitive, it must be recognized that linear damage accumulation is a significant assumption, the validity of which is likely to change depending on the conditions experienced and the levels/nature of the variability.

2.1.4 | Possible shortcomings

ISO bearing design standards must be recognized as phenomenal scientific achievements. The task of generating practical methods and equations to establish a universal framework for bearing design/selection, which includes myriad effects and complexities, is a challenging one indeed. Experience has shown that across many industries and applications, ISO 281, among other ISO standards, does just this. However, as is inevitably the case with such a general framework, which is also imbued with historical precedent, certain potential issues and omissions must be acknowledged. Perhaps the most important of these is the existence (or absence) of a fatigue load limit (C_u). While this concept has been adopted by the ISO standard, the available experimental evidence does not appear to support the hypothesis that a fatigue load limit exists for through-hardened bearing steels.[27,40] There is, therefore, concern that the inclusion of a fatigue load limit in ISO 281 may result in significantly overpredicted rating lives in some cases, introducing the risk of undersizing a bearing for its application.[41,42] Another issue concerns exponent values[**] in load-life equations (Equations (13)–(15)). While the Lundberg and Palmgren exponent values are used across the board in ISO 281, there is a significant body of evidence that shows (1) these values are not equally valid in all cases, especially for modern steels, and (2) the exponents used in ISO 281 have a conservative influence on rating lives in cases where a larger exponent is applicable/correct.[27,40] In addition, the influence of steel cleanliness (i.e., the prevalence of inclusions, which act as stress raisers) on load life exponents is also now better understood.[40] It has also been shown that exponent values may change in the presence of surface damage,[27] introducing further complexities into the problem of rolling bearing life prediction. Bearing life is also affected by the presence of residual internal stresses, which may result from manufacturing and treatment processes as well as operational conditions. Subsurface compressive residual stresses can help prolong bearing life,[†] whereas tensile residual stresses will shorten it. Generally, tensile stresses may be introduced into the inner ring subsurface as a result of press-fitting to the shaft or as a result of thermal or centrifugal effects.[27] These so-called hoop stresses can therefore negatively affect rolling contact fatigue life. Finally, uncertainties around the validity of linear damage accumulation for evaluating rating lives under variable operating conditions (refer to Section 2.1.3) are again emphasized, as are unknown levels of lubricant starvation. The listed effects are not accounted for in ISO 281 rating life predictions, nor are misalignment/tilt and bearing clearance (note that these latter effects are considered within ISO/TS 16281). These uncertainties only grow when considering larger bearings. This is because endurance tests are generally only carried out on small- to medium-sized bearings,[40] with results extrapolated using computational models.[36] In the context of large bearing rating lives, such as those used in modern wind turbines, there is therefore significant uncertainty stemming from a combination of known unknowns (fatigue load limit, exponent values, residual stresses) and possible unknown unknowns (new effects/interactions).

2.2 | Wind turbine design requirements—IEC 61400-1

IEC design requirements for wind energy generation systems[24] specify that the main bearing modified rating life at 90% survivability, L_{10m}, should equal or exceed the design life of the turbine (20+ years). The loads to be considered are specified in a set of design load cases (DLCs). For analysing fatigue of the main bearing, the most relevant DLC is 1.2, wherein normal power production simulations are undertaken across the wind speeds between turbine cut-in and cut-out. Design loads must be predicted using an aeroelastic and structural dynamics model that includes control system behaviour. Wind field turbulence must be represented by the Mann model,[43,44] Kaimal model,[45] or a similar kinematic turbulence model. The IEC standard provides turbulence standard deviation values (as a function of hub-height mean wind speed) in longitudinal, lateral, and vertical directions, corresponding to what is classed as high (a), medium (b), and low (c) turbulence conditions. Due to the stochastic nature of

**In addition to the e-value inconsistency raised in Section 2.1.2.

†It may even be the case that what was previously interpreted as a fatigue load limit during model fitting may in fact have been a reflection of the influence of compressive residual stresses.[27]

turbine inflow and the resultant load time histories, statistical convergence is sought by requiring that a minimum of six 10-min simulations are undertaken at each considered operating point. Appropriate safety factors, specified in the IEC standard, must also be applied as directed. The IEC standard indicates that, with respect to bearing rolling contact fatigue, the basis for rating life calculations should be ISO 281 and ISO/TS 16281. As discussed in Section 2.1, the focus of the current paper is the rating life provided by ISO 281. Possible shortcomings are also present for the load evaluation methodology of IEC 61400-1. This is mainly due to the necessary use of medium-fidelity modelling to allow for the evaluation of a large number of design load cases in reasonable time. Similarly, the kinematic wind fields used in aeroelastic simulations have the correct second-order statistics but do not contain true turbulent-eddy structure.[17] While experience has shown IEC design standards to provide a strong basis for wind turbine design, it is important to be aware of these limitations.

2.3 | ISO 281 main bearing life assessment

As indicated in Section 1, the work of Yucesan and Vianna[7] is relevant to the current study. This prior work presented an elegant and general framework for physics-informed cumulative damage modelling which accounts for the design, operation, and maintenance intervals of the considered component. Implications for optimal fleet maintenance scheduling were also considered. The study focused on the double-row, spherical roller main bearing in a three-point-mount, 1.5 MW wind turbine. Modified rating lives were determined similarly to those in IEC 61400-1 and ISO 281, but using the modification factor a_{SKF} (developed by SKF) rather than a_{ISO}. The impact of variable conditions on the main bearing modified rating life was considered assuming linear damage accumulation. The framework included grease progressing from fresh to degraded condition over time, until a regreasing event occurs and resets to fresh grease again. Variations in the grease temperature over time were estimated using an analytical model. This was necessary since measurements of main bearing temperature variations over time were not available. Crucially, in the context of this paper, their results predicted main bearing modified rating lives between 8 and 30 years (dependent on the wind speed, bearing temperature, and level of grease contamination). If this is indeed the case, then rolling contact fatigue may actually be responsible for many, even most, of the reported main bearing failures. There are, however, potentially problematic pitfalls in this previous work. While the framework itself remains a valuable contribution, the characterization of lubricant viscosity in degraded condition and the predicted main bearing operating temperatures appear to be at odds with the available data. Taking the former, Yucesan and Vianna treat the viscosity of fully degraded grease as being approximately half that of the fresh grease at the same temperature. However, the combined mechanical and chemical degradation of lubricating greases is complex and not well represented by a simple change in base oil viscosity.[46] Indeed, under chemical degradation oil viscosities can actually increase due to polymerization reactions.[47] With respect to main bearing temperatures, those used in the Yucesan and Viana study all fall between 60°C and 80°C, significantly higher than measured values for main bearings presented in other studies[13,16,48,49] where the standard range is 20°C to 40°C, with values consistently higher likely indicating damage. At a measured main bearing temperature of 60°C or 70°C, some main bearing monitoring systems will trigger an alarm and a recommendation to shut down the turbine.[49] It may therefore be the case that the low values of main bearing modified rating life were largely driven by unrealistic temperature (and hence viscosity) values. Note that studies also exist in the literature that undertake main bearing fatigue life assessment utilizing frameworks other than ISO 281.[11,50,51]

