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MAGNETIC SPRINGS – FAST ENERGY STORAGE FOR RECIPROCATING INDUSTRIAL DRIVETRAINS

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Abstract. *Industrial machines with reciprocating (oscillating) motion such as weaving looms tackle primarily high inertial loads, conventionally operating within frequency ranges of 5-15 Hz with relatively large strokes. Recent trends of individual electrification of parts of weaving loom drivetrains for reasons of increased flexibility of use make this problem even worse, as the inertial loads are less averaged out. Adding springs to such oscillating drivetrains can allow to improve the energy efficiency and downsize the actuators. To get an estimation of energy sinks and peak power consumption in a reciprocating drivetrain of a weaving loom, a spring assisted demonstrator available at Flanders Make has been modelled using a 1D multiphysical dynamic model. Next to energy requirements, industrial machines have strict lifetime demands. Target lifetime of 50 000 hours results in over 1E9 spring cycles. Mechanical spring design and fatigue modelling for this number of cycles is a difficult design problem with high levels of uncertainty. Therefore, magnetic springs are proposed instead of mechanical springs as a technological novelty with benefits of no material fatigue and additional flexibility in design. In the developed drivetrain model the mechanical spring is replaced by an off-the-shelf magnetic spring in order to perform a first estimation of the impact on the dynamic behavior.*

Keywords: *magnetic spring, spring assisted drivetrain , model based design, energy efficient oscillating drivetrain*

1. INTRODUCTION

At the time when the shortage of fossil fuels and global warming are on the horizon, and the energy demand keeps on expanding with the global industrial growth, the total cost of ownership for machines and cars is getting increasingly dominated by the costs of energy consumption (EU Parliament 2012). In order to increase energy efficiency, hybridization and electrification of traditionally mechanical drivetrains is becoming more popular. This step has been enabled by improvements in semiconductor, electrical storage and electrical drive technologies. The recent innovations in weaving loom market niche of machine building industry expose the same trends (Toyota 2005, Picanol 2008).

Specifically in weaving looms, increased flexibility of operation identified with electrification can improve weaving loom versatility and allow to outperform a variety of machines (Sabit 2001). With electrification, the number of rotating parts needed for conversion of mechanical energy from one movement to the other can be reduced. Energy saving potential of such reduction can be observed from work done by Croes et al. (2012), where bearing losses in complex mechanisms govern the overall losses. On the other hand, reducing machine complexity leads to less averaged reactive loads, thus demanding high power requirements for electrical drives. When a direct drive is used to accelerate and decelerate the load, the kinetic energy of the oscillating inertia can be repeatedly transferred to and from electrical energy storage at the DC-link. However, during every stated conversion the system generates losses in the inverter and the electric motor. Moreover, the motor is dimensioned according to the peak power and is operating in low efficiency conditions for the majority of the operating cycle.

By using a spring assisted drivetrain the kinetic energy of the load is converted into potential energy of the elastic component as in a simple mechanical harmonic oscillator. This way the electrical drive (both motor and power electronics) can be downsized since power requirements are averaged by the spring, while the motor only needs to compensate for the damping in the mechanical oscillator. Therefore, the motor has to be dimensioned according to the energy losses in the drivetrain and not the peak power. On Fig. 1 a comparison of power required by direct drive and spring assisted motor is shown, based on the existing spring assisted demonstrator.

This concept is inspired by natural muscle operation (Alexander 1990) and already applied in compliant robotics (Pratt and Williamson 1995, Mettin et al. 2010), but so far exists only as a prototype in other resonant machines (VVT 2015, Poltschak 2014). As perceived by authors, its full potential has not been exploited in industry due to the robustness issues that are twofold:

- Mechanical robustness – adding mechanical components to the drivetrain reduces the mechanical robustness of the system
- Control robustness – since the drivetrain has a passive characteristic it is more difficult to keep the tracking error within a set error margin

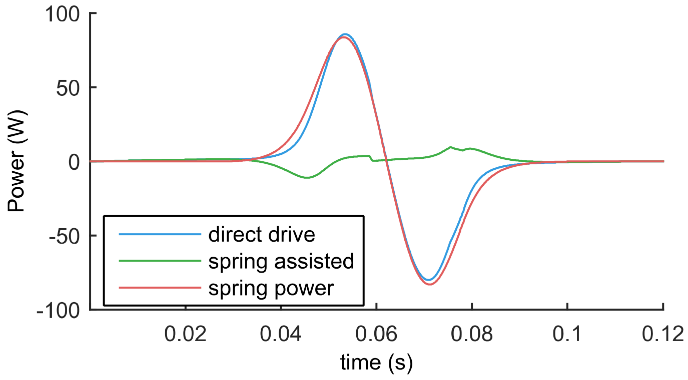

Figure 1. Comparison of mechanical power required by the electric motor for a spring assisted drivetrain and direct drive

