

Torsional Vibration Analysis by Bondgraph Modelling.

A practical approach

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Synopsis

The traditional modeling approach in engineering is mathematical, but thanks to the advances in computer technology it is now possible for the practical engineer to model physical systems in a more “user friendly” way.

This paper will demonstrate the power and simplicity of Bond Graphs in modelling technique.

Bond Graph modelling was originally developed in the late 1950s by the late Professor Henry M. Paynter of MIT.

Professor Paynter acted well before his time as the main advantage of his creation, other than the modeling insight that it provides and the ability of effectively dealing with mechatronics, came into fruition only with the recent advent of modern computer technology and the tools derived as a result of it, including symbolic manipulation, MATLAB and SIMULINK and the simulation package 20sim, see: www.20sim.com, which allows direct input of the Bond Graph.

Keywords: Electro-mechanical vibrations, Bondgraph modelling, the swing equation

1 Introduction

This paper will handle a practical modelling approach of parallel running generator sets as used in ship’s system for power generation.

A practical generator model for use in torsional vibration analysis is not a generator model modelled by using the matrices of Gabriel Kron, or the DQ transformation of Park but by regarding the generator as a torsional spring between the prime mover and the electrical grid, this is in line with the lumped mass-spring system used for torsional vibration analysis of a diesel drive line. The use of either finite element models or coupled finite element models are not considered as practical systems for a torsional vibration analysis.

Bond Graphs have the advantage of being interdisciplinary and are used for structured modeling of energy and power transfer processes, including mechanical, electrical, hydraulic and thermal processes and their combinations.

In the author’s view using bond graph is the way for a practical engineer to model a system.

But unlike block diagrams, bond graphs are derived directly from the physical system model, rather than through the equations of the system, and allows a more direct interface in tune with the engineer’s thinking.

Bond graphs, see reference [40] for an introduction, are a domain-independent graphical description of dynamic behavior of physical systems. This means that systems from different domains (cf. electrical, mechanical, hydraulic, acoustical, thermodynamic, material) are described in the same way. The basis is that bond graphs are based on energy and energy exchange. This energy flow can be described as the product of 2 variables (effort and flow), letting a bond be conceived as a bilateral signal connection. Analogies between domains are *more* than just equations being analogous: the used *physical concepts* are analogous.

Analogies among physical systems:

	Electrical	Mechanical translation	Mechanical rotation	Hydraulic	Thermal
Flow	current	velocity	Angular velocity	volume	Heat flow
Effort	voltage	force	torque	pressure	temperature
Power	power	power	power	power	Power · temperature
Inductive element	coil	inertia	Moment of inertia	Inertia of fluid	-
Capacitive element	capacitor	spring	Torsional spring	Tank, compressibility	Heat storage
Resistive element	resistance	friction	friction	friction	Thermal resistance
Transformer	transformer	lever	gears	transducer	-
Gyrator	Gyrator	-	gyro	-	-

Bond-graph modelling is a powerful tool for modelling engineering systems, especially when different physical domains are involved. Furthermore, bond-graph submodels can be re-used *elegantly*, because bond-graph models are *non-causal*. The submodels can be seen as *objects*; bond-graph modelling is a form of *object-oriented* physical systems modelling.

Bond graphs are labelled and directed graphs, in which the vertices represent submodels and the edges represent an ideal energy connection between power ports. The *vertices* are idealized descriptions of physical phenomena: they are *concepts*, denoting the relevant (i.e. dominant and interesting) aspects of the dynamic behavior of the system. It can be bond graphs itself, thus allowing hierarchical models, or it can be a set of equations in the variables of the ports (two at each port). The *edges* are called *bonds*. They denote point-to-point connections between submodel ports. When preparing for simulation, the bonds are embodied as two-signal connections with opposite directions. Furthermore, a bond has a power direction and a computational causality direction. Proper assigning the power direction resolves the sign-placing problem when connecting submodels structures. The internals of the submodels give preferences to the computational direction of the bonds to be connected. The eventually assigned computational causality dictates which port variable will be computed as a result (output) and consequently, the other port variable will be the cause (input). Therefore, it is necessary to rewrite equations if another computational form is specified then is needed. Since bond graphs can be mixed with block-diagram parts, bond-graph submodels can have power ports, signal inputs and signal outputs as their interfacing elements. Furthermore, aspects like the physical domain of a bond (energy flow) can be used to support the modelling process.

The concept of bond graphs was originated by Paynter (1961). The idea was further developed by Karnopp and Rosenberg in their textbooks (1968, 1975, 1983, 1990), such that it could be used in practice (Thoma, 1975; Van Dixhoorn, 1982). By means of the formulation by Breedveld (1984, 1985) of a framework based on thermodynamics, bond-graph model description evolved to a systems theory.

For a more thoroughly analysis of bond graphs the author refer to the extensive amount of literature available see among others the reference list: [1], [4], [5], [6], [9], [10], [12], [13], [14], [15], [16], [17], [19], [20]. Reference [40] and [42] gives a very good introduction in the use of bond graphs.

[41] is an example how to use bond graphs in electronics.

For Matlab users there is Bond graph add-on block library BG V2.1. see www.mathworks.com

As an introduction to the synchronous generator model the swing equation will be handled for a justification to consider the generator as a mass coupled with a torsional spring to the grid inertia. Another reason for a detailed explanation of the swing equation is because a literature review revealed that this important property of a synchronous generator is not get the required attendance in the author's view.

Another reason for this paper there is not much literature handling the weak grids on board of ship's and the author's involvement solving a power oscillation problem in a ship's power plant, see the oscillation in the figures 11, 12 and 13.

Also a practical diesel model with the dynamics of the governor and the fuel pump delay for TVA purpose will be presented. The model parameters as internal friction and damping are validated from torque measurements on a 12 cylinder generator set engine but because of a NDA agreement the author was not allowed to use the data of that interesting project.

The flexible coupling between the prime mover and the generator will also reviewed, it appears to be a highly nonlinear piece of equipment.

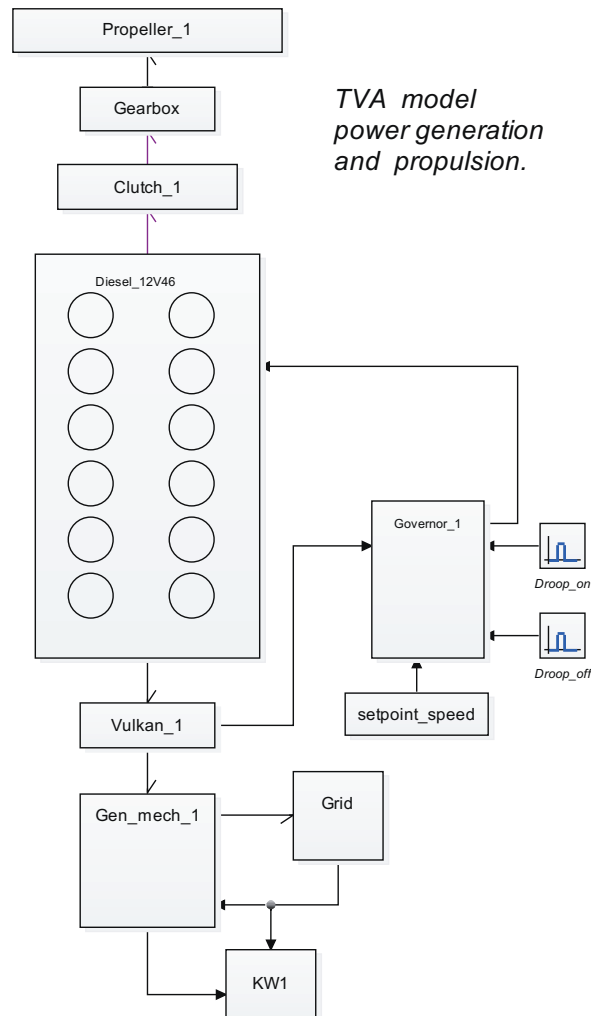


Figure 1. 20-sim iconic simulation model.

The system involved is the propulsion and power generation of a dredging vessel consisting of a PS and SB system. Each system, see figure 1, consisting of a Wärtsilä 12V46 engine with on the flywheel side a flexible coupling with clutch, propulsion gearbox and the propeller. On the PTO side a flexible coupling and the 6.6 kV 11350 KVA generator.

Author's Biographies

Tjeerd Heeringa currently working as a consultant in control engineering after his retirement. Before his retirement working in several positions as a senior superintendent of Boskalis dredging

2 Introduction to the swing equation

As mentioned in the introduction in most text books this subject is not very clearly explained therefore we will pay some more attention on this important subject.

Almost all energy consumed by various loads in an electric power system is produced by synchronous machines, or more correctly, the conversion from the primary energy sources, like water energy, nuclear energy, diesel engines, or chemical energy, to electrical energy is done in synchronous machines

2.1 Introduction

The tendency of a power system to develop restoring forces equal to or greater than the disturbing forces to maintain the state of equilibrium is known as stability

If the forces tending to hold the state of equilibrium with one other are sufficient to overcome the disturbing forces, the system is said to remain stable (to stay in synchronism).

2.2 Oscillations

The combination of an inertial and an elastic restoring torque makes possible the generation of oscillations by reason of the interchange of energy between the two.

Oscillatory conditions may arise in several ways, especially where there are forced vibrations near to that of the natural frequency.

Oscillation amplitudes may substantially increase by mechanical torque pulsating at low frequency, as in a generator driven by a diesel engine.

In ship propulsion with direct electric drive, fluctuating load torque may be impressed by the screws.

Rope elasticity in the vertical or inclined haulage of mine winders may resonate with the load mass at a frequency near to the natural frequency of the machine or of its closed-loop control drives.

Induction and synchronous machines respond in different ways to a change of load.

The induction machine responds by a change of slip, and therefore of speed, and oscillations are of purely mechanical origin

The synchronous machines, when operating in parallel with other synchronous machines or on supply bus bars, respond by adjusting its torque angle without change in mean speed: any divergence between actual and equilibrium angles calls into play a synchronizing torque proportional to a function of the angle of divergence and so adding to the system an elastic torque of electrical origin.

More general the synchronous machine as an energy converter and the coupling between the mechanical drive and the electrical grid, behaves like a torsional spring with damping.

A case of interest is that when the load change is cyclic.

2.3 Cyclic disturbing torque.

Cyclic irregularity in the driving torque of a generator (as when it is driven by a diesel engine), or in the load torque of a motor (as for an air compressor drive), or in a control system (e.g. governor 'hunting') excite small fluctuations in speed.

See reference [35].

With induction motors the speed variations are constrained by the system inertia and the inherent damping by I^2R loss is large.

With synchronous machines the synchronizing torque adds electrical elasticity and provides a combination with a natural oscillation frequency. If this is close to the frequency of the cyclic disturbance and if the damping is small the conditions may be such as to result in considerable phase-swing.

2.4 The generator.

The synchronous generator can be described by various mathematical models which have arisen from different methods of approach in the course of the development of the theory of electrical machines.

Depending on the task, the following methods are most frequently chosen;

Park's system of equations, see reference [27].

Model based on the rotating-field theory,
 Treatment as an asynchronous machine with distributed damper winding,
 Representation as a synchronous machine with distributed damper winding.
 The above mentioned descriptions are not easily suitable for use in torsional vibration calculations.
 We want to have a generator model which is in line with the used mass-rotating spring system or lumped mass-spring system of the diesel.
 Looking at energy level a generator can be viewed as rotating transformer, it transforms rotating mechanical energy by means of a rotating magnetic field in electrical energy.

Under normal operating conditions, the relative position of the rotor axis and the resultant rotor magnetic field is fixed.

The angle between the two is known as the power angle or torque angle. During any disturbance, for instance the unequal angle of rotation of the driving shaft of a diesel engine, the rotor will decelerate or accelerate with respect to the synchronously rotating air gap mmf, and a relative motion begins.

The equation describing this relative motion is known as the swing equation.

In addition to the state-space representation and model analysis, we will use the block diagram representation and torque-angle relationship to analyze the system stability.

While this approach is not suited for a detailed study of large systems it is useful in gaining physical insight and it fits in the mass-spring system approach of the mechanical torsional vibration calculations.

And it fits very well in system modeling by Bondgraph's which is an elegant tool for total system modeling.

A diesel generator set is a mechanical-electrical system and two or more generator set's running parallel form a mechanical – electrical system, magnetically coupled.

As an introduction to the theory of the generator rotor dynamics, "the swing equation", a brief description of electrical power generating systems will be given.

2.5 Electrical power generating systems.

1) A turbine driven electrical power generating system

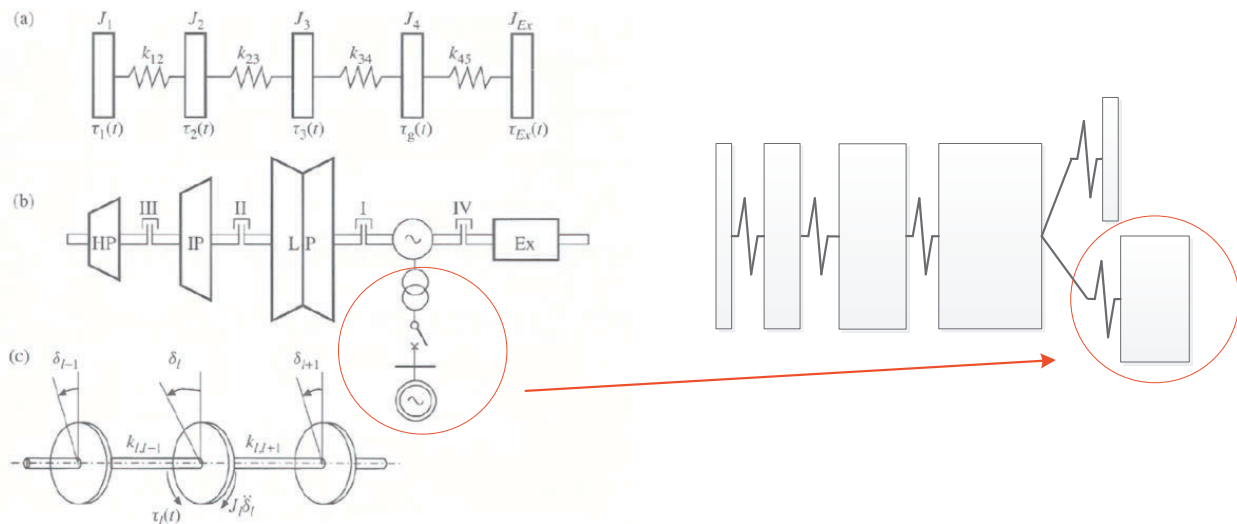


Figure 1. Turbo generator set. (copy from ref.[7] page 142, figure 5.1)

Generating unit as an oscillating system: (a) division of the rotor mass into individual sections; (b) schematic diagram; (c) torsional displacements. When the circuit breaker (in the red circle) is closed the right hand side of Figure 1 is valid, the grid inertia is coupled by a magnetic spring to the rotor inertia.

- HP = high pressure turbine section
- IP = intermediate pressure turbine section
- LP = low pressure turbine section
- G = Generator

Ex = rotating exciter
 k = stiffness shaft or coupling
 δ = angular displacement of a mass
 τ = external torque acting on a mass
 J = moment of inertia of individual sections

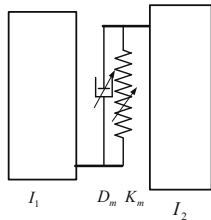


Figure 2 Turbine unit as an oscillating system when coupled to the electrical system.

As the stiffness of the shafts and couplings is much higher than the stiffness of the “magnetic spring” the system simplifies to a two masses one spring system.

