Modification of a compressor performance test bench for liquid slugging observation in refrigeration compressors

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Abstract. Compressor performance test procedures are defined by the standard DIN EN 13771, wherein a variety of possible calorimeter and flow rate measurement methods are suggested. One option is the selection of two independent measurement methods. The accuracies of both selected measurement methods are essential. The second option requires only one method. However the measurement accuracy of the used device has to be verified and recalibrated on a regular basis.

The compressor performance test facility at the Technische Universität Dresden uses a calibrated flow measurement sensor, a hot gas bypass and a mixed flow heat exchanger. The test bench can easily be modified for tests of various compressor types at different operating ranges and with various refrigerants. In addition, the modified test setup enables the investigation of long term liquid slug and its effects on the compressor. The modification comprises observational components, adjustments of the control system, safety measures and a customized oil recirculation system for compressors which do not contain an integrated oil sump or oil level regulation system. This paper describes the setup of the test bench, its functional principle, the key modifications, first test results and an evaluation of the energy balance.

1. Introduction

To determine the right refrigerant compressor for a given application most of the time one can only rely on data provided by the manufacturers. Since some applications cannot be operated statically over an extended period of time and tend to transcend the limitations of the specified “safe” area of operation, it is reasonable to examine the effects of these “unsafe” operation modes. However, standardized compressor performance test systems are designed to prevent operating modes which could result in a failure of the tested machine.

For the purpose to investigate a compressors possible failure mode due to such overload, the test facility has to be modified especially regarding its underlying system control.

The content of this paper will present all the modifications and measures taken, to provide a test facility which can be operated in different ways beyond the specified operation limits of the compressor. The most important aspect is the provision of liquid refrigerant at the compressor inlet to simulate short and long-term liquid slug. Furthermore the challenge is taken to operate compressors designed without an oil sump in the same test bench, which was designed and used for compressors which rely on an oil separation and return cycle.
2. Setup of the performance test bench

2.1. Components, pipework and instrumentation

Figure 1 shows the piping and instrumentation diagram of the compressor performance test bench, including the oil recirculation and measurement devices subsequent to their modification.

This particular compressor performance test bench was designed according to DIN EN 13771 [1]. The underlying measurement concept is described in subsection 5.6 of this standard. A refrigerant flow meter is used to determine the gas volume flow rate at the suction or discharge side of the compressor. The use of this method is limited by a few conditions. The first requirement states a maximum of 1.5% of the oil mass to be contained within the refrigerant gas at the flow rate measuring position. Another requirement is to prevent the refrigerant flow to include any liquid droplets to ensure a measurement in the superheated gas area. This second condition has a special significance for the aspired test of liquid slug, resulting in the situation that no performance measurement can be conducted while providing liquid. Merely the consequences of periodical liquid slug can be investigated with this method.

This setup allows the testing of various types of compressors. First tests were performed on reciprocating piston compressors while current tests are conducted on scroll compressors. The electrical power consumption of these compressors is limited to 30 kW, while most components are designed for a maximal conveyed volume flow of 150 m³/h. Additionally, tests with highly flammable refrigerants such as butane are possible.
Figure 1. piping and instrumentation diagram, compressor test bench, TU-Dresden
The evaporation pressure and outlet temperature are controlled by two motorized expansion valves located in the condensation segment (VII) and the bypass line for the hot pressurized gas (VIII). The condensation pressure and temperature are set by a valve which controls the flow of cooling water through the condenser (IX). In contrary to the use of a liquid line flow meter, in this set up the condenser does not need to subcool the liquid. A receiver (X) is installed to provide liquid refrigerant to the expansion valve VII.

The evaporation will be achieved by a mixed flow heat exchanger (IV). This module works as an internal heat exchanger where the primary flow from the condenser (6) is evaporated by gaining heat from a mixture of refrigerant consisting of the flow exiting the primary heat exchanger side (7) and the refrigerant flowing through the bypass section (3). This evaporator is used primarily to ensure the superheating at the compressor inlet. To provide liquid refrigerant for liquid slug tests with this kind of configuration a control system is required that is able to forcefully shut off the bypass gas flow.