3 | METHODOLOGY

Rating lives for a 240/630 and a 230/600 main bearing were assessed for a 1.5 MW wind turbine. These are double-row, spherical roller bearings installed in a three-point suspension (single main bearing) configuration. Both bearings have been used in practice for wind turbines of this size[1,4,10] and have appeared in previous studies.[9,19] Together, they represent both the larger (240/630) and smaller (230/600) limits of the spherical roller bearings selected for application in turbines around this power rating, and hence, they roughly span the design space. The 230/600 is the bearing considered in Yucesan and Vianna.[7] Details of these bearings are provided in Table 1. The bearings are grease-lubricated, with grease properties listed in Table 2 taken from a commercially available industrial grease specifically marketed for main bearing applications. Extrapolation to other temperatures was achieved using the ASTM method.[37] These same grease properties were applied in a previous main bearing study.[19]

The 1.5 MW wind turbine was simulated using DNV-GL Bladed, an industry-utilized wind energy modelling software certified for wind turbine design as codified in the IEC standards. Kinematic wind field turbulence was generated using a Kaimal spectrum. The turbine simulation model is identical to that used in two previous main bearing studies.[9,19] Complete DLC 1.2 (refer to Section 2.2) simulations were performed using the chosen turbine model for three cases: (1) a baseline case of medium turbulence and a 0.2 power-law shear exponent; (2) adjusted-shear cases using 0.1 and then 0.4 shear exponent values; (3) adjusted-turbulence cases using low and then high turbulence. In each instance, only a single variable is adjusted from the baseline case. Each set of DLC 1.2 results included six simulations at each hub-height mean wind speed[‡‡] considered

‡‡This is in order to reach some level of statistical convergence in results. Overall, for each combination of considered wind field parameters (mean wind speed, shear exponent, turbulence level) six different random wind fields were generated (and subsequently used to simulate main bearing loads) while keeping those parameters fixed. Refer to Section 2.2.

TABLE 1 Main bearing data.

Bearing model	Pitch diameter D_p (mm)	Contact angle α (deg)	Dynamic load rating C_D (kN)	Fatigue load limit C_u (kN)	"Limiting value"[a] of F_a/F_r
240/630	775	11.00	7530	1141	0.29
230/600	735[b]	8.34	6000	750	0.22

Abbreviations: deg, degrees; kN, kilonewton.
[a]Refer to Section 2.1.2 for explanation of the "limiting value."
[b]The pitch diameter for the 230 bearing is estimated as the mean diameter.

TABLE 2 Grease base-oil viscosity and contamination levels cases.

Case	Base-oil viscosity ($mm^2\,s^{-1}$)	Contamination level
#1	$\nu=460$ (40°C), $\nu=16$ (100°C)	Slight to typical
#2	$\nu=460$ (40°C), $\nu=16$ (100°C)	Very severe
#3	$\nu \rightarrow \nu/2$ everywhere	Very severe

between cut-in and cut-out (2 to 24 m/s in increments of 2 m/s). Sixty-six 10-min simulations were therefore performed for each set of wind field parameters, resulting in a total of 330 simulations. Hub loads (forces and moments) were extracted from each simulation at 20 Hz, and loading at the main bearing was estimated by applying a static force balance at each point in time.[5] In this context, the main bearing axial force magnitude is equal to wind turbine thrust and a contribution from rotor weight due to drivetrain tilt, $F_a=F_x$, and the main bearing radial force magnitude is given by

$$F_r=\sqrt{\left(\frac{F_y(L_1+L_2)-M_z}{L_2}\right)^2+\left(\frac{F_z(L_1+L_2)+M_y}{L_2}\right)^2}, \tag{36}$$

where F_y and F_z are hub forces and M_y and M_z are hub moments in the Bladed frame of reference.[§§] The distance from the hub centre to the main bearing centre is $L_1=2.145$ m, and the distance from the main bearing centre to the gearbox torque arm supports is $L_2=2.615$ m. Time histories of the low-speed-shaft rotational speed, Ω, were also extracted from each simulation. No safety factors are required by IEC 61400-1 on the calculated loads during bearing rating life assessment.

