Large Capacity Hydrostatic Transmission with Variable Displacement

Masashi Sasaki*, Atsushi Yuge*, Toshikazu Hayashi*, Hiroshi Nishino*, Michiya Uchida** and Toshihide Noguchi**

Mitsubishi Heavy Industries, Ltd., Nagasaki R&D Center, 5-717-1, Fukahori-machi, Nagasaki, 851-0392, Japan*
Mitsubishi Heavy Industries, Ltd., Renewable Energy Business Division, 12, Nishikicho, Naka-ku, Yokohama, Kanagawa, 231-8715, Japan**
E-Mail: masashi_sasaki@mhi.co.jp

In the wide range products such as wind turbine generator, engine generator, railway vehicle, ship and so on, the demands for large capacity hydrostatic transmission with high efficiency are increasing as a substitute of conventional drive train system such as gearbox for the purpose of improvement and differentiation of such products. For satisfying such demands, large capacity hydrostatic transmission with variable displacement was developed with applying the Digital Displacement ® technology /1/ of Artemis Intelligent Power, Ltd. The hydrostatic transmission introduced in this paper is comprised of original hydraulic pump and motors. As a result, the authors confirmed that it is possible to manufacture and provide the new hydrostatic transmission with large capacity over 7MW.

Keywords: Hydrostatic transmission, Large capacity, Variable displacement
Target audience: Power generating equipment, Marine machinery

1 Introduction

In the wide range products such as wind turbine generator, engine generator, railway vehicle, ship and so on, the demands for large capacity hydrostatic transmission with high efficiency are increasing as a substitute of conventional transmission such as gearbox in the drive train systems for improvement and differentiation of such products.

For example, in the gearbox of the wind turbine generator, input shaft torque is increased because the rotor speed is low even though a large output is required. In general, the size of gearbox becomes large in proportion to the input torque, therefore it becomes difficult to arrange the drive train system optimally to the limited space in nacelle of wind turbine. So there are few gearboxes with large capacity over 7MW and low input speed so far as the authors know.

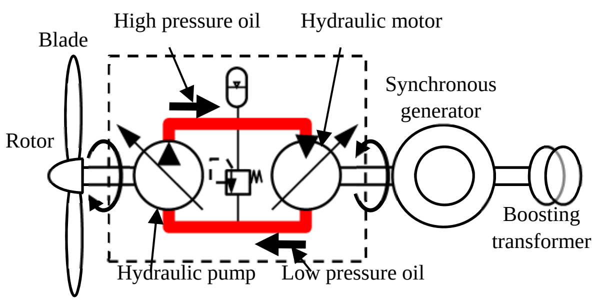
The hydrostatic transmission to be introduced here consists of auxiliary machinery including a hydraulic pump, a hydraulic motor, pipes, an accumulator and coolers. The input rotational power drives the hydraulic pump to generate high hydraulic pressure. The pressure travels through the pipes to rotate the motor at a constant speed and the power transmitted to the output shaft. The hydraulic pump is a ring-cam type multistage radial piston pump, and the hydraulic motor is a crank type radial piston motor. Therefore, it is possible to increase the capacity of the hydrostatic transmission easily with increase in the number of cylinders or stages constituting the hydraulic pump and motor. Additionally, applying the digital control technology of both oil pressure and displacement, it became to be able to transmit power efficiently with variable capacity.

The authors succeeded in the development of large capacity hydrostatic transmission with variable displacement over 7MW by applying these systems. In the following, an overview of the development status is described.

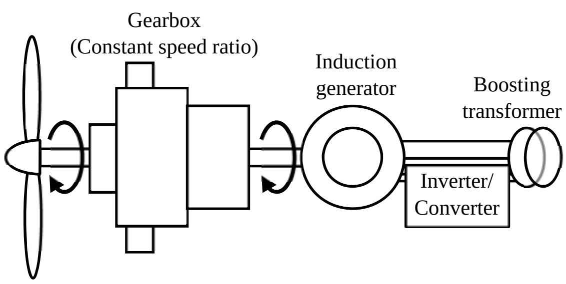
2 Basic Mechanism and Advantage of Hydrostatic Transmission

2.1 Basic mechanism of hydrostatic transmission

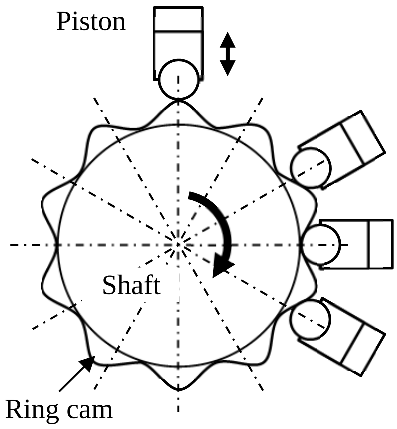
Figure 1 (a) shows the basic configuration of the hydrostatic transmission as a drive train system of wind turbine generator /2/. As mentioned above, the hydrostatic transmission consists of auxiliary machinery including a hydraulic pump, a hydraulic motor, pipes, an accumulator and coolers. The rotational power of the rotor drives the hydraulic pump to generate high hydraulic pressure. The pressure travels through the pipes to rotate the motor at a constant speed. The motor rotation drives the power generators to produce electricity. The accumulator absorbs any hydraulic pressure fluctuations. The hydraulic pump is a ring-cam type multistage piston pump, and the hydraulic motor is a crank type piston motor. Figure 1 (c) shows the basic configurations of the pump and the motor. The hydraulic pump consists of main shaft, ring-cam, pistons and cylinders etc. The main shaft is located at center of the hydraulic pump and directly connected to input shaft. The ring-cam is fixed to outer surface of the main shaft. Many pistons and cylinders are arranged radially around the ring-cam. The ring-cam is rotated by input torque, then each piston repeats reciprocating motion along the ring-cam. As a result, the oil pressure in each cylinder is increased. The high-pressure hydraulic oil is supplied to cylinders of the hydraulic motor through piping. In the hydraulic motor, cylinders and pistons are arranged radially around the crankshaft, and the crankshaft is connected to the generator. The crankshaft is rotated by oil pressure, then the motor rotation derives the generator to produce electricity. The basic mechanism of the hydraulic motor is the same as the reciprocating engine which changes reciprocating motion into rotational motion.