See figure 2:

I_1 = combined inertia turbo generator

I_2 = inertia electrical system

D_m = magnetic damping turbo generator

K_m = magnetic elasticity turbo generator

Both magnetic parameters D_m and K_m are dependent of the generator load.

Simulations revealed that the damping is also influenced by the grid inertia, in today’s ship’s systems with VFD as propulsion drive the inertia of the propulsion motor is decoupled from the ship’s grid.

2) A diesel driven electrical power generating system.

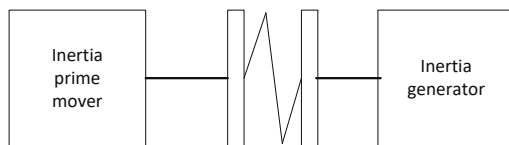


Figure 3. Diesel driven generator set.

The prime mover in this system is a diesel engine and because the unequal angle of rotation of a diesel engine (see reference [10]) a flexible coupling is needed between the diesel and the synchronous generator.

As the turbine has an equal angle of rotation the coupling between the turbine and the generator is a stiff one so, the inertia of the turbine can be combined with the inertia of the generator.

With a diesel as a prime mover for the generator this simplification is not allowed.

The simplification in this case is that the shaft stiffness is much higher than then stiffness of the coupling so the system of Fig. 3 is a two masses one spring system.

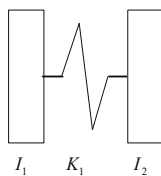


Figure 4 Generating unit from Fig. 3 as an oscillating system.

I_1 = inertia crankshaft + flywheel

I_2 = inertia generator

K_1 = stiffness flexible coupling

According reference [2] the resonance frequency of this system is:

$$\omega_0 = \sqrt{K \cdot \frac{I_1 + I_2}{I_1 \cdot I_2}} \quad (1)$$

When the diesel crankshaft has a torsional vibration damper on the non-driven side the simplified system is like this:

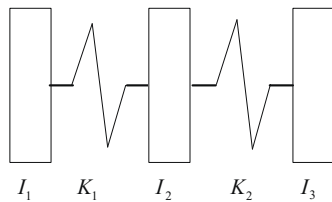


Figure 5. Generating unit as an oscillating system:

- I_1 = damper
- I_2 = inertia crankshaft + flywheel
- I_3 = inertia generator
- K_1 = stiffness torsional damper
- K_2 = stiffness flexible coupling

The resonance frequencies of this system (Figure 5) can still be calculated by a pocket calculator again according reference [2], see the formulae below.

This can be useful for a quick check of the simulation output.

$$\omega_{0,2} = \sqrt{0.5 \cdot A \mp 0.5\sqrt{A^2 - 4B}} \quad (2)$$

$$A = \frac{K_1}{I_1} + \frac{K_2}{I_3} + \frac{K_1 + K_2}{I_2} \quad B = \frac{K_1 \times K_2}{I_1 \times I_2 \times I_3} \cdot (I_1 + I_2 + I_3)$$

When the diesel generator set is coupled to the electrical system than the oscillating system becomes complex as depicted in Figure 6.

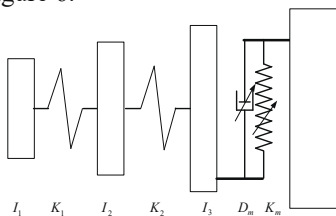


Figure 6 The rotor inertia I_3 is coupled magnetically to the “inertia” of the electrical system.

- I_1 = damper on the NDE of the crankshaft
- I_2 = inertia crankshaft + flywheel
- I_3 = inertia generator
- K_1 = stiffness torsional damper
- K_2 = stiffness flexible coupling
- K_m = stiffness magnetic “spring”
- D_m = magnetic damping

3 The Swing equation

The equation governing the motion of the rotor of a synchronous machine is based on the elementary principle in dynamics which states that accelerating torque is the product of the momentum of inertia of the rotor times its angular acceleration.

In the MKS system of units, this equation can be written for the synchronous generator in the form:

$$J \cdot \frac{d^2\omega_m}{dt^2} + D_d \cdot \omega_m = T_a = T_m - T_e \quad [\text{N.m}] \quad (3)$$

where the symbols have the following meanings:

- J the total moment of inertia of the rotor (kg.m²)
- D_d the damping-torque coefficient (Nms)
- ω_m the rotor shaft velocity (mechanical rad/s.).
- ω_s the synchronous speed.
- t time, in seconds.
- T_m the mechanical or shaft torque supplied by the prime mover, in Nm.
- T_e the net electrical or electromagnetic torque, in Nm.
- T_a the net accelerating torque, in Nm.

In the steady-state the rotor angular speed is synchronous speed ω_{sm} whilst the torque T_T is equal to the sum of the electromagnetic torque T_e and the damping (or rotational loss) torque $D_d \cdot \omega_{sm}$

$$T_t = T_e + D_d \cdot \omega_{sm} \quad \text{or} \quad T_m = T_t - D_d \cdot \omega_{sm} = T_e \quad (4)$$

where T_m is the net mechanical shaft torque i.e. the diesel torque less the rotational losses at $\omega_m = \omega_{sm}$. It is this torque that is converted into electromagnetic torque. If, due to some disturbance, $T_m > T_e$ then it accelerates, if

$T_m < T_e$ then it decelerates.

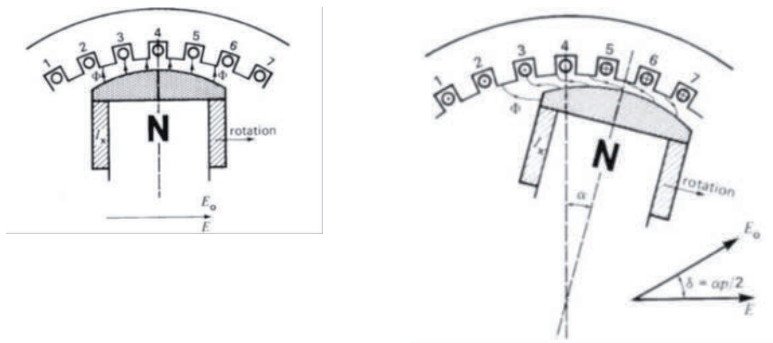


Figure 7. Left picture rotor with field in no load and left rotor with field in load condition.

It depicts the “magnetic spring” of the rotor field in a synchronous machine.

Left picture: rotor position or speed ω_m in synchronism with stator field ω_{sm}

Right picture: rotor position: $\omega_m + \Delta\omega_m = \omega_{sm} + \frac{d\delta_m}{dt}$ (5)

where δ_m is the rotor angle expressed in mechanical radians and $\Delta\omega_m = \frac{d\delta_m}{dt}$ is the speed deviation in mechanical radians per second.

Figure 7 is a practical abstract of chapter 5 of ref. [1] where the load angle is mathematically explained with the help of the phasor/vector diagram of the synchronous machine.

Substituting (5) in (3) gives:

$$J \cdot \frac{d^2\delta_m}{dt^2} + D_d \left(\omega_{sm} + \frac{d\delta_m}{dt} \right) = T_t - T_e \quad \text{or} \quad J \cdot \frac{d^2\delta_m}{dt^2} + D_d \frac{d\delta_m}{dt} = T_m - T_e \quad (6)$$

Multiplying (6) by the rotor synchronous speed ω_{sm} , results in:

$$J\omega_{sm} \cdot \frac{d^2\delta_m}{dt^2} + \omega_{sm}D_d \frac{d\delta_m}{dt} = \omega_{sm}T_m - \omega_{sm}T_e \quad (7)$$

Since angular velocity times torque is equal to power, we write the equation (7) in terms of power:

$$J\omega_{sm} \cdot \frac{d^2\delta_m}{dt^2} + \omega_{sm}D_d \frac{d\delta_m}{dt} = \frac{\omega_{sm}}{\omega_m} T_m - \frac{\omega_{sm}}{\omega_m} T_e \quad (7)$$

Where P_m is the net shaft power input to the generator and P_e is the electrical air-gap power, both expressed in watts.

During a disturbance the speed of a synchronous machine is normally quite close to synchronous speed so that $\omega_m \approx \omega_{sm}$ and equation (7) becomes:

$$J\omega_{sm} \cdot \frac{d^2\delta_m}{dt^2} + \omega_{sm}D_d \frac{d\delta_m}{dt} = P_m - P_e \quad (8)$$

The quantity $J\omega_{sm}$ is called the inertia constant and is denoted by M_m allows equation (8) to be written as:

$$M_m \cdot \frac{d^2\delta_m}{dt^2} = P_m - P_e - D_m \frac{d\delta_m}{dt} \quad (9)$$

Where $D_m = \omega_{sm}D_d$ is the damping coefficient.

Equation (9) is called the swing equation.

It is common practice to express the angular momentum of the rotor in terms of a normalized inertia constant when all generators of a particular type will have similar inertia values regardless of their rating. The inertia constant is given the symbol H defined as the stored kinetic energy in megajoules at synchronous speed divided by the machine rating S_n in MVA so that

$$H = \frac{0.5J\omega_{sm}^2}{S_n} \quad \text{and} \quad M_m = \frac{2HS_n}{\omega_{sm}} \quad (10)$$

The units of H are seconds. In effect H simply quantifies the kinetic energy of the rotor at synchronous speed in terms of the number of seconds it would take the generator to provide an equivalent amount of electrical energy when operating at a power output equal to its MVA rating. Also used is the symbol T_m as the mechanical time constant where:

$$T_m = \frac{J\omega_{sm}^2}{S_n} = 2H \quad \text{and} \quad M_m = \frac{T_m S_n}{\omega_{sm}} \quad (11)$$

Again the units are seconds but the physical interpretation is different. In this case, if the generator is at rest and a mechanical torque equal to $\frac{S_n}{\omega_{sm}}$ is suddenly applied to the turbine shaft, the rotor will accelerate, its velocity will increase linearly and it will take T_m seconds to reach synchronous speed ω_{sm} .

The load angle is normally expressed in electrical radians and there mechanical equivalent is;

$$\delta_m = \frac{\delta_e}{p/2} \quad \text{and} \quad \omega_{sm} = \frac{\omega_s}{p/2} \quad (12)$$

Where p is the number of poles.

Substituting equation (12) into equation (9) allows the swing equation to be written as:

$$\frac{2HS_n}{\omega_s} \cdot \frac{d^2\delta}{dt^2} + D \frac{d\delta}{dt} = P_m - P_e \quad \text{or} \quad \frac{T_m S_n}{\omega_s} \cdot \frac{d^2\delta}{dt^2} + D \frac{d\delta}{dt} = P_m - P_e \quad (13)$$

$$\text{When we neglect the losses : } \frac{2HS_n}{\omega_s} \cdot \frac{d^2\delta}{dt^2} = P_m - P_e \quad (14)$$

We can also write:

$$\frac{HS_n}{\pi \cdot f} \cdot \frac{d^2\delta}{dt^2} = P_m - P_e \quad \text{or} \quad M \cdot \frac{d^2\delta}{dt^2} = P_m - P_e \quad (15)$$

This equation, is called the swing equation and it describes the rotor dynamics for a synchronous machine (generating/motoring).

It is a second order differential equation where the damping term (proportional to $d\delta/dt$ is absent because of the assumption of a lossless machine).

$$\text{The electrical power equation in terms of } P_{e_{max}} \text{ is: } P_e = P_m - P_{e_{max}} \cdot \sin\delta_e \quad (16)$$

See for an extensive description reference [1], [3] and [7].

Since, the mechanical power input P_m and the maximum power output of the generator $P_{e_{max}}$ are known for a given system topology and load, we can find the rotor angle δ from (16) as:

$$\delta_e = \sin^{-1}\left(\frac{P_m}{P_{e_{max}}}\right) \text{ or } \pi - \sin^{-1}\left(\frac{P_m}{P_{e_{max}}}\right)$$

It can be observed from (15) that the rotor angle δ_e has two solutions at which $P_m = P_{e_{max}} \cdot \sin\delta_e$. Now, let us label

$$\delta_{e_0} = \sin^{-1}\left(\frac{P_m}{P_{e_{max}}}\right) \text{ and } \delta_{e_{max}} = \pi - \sin^{-1}\left(\frac{P_m}{P_{e_{max}}}\right)$$

There is a very important implication of the two solutions $(\delta_{e_0}, \delta_{e_{max}})$ of equation (15) on the stability of the system.

This can be clearly understood from what is called as “swing curve” or $P - \delta_{e_0}$ curve.

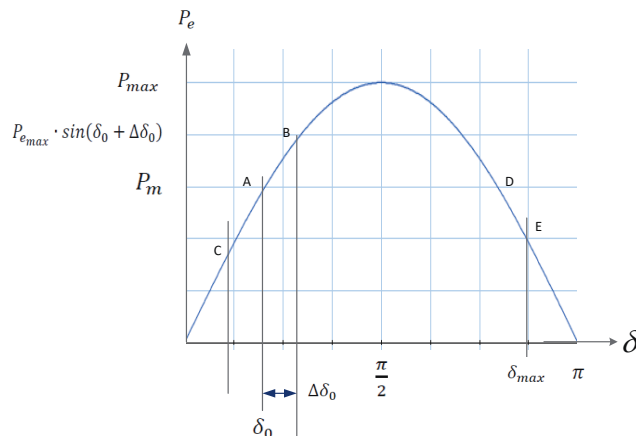


Figure 8. Swing curve or $P - \delta_{e_0}$.

The swing curve is depicted in Figure 8. The swing curve shows the plot of electrical power output of the generator P_e with respect to the rotor angle δ_e .

As can be observed from Figure 8 the curve is a sine curve with δ_e varying from 0 to π .

The maximum power output of the generator $P_{e_{max}}$ occurs at an angle $\delta_e = \pi/2$.

On the same curve the input mechanical power P_m can also be represented.

Since P_m is assumed to be constant and does not vary with respect to δ_e , a straight line is drawn which cuts the $P - \delta_{e_0}$ curve at points A and D. It can be seen from Figure 10 that at points A and D the mechanical power input P_m is equal to P_e . The rotor angles at point A and D are δ_{e_0} and $\delta_{e_{max}}$, the solutions of equation (15).

Let us try to understand which of the solutions $(\delta_{e_0}, \delta_{e_{max}})$ leads to a stable operation.

Let us take first point A. If we perturb the rotor angle δ_0 by a small positive angle $\Delta\delta_0$, so that the new operating point is at B, then electrical power will also increase to $P_{e_{max}} \cdot \sin(\delta_0 + \Delta\delta_0)$.

So after perturbation $P_m < P_{e_{max}} \cdot \sin(\delta_0 + \Delta\delta_0)$, for a positive value of δ_0 . Now the output electrical power will be more than the input mechanical power and hence the rotor starts decelerating due to which the angle δ will be pulled back to the point A. But since the rotor has certain inertia it cannot stop at the point A and decelerate further due to which the angle δ moves to the point say C.

At point C $P_m > P_{e_{max}} \cdot \sin(\delta_0 + \Delta\delta_0)$ that is input mechanical power will be more than the electrical power output and hence the rotor starts accelerating. Again the rotor angle δ starts increasing and will reach point A but due to inertia it will again move to point B.

This phenomenon repeats itself indefinitely if there is no damping to the rotor oscillations. This can be analogously understood from the pendulum motion in vacuum.

If a pendulum is perturbed from its steady state position then it swings to one extreme point reverses its direction, passes through its steady state position, goes to the other extreme, and reverses and passes through the extreme, and this happens indefinitely as it is oscillating in vacuum and there is no air friction to stop the oscillations.

