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Design Development of a Flywheel Energy Storage System for Isolated Pacific Island Communities

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Abstract— **Many Pacific Island Countries (PICs) lack the benefit of electrification in their outer lying regions, particularly on remote islands. Consequently, alternative power sources have been investigated for such situations and are widely reported; for example: wind, wave, tidal and solar generation. In isolation these technologies provide heavily disrupted electrical power, unless backed by a substantial and thereby expense power storage system. More recently, the emergent micro grid concept has facilitated the integration of these discrete power sources and has therefore extended their temporal utility, making them more useful to their host communities. However, with the addition of an energy storage device within the micro grid architecture (typically a battery bank) further enhancements to the around-the-clock accessibility of electrical power can be readily achieved for such system users.**

This paper explores the feasibility of developing a new, low speed Flywheel Energy Storage System (FESS) for use in the pacific region by small and isolated communities. Also, the specific electrical power needs, of the outlying communities, are presented prior to technical solutions being sought. Optimized flywheel performance, electro-mechanical coupling, safety and environmental impact are all addressed in the development, alongside other crucial technical consideration and state-of-the-art technological developments.

The reported work is leading to the practical implementation of a FESS bound micro grid in remote Oceania.

I. Introduction

Many Pacific Island Countries (PICs) lack the benefit of electrification in their outer lying regions, particularly on remote islands. Commonly, the affected communities try to compensate for this shortfall by intermittently using petrol or diesel powered generators, at great expense to themselves and the environment. In contrast, some of the remote communities have more recently adopted renewable energy sources (RES) in conjunction with micro grids to offset their dependence on fossil fuel powered generation. However, the combined RES and micro grid technologies must ultimately be tethered to an energy storage system in order to deliver a continuous or near-uninterrupted power source. In this configuration the RES power generation system provides much enhanced utility over the fossil fuel fired systems.

Of the available energy storage technologies, batteries are the most common in small-scale micro grid and renewable energy installations. However, at the end of their useful life, batteries are an environmental, health and prosperity hazard, particularly in the PICs due to the lack of recycling infrastructure and appropriate waste management. Though alternative power storage technologies exist they are largely unsuitable for PIC based micro grid applications. However, one technology shows potential above all others, the Flywheel Energy Storage System (FESS). A FESS offers such benefits as long life, low maintenance, high power density and the use of environmentally low-impact material in their construction, which makes them more suitable in Pacific islands than batteries [1].

This project focuses on the power storage needs of a small, Pacific island village with 10 households and an average occupancy of 5 persons per household (typically). The target village situation also has one community building and consequently needs a power storage system with a reserve, storage capacity of 25kW. Commonly, batteries would be used for energy storage in such situations. However, a major point of concern for battery utilization in the Pacific region is the environmental impact, as products transported to Pacific islands mostly never leave the islands. A sustainable and more environmentally friendly alternative is explored in the following sections of this paper [1].

When remote villages gain access to a continuous supply of electricity the inhabitants initially do not draw heavily upon the supply and generally only need power for a few light sources. However, as the villagers get used to the availability and reliability of electrical power their energy usage naturally increases gradually. The United Nations have described that Pacific rural households have an entry energy usage rate of 0.2 kWh/day and an established rate of 12 kWh/day [2]. From data of several rural area projects [3], [4], it can be concluded that the power demand of a typical small village could be around 10 kW as can be seen in Table I.

The amount of energy which needs to be stored depends not only on the demand, but also on production. In remote areas the energy production is increasingly being achieved sustainably by Renewable Energy Sources, like solar, wind or marine energy. However RES energy production is generally not constant. For example, the performance of a solar panel is dependent upon the intensity of the sun's radiation on the panel and upon the daylight hours per day. Within a Pacific context, the Weather Centre in Nadi, Fiji, has determined that the difference in daylight hours per day from winter to summer is only two hours. In this paper it is desired that the energy capacity should provide a back-up for several hours without the availability of any RES. In the case of a solar panel based

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energy production system, energy is always produced during day-light hours even during heavily overcast conditions. As a consequence the energy reserve within the FESS will actually be less than 8 hours. Since many villages have a diesel generator the FESS can be used to decrease the fuel consumption.