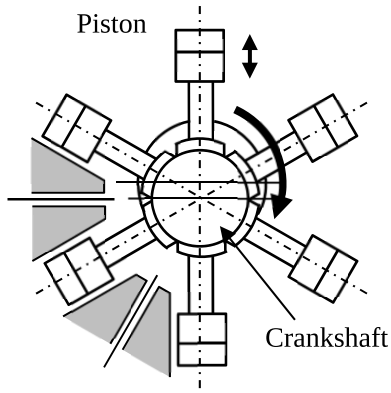
(a) Hydraulic drive train



(b) Conventional drive train (Gear type)



Hydraulic pump



Hydraulic motor

(c) Hydraulic pump and motor

Figure 1: Basic configuration of conventional type and hydraulic drive train for wind turbine generator.

2.2 Advantages of hydrostatic transmission

For example, in wind turbine generator, most of wind turbine manufacturers adopt the gearbox type or the direct drive type drive train systems conventionally. In the gearbox type wind turbine, the turbine rotor is rotated at low speed of about 10 to 15 rpm. On the other hand, the generator has to be rotated relatively high speed of about 1,000 to 3,000 rpm because the generator is connected to the grid. So the gearbox is used to increase the rotor speed to the generator speed. If the wind speed is low, the turbine rotor speed becomes lower than the speed described above in order to utilize the wind energy efficiently. If the wind speed increases, the turbine rotor is rotated at constant speed. Therefore, since the rotational frequency of the generator varies with the wind condition, it is necessary to change the frequency of the generated electricity to the appropriate value by using the inverter/converter.

In the direct drive type wind turbine, the turbine rotor drives the large generator directly without using transmissions. Since the rotational frequency of the generator varies with the wind condition, it is necessary to change the frequency of the generated electricity to the appropriate value by using the inverter/converter.as same as the gearbox type.

In the hydraulic drive type wind turbine, the speed ratio of the hydrostatic transmission can be adjusted by controlling the hydrostatic pressure and displacement. The rotational frequency of the generator is kept constant against the wind condition change. So we can use the versatile brushless synchronous generator at constant rotational speed and enable good power controllability that meets grid requirement without the inverter/converter. Specifically, the displacement of the hydrostatic transmission can be adjusted by changing the number of the cylinders to be used. In consideration of change of the input rotational speed and torque accompanying the wind condition change, the number of the cylinders to be used is adjusted by controlling digital valves attached on each cylinder. The rotational power of the hydraulic pump is transmitted to the hydraulic motor by hydraulic pressure. So, the rotational speed of the hydraulic motor and the generator can be kept constant with no reference to the wind conditions by controlling the oil pressure and displacement. Therefore, we can use inexpensive synchronous generators (using no rare earth materials decreases material sourcing risks). Since the wind condition changes at random rapidly, it was very difficult to enable efficient valve control in real time. This problem was solved by applying the Digital Displacement® technology of a UK venture company, Artemis Intelligent Power, Ltd. and we developed the hydrostatic transmission with variable displacement /1/.

Reliability of the conventional gearboxes and direct drive trains has been improved, but these systems require complete replacement after failure. On the other hand, the hydrostatic transmission is comprised of a lot of small hydraulic cylinders, so the redundancy of cylinders enables high availability to be maintained if some cylinders fail. Total replacement is not required and the hydraulic components are partially exchangeable. Moreover, even if the larger capacity is required, modular design concept of the hydrostatic transmission enables easier scale-up in shorter time and at lower development costs. For example, increasing the number of pump and motor stages would increase the output easily.

Advantages of the hydrostatic transmission above mentioned is shown in followings.

1. It becomes easy to maintenance or replace main components because the modular design concept can be applied to the hydrostatic transmission.
2. It becomes easy to scale up the capacity because the modular design concept can be applied to the hydraulic transmission.
3. The drive train configuration can be simplified because the hydrostatic transmission does not need a power converter and other devices.

3 Development of Large Capacity Hydrostatic Transmission for Wind Turbine

3.1 Development process of the large capacity hydrostatic transmission

In order to ensure product competitiveness, it is necessary to develop compact and reliable hydrostatic transmission compared to conventional drive systems. Figure 2 shows the design flow of the hydrostatic transmission. The design flow consists of three modules mainly, that is hydraulic system simulation module, multi-body dynamics (MBD) module and structural and vibration analysis module.

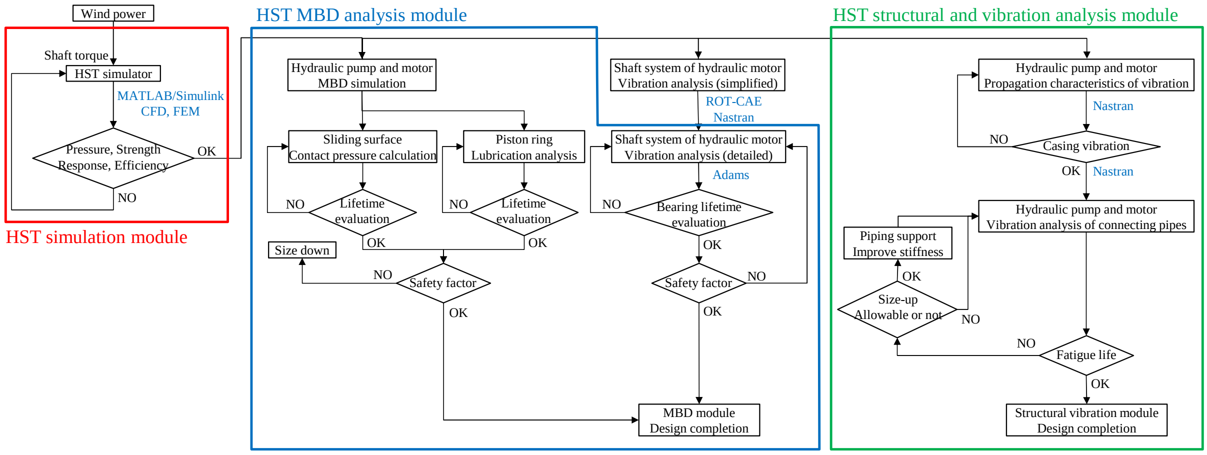


Figure 2: Design flow of the hydrostatic transmission.