Mechanical robustness of the spring assisted system is the first and critical operational condition. Due to the material fatigue, the elastic spring is the most critical component. A possible solution is to use accurate fatigue models for intended lifecycles during the design optimization. Another interesting solution for the material fatigue issue is using magnetic springs as a technological enabler of gigacycle operation (and possibly beyond).

In this paper we present a design exercise on an existing spring assisted demonstrator of a weaving loom drivetrain. The goal of the design exercise is to pinpoint issues and advantages of both approaches mentioned. This exercise consists of creating a simplistic 1D multiphysical dynamic model of the demonstrator in order to identify power flows in the machine (Section 2.2). In parallel, fatigue of the mechanical spring is analyzed using measurement data from related publications and Murakami fatigue models for crack initiation by nucleation due to the inclusions (Section 2.3). For an initial estimation , the mechanical spring is replaced by an off-the-shelf magnetic spring (Section 3), first in the lumped parameters computational model (Section 4.1), and afterwards validated with measurement using the experimental setup (Section 4.2).

Robustness issue should be a part of the multidomain design problem at hand, in order to find a global optimum for the problem of spring assisted energy efficient reciprocating machinery.

2. EXISTING DEMONSTRATOR MODELLING

2.1 Demonstrator overview

The demonstrator at the disposal of Flanders Make is an electrical leno selvedge (US English: selvage) drivetrain (Fig. 2) assisted by a torsional spring in parallel with the load. A conjugate cam-follower is used as a locking mechanism. Therefore, in the end positions no braking torque is required to hold the spring loaded. This is an important operational requirement, since the peak torque of the motor is *1.03 Nm* while the spring exerts a torque of *8.29 Nm* when the load is in the end position.

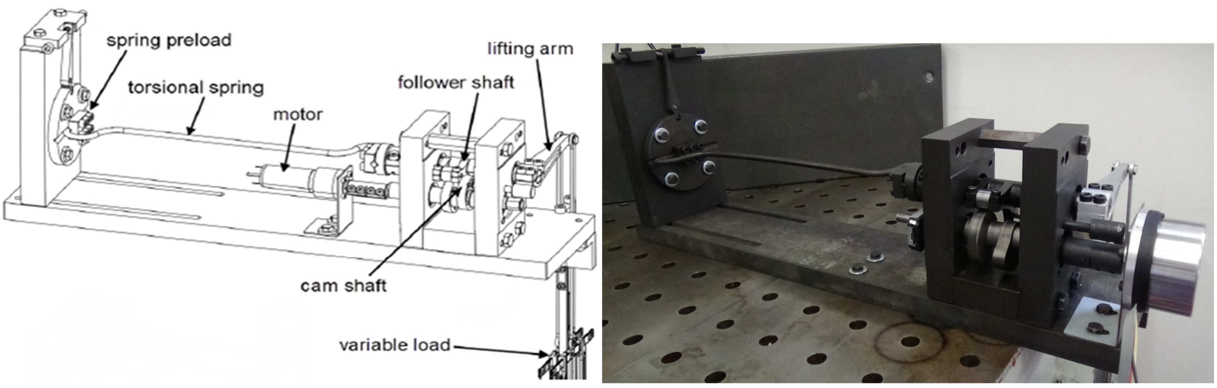

Figure 2. Sketch with labeled components and a photo of the of the spring assisted demonstrator

As compared to the principal drivetrains of a weaving loom, the load case is an order of magnitude smaller, but the stroke and trajectory are comparable to the principal motion of the weaving loom (i.e. shedding). With respect to lifetime constraints, we will consider an operational lifetime of *50 000 hours* as a minimum possible cutoff for design, which results in gigacycles in target frequency range of *5-15 Hz*.

2.2 1D multiphysical dynamic model in Simscape

Figure 3 shows a lumped parameter model created in Simscape using partly the existing CAD data, and identification of friction parameters with experimental data. Namely, friction on cam-shaft, follower-shaft and load slider were modelled each by a Coulomb and viscous friction block. The parameters were identified using non-linear least squares with different open-loop torque input profiles.