TABLE I. A TYPICAL VILLAGE POWER DEMAND SPECIFICATIONS

	Small village
Population	10 households
Average energy use	12 kWh/household/day
Total energy use	125 kWh/day
Delivery time	2 h/household
Village Load [kW]	10 kW

This paper explores the feasibility of developing a new, low speed FESS for use in the Pacific region for small and isolated communities. The development addresses some unique design, construction, operational and control considerations - set in the context of local conditions. Optimized flywheel performance, electro-mechanical conversion, longevity, safety and environmental impact are addressed in the development, alongside other crucial technical considerations.

II. FLYWHEEL DESIGN

The essential components of a simple flywheel energy storage system are the: rotor, shaft, bearings, electric drive motor/electric generator and enclosure/support frame. Despite this simplicity, further complexity can be found in developed and refined commercial flywheel systems. The work reported herein addresses some of the fundamental design decisions and optimizations of a simple flywheel for the above identified application and deployment situation.

The focus in the paper is directed at the rotor form and geometry so that it can operate at a high rotational speed (10,000rpm - considered low speed for some flywheel applications) without exceeding rotor material safe working stress limits. Simplicity of manufacture and construction have been factored into the early design decisions and solution, to ensure ease of producibility, with the available resources in the central Pacific region. Naturally, the objective of having the system developed and maintainable within the business centers of the central Pacific does not and cannot mean that all technological components and/or sub-systems are sourced locally, within this target region. Where possible and practical parts and systems will be made or assembled in the region. Outside of this possibility, parts may be sourced from relevant and qualified sources overseas. The ultimate aim is for a system that can be developed, manufacture (where practicable), assembled and maintained locally.

The associated energy losses are also considered in the development work and viable design solutions are investigated, modeled and optimized for the major contributors, namely the mechanical bearing and the rotor windage losses. Mechanical bearing power losses and severe life limitations are minimized through the introduction of electro-magnetic bearings, used in conjunction with mechanical bearings. The mechanical load sharing and self-stabilizing characteristics of this combined system make it an ideal solution. In addition, the windage losses, due to the high rotor speed when running in an air atmosphere, are reduced to a workable level through the adoption of a helium partial vacuum environment, contained by the flywheel enclosure. Various options and working vacuum pressures were investigated in conjunction with the practicality of their implementation. Some of the findings are presented in the later sections of this paper. The implications and design detailing of the enclosure and shaft have been considered but only at a fundamental level at this preliminary design development stage. Naturally, the electrical motor/generator and micro-grid elements are introduced and discussed in the context of the energy storage system.

A. Flywheel Rotor Materials and Configurations

In the design of a flywheel rotor two notable decisions have to be made, amongst many others. These decisions relate to the material and shape of the rotor due to the material composition and production processes used to shape it into the final form. Not all of the desired shapes can be made with the selected materials. Four of the most common shapes and their suggested production materials can be seen in Figure 1. In this figure the shape factor of each flywheel form is also listed. The factor relates the maximum working strength σ_u [Pa] and the density ρ_v [kg/m3], to the energy per mass W^M [J/kg], as in (1) below.

$$W = \frac{J\omega^2}{2} \tag{1}$$

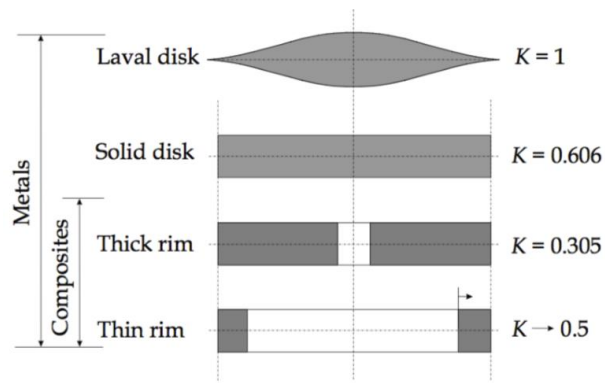

Figure 1. Shape factor of common flywheel shapes [5]

During operation, it is suggested that a maximum speed drop in the flywheel rotor should not be greater than 50% [6]. Since the stored energy is related to the square of the rotor angular speed, ω^2, the energy recovered from the rotor, in this case, is 75% of the total stored energy in the flywheel. In order to fulfil the design specifications the demanded energy should be equal to the required (specified) energy of 25 kWh, as derived in Section II above. The maximum rotational speed for a low speed flywheel is typically limited to 10 000 rpm [7] which gives a minimum rotational speed of 5 000 rpm as per the argument above.

