

Effect of Main Compressor Design on the Off-Design Operation Range of sCO₂ Recompression Cycle

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1. Introduction



Figure 1 Ever Ace A-class 24000 TEU container ship

In response to mounting concerns over anthropogenic climate change, the International Maritime Organization (IMO) has progressively introduced policies aimed at reducing CO₂ emissions from the maritime sector [1]. Civilian nuclear-powered vessels were developed and placed into commercial service as early as the late 1950s, demonstrating the technical feasibility of nuclear energy as a potential power source for future carbon-neutral marine propulsion systems [2].

A key operational difference between marine nuclear propulsion and stationary nuclear power generation is the frequent variation in load. Nuclear-powered ships require particular attention to the off-design performance and operating range of their power conversion systems [3]. sCO₂ Brayton cycle power conversion system may offer advantages for marine propulsion because its compact turbomachinery has relatively low rotational inertia, making it better suited to the rapid load-following requirements of ships than a conventional steam Rankine cycle [4].

This study investigated how a group of main-compressor designs, all of which satisfied the target design-point efficiency of an sCO₂ recompression cycle, differed in their off-design operating ranges. The main compressor was parametrically designed by varying the rotational speed, inlet hub diameter, and inlet axial velocity using a one-dimensional mean-line method combined with loss models. After the candidate designs were screened, quasi-steady-state analyses were performed to evaluate their off-design performance and operating ranges. At a fixed rotational speed and the minimum inlet hub diameter, increasing the design inlet axial velocity substantially expanded the operable range

at low turbine bypass ratios. The results also showed that the restricted operating range of compressors designed with low inlet axial velocities was attributable to the compressor-side quasi-steady stability limit.

2. Methods and Results

2.1 Design Condition of the System

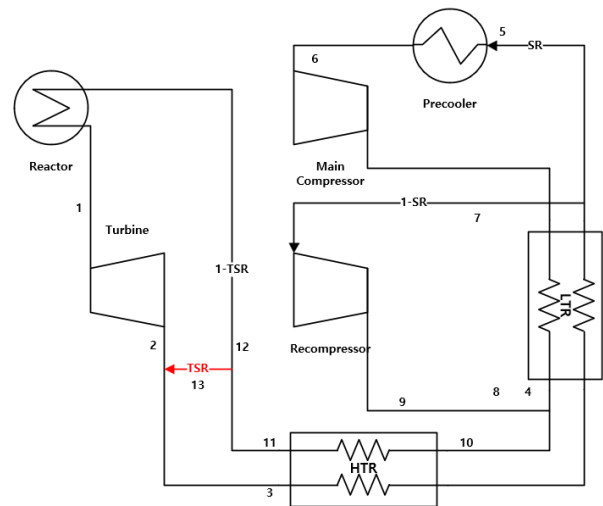


Figure 2 Recompression cycle with turbine bypass

Table 1 Cycle design parameters and results

Power output	75 MWe
Turbine inlet pressure	20 MPa
Turbine inlet temperature	600 °C
Turbine efficiency	0.92
Compressor inlet pressure	8.2 MPa
Compressor inlet temperature	35 °C
Compressor efficiency	0.84
Recompressor efficiency	0.82
HTR effectiveness	0.92
LTR effectiveness	0.92
Pressure drop across PCHE	150 kPa
Split ratio	0.67
Cycle efficiency	0.43
Mass flow rate	922.8 kg/s

The design results of the recompression cycle are summarized in Table 1. The rated output of 75 MWe was selected based on the maximum power output of the WinGE 92-B engine used in a 24,000-TEU container

ship, with additional allowance for auxiliary power consumption. The turbomachinery was designed for one-quarter of the total cycle mass flow rate. This configuration was adopted with reference to Section 5.9.7 of IMO Resolution A.491(XII), *Code of Safety for Nuclear Merchant Ships*, which requires the main electrical system to be divided into at least two independent systems [5]. Furthermore, when a double-suction, double-discharge configuration is employed to reduce the axial thrust acting on the rotating machinery, each turbomachinery unit handles one-quarter of the total cycle flow rate.

2.2 Main Compressor Screening

The turbomachinery design was performed using the KAIST Turbomachinery Design (TMD) code. It is a design model which combines 1D mean line model with sCO₂ fitted loss models. Among the compressor design-point parameters, the mass flow rate, inlet temperature, inlet pressure and outlet pressure were specified based on the closed cycle design results, while parametric designs were conducted by varying the rotational speed, inlet axial velocity, and hub diameter. The minimum shaft diameter required to withstand pure torsional loading was first calculated by applying a safety factor to the shaft power. The minimum hub diameter was then determined by multiplying the minimum shaft diameter by a hub-to-shaft diameter ratio of 1.3 [6, 7]. In addition, assuming that the compressor was subjected solely to centrifugal stress, the candidate designs were screened to retain only those hub diameters for which the impeller tip speed—calculated from the compressor outlet tip diameter obtained using KAIST TMD and the rotational speed—remained below the limit.