At first, hydraulic system simulator was developed for evaluating the dynamic characteristics of the hydrostatic transmission. The simulation for designed hydraulic circuit was carried out with use of MATLAB / Simulink. Figure 3 shows a block diagram of MATLAB / Simulink. The details are omitted here, but following factors are taken into consideration in the model.

- Compressibility of hydraulic oil: pressure dependence of the bulk modulus is considered.
- Operating characteristics of valves: relationship between valve displacement, pressure drop and flow rate are considered with referring the CFD analysis results.
- Effect of oil leakage from cylinders: the amount of oil leakage is estimated with consideration of clearance between piston ring and cylinder bore, gap at sliding surface of hydrostatic bearing with referring the MBD analysis results.
- Others: compressibility of gas in accumulators, pressure drop of piping.

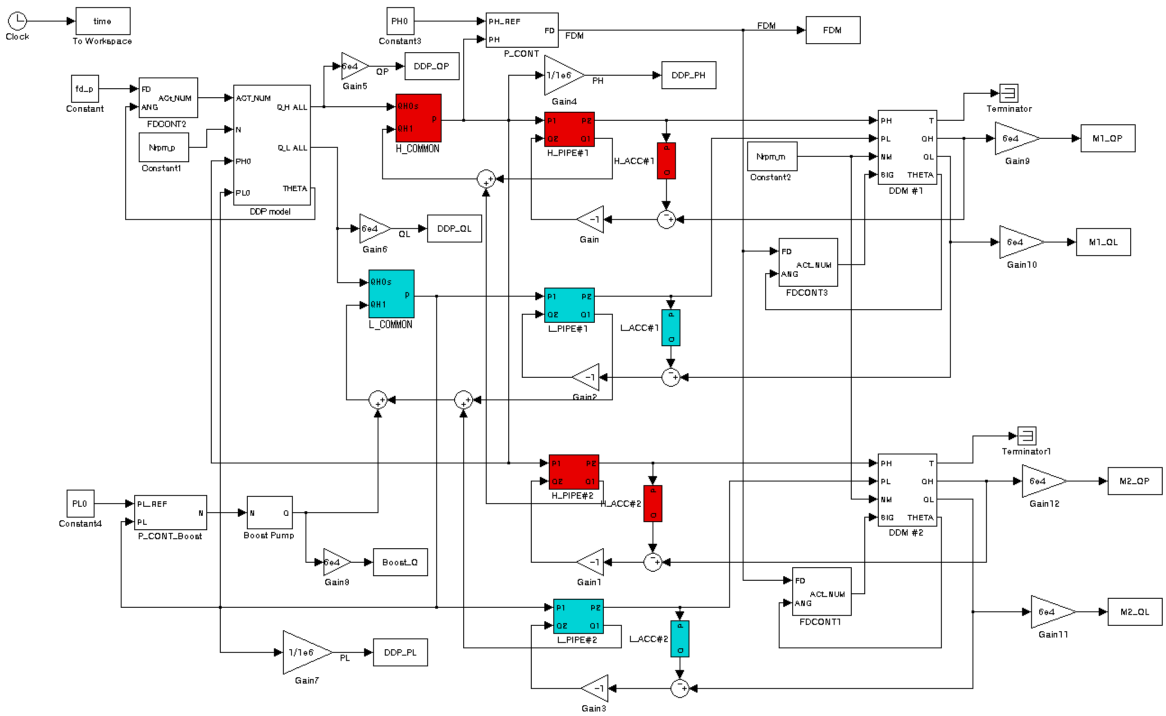


Figure 3: MATLAB/Simulink model of Hydrostatic Transmission.

Next, MBD analysis was conducted to estimate load and dynamic displacement acting on the key components of the drive train system with consideration of the hydraulic response obtained from hydraulic system simulation. Figure 4 shows an outline of MBD model around the piston assembly of the hydraulic pump. Here, piston assembly consists of piston, piston roller, piston ring and hydrostatic bearing. In MBD analysis, cam and roller are coupled by spring element in consideration of the contact stiffness determined by contact analysis. Roller and hydrostatic bearing surface are coupled by spring element in consideration of oil film stiffness. Piston ring and cylinder bore are also coupled by spring elements in consideration of the oil film stiffness. Frictional coefficient at the interface of cam and roller is estimated in consideration of surface roughness and oil film thickness. If roller skew has occurred, the roller end surface is pressed against the roller guide. So, roller end surface and roller guide are coupled by a spring element in consideration of the contact stiffness.

