Integrally-geared Centrifugal Compressor for Carbon Capture and Storage (CCS) and Long Duration Energy Storage (LDES) Applications

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Abstract

In recent years, as environmental awareness in society has increased, Carbon Capture and Storage (CCS) has been implemented to reduce greenhouse gas emissions. Kobe Steel has developed a high-pressure CO_2 compressor for injecting captured CO_2 into underground reservoirs and conducted pressure tests in a supercritical state using CO_2 gas in its factory. Meanwhile, the charging compressor for the mechanical battery system in Long Duration Energy Storage (LDES), which is expected to undergo technological advancement, faces technical challenges in handling high-temperature gas and operating conditions that require daily start and stop. Kobe Steel is addressing these challenges by combining its own technologies and know-how the company has cultivated to date. This article explains Kobe Steel's product development and approach to technical issues for centrifugal compressors to support its customers by creating a better environment.

Introduction

Carbon capture and storage (CCS) contributes greatly to advancing not only carbon neutrality but also the supply of energy via blue hydrogen and blue ammonia. As such, recent years have seen a surge in efforts toward related plant planning and operations, particularly in the United States and Europe. Additionally, Japan declared a goal in 2020 of achieving carbon neutrality by 2050. Consequently, the Act on Carbon Dioxide Storage Projects (CCS Business Act) was enacted in 2024, with a target for commencement of no later than 2030. It can thus be surmised that CCS projects will become a familiar presence in Japan as well in the near future.

In CCS applications, compressors are primarily used for transport and storage, specifically to pressurize CO_2 to transport it from an emission source via a vessel or pipeline, and to inject it into a reservoir. The equipment and operational costs associated with CCS generally reduce the overall economic efficiency of a plant. Therefore, the compressors in such systems must be not only mechanically reliable, but also economical and highly efficient.

The increasing prevalence of renewable energies such as solar and wind-power generation have placed increased attention on long-duration energy storage (LDES) as a means of storing surplus energy over extended periods. Several processes have been developed since the 2020s, some of which have been marketed commercially. One type of system is mechanical energy storage that uses compressors to store the pressure and thermal energy of gases. Advantages of mechanical systems over battery systems (e.g., lithium-ion batteries) include a high availability of materials for components, longer service life, and greater storage capacity. However, these systems have different considerations from machinery typically used in continuous operation, such as high-temperature gases and frequent start-stop cycles.

This paper first introduces the development details and key technological aspects of Kobe Steel's high-pressure CO_2 compressor designed for injecting CO_2 captured as part of CCS into underground reservoirs. It then covers the technical requirements and challenges for high-temperature compressors used in LDES, along with Kobe Steel's efforts to address them.

1. High-pressure CO_2 compressor for CCS

In Kobe Steel's flagship product, the integrally geared centrifugal compressor (hereafter, integrally geared compressor), the pinion rotors have impellers at the ends and a small gear in the center. The speed of the gear is increased via a large gear (bull gear), as shown in the compressor cutaway model in **Fig. 1**. Thus, the impeller's rotational speed can be adjusted per each pinion rotor by changing the gear ratio. As shown in **Fig. 2**, the efficiency of a centrifugal compressor is characterized based on the specific speed (n_s), which indicates how similar the impellers are in shape.[1] Adapting the impeller diameter and rotational speed for the required head and volumetric flow rate sets an appropriate specific speed, thus optimizing efficiency. Specific speed is defined by rotational speed, volumetric flow rate, head, and gravitational acceleration per Equation (1).

$$n_s = 2\pi N Q^{\frac{1}{2}} / (gH)^{\frac{3}{4}} \quad \cdots\cdots (1)$$

Here, N = rotational speed (1/s)
Q = volumetric flow rate (m^3/s)
g = gravitational acceleration (m/s^2)
H = head (m)

Furthermore, because the integrally geared compressor has independent stages, it is easy to incorporate interstage cooling between stages. This brings operation closer to isothermal compression, reducing power consumption. However, the rotor of this design tends to have a decrease in rigidity because it rotates at high speeds and has impellers at both ends of the shaft. Therefore, a challenge arises if efficiency is prioritized such that very high rotational speeds result, as it becomes difficult to ensure stability against rotor vibration.

