LARGE SCALE REFRIGERATION PLANT FOR GROUND TESTING THE JAMES WEBB TELESCOPE AT NASA JOHNSON SPACE CENTER

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ABSTRACT

The James Webb Telescope is the successor to the Hubble Telescope and will be placed in an orbit of 1.5 million km from earth. Before launch in 2014, the telescope will be tested in NASA Johnson Space Center's (JSC) space simulation chamber, Chamber A. The tests will be conducted at the deep space conditions of low pressure and temperature. Chamber A's helium cryo-panels are currently cooled down to 20 K by two Linde 3.5 kW refrigerators. The new 12.5 kW, 20 K Coldbox described in this paper is part of the upgrade to the chamber systems for this large test program. The Linde Coldbox will provide refrigeration power in several operating modes where temperatures and refrigeration power are controlled with high accuracy due to the demanding NASA test requirements. The implementation of two parallel expansion turbine strings and the Ganni floating pressure cycle results in a highly efficient and flexible process that reduces the electrical power input. This paper will describe the collaboration and execution of the Coldbox project.

KEYWORDS: Large Scale Refrigerator, Floating Pressure, NASA

INTRODUCTION

James Webb Telescope

The James Webb Space Telescope (JWST) will be a large infrared telescope with a 6.5-meter primary mirror. Launch is planned for 2014.

JWST will be the premier observatory of the next decade, serving thousands of astronomers worldwide. It will study every phase in the history of our Universe, ranging from the first luminous glows after the Big Bang, to the formation of solar systems capable of supporting life on planets like Earth, to the evolution of our own Solar System.

JWST is an international collaboration between NASA, the European Space Agency (ESA), and the Canadian Space Agency (CSA). The NASA Goddard Space Flight Center is managing the development effort. The prime contractor is Northrop Grumman; the Space Telescope Science Institute will operate JWST after launch.

Several innovative technologies have been developed for JWST. These include a folding, segmented primary mirror, adjusted to shape after launch; ultra-lightweight beryllium optics; detectors able to record extremely weak signals, microshutters that enable programmable object selection for the spectrograph; and a cryocooler for cooling the mid-IR detectors to 7 K. The long-lead items, such as the beryllium mirror segments and science instruments, are under construction. All mission enabling technologies were demonstrated by January 2007, and the Project was confirmed to enter its implementation phase in July 2008.

Refrigerator and Space Simulation Test Chamber

JSC space simulation chamber, Chamber A, is being prepared to test the JWST Observatory. The Observatory consists of the primary optics called the OTE (Optical Telescope Element) and the science instrument package called ISIM (Integrated Science and Instrumentation Module). The primary objective is to test the OTE in its flight like environment. The OTE test will align and focus the 18 mirror segments at cryogenic conditions, verify and measure the cryo-positioning of the mirror structure to see the integrated full optics act as a monolith at cryogenic temperatures. The OTE and ISIM testing will verify the integrated path of the light through the primary mirrors to the science instruments. Another objective of the test will be thermal balancing to simulate the thermal performance on flight and to correlate the observatory thermal models. To create the required deep space environment, Ch-A at the Johnson Space Center is undergoing large scale modifications. A new gaseous helium cooled shroud will be installed in the chamber. The new refrigerator will provide the active cooling (<20 K) for the shroud and also several pieces of test specific ground support equipment.

PROJECT EXECUTION

NASA and the US Department of Energy agreed to have Jefferson National Lab's (JLab) cryogenics experts consult on the helium refrigerator Chamber A upgrade for the JWST test program. JLab is consulting to NASA and Jacobs Engineering for the system design, system integration, and controls. Jacobs Engineering awarded the Coldbox contract Linde Cryogenics. Coldbox engineering and startup support will be performed by Linde Kryotechnik. Coldbox fabrication, testing, and installation support will be performed by Linde Cryogenics. Process engineers from JLab and Linde are cooperating under the Linde license of the "Ganni Floating Pressure Cycle" to optimize the process and controls for the new plant. The Coldbox is scheduled to be delivered to NASA JSC early in 2010.