In this report three potential candidate materials are initially considered: High tensile Steel AISI 4340, S2-glass and Carbon T1000. The last two materials listed are composites and therefore can be applied to thick rim or a thin rim rotor design solutions, as per Figure 1. The High tensile steel is more suited to a Laval disk. The solid disk has been

removed from the development since the Laval disk has all the advantages of the solid disk and fewer disadvantages. The specifications of all the three materials are listed in Table II. Here, the maximum allowable working stress σ_u is determined by dividing the yield strength by a safety factor, n. The value of n is material and application specific. For this scenario a typical value of 1.5 has been chosen for the metallic material, however close attention must be paid to the selection of this value in a commercial design solution. For the composite materials the 'ultimate tensile strength' is divided by a safety factor, n, of 2.0. Normally, composite material safe working limits demand thorough investigation to determine.

TABLE II. MATERIAL SPECIFICATIONS [6, 8]

	Density, ρ	Working Stress, σ_u	Poisson Ratio, ν	Unit Price
	[kg/m3]	*[MPa]*		*[$/kg]*
Steel AISI 4340	7830	486	0.29	1.00
S2-glass	1920	735	0.22	24.60
Carbon T1000	1520	975	0.20	101.80

B. Thick Rim Flywheel

There are two stress components that are considered important in the design of a flywheel rotor. The radial stress presented in (2) and the hoop stress expressed by (3) [6].

$$\sigma_r(r) = \frac{3+\nu}{8}\rho\omega^2\left(r_0^2 + r_1^2 - \frac{r_0^2 r_1^2}{r^2} - r^2\right) \quad (2)$$

$$\sigma_\theta(r) = \frac{3+\nu}{8}\rho\omega^2\left(r_0^2 + r_1^2 + \frac{r_0^2 r_1^2}{r^2} - \frac{1+3\nu}{3+\nu}r^2\right) \quad (3)$$

Where ρ is the mass density [kg/m3], ω is the rotational speed [rad/sec], ν is the Poisson ratio, r_0 is the outer radius of the rotor [m], r_1 is the inner radius of the rotor [m] and r represent any radius within the rotor [m]. It can be seen from simulations that the hoop stress in a thick rim rotor is much higher than the radial stress. Furthermore, the hoop stress is at its maximum at r_1, the inner radius.

TABLE III. CONFIGURATIONS OF A THICK RIM FLYWHEEL, W = 25KWh

	Inner radius, r_1	Outer radius, r_0	h	m
	[m]	*[m]*	*[m]*	*[kg]*
Steel AISI 4340	0.050	0.260	7.645	12370
S2-glass	0.468	0.617	1.5	1460
Carbon T1000	0.217	0.594	1.5	2190

Consequently, when designing for the limiting hoop stress at the inner radius the total stored energy W [J] can be calculated by: $W = ½J\omega^2$. Where J is the flywheel moment of inertia [kgm^2]. The best rotor configurations are calculated per material specifications. Here the active energy storage is kept to 25kW at a maximum rotational speed of 10 000rpm. Table III gives the computer results for the overall dimensions and mass of the flywheels with a thick rim produced in the three target materials.

C. Thin Rim Flywheel

When designing a thin rim flywheel the procedure is similar to the thick rim configuration. The only simplification which is applied is due to the small cross-sectional area, resulting in a hoop stress that can be taken as constant throughout the rim thickness. Then (3) can be simplified to (4) [5]

$$\sigma_\theta = \rho r_1^2 \omega^2 \quad (4)$$

With this equation the best configurations of a cylindrical flywheel can be calculated and are shown in table 3.