Using an integrally geared compressor with these characteristics for high-pressure CO_2 compression in CCS applications presents the following technical challenges:

i) Ensuring accurate prediction of aerodynamic performance accounting for changes in CO_2 properties at high pressure
ii) Ensuring vibrational stability of the rotor and vibrational strength of the impeller blades, which are subjected to fluid forces from high-pressure, high-density CO_2
iii) Ensuring seal performance sufficient to inhibit the leakage of high-pressure CO_2 gas
iv) Reducing life cycle costs

To validate the solutions for overcoming these challenges, Kobe Steel designed and manufactured a prototype compressor and conducted in-plant pressure testing using CO_2 gas.[2)] **Table 1** shows the target specifications for the prototype CO_2 compressor. **Fig. 3** shows the CO_2 compressor and associated equipment in the test facility. The next section describes the results of pressure testing as well as the key technologies of the high-pressure CO_2 compressor (hereafter, prototype unit) that address the aforementioned technical challenges.[2)]

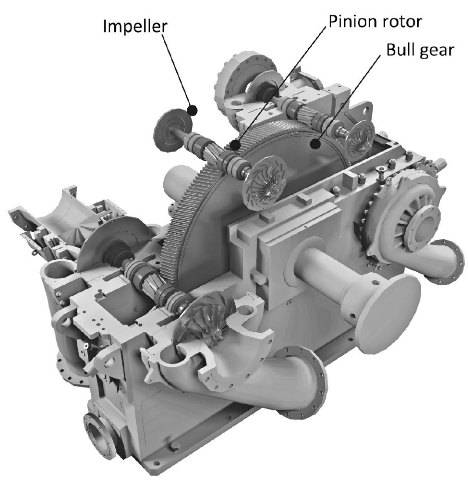


Fig. 1 Section model of integrally geared centrifugal compressor

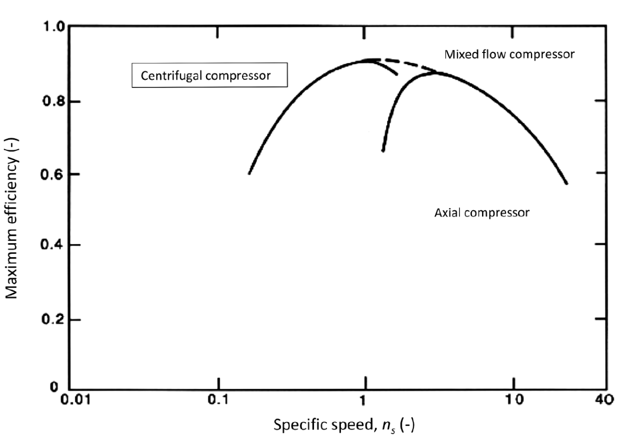


Fig. 2 Specific speed vs maximum efficiency [1)]

Table 1 Specification of development target for high-pressure CO_2 compressor

Application	Prototype
Type	Integrally geared
No. of stage	8 stages
Gas handled	CO_2
Capacity	140,000 kg/h (72,500 m^3/h)
Suction / discharge pressure	0.01 / 20 MPaG
Shaft power	16,600 kW
Driver	Motor (18,500kW)

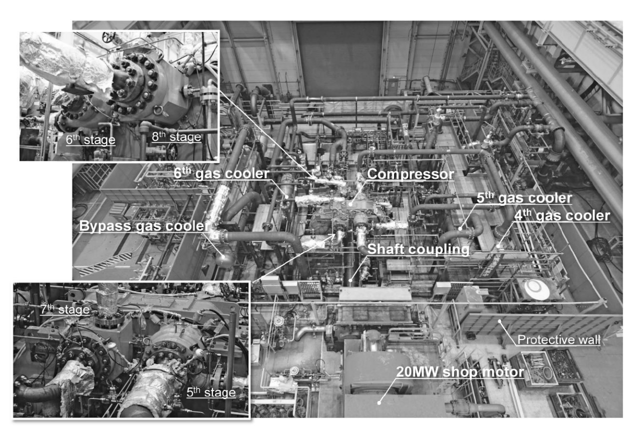


Fig. 3 Overview of high-pressure CO_2 prototype compressor and test facility

1.1 Aerodynamic design and results of performance testing

Key components that define compressor performance include the impeller, which imparts kinetic energy to the gas, and the diffuser, which converts this kinetic energy into an increase in static pressure. The physical properties of CO_2 gas change significantly in the high-pressure supercritical region. Regarding it as an ideal gas can result in a compression ratio that differs greatly from calculated values. Accounting for these changes in the physical properties of the gas in the aerodynamic design yields an impeller design that more accurately achieves target specifications.