PROCESS DESIGN

Refrigerator Design Criteria

The refrigeration plant was designed to provide the required refrigeration power with good efficiency over a wide temperature range. The JWST test requirements are under development and the anticipated load at 20 K is between 8 kW and 12 kW. The main features of the refrigerator are:

- Two parallel TED45 turbines provide optimal refrigeration performance in the 100 K range as well as in the 20 K range.
- A large LN2 vessel, two parallel warm end heat exchangers and three load return valves at different temperature stages ensure an effective large cool down capacity.
- Two parallel 80 K charcoal adsorbers and a subsequent bypass line are used to purify the circulating helium before the actual cool-down of the chamber when the turbines are started.
- Additional cold gaseous nitrogen supports the first helium / nitrogen heat exchanger which results in a lower LN2 consumption.

Process Arrangement

The Process Flow Diagram in **FIGURE 1** shows a simplified concept of the process arrangement. Warm high pressure helium enters the Coldbox and is divided into two streams that are being cooled down by the low pressure return flow and the nitrogen vapor respectively. Connected to one stream again the high pressure helium is cooled down to 80 K in the nitrogen evaporator before it passes one of the two parallel adsorbers. Being further cooled down in the subsequent heat exchangers the helium is expanded in one or two turbines, depending on the temperature range and fed to the Space Chamber. The low pressure helium return joins the process at 20 K, 80 K or 300 K level depending on its temperature.