Now take the case of second operating point δ_{max} . Again if we perturb δ_{max} by a positive angle $\Delta\delta_{max}$ then the electrical power output will be $P_{e_{max}} \sin(\delta_0 + \Delta\delta_0)$ and the rotor starts accelerating due to which the angle will increase further and this will lead to further decrease in electrical output. Hence, for a small positive perturbation in the rotor angle at the operating point D leads to continuous increase in the speed of the rotor thereby leading to unstable operation of the generator.

This discussion leads to an important conclusion that out of the two operating point B is unstable for small disturbances. Hence, point A is called stable equilibrium point and operating point D is called unstable equilibrium point.

Similarly, δ_0 is a stable steady state rotor angle and δ_{max} is an unstable rotor angle.

From $P - \delta$ curve it can be observed that if we disturb the rotor angle from point A by a small angle $\Delta\delta$ then the rotor angle starts oscillating between points B and C. If observed closely it can be seen that $P - \delta$ curve falling between the points B and C is almost linear if $\Delta\delta$ is very small. Hence, if the rotor angle perturbation $\Delta\delta$ is very small then we can assume that the system behaves in a linear fashion around the operating point A between B and C. Mathematically this can be written by supposing the angle δ_0 is perturbed by an angle $\Delta\delta$ then we can express the swing equation as:

$$M \cdot \frac{d^2(\delta_0 + \Delta\delta)}{dt^2} = P_m - P_{e_{max}} \cdot \sin(\delta_0 + \Delta\delta) \quad (17)$$

using the formulae: $\sin(\alpha + \beta) = \sin\alpha \cdot \cos\beta + \cos\alpha \cdot \sin\beta$ we get:

$$M \cdot \frac{d^2\delta_0}{dt^2} + M \cdot \frac{d^2\Delta\delta}{dt^2} = P_m - P_{e_{max}} \cdot (\sin\delta_0 \cdot \cos\Delta\delta + \cos\delta_0 \cdot \sin\Delta\delta) \quad (18)$$

since $\Delta\delta$ is small, $\cos\Delta\delta \cong 1$ and $\sin\Delta\delta \cong \Delta\delta$ with $\Delta\delta = rad$, we have:

$$M \cdot \frac{d^2\delta_0}{dt^2} + M \cdot \frac{d^2\Delta\delta}{dt^2} = P_m - P_{e_{max}} \cdot \sin(\delta_0) - P_{e_{max}} \cdot \cos(\delta_0) \cdot \Delta\delta \quad (19)$$

Since at the initial operating point A,

$$M \cdot \frac{d^2\delta}{dt^2} = P_m - P_{e_{max}} \cdot \sin\delta = 0$$

$$M \cdot \frac{d^2\delta}{dt^2} = -P_{e_{max}} \cdot \cos\delta \cdot \Delta\delta \quad (20)$$

We can label, $P_s = P_{e_{max}} \cdot \cos\delta_0$ P_s is called as the synchronizing torque or power, as per units torque and power will be the same. P_s is the torque required to pull the rotor in to synchronization.

Substituting P_s in (20) and rewriting it we can get:

$$M \cdot \frac{d^2\delta}{dt^2} + P_s \cdot \cos\delta_0 = 0 \quad (21)$$

The solution of the above second order differential equation depends on the roots of the characteristic equation given by

$$s^2 = -M \cdot P_s \quad (22)$$

When P_s is negative, we have one root in the right-half s-plane, and the response is exponentially increasing and stability is lost. When P_s is positive, we have two roots on the $j - \omega$ axis, and the motion is oscillatory and undamped. The system is marginally stable with a natural frequency of oscillation given by:

$$\omega_n = \sqrt{M \cdot P_s} \quad (23)$$

In case of a synchronous generator, the rotor has damper windings due to which it acts as an induction motor during transients and this effect has to be considered while evaluating the stability of the system. This effect of damper winding is called as damping torque which depends on the rate of change of rotor angle.

$$P_d = D \frac{d\delta}{dt} \quad (24)$$

Where, D is the damping coefficient. After including damping torque, the swing equation (15) can be written as:

$$M \cdot \frac{d^2\delta}{dt^2} + D \cdot \frac{d\delta}{dt} + P_s \Delta\delta = 0 \quad (25)$$

4 Introduction to the modelling

From the mathematical exercises it is clear that a synchronous generator needs damping for a proper operation. Damping of the rotor motion by mechanical losses is small and can be neglected for all practical considerations. The main source of damping in a synchronous generator is provided by the damper, or amortisseur, windings see figure 9. The damper windings act in a similar way as the short circuited squirrel rotor windings in an induction motor. As the induced current in these windings only appears when $\omega \neq \omega_s$ it is proportional to $\Delta\omega = d\delta/dt$ and is referred to as the asynchronous torque or damping torque.

Machines with laminated salient poles require explicit damper windings for effective damping.

Again from reference [7] page 145:

Rigorous derivation of an expression for damping power is long and complicated but an approximate equation for the generator-infinite busbar is:

$$P_D = V_s^2 \left[\frac{x_d'' - x_d'' \cdot x_d'}{(x + x_d) \cdot x_d''} \cdot T_d'' \sin^2 \delta + \frac{x_q'' - x_q'' \cdot x_q'}{(x + x_q) \cdot x_q''} \cdot T_q'' \cos^2 \delta \right] \cdot \Delta\omega \quad (26)$$

The reader will agree that above equation is not very practical for use in TVA, the author likes to refer to a rotor drawing of a 11 MVA, 6.6 kV synchronous generator, see figure 9, which has shown to have sufficient damping for a “swing free” operation. See the measurements in figure 10.

It appears that it is not easy to get from the machine manufacturer a rotor drawing with the dimensions of the damper bars.

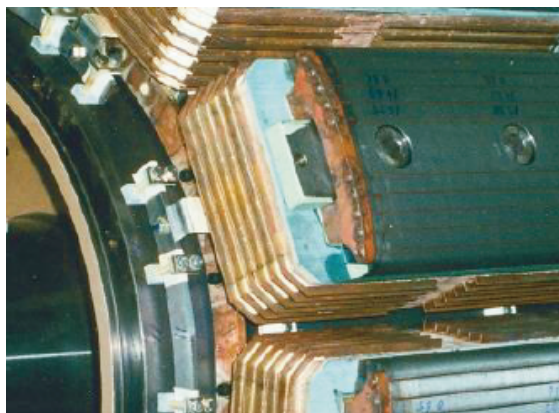
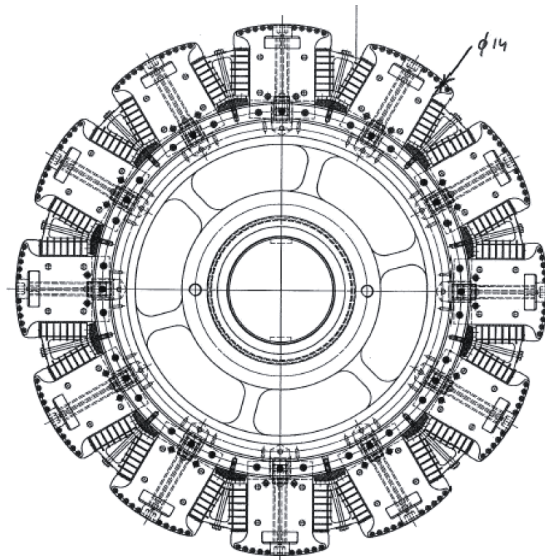


Figure 9 Salient pole rotor of a 11.35 MVA 6600 V 500 rpm generator. Drawing and pole picture

Each pole has 10 damper bars with a diameter of 14 mm.

Also the mentioned exercises will not give a model for use in torsional vibration analysis. As mentioned in the introduction a practical approach is to regard the synchronous generator as a torsional spring, the rotor inertia is coupled magnetically to the grid inertia and through the grid inertia magnetically coupled to another generator in case of parallel service.

So a ship's power plant with generator sets running parallel appears to be a complicated dynamic system and needs proper design and engineering to ensure stable parallel operation of the generator set's.

The magnetic stiffness can be derived from the load angle data at different loads, see the calculations at page 15.

4.1 Measurements showing a swing free parallel operation

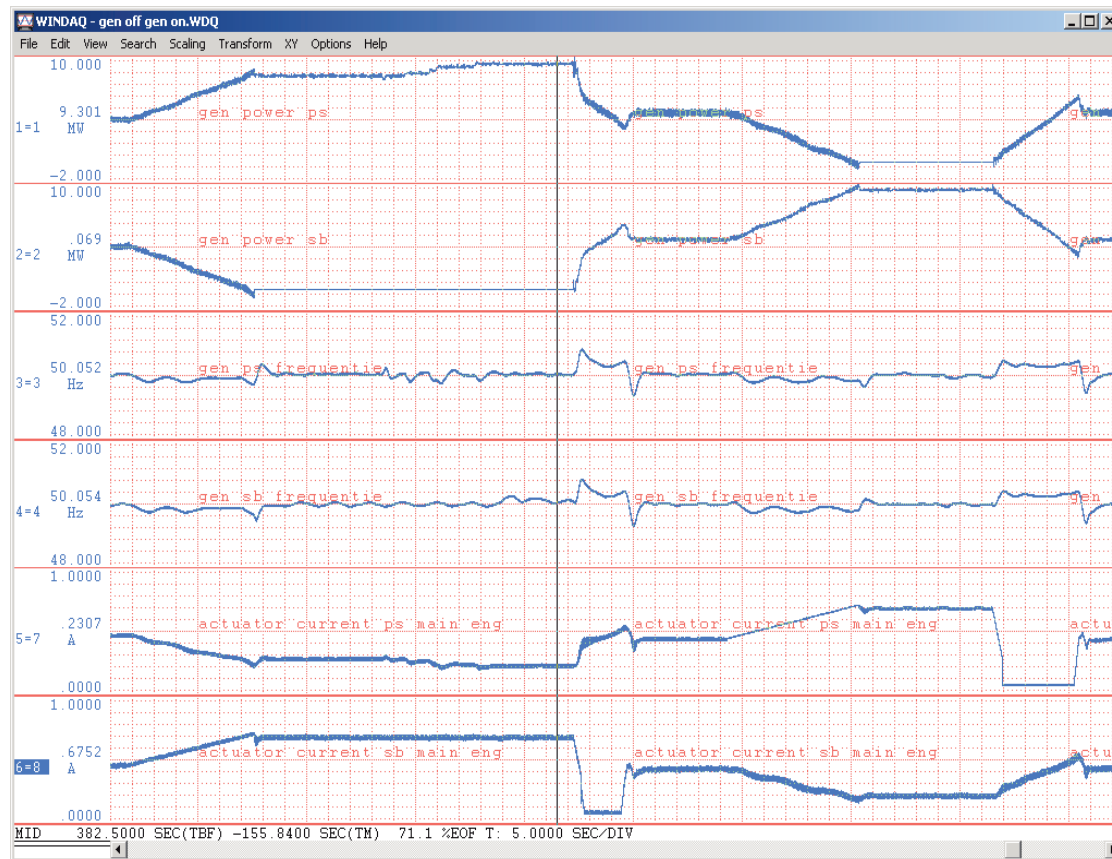


Figure 10. Measurements on a power and propulsion system of a dredging vessel.

The dredging vessel was equipped with two systems from figure 1 running electrically parallel with active electric load sharing. The diesels are supplied with RE Viking 25 governors.

- Channel 1: PS generator power
- Channel 2: SB generator power
- Channel 3 : PS generator frequency
- Channel 4 : SB generator frequency
- Channel 5: PS main engine governor actuator control current.
- Channel 6: SB main engine governor actuator control current.

The governors are reverse acting, minimum actuator current is maximal fuel rack, channel 5 and 6. The measurements start with SB and PS generators running parallel and both about 4 MW electrical load and off-loading the SB generator load to the PS generator, then CB opens and just after the vertical line the SB generator synchronizes with the PS generator and the active load sharing starts. Then the same procedure starts with the PS generator.

The measurements reveal that after closing the CB the system is stable there is no unwanted power swing after closing the circuit breaker.

The next page demonstrates what can go wrong when the design and the engineering of a ship's power system was not handled with the necessary care.

4.2 Undocumented measurements of a dynamic system with among other things insufficient damping

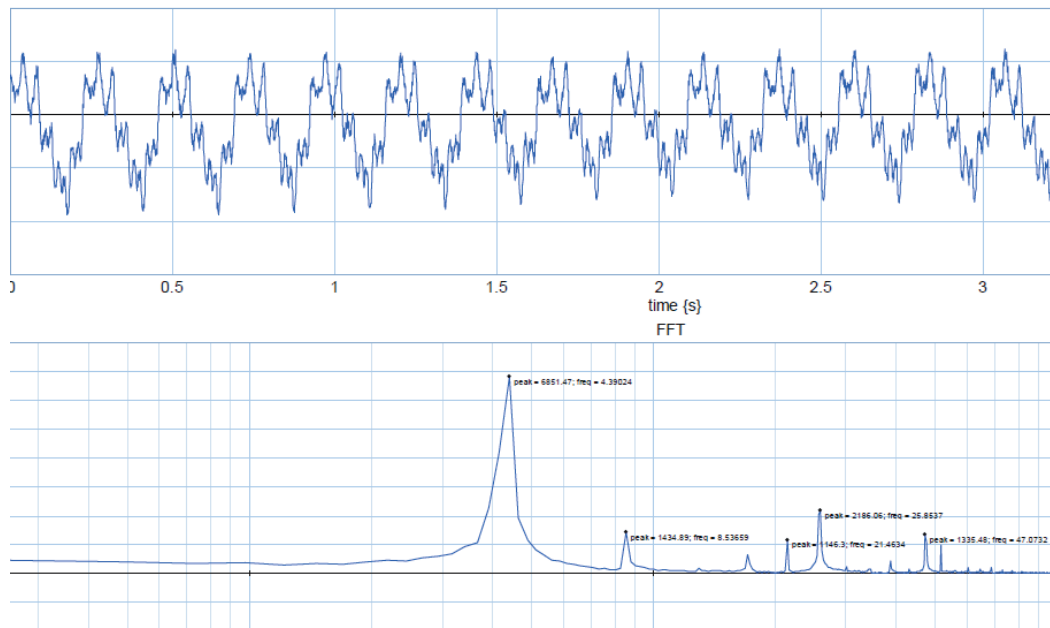


Figure 11. No load generator shaft torque

There was an offset in the torque measurement but not enough to bring the torque signal above the zero line, the mass spring system rotor mass, stiffness flexible coupling and diesel mass was almost in resonance. The TT value of the torque signal was about 30 kNm

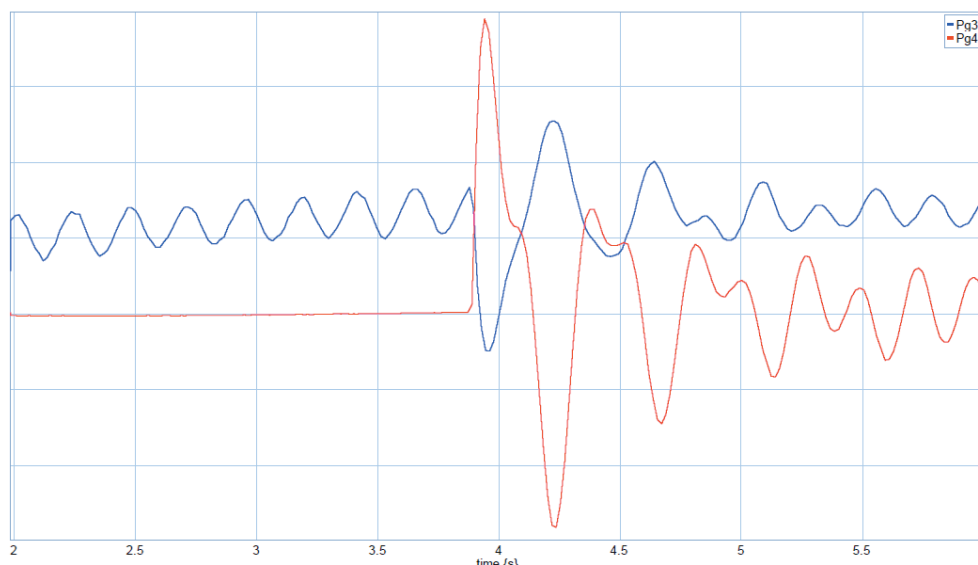


Figure 12. at $T = 3.8$ sec. the 2 generator sets are switched parallel .