$$\tau = 9550 P_{shaft} (kW) / RPM (N \cdot m)$$

$$\min(D_{shaft}) = \left(\frac{16\tau \times \text{Safety Factor}}{\pi \sigma_{yield}} \right)^{1/3}$$

$$\min(D_{hub}) = 1.3 \min(D_{shaft})$$

The design space for the main compressor of the recompression cycle covered rotational speeds ranging from 3,000 to 24,000 rpm and inlet axial velocities ranging from 2 to 50 m/s. For each combination of rotational speed and inlet axial velocity, calculations were performed by increasing the hub diameter from its minimum value to the maximum allowable value in increments equal to 0.25 times the minimum hub diameter. The resulting designs were then screened based on whether they satisfied the target compressor efficiency. At a given rotational speed and inlet axial velocity, the compressor efficiency at the design point was found to decrease monotonically as the hub diameter increased. Among the screened designs, the case with the minimum hub diameter at 7000 rpm was selected for the off-design performance analysis because the selected rpm condition yielded the largest number of inlet axial

velocity–hub diameter combinations that met or exceeded the target compressor efficiency.

2.3 Off Design Performance Analysis

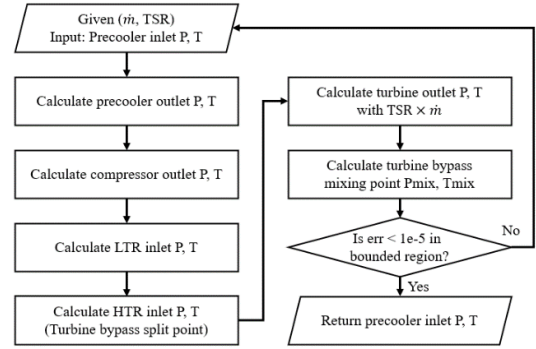


Figure 3 KAIST QCD Algorithm flow chart

The off-design performance analysis was conducted using the KAIST QCD code, a quasi-steady-state cycle analysis tool. Quasi-steady-state analysis evaluates off-design performance by considering only the new steady-state operating condition to which the cycle converges under a given set of boundary conditions. Owing to its relatively low computational complexity and short calculation time, this approach is well suited for developing control strategies and performing sensitivity analyses of the effects of individual components on cycle performance. Unlike transient analysis, however, it cannot evaluate time-domain behavior or dynamic control performance. In this study, inventory control and turbine bypass control were employed for load regulation because inventory control generally provides higher off design cycle efficiency than turbine bypass control or rotational-speed control at the same net power output [8]. The off-design performance was held by fixing other components - high temperature recuperator (HTR), low temperature recuperator (LTR), precooler, turbine and recompressor -, except for the main compressor. The rotational speed of each turbomachinaries were held in the designed value.

2.4 QCD Result

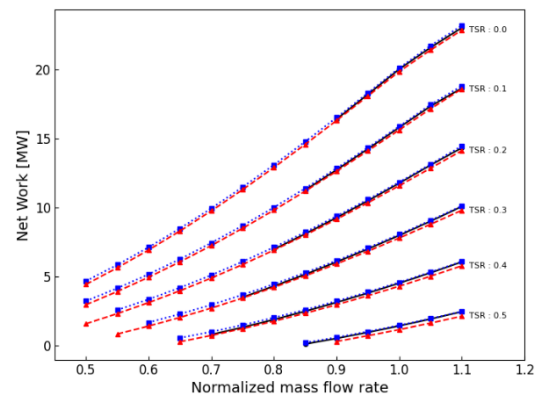


Figure 4 Net power output of along normalized mass flow rate and turbine bypass split ratio (TSR)

Vin=18m/s (black), Vin=22m/s (blue), Vin=34m/s (red)

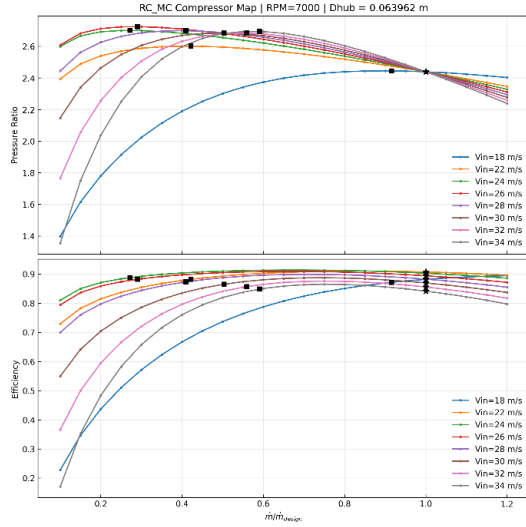


Figure 5 Main compressor performance curve at 7000 RPM (black square: surge point, black star: designed point)