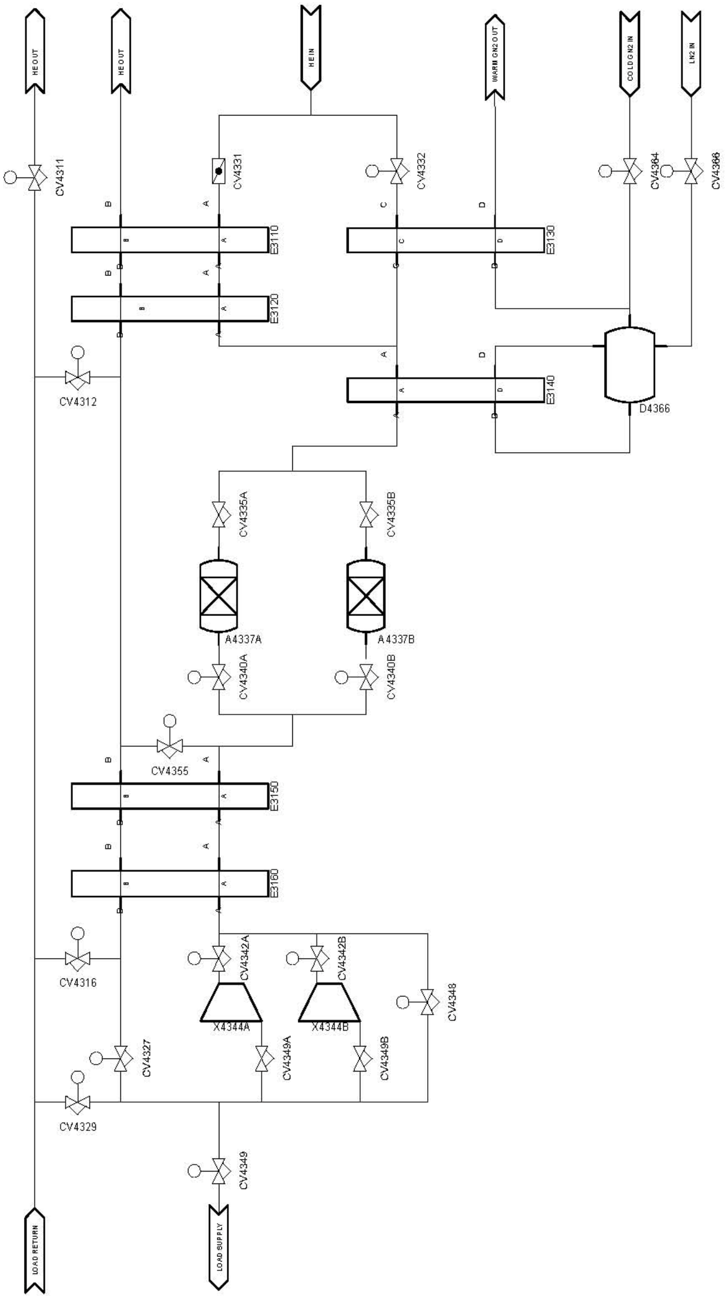

FIGURE 1: Process Flow Diagram

Floating Pressure Concept

It is not unusual that the primary design conditions "Refrigeration Load" and "Load Return Temperature", given by the customer, vary in a considerable range not only in the sales phase but also during project execution as not all impact variables are known in the beginning. To be on the safe side a "worst case scenario", via highest load and coldest temperature is defined to specify the demand for the refrigerator. Hence the system consisting of refrigerator, compressor(s) and load must possess sufficient flexibility to cope with all scenarios in an efficient way. The Ganni Floating Pressure Cycle, well described in [1 & 2] is a useful control strategy to provide this flexibility and energy saving potential. The implementation of the floating pressure theory to the existing 3.5kW system at NASA, JSC and the resulting gains are presented in [3]. Its limitations in actual built plants are pointed out below.

Basis for the System Efficiency and Exergy Analysis

The intrinsic fluid property "(mass)specific exergy" is defined as

$$e = h - T_0 \cdot s\,,$$

where h is the specific enthalpy, s is the specific entropy and T_0 is the ambient temperature.

The exergy flow between two states 1 and 2, used to quantify the process efficiency, is defined as

$$\Delta E = \dot{m}\left(h_1 - h_2 - T_0\left(s_1 - s_2\right)\right)$$

The exergetic efficiency of the Coldbox $\eta_{ex,CBX}$ can then be defined as the ratio of the exergy flow leaving the Coldbox at the cold end and the exergy flow into the Coldbox at the warm end.

$$\eta_{ex,CBX} = \frac{\Delta E_{HE,COLD}}{\Delta E_{HE,WARM} + \Delta E_{N2}}$$

Most interesting for the operator of the refrigeration system however is the exergetic efficiency of the entire system $\eta_{ex,SYS}$, including the energy power input for the recycle compressor and for the production of liquid nitrogen. These input powers are characterized by efficiencies of compressor and LN2 suppliers gained on numerous executed projects and can be used to define the system efficiency as

$$\eta_{ex,SYS} = \frac{\Delta E_{HE,COLD}}{\dfrac{\Delta E_{HE,WARM}}{\eta_{is,COMP}} + \dfrac{\Delta E_{N2}}{\eta_{ex,LN2}}}$$

For the NASA project the isothermal efficiency of the compressor, between 0.4 and 0.55 depending on the pressure ratio, is defined in the customer's specification [4] as well as the exergetic efficiency for the LN2 production being 0.35.

System Efficiency and Boundaries of the Floating Pressure Concept

Apart from the design conditions T-s-diagrams for various different load cases were calculated in order to obtain information about the efficiency behavior depending on the load cases. Exemplarily various constraints of the floating pressure concept are shown.

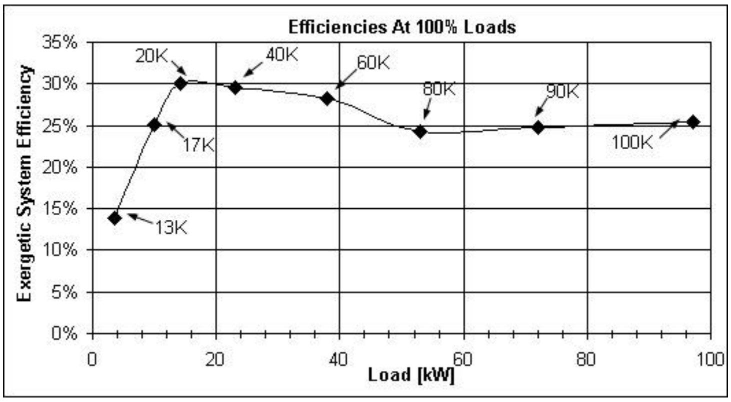

FIGURE 2: 100% Loads at Various Load Return Temperatures

In **FIGURE 2** the refrigeration loads and efficiencies at different load return temperatures are shown. The exergetic system efficiency is expected to be between 24% and 30% in the large temperature range between 17 K and 100 K. For lower temperatures the efficiencies and possible refrigeration loads reduce drastically as the turbine reaches its axial thrust restriction. An interesting observation is that there is a local efficiency minimum in the 80 K range. That is when the LN2 pre-cooling stage is least effective. For load return temperatures higher than 80 K a significantly higher LN2 consumption is required in order to maintain the turbines inlet temperature at 80 K, which is boosting the system performance in turn.

