Control and Commissioning of a Hot Gas Bypass Compressor Load Stand for Testing Light-Commercial Compressors on Low-GWP Refrigerants

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Abstract. Because of changing efficiency standards for HVAC&R equipment, hydrofluorocarbon refrigerants such as R134a and R410A are in the process of being phased out because of their high Global Warming Potential (GWP). Many low-GWP refrigerants, such as R1234yf, R1234ze(E), R1234ze(D), R32, and several blends of these, are being considered as replacements. This creates a need for design changes to compressors. Recent work by Schmidt et al. (2018) presented a hot-gas bypass compressor load stand constructed at Oklahoma State University to facilitate testing of compressors with their new design changes.

The load stand has a design capacity range of 10-80 tons. This wide spectrum of testing conditions necessitates a comprehensive controls package to maintain stability and control over the simulated operating condition. This controls package was designed in LabVIEW and allows independent control over the compressor suction pressure and temperature and the discharge pressure using PI controllers. The controllers operate to both set the desired operating condition as well as minimize the variation of the parameters with time (i.e. random uncertainty).

A series of commissioning tests were performed to validate the stand operation and performance metric measurements using a 40 ton scroll compressor. The initial results suggested that appropriately balanced cooling water flow and appropriately tuned controllers are critical to minimize random uncertainty of the data. Without appropriate tuning or cooling water control the load stand was unstable. Adjusting the water flow provided stability but yielded random uncertainty of 0.014% compared with both water control and appropriately tuned controller which yielded random uncertainty of 0.0041%.

1. Introduction
With a global effort to reduce the environmental impact of HVAC&R equipment, lowering the global warning potential (GWP) of refrigerants was introduced in the Kigali Amendment of the Montreal Protocol, which requires countries to reduce the use of HFCs by 85 percent between 2019 and 2036. Because of this, Schmidt et al. [1] developed a hot-gas bypass compressor load stand for the purpose of testing the effect of low-GWP refrigerants on compressor performance, for which the controls and commissioning is being addressed in this project.

The hot-gas bypass load stand is a flowmeter type of load stand which is one of two main methods for testing compressors. The other is the calorimeter type. These two compressor
testing methods differ in the way that they determine mass flow rate. The flowmeter type uses direct measurements of refrigerant mass flow rate while the calorimeter type uses heat balances on one of the heat exchangers to determine the flow rate. The calorimeter load stand is typically cheaper to develop than the hot-gas bypass stand because it eliminates the need for expensive mass flow meters. However, as Sathe et. al.[2] describe, the hot-gas bypass stand has fewer components than a calorimeter, making it easier to build and operate. They also explain how the hot-gas bypass stand moves between test conditions faster than a calorimeter because it directly controls the three operating conditions (suction pressure, discharge pressure, and suction superheat). The last difference they observed is that hot-gas bypass stands require much less condenser cooling capacity than calorimeters because a majority of the discharge gas is bypassed, so the amount of refrigerant flowing through the condenser is lower than that of a calorimeter. Marriott[3] also explains that typical calorimeters are not effective at stabilizing quickly, making them less efficient at collecting numerous data points.

The desired operating ability of the load stand is to be able to move rapidly between operating conditions, so that a matrix of conditions can be tested quickly, with little down time between test points. However, in order to achieve that ability, a controls scheme had to be developed that would allow for automatically moving between points and stabilizing, to greatly reduce the amount of time required for each test. However, it is not sufficient to only move between points quickly. The load stand must also produce reliable and accurate results. Because of this, the controls scheme was also designed to reduce the amount of random uncertainty of the measurements enough, so that systematic uncertainty is the effectively the only source of uncertainty affecting the results.

After the load stand and the controls were completed, commissioning activities had to take place to verify that the controls program reaches operating conditions in accordance with ASHRAE Standard 23.1[4]. However, for this load stand it was desired that the operating conditions are met with much higher accuracy (within the limits of the sensor uncertainty), which is a defining feature of this project.

2. OSU Load Stand

The design of the compressor load stand was presented by Schmidt et al.[1] and operates on the principle of a hot-gas bypass cycle. The hot-gas bypass cycle operates similarly to a normal vapor compression cycle, except at the exit of the compressor, refrigerant vapor is separated into two flow paths. The majority of the refrigerant is part of the bypass line that will be used to heat the rest of the refrigerant to a specified compressor inlet temperature after the smaller portion passes through the condenser and is expanded to suction pressure in the liquid expansion valve.