Fig. 4 shows the aerodynamic performance test results from pressure testing of the prototype unit. At the specified flow rate (ϕ/ϕ_{design} = 1), the discharge pressure of the final stage (eighth stage) reaches 20 MPaG, validating the aerodynamic design.

1.2 Evaluating rotor design and stability

Hydrodynamic forces generated by the impellers and seals in the rotor package act as destabilizing forces. The stability of the rotor must be sufficient to overcome the resulting unstable vibration. Stability is generally evaluated in terms of the logarithmic decrement of the rotor system, which consists of the rotor and bearings. A positive log decrement indicates a stable rotor system, a negative log decrement an unstable system. **Fig. 5** shows the stability analysis results for the seventh-to-eighth stage rotor of the prototype unit. The horizontal axis shows the destabilizing force (cross coupling) as specified in the widely adopted centrifugal compressor standard API 617.[3] The vertical axis shows the log decrement. As gas pressure increases, power and density also increase, in turn increasing the destabilizing forces acting on the impeller and seals. API 617 requires a design log decrement of 0.1 or greater. However, calculations show that achieving this value is difficult for tilting pad journal bearings (TPJB) because of excessive destabilizing forces. Hence, the labyrinth seal at the impeller suction has a swirl brake with a large number of ribs to suppress swirling flow and thereby suppress destabilizing forces. Additionally, the ISFD (integral squeeze film damper) bearing imparts excellent damping characteristics to the rotor, resulting in a highly stable rotor design.

We conducted excitation tests to measure the rotor's vibration characteristics. **Fig. 6** depicts Kobe Steel's test method for exciting the rotor to evaluate rotor stability. We arranged four electromagnets in the circumferential direction on the front of the impeller. Electromagnetic force during compressor operation applied a bending moment to the rotor via the impeller, inducing excitation perpendicular to the shaft. This setup enabled us to obtain precise data regarding the rotor's damping characteristics. The horizontal axis in **Fig. 7** shows the discharge pressure of the last stage. The vertical axis shows the natural frequency and log decrement of the rotor's first forward rotation mode (1st FW) and second forward rotation mode (2nd FW) measured during the excitation test. The excitation test results confirm

