Hybrid heave drilling technology reduces emissions, operating costs for offshore drilling

By Kevin R. Williams, H.J. de Jone,
KRW Technologies

ABSTRACT

In order to analyze the anticipated performance of a large scale flywheel-based energy storage system in a real-world application, a detailed simulation of a heave compensating drawworks developed. The simulation as based on an actual drawworks, HITEC AHC-1000® to ensure relevance to practical applications. The simulation also includes a mathematical model of the flywheel dynamics and a proprietary power routing algorithm which holds the machine's total power consumption constant. Fuel consumption data can also be analyzed by including the characteristics of a Caterpillar diesel generator set. The simulation results show a dramatic reduction (up to 75%) in average electric power demand, and even greater reductions (often up to 90%), in peak power draw.

INTRODUCTION

Many electro-mechanical cyclical processes could potentially benefit from the use of an energy storage system able to absorb, store, and relinquish large amounts of energy. Commonly used energy storage systems such as batteries (chemical), hydraulic accumulators, or capacitors (electrical) are limited by size, weight, cost, capacity, power output, cycling or efficiency considerations. A spinning flywheel (kinetic energy storage) is an ideal, cost-effective way to store large amounts of readily-available energy.

A deep-sea drilling vessel fitted with a drawworks-based heave compensation system is an ideal candidate for the application of such an energy storage system. The highly periodic and predictable characteristics of ocean swells readily allow for highly efficient energy recovery; thereby eliminating the need for wasteful dissipation schemes.

THE DRAWWORKS

Dynamics

To ensure results relevant to the real world, the specifications of the drawworks model used are based on an actual drawworks HITEC AHC-1000®, a 1000 ton drawworks, in operation.

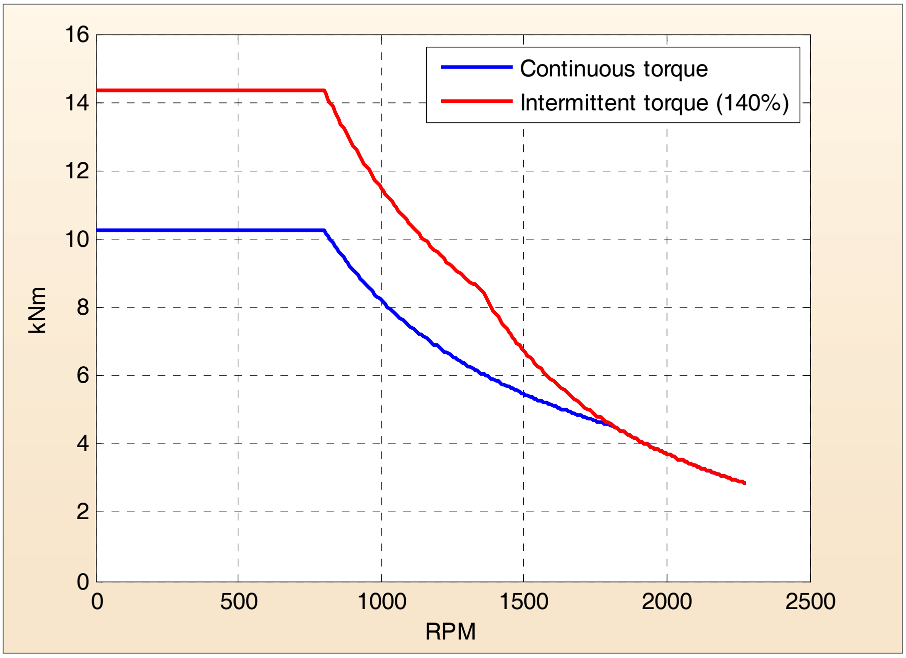


Figure 1: GEB22® AC motor speed-torque characteristic.

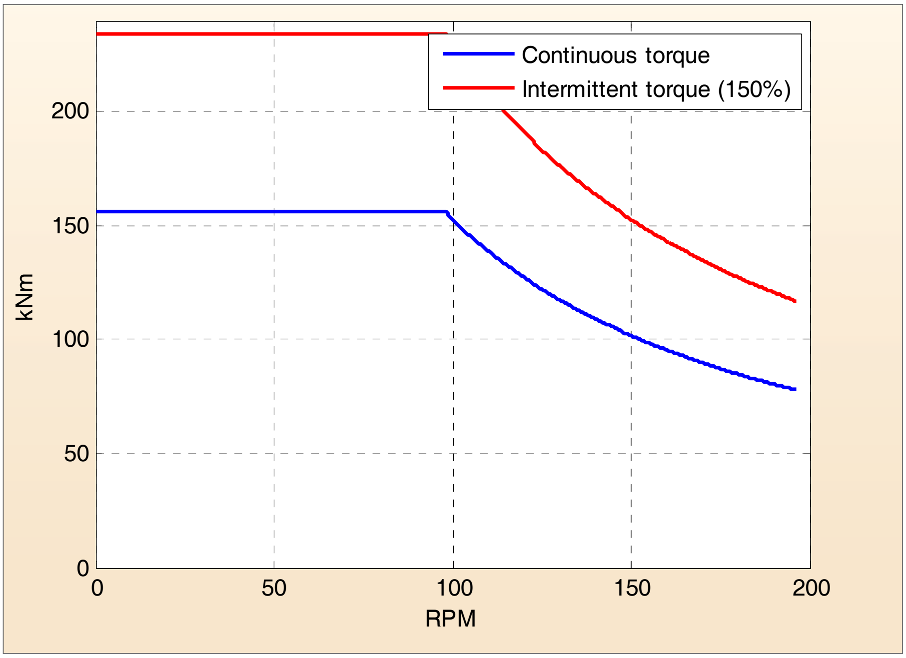


Figure 2: Proposed high torque AC motor speed-torque characteristic.

All relevant mechanics of the drawworks are considered and described mathematically. The model consists mainly of small, mathematically simple parts such as a winch drum or a gear reduction, coupled together mathematically via resultant forces. Main sources of friction and drag are included, such as wire rope elasticity and damping and sheave friction.

Mechanical systems

The dynamics model consists of the following simple mechanical systems:

- The AC motors
- Gear reductions
- The winch drum and winding mechanics
- Wire rope, sheaves and hook load

The systems are separately modeled and coupled via force balances. The mechanical model is essentially a classic mass-spring-damper problem, with the inclusion of several important real-life non-linearity.

Motors and drives

The HITEC AHC-1000® drawworks uses six GE GEB22® AC induction motors. They are modeled as devices that convert electrical power into a mechanical torque directly applied to a rotating inertia (the rotor) at a constant efficiency of 95%. The motors and drives are able to fully regenerate power, also at an efficiency of 95%. The motor models are speed-controlled via PID controllers. The outputs are limited by speed-torque characteristics and major drive parameters (slew rate, speed limits, etc.). Due to the periodic nature of heave compensation, the motors are allowed run at 140% load while heave compensating.