There are a minimum of three parameters that are required to be manipulated during compressor testing to gather performance data over the full range of the compressors operating capabilities. These include, a simulated evaporating temperature, the condensing temperature, and suction temperature (i.e. the suction superheat of the compressor).

The simulated evaporating temperature is the saturation temperature at the suction pressure, which represents the pressure a compressor would be exposed to at a particular evaporating temperature. Control of the simulated evaporating temperature is achieved through manipulation of the suction pressure. This is accomplished by adjusting the expansion valves on the bypass line. Figure 1 shows a schematic of the load stand and all of its control mechanisms, the gas bypass valves operating between state 2 and 3. These valves directly control P01, which is the suction pressure in Figure 1.

The condensing temperature is the saturation temperature at the discharge pressure of the compressor, which is controlled to set the desired condensing temperature. The compressor discharge pressure is manipulated through the control of the cooling water provided to the condensers shown in Figure 1. The flowrate of water going through the condensers will cause a
change in the discharge pressure (P02 in Figure 1) of the refrigerant.

Finally, the amount of superheat desired will affect the suction temperature of the compressor. The suction temperature can be controlled by three liquid expansion valves, which operate between state 4 and 5 of Figure 1, where T01 is the suction temperature being measured.

3. Load Stand Control Scheme

The load stand is equipped with a comprehensive data collection and controls scheme, shown in Figure 2 that uses a variety of sensors which are used to measure desired variables. These sensors are recorded for the sake of characterizing compressor performance as well as to be process variables in the PI controller. Refrigerant enters the compressor at a desired suction temperature and pressure (T01 and P01) at state 1 and exits the compressor at the desired discharge condition (P02). From there the refrigerant flow splits into two paths, the bypass line to point 3 and the liquid line to point 5, where the suction pressure and temperature respectively are controlled. States 3 and 5 then mix together to create the suction condition.

The load stand uses gauge pressure transducers as well as T-type thermocouples and RTDs for measurements of pressures and temperatures. It also uses Coriolis mass flow meters for high accuracy mass flow measurements on the suction, discharge, economizer gas and liquid lines, and the oil line. An accelerometer measures the output frequency of the compressor. Finally, a watt transducer is used to measure the power used by the compressor during a test. The specific model of instruments and their systematic uncertainty has been previously reported by Schmidt et al.[1]. Schmidt et al.[1] also conducted an uncertainty analysis to quantify the systematic uncertainty propagated to the calculation of both the isentropic and volumetric efficiencies. This work concluded that the systematic uncertainty propagated to these metrics is acceptable for testing but assumes a negligible contribution of random uncertainty.

The three load stand control variables previously discussed are controlled using a Proportional-Integral (PI) controller. The control scheme is executed by code developed in
Figure 2. Flow chart of data acquisition and controls

LabVIEW[6], where measurements are taken from the load stand and the PI loops create changes in the testing conditions. In order to accurately determine performance metrics, the load stand must be operating at steady-state, meaning there is no significant change of any of the control variables over the length of the testing period. According to ASHRAE Standard 23.2[5], steady-state operation requires that the data points collected must not be successively increasing or decreasing and must be within the specified tolerances. While ASHRAE 23.1[4] specifies the required tolerances for testing compressors, the target tolerance for this load stand is within the range of uncertainty of the measurement devices (i.e. systematic uncertainty). To achieve this, the random uncertainty must be minimized for the testing period. Using PI controllers, the system is able to reach steady-state, with minimal amounts of random uncertainty for a wide range of operating conditions.

3.1. Suction Pressure
The suction pressure is controlled by three gas expansion valves. Because of the range of allowable capacities on the load stand, three expansion valves can be used to make large, medium, and small adjustments. During operation, only one valve is controlled at a time and the decision of what valve to use is decided by a threshold logic and that utilizes far away the measured suction pressure is from the setpoint. To determine which valve to use measured suction pressure plus a range dictated by a threshold value of 34.5 kPa (5 psia) is calculated as,

\[ P_{\text{suc,range}} = P_{\text{suc,setpoint}} \pm P_{\text{suc,threshold}}. \]

If the suction pressure is outside of the range given by \( P_{\text{suc,range}} \) then the larger valve is used, and the smaller valve if less than the range. Each of these valves has its own PI controller, with independent gains, which adjusts the suction pressure toward the set point.