Basic and modified rating lives were then calculated per ISO 281 as described in Section 2.1. Checks were undertaken to confirm that at no point was $P_{eq}>C_D/2$ for either bearing. Modified rating lives were calculated assuming a constant temperature throughout the bearing's operational life, with considered temperatures ranging from 30°C to 70°C. Additionally, three lubricant property and contamination level cases summarized in Table 2 were considered. Case #1 is for the baseline grease viscosity with slight contamination while Case #2 increases the contamination level to very severe. Case #3 further reduces the viscosity by a factor of 2 at all temperatures, which is similar to the representation of fully degraded grease applied in Yucesan and Vianna.[7] As discussed in Section 2.3, this is not necessarily a valid characterization of degraded grease. As such, in the current paper, Case #3 is interpreted as simply a case of a *reduced viscosity lubricant with severe contamination*, without claiming any particular link to grease degradation. Load and speed variability were handled via multistage linear damage accumulation (refer to Section 2.1.3), as will be described. First, within each simulation and at each point in time, t, L_{10} and L_{10m} values were calculated for the value of load and speed occurring at time t. These were then converted to units of years by scaling by $10^6/(\Omega(t)\cdot 60\cdot 24\cdot 365)$, for Ω in rpm. All calculated rating lives within each simulation were combined using Equation (32) and equal weightings of $\phi_t=1/(600\cdot 20)$, which are 10-min simulations at 20 Hz. The six resultant rating lives corresponding to each value of hub-height mean wind speed (recall that for each combination of wind parameters, simulations were performed for six different randomly generated wind fields with parameters in common) were combined using Equation (32) and equal weightings of $\phi_i=1/6$. At this stage, rating lives have been combined to the point where a single resultant rating life has been determined for each value of hub-height mean wind speed. Applying Equation (32) to this final set of values requires appropriate weightings, ϕ_v, for the time spent at each mean wind speed, v. Coincidentally, 10-min mean wind speeds are also commonly described using a Weibull distribution,[52] albeit with a different parameterization to that of Section 2.1. Weibull wind speed distributions are parameterized according to a shape factor, k, and scale factor, C. A standard wind site value of $k=2$ was assumed. C may then be approximated from k and the site annual mean wind speed,[52] taken here to be 10 m/s. Having identified wind site distribution parameters, ϕ_v values are readily obtained by differencing cumulative Weibull distribution values, that is,

[§§]$x=$ downwind, $z=$ vertically upwards, $y=$ lateral (conforming to right-hand rule).

$$\phi_v = \exp\left(-\left(\frac{v-1}{C}\right)^k\right) - \exp\left(-\left(\frac{v+1}{C}\right)^k\right), \tag{37}$$

for each v. Note that the terms $v \pm 1$ are due to the mean wind speed being incremented by 2 m/s. Rating lives across the mean wind speeds were therefore combined using the previously mentioned weighting factors and Equation (32), arriving at the final resultant basic and modified rating lives (in years) for the selected bearing under specified conditions of wind characteristics, temperature, lubricant properties, and contamination. Finally, the formulations of Section 2.1.2 were applied to resultant rating lives in order to also determine the expected percentage of failures in the main bearing population at year 20 for each case and at each temperature. This analysis was undertaken for two values of Weibull slope, $e = 1.118$ and $e = 1.5$.

4 | RESULTS

A detailed breakdown of basic rating life, L_{10}, for each main bearing is provided in Table 3 for the different wind field parameters. Basic rating lives all fall between 126 to 174 years for the 240 bearing and 34 to 44 years for the 230 bearing. The effects of wind field parameters are consistent across all cases and bearings, and these will be considered first. The rating life is only weakly affected by changes in turbulence level, with a slight reduction under higher turbulence and a slight increase for lower turbulence. Changes in the power-law shear exponent value elicit a more pronounced effect. It is interesting to note that the rating life increases when considering the larger shear exponent of 0.4 and decreases for a shear of 0.1. This result seems counterintuitive, since higher shear is normally associated with larger fluctuations in load and therefore shorter component lives. However, with respect to rolling contact fatigue, cycles are principally considered in the context of rollers orbiting the bearing under a constant applied load. In this setting, the bearing rings experience fluctuating internal/surface stresses due to the passage of individual rollers, and the rollers experience fluctuating stresses and forcing as they rotate and orbit, passing in and out of the loaded zone. The resulting material stress fluctuations, under a constant applied load, lead to failure via rolling contact fatigue. It is in this context that ISO 281 seeks to predict the rating life of a bearing population. In ISO 281, fluctuations in bearing applied loads and their influence on rolling contact fatigue life are therefore not considered. Even when accounting for variable operating conditions by assuming linear damage accumulation (refer to Section 2.1.3), one is effectively still considering a series of independent, constant load cases and simply undertaking a process of weighted averaging to account for the duration spent in each. The upshot of this, in the context of wind-induced loading, is that the principal driver of rating life here is the mean load experienced by the main bearing. This is consistent with the low sensitivity observed between rating life and turbulence level, since kinematic turbulence levels mostly influence the fluctuations rather than the mean value. The shear results may be interpreted in the same light, since a higher value of shear exponent leads to a rotor moment that counteracts the rotor weight moment and so reduces the mean radial load on the main bearing.[15,17] Hence, a larger shear exponent leads to a reduction in the main bearing mean load and an increase in the rating life, whereas the opposite occurs when the shear exponent is reduced. This is exactly the behaviour observed in rating life results when the shear exponent is varied.

Modified rating lives at 90% survivability, L_{10m}, for each main bearing are presented in Figure 2, along with the range of basic rating lives from Table 3 and the desired 20-year design life. As described in Section 2.1.2, L_{10m} includes the effect of temperature on lubricant viscosity, lubricant contamination, and the bearing fatigue load limit. Each of the three viscosity contamination cases are now examined. Case #1 provides the most accurate representation of the viscosity and contamination levels sought during operation. The Case #1 contamination level corresponds to "clean

TABLE 3 Basic rating life, L_{10}, results.

Bearing model	Shear exponent	Turbulence level	L_{10} (years)
240/630	0.1	Med.	126
"	0.2	Low	142
"	0.2	Med.	141
"	0.2	High	139
"	0.4	Med.	174
230/600	0.1	Med.	34
"	0.2	Low	37
"	0.2	Med.	37
"	0.2	High	37
"	0.4	Med.	44

