Conceptual design of cooling system for superconducting wind turbine generator

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Background

The wind energy sector is one of the main energy sources concerned with environment and energy.
However a particular trend is increasing turbine ratings, this trend causes the increase of the nacelle weigh.
To solve this problem, superconducting wind turbine generators have the potential to provide a compact and light weight drive train.

Objectives

We propose a method of supplying a coolant in the cooling channel of the superconducting rotor by circulation pumps built in the rotor and rotating-stationary heat exchangers placed in the rotor to separate the refrigerant of the stationary system and the rotational one.

Problems

☆Nevertheless of these advantages, the cryogenic system has some problems. Because the use of a stationary refrigerator requires that a means be provided for the transfer of cooled helium gas from the stationary supply to the rotating field winding and for return of the gas from the rotor to a stationary reference frame.

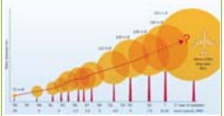
☆ It was possible by using the centrifugal force due to high speed rotation, so-called self -pumping effect that causes the refrigerant circulation. However the wind turbine speed is later as two orders of magnitude than normal generators or motors, the superconductors of the rotor must be force-cooled by cooling channel.

☆ Moreover it is necessary to supply a high pressure helium gas, the sealing technology of high pressure cryogenic refrigerant has not been established yet.

Merits

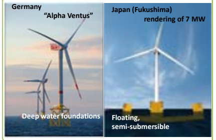
Recent trends in wind generation

Larger wind turbine



Why?
- ➢ Larger electricity generation
  (Power generation) ∝ (Rotor diameter)^2
  while (Interval of turbine) ∝ (Rotor diameter)
- ➢ Improvement of capacity factor
  Higher altitude, higher wind speed

Offshore



Why?
- ➢ Better wind speeds
- ➢ Space scarcity for installation of onshore (opposition to construction is much weaker)

Japan's Exclusive Economic Zone (EEZ) is the 6th largest in the world.

Impact of introducing large generator

- Trial calculations assuming replacement of 1.5MW wind turbines with larger capacity turbines. (Bay area with prevailing winds)

	1.5 MW (datums)	2.5 MW	5 MW	10 MW
Rotor diameter (D)	70 m	90 m	129 m	180 m
Interval of turbines (3D)	~ 210 m	~ 270 m	~ 387 m	~ 540 m
Number of turbines	17	12	9	6
Total output	25.5 MW	30 MW	45 MW	60 MW
Initial costs	5.1 billion Yen	6.0 billion Yen	9.0 billion Yen	12.0 billion Yen
Capacity factor	28.7%	29.8% X 1.7	34.8% X 1.5	40.2%
Annual generation	64.1GWh	78.4 GWh	137 GWh	211 GWh
Annual income (10 円／kWh)	0.6 billion Yen	0.8 billion Yen	1.4 billion Yen	2.1 billion Yen
Profit for 20years	7.7 billion Yen	9.7 billion Yen	18.4 billion Yen	30.2 billion Yen

Hachiryuu wind farm (Akita, Japan)
17 wind turbines along 2.9 km coast

Problems of Large capacity wind turbine

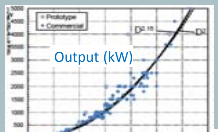
Mass of large capacity wind turbine

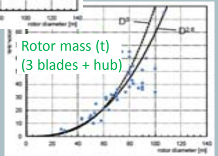
- Output ∝ $D^{2.1}$
- Rotor mass ∝ $D^{2.6}$, Head mass ∝ $D^{1.95}$

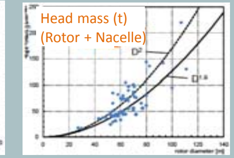
D: Rotor diameter

- Estimated size of 10 MW wind turbine
  - D ~ 180 m
  - Head mass ~ 400 t

Feasible ?

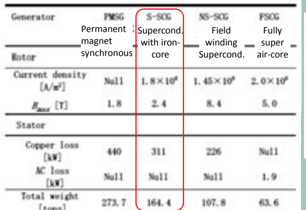




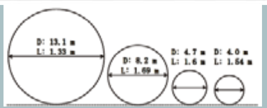


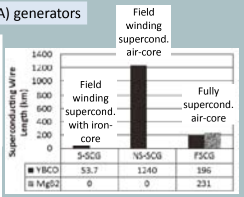
Comparison of generators

Design of 10 MW (3.3 kV, 1.75 kA) generators

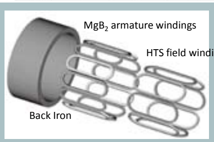


Generator	PMSG Permanent magnet synchronous	S-SCG Supercond. with iron-core	NS-SCG Field winding Supercond.	FSCG Fully super air-core
Rotor				
Current density [A/m²]	Null	1.8×10^8	1.45×10^8	2.0×10^8
B_{max} [T]	1.8	2.4	8.4	5.0
Stator				
Copper loss [kW]	440	311	226	Null
AC loss [kW]	Null	Null	Null	1.9
Total weight [tons]	273.7	164.4	107.8	63.6