3.2. Suction Temperature
Similar to the suction pressure, the suction temperature is controlled using three liquid expansion valves on the liquid line that exits the condenser. Adjusting the amount of liquid that mixes with the bypassed discharge gas will have the greatest effect on the suction temperature (i.e. superheat), so the liquid expansion valves were selected to control this process variable. Because there are three liquid lines to control the capacity of the compressor load stand, the selection
process for which expansion valve to use is the same thresholding procedure as the suction pressure with a threshold of 2 °C (3.6 °F).

3.3. Discharge Pressure
The discharge pressure is controlled by adjusting the flow of water through the condensers. The PI controller adjusts the flow rate of water to meet the desired condition by adjusting two devices: a bypass valve and the rotational speed of a pump. A schematic of this setup is shown in Figure 3. The condensers of the load stand are connected to an intermediate cooling water supply line, which then interacts with two intermediate heat exchangers that are subsequently connected to the building chilled water supply shown in Figure 3. The amount of heat transfer within the condensers is dependent on the mass flow rate of water, seen in Equation 2.

\[
\dot{Q} = \dot{m}_w c_p (T_{w,exit} - T_{w,in}).
\]  

(2)

This expression suggests, as more water is bypassed, less heat will be transferred from the load stand to the cooling water, thus increasing the condensing temperature of the load stand, which in turn increases the discharge pressure.

The bypass valve acts as the coarse adjustment of the flow rate and the pump speed is used to fine-tune the water flow. The bypass valve position is adjusted from fully closed to fully open. The pump acts as fine adjustment because changes in the rotational speed of the pump tend to create smaller changes in the system than the bypass valve. By increasing the rotational speed of the pump, the flow rate of water will increase, which will cause an increased amount of heat transferred from the refrigerant to the cooling water in the condensers. Therefore, the pump speed is indirectly proportional to the condensing temperature.

The purpose of having two intermediate heat exchangers and two condensers is to be able to control within the range of capacities. By using only one of the condensers or intermediate heat exchangers, the size of compressor that can be supported by the load stand is reduced, which is beneficial for testing smaller compressors that have a lower capacity rating.

The process for selecting which of these devices to use is similar to that of the expansion valves. A threshold is set, which defines when the controller switches from coarse to fine control. Then, by setting correct proportional and integral gains, all of the controllers move their respective devices to bring the system to a desired set point.
3.4. PI Controller Tuning
To adequately control the three parameters of the load stand automatically, the controllers had to be tuned with appropriate gains so that the control variable, $u(t)$ in Equation 3 would converge to the desired setpoint by minimizing the error, $e(t)$, the difference between the setpoint and the current value of the process variable.

$$u(t) = K_p e(t) + K_i \int_0^t e(t')dt' + K_d \frac{de(t)}{dt} \tag{3}$$

The controllers do not use the derivative gain, $K_d$, they only use proportion and integral gains, $K_p$ and $T_i$ respectively, where

$$K_i = \frac{K_p}{T_i}. \tag{4}$$

Proportional and integral gains were found using the tuning method described by Ziegler and Nichols[7]. During this method the controllers started with initial values of $K_p = 0$ and $T_i = \infty$, while the system was stabilized at some arbitrary point. Then the value of the controller was increased by 10 percent and the proportional gain, $K_p$ was changed until the critical gain, $K_{pu}$, was reached, where sustained oscillations were observed. The period of oscillation at this point, $T_u$, was found for each of the controllers separately. Ziegler and Nichols[7] determined that the optimum values for the proportional and integral gains, respectively, are

$$K_p = 0.45K_{pu}. \tag{5}$$

and

$$T_i = \frac{1.2}{T_u}. \tag{6}$$

4. Test Results and Discussion
To test the efficacy of the control strategy a series of preliminary tests were performed using a Bitzer GSD80485VAB4-1 40-Ton scroll compressor as a way to verify that all of the equipment works properly and that steady-state conditions can be reached using the controllers, within a reasonable degree of accuracy. According to ASHRAE Standard 23.1[4], the suction and discharge pressures must be maintained within $\pm 1$ percent of the absolute pressure that is specified and the suction temperature must be maintained within $\pm 1$ K ($\pm 2$ °R) of its specified value. This is the maximum error that can be allowed for each set point when testing. However, the instrumentation selected for the load stand is capable of significantly better systematic uncertainty than this requirement ($\pm 0.05$ percent of full-scale for pressures and $\pm 0.2$ K for the suction temperature ). Thus, to meet standard 23.1 requirements the controllers need to only reduce the random uncertainty such that it does not add significantly to the instrumentation systematic uncertainty. Using the methods described in ASME PTC 19.1[8] the random uncertainty ($\bar{s}_x$) for each operating condition was calculated using Equation 7,