The above measurement demonstrates among other things the insufficient damping in the synchronous generator. There was no rotor drawing available for comparison with the rotor in figure 9.

The power swing was about 3.5 MW (!).

The other remarkable fact was the power swing of one generator feeding the grid, the simulation model confirms this.

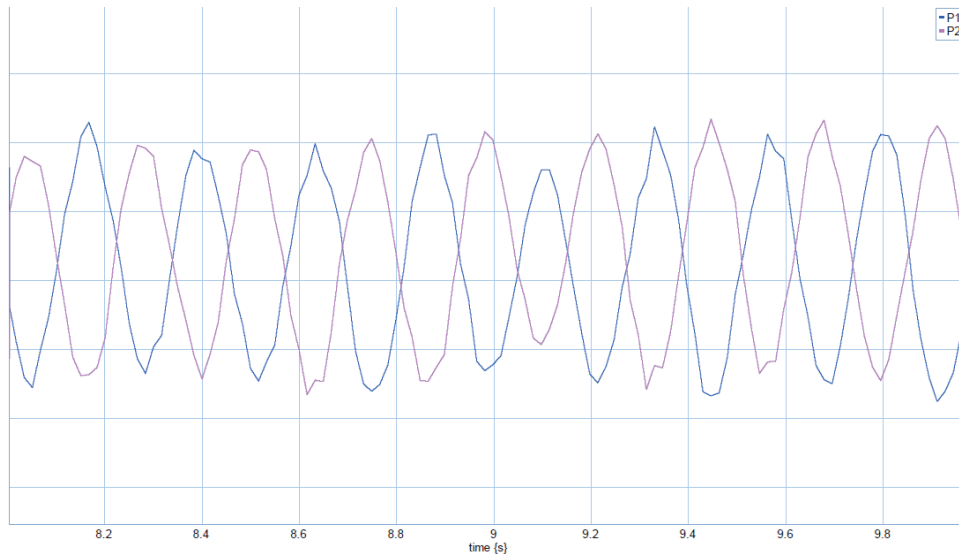


Figure 13. Steady state parallel operation 2 generator set's .

The above measurements demonstrates that in a dynamic system something can go wrong if there is insufficient system damping and the resonance frequency of the system is too close to the diesel excitation . The frequency of the steady state oscillation is the half order of the prime movers . Because of the low internal damping in the synchronous generator the load angle of the generator oscillates. Simulations revealed that the load angle of the generator is swinging with an amplitude of about 0.1° at a frequency of 4.3 Hz.

5. The synchronous generator model

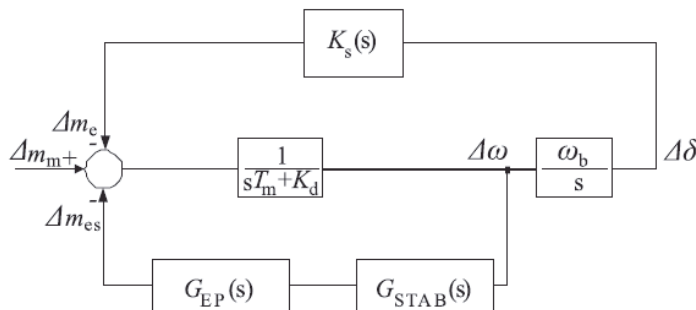


Figure 1. Relationship between electromechanical torque m_m , generator rotor speed ω and load angle δ

Figure 14, from reference: [8]. This block diagram is close to the “mechanical” generator model used for TVA analysis.

This paper, reference [8], is handling the electromechanical oscillations as they are manifested in land based power systems where they occur at frequencies in between 0.1 and 0.3 [Hz].

It appears that there is not much literature available handling the possible problems in ship's systems with weak grids and nowadays with the use of frequency drives with also a low grid inertia.

The author was inspired to write a paper from the involvement with a ship's system which showed a power oscillation between the parallel running generators with a frequency at the halve order of the diesel.

5.1 Model parameters of the synchronous generator

Calculating of the model parameters by use of the load angle parameters for the “mechanical” synchronous generator model.

The load angle data is not a usual content of the nameplate of the generator, it requires some effort to get these data.

Generator:

HOLEC DG225Y , 11350 kVA, 6600 V, pf = 0.9 , n = 500 , see the rotor drawing at figure 9.

Data of the load angle and the calculated stiffness of the magnetic spring:

Load	δ_e	δ_m	δ_m [rad]	Torque [Nm]	stiffness [Nm/rad]
100%	16.5°	2.75°	0.047996	191e3	3.98e6
75%	13.8°	2.3°	0.0401	143e3	3.567e6
50%	10.3°	1.716°	0.02996	95.5e3	3.187e3
25%	5.7°	0.95°	0.01658	47.75e3	2.88e6
-25%	6.4°	1.066°	0.01861	-47.75e3	2.57e6

δ_e = electrical load angle, δ_m = mechanical load angle see formulae (12)

Calculation the synchronizing power at different loads (see page 10):

$$100\% \text{ load: } P_s = 35.91 \times \cos 16.5 = 34.4 \quad T_s = 657.56 \text{ [KNm]}$$

$$75\% \text{ load: } P_s = 35.91 \times \cos 13.8 = 34.9 \quad T_s = 666.54 \text{ [KNm]}$$

$$50\% \text{ load: } P_s = 35.91 \times \cos 10.3 = 35.5 \quad T_s = 674.75 \text{ [KNm]}$$

$$25\% \text{ load: } P_s = 35.91 \times \cos 5.70 = 35.7 \quad T_s = 683.40 \text{ [KNm]}$$

With the Mathcad function: polyfitc we derive a function: $\delta_m = f(T)$, and $C_m = f(T)$ to use as a function block in the simulation.

See the figure 22. the function block Kc generates the mechanical load angle δ_m and the block Ks

6. The model for torsional vibration analysis.

The purpose of this modelling is to investigate the torsional vibration behaviour of the system, a diesel engine with on the flywheel side a flexible coupling with a clutch then a reduction gearbox, propeller shaft with the propeller, and on the PTO side a flexible coupling with a generator. See figure 15.

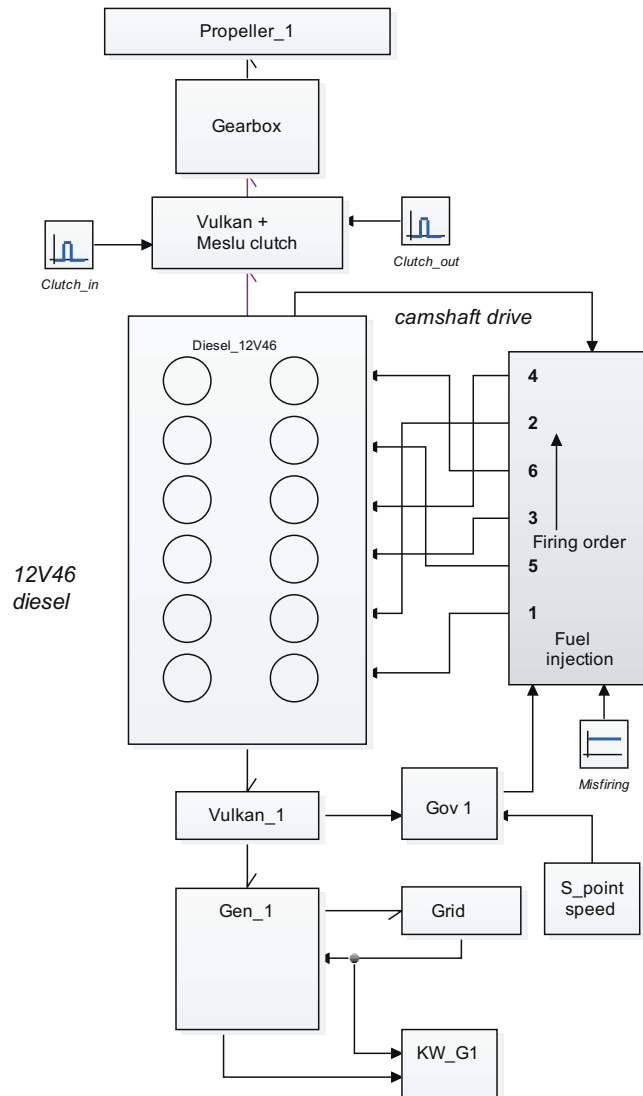


Figure 15. Complete torsional vibration analysis model.

The iconic model is self explaining and on the next pages the bond graph within the icons will be clearlyfied.

It appears that a thermodynamic model of the combustion to generate the combustion force on the piston is interesting but too complicated for this purpose.

A practical approach to model the combustion pulse was found in reference [4], page 198 .

For the piston force a cosine shaped variable is chosen, to be max. in the upper dead point and min. in the lower dead point. Another advantage is that for dynamic analysis of the speed control of the diesel the fuel pump delay is automatic taken into account.

Figure 16. depicts how to generate this combustion signal with the help of some iconic models from the 20Sim library.

Torque measurements reveals that this approach generates a simulation output close to the measurements.

As the inertia of the generator is much more than the diesel inertia the rotational speed is more even, thanks to the Vulkan coupling, the governor speed sensing is performed from the shaft of the generator rotor, the back-up speed sensing is from the diesel flywheel.

6.1 Combustion pulse generator

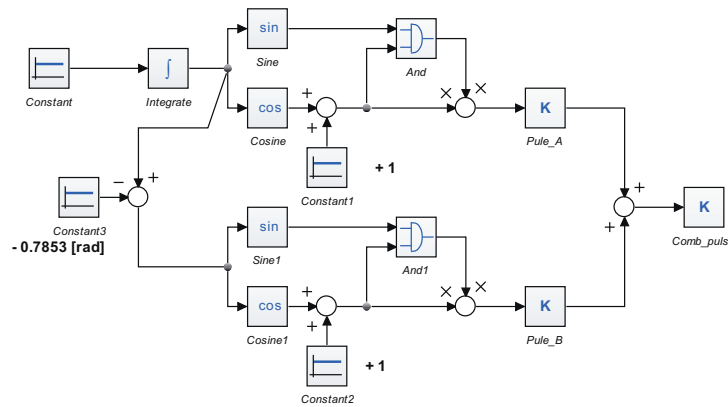


Figure 16 . Iconic model to generate the combustion pulse.

The diesel engine involved is a Wärtsilä 12V46 engine, the V angle is 45° degrees = 0.7853 rad..

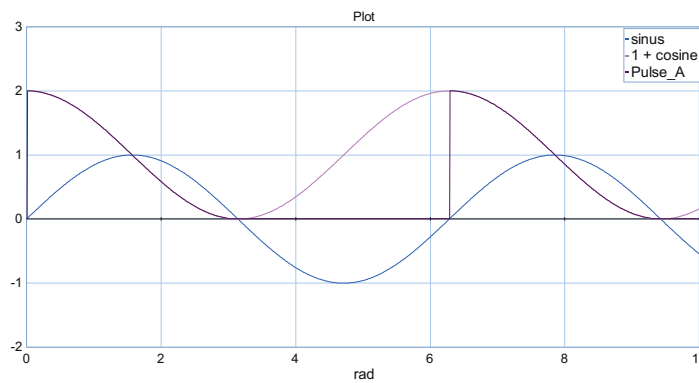


Figure 17 Simulation output of the model of figure 15.

The black line is the combustion pulse, max. at upper dead point an min. at lower dead point. Comparing the FFT of the output torque of the diesel measurements and the FFT of the simulation of the diesel output torque this matches realistically.

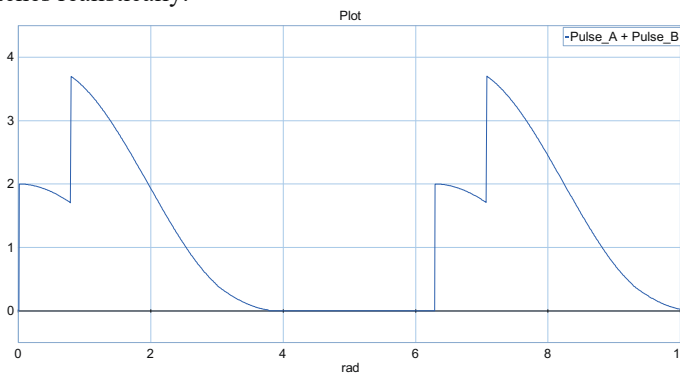


Figure 18. The combined combustion pulse for a V engine.

To speed up the simulation process the iconic blocks are replaced by 20Sim code to generate the combustion pulses, see **appendix 1**.

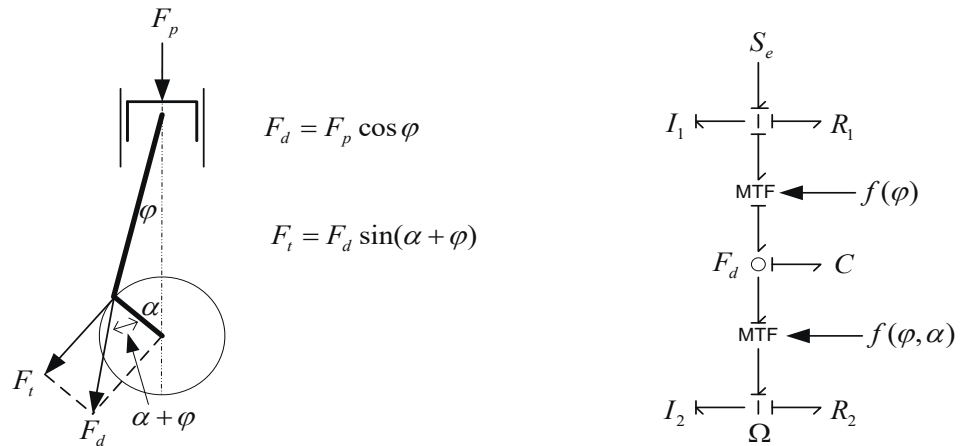


Figure 19 Piston crank mechanism and the Bond Graph

S_e = force on piston from combustion
 I_1 = mass of the piston I_2 = inertia crank + flywheel
 C = stiffness rod + oil film

See reference [4].

To model the piston – crank mechanism in this manner needs information that is not easy available from the engine manufacturer and for only TVA such a detailed model delivers not more information and takes more computation time. Therefore a simplified model is used see figure 20 below.

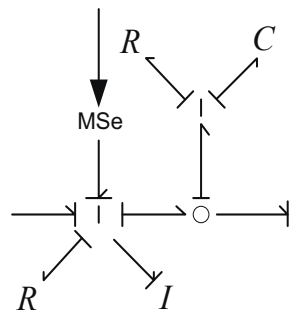


Figure 20 Bond graph inertia piston and stiffness crank.

I = reduced inertia piston + crank.
 C = stiffness crank shaft.
 MSe = combustion force.

