Design Methodology of Supercritical CO_2 Brayton Cycle Turbomachineries

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1. Introduction

The Supercritical CO_2 Brayton Cycle (S-CO_2 Cycle) has been gaining substantial interest for applying the technology to the next generation nuclear systems due to its high efficiency and compactness. The cycle gains high efficiency owing to low compressor work which originates from non-linear property variation of CO_2 near the critical point. Therefore, the design of a main compressor becomes important to ensure high efficiency of the cycle [1]. In this paper, the design methodology of S-CO_2 Cycle turbomachineries will be briefly discussed in the viewpoint of non-linear property variation of the fluid.

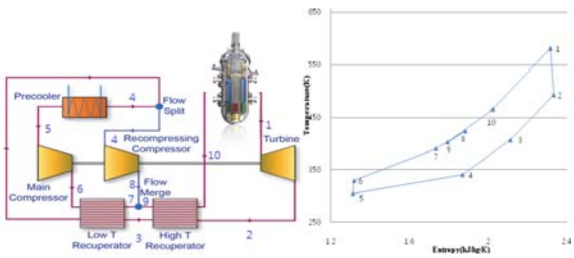

Fig. 1. S-CO_2 Recompressing Cycle layout and operating points

2. Design Methods

Existing generally accepted design methodologies of turbomachineries usually assume either ideal incompressible fluid (water case) or ideal gas (air case). However, in the S-CO_2 Cycle the non-linear property variation becomes significant and therefore, both assumptions can lead to erroneous results.

2.1 Stagnation-to-Static

When designing turbomachineries, both stagnation and static conditions of fluid are equally important. This is because many state variables (e.g. enthalpy, entropy, etc.) are based on the stagnation conditions while fluid properties (e.g. density, specific heat, etc.) are based on the static conditions. The stagnation enthalpy is summation of static enthalpy and the fluid kinetic energy (Eq.(1)).

$$h_o = h_s + \frac{V^2}{2} \qquad (1)$$

The stagnation condition is defined when the fluid at rest (at static condition) is adiabatically and reversibly (isentropic) accelerated to velocity V. For an ideal gas assumption following relations hold between stagnation (subscript o) and static conditions (subscript s).

$$\frac{T_o}{T_s} = 1 + \frac{V^2}{2C_p T_s} = 1 + \frac{\gamma - 1}{2} M^2 \qquad (2)$$

$$\frac{P_o}{P_s} = \left(\frac{T_o}{T_s}\right)^{\frac{\gamma}{\gamma-1}} = \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{\gamma}{\gamma-1}} \qquad (3)$$

where, T: temperature, P: pressure, C_p: specific heat, V: flow velocity, γ : ratio of specific heats, M: Mach number.

However, to reflect real gas properties with high non-linearity variation, equations (2) and (3) are not applicable [2]. This is because the specific heat ratio is not a constant near the critical point as it is shown in Fig. 2. Thus, the static condition should be calculated from the stagnation condition based on the original definition with property package which is graphically introduced in Fig. 3.

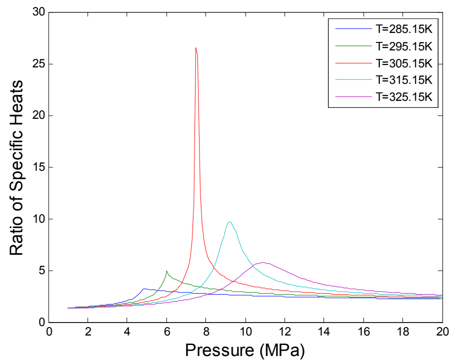

Fig. 2. Ratio of specific heats variation near critical points

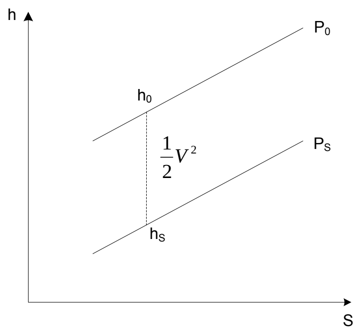

Fig. 3. Finding static condition from stagnation condition using property package