TABLE IV. CONFIGURATIONS OF A THIN RIM FLYWHEEL, W = 25KWh

	Inner radius, r_1	Outer radius, r_0	h	m
	[m]	*[m]*	*[m]*	*[kg]*
Steel AISI 4340	0.135	0.23	13.10	11920
S2-glass	0.490	0.59	2.29	1488
Carbon T1000	0.440	0.54	3.85	1804

D. Equal Stress Flywheel

The equal stressed disk is also known as the Laval disk. As has been shown in figure 1, this shape is ideally made from metals. The main reason for the usage of the Laval disk is the high shape factor K which results in a higher energy storage per mass. The shape of an equally stressed disc can be obtained from the known equations of equilibrium and compatibility written in terms of stresses for linear, isotropic materials with constant characteristics [9], as follows.

$$\frac{d(\sigma_r r h)}{dr} - \sigma_\theta h + \rho\omega^2 r^2 h = 0 \quad (5)$$

$$(\sigma_\theta - \sigma_r)(1+\nu) + r\frac{d\sigma_\theta}{dr} - \nu r\frac{d\sigma_r}{dr}\sigma_\theta = 0 \quad (6)$$

If $\sigma_r = \sigma_\theta = \sigma_u$ = const then from (5) the profile h(r) [m] can be computed. This profile is given in (7).

$$h(r) = h_c e^{-B\chi^2} \quad (7)$$

With dimensionless constants $B = \rho\omega^2 r_0/2\sigma_u$ and $\chi = r/r_0$.

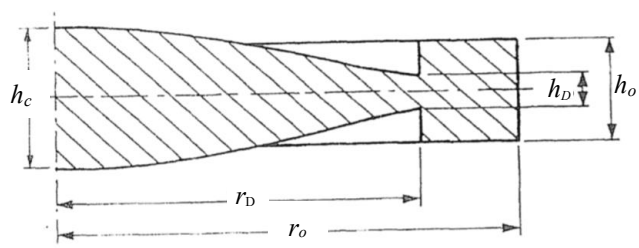

Figure 2. Constant-stress disc with constant-thickness outer rim [9]

The problem with this profile is that the radius goes to infinity. To create a boundary on the infinite profile a rim is needed at the end of the profile [9]. By creating a rim the shape factor is reduced. With $\beta = r_D/r_0$ as the ratio between the inner radius of the rim and the outer radius and $\alpha = h_0/h_D$ as the ratio between the thickness of the rim to the connection point. Both can be seen in Figure 2. The radial thickness β depends not only on B but also on α [9]. This relation is shown in (8) and with this the link to the revised shape factor can be determined as is shown in (9).

$$\beta = \sqrt{\frac{1}{B\alpha}\left[\alpha - 1 + 2\sqrt{\frac{\alpha^2 B(B-1+\nu)}{(1-\nu)^2} + \frac{(\alpha-1)^2}{4}}\right] - \frac{1+\nu}{1-\nu}} \quad (8)$$

$$K = \frac{1+[\frac{\alpha B^2(1-\beta^4)}{2} - B\beta^2 - 1]e^{-B\beta^2}}{1+[\alpha B(1-\beta^2)-1]e^{-B\beta^2}} \quad (9)$$

In table V are the two optimum configurations for a steel, equal stress disk with and without a small rim are given. The resultant profile of the flywheel rotor with a rim can be seen in Figure 3.

TABLE V. CONFIGURATIONS OF LAVALDISK FLYWHEEL, $W = 25$KWh

Steel flywheel AISI 4340	r_0 [m]	r_D [m]	h_c [m]	h_D [m]	m [kg]
Without rim	1.0	-	0.737	-	1934
With rim	0.525	0.391	0.752	0.195	2140