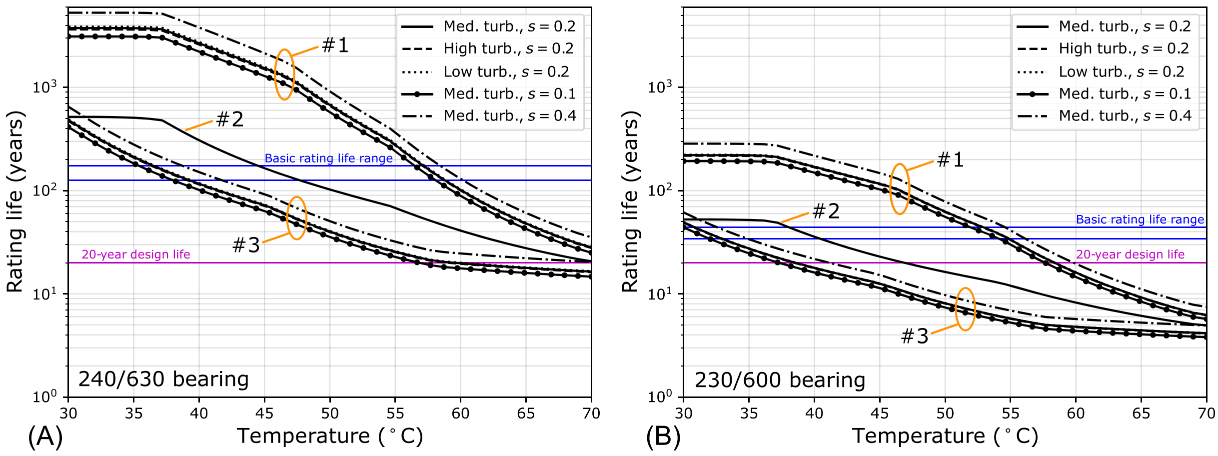

FIGURE 2 Rating lives at 90% survivability, that is, L_{10} (blue) and L_{10m} (black), for bearing (A) 240/630 and (B) 230/600 under each of the cases listed in Table 2. For the sake of clarity, only the medium turbulence and shear 0.2 result is shown for Case #2.

assembly; moderate sealing capacity in relation to operating conditions; regreasing according to manufacturer's specifications,"[25] all of which appear entirely reasonable. The highest sensitivity for Case #1 is from temperature, with modified rating lives changing from around 4000 and 220 years to around 28 and 6 years for the 240 and 230 bearings, respectively, as the temperature increases from 30°C to 70°C. This strong dependence on temperature is not surprising, given the high sensitivity of lubricant viscosity, and hence lubricant film thickness, to temperature.[18,19] That being said, the scale of the changes in Figure 2 for modified rating life with temperature are dramatic. Case #1 modified rating lives only fall below basic rating lives at higher temperatures, around 59°C and 55°C for the 240 and 230 bearings, respectively. For the 240 bearing, the modified rating life will not fall below 20 years until temperatures reach >70°C. For the 230 bearing, this occurs around 58°C. Case #2 has the same viscosity characteristics as Case #1, but with very severe contamination representative of "assembly in contaminated environment; inadequate sealing; long regreasing intervals."[25] Such conditions would only normally be expected to occur for more extreme applications, such as mining, and/or if regreasing activities are neglected. While severe contamination could conceivably result from a buildup of damage and/or wear particles, this would require damage of some sort to already be present. This introduces potential issues around the conceptual consistency of predicting the onset of damage if damage is already present, as well as completeness issues with regards to the types of damage accounted for (i.e., if non-rolling contact fatigue damage, such as wear, ultimately determines bearing life, then that primary mechanism should ideally be understood and its onset predicted prior to analysis of any subsequent impacts on fatigue). Modified rating lives for Case #2 are significantly reduced compared to Case #1, especially at lower temperatures. Case #2 modified rating lives fall below basic rating lives around 48°C and 41°C for the 240 and 230 bearings, respectively. They then fall below 20 years for temperatures of around 70°C and 47°C, respectively. Case #3 conditions compound the severe contamination of Case #2 with a viscosity reduced by a factor of 2. Case #3 modified rating lives experience a further reduction from Case #2, but less than the change between Cases #1 and #2. Case #3 results fall below basic rating lives around 39°C and 33°C for the 240 and 230 bearings, respectively. They then fall below 20 years for temperatures of around 59°C and 38°C, respectively. While temperature represents a strong sensitivity in the rating life results, it is arguably not a significant source of uncertainty. As outlined in Section 2.3, data available in the literature regarding main bearing operating temperatures put the value for undamaged bearings somewhere between 20°C and 40°C, in general. It should be understood that reported values tend to be those measured on the bearing housing[13,48] or side-face,[16] as opposed to the bearing raceway. However, for low-speed bearings, any discrepancy is not expected to be large. Internal bulk lubricant temperature values would therefore be expected to be perhaps only a few degrees above what is measured externally. Second, because in the current analysis a constant temperature is assumed throughout the operational life, these temperatures represent the mean/effective value over the bearing life. As such, the most representative value will lie somewhere between temperature extremes. Taking both points into account, it seems reasonable to infer that a mean temperature in excess of 40°C is unrealistic in the context of this analysis. Indeed, the true upper limit for a reasonable bulk-lubricant mean operating temperature may in fact be less than 40°C. The above modified rating life results, especially when considering realistic temperature ranges, are in stark contrast with the field data findings of Hart et al,[6] where 90% survivability (i.e., the field L_{10} life) occurred at 10.5 years on average, and the field L_{10} lives of 8–11 years reported in Chovan[4] for 1.5 MW wind turbines with 240 and 230 series main bearings.

Figure 3 shows the percentage failures expected at year 20, obtained from the rating lives of Figure 2, for each case under the baseline inflow conditions of medium turbulence and a 0.2 shear exponent. Note, in Figure 3A, the percentage of failures tends to zero at lower temperatures. Plotted values are limited to 10^{-3} so as to remain visible. Results are shown for Weibull slopes of 1.118 and 1.5. As expected from the analysis of Section 2.1.2, the smaller value of Weibull slope is more conservative (regarding survival probabilities) where failures are less than 10%, with the