The simulation will also be run using an innovative new dual-stator AC motor currently in development by KRW Technologies, Inc. This motor is a large, low speed, very high torque machine that will be directly coupled to the winch drum, eliminating the necessity for a gear reduction. The rotor is ring shaped, with a concentric outer stator and inner stator. The outer stator will deliver 68% of the total torque, while the inner stator delivers the remaining 32%.

The drawworks will require four such motors, two on each side of the drum, coupled in series to a common shaft. Since rotational inertia on the high speed side of a gear reduction is effectively multiplied by the square of the gear ratio, this direct drive setup will have a significantly lower total inertia (approximately 35% lower than the GEB22 equipped system) while the overall weight stays roughly the same. This system will also benefit from the elimination of gear backlash, greater redundancy; lower cost drive system and high breakdown torque (225% of nominal).

Torque characteristics of both types of motors are plotted in figures 1 and 2. General drawworks specifications are listed in tables 1 and 2. The specifications for both drawworks mirror the actual HITEC AHC-1000® specifications as closely as possible (with the obvious exceptions in the direct drive system).

Gear reductions

Gear reductions simply multiply speed and torque by reciprocal ratios. Gearbox

Drawworks specs			
	Symbol	Value	Unit
Gear ratio	GR	10.5	
Number of line parts	N	14	
Drum radius	R	1.867	M
Drum length	L	2.057	M
Drum inertia	I_d	12000	$kg.m^2$
Wire rope diameter	2.Rw	50.8	Mm
Wire rope elastic modulus	E	90	GPa
GEB22 AC induction motor			
	Symbol	Value	Unit
Nominal speed		800	RPM
Continuous torque	T_c	10,260	N.m
Continuous power	P_c	858	kW
Intermittent overload capacity		140	%
Intermittent torque	T_i	14,364	N.m
Intermittent power	P_i	1,201	kW
Inertia	I_m	18.2	$kg.m^2$
Number of motors	N_m	6	

Table 1: HITEX AHC-1000® ton drawworks specifications.

Drawworks specs			
	Symbol	Value	Unit
Gear ratio	GR	1	(direct drive)
Number of line parts	N	14	
Drum radius	R	1.867	M
Drum length	L	2.057	M
Drum inertia	I_d	12000	$kg.m^2$
Wire rope diameter	2.Rw	50.8	Mm
Wire rope elastic modulus	E	90	GPa
High torque/dual stator AC induction motor			
	Symbol	Value	Unit
Nominal speed		98	RPM
Continuous torque	T_c	155,900	N.m
Continuous power	P_c	1600	kW
Intermittent overload capacity		150	% (conservative rating)
Intermittent torque	T_i	233,900	N.m
Intermittent power	P_i	2400	kW
Inertia	I_m	1950	$kg.m^2$
Number of motors	N_m	4	

Table 2: Proposed direct drive drawworks specifications.

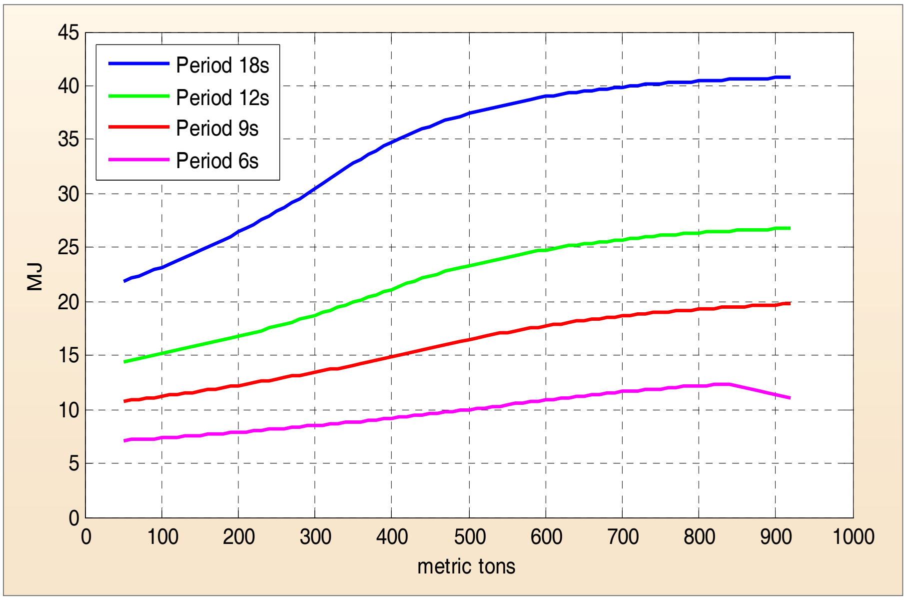


Figure 3: Maximum energy fluctuation.

inertias are included. Associated non-linearity includes friction and gear backlash.

Winch and cables

The winch drum is modeled as a rotating device with a rotational inertia that converts torque into linear force. Furthermore, the winding of cable layers is included in the model. When a layer fills up, the overall radius of the winch drum increases accordingly. The effective inertia of the drum increases as cable is wound on it. In the case of the parameters considered here, the inertia of the fully wound drum is almost twice the inertia of the empty drum.

The cables are modeled as damped springs with a variable length (and therefore, variable spring and damping values). The modulus of elasticity for the cables used is 90 GPa, and the damping ratio is approximately chosen so that the overall system is slightly underdamped.

Control

The simulation includes several common control and safety systems, such as an acceleration limiter, floor/crown saver and breakaway protection. For the situation being considered most of these systems are disabled except for the active heave compensating system. This system uses a heave velocity signal to modulate the speed demand signal that is sent to the motor drives.

ESTIMATED POWER AND ENERGY DRAW

For proper flywheel sizing, an estimate should be made of the drawworks' peak power draw and energy fluctuation. For a rough estimate it is adequate to consider the potential (block height relative to ship deck) and kinetic (rotation of drum and motor) energy contained in the drawworks system, neglecting friction:

$$E = \frac{1}{2} I_t \dot{\theta}^2 - MgH(t) \tag{1}$$

where

$$\dot{\theta} = \frac{N}{R}\frac{dH(t)}{dt},$$

$$H(t) = A\sin\left(\frac{2\pi}{P}t\right),$$

$$I_t = I_m GR^2 + I_d$$

In this simple approximation, the hook is assumed to be held absolutely motionless while the ship heaves sinusoidal with heave height *H(t)*. Energy is contained as potential energy in the relative hook height and as kinetic energy in the rotating motors and winch drum. Drum speed $\dot{\theta}$ is assumed to be exactly proportional with heave speed, with a factor based on the number of line parts *N* and drum radius *R*. The total effective inertia I_t stems from the drum inertia plus the total effective inertia of the motors seen through a gear reduction of *GR*.