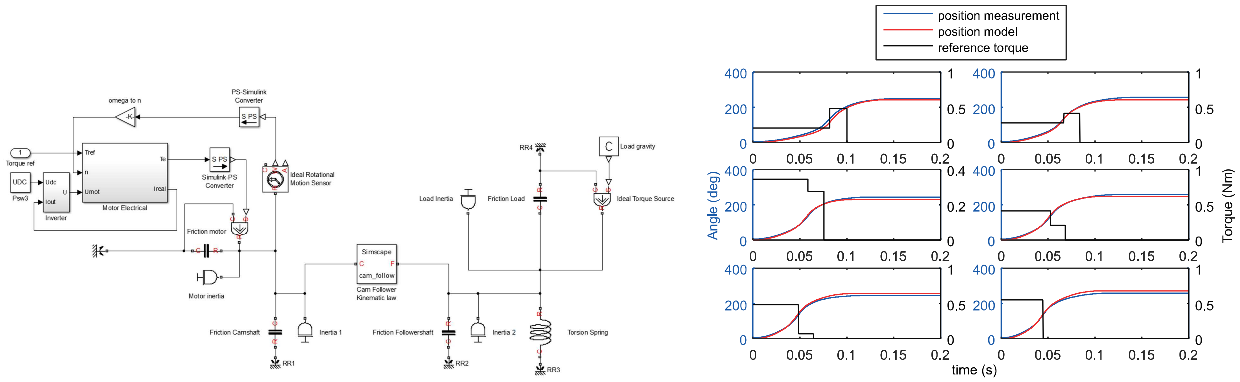

Figure 3. Simscape model of the spring assisted demonstrator and identification of friction parameters

Steady-state electrical model is used as the dynamics of the electrical circuit are significantly faster than the mechanical dynamics. The rolling losses of the cam-follower were not identifiable with the current model structure and setup at hand, and were therefore fitted with the before mentioned friction blocks.

Thus, the model is biased and is not valid out of the current operational points and does not provide a clear idea of the system's mechanical losses distribution. It can however be a good tool for better understanding of the dynamics of the system and give an overview of energy flows. On Fig. 5 it can be seen that the reactive load is dominant, and that the energy oscillates between potential energy of the spring and the kinetic energy of the moving inertia. The energy distribution between electrical and mechanical losses is considered reliable since the electrical model is built using only the physical data of the electrical drive.

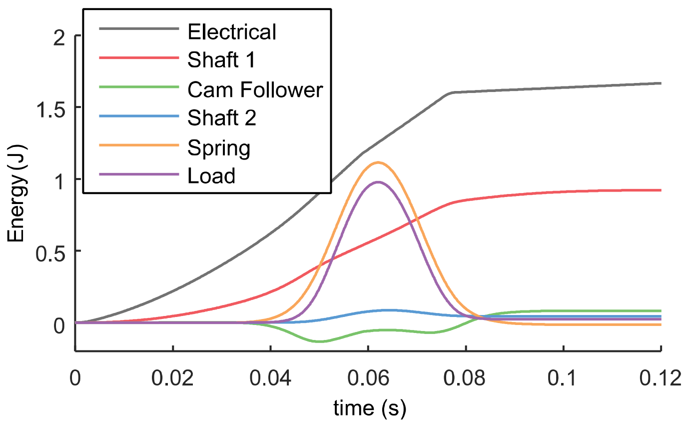

Figure 4. Energy flows based on 1D multiphysical model

2.3 Fatigue analysis

As a design exercise, it is possible to calculate the dimensions of the torsion bar spring for the minimum lifetime of a spring assisted drivetrain. This can be done by parametrizing the stress in torsion bar and finding the stress boundary for the required number of cycles.

The stress and stiffness of a simple torsion bar can be obtained from constitutive equations (Eq. 1 and Eq. 2) if we consider that the torsion bar with circular cross section is in small rotations and pure torsion.

$$\tau_{max} = G\varphi \frac{r}{L} \sim \frac{r}{L} \tag{1}$$

$$K_{spring} = \frac{T}{\varphi} = G\frac{r^4 \pi}{L} \sim \frac{r^4}{L} \tag{2}$$

The spring material is oil-tempered super clean Cr-Si high strength steel, under commercial name Oteva 70 Sc. This material is commonly used for high-demanding motor valve springs. On Fig. 6 different modelling approaches are shown classical S-N curve with fatigue limit and *3s* boundaries obtained from material manufacturer data (Oteva 2015).