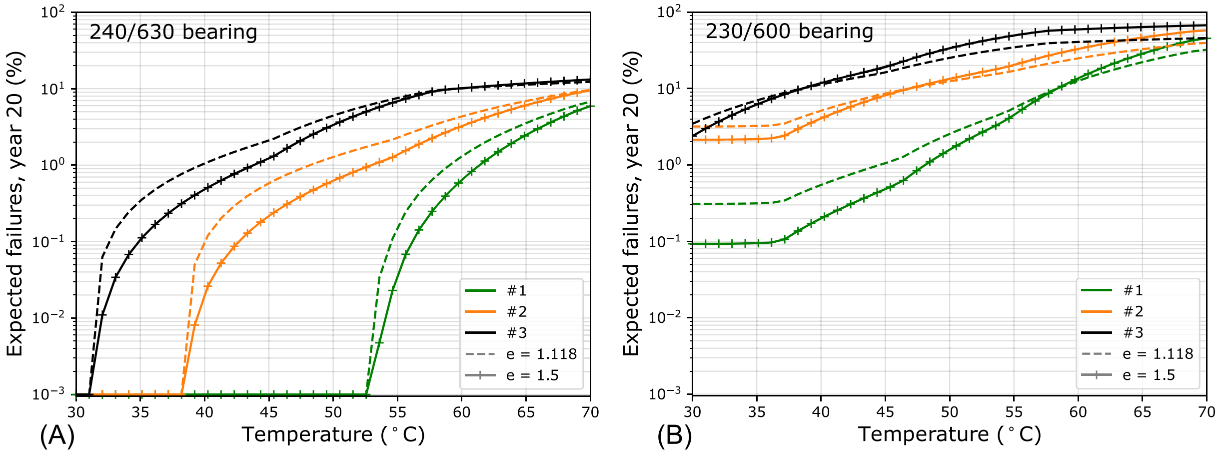

FIGURE 3 Expected failures at year 20 for bearing (A) 240/630 and (B) 230/600, obtained from L_{10m} results assuming Weibull slopes of 1.118 and 1.5. Presented results are under medium turbulence and a 0.2 shear exponent. Note that in (A), expected failures tend to 0 at lower temperatures. These values are limited to 10^{-3} so as to remain visible.

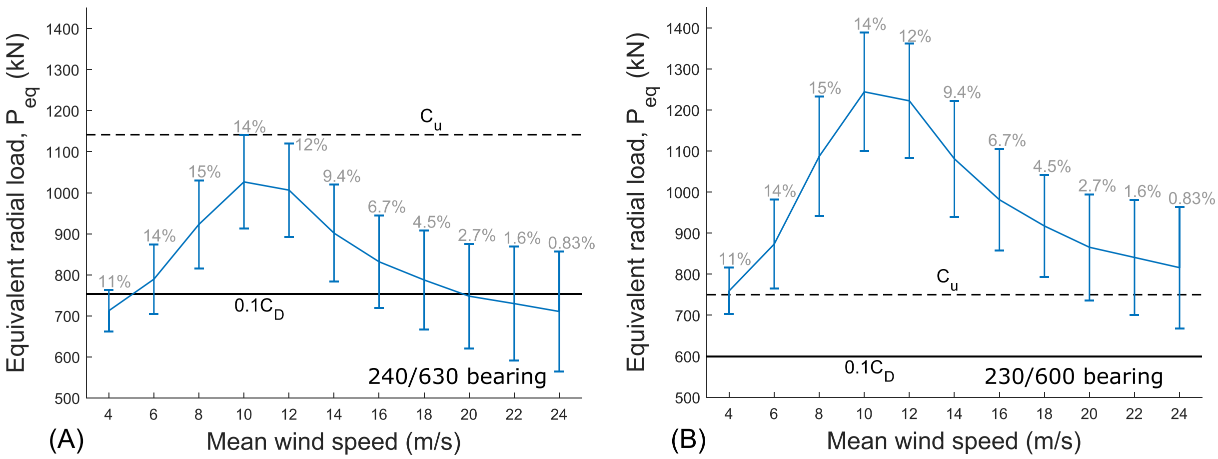

FIGURE 4 Main bearing equivalent radial loads, for bearing (A) 240/630 and (B) 230/600, across the turbine operational envelope and for the baseline case of medium turbulence and a shear exponent of 0.2. The mean value plus/minus one standard deviation is plotted at each hub-height mean wind speed. The expected proportion of time spent at each operating point is also indicated, as well as each bearing's fatigue load limit, C_u, and 10% of its dynamic load rating, C_D.

larger slope more conservative where failures increase above 10%. Temperature has a strong influence on year-20 failures, consistent with the temperature sensitivities of previous results. Under Case #1 conditions, very few failures are expected to occur by year 20 across most of the temperature range. For the 240 bearing, Case #1 expected failures at year 20 remain below 1% until temperatures reach 59°C and do not exceed 10% below 70°C. For the 230 bearing and Case #1 conditions, 1% is exceeded at around 45°C, and 10% is exceeded around 58°C. Temperature thresholds for 1% and 10% reduce under Case #2 and #3 conditions. For the 240 bearing and Case #2, the temperature thresholds for 1% and 10% occur around 48°C and 70°C, and for Case #3, they occur for 40°C and 59°C, respectively. For the 230 bearing, the percentage failures at year 20 always exceed about 2% for Cases #2 and #3. The 10% thresholds are exceeded at 47°C for Case #2 and 38 °C for Case #3. The associated expected failures, at year 20, obtained from L_{10} values, were 0.3%–0.8% for the 240 bearing and 3.8%–4.9% for the 230 bearing.

Having studied the bearing rating lives, it is instructive to also consider the loads that drive them. Figure 4 presents main bearing equivalent radial loads, P_{eq}, for each bearing across the turbine's operational envelope in the baseline case of medium turbulence and a shear exponent of 0.2. Results are plotted as mean values, plus/minus one standard deviation, calculated from the six 10-min simulations at each hub-height mean wind speed. Fatigue load limits, C_u, and 10% of the dynamic load rating, C_D, are also shown for each bearing, along with the expected proportion of time spent at each operating point (based on the assumed Weibull wind speed distribution). The equivalent radial load is strongly dependent on the turbine operating point, with a peak in P_{eq} occurring close to the rated wind speed. Load variability increases with wind speed, indicated by increasing standard deviations. P_{eq} remains well below C_D for both bearings, and its relationship to the fatigue load limit, C_u, is notable. These results indicate that the 240/630 bearing is operating at loads that fall below its fatigue load limit for the vast majority of its operational life.