Using the following equations, we can find the absolute minimum and maximum of *E(t)*, and the total (peak-to-peak) energy fluctuation:

$$\Delta E = 2C_2 \qquad \text{if} \quad C_2 > 2C_1 \tag{2}$$

$$\Delta E = C_1 + \frac{C_2^2}{4C_1} + C_2 \qquad \text{if} \quad C_2 \le 2C_1$$

where

$$C_1 = \frac{2\pi^2 N^2 A^2 I_t}{R^2 P^2}$$

$$C_2 = MgA$$

Using these equations and taking the torque and power limits into account, an estimate can be made as to the maximum energy fluctuations. Since the energy fluctuation can become arbitrarily large with increasing heave period (allowing high-amplitude, but low-speed heaving motion), an upper limit to the heave period must be set. This limit is chosen as 18 seconds, corresponding to the maximum period considered in the HITEC AHC-1000® specifications. Figure 3 shows the maximum possible energy fluctuations for varying hook loads and heave periods. For an 18 second period, the maximum energy fluctuation is around 41 MJ. Note that none of these calculations take friction into account, so the actual number will be somewhat lower. The maximum power draw can of course directly be determined from the specifications of the motors. Accounting for 6 motors at 140% capacity and 95% efficiency, the maximum power draw will be approximately 7600 KW.

FLYWHEEL SYSTEM

Dynamics

Dynamically the flywheel is a very simple device. It is a simple torque device (an AC induction motor/generator) coupled to a large rotational inertia. The governing equation is:

$$T = I\dot{\omega} + T_d \tag{3}$$

where *T* is shaft torque, T_d is aerodynamic drag torque, *I* is the rotational inertia and $\dot{\omega}$ denotes the time derivative of angular velocity. The total amount of kinetic energy contained in the rotating mass is

$$E = \tfrac{1}{2} I\omega^2 \tag{4}$$

and the power transfer is the time derivative of this.

The aerodynamic drag of the spinning flywheel is estimated by considering the shear drag on a flat plate, aligned parallel to a fluid stream:

$$F_{plate} = \tfrac{1}{2} C_{Df} \rho A V^2 \tag{5}$$

where F_{plate} is the drag force, C_{Df} is the shear drag coefficient, ρ is fluid density and *V* is linear velocity. Assuming the flywheel is cylindrical with thickness *D*; we can integrate equation (5) over the entire surface of the flywheel, and deduce the total drag torque:

$$T_d = \iint_S F_{plate} r\,dA = \pi\rho C_{Df}\omega^2\left(\tfrac{2}{5}r^5 + Dr^4\right) \tag{6}$$

A number of empirical formulas exist to determine the drag coefficient and in this case the following formula for turbulent flow is used (from Munson et al. (1990)):

$$C_{Df} = \frac{0.455}{(\log(\mathrm{Re}))^{2.58}} \tag{7}$$

where the Reynolds number is based on the flywheel radius and tip speed. A much more detailed analysis can be found in Dorfman (1963).

Details on the simulated flywheel are listed in table 3. These specifications were chosen as a compromise between several factors, such as overall weight, energy capacity and (aerodynamic) power dissipation. Note that the induction motor mostly operates in a speed range above its nominal speed, in the constant-power region. This is preferable for a flywheel system that should be able to output nominal power regardless of speed.

Control System

The flywheel system has two main goals. First, it should store and reuse regenerated power realizing a lower overall average power consumption. Second, it should buffer the equipment's power requirements in such a way that the power source sees a relatively constant load profile free of extreme peaks and valleys. Given unlimited (or very high) energy storage capacity or perfect predictability of the equipment power demand, it is a trivial task to achieve these two goals. However, in the interest of practicality and cost, it is desirable to dimension the device such that its energy and power capacity is consistent with the machine's maximum estimated peak-to-peak energy fluctuations and power demand. In the previous section, the energy fluctuation is estimated to be approximately 41 MJ and a flywheel capacity of roughly double that amount will ensure a sufficient energy range. Maximum power to be delivered by the flywheel is roughly equal to the maximum power rating of the equipment it will be used with.

The flywheel system being considered will consist of the same GEB22 AC motors as used in the drawworks coupled to a large steel disc. Two motors facing each other will drive a common flywheel situated in between them. Three of these units are sufficient to provide the power and energy storage capacity needed for the drawworks. The specifications are listed in table 3. The motors all operate in parallel.

Topology

There are four main power generating or consuming systems:

A. The main power source: electrical grid, diesel generators, etc.

(power generation)

B. The "equipment" (in this section, "equipment" will refer to drawworks system as whole)

(power consumption and (re)generation)

C. The flywheel energy storage system

(power consumption and (re)generation)

D. An excess power sink (power consumption)

Note that this categorization pertains purely to the operating principles of the flywheel control algorithm; the topology of the actual power electronics equipment is not being considered here. The four systems labeled A through D in figure 4) are connected to a (hypothetical) central black box which coordinates energy transfer between the systems.

Algorithm

The controller routes power to and from the four systems according to the following priorities in order of importance:

-1- the power demand of the application (B) is satisfied.

Motor/generator (2 per flywheel)		
Nominal speed	800	RPM
Nominal power	746	kW
Intermittent power	1,201	kW
Flywheel		
Diameter	1.9	m
Thickness	0.20	m
Inertia	2000	$kg.m^2$
Weight	4555	kg
Speed range	500-1650	RPM
Energy capacity	28	MJ
Power dissipation @ max speed	71	kW

Figure 3: Maximum energy fluctuation.

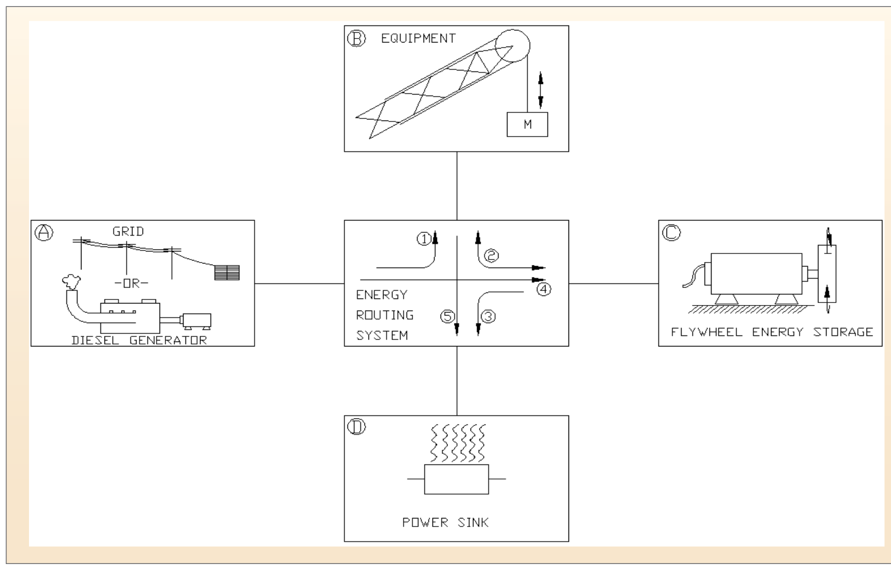


Figure 4: Power routing topology.