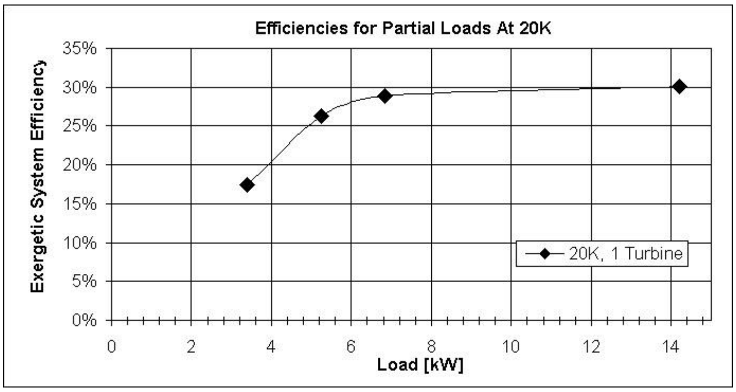

FIGURE 3: Turndown Modes at 20K Load Return Temperature

In **FIGURE 3** the turndown modes or partial loads at the design load return temperature of 20 K are shown. The exergetic efficiency is expected to be high and stable for as low as 50% of the maximal load at this temperature. For lower refrigeration loads the efficiency decreases because the suction pressure of the compressor shall not be sub atmospheric. The residual mass flow that is not necessary for the refrigeration load has to circulate in the compressor bypass. Furthermore the high pressure of the compressor outlet shall not fall below 8 bar to ensure proper functioning of the oil removal system.

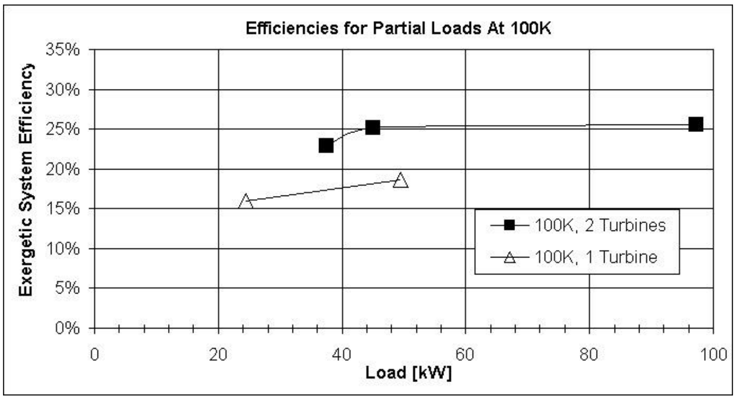

FIGURE 4: Turndown Modes at 100K Load Return Temperature

In **FIGURE 4** the turndown modes or partial loads at the off-design load return temperature of 100 K are shown. As an option also the use of only one expansion turbine instead of two as recommended is shown. The exergetic efficiency is expected to be high and stable for as low as 50% of the maximal load at this temperature when two turbines are in use. For lower refrigeration loads and particularly for operation of only one expansion turbine the efficiency decreases because of a high pressure ratio restriction of the turbine(s). The pressure ratio also contributes to the axial thrust which limits the performance of the large turbines. The use of only one expansion turbine does not automatically lead to 50% of the refrigeration load. Because of the mentioned pressure ratio constraint the low pressure (respectively turbine outlet pressure) has to be increased.

PROCESS CONTROL

The process control of the cold box is based on the Ganni floating pressure cycle philosophy (simplified schematic: **FIGURE 5**). As the floating pressure process is a constant pressure ratio and variable gas charge process, the compressor discharge pressure is adjusted to match the required load as indicated by the shield return temperature. Of course, the minimum compressor discharge pressure is constrained by the design of the oil removal system and the maximum by the system design pressure (i.e., the pressure rating of the compressor and components downstream of the compressor discharge). Additionally, there may be other compressor constraints, such as preventing the suction from becoming sub-atmospheric (e.g., for rotary screw compressor designs using a suction shaft seal). Turbine constraints such as maximum tip speed and bearing capacity are normally handled by the local turbine control elements (i.e., inlet and brake valves). These are set so as to allow the widest possible operational envelope for the turbine. The Floating Pressure process typically utilizes two control elements to add or remove gas charge from the system; namely the 'mass-in' (MI) and 'mass-out' (MO) control valves, respectively. The compressor bypass valve operates only to prevent a sub-atmospheric suction condition. As such it is normally closed, except under greatly reduced loads (i.e., less than approximately 50% of the maximum load) or in cases of manual intervention (e.g., such as required in a single turbine operation at 100 K for maximum capacity due to a turbine pressure ratio limitation). During steady (fixed temperature) operation, additional measures of capacity control, such as turbine bypass, are not implemented until the compressor suction pressure reaches its minimum allowed setting (say, 1.05 bar). During a cool down to a desired shield return temperature set-point, the discharge pressure remains at its maximum limit, until the load return temperature reaches the given set-point (say, 20