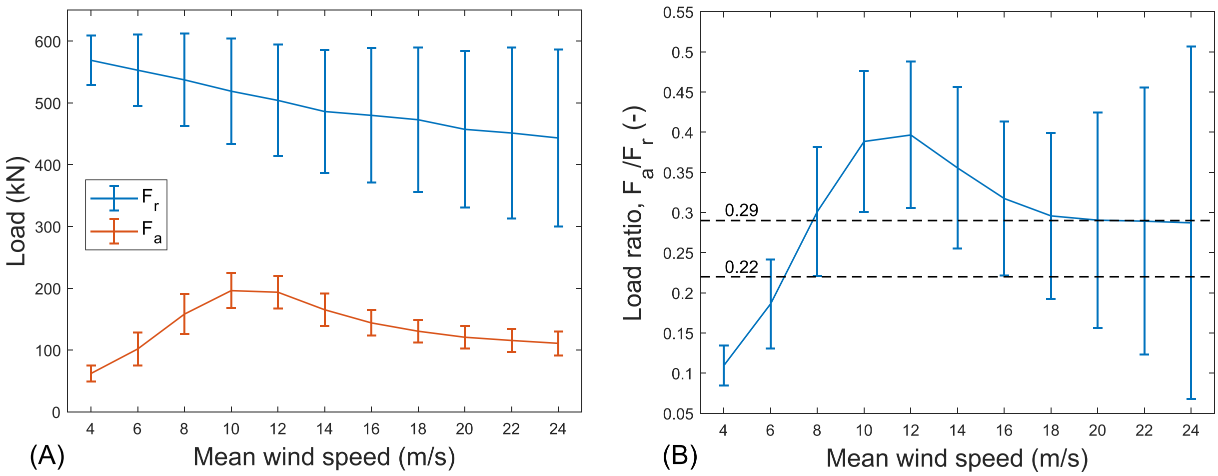

FIGURE 5 (A) Constituent axial and radial loads, F_a and F_r, respectively, of which P_{eq} is composed. (B) Load ratios, F_a/F_r, at each turbine operating point, along with the "limiting values" of F_a/F_r for the two bearings. All results correspond to the baseline case of medium turbulence and a shear exponent of 0.2. Results are presented as mean values plus/minus one standard deviation.

Considered alongside the fact that one source[3,4] has reported lower field L_{10} lives for the 240 bearing series (compared to the 230 bearing series in 1.5 MW turbines), it seems reasonable to ask whether underloading may be exacerbating 240 series failures. Alternative hypotheses related to the microslip tendencies of 240 versus 230 series bearings have also been proposed.[1] Given the 240 bearing is generally operating at loads below C_u, the generally high values of rating life noted in previous results is unsurprising, especially for clean, normal temperature grease. The 230/600 bearing has a smaller fatigue load limit that operational loads remain above a vast majority of the time. Figure 5A shows the constituent axial and radial loads, F_a and F_r, respectively, that make up P_{eq}. F_a mean values follow the turbine's thrust operating curve, as would be expected. F_r mean values steadily decrease as the wind speed increases, because the shear-induced rotor aerodynamic moment increasingly counteracts the rotor weight moment as the wind speed increases. While the variability in F_a remains similar across the operational envelope, the variability in F_r increases dramatically with wind speed.[¶¶] In all instances, the load distributions at each operating point were checked to ensure approximate normality. The presentations of load results as mean values plus/minus one standard deviation are therefore appropriate. Axial-to-radial load ratios, F_a/F_r, experienced during operation are also important. These ratios may directly impact bearing abrasion and micropitting[1] and are used to determine the coefficients from which P_{eq} is calculated.[25] In the case of the 240 bearing, ISO 281 prescribes that when $F_a/F_r \le 0.29, X = 1$ and $Y = 2.32$, whereas when $F_a/F_r > 0.29, X = 0.67$ and $Y = 3.45$ (refer to ISO 281 Table 8 entries for "Double-row" bearings[25]). Similarly, for the 230 bearing, ISO 281 prescribes that when $F_a/F_r \le 0.22, X = 1$ and $Y = 3.07$, whereas when $F_a/F_r > 0.22, X = 0.67$ and $Y = 4.57$. In the standard, the "damage contribution" from axial loading therefore increases (relative to radial loading) when the ratio F_a/F_r moves above the specified limiting value. The load ratios occurring at each turbine operating point, for the baseline case of medium turbulence and a shear exponent of 0.2, are shown in Figure 5B. Results are again presented as mean values, plus/minus one standard deviation, from the six 10-min simulations at each hub-height mean wind speed. Limiting values of load ratio for the two main bearings are also shown. Note that the load ratio tends to have a skewed distribution.[5] This should be kept in mind when interpreting these results. The mean load ratio is initially below the limiting value of each bearing at low wind speeds, before increasing steadily with wind speed up to around 12 m/s. The limiting value is passed at around 8 m/s for the 240 bearing and just beyond 6 m/s for the 230 bearing. At higher wind speeds, the mean load ratio drops and then levels off at around 0.29. Load ratio variability increases dramatically and monotonically with wind speed. From these results, in combination with those for P_{eq}, F_a, and F_r, it follows that trends in the mean values and variability of P_{eq} are driven by contributions from both F_a and F_r, with neither necessarily being dominant everywhere. The relative importance of each is strongly tied to X and Y dynamic load factors, and hence to the load ratio, F_a/F_r, at each point in time. Finally, it is worth noting that the largest load ratios observed in the data were as much as $F_a/F_r = 8$ in the tails of the load distributions. These more extreme values likely occur during radial unloading events, which have been previously documented.[9,17]

5 | DISCUSSION

The results presented in the previous section must be interpreted with care, owing to a number of uncertainties that have been discussed throughout the paper. First, there is the question of the existence and/or validity of the fatigue load limit. As described in Section 2.1.4, there is