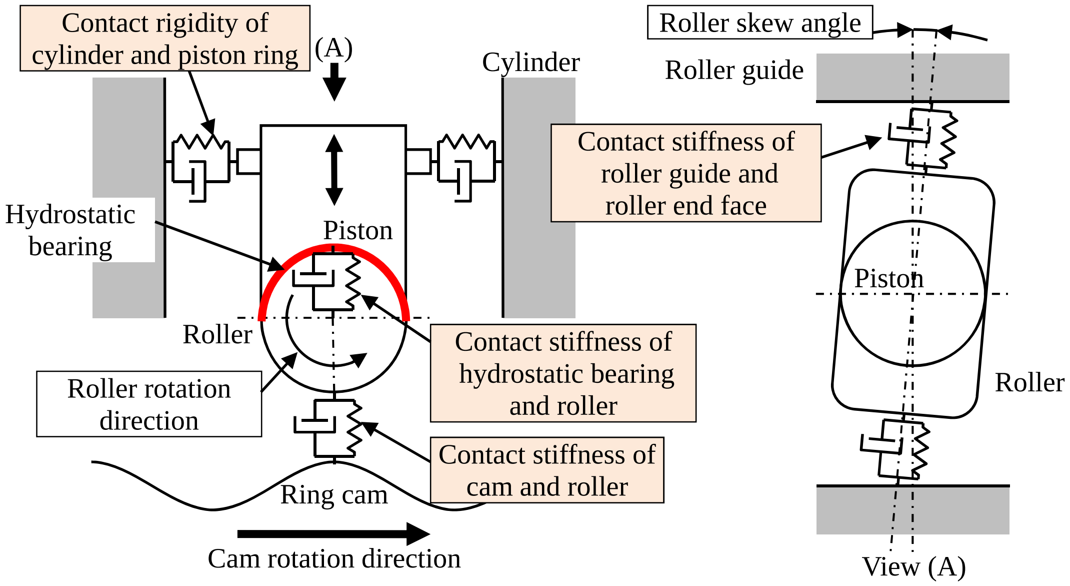


Figure 4: MBD model around the piston of hydraulic pump.

Figure 5 shows an outline of MBD model about the piston assembly and the crankshaft of the hydraulic motor. The piston of hydraulic motor is con rod type piston which has a hydrostatic bearing in sliding surface with crankshaft. The crankshaft is supported by journal bearings in the motor housing. These bearing load acting on crankshaft are estimated with consideration of oil film pressure distribution.

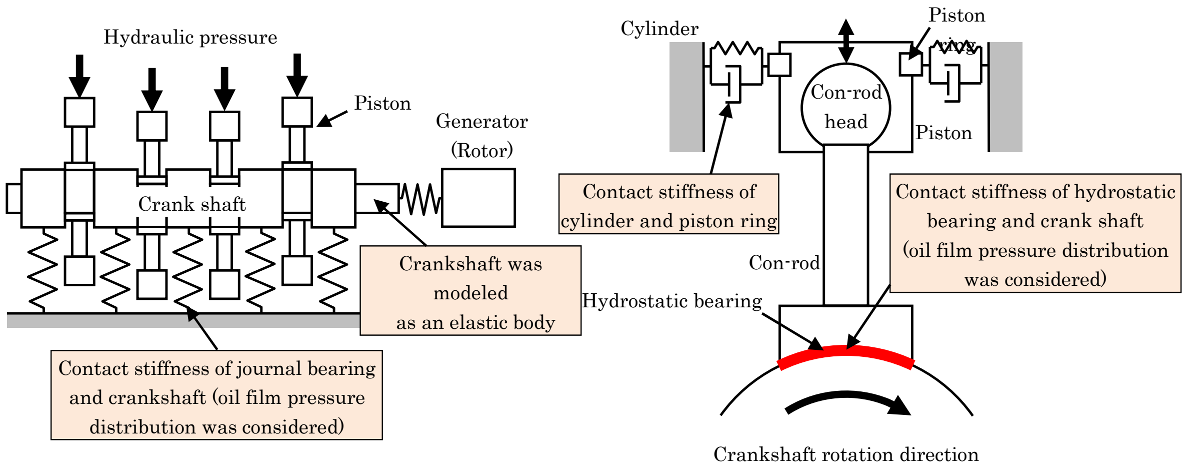


Figure 5: MBD model around the con rod and piston of hydraulic motor.

Subsequently, the reliability of each part is evaluated in consideration of the load and displacement estimated by the MBD analysis explained above. Some examples of analysis results are described below.

In the hydraulic pump, rolling contact fatigue life of cam and rollers is usually estimated under the condition of rolling without relative slippage. However, there is a possibility that slippage between cam and roller will occur because contact load and roller speed are fluctuated according to cam shape. Excessive roller slippage may cause smearing or pitting damage, then rolling contact fatigue life will decrease. Figure 6 shows the analysis results of the slip ratio between roller and cam considering the time-history data of the cylinder internal pressure. As a result, it is found that there is a possibility that the relative slip of about 0.5% occurs between roller and cam. In the design process of the roller and cam, the rolling fatigue life is estimated with consideration of the result of these analysis.

