Practical Implementation and Associated Challenges of Integrated Torque Limiter

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Abstract

Evolving of aircraft design towards further electrification requires safe and fault-free operation of all the components. More electric aircraft are increasingly utilizing electro-mechanical actuators (EMA). EMAs are prone to jamming and subsequent failure due to large forces on the shaft. Large forces are generated due to the high reflected inertia of the electric machine rotor. To limit the force acting on the shaft, a torque limiting device is connected to the power train which can separate the rotating mass of the electric machine from the power train.

In this paper, a concept of integration of torque limiter and the electric machine rotor is presented to reduce overall volume and mass. It is connected closely with the rotor, within the motor envelope. A commercially available torque limiter and an electric machine designed for actuator application are used to demonstrate the concept. While essential for safety, the torque limiter adds to the mass and size of the overall EMA. Conventionally, the torque limiter is connected externally to the motor shaft. Key performance requirements of the machine and torque limiter are provided. Structural analysis of the proposed integrated system is carried out to show the viability. Considering the high-speed operation of the motor, rotor dynamic is analyzed to ensure resonant modes are not encountered within operating speed. Mechanical design of the system, considering assembly, is presented. Integration is shown to reduce the overall mass and size of the system compared with a conventional system, as well as better dynamic behavior and higher bearing life.

Introduction

Conventionally in a typical civilian commercial aircraft a gas turbine engine provides the propulsion power. Multiple power sources such as electrical, pneumatic, hydraulic, and mechanical power are derived from the engine [1]. The electrical source powers avionics, lighting, and other electrical loads. The pneumatic power, obtained from engine bleed air, caters to wing de-icing and cabin air-conditioning. The hydraulic power operates the many different actuators within the aircraft [2].

Replacing the many different sources of power with electrical power provides certain advantages such as higher efficiency, lower emissions, and lower fuel consumption. Engine without any bleed air system has been shown to have better efficiency [3]. Removal of hydraulic system can lead to easier maintenance as well as better integration of actuator within the airframe [4]. Electric power also offers easier control, reconfiguration, continuous monitoring and diagnostics. These advantages are leading to development of more electric aircraft (MEA) and all electric aircraft (AEA) designs [5], [6].

The different actuators in MEA now must operate using electrical power source, leading to development of electro-mechanical actuators (EMA). These actuators position the flight control surfaces, landing gear, different nozzles and guide vanes [7]. Accurate and reliable positioning of control surfaces are critical for safe operation of aircraft. Reliable and fault-free operation of EMA is therefore crucial for success of MEA concept. On the other hand, it is also desirable to have compact and low mass of the EMA.

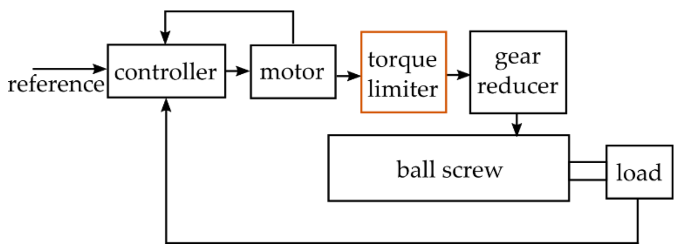


Figure 1. Block diagram of an EMA showing controller, motor, torque limiter, gearbox, and ball screw.

In a typical EMA, a power electronic converter drives an electric motor. The electric motor then connects to the different control surfaces through reduction mechanism. Block diagram of a typical EMA configuration is shown in Figure 1. The power electronic converter receives feedback from the motor and the load and applies pulse-width-modulated voltage pulses to the motor. The torque of the motor is increased by gears and converted to linear motion by ball screw mechanism. The reduction mechanism lowers the mass and torque requirement of the electric machine. At the same time, the rotating inertia of the motor is multiplied by the reduction mechanism.

One of the major causes of failure of EMA is jamming of ball screw [4], [8], [9]. This will cause sudden deceleration of the EMA drive shaft. Any block or end stop on the load will also cause fast deceleration. The large reflected inertia of the electric motor rotor leads to large forces on the shaft and subsequent damage to EMA. A torque limiter is thus introduced within the shaft, as shown highlighted in the same figure. If the torque transmitted through it exceeds a certain set-point, the input and output shaft of the torque limiter disengage and can rotate independently. The torque limiter prevent any damage to the electric machine because of any locking in the rest of the chain.