Energy transfer between the four systems				
	To A	To B	To C	To D
From A		R1	R4	-
From B	-		R2	R5
From C	-	R2		R3
From D	-	-	-	

Table 4: Energy transfer routes.

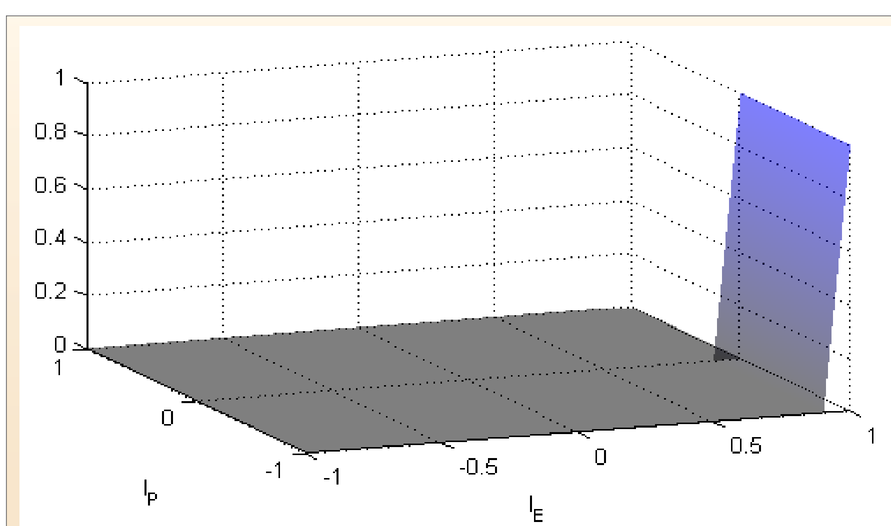


Figure 5: Power transfer surface for route R5.

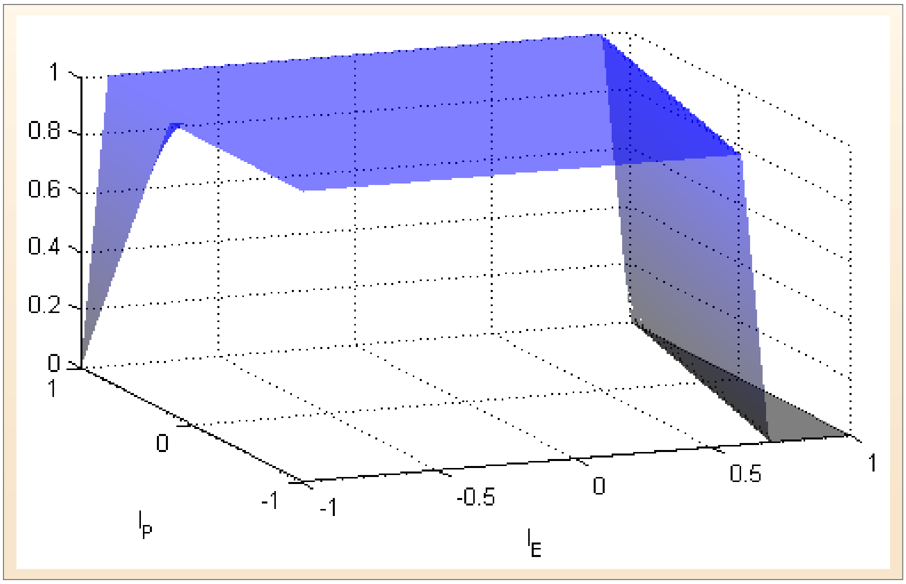


Figure 6: Power transfer surface for route R4.

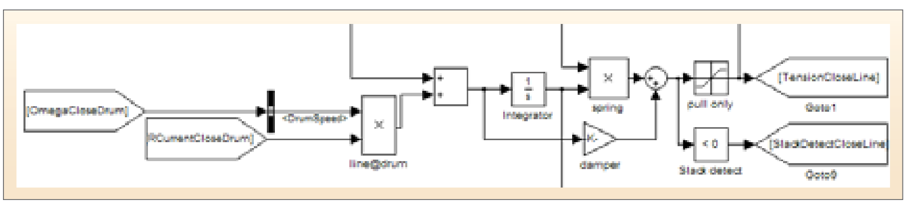


Figure 7: Simulink block diagram.

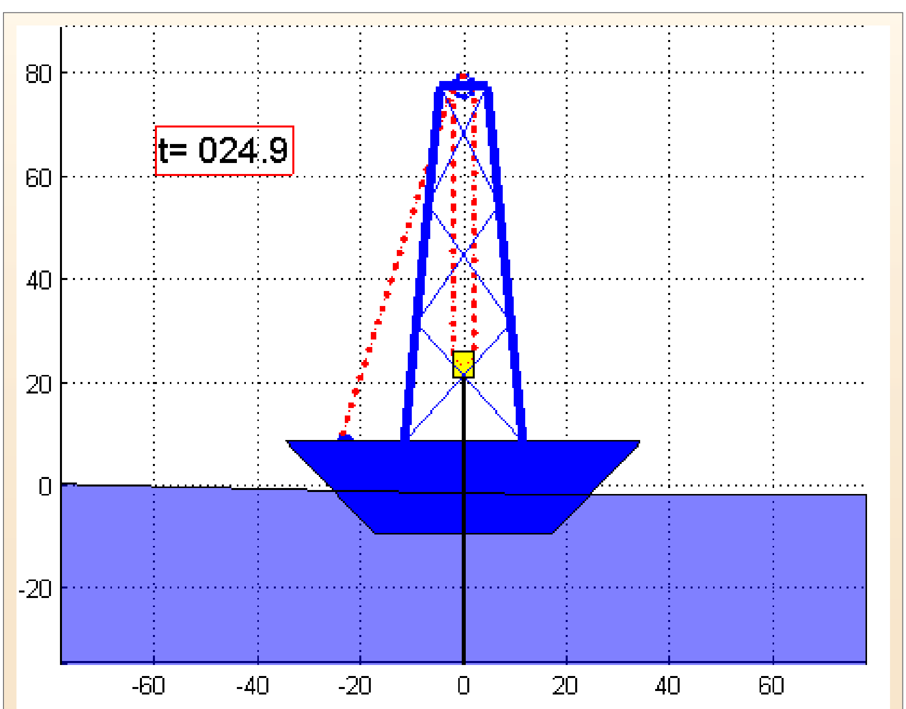


Figure 8: Screenshot of the simulation output animation.