However, it has been shown repeatedly in recent material studies (Bathias and Paris, 2005), that the true fatigue limit does not hold in gigacycle regime. In low cycle regime fracture originates in surface cracks and propagates towards the center of the structure. On the other hand, in gigacycle regime the unavoidable inclusions in steel can cause micro cracks within the specimen and fish-eye fracture can thus occur. Although the manufacturer data for super clean steel states that there should not be inclusions larger than *25* μm, it is generally accepted that they will still occur. In experimental work dealing with the exact valve spring material observed hereby (Puff and Barbieri 2014), SiO_2 inclusions of *144* μm have been reported as a cause of nucleation cracks. This dimension is used for Murakami and improved Murakami model (as described by Bathias and Paris 2005). Empirical data in form of a logarithmic fit of measurement recorded by Serbino and Tschiptschin (2013) is also shown. As expected, the improved Murakami model shows the best fit, although the exact knowledge of non-metallic inclusions and its dimensions is not straightforward to obtain.

From Fig. 6 we can conclude that the current design is over-engineered. At the moment the spring length of *310 mm* is quite large compared to the size of the drivetrain, but it could probably be downsized to *200 mm* (though this is still considerably large compared to the remainder of the machine).

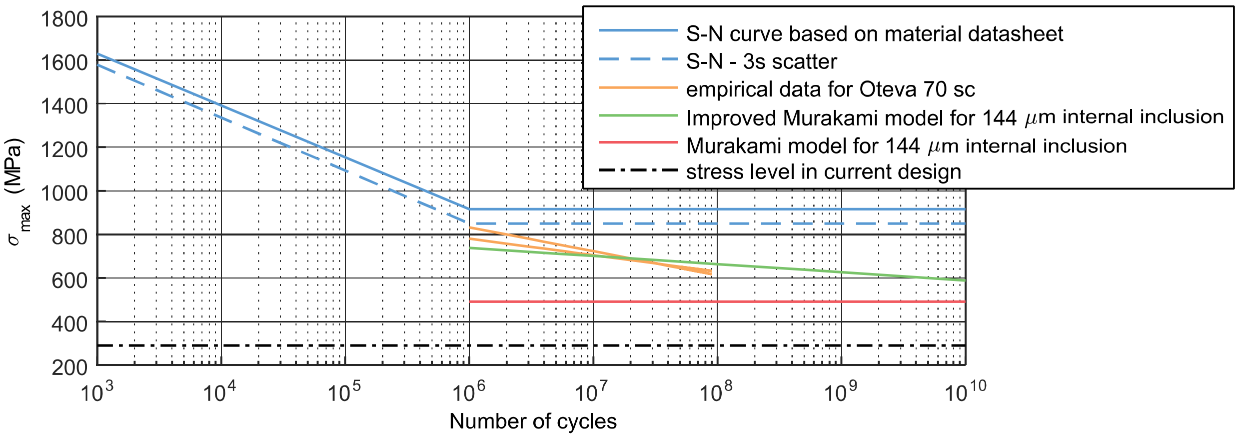

Figure 5. Maximum stress limits calculated with various fatigue models and experimental data for EN10270-SiCrVD (Oteva 70 sc)

3. MAGNETIC SPRING

3.1 Concept of Magnetic Spring

Magnetic spring is a component with elastic characteristic (i.e. resembling the characteristic of an elastic spring) that results from the attractive reluctance force between a permanent magnet and soft magnetic material or attractive/repulsive force of two permanent magnets.

The concept resembles an electric motor where controllable electromagnetic flux is replaced by PM flux. The result is a permanent force or torque characteristic. Therefore, the moving inertia/mass of magnetic spring should generally be lower compared to the electric motor as PM flux density is significantly higher than the flux of electromagnets in electric motors. Depending on the magnetic topology different characteristics are obtainable, ranging from multi-equilibrium springs, quasi-linear springs to constant forces. Like electric motors, the magnetic springs can be both linear and rotational (torsional). The same physical principles can be observed at different scale in other electromagnetic devices such as electric motors as cogging torque (due to reluctance forces), magnetic couplings (smaller stroke and higher stiffness), or passive magnetic bearings.