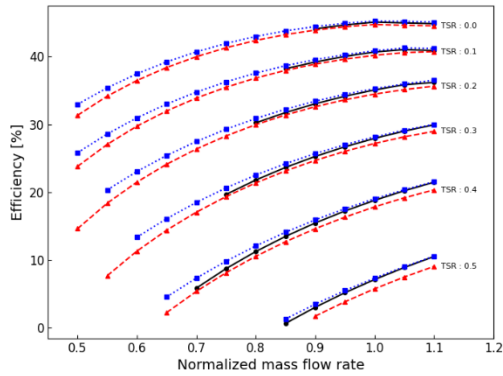


Figure 6 Cycle efficiency of along normalized mass flow rate and turbine bypass split ratio (TSR)
Vin=18m/s (black), Vin=22m/s (blue), Vin=34m/s (red)

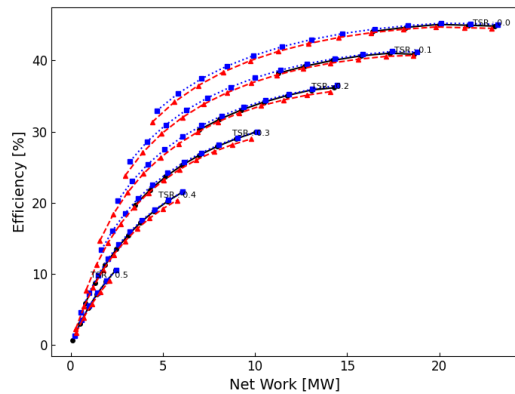


Figure 7 Cycle efficiency of along net power output and turbine bypass split ratio (TSR)
Vin=18m/s (black), Vin=22m/s (blue), Vin=34m/s (red)

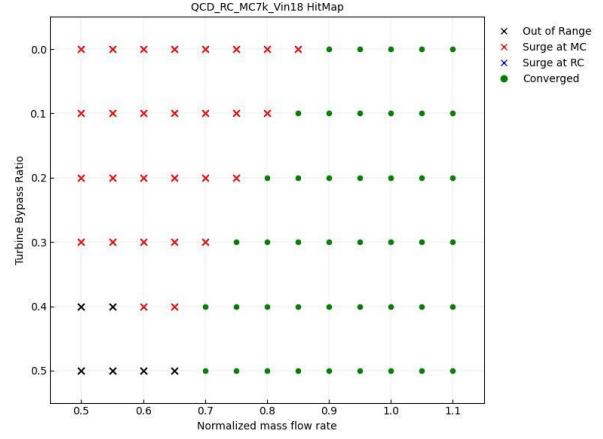


Figure 8 RPM=7000, Vin=18m/s Convergence map

Figure 4 presents the net power output as a function of the normalized mass flow rate and TSR at 7,000 rpm. The differences in efficiency among the compressor designs are negligible because, as shown by the main-compressor performance curves in Fig. 5, the efficiency characteristics as a function of mass flow rate do not differ significantly over the examined operating range. Figures 4, 6, and 7 show that compressors designed with higher inlet axial velocities have wider operating ranges at low TSR values. Figure 8 presents a map of the operating points at which the QCD calculations converged or failed to converge for the compressor designed with an inlet axial velocity of 18 m/s. The results indicate that the off-design operating range of this main compressor at low turbine bypass ratios is limited by main-compressor surge. Figure 7 also shows that compressors designed with higher inlet axial velocities can achieve greater power reductions at low TSR values, thereby taking advantage of the high off-design efficiency provided by inventory control.

However, an increase in the design inlet axial velocity should not be interpreted as direct surge suppression. Surge is a coupled compressor-system dynamic instability, whereas QCD cannot capture the effect of the duct/plenum inertance and compliance [9, 10]. Thus, the present low-flow boundary is more appropriately interpreted as a compressor-side quasi-steady stability limit. In addition, because the design volumetric flow rate, hub diameter, and rotational speed are fixed, increasing the design inlet axial velocity reduces the inlet annulus area and eye-tip diameter, changes the eye-tip speed and inlet velocity triangle. Therefore, inlet axial velocity is not an isolated aerodynamic variable. Consistently, Romei et al. showed that compressor rangeability is not a monotonic function of inlet flow coefficient across different sCO₂ design families [11].

3. Conclusions

This study evaluated the effect of main-compressor design on the off-design operating range of a 75-MWe sCO₂ recompression cycle for marine propulsion. Candidate compressors satisfying the design-point

efficiency target were generated by varying rotational speed, inlet hub diameter, and inlet axial velocity. At 7000 rpm and the minimum hub diameter, increasing the design inlet axial velocity produced little difference in cycle efficiency but substantially extended the operation range at low turbine bypass ratios. This enabled greater power reduction while retaining the off-design efficiency advantage of inventory control. The resulting low-flow boundary should be interpreted as a compressor-side quasi-steady stability limit rather than direct suppression of dynamic surge, because QCD does not represent duct and plenum dynamics. The range extension therefore reflects the combined effects of inlet axial velocity and the associated changes in compressor geometry and inlet velocity triangle.

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