While torque limiter is essential for safety, it also increases the mass and volume of the EMA. Size of the EMA can be reduced through integration of the torque limiter within the electric motor. Power density improvement through integration of power electronics [10], passive components [11] and impeller [12] within the motor are explored in recent works. Integration of torque limiter has been explored earlier in [13], [14]. However, detailed mechanical design and dynamics are not presented.

In this paper detailed mechanical design and analysis of integrated torque limiter is presented. Different possible configurations for integrating frictional torque limiter into the machine are explored and advantages and disadvantages have been studied. Dynamic and vibrational behavior of the feasible options have been investigated and finally compared with the common solution dynamics behavior.

Machine Design

Base electrical machine

For the integration, an electric motor designed for control surface actuation is considered. It is a permanent magnet surface mount (PMSM) type electric motor. Few important ratings of the machine are presented in Table 1. The machine torque is 3.4 Nm. Considering some margin, the torque limiter must be capable of transmitting at least 5 Nm.

Table 1. Details of the electric motor

Power	5 kW
Speed	14 000 rev/min
Operating Duty	30 second

Torque Limiter Integration Concept

As mentioned, torque limiter is a safety device that protects the drive and driven components from damage in the event of an overload, crash, or jam. There are different kinds of torque limiters with different abilities [13]. The first type is ball detent torque limiters that generally come in 3 main types ratcheting, synchronous and overload. In the ratchetting type – when the torque value reaches the set limit, the torque limiter slips and ratchet round until the toque overload is removed. In this type, when the overload is gone it is automatically reset and ready to drive again and during tripping it transmits approximately 5% to 15% of their trip torque. The synchronous one works in the same way as the ratchetting limiter described above but when they trip, they only re-engage every 360 degrees so the timing between input and output is maintained. And the last one, the overload type, has the same mechanism but when the torque limiter trips the driven and driving side of the limiter is completely disconnected and allows one side to spin freely against the other. Also in the overload type, when the torque overload is removed, the limiter needs to be re-engaged manually or by reversing the drive side.

Another mean of limiting torque is electromagnetic clutch or coupler. A magnetic field couples the input and output shaft. This type of torque limiter is generally large in size, it also adds to system mass due to the electronics and control needed.

Another type is frictional torque limiter. In this category the torque is transferred by friction force between the frictional pad and the output hub. The amount of torque limit is adjusted by locking nut that pushes the disk like spring behind the frictional pads. In case of exceeding the limit, the mechanism doesn't allow to transfer more torque through the fictional pad. The main difference between the frictional category and the ball detent one is that they can hold the torque on the drive line by almost 10% deviation, although the drive side slips relative to the driven side. Also, friction limiters are cheaper and more compact that make it a good option for considering integrating electric drives for EMA application. A sectional view of the torque limiting device is illustrated in Figure 2.

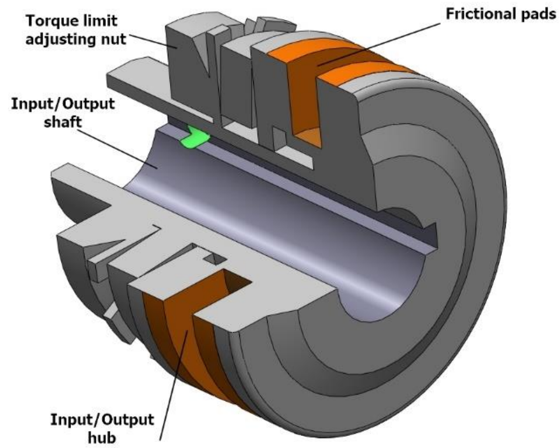


Figure 2. Cut away view of a friction torque limiter

The integration of torque limiters within the electric motor should not affect the motor performance. Therefore, modification of the active part of the machine should be avoided. Further, the modifications should not introduce new resonant modes within the machine operating speed. Assembly, design complexity and manufacturing constraints are other aspects which need to be addressed for successful integration. Some of these challenges and limitations are discussed in this paper.