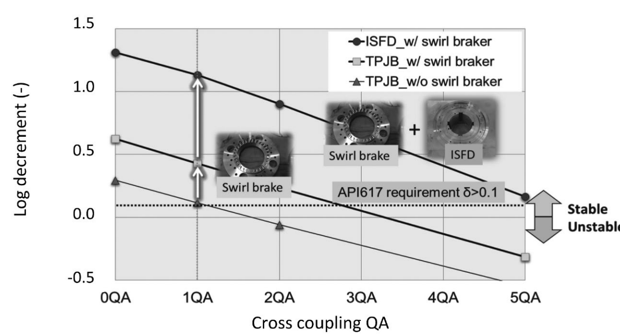


Fig. 5 Result of rotor stability analysis

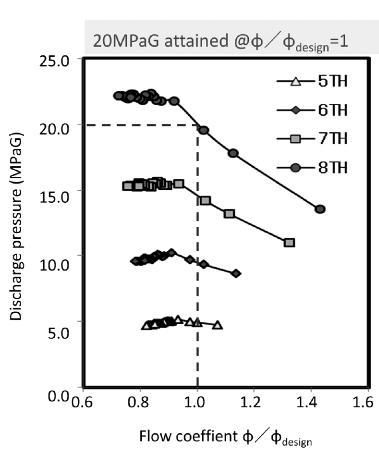


Fig. 4 Discharge pressure of each stage

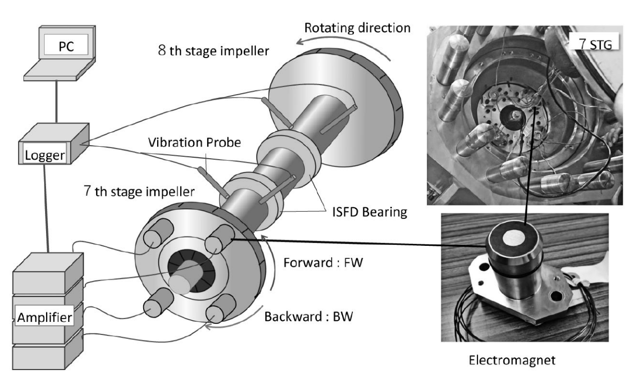


Fig. 6 Developed magnetic excitation system

that the rotor's stability exceeds the log decrement specified in API 617 even under high-pressure conditions.

1.3 Measuring impeller blade vibration and evaluating strength

High-density gas exerts significant forces on the blades of open impellers because they are cantilevered. Increased vibration can cause excessive dynamic stress and potentially lead to failure. Therefore, we performed strength testing by measuring blade vibration in the fifth- and sixth-stage impellers of the prototype unit.

Fig. 8 shows a conceptual diagram of the blade vibration measurement system. An optical displacement sensor on the casing opposite the impeller measures the frequency at which the blades pass the sensor during vibration (dashed line). Analyzing the difference in time (ΔT) as compared with the non-vibrating blades (solid line) indicates the impeller's vibration amplitude (ΔX) and vibration frequency. **Fig. 9** shows the blade amplitude of the impeller at each operating point, with the ratio of the flow rate to the design flow rate on the horizontal axis and the discharge pressure of the sixth stage on the vertical axis. The size of the circle at each point represents the magnitude of the blade amplitude. We used finite element analysis to calculate dynamic stress based on the acquired amplitude. **Fig.10** shows the resulting modified Goodman diagram, with normalized mean stress on the horizontal axis and normalized dynamic stress on the vertical axis. Alongside vibration in the normal operating range, we also calculated surge point stresses, confirming that stress is below the fatigue limit of the impeller material at all operating points and that the load on the impeller is sufficiently low. This validation demonstrates

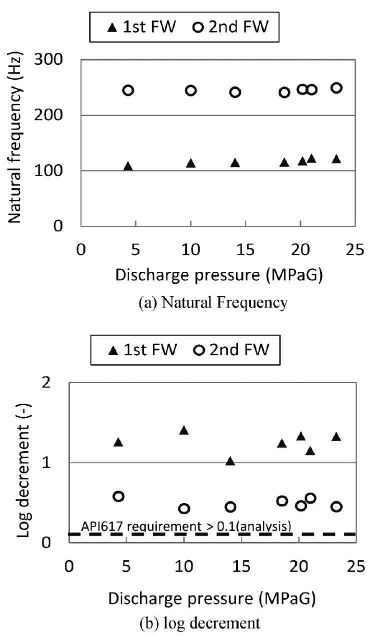


Fig. 7 Measured dynamic characteristics of rotor vs discharge pressure (ISFD bearing)