6.2 Diesel rotating parts model

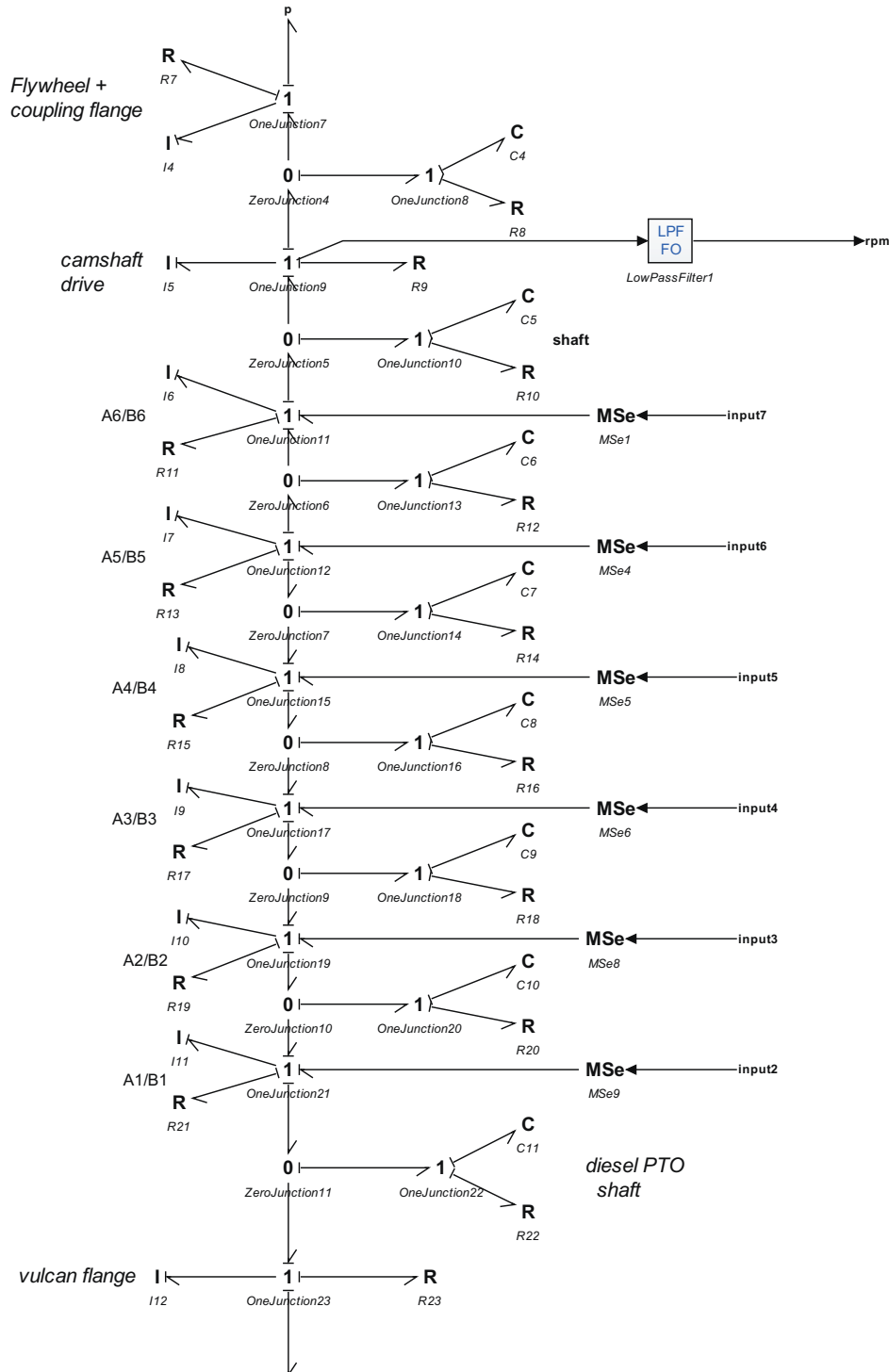


Figure 21. BG model elastic system 12V46 engine crank shaft.

On the right side the 6 combined cylinder inertia's and on the left side the 6 torque pulses.

6.3 Mechanical synchronous generator model

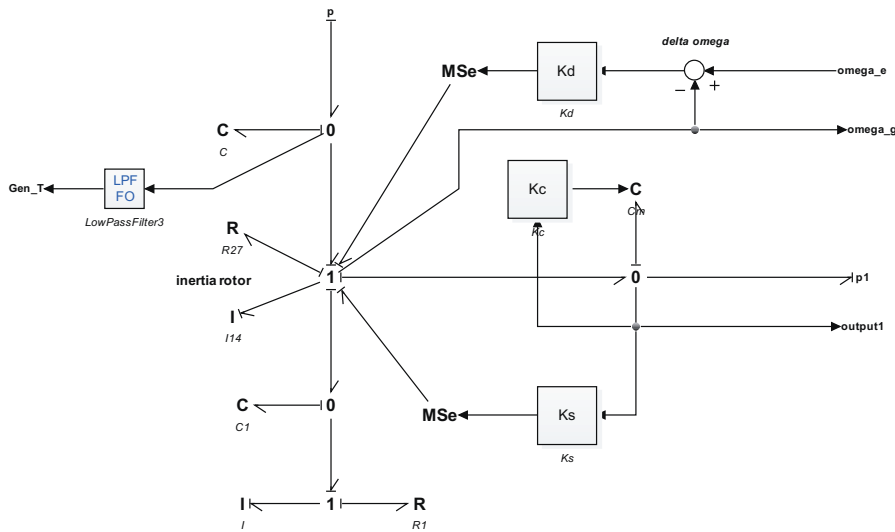


Figure 22. BG model synchronous generator for TVA purposes

Description of the model, a "mechanical" synchronous generator model.

- I14 = rotor inertia with some friction (R27)
- Kc = magnetic spring, stiffness $Kc = f(T)$
- Kd = damping torque = magnetic damping proportional with d_{ω_n}/dt
- Ks = generation of the load angle δ . $\delta = f(T)$ and "synchronizing" torque from: $\Delta\delta \quad Ts = ks \cdot \Delta\delta$
- I = inertia exciter
- C = rotor shaft stiffness DE
- C1 = rotor shaft stiffness NDE

The function that calculates the load angle δ , from the delivered torque is derived by the Mathcad function "polyfit" from the manufactures data of the load angle at different loads of the generator, 10%, 25%, 50%, 75% and 100% load.

Also the function that calculates the magnetic spring stiffness is derived by using the Mathcad function "polyfit" The model is validated from torque measurements on a 12 MVA generator shaft. For a "true" 3 phase generator model see **appendix [3]**.

6.5 Flexcible coupling and clutch model

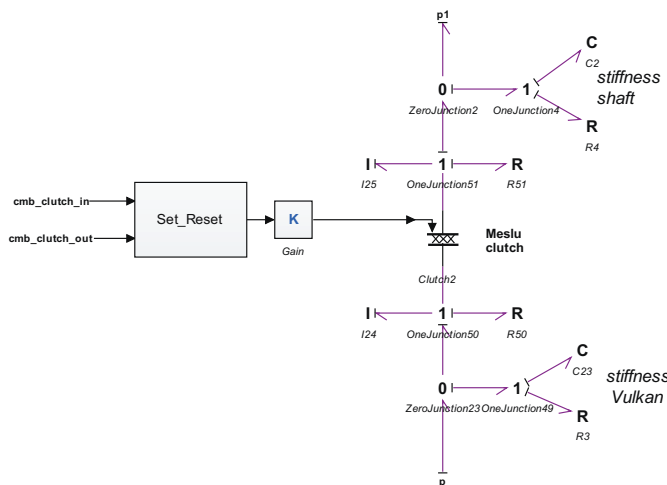


Figure 23 BG model Vulkan coupling with clutch.

The used clutch model is a simple clutch model from the 20Sim librarie and its suits for this purpose. Friction is not easy to model it in a proper way and again the auther refers to the literature available on this subject see [37]. And also the stiffness of the elastic element is modeled as a simple spring.

6.6 Gearbox model

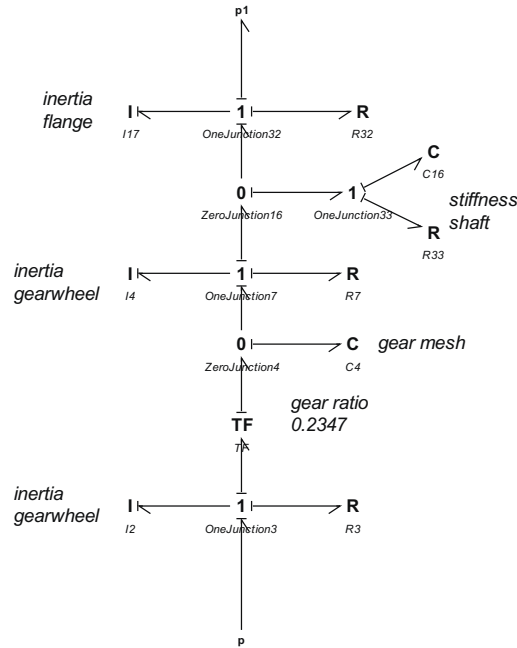


Figure 24. BG model gearbox.

The model can be simplified by reducing the system to engine speed then the system simplifies to the gearbox inertia the propeller shaft stiffness and then the propeller inertia.

6.7 Propeller model

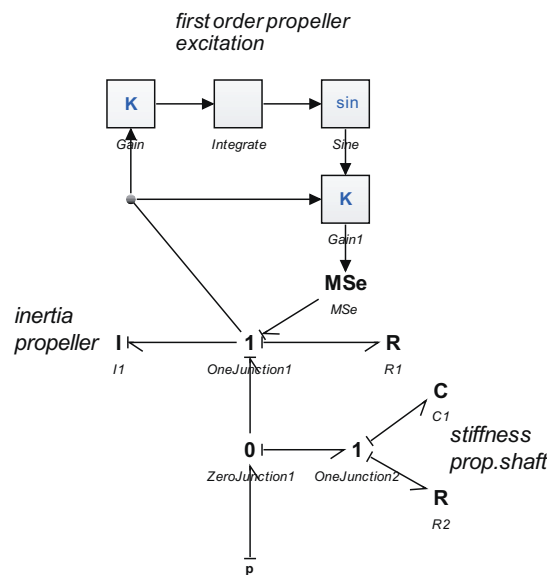


Figure 25. BG model of propeller shaft and propeller inertia and the 20Sim blocks to generator the first order propeller excitation.

6.8 Simple speed governor model

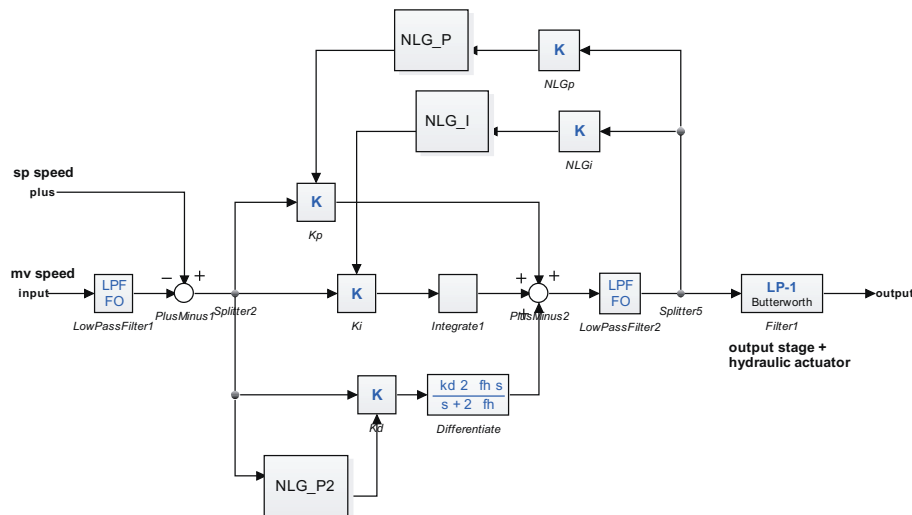


Figure 26. Simplified model of the Viking 25 governor.

As depicted this is a simplified model, normally the governor has a some more variable gains as there are: error dependable gain(small error low gain big error high gain), speed dependable gain, error dependable integration time. See **appendix 2** for a detailed block diagram.

In the model there is only load dependable gain and load dependable integration time and error dependable derivation time (big error no derivation , small error some derivation) .

The dynamics of a governor are determined by the actuator to move the lever of the fuel pumps see again **appendix 2** for a block diagram of the hydraulic actuator, the up/down time of the actuator is about 80 msec. so the actuator is modelled as a first order system with a time constant of 80 msec.

The modern diesel is now a common rail engine and this engine is much better to control, today necessary to comply with the exhaust emission rules, the controller output is direct input to the fuel injector.

6.9 The flexible coupling

Another object of interest is the flexible coupling, in this simulation model, modelled as a simple spring with damping. As depicted in the icon in figure 15 there is a flexible coupling between the diesel and generator and a second one between the diesel and the propulsion gearbox.

The author was able to do some measurements during a commissioning at the Vulkan factory in Herne (years ago but the physics of the rubber elements do not change) and an abstract of the analysis of the measurements will be given see pages 25 and 26.

For interesting literature regarding this subject see [30], [31], [32], [33], [34].

Unfortunately this literature does not present a “usable” rubber model for use in a TVA model.

The analysis of the measurements revealed that a rubber coupling element is a highly nonlinear piece of material, at low oscillating torque amplitude there is no damping and at high oscillating torque there is considerable damping.

Regarding the stiffness at low oscillating torque amplitude there is high dynamic stiffness and at high oscillating torque amplitude the dynamic stiffness is lower.

This nasty habit of the rubber elements in a flexible coupling played an important role in the project with unpleasant oscillations (see figure 11 , 12 and 13), unfortunately the flow of information is blocked by a NDA agreement.

The flexible coupling (Vulkan_1) between the diesel and generator is a model with stiffness and damping adaption derived from the Vulkan measurements and the Vulkan data sheet during a commissioning at their factory. The Author developed a “usable” coupling model

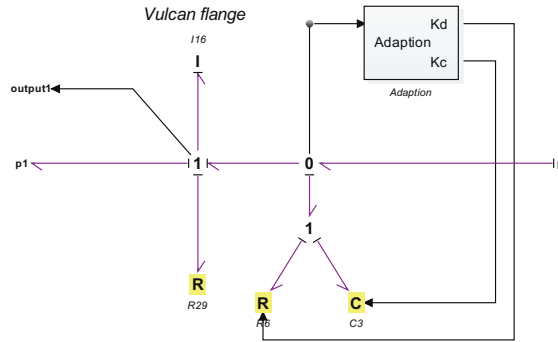


Figure 27. BG model flexible coupling with adaption block.

The adaption block has 2 outputs, output Kd is the elastic element damping adaption signal and output Kc is the elastic element stiffness adaption signal.

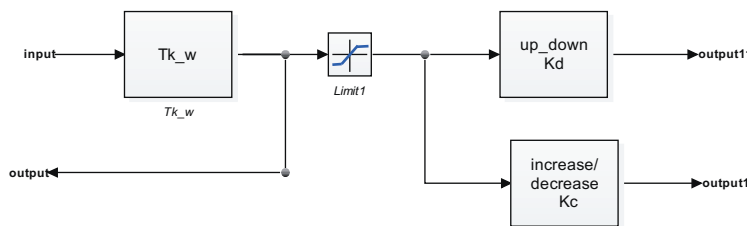


Figure 28. inside the adaption block.

The Tk_w block determines the amplitude of the oscillating torque superimposed upon the mean torque and the right blocks determines the adaption factor for the damping and the stiffness of the elastic element.

In the Vulkan performance data the Vibratory Torque T_w is specified as: “The vibratory torque T_w is the amplitude of the fluctuating torque superimposed upon the mean torque T_m in the stationary condition, T_w should not exceed the permissible maximum vibratory torque T_{kw} ”. See reference [38].