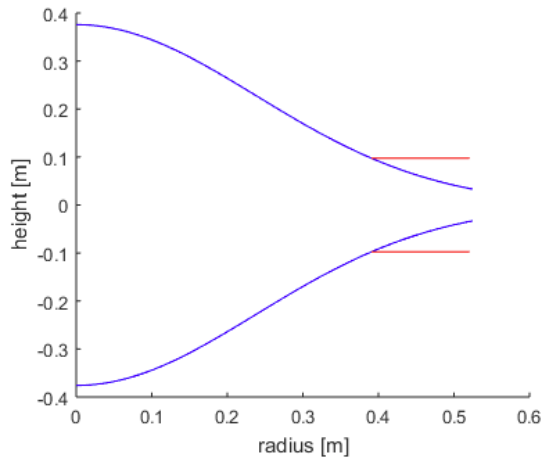

Figure 3. The profile which results from (7) and rim added in red

III. FLYWHEEL LOSSES

For a flywheel to be efficient and useful the losses must be estimated and any 'high-loss' problem areas addressed at the design stage of the flywheel development. In this study an estimation of the air friction and bearing friction losses were made. In turn, this influenced critical design decisions and informed the resultant design solution.

A. Air Friction

The friction which results from the differential speed between the flywheel rotor and the surrounding air can cause high frictional losses. The equation that is used to determine the air friction losses of a rotating circular disk in air is given below (10) [10]. The flywheel design report herein is not a circular disk but the suggested approach gives a close first estimate of the anticipated losses. A refined analysis and simulation will be conducted later in the project. In the equation the terms are as follows: power loss $P_{a,l}$ [W], ρ_a is the gas density [kg/m^3], β_a is the dynamic viscosity of the gas [Pa · s], ω the angular speed of the rotor [rad/s], r is the cylindrical flywheel rotor radius [m] and h the thickness (height) of the flywheel [m].

$$P_{a,l} = 0.04\rho_a^{0.8}\beta_a^{0.2}(\omega r)^{2.8}(2r)^{1.8}(\alpha + 0.33) \quad (10)$$

with:

$$\alpha = \frac{h}{2r} \quad (11)$$

It is commonly understood that air friction losses are considerable for flywheel systems, particularly when operated at rotational speeds as high as 10 000 rpm. Consequently the effects of different 'housing' gases and levels of vacuum were explored. Within the comparative analysis, four different gases were considered and it was assumed that the ambient operating temperature expected in the Pacific would be around 50°C. The four gases under consideration were Air, Ammonia, Helium and Hydrogen. These gases were chosen for their low density and low viscosity at 50°C. In Figure 4 it can be seen that when the flywheel housing pressure is lowered the enveloping gas frictional losses are decreased. It can also be seen that when the flywheel rotates at atmospheric pressure in air; the power loss will be considerable (261kW) for the case introduced above. This is over 10 times the storage capacity of the system and clearly illustrates the deficiency of the FESS operated in an air environment at ambient pressure. A combination of other gases and a lower pressure reveals a solution to this problem. Under these revised conditions the relative losses were significantly lower.

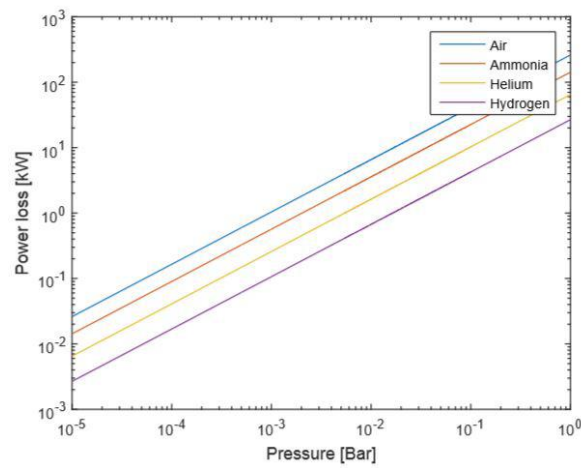

Figure 4. Extra torque force due to the air friction

In Figure 5 the gas friction is plotted as a function of flywheel rotational speed. In this plot the gas is Helium at a pressure of 0.001 bar.