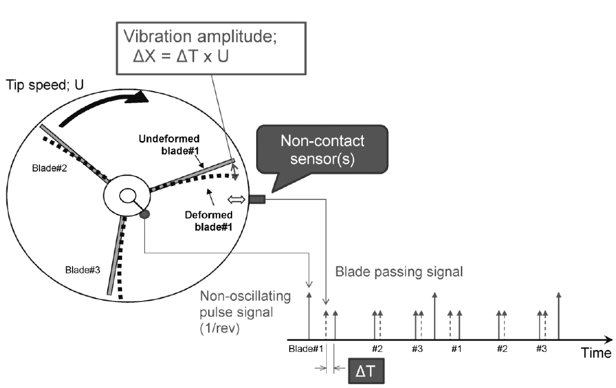


Fig. 8 Schematic of blade vibration measurement system

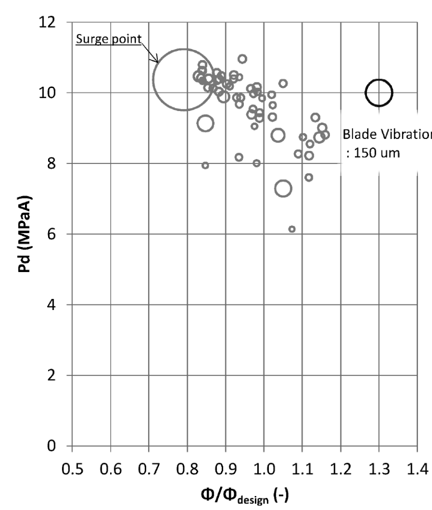


Fig. 9 Blade vibration of 6 th stage open impeller at each operation point

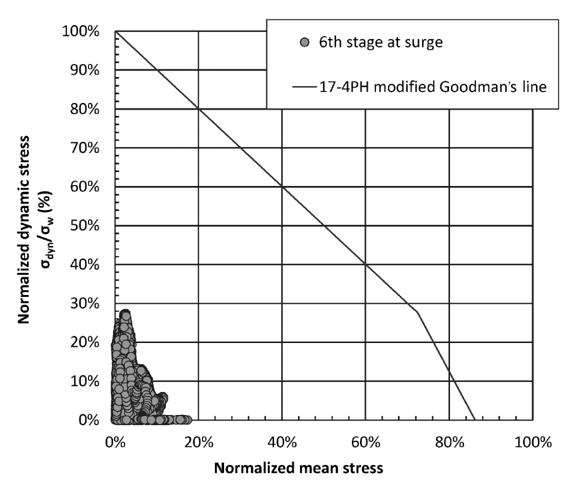
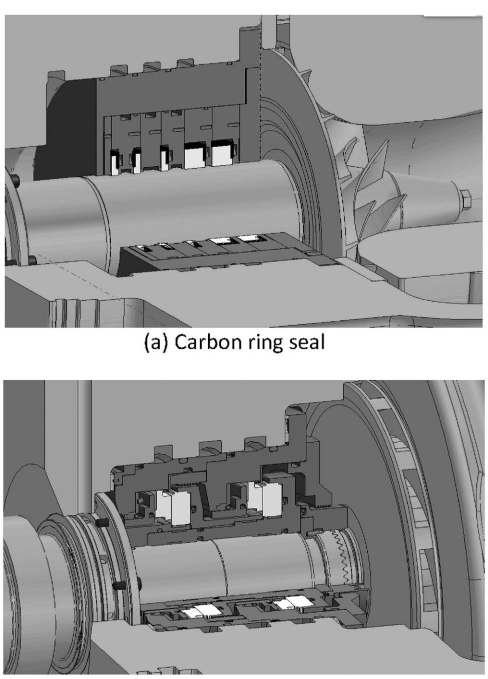


Fig.10 Modified Goodman diagram of open impeller

that by expanding the application of open impellers, which can operate at higher rotational speeds than closed impellers, it is possible to reduce the number of stages and achieve a more compact compressor.

1.4 Selecting shaft seals and performing leak testing

Fig.11 shows a carbon ring seal and a dry gas seal, which are commonly used in CO_2 compressors. Low system pressures are near atmospheric pressure. Therefore, carbon ring seals are used because they are inexpensive and provide adequate protection against leakage. Carbon ring seals are a type of annular non-contacting seal. They reduce leakage to atmosphere because the difference between the coefficient of liner thermal expansion between the shaft material and the carbon causes the gap in the sealing area to close during operation. When sealing pressure is high, the number or thickness of the carbon rings must be increased. For integrally geared compressors with impellers at both ends, this leads to longer overhang, generally reducing rotor rigidity and stability. Therefore, dry gas seals are used for high-pressure conditions, as their sealing width is constant regardless of pressure and they inhibit leakage better than carbon ring seals. Dry gas seals are non-contacting radial flow seals that exploit the dynamic pressure generated by gas trapped between the sealing surfaces during equipment rotation. This dynamic pressure causes the sealing surfaces to lift a distance on the order of microns.


(a) Carbon ring seal

(b) Dry gas seal

Fig.11 Gas seal

We used a dry gas seal as the shaft seal for the prototype unit. We measured leakage at a discharge pressure of 20 MPaG and verified the high reliability and low leakage of the dry gas seal. **Fig.12** shows the discharge pressure (horizontal axis) and the leakage of the dry gas seal (vertical axis) of the last stage (eighth stage).