The adaption block is not complete, the internal warming up of the element is not yet modelled . To use this elastic coupling model is also useful at starting the model, starting of the diesel generates high oscillating torque in the element and can cause numeric instability

6.10 The grid model.

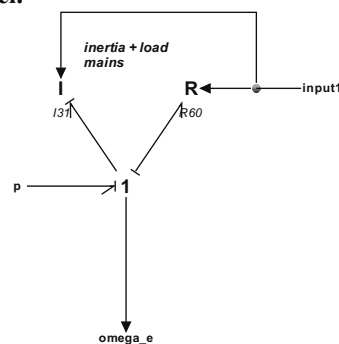


Figure 29. Simple grid model.

The grid is modelled as an inertia (I31) with a resistive load (R60) ,input1 is the increasing load command input and the grid inertia is also increased with the load.

Please note that the grid inertia has to be transformed to the diesel speed and that means that the grid inertia has to be multiplied by a factor 36.

The generator has 6 pole pairs (50 Hz), so the factor is 6².

When there are some big direct on-line electrical motors than the grid inertia can be considerable and has a low damping to the possible electrical power oscillations .

Simulations depicts that a low grid inertia has more damping than a high grid inertia.

7. Some measurements of a Vulkan flexible element.

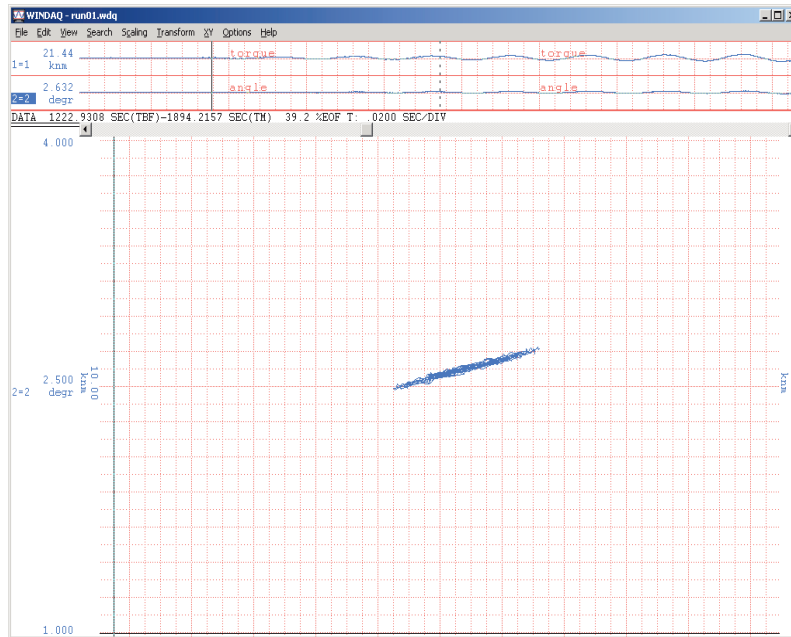


Figure 30. Element 27RTR , x axis applied torque y axis angle of deflection
 Temperature element is 30°.
 Pre torque of 20 kNm.
 Amplitude oscillating torque: 4.3 kNm top-top.
 Dynamic stiffness is 1411 kNm/rad.
 Notice that the element has almost no damping.

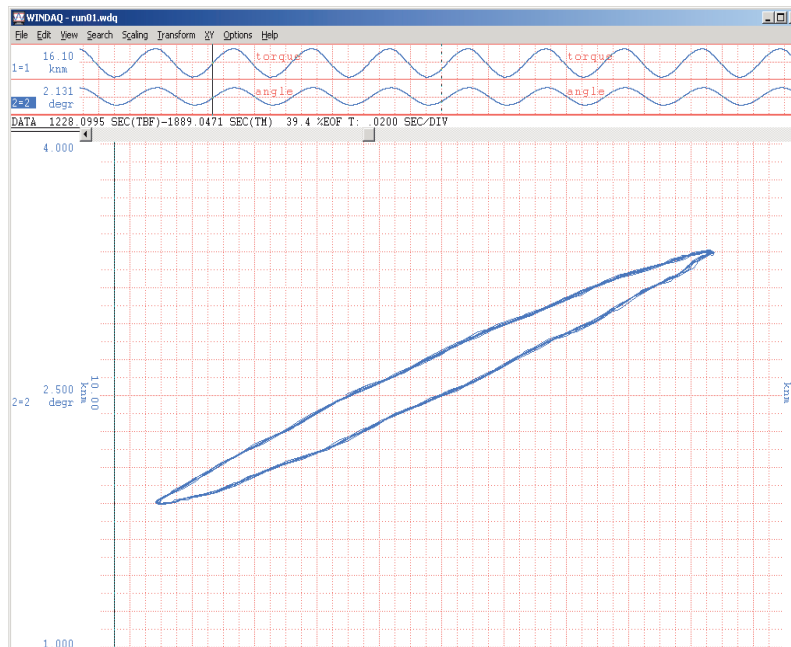


Figure 31 Element 27RTR. , x axis applied torque y axis angle of deflection
 Temperature element is 30°.
 Pre torque of 20 kNm.
 Amplitude oscillating torque: 16.67 kNm top-top.
 Dynamic stiffness is 645.1 kNm/rad.
 Notice that the element has now damping, the width of the banana is a the amount of damping.

7.1 Mathcad plot of damping versus oscillating torque

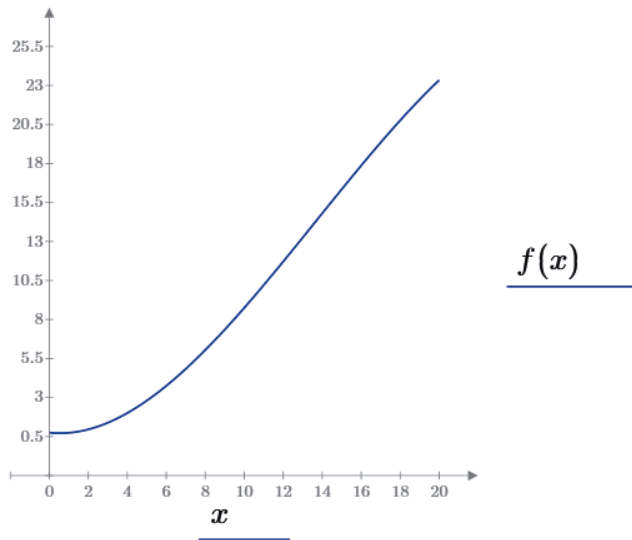


Figure 32. Plot from the Vulkan measurements.

x axis: ΔT [kNm]
y axis: damping amplification factor

7.3 Mathcad plot of stiffness versus oscillating torque

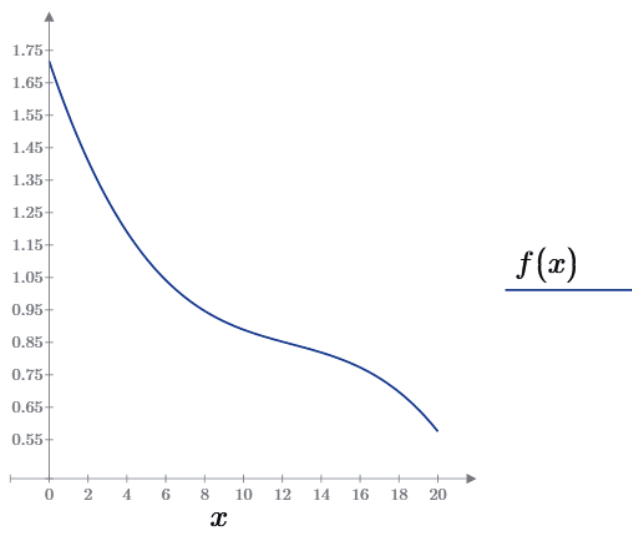


Figure 33. Plot from the Vulkan measurements.

Stiffness as a function of the amplitude of the oscillating torque
x axis: ΔT [kNm]
y axis: stiffness reducing factor

The above plots demonstrates the nonlinear properties of rubber elements of a flexible coupling.

In a proper engineered system the resonance frequencies are more than a factor $\sqrt{2}$ from the system excitations than the system works fine with the stiffness parameters from the manufactures data sheet.

Also an important property is that at increasing temperature of the elements the stiffness is decreasing. See also reverence [34].

8. Simulation results.

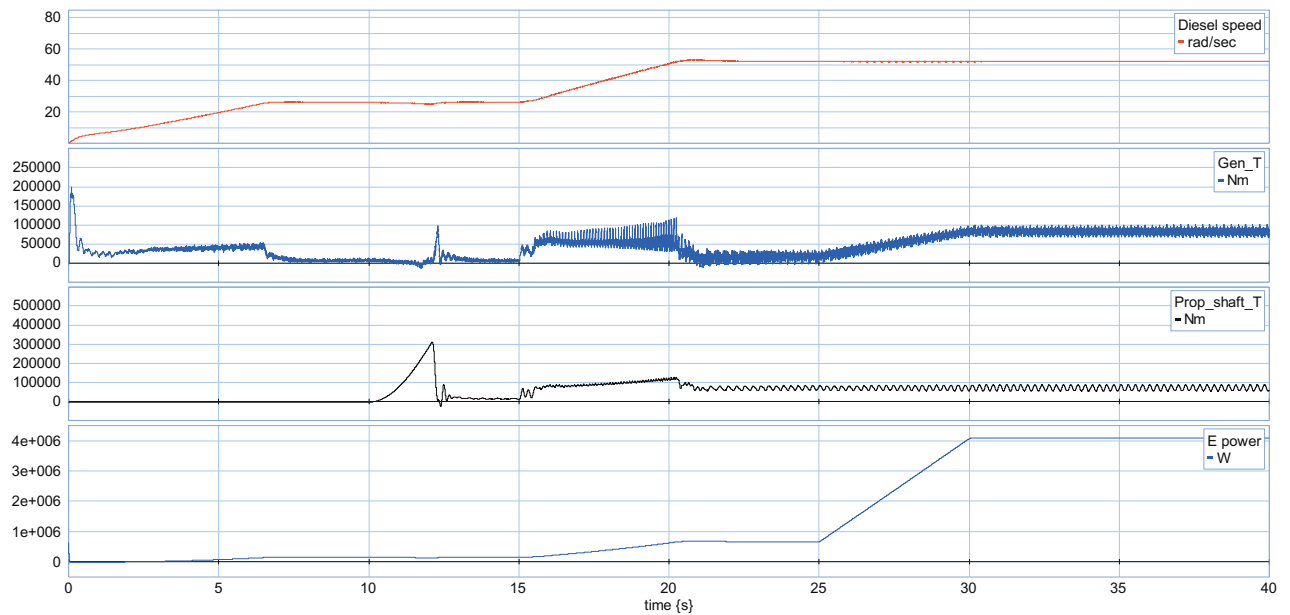


Figure 34. Simulation output.

- Plot 1: diesel speed [rad/sec]
- Plot 2: generator shaft torque [Nm]
- Plot 3: propeller shaft torque [Nm]
- Plot 4: generator electrical power.[W]

At $t = 0$ the diesel starts and speed up to half nominal speed, at $t = 10$ seconds the propeller is clutched-in and at $t = 15$ seconds the diesel speeds up to nominal speed = 500 rpm = 52.4 [rad/sec].

At $t = 25$ seconds the electrical generator load is increased to $4e6$ [W].

The simulation depicts that there are no resonances in the speed range of the diesel.

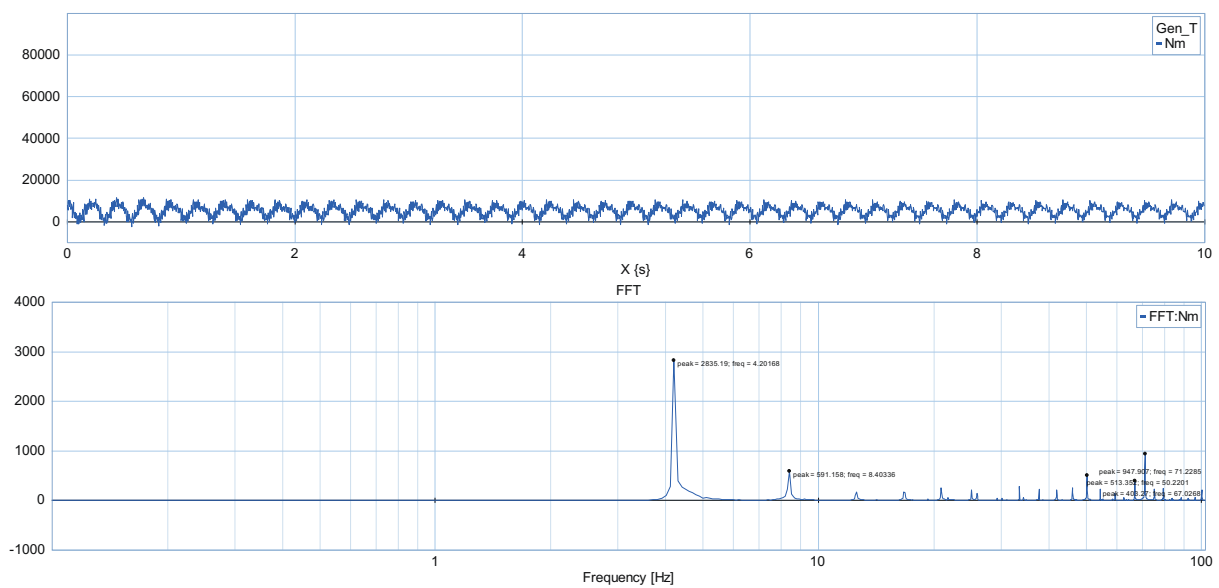


Figure 35. Simulation output no load torque in generator shaft.

The first peak in the FFT plot is the half order of the diesel, the second smaller peak is the first order of the diesel.

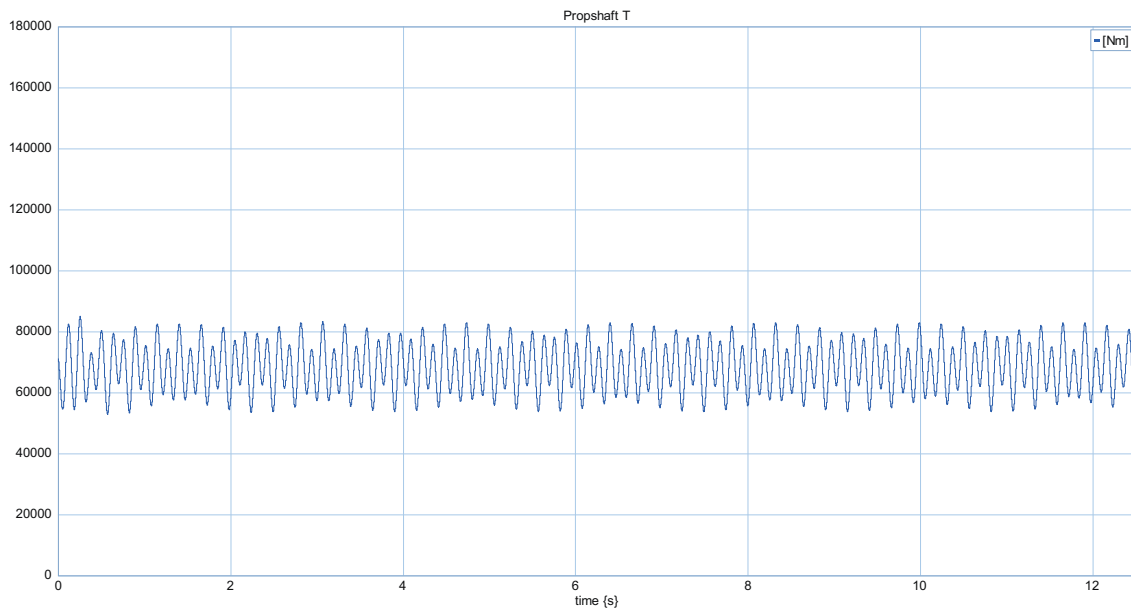


Figure 36. Simulation output no load torque propeller shaft.
The model is generating the same torque signal as measured in the propeller shaft see figure 37.