¶¶Note that the characteristics and structure of main bearing radial load variability have been studied in detail in previous work.[9,17]

ongoing debate surrounding this concept. Especially for the 240 bearing, where loads fall below the fatigue load limit a majority of the time (and hence fall into the “low load” region), there is a direct risk of life overprediction. Note that the form of the semiempirical a_{ISO} equations (refer to Equation (30)) means it is not possible to isolate or remove the influence of the fatigue load limit on modified rating lives. A second source of uncertainty regards the validity of resultant rating lives obtained under the assumption of linear damage accumulation, discussed in Section 2.1.3. As was highlighted, this approach accounts for operational variability by assuming a series of constant-condition operating cases, from which the resultant rating life is determined via a process akin to weighted averaging. While this approach has proved sufficient for many applications, the limits of its applicability are not clear. Wind turbine main bearings have been shown to experience rapid and continual changes in loading throughout operation,[9,17] on timescales which are of the same order as the bearings' rotation. It is not clear to what extent linear damage accumulation is able to capture bearing fatigue life in this setting. Furthermore, resultant rating lives obtained this way would not be expected to account for damage initiated as a result of moving from one load case to another (e.g., if fatigue were initiated as a direct consequence of the rapid changes in load). For this same reason, caution is advised when interpreting the low sensitivity of rating lives to turbulence levels. If effects due to changing loads are indeed important here, then true sensitivities to turbulence levels may be higher than indicated by the presented results. Finally, and perhaps most importantly, there is significant uncertainty around the true lubrication conditions present within operating wind turbine main bearings as well as how these conditions change between relubrication events. This uncertainty includes the real-world effects of starvation, grease life and contamination, and “methodological uncertainty” with respect to their inclusion in the modified rating life via the ISO 281 viscosity and contamination factor variables. Related to this, Section 4 results appear to confirm that the low rating lives predicted by Yucesan and Vianna[7] were indeed the result of unrealistic temperatures. This follows from the fact that results in the current work are consistent with those of Yucesan and Vianna[7] only at temperatures above 55°C.

Considering the results of the current paper, rating lives were found to be sufficiently above the desired 20-year design life for the 230/600 bearing and far above this for the 240/630 bearing under expected operating conditions (Case #1, mean temperature over life < 40°C). For the 240/630 bearing, this appears to be strongly driven by P_{eq} being below or close to the fatigue load limit a vast majority of the time. Within the reasonable range for mean main bearing operating temperatures (<40°C), only in Case #3 for the 230 bearing do year-20 failures begin to approach the 22 to 30% reported in the literature.[5,6] However, as discussed, Case #3 includes the use of a reduced-viscosity grease and very severe contamination. While this was applied in a previous study to represent a degraded grease,[7] it is argued in Section 2.3 that the validity of this representation has not been demonstrated, with some results in the literature actually to the contrary. Case #3 results cannot, therefore, be considered a reasonable representation of main bearing conditions at this stage. Instead, they must simply be interpreted as a sensitivity indicator for the effect of halving the grease base-oil viscosity in the presence of very severe contamination. For the 240 bearing, and all other cases for the 230 bearing, only minimal instances of failure are predicted by year 20. These results only become more pronounced if one considers that the rating life corresponds to the first sign of fatigue damage, whereas main bearings in the field will be routinely operated for 6 to 9 months (or more) beyond the point at which damage is first observed. The proportions of main bearings which have damage present at year 20 will therefore be larger than the reported 22 to 30% figures, which generally corresponds to replacement after the onset of more severe damage.

Aside from the fatigue load limit, the influence of which is difficult to disentangle, the key sensitivities for rating life values were observed to be temperature (even between 30°C and 50°C) and contamination. Both of these variables were found to have a stronger influence on L_{10m} than shear and turbulence. As discussed in Section 4 and the current section, the reasons for this appear strongly linked to the application of linear damage accumulation in order to account for variable operating conditions. Again, caution is advised when interpreting this particular finding. The impact of wind shear on rating life values is also worth considering in the context of other flow phenomena a turbine may experience. In the case of vertical wind shear, a larger shear exponent increased the rating life by producing an overturning moment that counteracted rotor weight and so reduced the mean radial main bearing load. If, instead, *horizontal* wind shear was present, the resulting yawing moment would likely increase the mean radial main bearing load and hence reduce the main bearing rating life. Horizontal shear occurs for partially waked turbines, which may be the case for a majority of turbines in a wind farm at any one time. The impact of partially waked operation on main bearing rating lives should therefore be considered in future work. Similarly, yaw misalignment and wind veer have both been shown to affect main bearing loads[8,17] and should also be considered in the context of main bearing lives. Yaw misalignment has been found to principally drive changes in mean loading,[17] so it would be expected to impact the rating life. Veer, on the other hand, most strongly influences load variations about the mean,[17] so its impact on the rating life obtained via linear damage accumulation would likely be small. If, however, it transpires that changes in main bearing loading are an important driver of service life, then veer may play an important role in main bearing reliability.

While this analysis was undertaken for a single 1.5 MW aeroelastic model and two main bearings, it is argued that the findings presented here would be expected to hold more generally for two reasons: First, the selected bearings approximately span the design space for main bearings in three-point-mounted wind turbines of this power rating. As such, the results for other bearings would be expected to fall between those presented here. Second, the presented results show a low sensitivity of rating lives derived from ISO 281 to variations in turbine operating conditions; the mean load at each operating point is the principal driver of rating life values. While turbine models will differ in terms of their aerodynamic characteristics and controller specifics, in general, their operating strategies and rotor masses (which together drive the mean load levels in the drivetrain) will not deviate significantly at the 1.5 MW rating level, where turbine technology is relatively standardized.