The common way for having a torque limiter in drive line is to put it in the middle of the drive train somewhere between the drive motor and the load as shown in Figure 3. In this arrangement the toque limiter is between two flexible couplings that add considerable mass and inertia. It also increases the rotating chain length which lead to more mechanical vibrations and stress. Any solution with lower length and mass help to reduce loads on the bearing and consequently less vibrations and stress.

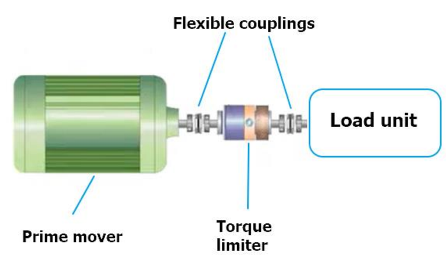
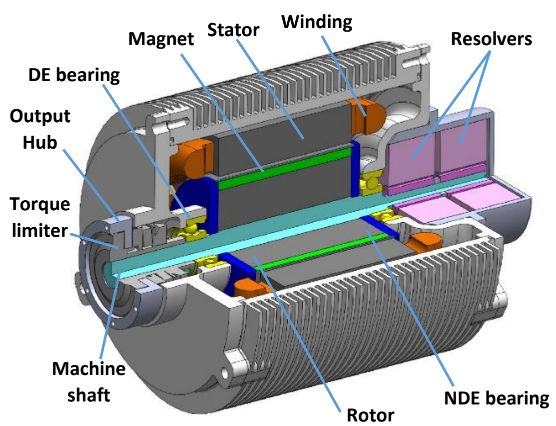
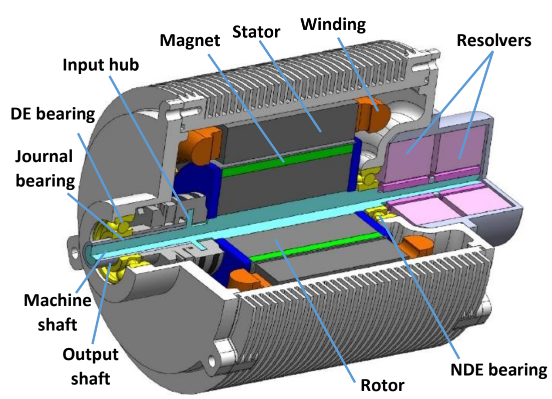


Figure 3. Common configuration for using torque limiter in the drive chain.

As mentioned, a 5-kW electric drive has been considered as the base machine and feasible options for integrating the torque limiter into the drive have been investigated. Figure 4a and 4b shows two practical options for integrating torque limiter into the machine. In the first arrangement, Figure 4a, the active parts are directly connected to the drive shaft for transferring torque and speed directly to the shaft. The torque limiter has been placed after the drive-end bearing that transfers the torque to the output hub through the friction pads. In case of exceeding the torque limit, the shaft and torque limiter slip relative to the output hub whilst it still transfers the torque below limit level. In this configuration the torque limiter is closer to the bearing rather than the common solution. It eliminates the need for a flexible coupling between the torque limiter and machine, just one flexible coupling would be needed at the end of the machine to connect it to the load. Also, the output is a hub instead of shaft that because of its greater diameter respect to the shaft leads to a stiffer rotating chain. Additionally, from maintenance point of view the torque limiter parts, especially the frictional pads that could experience wear, are easily accessible for repair or replacement. On the other hand, torque limiter and flexible couplings at the drive-end and the resolver at the non-driven side behave like overhung masses that their load and vibration are mostly transferred to the machine bearings.



(a) Option 1



(b) Option 2

Figure 4. Overall cutaway view of the possible options for integrating torque limiter in the machine.