After being evaporated the combined refrigerant flow from the bypass and condensation segment enters the volume flow meter (V). For an automated operation during the performance tests including temporary liquid slugging, the volume flow meter has to be protected against any liquid refrigerant entering with the gas flow. If liquid refrigerant would get in between the two rotary pistons of the flow meter, an evaporation of the fluid would occur which leads to a deformation and failure of the flow meter. Because of that two shut-off valves have to be operated during the bypass shut off mode as well. These valves will respectively close the inlet line of the flow meter and open a line leading directly to the compressor inlet.

Shock absorbers (flexible lines) are used to reduce the stress from vibrations on the pipework. While most connections between pipes and fittings are soldered to achieve a high overall system leak tightness, the compressor is connected by screwed fittings. The intention is to improve the flexibility of the test bench and enable a fast exchange of test objects.

During the measurement procedure pressure and temperature values are recorded at all times. The system is completely insulated to reduce the influences of any possible heat flow across the system boundaries.

![Figure 2](image)

**Figure 2.** Isometric view of the original compressor test bench, TU Dresden

### 2.2. Key Modifications

The mostly automated testing of compressors requires from the control system to automatically operate the valve needle positions and other variables like the frequency of a variable speed
compressor. Since the compressors that are going to be tested on this bench will be operated beyond their specified application limitations the use of a conventional control system is not recommendable and the controller was programmed especially for this performance test bench using the software DASYLab®. The use of a customized control system allows the operator to intervene in the process at any time, improving its flexibility.

In case of the described test bench, the before mentioned manually operated shut-off valves had to be replaced by solenoid valves, which can be automatically controlled according to the operation mode and the refrigerant state at the flow meter inlet. The data acquisition system is a modular designed system to allow an easy extension for future needs.

A frequency inverter was implemented in the system for tests with variable speed compressors. With regard to the controller the speed regulation is based on an analogue output signal. Because the test bench is built for the testing of many different compressors of a wide power range, the frequency inverter is not permanently installed within the power panel but used as an exchangeable component to always provide the best fitting combination with the compressor.

The measurement and control program was adjusted to fit the new requirements of the automated operation. In the basic operation mode the valves, which are addressed by analogue signals, are controlled by PID modules. For the instance of a liquid refrigerant flow at the compressor inlet, the valves need to be forced into specific positions. To enable the liquid refrigerant to flow through the system while generating as little gas as possible at the compressor inlet, the expansion valve VII of the condensation segment has to be fully opened. At the same time the bypass valve VIII has to be closed. This in return will prevent the evaporation of the liquid refrigerant when traversing the mixed flow heat exchanger.

Due to the amount of the liquid stored in the receiver, the described operation of the expansion valves can feed liquid to the suction line for a longer period of time. Meanwhile the before mentioned solenoid valves are activated simultaneously and are preventing the liquid refrigerant from entering the volume flow meter by opening the bypass. Another method to induce liquid to the suction line is to feed the controller with adjusted control parameters. When the saturation temperatures are set and the controller receives signals from the sensors which do not fit the actual state of the refrigerant, the control can be used to create a two phase refrigerant flow. An example is given in Figure 3. To produce a partial liquid flow at the compressor inlet the pressure signal from which the control parameter for the PID module is derived, has to be adjusted to a lower value. At the same time the amount of superheat has to be set to a minimum. The difference between the transferred pressure signal $p_{\text{trans}}$ and the real operating pressure $p_{\text{real}}$ will then result in a two phase refrigerant flow at the compressor inlet.

The sight glass (VI) is used to monitor the state of the refrigerant at the suction side of the compressor and to evaluate the oil return rate qualitatively. The integration of another sight glass within the oil return line is planned to provide an independent verification option for the oil return rate.

However, the most significant modification took place at the oil separation cycle itself. Because of the lack of an oil sump and the corresponding oil level regulator in certain compressor models, the oil
has to be fed into the suction line directly from the oil receiver (XI). To control the oil return rate a manually operated needle valve (XIII) is used. The valve is supposed to be set to a specific position and not to be changed during testing. The valve itself separates the oil receiver on the high pressure side from the suction line’s low pressure level and can influence the controller upon a change of its opening degree. A view of the rebuild suction line is given in Figure 4. Suction line of the test bench including the sight glass, oil return line and suction state sensor.