The temperature of the barrier gas must be controlled because dry gas seals generally require a dry environment. Measures are taken to prevent the liquefaction of gas due to the Joule-Thomson effect when diverting discharge gas from the high-pressure stage to purge the low-pressure stage with barrier gas.

Compressor shaft seals used in CCS are specified to optimize leakage characteristics, cost, and mechanical reliability at the respective pressure levels for separation, capture, transport, and storage.

1.5 Skid-mounted compressor packages

One means of reducing compressor footprint and installation time is to use a package, which contains the system in a two-level steel frame structure (hereafter, skid) to achieve a compact, integrated compressor unit. The first level usually

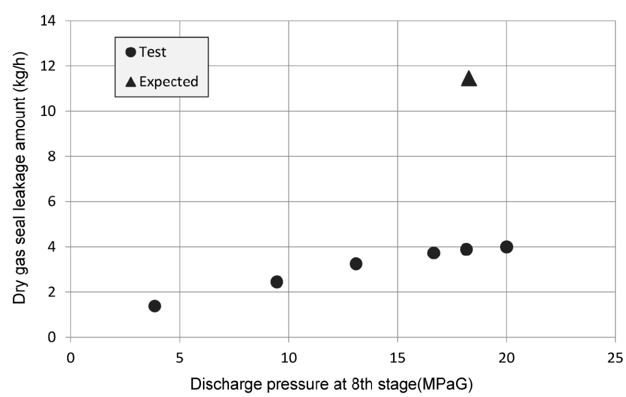


Fig.12 Dry gas seal leakage amount of 8th stage

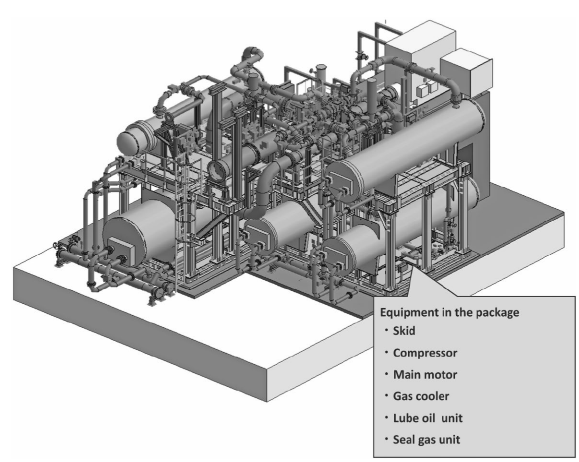


Fig.13 Overview of compressor package

contains the lubricating oil unit, barrier gas unit, and interstage cooler for the low-pressure stage; the second level usually contains the compressor, motor, and interstage cooler for the high-pressure stage. **Fig.13** shows the structure of the skid-mounted compressor package.

The two-level structure of this compressor package reduces the floor space required for the compressor and auxiliary equipment. Furthermore, since the compressor package is fully assembled including final piping connections before leaving the factory, there is no need for on-site pipe welding or pressure and leak testing. On-site assembly is relatively easy because components are installed in the same sequence as at the factory. This reduction in on-site working hours helps lessen the burden of labor costs, which have recently been on the rise. However, special considerations are required for maintenance: The compressor and auxiliary equipment must be able to be rigged and lowered easily, workers must be able to access equipment easily, and special safety considerations such as handrails must be in place. Furthermore, methods such as eigenvalue analysis must be used during design to ensure resonance will not occur between the natural frequency of the skid and the operating frequency since the compressor package is a two-level structure. Additionally, the foundation material and the size of the bolts used in the structure must be validated as capable of withstanding the dynamic loads generated by rotating machinery. Kobe Steel provides highly reliable compressor units through these technical approaches.

2. High-temperature compressor for LDES

Recent years have seen an increase in studies and demonstration projects for systems for long-term storage of renewable energy.[4] Mechanical charging systems combine a heat exchanger, a pressure and heat storage system, and a power generation turbine to perform charging and power generation. Specifically, these systems separate and store the pressure energy and thermal energy imparted to high-temperature gas by a centrifugal compressor. Advantages of mechanical energy storage systems over batteries (e.g., lithium-ion batteries) include the elimination of procurement constraints for materials, no degradation in charge/discharge efficiency over time, and lower operating costs as charge/discharge capacity increases.