Figure 4b shows the second option for integrating the torque limiter into the machine. In this arrangement the torque limiter has been placed between the bearings, at the back of the drive-end bearing. It is connected to the machine shaft through a hub internally. Indeed, in this option the torque limiter is placed in a reverse position that hub side of the torque limiter has been used as the input and the shaft side as the output. So, the torque and speed are transferred to the hub that is shrink-fitted to the machine shaft, torque limiter, and then to the output shaft. In the ideal situation that the machine torque is less than the limit the torque is transferred from the hub to the limiter and then to the output shaft. There is a journal bearing between the machine shaft and output shaft that transfers the rotating parts weight to the output shaft and then to the drive-end bearing. On the other hand, when torque exceeds the limit, the journal bearing lets the machine and output shafts rotate independently with a relative speed. Like the first option, this configuration is more compact with respect to the common solution, and most of the inertia and mass are placed in bearings span that can give better dynamics behavior. But as a journal bearing has been considered in the machine, an oil feed and sealing need to be considered, which could be challenging depending on the end application. For low-speed machines and when the machines have a control system that can detect disengagement the journal bearing can work with grease for a brief period, too.

As discussed earlier the first option is an easier option in terms of assembly and maintenance but the second option has a better mass inertia distribution on the bearings. In the next section their dynamic performances have been studied.

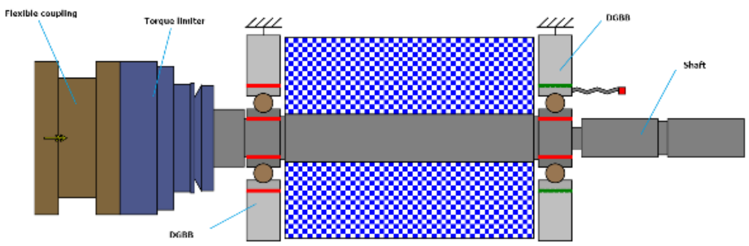
Discussion

From the mechanical point of view, the dominant loads on these machines are the torque and rotating parts mass and inertia. Regarding torque the machine shafts are in safe area and far from failure, but as suggested option has different mass and inertia distribution especially respect to the common solution, in this section the vibrational behavior and bearings performance for the proposed options are studied and compared with the common solution.

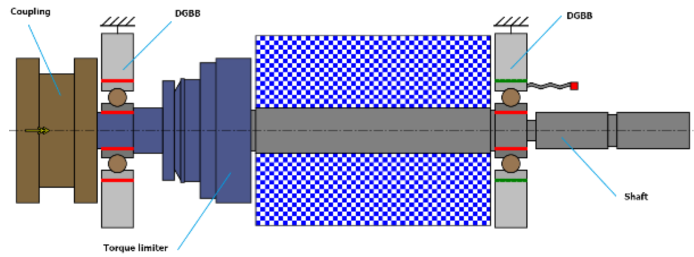
In order to analyze the vibrational behavior and bearings' performance, a finite element model has been developed for two options and the common solution. Figure 5a-c shows an overall view of the developed models for all three cases. A Timoshenko beam element model has been used for shaft modeling that includes shear deformation and rotary inertia. Gyroscopic effect has been considered in finite element formulation. Each element has two nodes, and each node has two translational and two rotational degrees of freedom. Proper steel material has been considered for shaft, limiter and coupling. The laminated iron, magnets and sleeve are replaced by a mass-inertia element with assigned material properties.

Deep groove ball bearings are replaced by an equivalent five-by-five stiffness matrix that covers radial, axial and tilting stiffnesses. Also, bearing performance is analyzed based on race control hypothesis to extract the stress in bearing, bearing life and dynamic parameters of the bearing. For flexible coupling in addition to the mass and inertia their equivalent stiffness has been considered, too.

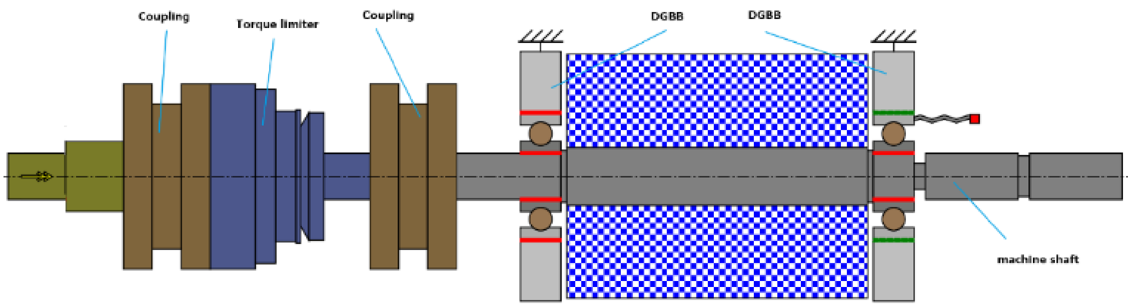
For the common solution that is shown in Figure 2, the drive machine is connected to the load through two couplings with a torque limiter in between. So as shown in Figure 5 the same configuration is considered in its finite element model and at the end of the rotating chain.