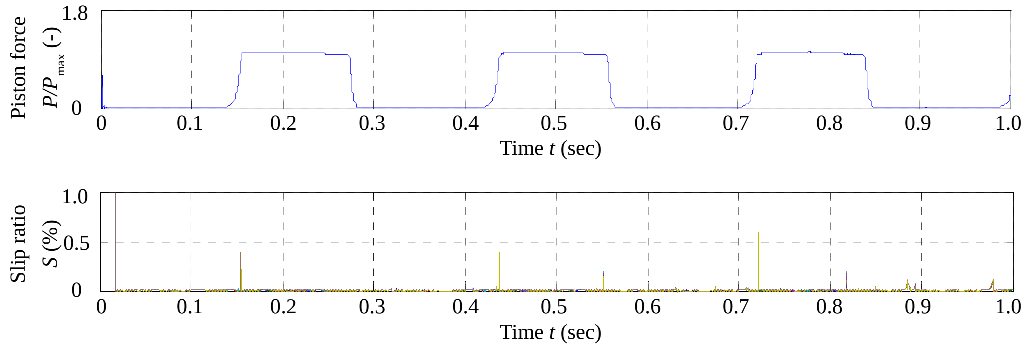
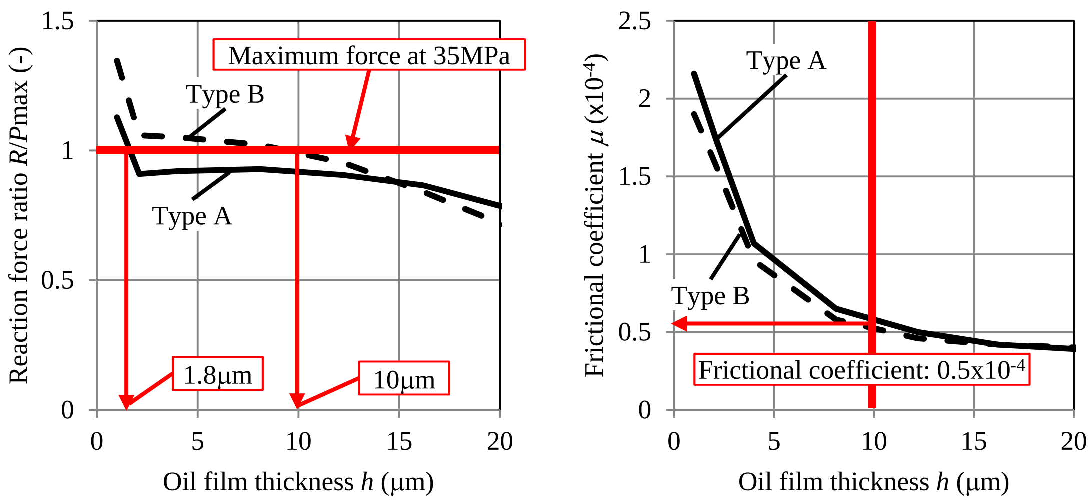


Figure 6: Slip ratio between cam and roller of hydraulic pump calculated by MBD analysis.

In the hydraulic motor, oil film thickness and temperature distribution of the hydrostatic bearing are estimated by EHL analysis considering the bearing force and tilting angle obtained from the MBD analysis. Figure 7 shows the analysis results about two different hydrostatic bearing groove shapes. The relationship between oil film thickness and bearing force is shown in Figure 7 (a), the relationship between the sliding surface friction coefficient and oil film thickness is shown in Figure 7 (b). From Figure 7 (a), it is found that shape B is suitable in lubrication property at maximum load condition compared with shape A. Thus, it became possible to evaluate the validity of the bearing groove design to ensure the necessary oil film thickness.



(a) Reaction force *(b) Friction coefficient*

Figure 7: Oil film thickness and friction coefficient between crankshaft and hydrostatic bearing.

After designing the main components of hydrostatic transmission, shaft vibration analysis of the wind turbine drive train system, housing vibration analysis of the hydraulic pump and motor and structural strength analysis were conducted for evaluating reliability of these systems. After that, experimental verification was conducted with use of the hydrostatic transmission manufactured for existing demonstration wind turbine. In the following, the vibration response analysis of the hydraulic motor shaft is described as an example.

3.2 Drive train dynamic analysis

Dynamic motion of the crankshaft and piston assembly of the hydraulic motor was evaluated by MBD analysis tool, Adams. Analysis results are taken into consideration as boundary conditions in the design of journal bearings supporting crankshaft and flexible coupling connecting the crankshaft to generator. Items to be evaluated in MBD analysis are as follows.

- Dynamic displacement of the crankshaft
- Torsional vibration of the crankshaft

3.2.1 Analysis model

Figure 8 shows an outline of the analysis model. As shown in Figure 8, crankshaft, piston assembly and the generator rotor are modelled in MBD analysis. EHL analysis model and MBD model were coupled in order to estimate oil film stiffness between journal bearings and crankshaft with consideration of the pressure distribution.

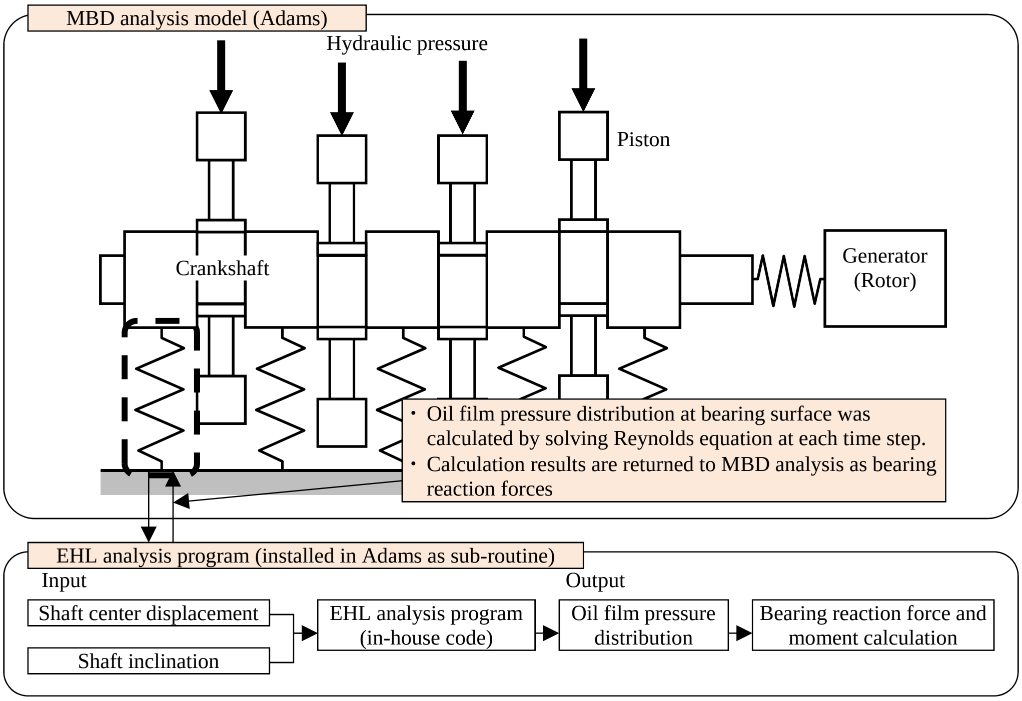


Figure 8: MBD and EHL interaction analysis for journal bearings.

