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Design of a Compressor Load Stand Capable of Supplying Two-Phase Refrigerant at Two Intermediate Pressures

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ABSTRACT

The development of compressors with refrigerant injection ports provides a less complex and less costly alternative to implementing multi-stage compressors with economization. The ports can be used to inject economized refrigerant during the compression process, which provides the desired cooling effect and decreases the work required to compress the gas per unit mass. Therefore, this paper presents the design of a compressor load stand for testing compressors with multiple injection ports. The load stand is based on a traditional hot gas bypass configuration but is capable of supplying refrigerant to injection ports at two different pressures between the compressor suction and discharge pressures. In addition, the state of the injected refrigerant can be controlled such that it is either superheated vapor or a saturated liquid-vapor mixture. To guide the design of the bench and size system components, a model was developed to predict the system performance with a commercially available R-410A compressor that has a single injection port. The model is used to predict the range of injection conditions that the load stand will be able to achieve over a range of operating conditions.

1. INTRODUCTION

With continued concern over the energy efficiency of refrigeration cycles, much research has been done to further improve the performance of the traditional vapor compression system. The application of compressors with refrigerant injection has been considered as a substitute for multi-stage compressors, which are the conventional technology used for economization in vapor compression cycles. The compressors with injection ports provide a less complex and less costly alternative to multi-stage compressors. The injected refrigerant provides the desired cooling effect and decreases the work required to compress the gas per unit mass. Depending on the configuration of the cycle modifications, the injected refrigerant either can be a vapor or saturated liquid-vapor mixture.

Experiments have shown that injecting liquid or low quality refrigerant is effective for reducing the compressor exit temperature and improving system reliability. Cho and Kim (2000) experimentally investigated the impact of liquid injection on an inverter-driven scroll compressor at different frequencies and concluded that liquid injection improves the coefficient of performance (COP) of the air-conditioning cycle at high frequency and reduces the compressor discharge temperature, which will improve the reliability of the compressor. However, the benefits of liquid injection are lost at low frequencies because of high leakage flow rates. Liu et al. (2008) performed experiments employing a rotary compressor with a liquid injection port for heat pump water heater applications in cold regions. The experiments found that the discharge temperature drops significantly because of the injected liquid refrigerant, but the capacity will be almost the same as that without injection because the mass flow rate of injection is very small. Cabello et al. (2010) experimentally compared the direct liquid injection and sub-cooler systems in two-stage vapor compression cycles. Although the liquid injection provided a reduction in discharge temperature, the authors found that the sub-cooler system had a higher COP and cooling capacity that the direct liquid injection system.

While liquid injection reduces the compressor discharge temperature, previous studies have demonstrated that injecting refrigerant vapor improves the cooling or heating capacity of the system. Heo et al. (2010a) experimentally investigated the impact of flash tank vapor injection on the heating performance of a two-stage heat pump with an inverter-driven twin rotary compressor and observed an increase in both the COP and heating capacity with injection. Liu et al. (2010) performed a series of experiments employing a two-stage rotary compressor with flash tank vapor injection for a heat pump water heater application and found that it is mostly the vapor injection which made a contribution to enhancing the heating capacity and improving the system performance in cold region. In addition, the paper concluded that the intermediate pressure should be adjusted at intervals based on condensing pressure in order to get the best operation. Baek et al. (2008) tested the transcritical CO_2 heat pump system with a twin-rotary compressor by varying the gas injection ratio and outdoor temperature and found that the transcritical CO_2 heat pump with gas injection had better performance at low outdoor temperature than normal situation. The results showed that the ratio of heating capacity and COP rose with an increase in the gas injection ratio for the increasing total mass flow rate.

Despite the many studies on cycles operating with liquid or vapor injection, very little information is available for cycles operating with injection states between these limits. Liu et al. (1994, 1995) studied the compression of two-phase refrigerant by developing a mathematical model; the model was used to analyze the factors causing slugging problems and the effect of compressor kinematics on slugging. Dutta et al. (1996) studied a two-phase refrigerant injection compression process through experiments and simulations. Three mathematical models, the droplet model, homogeneous model and slugging model, were proposed. The droplet model assumed that the gaseous and liquid refrigerant exist in the control volume separately with different temperatures. The homogeneous model assumed that the two-phase refrigerant had the same temperature throughout. The slugging model assumed that the liquid refrigerant has the same temperature as the gas and the gas is always saturated during the compression process. They found that the homogenous model had good agreement with the experimental results.

Theoretical work suggests that cycle performance with two-phase refrigerant injection can provide greater improvements in COP than vapor injection. Ozaki et al. (1990) investigated two-phase compression heat pump cycles theoretically, in which part of the liquid from the condenser and the gas separated from the economizer are injected into the compressor at an intermediate pressure. It was determined that the improvement of COP and the reduction of the discharge superheat can be achieved by two-phase injection and there occurs an optimum gas-liquid mass flow rate ratio that brings the maximum COP. After that, Ozaki (1992) studied the two-phase compression process experimentally using an injector and pressure chambers consisting of a test section and compression cylinder. Although there were some differences from the real compression cycle, it provided data on a two-phase compression process.