(a) Option 1 including coupling



(b) Option 2 including coupling



(c) Common method of integrating torque limiter in drive chain

Figure 5. Finite element model of possible options for integration of torque limiter in drive chain.

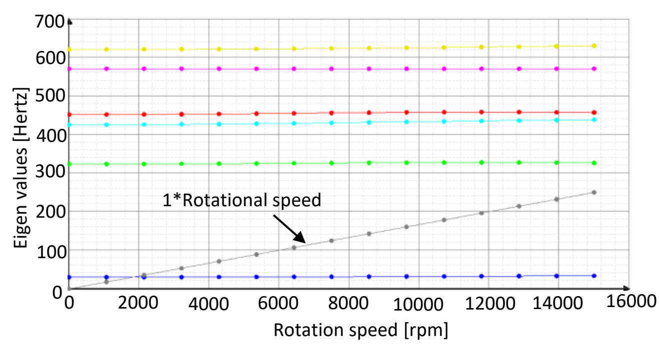
The derived mathematical model for all these options is in the general form of:

$$[M]\ddot{q} + [C]\dot{q} + \Omega[G]\dot{q} + [K]q = 0 \qquad (1)$$

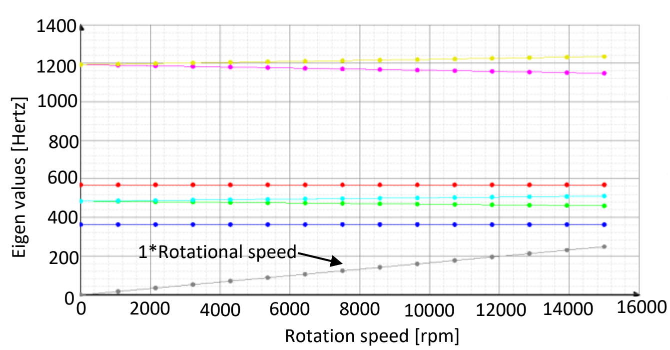
Where [M] is the mass and inertia matrix, [C] is damping matrix, [G] is gyroscopic effect matrix, [K] is the total stiffness matrix of the model, **Ω** is speed and q is the vector of degrees of freedom. To solve this set of equations it can be rewritten in state-space formation and then is solved as an eigenvalue problem.

Vibrational analysis

The eigenvalues of eq. 1 are natural frequencies of the system that are speed dependent. So, for each rotational speed there is a corresponding natural frequency. If this frequency becomes equal to the rotational speed of the machine, the system will experience a high level of vibration called critical speed. The graph that shows the variation of machine frequencies against speed is called Campbell diagram. This is plotted for all these three cases in Figure 6 to Figure 8. Graphs are plotted for bearings without preload and moderately preloaded at 55 N. The grey line is a speed line that intersections with any of the frequency lines is critical speed value. The same amount of preload is considered for all cases to achieve a better comparison. The no-preload cases are investigated as the bearings are deep groove ball bearings and applying preload is optional and depends on required performance.