Since the origin of force/torque is not the elastic deformation of material, fatigue does not occur in magnetic springs. Therefore, the magnetic springs can be thought off as having virtually infinite lifetime. This assumptions only holds for properly designed magnetic springs, with no overheating or mechanical damage due to high impact forces.

3.2 Off-the-shelf Magnetic Spring in the Existing Setup

For a quick first estimate of magnetic spring dynamic behavior, the only commercially available magnetic spring (to the knowledge of authors) has been used (MagSpring 2015). The selected component is intended for compensation of static gravitational load in industrial automation applications. As such, the spring is designed to have a constant force characteristic with displacement for a specified stroke (*10 mm* to *50 mm* or *-10 mm* to *-40 mm* on Fig. 6). To improve model accuracy, the static characteristic of the magnetic spring is measured using a linear positioning system and a strain gauge. According to the measurement, the force roughly fits into the *±10%* characteristic provided by the manufacturer. Overall, the spring does have a stable equilibrium point and adequate potential energy storage (although with a softening characteristic) which is deemed a sufficient condition for use with the oscillating dynamic setup at hand.

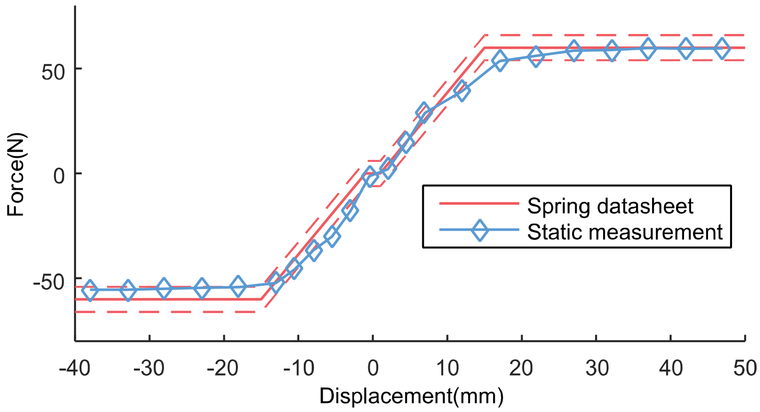

Figure 6. Characteristic of a commercially available magnetic spring

Unlike the spring used in the setup, the described magnetic spring is a translational and not a torsional spring. Therefore, the existing lever mechanism attached to the load has to be updated with a rotational joint as shown on Fig. 7 in order to connect the translational spring to it. The stator of the magnetic spring is attached to yet another rotational joint in order to minimize the radial loading of the linear bearing of the magnetic spring.

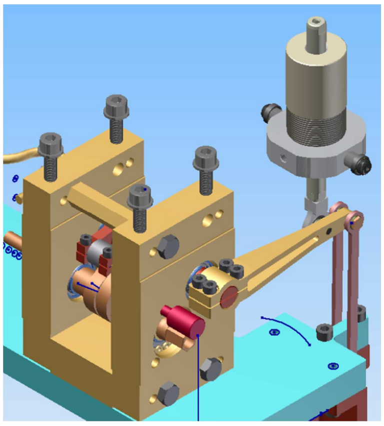

Figure 7. CAD Design of spring assisted demonstrator with linear magnetic spring

The magnetic spring selection is limited by discrete components given in Tab. 1. However, continuous design tuning is possible by changing the lever length. Only the highest spring stiffness per mass of translator from Tab 1. are considered (Fig 8). since they offer the best dynamics. An equivalent inertia of the existing machine has been calculated for a linearized approximation as a spring-mass oscillator. For the specific magnetic spring attached to the system, inertia and torsional stiffness can be tuned (with lever length) for the desired natural oscillation frequency. The final design can be tuned to mimic the dynamics of the original setup.

Table 1. Commercially available magnetic spring selection

Stroke (mm)	Forces (N)	Translator mass(g)	Dimension(mm)
50	40 50 60	75	37x37x80
50	11 17 22	75	20x20x80
125	40 50 60	155	37x37x155
130	11 17 22	155	20x20x155
200	40 50 60	220	37x37x230
210	11 17 22	220	20x20x230