3. Experimental results & functionality tests

3.1. First experimental results
Before the modifications were realized the performance test bench was operated to provide a reference measurement. The power consumption of a piston compressor from Bitzer®, the 4NES-20Y-40P was measured at different operation points. Afterwards the measured value was compared to the data sheet of the compressor [2]. To be in compliance with the standard DIN EN 13771, the deviation of the measurement has to be below 4 % from the data sheet. The data was averaged over a runtime of 15 min as required by the standard. The results are shown in Table 1.

| Variable                      | Target value | Measured value |
|-------------------------------|--------------|----------------|
| Compressor inlet temperature [°C] | 20.00        | 20.16          |
| Condensation temperature [°C]   | 70.00        | 71.05          |
| Evaporation temperature [°C]    | 15.00        | 14.17          |
| Power consumption [kW]         | 13.03        | 13.73          |
| Rel. error (power consumption) [%] |              | 5.39           |

The manual control of the expansion valves (VII and VIII) increases the oscillating characteristics of the machine and thereby the settling time of the values. As a consequence the achieved temperature settings have a high deviation from the target values. When a corrected value is used to calculate the power consumption for a condensation temperature of 71 °C the calculated relative error is reduced to 4.7 %. To reach the desired maximum error of 4 % the sensors need to be recalibrated.
In general for a test bench working with a single volume or mass flow meter the standard defines a period of 6 months as the recalibration interval for the flow meter. It is assumed that the prolonged downtime without the recalibration of the bench is the cause for the observed deviations in the measured data. Thus the flow meter has to be recalibrated to reach the desired measurement accuracy.

3.2. Functionality tests

After the performance test of the piston compressor the controller was reprogrammed. The functionality was tested using a relative low capacity scroll compressor of the ELV21 series manufactured by Bitzer. This compressor has no oil sump and a speed control working on either a pulse width modulation or a 0 to 10 V signal.

An important test was the investigation of the PID modules to control the expansion valves. The proportional, integral and derivative terms were specified using iterative experiments. Chart 1 shows a temperature profile for an experiment determining the functionality of the PID controller for the bypass expansion valve. The error band is below 10 % at a target value of 10 °C (s. Chart 2). The mean temperature for the runtime of 15 minutes at the compressor inlet is 10.13 °C. While the accuracy of this value is no big improvement in comparison with the compressor inlet temperature measured during the first experiments with the 4NES-20Y-40P, a very high improvement was realized in regards to the run-in period. It took only about 200 s from standstill until a quasi-stationary process whereas some manually operated tests had taken more than 6 fold of that time. Since the PID control was implemented for both expansion valves as well as for the water flow control valve, the coordination of all the valves had to be adjusted during following experiments. In the end the same accuracy of the controller is desired for the condensation and evaporation temperature without increasing the settling time.

![Chart 1. Compressor inlet temperature profile, PID controller bypass expansion valve](chart1.png)

![Chart 2. Compressor inlet temperature profile, detailed error band](chart2.png)

Other functionality tests included the automatic switch for the relay-controlled solenoid valves as well as for the liquid slug activation process. DASYLab® is a data acquisition and control system based on a visual programming language. Figure 5 shows a section of the program depicting the elements responsible to process the control system.

The control module (Control1) is used to provide a target value for the condensation, evaporation and compressor inlet temperature for the PID controller. However, the frequency signal is predefined by the user and transmitted to the analogue output module as a constant value. The expansion valve signals are forwarded to a formula editor module. A switch (EXV-Switch) is used as a third input variable and primary condition for an if-then-function (EXV If/Then).

