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# Operational Characteristics of the MTF Liquid Helium Plant

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## Abstract:

Operational characteristics of the MTF liquid helium plant have been studied and documented: helium flow rate through the $LN_2$ precooler vs. measured differential pressure, cold box turbine flow rates, and cold box turbine performance.

## 1. Introduction

As a prelude to modeling the MTF cryogenics system, some important operational characteristics of the helium plant have been studied using archived MTF operating data: helium flow rate through the $LN_2$ precooler vs. measured differential pressure, cold box turbine flow rates, and cold box turbine performance. These operational characteristics are presented here both as a matter of documentation as well as to form the basis of a model developed in TD-05-008, "Modeling and Parametric Studies of the MTF Cold Box."

## 2. MTF $LN_2$ Precooler Helium Flow Rate

The MTF CTi 1500 cold box originally used $LN_2$ in the first two heat exchangers, HX-1/1A and HX-RC. At some point the heat exchanger HX-1/1A failed, allowing high-pressure helium to enter the liquid nitrogen stream and vent outside. An external $LN_2$ precooler was integrated into the system in the early 1990's, and the cold box heat exchanger $LN_2$ passes are now backfilled with high-pressure helium during normal operations.

Additional instrumentation was added to the $LN_2$ precooler during the 1999 cold box controls upgrade. One of these instruments was a differential pressure transducer to record the pressure drop of the helium as it passes through the $LN_2$ precooler. There was no correlation between helium flow rate and pressure drop, however.

Archived operations data allowed this relationship to be determined. Liquid nitrogen usage rates were determined using the rate of change in the measured liquid level, and an energy balance allowed the helium flow rate to be calculated. It was assumed the helium enters the precooler at a temperature of 285 K and the nitrogen gas leaves the precooler at 280 K. The calculated helium flow rate vs. measured helium pressure differential is shown in Figure 1.

It is also important to note that the precooler $LN_2$ supply valve leaks, so the change in $LN_2$ liquid level during normal operations is misleading. The data used for these calculations were taken from periods when the $LN_2$ supply to the precooler was known to be off: beginning a scheduled warmup or closure of the solenoid supplying $LN_2$ to IB1.

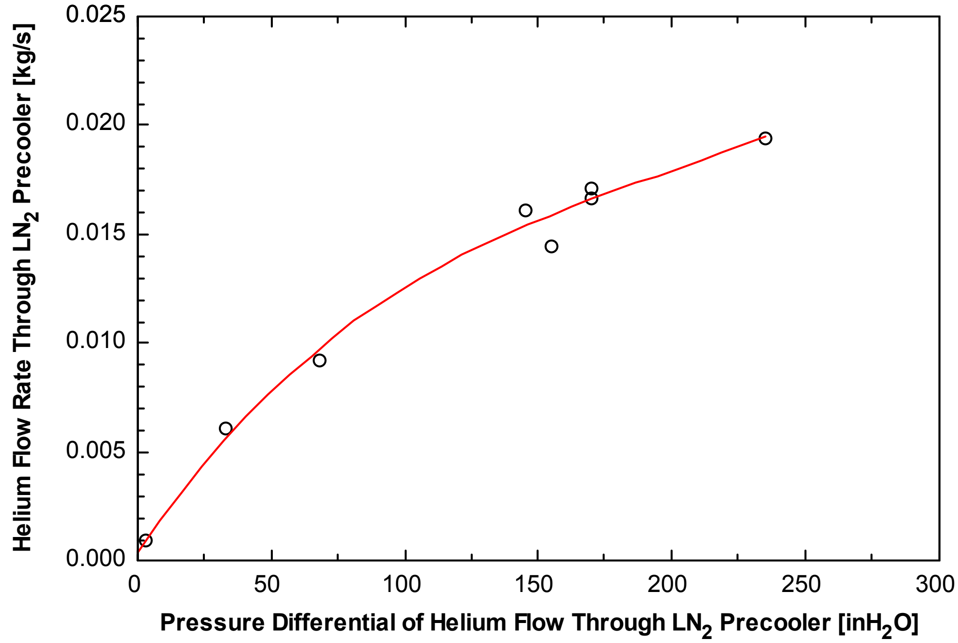


**Figure 1** Helium flow rate vs. measured helium pressure differential through the $LN_2$ precooler.

## 3. MTF Cold Box Turbine Flow Rates

The high-pressure flow rate through the cold box is dictated by the turbine inlet valves, the turbines themselves, and T2 bypass valve PCV56. Flow through the turbines has been found to follow the relationship of Equation 1 for an orifice with sonic flow:

$$\dot{m} = K_n \sqrt{\rho_{in} P_{in}} \tag{1}$$

where $\dot{m}$ is the mass flow rate through the turbine, $\rho_{in}$ is the helium density at the turbine inlet, $P_{in}$ is the turbine inlet pressure, and $K_n$ is an empirical constant for each turbine.

As shown in Figure 2, there is a venturi downstream of turbine T1. The known geometry of this venturi allows the T1 mass flow rate to be calculated according to Equation 2:

$$\dot{m} = Y F C_D A_t \sqrt{2 \rho \Delta P} \tag{2}$$

where $\dot{m}$ is the mass flow rate through the venturi, Y is the flow compressibility factor (Y = 1 for the flow Mach numbers encountered in the cold box), F is the velocity of approach factor (F = 1.028) which is a function of the ratio of the venturi throat diameter (1.085 in) to the pipe inner diameter (2.245 in), $C_D$ is the discharge coefficient ($C_D$ = 0.98), $A_t$ is the throat area ($A_t$ = 0.925 in$^2$), $\rho$ is the helium density at the venturi, and $\Delta P$

is the pressure differential between the inlet pipe and the venturi throat. The helium density at the venturi is calculated using the T1 outlet pressure and temperature.

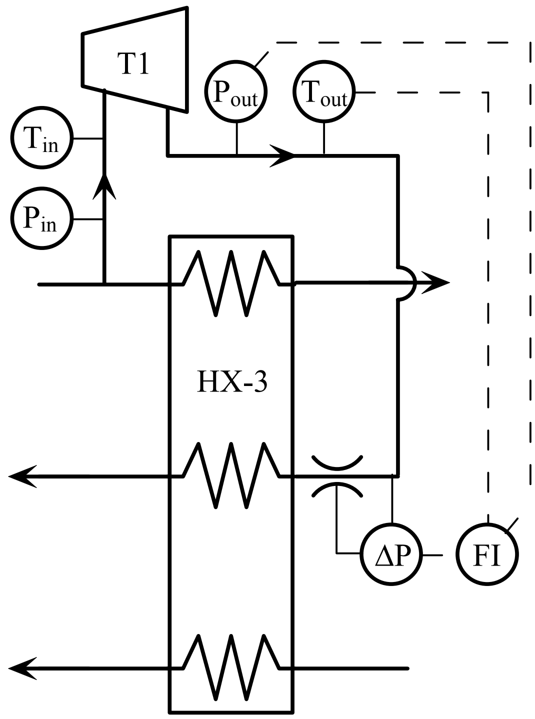


**Figure 2** Medium-pressure flow path and instrumentation at the T1 turbine and the downstream venturi.

The empirical constant $K_1$ for the T1 turbine can then be calculated, yielding the final relationship of Equation 3:

$$\dot{m}\,[\text{g/s}] = 1.32\sqrt{\rho_{in}\,[\text{kg/m}^3]\,P_{in}\,[\text{psia}]} \tag{3}$$

Figure 3 compares the T1 flow rate calculated using Equation 3 with T1 inlet conditions against the venturi flow rate calculated using Equation 2 with T1 outlet conditions. Each data point represents a steady operating condition between March 2001 and March 2004, usually during the overnight. There is very good agreement between the two calculations. The empirical relationship of Equation 3 appears to be valid.