-2- the flywheel (C) operates within its preset limits.

-3- the power drawn from the main power source (A) is constant (as possible).

-4- the flywheel contains sufficient energy to supply the next demand peak.

-5- the flywheel has sufficient "headroom" to absorb the next regeneration peak.

-6- a minimal amount of power is routed to the power sink (D).

There are several possible routes along which the black box can transfer energy, denoted as 1 through 5 in figure 4 and table 4. The unlabeled routes are either impossible or impractical in real-life situations and are not considered.

For each route listed above an energy index (I_E) and a power index (I_P) is defined, both ranging from -1 to +1. The energy index is related to the flywheel charge and the power index is related to the power being demanded by the application. The amount of power to be transferred along each route is a function of these two indices. Graphically a surface in three dimensional space is defined for each route. The value of IE and IP can be seen as coordinates defining a point on this surface. The height of the surface (ranging between -1 and 1) at this point is then a measure of the amount of power to be transferred along that route. These surfaces have been chosen so that the priorities listed above are satisfied.

As an example consider a typical surface for route R5 which controls the amount of power routed from the equipment to the power sink (figure 5). In this case IE and IP are directly linearly related to the flywheel charge and the machine's power demand. The associated surface has height 0 almost everywhere, except for where I_E>.9 and I_P<0. In other words when power demand is negative (the drawworks is regenerating) and the flywheel is almost fully charged; the system begins to dissipate power through the sink.

The power and energy indices can be directly linearly related to the current power demand or flywheel charge respectively (such as in the case of R5), or they can be filtered in some way. Similarly, the surface height is somehow related to the amount of power to be transferred along that route. For example for route R4 the power transfer is scaled by a factor obtained by passing the equipment's power demand signal through a low-pass Butterworth filter with a cutoff frequency below the machine's typical operating cycle frequency. The surface (figure 6) is shaped so that power is continuously routed at a fairly constant trickle through R4 (approximately at the rate of the machine's average power consumption) except for when the power demand is high and flywheel is almost depleted (diverting power generation resources straight to the equipment) and when the flywheel is near maximum charge.

Besides the main control system, some minor logic can be added to further optimize the system's performance. Two such additions are a grid peak limiter, which limits the maximum power draw from the grid (or generator set) to a preset value and a precharge unit which precharges the flywheel to compensate for initial filter start-up transients.

The system is easily adapted to other applications. The most important parameters to consider are those for the low-pass filters (filter order and cut-off frequency). The power routing surfaces as developed for this system are generally suitable to most cyclical applications. Usually only minor parameter adjustments are required to optimize performance.

IMPLEMENTATION

Simulink

All simulations were developed in and run in Simulink which is a software package that is used in conjunction with MATLAB (both by the MathWorks). It provides a graphical interface to model highly complex dynamic systems as the familiar block diagrams (figure 7). It is a very extensive numerical ordinary differential equation (ODE) solver. As long as a complex system can be broken down into mathematically simple components (masses, springs, etc.) it can be modeled in Simulink in a fairly straightforward manner.

The simulation is divided into two (2) separate parts. First, the drawworks dynamics simulation is run for a certain amount of time (for example, 400 simulated seconds). This simulation outputs, among other things, the drawworks' power requirement load profile. This load profile is subsequently used as an input for the simulation of the flywheel dynamics and the flywheel control system. The simulation is split up for several reasons, the first reason being modularity. The flywheel simulation can accept any time-based load profile as input, be it generated by simulation or actual measured data. The other reason is computational efficiency. Since there is only a one-way dependency between the drawworks dynamics and the flywheel system each part of the simulation can utilize its own optimal solver.

Solver

The mathematical nature of the drawworks dynamics simulation is different from that of the flywheel control system.

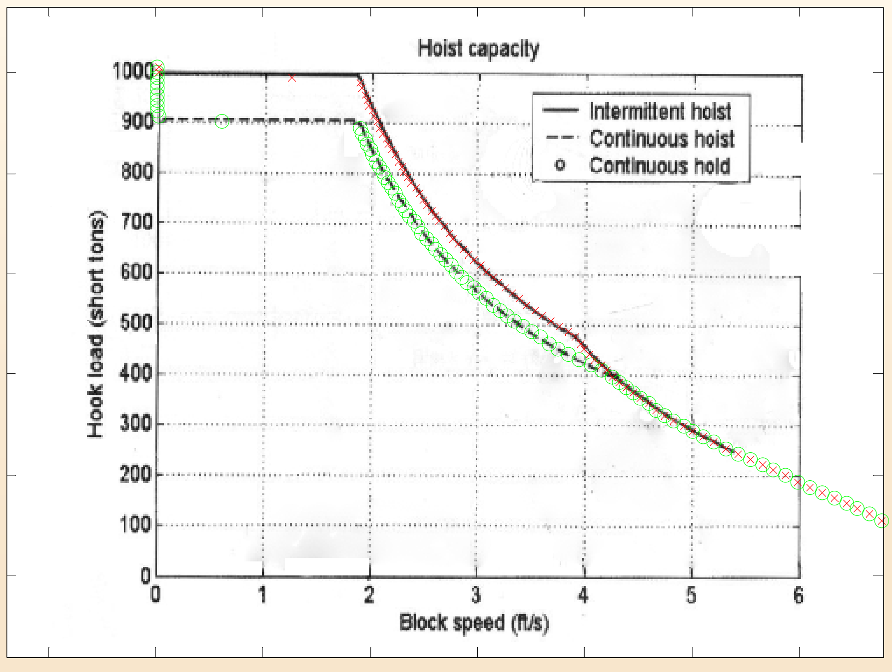


Figure 9: Hoisting capacity verification.