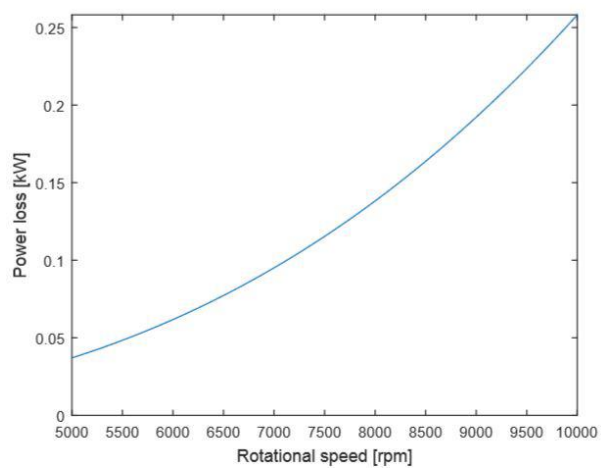

Figure 5. Enveloping gas friction on flywheel with Helium at 10^{-3} bar

This pressure was chosen as it is within the range of mechanical pumps which can draw vacuums down to 10^{-5} bar. It can be concluded from this graph that the flywheel will slow

down in 48 hours from maximum speed to the minimum speed (6.1 kWh is lost due to gas friction). This is an average of 0.13 kW. Note that hydrogen demonstrates better operational results than helium but the explosive potential when pure hydrogen blends with air is a major safety concern and consequently it is not considered further. Helium is not as flammable as Hydrogen and is for this reason a better option.

B. Bearing Losses

Since in the previous section it is concluded that low pressure conditions are needed around the flywheel rotor the mechanical flywheel bearings (if used) must consequently be sealed and isolated from the vacuum in order to prevent the lubrication from outgassing and affecting the quality and condition of the vacuum.

As a consequence, a form of lip seal is needed in order to isolate the bearing lubrication from the vacuum. The frictional losses, in the case of the mechanical bearings, are the rolling friction and hydrodynamic losses in the bearing due to the associated lubrication – particularly in the higher loaded bearing at the base of the flywheel, assuming a vertical rotational axis. These losses comprise a rolling frictional torque M_{rr}, a sliding frictional torque M_{sl}, a seal frictional torque M_{seal} and the frictional torque due to drag losses M_{drag}. The equations for all these individual losses can be found in the SKF catalogue [11]. Through the related analyses a mechanical bearing was identified with relatively low losses. The calculated total frictional torque amounts to 1.41Nm. From this value it was determined that the power loss would be 1.4kW, which would require an increase of 5.6% of the stored energy per hour for compensation purposes. This is a high loss value. Subsequent analyses validated the use of magnetic support bearings to carry of 87.5% of the weight of the flywheel and therefore reducing the mechanical bearing losses down to 170W - due to support load reduction. This proposal introduced a reduction of the stored energy losses to 0.68%. Further investigations are due to be conducted on the feasibility of using a bearingless motor.

IV. Flywheel Dynamical Model

Generally a FESS is made up of a flywheel connected to an electrical machine M and of back-to-back bidirectional converters to the grid or local load (Fig.6).. Since the flywheel can vary in speed and the load or the grid must be supplied at a certain fixed frequency and voltage, the electrical machine is connected to an AC/DC rectifier a capacitor DC link and another DC/AC converter which interfaces with the grid. Each converter must be operated in a bidirectional mode (back-to-back converters). Loads can be either directly connected just after the grid-interface converter or directly to the DC link. Furthermore RES are generally connected to the DC link, the voltage of which must therefore be controlled to manage the energy fluxes. With this configuration the electrical machines connected to the fly-wheel can work either as a motor or as a generator according to the active power fluxes. If no exchange of energy occurs, the FESS is at idle mode and rotates at constant speed

In general, for low-speed applications, like this one, an induction motor is used especially for its robustness to sudden change of operational mode, and is operated in field-weakening mode, [12] [13]. Magnetic or superconducting bearings should be used to decrease friction losses, at the expense of an increase of the cost of the system and the complexity of the control systems.

Typically both converters are controlled with high performance methods to ensure proper active and reactive power fluxes: the generator and its converter can be either controlled with a FOC (Field Oriented Control) or a DTC (Direct Torque Control) with a control in speed and flux. The grid converter can be correspondingly controlled with a VOC (Voltage Oriented Control) or a DPC (Direct Power Control). The DC link control is provided by the grid side inverter [12].