$$\bar{s}_x = \frac{s_x}{\sqrt{N}} \tag{7}$$

where $s_x$ is the sample standard deviation over the sample length, $N$ (1000 points), given by Equation 8, in which $x_j$ is the measurement value and $\bar{x}$ is the sample mean.

$$s_x = \sqrt{\frac{\sum_{j=1}^{N} x_j - \bar{x}}{N - 1} \frac{N}{N - 1}} \tag{8}$$
4.1. Control Strategies for Resolving Instability

Because the control variables for the compressor load stand are connected, it is challenging to control each variable independently because a change in one variable causes changes in the others as well. Throughout the preliminary testing process a couple of issues were experienced that provided unsatisfactory results. The first is the coupling between suction temperature and pressure and the second is the control over cooling water flow. Without addressing both these issues, a steady-state data point was never able to be reached with acceptable setpoint error and minimized random uncertainty.

4.1.1. Coupling of suction temperature and pressure. One of the difficulties in maintaining steady data throughout the duration of the test was the connection between suction temperature and pressure. As the suction temperature changes, a significant change in the pressure can be observed as well. As a result, rapid changes in the temperature causes the pressure to change rapidly, which the suction pressure controller responds, creating a cycle of one controller responding to the other. This creates periodic instability, where the values start near the set point but diverge as the controllers fight each other. This is visualized in Figure 4 which shows the suction pressure at 10 °C (50 °F) evaporating temperature, 32.2 °C (90 °F) condensing temperature, and 11.1 K (20 °R) superheat.

The instability made it impossible to collect a steady-state average or estimate the random uncertainty of the data. To correct this instability, the PI gains for the suction temperature controller were lowered below that of the suction pressure by roughly an order of magnitude. The objective of this change is to control the temperature very gradually so that the pressure response is not large enough to cause over-correcting by its controller. The resulting adjustments to the PI gains resulted in the data presented in Figure 6.

4.1.2. Sensitivity of discharge pressure to cooling water temperature. Another element of instability in the load stand during the preliminary test phase was the sensitivity of the discharge pressure to changes in the cooling water temperature. Initial tests recorded a large difference in the condenser water inlet and exit temperature. The test shown in Figure 5 had an average entering water temperature of 11.94 °C (53.5 °F) and an exiting water temperature of 28.06 °C (82.5 °F). Using Equation 2, a large temperature difference will be more sensitive to changes in mass flow rate of water. Because of this, small changes in the flow rate of water will have a large effect on the heat transfer rate. This causes large, rapid changes in the discharge pressure, which the controllers respond to, creating more large changes. As a result, the system never completely stabilizes. As Figure 5 shows, the system had an undesirable, large oscillatory behavior because of this large temperature difference of water across the condenser. The steady-state setpoint error was negligible and the random uncertainty was 0.049 kPa (0.0071 psia).

To resolve this issue, the flow rate of chilled water through the intermediate heat exchangers in Figure 3 needed to be reduced significantly, to increase the temperature of water entering the condensers. One of the intermediate heat exchangers was shut off entirely from the building chilled water supply and the other had its chilled water supply choked by partially closing the shutoff valve. With the flow rate of chilled water, the amount of heat dissipated from the load stand’s water supply decreased, which increased the water temperature at the inlet of the condensers to 23.89 °C (75 °F). Doing this led to increased stability of the discharge pressure, which yielded better steady state results as shown in Figure 7.

4.2. Steady-State Results

A matrix of data was tested on the compressor to validate that the load stand can provide control of the testing conditions using R134a as the working fluid. Figures 6-8 show the one such condition that was tested, 10°C (50 °F) evaporating temperature, 32.2°C (90 °F) condensing...
instability in suction pressure caused by controller over-correction

Figure 5. Discharge pressure of a system with a large condenser water temperature difference.

temperature, and 11.1 K (20 °R) of superheat at the inlet of the compressor. From this condition, the suction pressure setpoint is 415 kPa (60.17 psia), the discharge pressure setpoint is 821 kPa (119.1 psia), and the suction temperature setpoint is 21.1 °C (70 °F). The figures show how the data compares to the set point over the test length of 1000 samples at 0.37 samples/second sample rate. The orange line represents the set point and the red lines represent the systematic error for the instruments associated with each measurement. Along with these figures, the random uncertainty was calculated.