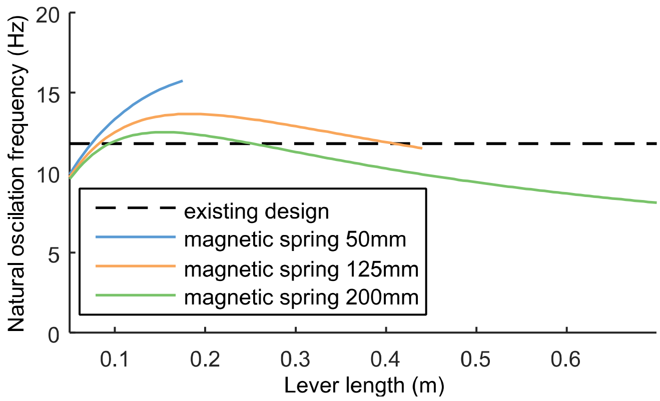

Figure 8. Spring selection and tuning of natural oscillation frequency

4. DYNAMIC BEHAVIOUR COMPARISON

4.1 Model based approach

The proposed magnetic spring is introduced into the existing 1D model of spring assisted drivetrain as an idealized mass-stiffness since no friction model is available for it. Although the spring characteristic differs significantly from the linear mechanical spring, the dynamic trajectories show very small differences (Fig. 8). Instantaneous power stored in the spring is different due to the change in spring characteristic but the end-effector dynamic response is very similar to the original setup (less than 2 % error).

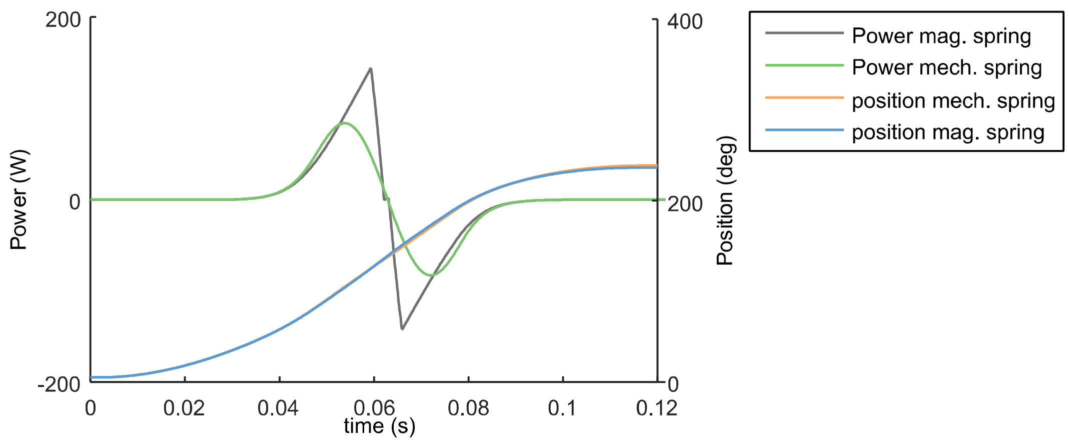

Figure 9. Modelled dynamic behavior of spring assisted demonstrator with mechanical and magnetic spring

Due to the simplistic model nature, explained in Section 2.2, the difference in loading the cam-follower rollers and the follower shaft bearings (due to the softening spring characteristic of the magnetic spring), is not modelled. Also, the losses estimate for the linear bearing of magnetic spring is not included in the model and it remains to be identified with the experimental setup. Nonlinear behavior from magnetic spring could lead to additional losses as the steep power curve might create high frequency excitations of the setup around the spring equilibrium position.

4.2 Experimental validation

Prototype of magnetic spring setup is used to validate the model based design approach. The shown measurement values are averaged after 50 cycles of operational behavior.

In the measurement considered here the magnetic spring setup is run with lever length of *90 mm*, therefore the response is expectedly slower than in the original setup. For different lever lengths faster response should be achieved (as predicted in Section 4.1).

Initial dynamic measurement of the magnetic spring setup is used to re-tune the friction parameters of the model. In Fig. 10 the dynamic response of the two discussed setups is compared. The load position is of higher interest for the machine performance, but in this specific case the motor position gives a better idea of system dynamic behavior and energy consumption. With the same updated friction parameters, response of the magnetic spring show a good fit with the modelled behavior (within *5 %* error boundary) for majority of the operational cycle (*0-0.07 s*). However, there is a mismatch between the model and measurement in the last *0.03 s* when the load has already reached the end position.