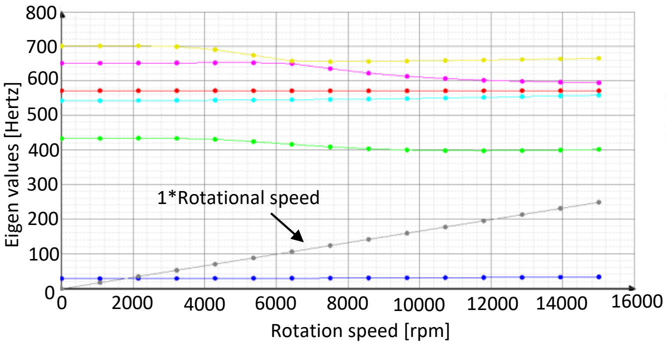
(a) without preload



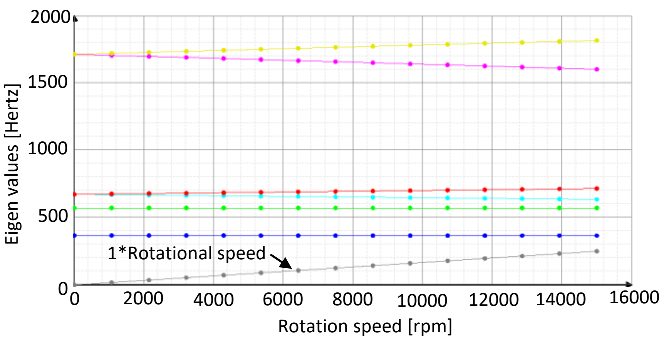
(b) 55 N preload

Figure 6. Campbell diagram for option 1 of integrating torque limiter into the rotating chain with and without preload.

Figure 6 and Figure 7 shows that without preload both options meet one critical speed. While for the common solution in Figure 8, it increases to three critical speeds which two of them are very close to the rated speed of the machine. By applying a moderate value of preload to the bearings the frequency lines shift up for the proposed options, and the machine works in sub-critical state. But for common solution still the machine should work closely to the critical speeds. It should be mentioned that increasing the preload in common cases can push the frequency lines far from the working speed, but the amount of preload increase is limited by the bearing performance parameters that are discussed in the next section.

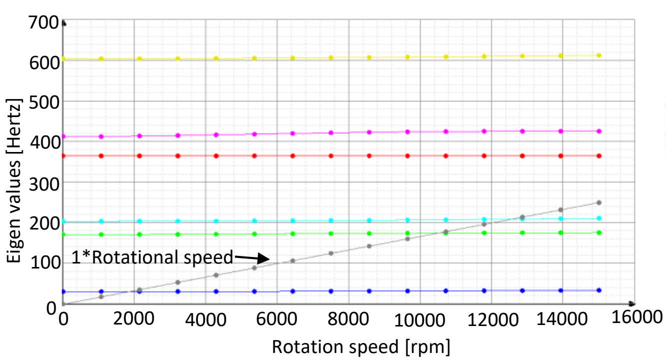


(a) without preload

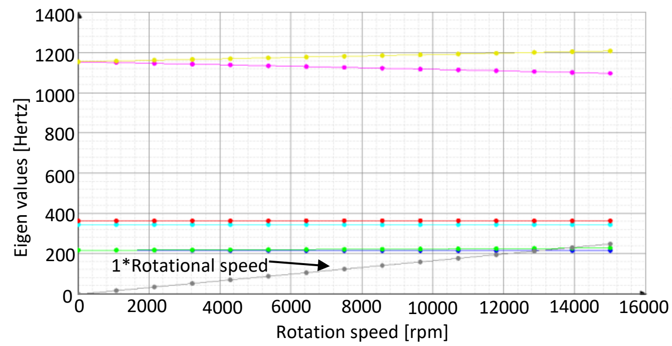


(b) with 55 N preload

Figure 7. Campbell diagram for option 2 of integrating torque limiter into the rotating chain with and without preload.



(a) without preload



(b) with 55 N preload

Figure 8. Campbell diagram for common solution of connecting torque limiter into the rotating chain with and without preload

It should be mentioned here that another way to make the common solution viable is to consider another bearing at the end of the rotating parts. But one objective is to have a system with the same number of parts especially for the number of bearings. Adding another bearing to the chain create more difficulty in terms of assembly and maintenance.