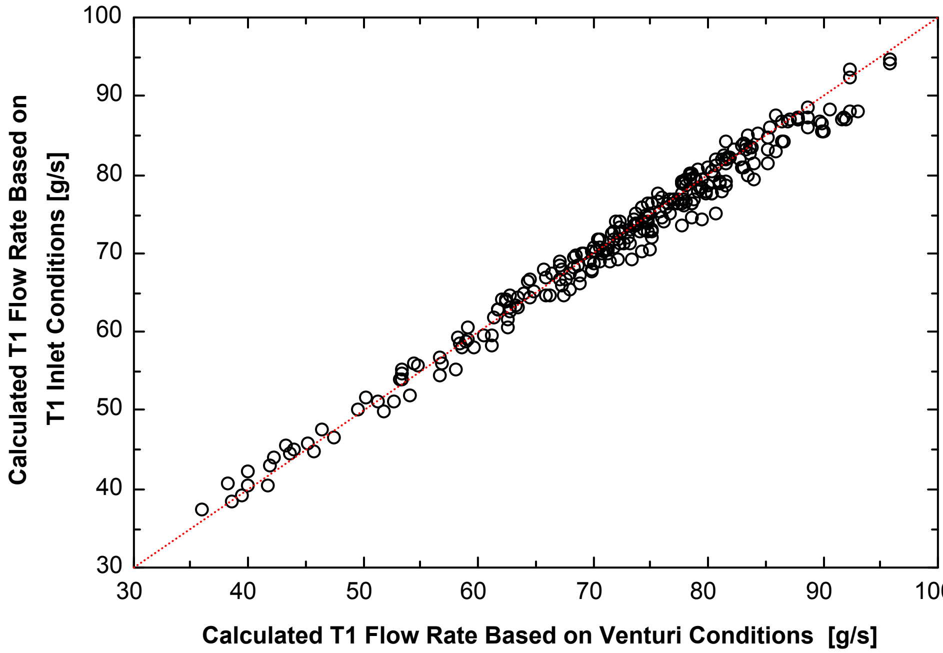


**Figure 3** Calculated T1 flow rate based on T1 inlet conditions vs. calculated flow rate based on venturi conditions.

The empirical constant $K_2$ to describe the flow through the T2 turbine cannot be determined in such a straightforward manner because there are no local flow elements. The combined mass flow rate through the T2 turbine and the bypass valve PCV56 must be first calculated with a mass balance. An orifice plate in the high-pressure piping of the compressor system discharge is used to calculate the mass flow rate entering the cold box. Subtracting the mass flow through the T1 turbine leaves the mass flow through the T2 turbine and PCV56. Logged historical data can then be sorted to isolate data points when PCV56 was closed, leaving only the flow through the T2 turbine. This mass flow rate and the measured T2 turbine inlet conditions are used to calculate a value of empirical constant $K_2$, yielding the final relationship of Equation 4.

$$\dot{m}\left[\mathrm{g/s}\right] = 0.753\sqrt{\rho_{in}\,[\mathrm{kg/m^3}]\,P_{in}\,[\mathrm{psia}]} \tag{4}$$

It is important to note that the value of $K_2$ = 0.753 was calculated with the inlet density $\rho_{in}$ determined using real fluid properties, not the ideal gas equation. Figure 4 shows the fit of $K_2$ = 0.753 for operating data collecting during the periods of January-February 2004 and August-September 2004.