Figure 6 shows that the suction pressure holds well within the error range of ±0.5 psia for the duration of the test, with an average value of 415 kPa (60.17 psia) (zero error relative to the setpoint). The calculated random uncertainty for this test is 0.016 kPa (0.0023 psia). This provides verification that the suction pressure can be precisely controlled.

Figure 6. Suction Pressure at 60.17 psia

Figure 7. Discharge pressure at 119.1 psia

The discharge pressure, with small exceptions, stays within the desired range of error as well for the 1000 samples tested as is seen in Figure 7. It had an average value of 820.8 kPa (119.05
psia) with a random uncertainty of 0.038 kPa (0.0055 psia). A 0.2 kPa (0.05 psia) setpoint error is well within the desired range of acceptable values and the random uncertainty does not add significantly to the systematic uncertainty, so the control scheme for the discharge pressure can be used to collect accurate steady-state values.

Likewise, the suction temperature maintains a reasonable amount of steadiness throughout the duration of the test. The suction temperature was more difficult to control because of its connection with suction pressure. Temperature changes are very sensitive to changes in the pressure, so as the controllers for suction pressure change, they can cause undesirable large changes in the suction temperature. As a result, keeping the suction temperature within the desired amount of error is more difficult. However, as is seen in Figure 8, the control scheme keeps the temperature for the current test at the setpoint within a reasonable degree of error. The average suction temperature for the test was 21.08°C (69.94°F). Compared to the desired temperature of 21.1°C (70°F) the load stand is able to test within the desired parameters with an error of 0.09 percent and a random uncertainty of 0.0028 psia.

Figure 8. Suction Temperature at 70°F

The metric by which data can be considered good is the random uncertainty. The target amount of random uncertainty that was decided upon was 0.01 percent of the systematic uncertainty. This would effectively eliminate any uncertainty other than that of the measurement device. By resolving the issues that lead to instability, the amount of random uncertainty was reduced to within that target as is evident in Figures 9 and 10. These figures show how a finely tuned control scheme will affect the measurement uncertainty. From using Equations 7 and 8, it is seen that as more samples are taken for a test, the amount of random uncertainty should decrease and would cause the total uncertainty to asymptotically approach the systematic uncertainty. Figure 9 shows the case before resolving the controls issues. It can be seen the system total uncertainty to diverge from the systematic uncertainty, indicating that this controls scheme is not viable for collecting steady-state results. In contrast to this, Figure 10 shows how the total uncertainty does approach the systematic uncertainty over the length of the test period, giving validation that after correcting the instability, the control scheme is able to produce accurate steady-state results. This also shows how the testing length can be optimized. In this test, the total uncertainty reaches the target uncertainty of 0.01 percent more than the systematic uncertainty at about 275 samples. Being able to know when the uncertainty reaches this point could significantly decrease the amount of time that it takes to collect a steady-state condition.

5. Conclusions
A controls system for a hot-gas bypass compressor load stand was created in order to quickly and accurately move between design conditions. Using a range of controls from coarse to fine, the stand is able to automatically reach test conditions for capacities from 10-80 tons. A series of valves and pumps are controlled with an array of PI controllers which were tuned using
preliminary testing on a 40 ton scroll compressor with R134a as the working fluid. Initial
tests were done with the load stand to determine its ability to maintain control over specified
setpoints. It was found that accurate steady-state data was unobtainable without further tuning
of controllers and resolving other reasons for instability, such as sensitivity of the conditions to
changes because of a cooling water temperature that was too low. By resolving the issues of
instability, the system was able to achieve a total uncertainty for a 45 minute test period of
0.62325 percent, compared to 0.62323 percent if random uncertainty was neglected.

In the future, the load stand will undergo a final commissioning with a 75-ton screw
compressor. This will push the systems limits and validate the design of the load stand. After
commissioning, a series of tests using Low-GWP refrigerants will begin, so as to characterize
the performance of compressors using this new line of refrigerants.

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