Oil film thickness h at the sliding surface is obtained by equation (1) with use of shaft displacement shaft inclination from MBD analysis.

$$h = c_r\left\{1 + \left(\varepsilon_x + \alpha_y z\right)\cos\theta + \left(\varepsilon_y + \alpha_z z\right)\sin\theta\right\} + \delta + \mathbf{L}\cdot p \tag{1}$$

Oil film pressure distribution is obtained by solving Reynolds equation shown in equation (2) with finite difference method.

$$\nabla\left(\frac{\rho h^3}{12\mu}\nabla p - \frac{\rho U h}{2}\right) = \frac{\partial(\rho h)}{\partial t} \tag{2}$$

Oil film reaction force w is obtained by integrating the oil film pressure at the sliding bearing surface as following equation (3).

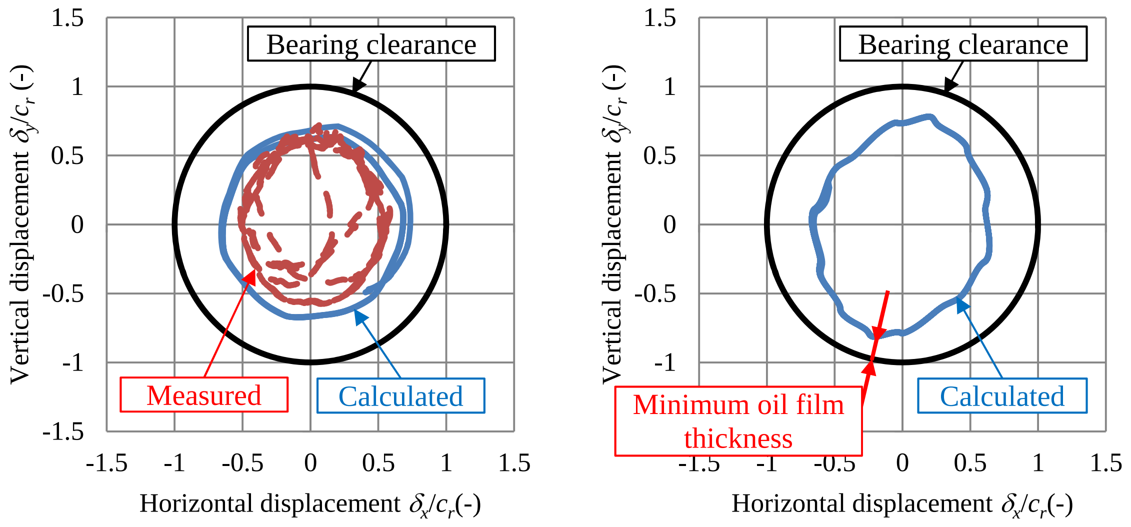
$$w_x = \int_S p\cos\theta\, ds,\ w_y = \int_S p\sin\theta\, ds \tag{3}$$

Bearing reaction force and moment obtained by EHL analysis are returned to MBD analysis as boundary conditions. These calculations are conducted at each time step.

3.2.2 Dynamic displacement of the crankshaft

Figure 9 (a) shows the crankshaft displacement at journal bearing position calculated by MBD analysis for the hydraulic motor testing machine as compared with measurement value. From Figure 9 (a), it is found that the maximum difference between calculation value and measurement value is about 15 % and it is able to estimate the dynamic displacement of the crankshaft within practically allowable accuracy.

Figure 9 (b) shows the calculation results of the crankshaft displacement of hydraulic motor developed for 7MW wind turbine. From Figure 9 (b), it is found that the crankshaft displacement is smaller than bearing clearance and it is able to keep required oil film thickness.



(a) Verification results for testing machine

(b) Calculation results for 7MW hydraulic motor

Figure 9: Crankshaft displacement of hydraulic motors.

3.2.3 Torsional vibration of the crankshaft

Figure 10 shows the crankshaft torque calculated by MBD analysis for the hydraulic motor testing machine as compared with the measurement value. From Figure 10, it is found that the maximum difference between calculation value and measurement value is about 10 % and it is able to estimate the dynamic shaft torque within practically allowable accuracy. The relationship between the coupling stiffness and the torsional vibration was evaluated with MBD analytical model verified above, and these results were applied to coupling design.

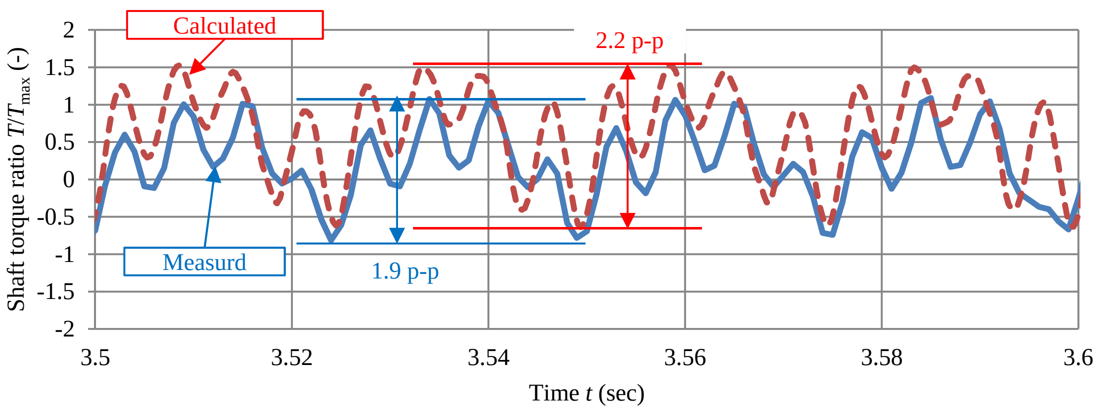


Figure 10: Shaft torque of the hydraulic motor testing machine.