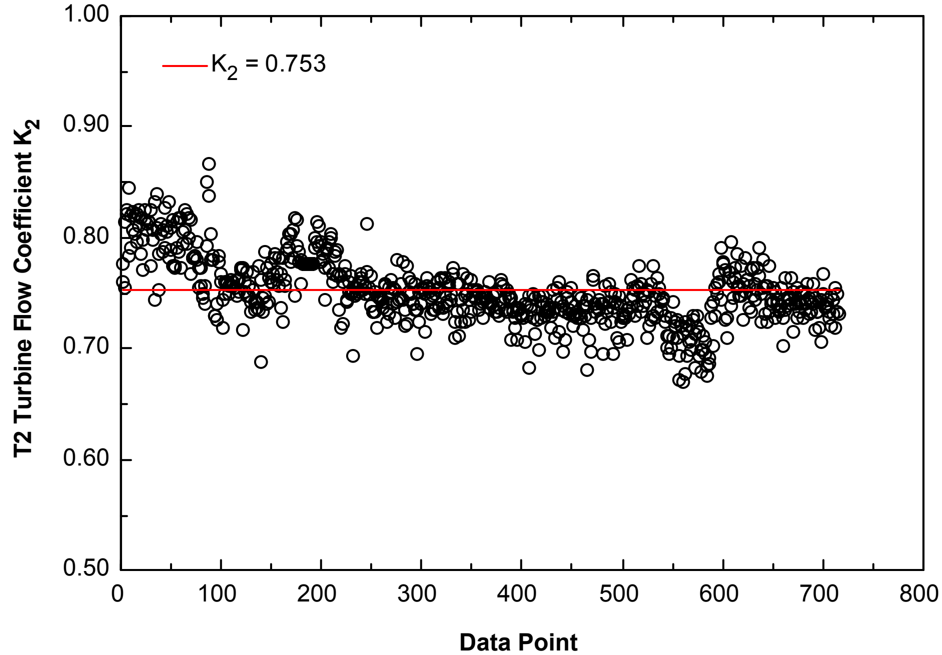


**Figure 4** Fit of T2 turbine flow coefficient $K_2$.

## 4. MTF Cold Box Turbine Performance

Figure 5 plots the T1 isentropic efficiency vs. speed. Like the data used to study the T1 turbine flow rate, each data point here represents a steady operating condition between March 2001 and March 2004, again usually during the overnight. This plot indicates that under normal operating conditions, running the T1 turbine at higher speeds will lead to higher efficiencies and therefore better cryogenic system performance.

This plot is also useful in plant troubleshooting by helping determine whether poor system performance (e.g., a low liquefaction rate) is due to a T1 turbine performance problem.

Similar data could not be collected for the T2 turbine. The measured inlet and outlet temperatures are not accurate enough to allow meaningful efficiency calculations. Calculated isentropic efficiencies for the T2 turbine tend to be greater than one or less than zero.

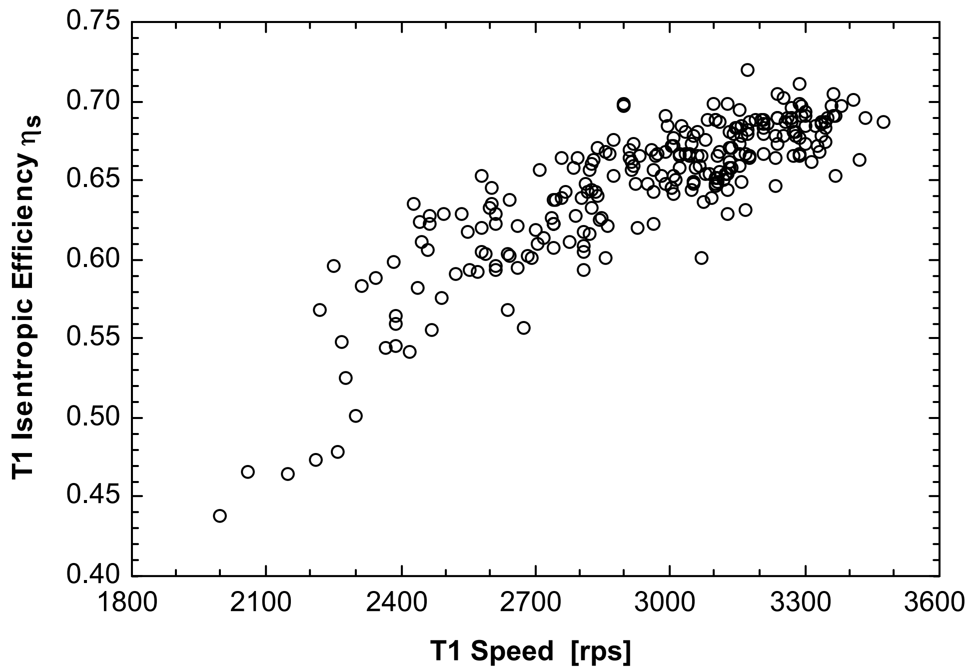


**Figure 5** T1 isentropic efficiency vs. speed.

## 5. Conclusion

A number of operational characteristics of the MTF liquid helium plant have been studied and documented. It is hoped that documentation of these characteristics will be useful in troubleshooting and further system study.