K). Upon reaching the set-point, the MI and MO adjust the discharge pressure to maintain the desired shield return temperature so that the refrigeration capacity matches the actual heat load.

Liquid nitrogen (LN2) is used for pre-cooling the high pressure helium stream down to 80 K. The LN2 pre-cooler implements nitrogen phase-separation using a thermo-siphon and directs the total high pressure helium flow through the latent nitrogen heat exchanger section. The LN2 supply maintains a pre-determined liquid level set-point in the phase-separator vessel (i.e., the 'LN2 pot'). The usage of the LN2 is regulated by the high-pressure helium bypass between the large 300-80 K helium-helium heat exchanger and the smaller 300-80 K helium-nitrogen vapor heat exchanger. This valve uses the warm-end helium-helium stream temperature difference as the primary process variable to affect minimum LN2 usage. Cascade control is implemented on this valve to prevent the warm-end helium-nitrogen vapor stream temperature difference from becoming too extreme (say, a 240 K nitrogen vent temperature).

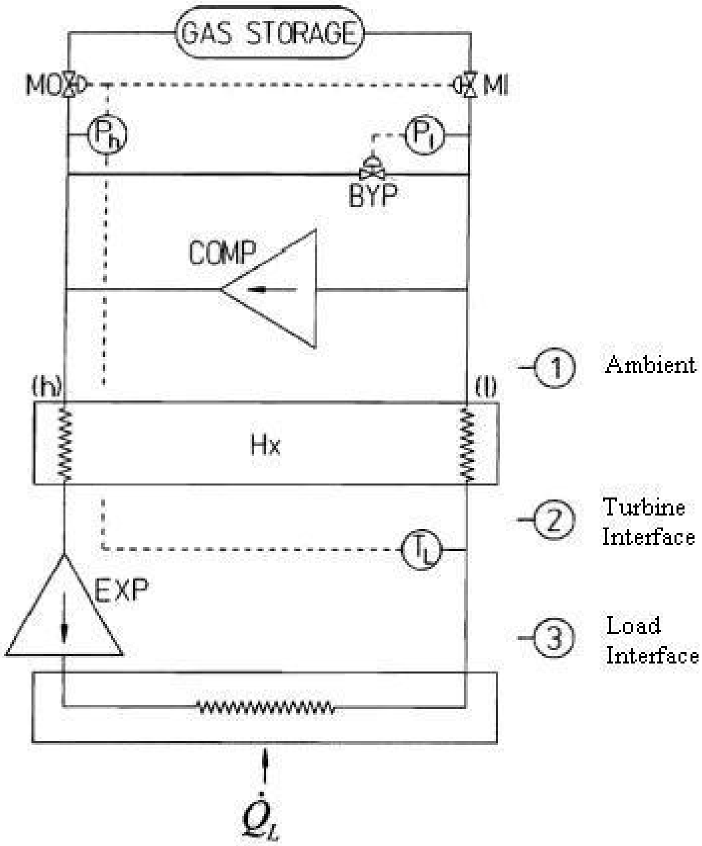

FIGURE 5: Floating pressure control schematic

SUMMARY

The system is designed to meet the needs of flexible load (both in meeting the actual load and the required operational temperature for the experiment) with good efficiencies. However, it is important for the supplier of cryogenic plants to receive as much information as possible in the early phases of a project in order to design the process and particularly the expansion turbines to achieve optimal results in the most relevant load cases. In addition the floating pressure philosophy is expected to provide improved temperature stability and the reliability as proven in other systems operating under this principle, and these are of critical importance to this system.

ACKNOWLEDGEMENTS

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