Then, the dynamic shaft torque of the hydraulic motor for 7MW wind turbine was analysed with same manner. Figure 11 shows the calculation results of dynamic response factor of shaft torque. In Figure 11, horizontal axis represents the rate of cylinders to be used, which is almost same meaning as the displacement rate of hydrostatic transmission. As shown in Figure 11, it is found that the dynamic torque increases up to 3.5 times because of resonance with torsional natural frequency and excitation force. Consequently, it was verified that the dynamic torque calculated by MBD analysis is lower than the allowable torque specified by the frictional torque at fitting surface between the motor output shaft and the coupling.

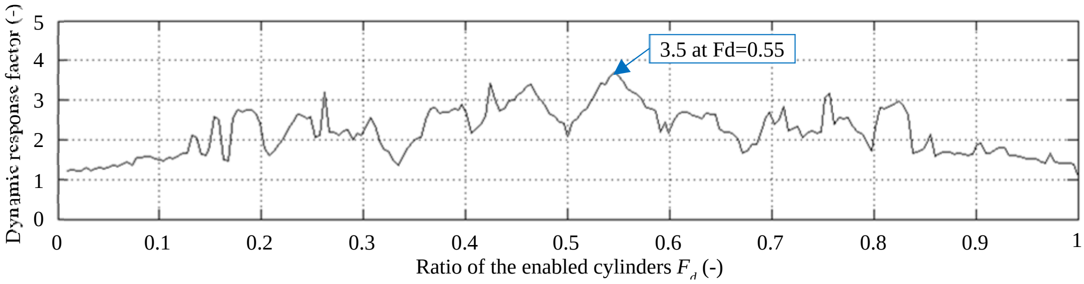


Figure 11: Frequency analysis results of the 7MW hydraulic motor shaft torque.

4 Experimental Verification

The first step is assembling a closed circuit consisting of an existing 1.6 MW hydraulic pump and motor. Next, the circuit is used to simulate the hydraulic drive train to measure the energy transmission efficiency and to check the compatibility of the hydraulic drive train with the drive train of wind turbine. The development steps include elemental tests for components such as the hydraulic piston, valves, cams and rollers. The existing demonstration wind turbine is used to verify the operation of the hydraulic drive train at MHI Yokohama Dockyard & Machinery Works. The hydraulic drive train was downsized from 7 MW, and underwent a response test and a load test by simulating wind turbine input to the hydraulic pump at the Machinery Works. After the tests the hydraulic drive train was installed in the demonstration wind turbine. The world's first wind turbine with a hydraulic drive train started operation in January 2013.

7 MW hydraulic drive train was also assembled and underwent an operation test and a load test using simulated input. Figure 12 shows the nacelle with the 7 MW hydraulic drive train consisting of 7 MW hydrostatic transmission (a 7 MW hydraulic pump and two 3.5 MW hydraulic motors) and two 4.2 MVA induction generators. As shown in Figure 13, it has succeeded in starting synchronization at rating output in operating tests. After the verification test about reliability, the nacelle will soon be shipped to the 7 MW wind turbine model construction site in the UK.

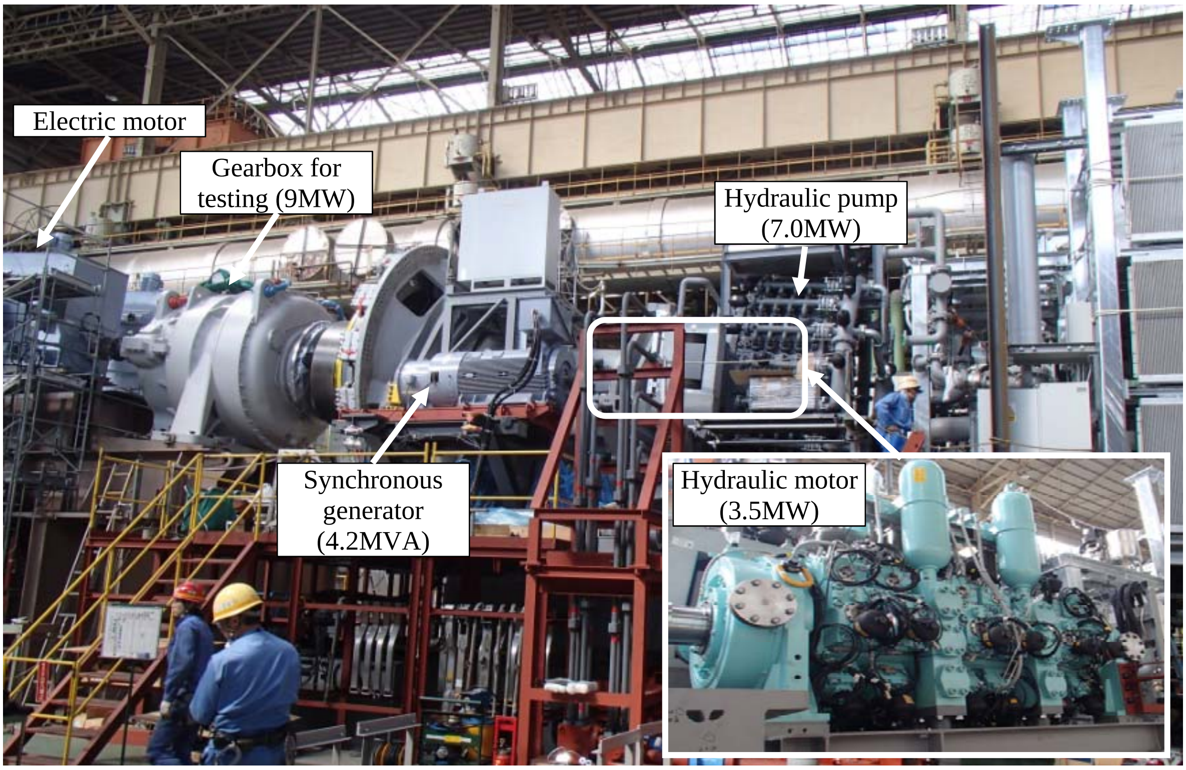


Figure 12: Hydraulic drive train for 7MW wind turbine.