	S-SCG	NS-SCG	FSCG
■ YBCO	53.7	1240	196
▨ MgB2	0	0	231



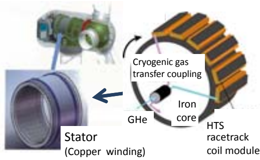
Fully superconducting generator

Methods

Components of HTS-G to be developed

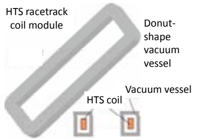
- **HTS racetrack coil module**
- **Cryogenic gas transfer coupling**
- **Highly reliable refrigerator**

Compressor for cryogenic gas



Stator
(Copper winding)

Rotor
(HTS coil modules and iron core)



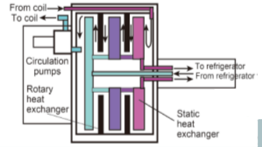
Rotational-Stationary Heat Exchangers

Design conditions
・Stationary side Inlet temperatureT_{si} : 25 K, Inlet pressureP_{si} : 2.1 atm
・Rotational side Inlet temperaruteT_{ri} : 40 K, Inlet pressureP_{ri} : 1.0 atm

・Stationary side outlet temperatureT_{so} : 35 K, Outlet pressure P_{so} : 2.1 atm
・Rotational side outket temperatureT_{ro} : 30 K, Outlet pressureP_{ro} : 1.0 atm

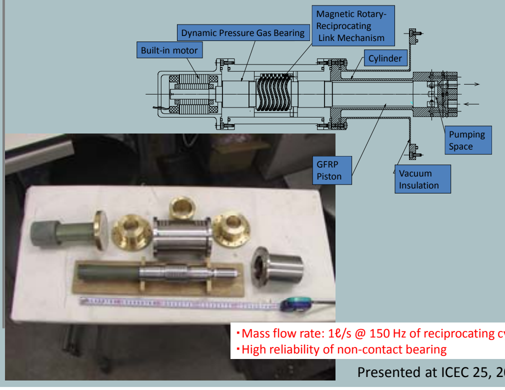
・Heat exchange amount Q : $C_{ps}m_s(T_{si}-T_{so})=C_{pr}m_r(T_{ri}-T_{ro})$= 700 W

m_s = 13.5 g/s （Mass flow rate of refrigerator）
m_r = 13.5 g/s （mass flow rate of pumps 1.1ℓ/s × 8 pumps × 1.5 kg/m^3 @ 35 K）
C_{ps} ~ C_{pr} = 5.2 kJ/kg/K @ 25〜40 K



Results

Circulation Pump

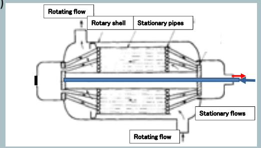


・Mass flow rate: 1ℓ/s @ 150 Hz of reciprocating cycle
・High reliability of non-contact bearing

Shell & Tube Heat Exchanger

・Heat transfer rate $Q = UA\,(T_h-T_c)$=700 W
$1/U = 1/(1/h_h + \delta/\lambda_w + 1/h_c)$
・Overall heat transfer coefficient U = 1.9 W/m^2/K
・Heat transfer surface area $A = Q/U/\Delta T$= 700/1.9/5= 74m^2
(Surface area of pipes in 1.2 cm d$_o$: 37 m^2) → Fins or baffle plates are necessary to increase surface area
・Heat transfer coefficient of flow circular inside tube (Re <2300)
$Nu = 3.657+0.0668Gz/(1+0.04Gz^{2/3})$
Graetz number $Gz = RePr\,(d_i/L)$
$h = Nu\,\lambda_b/d_i$

・Pressure drops $\Delta P = 2f\rho v^2\,d/L$
Friction coefficient $f = 16/Re$
Stationary side ΔP_c =0.075 (Pa)
Rotational side ΔP_h =0.026 (Pa)



・Weight of heat exchanger
Weight of aluminum pipes : 34.6cm^3 × 2.7 × 1,000〜 93 kg
SUS Shell（D$_o$:1m, L:1.5 m, t:2mm) : 12,600cm^{3}x7.8
〜100 kg

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