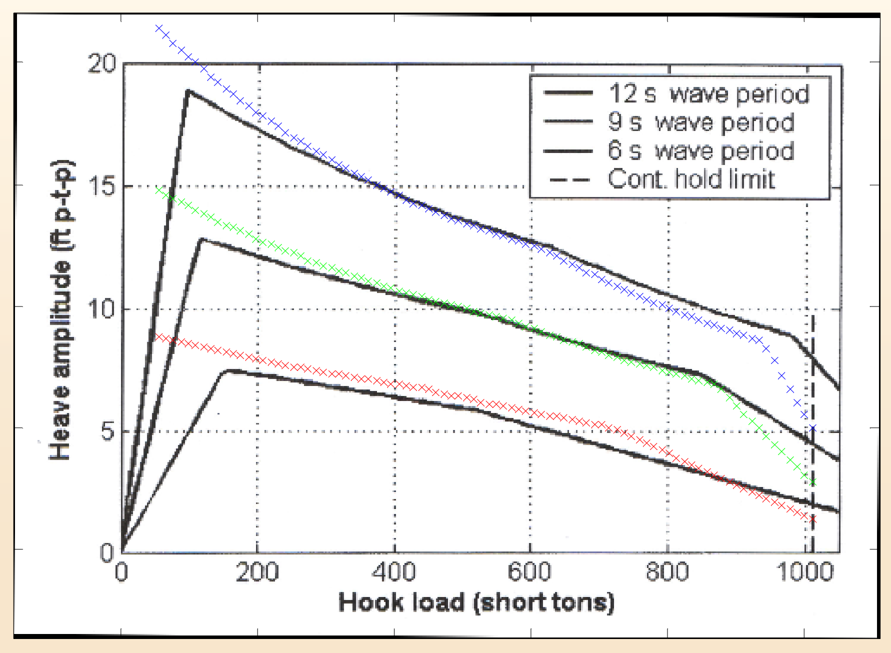


Figure 10: Maximum heave amplitude.

		Generator load with energy storage		Generator load without energy storage	
AHC1000 6s – 300 tons	Peak	488	kW	7119	kW
	Average	413	kW	1684	kW
Direct drive 6s – 300 tons	Peak	505	kW	6531	kW
	Average	390	kW	1536	kW
AHC1000 6s – 900 tons	Peak	470	kW	4510	kW
	Average	391	kW	1402	kW
Direct drive 6s – 900 tons	Peak	469	kW	4475	kW
	Average	391	kW	1401	kW
AHC1000 18s – 900 tons	Peak	933	kW	7328	kW
	Average	661	kW	2331	kW
Direct drive 18s – 900 tons	Peak	933	kW	7310	kW
	Average	661	kW	2331	kW

Table 5: Average and peak loads.

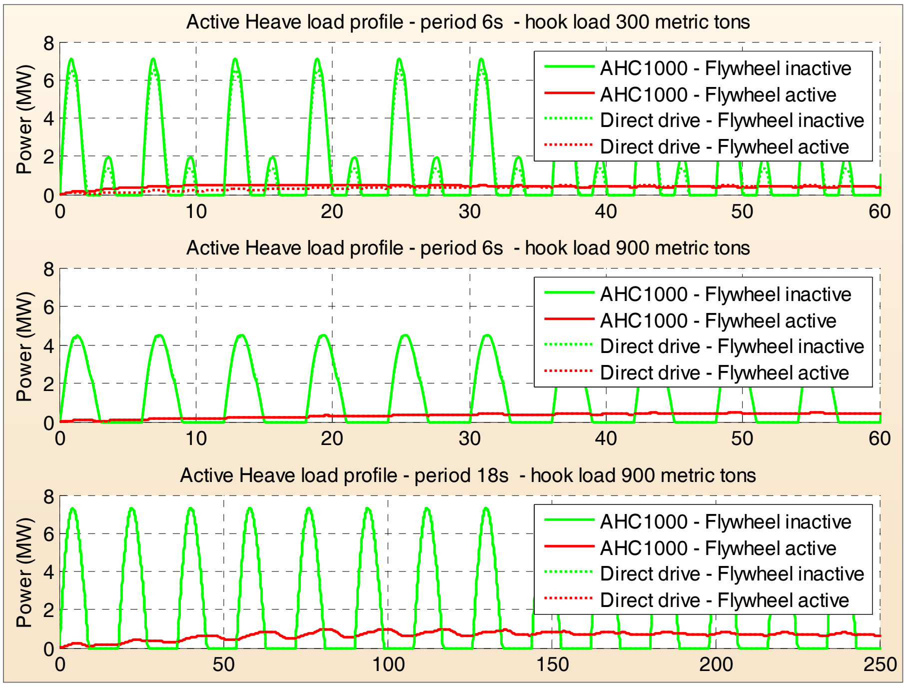


Figure 11: Load profile plots.

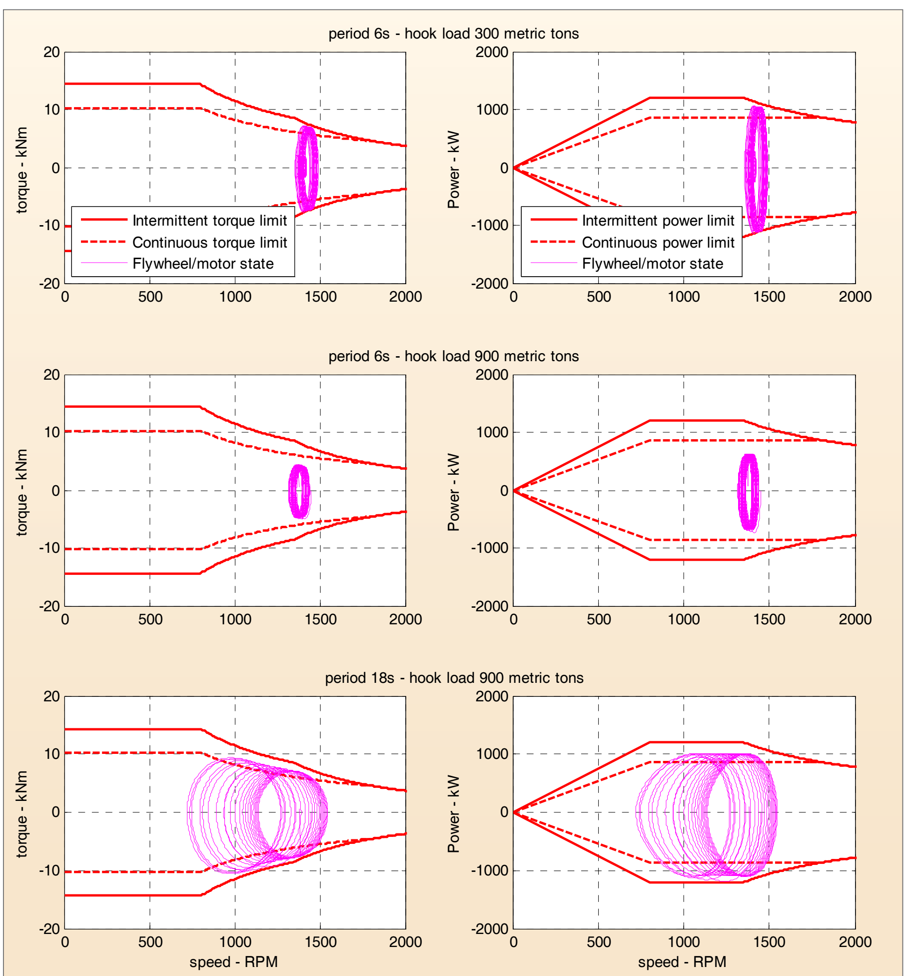


Figure 12: Flywheel operating range.