Fig.14 shows the schematic of one type of mechanical energy storage system, a CO_2 battery.[5] During the charging phase, CO_2 gas is drawn from a large-capacity CO_2 gas holder, called a dome, and sent to a compressor that increases its pressure and temperature. The thermal energy is stored in a heat exchanger called a TES (thermal energy storage unit). CO_2 is then liquefied and stored under high pressure. During discharge, the TES vaporizes the stored CO_2. The heated gas expands to drive a turbine and generate electricity. **Fig.15** shows the compressor of a CO_2 battery for this application.

Fig.16 shows the schematic of an LAES (liquid air energy storage) system, which uses air to charge a cryogenic liquefaction system.[6] Two major technical challenges for compressors used in these systems are high operating temperatures and frequent start-stop cycles. The following subsections explain each challenge and Kobe Steel's solutions.

2.1 High temperature and thermal insulation design

Increased temperatures cause a decrease in the strength of the materials of the impeller in a centrifugal compressor. Therefore, the material and rotational speed are specified such that the stress of the impeller remains below the permissible value at operating temperature. The rear of the impeller in particular is exposed to high temperatures due to the impeller outlet temperature. Furthermore, the interior and rear of the impeller experience constant, high stress during operation due to the centrifugal loads from the rotation of the impeller itself. Strength analysis using finite element analysis is necessary to account for creep rupture

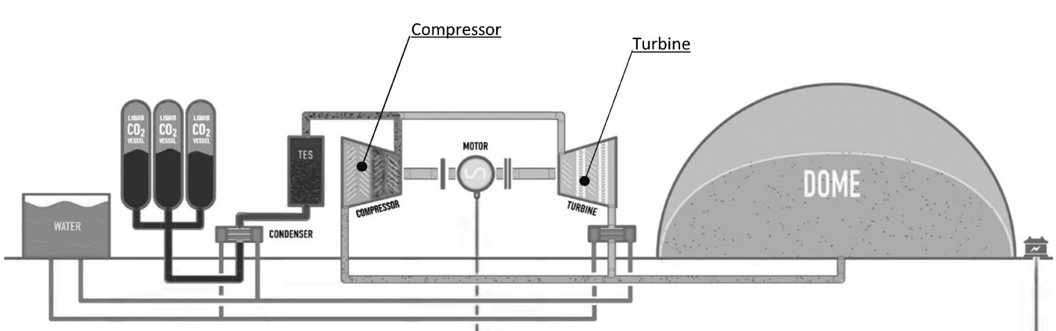


Fig.14 System schematic of CO_2 battery

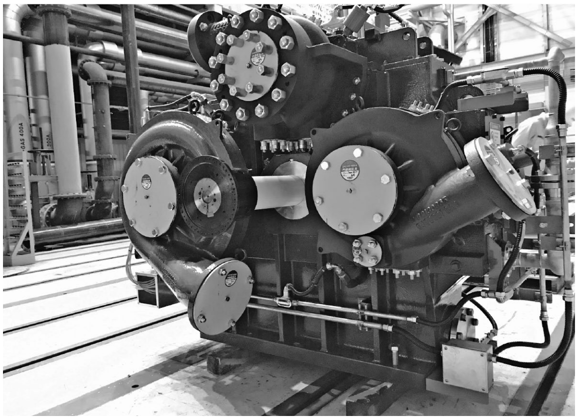


Fig.15 Centrifugal compressor for CO_2 battery

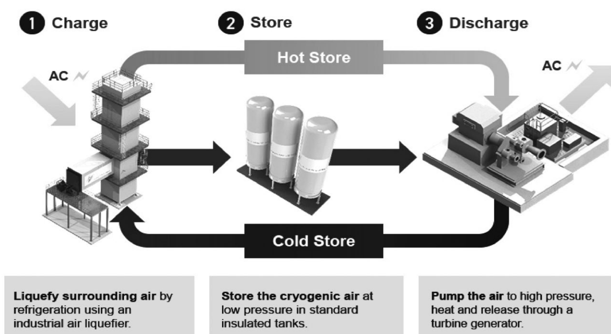


Fig.16 System schematic of LAES

life and ensure that permissible stresses are not exceeded. Kobe Steel addresses high temperatures by specifying heat-resistant steel for the impeller and surrounding components based on design temperature.