With regard to the research question posed in Section 1, the findings provide evidence to support a negative answer. More specifically, the results of this study contribute evidence that supports the position that rating life assessment, as codified in IEC 61400-1 and ISO 281, does not account for the reported rates of main bearing failures in 1 to 3 MW wind turbines. As outlined in Section 1, if this is indeed the case (and if these findings extend to ISO/TS 16281), then one of the following scenarios (or a combination of both) must therefore hold: Either (1) rolling contact fatigue is not a dominant driver for premature main bearing failures (meaning observed failures are principally caused by other damage mechanisms) or (2) rolling contact fatigue (surface and/or subsurface) does contribute to the observed failures in wind turbine main bearings, but current rating life assessment methods do not capture it sufficiently and hence are unable to provide realistic rating life predictions. It seems likely that both cases hold to some degree. For instance, results showed that grease contamination had a strong influence on rating lives. If wear or other non-fatigue mechanisms cause a critical amount of metallic particles to enter the lubricant, entrainment and over-rolling may accelerate surface-initiated fatigue damage, which in turn releases further hard particles, and so on. Such a mechanism of evolving contamination, and its impact on rating life, is not dealt with by ISO 281. Similarly, ISO 281 does not address wear-driven failures, which could be a large portion of the observed failures. In this case, ISO 281 may provide an accurate rating life, since the rating life is, by definition, the operational duration prior to the onset of *fatigue* damage. As such, the results of this work do not necessarily imply an insufficiency with regards to ISO 281 itself. Aspects of the above discussion are considered in the analysis of main bearing damage reports in Hart et al.[6] Interestingly, spalling is reported in 80% of failure cases, with evidence of surface damage also obtained. This indicates that rolling contact fatigue is an important mechanism in main bearing damage and replacements. However, the study also highlights that spalling may be a secondary damage mode induced by wear and that the proportions of spalling cases resulting from "wear-induced," surface-initiated, or subsurface-initiated rolling contact fatigue are not currently known. As has been pointed out, ISO 281 explicitly states that it does not address wear damage. The extent/limit of its ability to account for surface-initiated rolling contact fatigue (via the modified rating life) is also not easily determined. In summary, ISO 281 has a number of limitations (refer to Section 2.1.4), which may result in the second scenario outlined above. However, the issue may instead be that main bearing operational conditions and damage mechanisms lie beyond where ISO 281 claims to be applicable and/or appropriate (e.g., if spalling is principally wear-induced). The resolution of this problem may therefore lie not in the enhancement of ISO 281 (and possibly ISO/TS 16281) but in the development of a new application standard which is specific to wind turbine main bearings. This latter route is non-trivial and would require extensive work to better characterize the true internal conditions for main bearings in wind turbines, including global bearing deflections, evolving grease contamination via the shedding of hard particles, and, critically, the identification of principal root causes of failure. Only then could the problem of predicting resulting service life be tackled. If a comprehensive enough database of main bearing failures and damage were collated, a *service life*[53] approach to main bearing reliability may also become possible. Middle-ground approaches could also prove fruitful—for example, a methodology to allow for metallic wear particle contamination and/or starvation and grease degradation effects to be accounted for via adjustments to the contamination factor and viscosity values applied in ISO 281 equations. This would constitute an "effective parameter-value" approach, that is, for a given set of conditions, the *effective contamination* and *effective viscosity* would be those which result in an ISO 281 predicted life that matches the actual life of the real-world component.[##] Such an approach would require careful validation.

ISO/TS 16281 provides methods that seek to account for additional effects. As such, a similar critique and analysis of main bearing rating lives under ISO/TS 16281 should be undertaken in future work. However, ISO/TS 16281 methods do not directly address the principal concerns which have been raised. Therefore, while it would certainly be of interest for future work to extend the current treatment to include ISO/TS 16281, the principal findings and conclusions may remain largely unchanged.

6 | CONCLUSIONS

This paper has studied the rating lives of wind turbine main bearings as determined by IEC 61400-1 and ISO 281. A review of bearing rating life theory was provided, followed by a critical discussion of the ISO 281 equations for basic and modified rating lives. Accounting for variable operating conditions under the assumption of linear damage accumulation was discussed, and a new equivalence result was presented, which proves useful in rating life implementations. Shortcomings of the ISO 281 standard were outlined, particularly with respect to the existence or absence of the fatigue load limit. The validity of the linear damage accumulation assumption was also considered. An exploratory case study was then undertaken to determine rating lives for two models of main bearing—240/630 and 230/600 bearings, which together approximately span the design space—in a 1.5 MW wind turbine. Rating life assessment was carried out in accordance with IEC 61400-1 and ISO 281. Various cases were considered, including varying the bearing temperature, wind field characteristics, lubricant viscosity, and contamination levels. Rating lives were found to be sufficiently above the desired 20-year design life for the 230/600 bearing and far above this for the 240/630 bearing, under expected operating conditions. For the 240/630 bearing, this appears to be strongly driven by the loads being below or close to the fatigue load limit a vast majority of the time. Key sensitivities for rating life values were temperature and contamination, both of which were found to have a stronger influence than shear and turbulence. Some level of caution was advised when interpreting the observed lower sensitivities to shear and turbulence, since these results may be due (in part or whole) to the assumption of linear damage accumulation. Overall, the results of this study contribute evidence supporting the position that rating life assessment, as codified in IEC 61400-1 and ISO 281, does not account for the

[##]In such cases, the viscosity in the real world bearing and the value of effective viscosity may differ.

reported rates of main bearing failures in 1 to 3 MW wind turbines. In future work it was recommended a similar analysis be performed for ISO/TS 16281, and that further efforts be undertaken to identify principal root causes of main bearing failures—possibly leading to a new application standard specific to this component. Extrapolation of wind shear results also led to a recommendation that impacts of partial wake impingement on main bearing rating lives be considered in future work.

ACKNOWLEDGMENTS

This work forms part of project AMBERS (Advancing Main-BEaRing Science for wind and tidal turbines). Jarred Kenworthy is funded by the EPSRC (grant no. EP/S023801/1). Edward Hart is funded by a Brunel Fellowship from the Royal Commission for the Exhibition of 1851. This work was also authored in part by the National Renewable Energy Laboratory operated by the Alliance for Sustainable Energy, LLC, for the US Department of Energy (DOE) under contract no. DE-A#36-08GO28308. Funding was provided by the US Department of Energy Office of Energy Efficiency and Renewable Energy Wind Energy Technologies Office. The views expressed in the paper do not necessarily represent the views of the DOE or the US Government. The US Government retains and the publisher, by accepting the paper for publication, acknowledges that the US Government retains a nonexclusive, paid-up, irrevocable, worldwide licence to publish or reproduce the published form of this work, or allow others to do so, for US Government purposes.

CONFLICT OF INTEREST STATEMENT

The authors declare no potential conflict of interests.

PEER REVIEW

The peer review history for this article is available at https://www.webofscience.com/api/gateway/wos/peer-review/10.1002/we.2883.

DATA AVAILABILITY STATEMENT

The authors are happy to make their data and code available. To arrange access, please contact jarred.kenworthy@strath.ac.uk and/or edward.hart@strath.ac.uk.

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How to cite this article: Kenworthy J, Hart E, Stirling J, et al. Wind turbine main bearing rating lives as determined by IEC 61400-1 and ISO 281: A critical review and exploratory case study. *Wind Energy*. 2024;27(2):179-197. doi:10.1002/we.2883