The drawworks model is a "stiff" nonlinear differential equation. This means that the solution can sometimes change very abruptly on a time scale that is very short compared to the time scale of interest. For example, when a new layer forms on the winch drum, the velocity of the wire rope changes very abruptly, and therefore needs to be calculated in simulation time steps in the order of microseconds. However, it would take very long time to compute the entire simulation (400 seconds) in simulation steps of 1 microsecond. A variable-step stiff solver is used which takes big calculation steps (for example, 0.1 seconds); until a "stiff" nonlinearity is encountered at which time it then calculates in much shorter steps.

The flywheel control system simulation contains many logic-based components that can change their output between discrete states instantaneously. This would pose a problem to a variable-step stiff solver that would reduce its step size indefinitely whenever such a discrete transition occurs (which is often). Therefore the flywheel control system is simulated using a fixed-step discrete solver.

Simulation process

Before the simulation is started a parameters file that includes simulation settings (solver, step time, simulation duration), drawworks parameters (inertias, geometry, speed-torque characteristics, etc.) and flywheel control system settings (power routing surfaces, filter settings, etc.) is loaded. First the drawworks simulation runs, then its power demand output is loaded into the flywheel simulation as an input. The most important output of the flywheel simulation is the systems total power draw. This output and others is used in the post-processing where resulting electric power draw and diesel generator set fuel consumption and emissions are calculated and plots of this data are generated. Also a simple animation of the resulting data is generated (figure 8), which is useful as a quick check for any obvious errors in the simulation output.

RESULTS

Verification

To verify the accuracy of the simulation, its results were first compared to the official HITEC AHC-1000 specifications. Figure 8 shows the simulated maximum hoist capacity overlaid on a scan of the official specifications. The results agree well with the specifications.

Figure 10 shows the heave compensation limits calculated through simulation, overlaid onto the scanned AHC 1000® plot. The areas where the results diverge slightly from the specifications can be attributed to differing friction factors, motor specifications and control parameters. Since not every parameter was given in the AHC-1000® specifications, some had to be assumed or calculated. It is likely that some of these parameters do not exactly match those used to calculate the official specifications, causing the discrepancies seen in figure 10.

Power usage

The simulation was run for several different limit cases for both the original HITEC AHC-1000 and direct drive configurations to ensure that the system performs adequately under different conditions.

The overall power draw profile for the drawworks system is significantly improved when the flywheel system is implemented (table 5, figure 11). The high peaks (which coincide with the downward heaving movement) and deep valleys (upward heave) are fully buffered by the stored energy in the flywheels and the resulting power draw profile shows only minor fluctuations. The effect of the flywheel system's start-up transient can be seen in figure 11, which plots the total external power draw of the drawworks (with (red) and without (green) the flywheel system enabled). Initially, the flywheel's precharge is supplying most of the power to the drawworks, while external power draw slowly picks up. Table 5 shows that the peak load is reduced by a factor of 10 or more in most cases while average power draw is reduced approximately by a factor of 4. At low hook loads the effect of the lower rotational inertia of the direct drive system can be seen in the load profile, but at higher hook loads this effect is overshadowed by the power demanded by the linear heaving motion.

Figure 12 shows the range of operation for a single flywheel motor during heave compensation operation. All six motors operate in a paralleled configuration. As the flywheel accelerates and decelerates as a result of power transfer, the driving motor's state traces out an ellipse in the speed-torque (and speed-power) plot. Depending on conditions the ellipse changes shape. A tall and narrow ellipse means a small amount of energy is being transferred, but at a high rate (high power). Conversely, in high load or high wave amplitude situations, large amounts of energy are transferred making for a wide ellipse.

Generator requirements

The drastic reduction in peak power draw means that power production capacity can be reduced, for example, by putting generator sets offline. As a hypothetical example, we assume that the drawworks is powered by a number of CAT 3516 1280 KW diesel generator sets. During active heave compensation operations in 4 meter peak-to-peak waves with a period of 18 seconds and a 900 ton hook load (the bottom plot in figure 11) 6 of these gensets would normally need to be online. With the energy storage system in use only a single genset needs to be online due to the greatly reduced peak power draw.

Figure 13 shows the fuel efficiency curves of these generator sets superimposed on histogram plots of the draw-

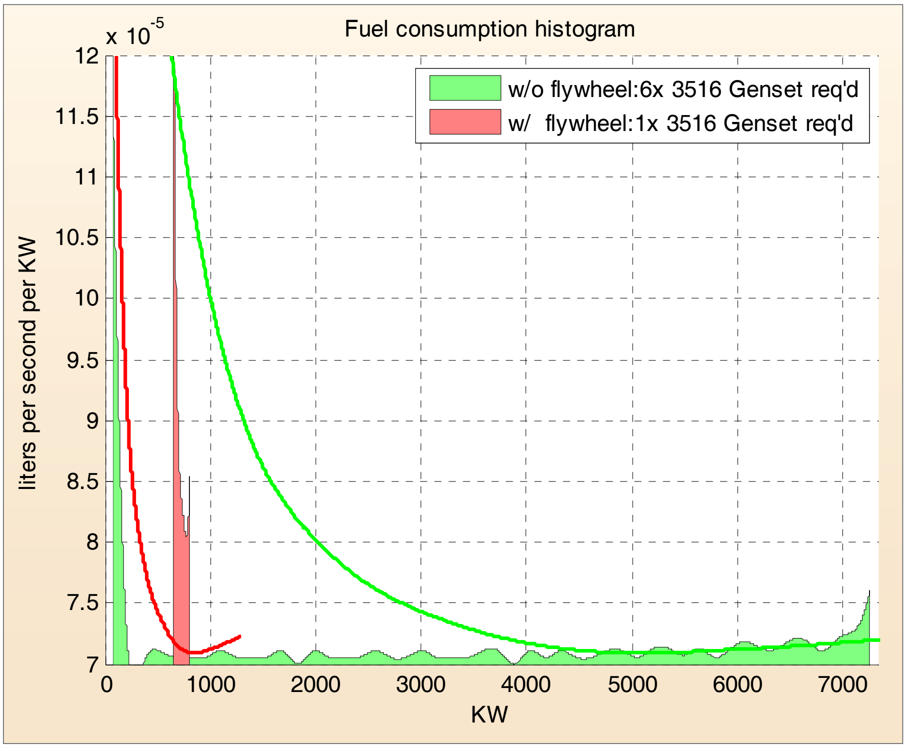


Figure 13: Fuel consumption histogram.