In any case a dynamical model of the flywheel suitable for control applications must be used. In section IV losses are discussed: these where the bearing frictional losses and air frictional losses. For the dynamic model presented in this paper, the pressure around the flywheel rotor is chosen to be 1 mbar Helium, while the magnetic support on the bearing is 22 kN. This makes the resulting axial force be reduced from 25 kN down to 3 kN on the bearing. The bearing frictional torque M_b [Nm] is given by data from the catalogue of SKF [11] as follows:

$$M_b = 5.4\ 10^{-6}\omega + 0.106 = \alpha\omega + \beta \quad (12)$$

From eq (11) the following equation results:

$$T_{air} = 0.1393\ \rho_a^{0.8}\beta_a^{0.2}r^{4.6}(\alpha + 0.33)\ \omega^{1.8} = \gamma\ \omega^{1.8} \quad (13)$$

where T_{air} [Nm] is the torque due to the air friction on the flywheel rotor. The resulting mechanical equation then results:

$$J\frac{d\omega}{dt} = M(t) - M_b - T_{air} = M(t) - \alpha\omega - \beta - \gamma\ \omega^{1.8} \quad (14)$$

where $M(t)$ [Nm] is the electromagnetic torque of the electrical machine. If $\omega = d\theta/dt$, then the following state equations can be written in standard form:

$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\end{bmatrix} = \begin{bmatrix}\dot{\omega}\\ \dot{\theta}\end{bmatrix} = \begin{bmatrix}\frac{\alpha\omega}{J} - \frac{\beta}{J} - \frac{\gamma\omega}{J}\sqrt[5]{\omega^4}\\ \omega\end{bmatrix} + M(t)\begin{bmatrix}\frac{1}{J}\\ 0\end{bmatrix} \quad (15)$$

With the help of Solidworks® the value J = 219.3 kg m^2 is obtained for the flywheel rotor.

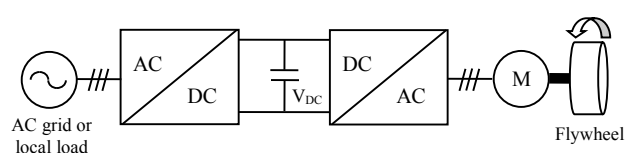

Figure 6. Block diagram of a FESS

Eq. (15) gives a set of nonlinear equations, and this characteristic, together with the well-known nonlinear dynamical model of AC machines, suggests that nonlinear control methods be used, like the feedback linearization.

V. CONCLUSION

This work has contributed to the first steps in realizing an electro-mechanical flywheel system for energy storage in small Pacific island communities. The flywheel was optimized for the projected storage capacity of 25kW. The optimization was primarily driven by a limiting stress criterion within the flywheel rotor body, as induced by the rotational speed (10,000rpm). Furthermore, the construction materials, geometry and general configuration were considered in context of Design For Manufacturing (DFM), relevant to the target environment and a suitable configuration, Laval disk with favorable shape factor, was identified and developed into a virtual model in alloy steel. The shortfalls and losses associated with mechanical flywheel storage systems were also addressed and evaluated in context of mechanical bearings and rotor windage. Specifically, through the elimination of the all-mechanical bearings and consequent introduction of a validated electro-magnetic and mechanical bearing system, the associated losses were brought to a practical and workable level. Additional gains where embodied into the design solution through the proposal of operating the flywheel in a partial vacuum, helium atmosphere - the implementation and maintenance of which is aligned with readily available technologies. The implications and design detailing of the enclosure and shaft have been considered but only at a fundamental level at this preliminary design development stage - this work will be presented in more detail, once sufficient validation work has been completed. The electrical motor/ generator and micro-grid elements have been introduced and discussed in the context of the energy storage system. A model for the dynamical behavior of the system with an induction motor electrical drive working in deep field weakening was presented with simulated and emulated test results in [14]. The reported work is leading to the practical implementation of a FESS bound micro grid in remote Oceania.

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