The simultaneously reversed change of the magnetic valve positions is achieved by using one switch module (MV-Switch) in combination with a logical link (Bit-Logik00) which operates one of the valves contrarily to the input signal.
Figure 5. Program section DASYLab®, control system

The DASYLab® program communicates with a Microsoft Excel worksheet. The worksheet is fed with data from the acquisition modules and uses an add-in for the property program Refprop of the National Institute of Standards and Technology (NIST). The measurement data are used in combination with Refprop to determine all other necessary properties (e.g.: specific enthalpy, specific entropy and specific volume).

4. Energy and mass balance

When a compressor performance test facility is operated in compliance with the standard the user also has to calculate the energy and transport values in a specified way. Values defined by the standard [1] are the refrigerant mass flow, cooling capacity, volumetric efficiency/efficiency of the compressor and the COP of the standardised cycle.

At this test bench with a suction line gas flow meter, the mass flow \( \dot{m}_R \) has to be calculated using the measured refrigerant volume flow \( \dot{V}_R \) and the mean gas density across the volume flow meter \( \rho_m \):

\[
\dot{m}_R = \dot{V}_R \rho_m
\]

(1)

The cooling capacity \( \dot{Q}_0 \) is not determined by the actual evaporation capacity of the installed heat exchanger. The same applies to the condensation capacity. To standardise all compressor performance tests, the following calculation uses instead the specific enthalpy of the saturated liquid state at high pressure which is also the evaporator inlet enthalpy \( h_{l,h} \). The high pressure is measured at the compressor outlet, neglecting further pressure losses until reaching the expansion valve. The evaporator outlet enthalpy is taken as the measured compressor inlet state.

\[
\dot{Q}_0 = \dot{m}_R (h_{c,in} - h_{l,h})
\]

(2)

To calculate the volumetric efficiency \( \lambda_c \) the theoretical volume flow \( \dot{V}_{c,th} \) of the tested compressor and the inlet density \( \rho_{c,in} \) have to be known.

\[
\lambda_c = \frac{\dot{m}_R}{\dot{V}_{c,th} \rho_{c,in}}
\]

(3)
An overall efficiency of the compressor $\eta_c$ can be calculated with the measured data of the compressor inlet state and the isentropic compressor outlet enthalpy ($h_{c,\text{out},s}$). Furthermore the measured power consumption of the electric motor $P_{c,\text{el}}$ is needed to provide the reference value.

$$\eta_c = \frac{\dot{m}_R (h_{c,\text{in}} - h_{c,\text{out},s})}{P_{c,\text{el}}} \quad (4)$$

Finally the coefficient of performance is determined from the cooling capacity and the measured power consumption, according to equation (5).

$$\text{COP} = \frac{Q_o}{P_{c,\text{el}}} \quad (5)$$

5. Conclusion

The modifications of the data acquisition and control system resulted in an overall improvement of controllability of the presented customised performance test bench with newly achieved settling times of about 200 s for a single operation state.

The most important modifications were carried out at the suction line and the oil recirculation system. These modifications were verified in a series of functionality tests and enable the test bench to be used for tests of compressors which are not equipped with an oil sump. An analysis of such a compressor was done after 80 h of operation time. The draining and measuring of the oil load from the compressor did result in the same amount that the compressor was filled with initially by the manufacturer to prevent an unlubricated start.

The operation of compressors without an oil sump results in the demand for special modifications such as monitoring options to assure the lubrication of the compressor at all times. The installation of two sight glasses, one of which is located in the oil recirculation line, can provide the needed verification for the oil return rate. The second sight glass which is located in the compressor suction line allows for the qualitative evaluation of the liquid refrigerant ratio at the suction state when the liquid slug operation mode is engaged. At the same time the newly implemented magnetic valves offer an automatic protection mechanism for the rotary flow meter. These changes to the test bench provide the necessary conditions to conduct liquid slug tests.

With the recalibration of the flow meter the test bench allows for compressor performance tests within the defined tolerance of the standard DIN EN 13771 to be conducted. After the implementation of a frequency converter the test bench can be used for the testing of compressor with variable speed operation modes.

References

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[2] Bitzer Kühlmaschinenbau GmbH 2012 Data Sheet: Semi-Hermetic Reciprocating Compressors – New Bitzer Ecoline R134a/R404A