Bearing performances

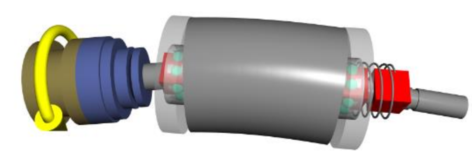
In addition to the vibrational behavior of the machine, the bearings performances need to be studied and compared to the common solution. Table 2 compares the important performance parameters of the bearings in preloaded condition for all three cases. Table 2 shows that options one and two have better life with respect to the common solution. Although the NDE bearing of the common solution has higher life, it is a sign of unequal load distribution between the machine bearing. Unequal load distribution is undesirable as usually the designers prefer to have bearings with similar lifetimes that lead to less maintenance times. Also, between the proposed options, option 2 has similar lifetimes for the two bearings. Maximum pressure at the inner and outer race of the bearing is another sign of even load distribution in common solution and better load distribution in option 1 and 2.

As the bearings don’t experience any radial and axial loads the spin to roll ratio is the same. The bearings misalignment and shaft tip deflection in common solution is greater than the other options, indicating uneven load distribution between the bearings. In option 2 the maximum deflection is in the middle of the bearing span whilst in option 1 and the common solution it is at the tip of drive-end of the rotating part.

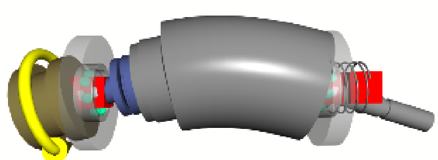
The first mode shape of these cases that are shown in Figure 9. In case vibration excitation like unbalance force, the maximum displacement of the common solution is at the drive-end of the rotating chain while for the proposed options it is in the middle of the machine and between the bearings. So, the critical area of the machine from deflection and displacement point of view for option 1 and 2 is in the middle of the machine and for the common solution is in the drive-end side.

Table 2. Comparison of bearings performance in different options of integrating torque limiter into the rotating chain for preloaded condition.

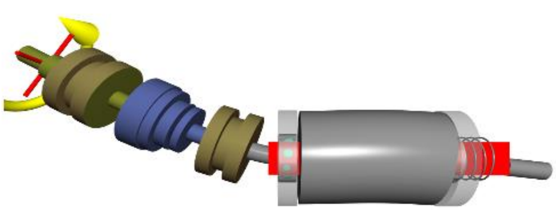
No	Parameter	Option 1		Option 2		Common solution	
		DE bearing	NDE bearing	DE bearing	NDE bearing	DE bearing	NDE bearing
1	Bearing L10 life (h)	73200	78200	73500	73300	70300	80400
2	Max pressure on the inner race of bearing (MPa)	1306	1295	1304	1305	1313	1290
3	Max pressure on outer race of bearing (MPa)	1448	1436	1347	1347	1456	1431
4	Spin to roll ratio	0.05	0.05	0.05	0.05	0.05	0.05
5	Bearing misalignment (min)	0.01	0.03	0.04	0.06	0.1	0.01
6	Shaft tip deflection (µm)	1.1 at DE		0.6 at Middle		8 at DE	



(a) Option 1 1st mode shape



(b) Option 2 1st mode shape



(c) Common solution 1st mode shape

Figure 9. Mode shape comparison between the common solution and proposed integrated torque limiter configuration.

Conclusions

In this paper two suitable options for integrating a torque limiter in the drive train of an electric motor actuator have been studied and compared with the traditional method. Among the available torque limiters in the market, frictional torque limiter has been selected because of its compact and simple architecture. The torque limiter has been integrated into the machine in two different ways. Option 1, in which the torque limiter is seated at the drive end side of the machine beside the bearing, showed better dynamic performances against the traditional architecture. Putting the torque limiter beside the machine bearing eliminates the need to have a flexible coupling before the torque limiter and presents easy assembly and maintenance. For option 2, the torque limiter sits in the bearings span, also offer good dynamic behavior and bearing performance with respect to the existing method. Second proposed solution showed a better performance with respect to the first solution, but it needs an intermediate journal bearing and oil feeding path in the housing. This makes option 2 more complicated, and suitable only for oil cooled machine as it doesn't add any extra requirement for oil auxiliary system. Both proposed options show better dynamics behavior respect to the common solution especially when less preload is applied to the bearings, resulting in better bearings life and load distribution. Also, considering the same base machine is used for analyzing all the solutions, it can be concluded that the proposed options have better vibrational behavior in terms of no critical speed are encountered in the working speed range.

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