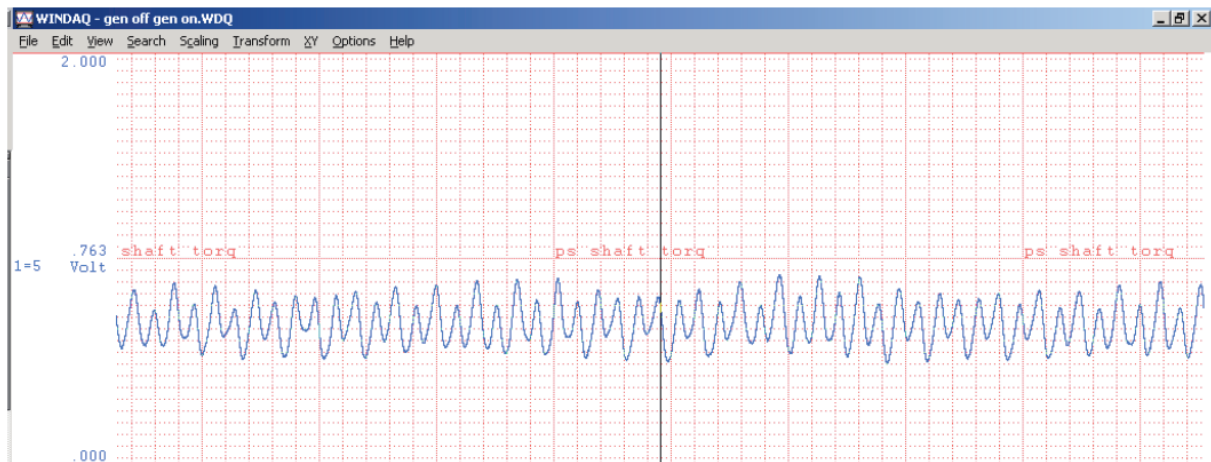


Figure 37. no load torque measurement propeller shaft.

The shaft torque measurement had an output of 118.5 kNm/V, so the vertical scaling is 6.97 kNm/div.
The horizontal scaling is 0.28 sec/div.

Literature/manufacture's documentation revealed that the entrained water of the propeller should be taken into account and in case of a controllable pitch propeller the entrained water depends on the pitch. That implies that the resonance frequency in the system is changing with the pitch of the propeller. The Author has never measured a changing resonance frequency due to the changing pitch of the propeller.

9. Torsional vibration analysis complete power and propulsion plant.

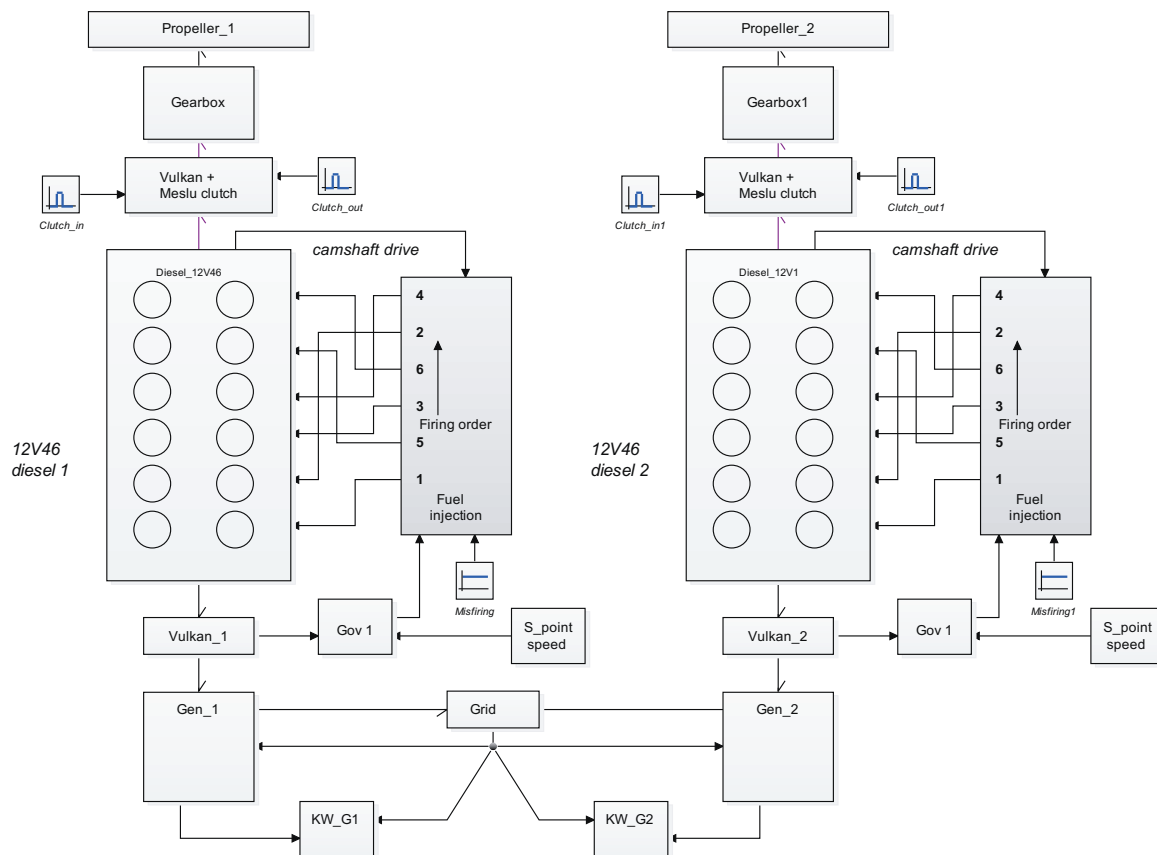


Figure 38. Iconic simulation model propulsion and power generation.

9.1 The grid model.

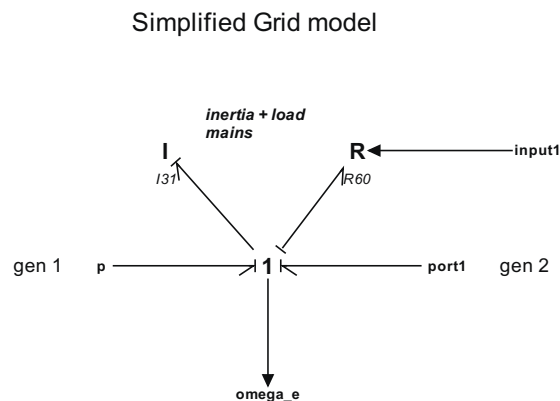


Figure 39. Grid model with two power inputs.

Building a model for torsional vibration analysis of parallel running generator set's is simple (thanks to the use of bondgraphs) as the iconic model depicts.

Just copy/paste the single engine model and add a power input port to the grid model and connect.

This model is for TVA purposes, to investigate the dynamics of parallel switching/load sharing of generator set's a more complicated model has to be used. The generator has to be modelled as a true 3 phase model, that's no problem in 20Sim but for torsional vibration analysis purposes to complicated.

And to model a synchronous generator as a true 3 phase model requires more system parameters not easily available from the manufacturer. In other words you need good personal contacts to get these information.

Another challenge is with the true 3 phase model to model the process of synchronizing the two generators.

And to find a good model of a three phase circuit breaker or to model a switch in a proper way.

To get a good block diagram of the working of a synchronizer is not easy.

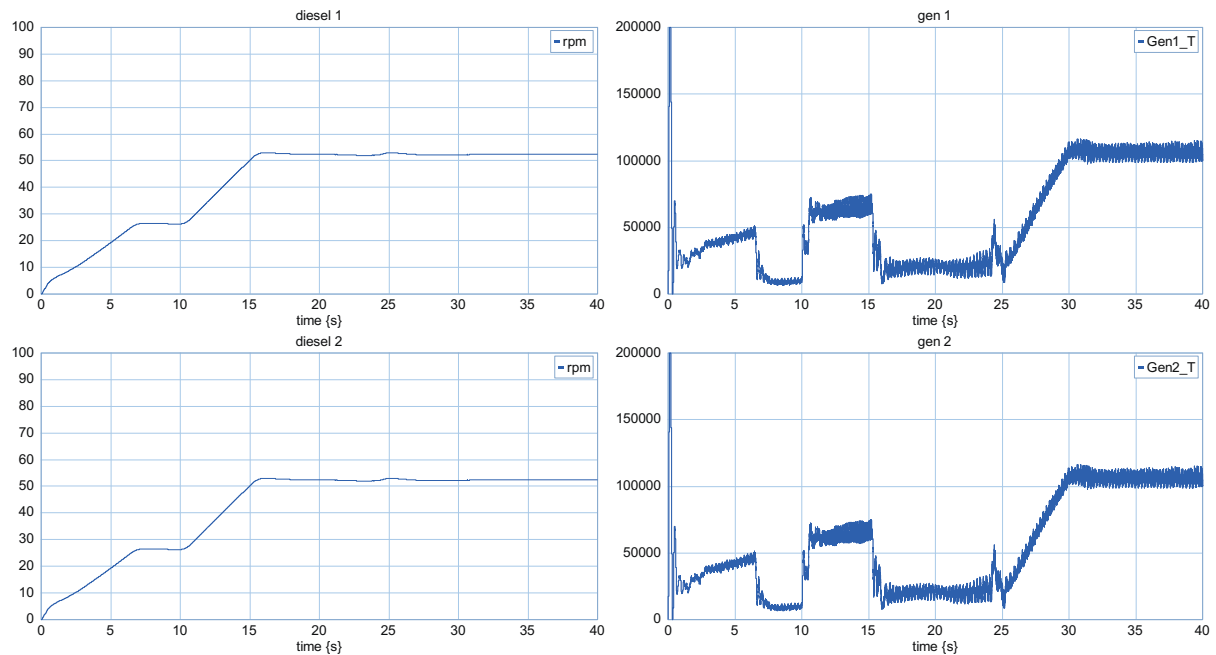


Figure 40. Simulation output .

Above plots are speed of diesel 1 with the generator 1 shaft torque.

The below plots are speed of diesel 2 with the generator 2 shaft torque

As the interest are the torsional vibrations the diesels starts simultaneously and speed up to nominal speed.

To investigate the dynamics of parallel switching of the generators than this model is to simple.

Then there is the need for a “true” 3 phase synchronous generator model see **appendix 3** .

Conclusions.

The synchronous 3 phase generator appears to be a highly dynamic system.

It is not easy to get “usable” data regarding the damping torque of a synchronous generator, see for an explanation [39].

This paper depicts the power of Bond Graphs in modeling marine engineering systems. The first problem for a control engineer or systems engineer in front of a process to control is to obtain a precise and easily manipulated model with predictions corresponding to the real observations.

The use of 20sim as a simulation package make live a lot easier because of the ability of direct input of the Bond Graph.

For people not familiar with Bond Graphs there is a large library of iconic models see www.20Sim.com.

The modeling itself is broaden the insight and understanding of the system.

The most time consuming part of modeling is to get the right parameters of the system or the systems components. It depends largely on knowing the right man on the right place to get the information needed to set up a model that represent the physical system.

Measurements revealed that the simplifications, among other things the generation of the combustion pulses, in the modeling where allowed for this reason finding the resonance frequencies of the system, as mentioned before one looks for relevant analogies, not for complex identities.

Unfortunately the system involved and the mentioned measurements were not allowed to use in this paper.

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²⁾Faculty of Engineering University of Ryeka
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Appendix 1

20Sim code of the combustion pulse generator for a V12 engine

```

variables
  real a,c,d,f,g,i,j,l,m,o,p,x,z;
  real aa,cc,dd,ff,gg,ii,jj,ll,mm,oo,pp,xx,zz;
  real ww;
  boolean b, bb, e, ee, h, hh, k, kk, n, nn, q, qq;
  real port1, port2, port3, port4, port5, port6, port8, port9, port10, port11, port12, port13;
equations
  z = misfiring;
  zz = fuel;
  ww = speed * 0.5;
  x = sin ( ww * time);
  a = 1 + cos ( ww * time);
  b = if x > 0.01 and a > 0.01 then
      true
    else
      false
    end;
  if b then
    port1 = zz *(0.99 * a);           // pulse 1A
  else
    port1 = 0;
  end;
  xx = sin ( ww * time - 0.43633);  // Vee angle 45 degrees.
  aa = 1 + cos ( ww * time - 0.43633);
  bb = if xx > 0.01 and aa > 0.01 then
      true
    else
      false
    end;
  if bb then
    port8 = zz * aa;                 // pulse 1B
  else
    port8 = 0;
  end;
  cylin_1 = port1 + port8;
  c = sin ( ww * time - 1.0473);
  d = 1 + cos ( ww * time - 1.0473);
  e = if c > 0.01 and d > 0.01 then
      true
    else
      false
    end;
  if e then
    port2 = zz*( 0.98 * d);          // pulse 2A
  else
    port2 = 0;
  end;
  cc = sin ( ww * time - 1.4835);
  dd = 1 + cos ( ww * time - 1.4835);
  ee = if cc > 0.01 and dd > 0.01 then
      true
    else
      false
    end;
  if ee then
    port9 = zz * (1.03 * dd);        // pulse 2B
  else
    port9 = 0;
  end;
  cylin_2 = port2 + port9;