All of the aforementioned studies considered injection processes at one intermediate pressure. However, work also has shown that increasing the number of stages in an economized cycle with a multi-stage compressor improves the cycle performance (Jung et al., 1999; Mathison et al., 2011). It is expected that increasing the number of injection ports would have a similar effect on system performance. Therefore, a compressor load stand will be designed and built for studying the impact of injected refrigerant quality and the number of injection ports on vapor compression cycle performance.

2. LOAD STAND DESIGN

The load stand has built to test compressors with injection ports that operate at up to two different injection pressures. The hot gas bypass test stand, shown in Figure 1, can supply two-phase refrigerant to each injection line over a range of pressures and conditions from saturated liquid-vapor mixture (SLVM) to superheated vapor.

The main loop of the test stand controls the conditions at the inlet to the compressor. After the refrigerant comes out of the compressor, it passes through a set of electric expansion valves (EEVs) used to control the discharge pressure; the two valves are configured in parallel and sized to provide coarse and fine control over the flow. The refrigerant flows through an oil separator and then enters a Coriolis flow meter that measures the total refrigerant flow rate in the system. Next, the refrigerant splits into two streams; a portion of the flow bypasses the condenser and expands directly to the suction pressure, while the remaining flow is condensed in a water-cooled heat exchanger before expanding to the suction pressure. The EEVs used to expand these streams to the suction pressure not only control the pressure drop across the system, but also control the division of refrigerant between the bypass and condensing streams. The refrigerant should be split between the streams such that the desired suction state is achieved when the streams are allowed to mix. Following the mixing process, the mass flow rate of the refrigerant is measured again and it returns to the suction port of the compressor. Hand valves close to the inlet and exit of the compressor can be used to isolate the compressor if it needs to be evacuated and removed from the system.

The two injection loops operate using the same principle as the main loop that supplies refrigerant to the compressor suction state. Before the refrigerant enters the condenser, a portion of the vapor is drawn off into each injection line. Similarly, a portion of the liquid exiting the condenser is drawn off for each injection line. Therefore, each injection loop is composed of a vapor line and a liquid line; a pair of EEVs in each line provides control over the injection pressure and the division of flow between the vapor and liquid streams. In order to determine the division of refrigerant between the two lines, each contains a Coriolis flow meter; the meters are located upstream of the EEVs, to ensure that the fluid enters the meter in a single phase. In addition, the pressure and temperature of the liquid stream is measured upstream of the EEVs since the liquid will enter the two-phase region as it expands, at which point temperature and pressure do not provide sufficient information to fix the state. In the vapor stream, the temperature and pressure measurements can be taken at the EEV exit to determine the vapor properties just before the liquid and vapor streams mix. These property measurements, along with the mass flow rate measurements, will facilitate the application of mass and energy balances to the mixing processes for comparison to the idealized, adiabatic mixing process in the load stand model. Finally, these loops can be shut down in order to isolate the compressor or run tests without injection by using the hand valves located close to the injection ports at the compressor and at the inlet to each injection line.