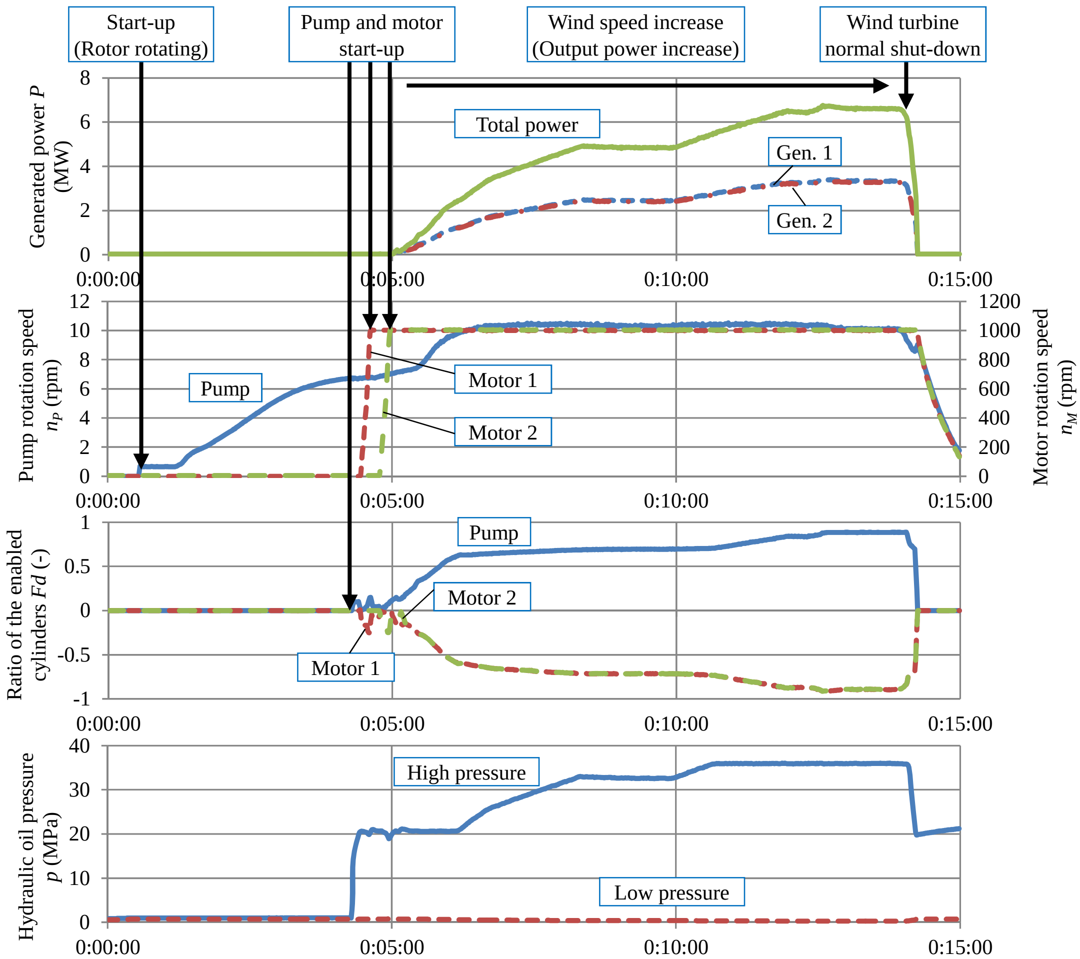


Figure 13: Time-history data of generated power of 7MW drive train at operating test.

5 Summary and Conclusion

In this paper, we introduced the large capacity hydrostatic transmission with variable displacement as substitution of the conventional systems, such as gearbox and so on. The hydrostatic transmission consisting of hydraulic pump and motors was developed based on the Digital Displacement ® technology of Artemis Intelligent Power, Ltd. and we attempted to use the hydrostatic transmission as a drive train system for a wind turbine.

Consequently, we succeeded in the development of large capacity hydrostatic transmission over 7 MW and starting synchronization at rating output in the operating test. Subsequently, durability test is ongoing at the Machinery Works. After the verification test about reliability, the nacelle will soon be shipped to the 7 MW wind turbine model construction site in the UK.

We wish to thank NEDO (Japan New Energy and Industrial Technology Development Organization) for their support in developing the 7 MW wind turbine.

Nomenclature

Variable	***Description***	***Unit***
c_r	Bearing Initial Radial Clearance	[mm]
h	Oil film thickness	[mm]
$\mathbf{L}$	Compliance Matrix	[-]
p	Oil film pressure	[MPa]
t	Time	[sec]
U	Velocity	[mm/sec]
z	Shaft Axial Position	[mm]
α	Shaft Inclination Angle	[rad]
ε	Shaft displacement	[mm]
η	Viscosity Coefficient	[Pa·sec]
μ	Friction Coefficient	[-]
ρ	Oil Density	[g/cm^3]

References

/1/ Taylor, J., Rampen, Win., Robertson, A. and Caldwell, Niall. Digital Displacement ® Hydraulic Hybrids – Parallel Hybrid Drives for Commercial Vehicles. In: *Society of Automotive Engineers o Japan Annual Congress in Yokohama*. 2013.

/2/ Umaya, M., Noguchi, T., Uchida, M., Shibata, M., Kawai, Y. and Notomi, R. Wind Power Generation - Development Status of Offshore Wind Turbines -. In: *MHI Technical Review*. Vol. 50, No. 3, pp. 29-35, 2013.