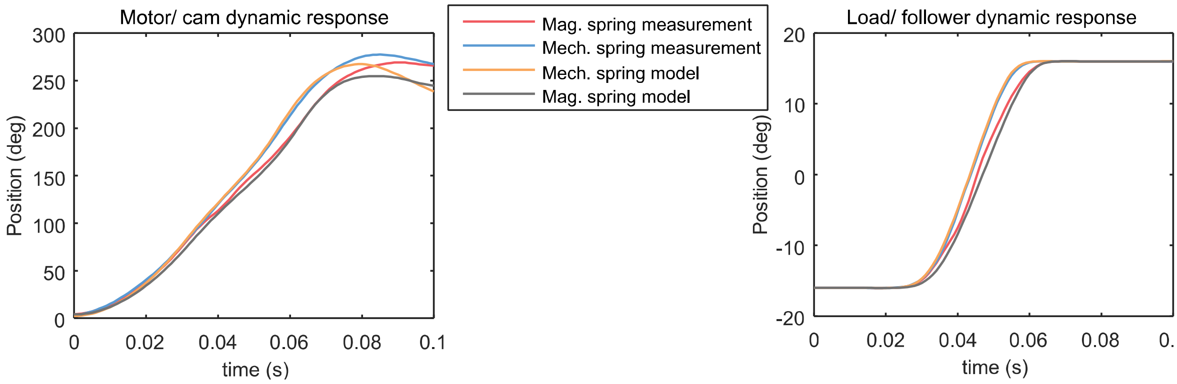

Figure 10. Measured and modelled dynamic response of the spring assisted demonstrator for mechanical and magnetic spring

The same behavior can be observed in power and energy consumption. The model shows a good quantitative fit of the losses within first *0.07 s*, but both qualitative and quantitative fit of last *0.03 s* are inaccurate as seen in Fig 11.

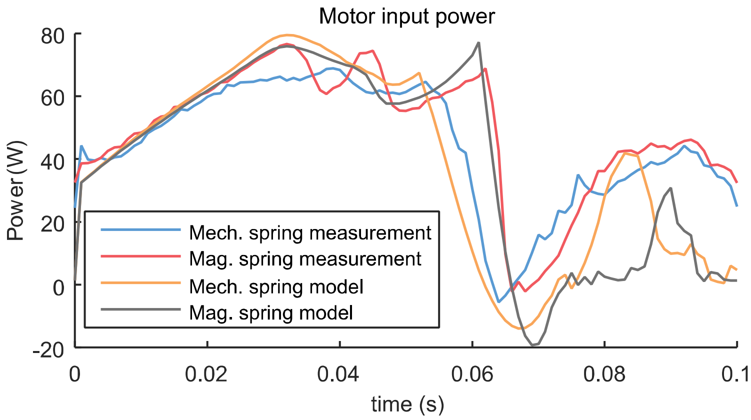

Figure 11. Power consumption measurement and model of the spring assisted demonstrator for mechanical and magnetic spring

The energy consumption is increased for *10.27 %* with magnetic spring due to overall increase in friction. This is probably the result of the relatively high friction in plain sleeve bearing of magnetic spring as compared to material damping of the mechanical spring. Additionally, linear sleeve bearings are not necessarily durable and are hard to manufacture adding up to the cost. Therefore, rotational magnetic spring should be considered in future work as a possible improvement.

5. CONCLUSION

Spring assisting the load is an interesting option for increasing the power density of the reciprocating machinery and reducing the energy consumption by downsizing the electric motor. Material fatigue in elastic spring is a substantial obstacle to this, considering high operational frequencies and long lifetime requirements of the weaving machines.

It has been shown one specific case of reciprocating drivetrain, both experimentally and using a simplistic lumped parameter model that the mechanical spring can be successfully replaced with a more compact and fatigue free magnetic spring in a way that does not significantly diminish system's dynamic performance. The usage of magnetic spring increases the energy consumption compared to the original setup for *10 %*. However, the specific magnetic spring component has not been designed to fit this specific application and in the opinion of the authors there could be additional benefits resulting from custom magnetic spring design (e.g. rotational magnetic spring with optimized geometry), such as lower losses, higher natural frequencies or shapeable non-linear characteristics (hardening or softening springs).

Before continuing the future studies, more accurate model of magnetic springs on component level and the system level dynamics should be developed for better understanding of underlying physical phenomena and better energy estimation. Hereby presented results will be used as a foundation for a scalability study of both explored technologies as a step towards finding the global optimum for the problem of spring assisted energy efficient reciprocating machinery.

6. ACKNOWLEDGEMENTS

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