```

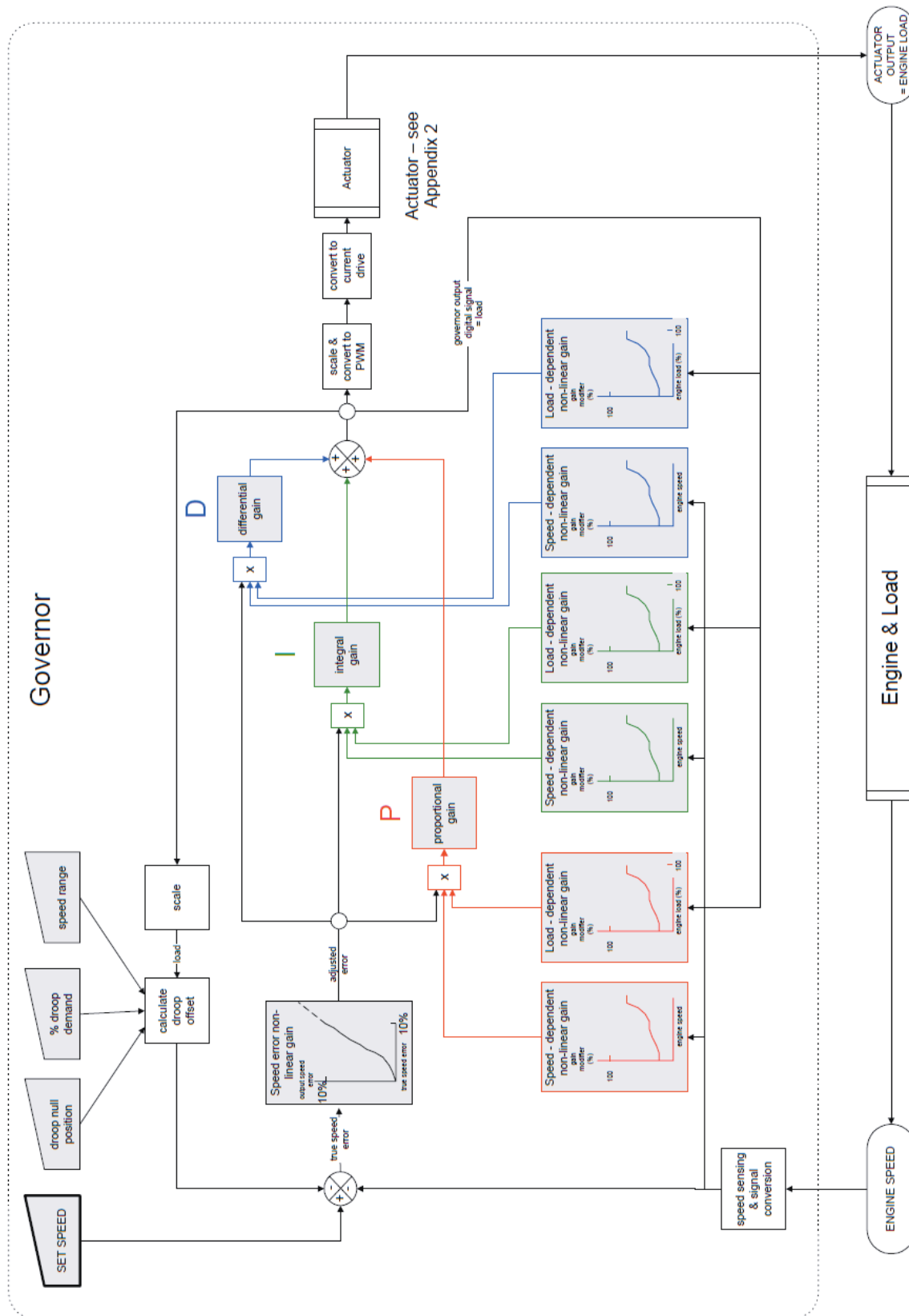
```

f = z * sin ( ww * time - 2.0944);
g = z * ( 1 + cos ( ww * time - 2.0944));
h = if f > 0.01 and g > 0.01 then
    true
    else
        false
    end;
if h then
    port3 = zz * (0.97 * g);           // pulse 3A
else
    port3 = 0;
end;
ff = sin ( ww * time - 2.503);
gg = 1 + cos ( ww * time - 2.507);
hh = if ff > 0.01 and gg > 0.01 then
    true
    else
        false
    end;
if hh then
    port10 = zz * gg;                // pulse 3B
else
    port10 = 0;
end;
cylin_3 = port3 + port10;
i = sin (ww * time - 3.1416);
j = 1 + cos (ww * time - 3.1416);
k = if i > 0.01 and j > 0.01 then
    true
    else
        false
    end;
if k then
    port4 = zz * (0.98 * j);         // pulse 4A
else
    port4 = 0;
end;
ii = sin (ww * time - 3.5773);
jj = 1 + cos (ww * time - 3.5773);
kk = if ii > 0.01 and jj > 0.01 then
    true
    else
        false
    end;
if kk then
    port11 = zz * (1.02 * jj);      // pulse 4B
else
    port11 = 0;
end;
cylin_4 = port4 + port11;
l = sin (ww * time - 4.1888);
m = 1 + cos ( ww * time - 4.1888);
n = if l > 0.01 and m > 0.01 then
    true
    else
        false
    end;
if n then
    port5 = zz * (0.97 * m);        // pulse 5A
else
    port5 = 0;
end;

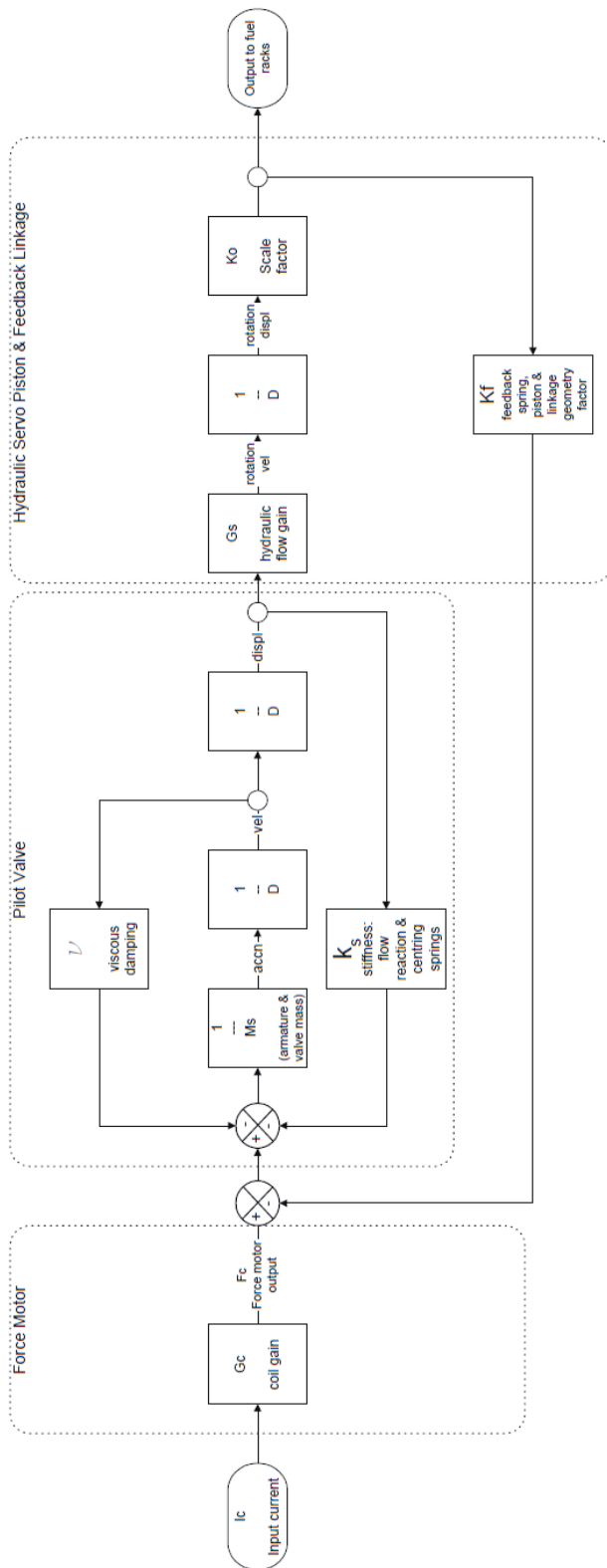
```

```
ll = sin (ww * time - 4.6251);
mm = 1 + cos ( ww * time - 4.6251);
nn = if ll > 0.01 and mm > 0.01 then
    true
    else
        false
    end;
if nn then
    port12 = zz * (1.02 * mm); // pulse 5B
else
    port12 = 0;
end;
cylin_5 = port5 + port12;
o = sin ( ww * time - 5.2360);
p = 1 + cos ( ww * time - 5.2360);
q = if o > 0.01 and p > 0.01 then
    true
    else
        false
    end;
if q then
    port6 = zz * (0.97 * p); // pulse 6A
else
    port6 = 0;
end;
oo = sin ( ww * time - 5.6723);
pp = 1 + cos ( ww * time - 5.6723);
qq = if oo > 0.01 and pp > 0.01 then
    true
    else
        false
    end;
if qq then
    port13 = zz * (1.02 * pp); // pulse 6B
else
    port13 = 0;
end;
cylin_6 = port6 + port13;
```

Appendix 2

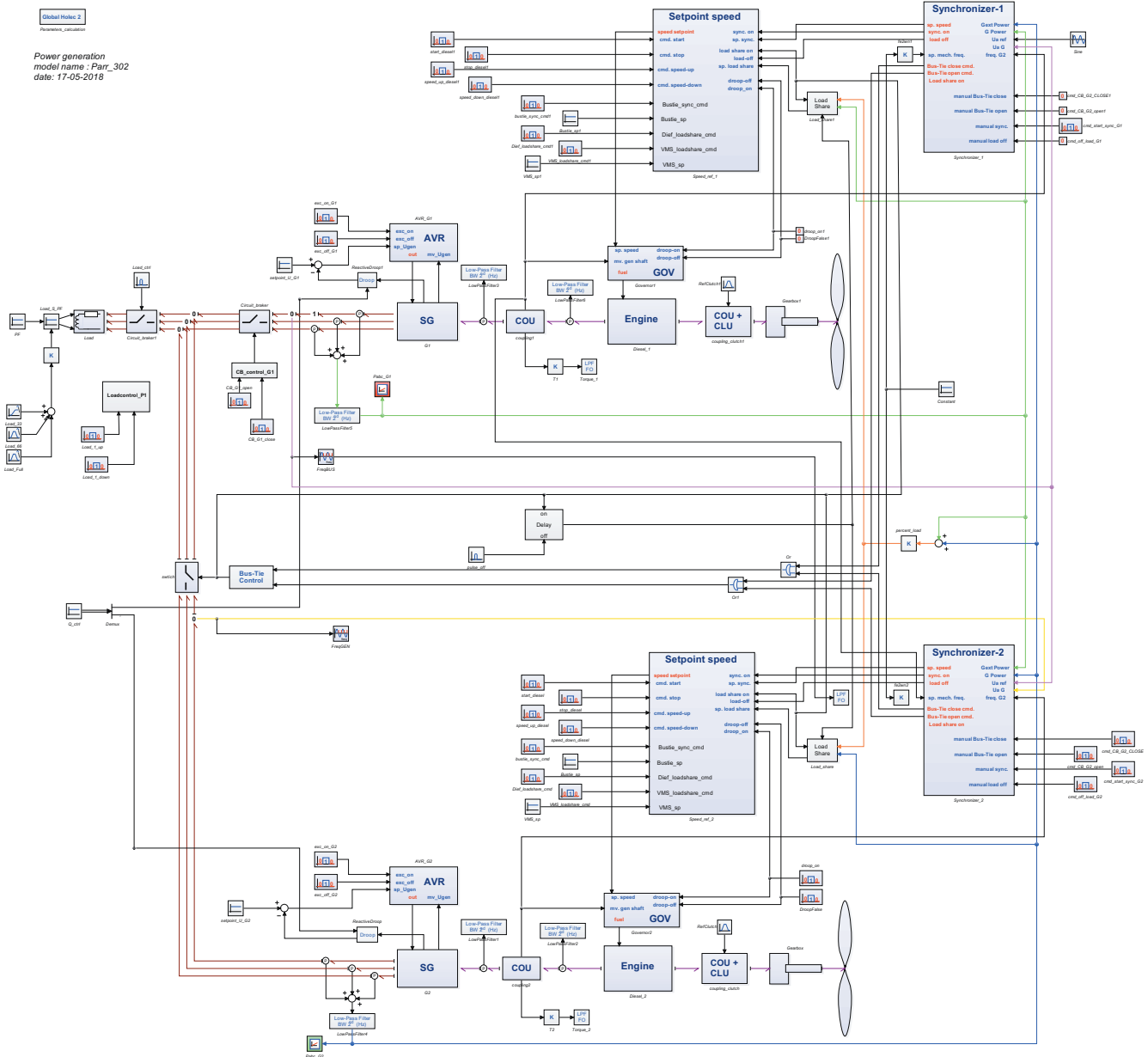


Software block diagram Viking 25 governor



Type 2221 Actuator control block diagram.

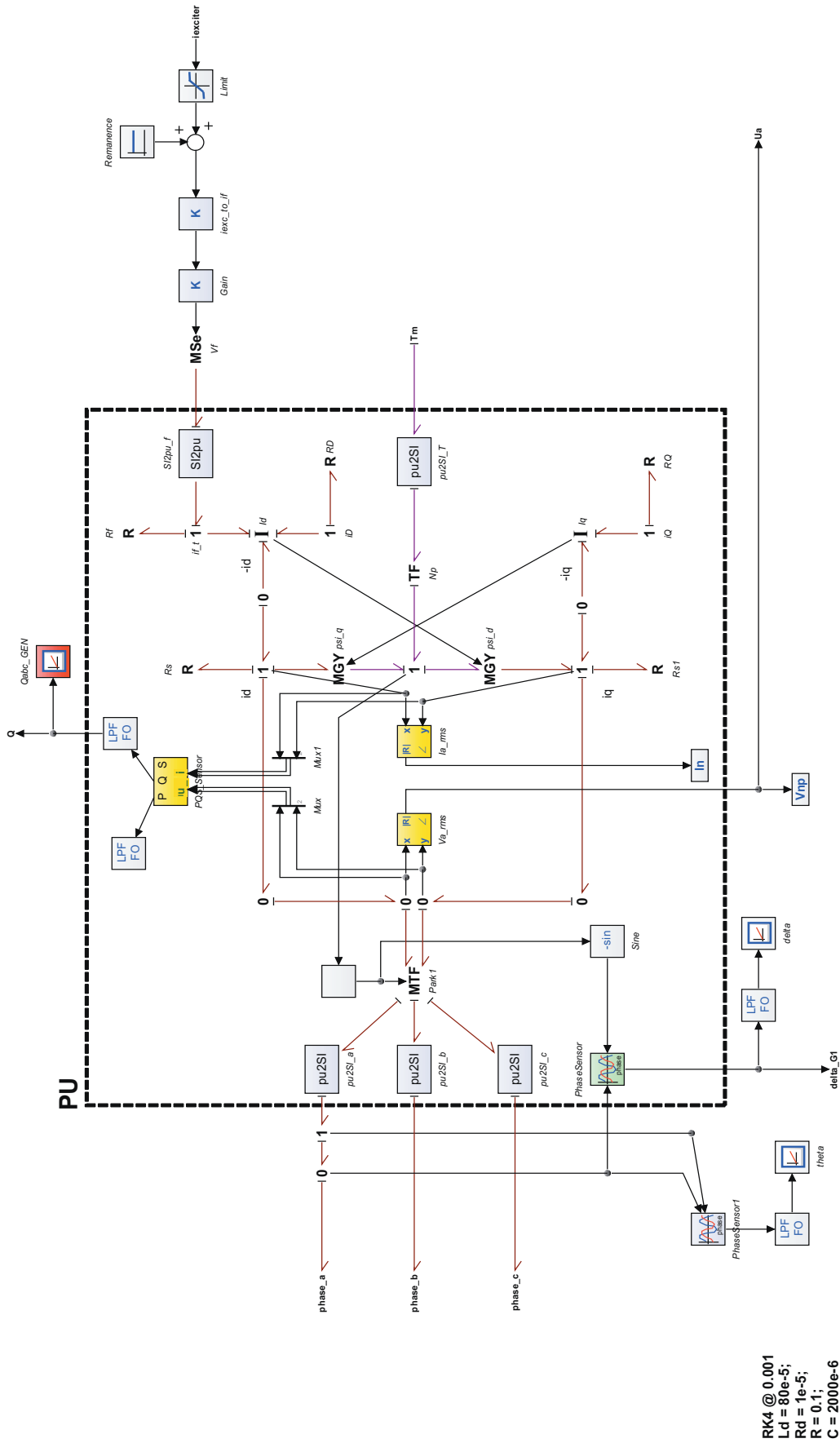
Appendix 3



True 3 phase power generation and propulsion model in 20-sim.

This model has the ability to simulate the transients at parallel switching of the generators and the dynamics of active loadsharing between the generators. The other interesting possibilities of this model are the investigation of short circuit events, and with the package 20-sim 4C it opens the possibility to do HIL simulations.

From the 20 sim website: 20-sim 4C enables rapid prototyping for control engineers. With 20-sim 4C you can run c-code on hardware to control machines and systems.



True 3 phase synchronous generator BG model in 20-sim.

The model is based on the equations of Park, it is not easy to use this model because it uses generator parameters not easily available from the manufacturer.