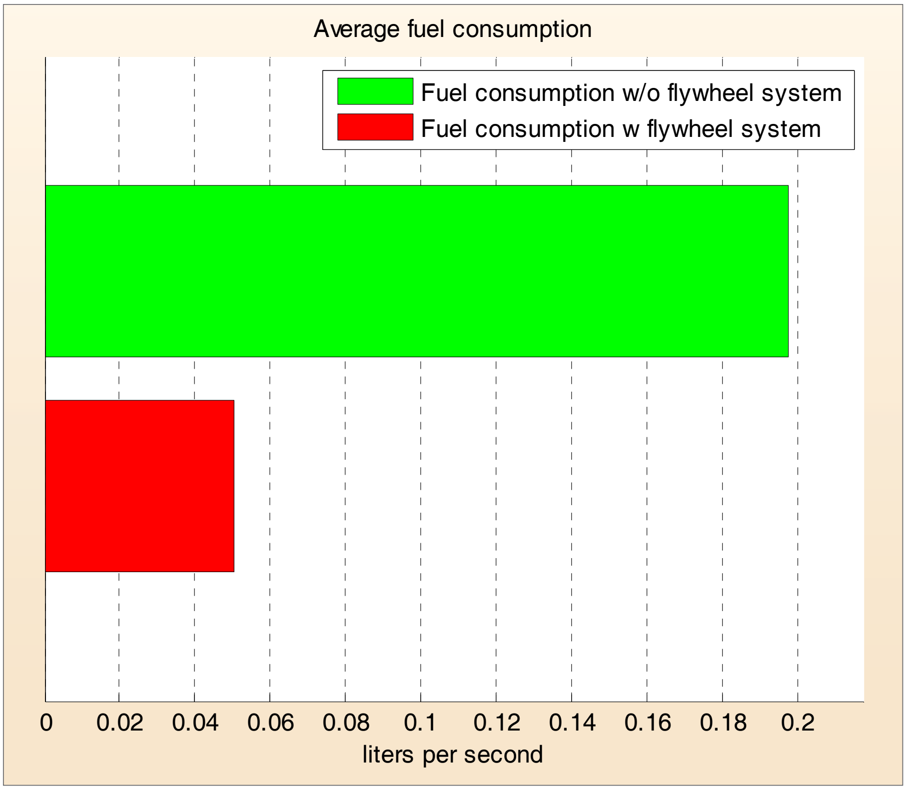
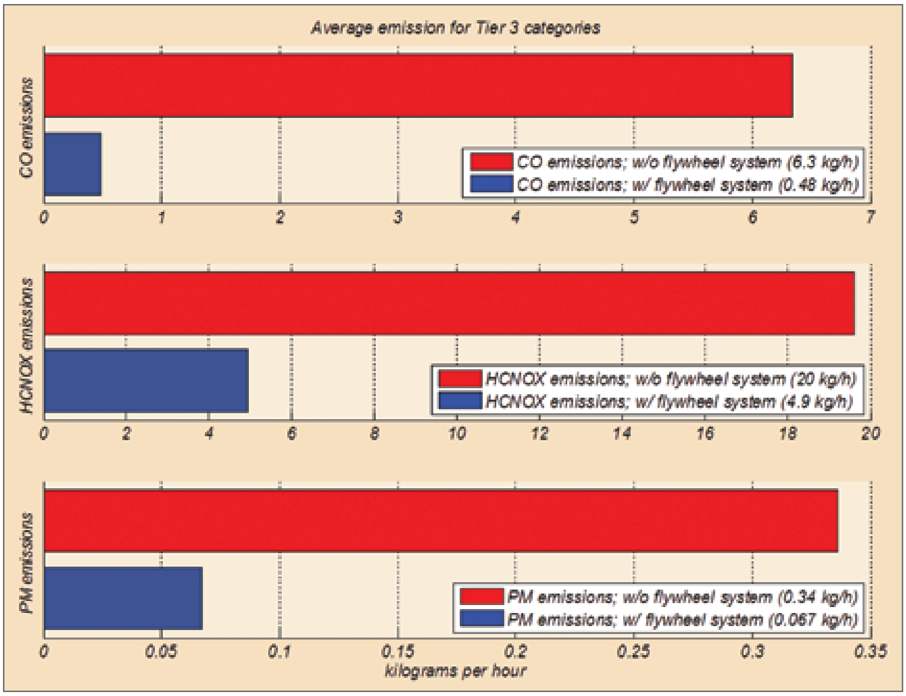


Figure 14: Average fuel consumption.



Hydrocarbon and NoX emissions were reduced by the same amount as the fuel savings (75%) while carbon monoxide and particulate matter were reduced by 92% and 80%, respectively, due to the optimized operating range of the genset engines.

works' power draw (with and without the flywheel system). The efficiency curves show that a generator set is most efficient when operating near its maximum power rating. The histograms are a measurement of the relative amount of time that the drawworks is demanding a certain amount of power. When unaided by the flywheel device large amounts of power (up to 7500 KW) are drawn for short periods of time, but the overall power requirement is very low. With the flywheel-equipped drawworks the power usage is much more consistent (approximately 700 KW). The consistent power draw and the reduced average load results in a 75% lower fuel consumption with only a single genset online instead of the usual six.

The tier 3 emissions for both situations were calculated as well. Hydrocarbon and Nox emissions are reduced by the same amount as the fuel savings, while CO and particulate emissions are reduced by even greater amounts (92% and 80%, respectively), due to the optimized operating range of the genset engine.

CONCLUSION

This simulation analyzes the anticipated performance of a flywheel system in a heave-compensating drawworks. The drawworks and flywheel simulation were modeled and work well. Average power demand during heave compensating operations is greatly reduced and stabilized eliminating the large load peaks inherent in a drawworks-based heave compensation system. The results show a very stable and much-reduced power demand potentially reducing operating costs, capital and environmental impact.

The authors would like to note that since these simulations and article was written, a newer, much lower inertia motor has been developed and currently being proto-typed, therefore; much better results are achieved.

REFERENCES

Munson, B.R., Young, D.F., Okiishi, Th. H. (1990). Fundamentals of Fluid Mechanics. New York: John Wiley & Sons, Inc.

Dorfman, L.A. (1963). Hydrodynamic Resistance and the Heat Loss of Rotating Solids. Edinburgh: Oliver & Boyd.