2.2 Axial and Radial Turbomachinery Design Methods

The mass conservation and Euler equations are always applicable regardless of ideal gas or real gas conditions. Therefore, the most basic equations for designing turbomachineries based on 1-D mean line analysis are the following:

$$\dot{m} = \rho(h_s, P_s)AV \tag{4}$$

$$h_{o2} - h_{o1} = U_2V_{\theta 2} - U_1V_{\theta 1} \tag{5}$$

where $\dot{m}$: mass flow rate, *A*: flow area, *U*: rotor speed

However, since not all work done to or by fluid are isentropic since some losses are always involved during the process. Regarding the losses in a turbomachinery there are two types of losses that can be defined: Pressure losses and Enthalpy losses. These losses carry the same meaning which indicates how much the process in a turbomachinery departs from the ideal (isentropic) machine. By selecting appropriate sets of loss models for each turbomachinery: (1) axial compressor, (2) axial turbine, (3) radial compressor, and (4) radial turbine, all axial type turbomachineries are based on enthalpy loss models while loss models for radial type turbomachineries are pressure loss models. The difference in these two loss models are shown in Fig.4.

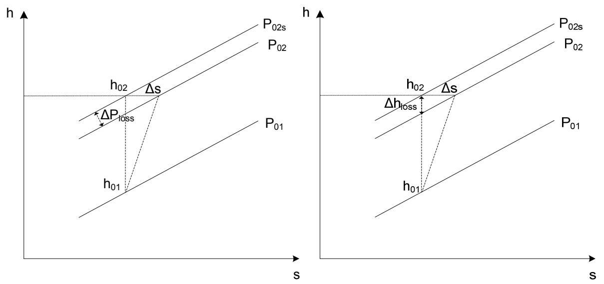

Fig. 4. Applying loss models to identify the irreversibility in axial (left) type and radial (right) type compressors

Based on the abovementioned equations and loss models, a turbomachinery design code that can design and predict off-design performance for both axial and radial type S-CO_2 cycle components was written in MATLAB environment. The flowchart of the Turbo_Design code is shown in Fig. 5.

3. Results & Summary

Fig. 6 summarizes a sample calculation results for 10 stage axial compressor. The designed geometry and the trend of the turbomachinery off-design map are similar to Ref. [2]. Thorough Validation and Verification of the Turbo_Design code will be followed in near future. Furthermore, detail comparison of axial type turbomachineries to radial type turbomachineries in the S-CO_2 cycle will be performed to identify the best suitable type for the cycle.

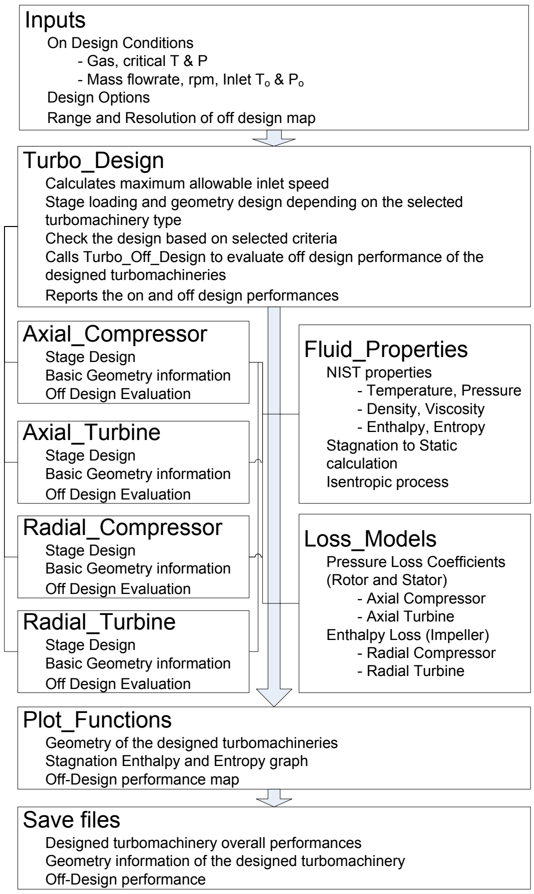

Fig. 5. Flowchart of the Turbo_Design program

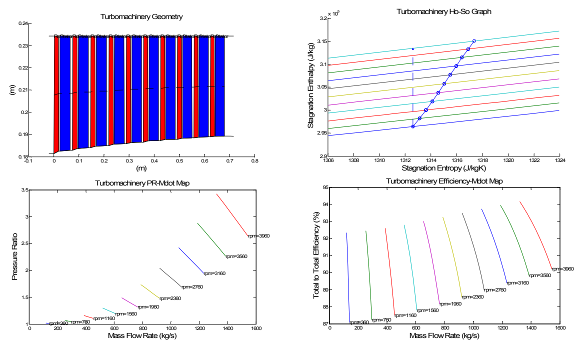

Fig. 6. Sample calculation result for 10 stage axial type compressor

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