The pinion rotor also becomes hot due to thermal conduction from the impeller and heat transfer with the gas. This component is heat treated to increase its gear strength, so the operating temperature must stay below the tempering temperature of the material. It is also critical to evaluate the strength of the shaft end, which is the area of torque transmission, and to prevent heat transfer to the shaft area. To address this challenge, Kobe Steel incorporated a sleeve made of heat-resistant steel axially between the impeller and pinion rotor to increase the distance between these components, thereby suppressing heat input to the rotor.[7)]

Additionally, excessive heat transfer from the compressor casing to the gear case causes thermal deformation in the gear case. This in turn causes misalignment of the bearing section, potentially leading to contact between rotating and stationary parts. For this reason, the gear case must be thermally insulated. Thermal insulation between the compressor casing and gear case suppresses heat input to the gear case and thus minimizes the effects of thermal deformation.

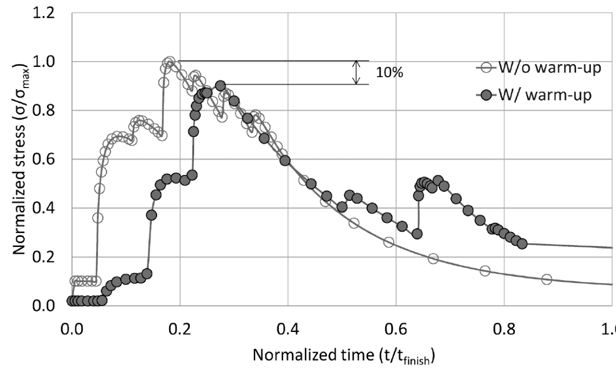


Fig.17 Stress at root of impeller blade during start-up

2.2 Thermal stress and frequent start-stop cycles

A large temperature difference between the stopped and operating states can lead to very large loads and moments on the pipe nozzles due to thermal expansion. In this case, the nozzles must be supported, or the loads must be reduced. We investigated support plans to relieve external forces on piping and support piping loads using expansion joints and bent pipes.

Furthermore, for this application, curtailing start-up time is important to minimize energy loss. However, rapid temperature shifts during start-up and shutdown cause thermal stresses, necessitating an understanding of when and where high-stress conditions occur. We calculated thermal stresses by combining transient thermal analysis, fluid analysis to determine the temperature distribution of parts in contact with gas, and structural analysis. **Fig.17** shows a comparison of the change in stress during start-up with and without a warm-up operation. On the horizontal axis is the time, normalized by the time it takes for the compressor to reach rated speed, and on the vertical axis is the stress generated in the impeller, normalized by its maximum value. We optimized the temperature and duration of the warm-up to reduce the stress level ratio at the root of the impeller blade by 10%, which should extend impeller service life.

LDES systems are charged using surplus electricity and serve to balancing power supply sources. Because they undergo frequent start-stop cycles, the number of daily start and stop cycles (DSS) must be considered during the design phase. The aforementioned thermal stresses are iteratively applied with each start-stop cycle, meaning that the fatigue design must account for the number of cycles. Furthermore, the starting torque of the shafts and couplings that transmit torque is several times greater than the rated torque. Therefore, design strength must account for both the starting torque and the number of starts and stops.

When the time between shutdown and the next startup is relatively short, the compressor's internal components do not cool uniformly. This uneven temperature distribution can cause the rotor to warp. Thermal distortion causes imbalances, leading to increased vibration during startup and potentially preventing the equipment from reaching rated speed. Applicable countermeasures are necessary, particularly with large units where the effects of thermal distortion are significant. Instead of completely stopping the rotor after operation, a turning device turns the rotor at low speed to equalize its temperature, enabling a smooth subsequent start-up.

Conclusions

This paper describes the technical challenges and their countermeasures for high-pressure CO_2 compressors used in CCS and high-temperature compressors for LDES. These technologies are critical for a carbon-neutral, sustainable society. Kobe Steel will continue tackling new demands and advancing high-efficiency, high-end, reliable compressor technologies and products to support customers' manufacturing needs.

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