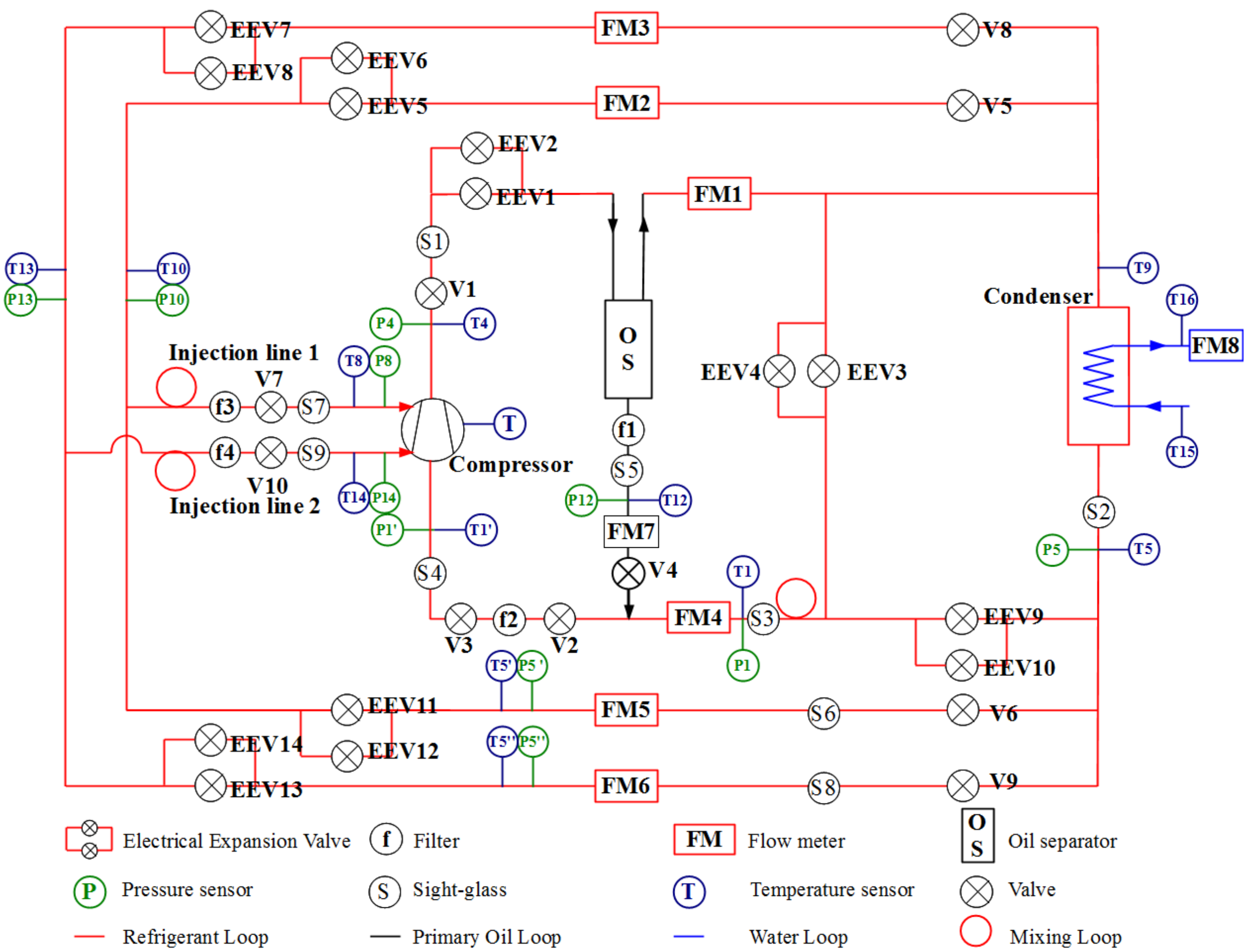


Figure 1: Schematic of load stand

A primary benefit of the hot gas bypass configuration is eliminating the requirement for an evaporator, which reduces the complexity and cost of the system. Instead, the two-phase mixture that results from expanding the condensed refrigerant is heated by the superheated vapor that bypasses the condenser. Additionally, the electric expansion valves can provide precise control over the suction pressure, discharge pressure, and intermediate pressure, which determines the injection mass flow rate. The valves also can be used to control the mixing of superheated vapor and liquid refrigerant to obtain the desired quality for two-phase injection, or superheat for vapor injection.

3. LOAD STAND MODEL

To guide the design of the load stand and assist in sizing system components, a model has been developed to predict its performance over the range of anticipated operating conditions. The load stand is intended for use with R-410A as the working fluid and will be sized to test compressors with capacities up to approximately 75,000 Btu/hr. However, because the load stand must be capable of testing a variety of different compressors both with and without injection, the model should be easily adaptable to serve as a tool for evaluating the impact of compressor selection on system performance.

To accomplish this goal, the model uses manufacturer-supplied data to characterize the compressor performance. This data is typically provided over a range of condensing and evaporating temperatures with a specified superheat at the compressor inlet and subcooling at the condenser exit. For a compressor without injection ports, manufacturers may report the expected cooling capacity, power consumption, current draw, mass flow rate, EER and isentropic efficiency of the compressor under each condition. However, the performance of a compressor designed to operate with economized vapor injection cannot be characterized as succinctly. Because of the economizer, the enthalpy of the refrigerant supplied to the evaporator no longer depends on the degree of subcooling at the condenser exit alone. Therefore, the manufacturer must supply much more information to completely specify the conditions entering the evaporator and the injection line.

Although the manufacturer may supply information that can be used to determine the conditions entering the evaporator, additional information is needed to specify the state of the injected refrigerant. Therefore, providing a detailed description of the compressor performance is much more complex with injection.

It follows that completely describing the performance of a compressor with injection within the load stand model would require significantly more inputs than describing a compressor without injection. However, it was desired to use the same load stand model, and thus the same inputs, for compressors both with and without injection. Furthermore, the load stand model must predict system performance with two-phase economized refrigerant injection, for which compressor performance data is not available. Therefore, it was decided to characterize compressor performance in the load stand model using isentropic efficiency alone. When the compressor inlet conditions (state 1) are known and the discharge pressure (state 2) is specified, the isentropic efficiency can be used to determine the discharge enthalpy:

$$\eta_s = (h_{2s} - h_1)/(h_2 - h_1) \tag{1}$$

In this equation, h_{2s} represents the enthalpy of the refrigerant exiting an isentropic compression process from the inlet state to the exit pressure.

In order to apply this definition to a compressor with injection, the injection process is modeled as an adiabatic, isobaric mixing process between compressor stages, and Equation (1) is applied to each stage of the compressor. For example, Equation (1) can be applied to a compressor with a single injection port by letting state 2 represent the state of the refrigerant in the compressor as it reaches the injection pressure. If state 8 represents the state of the injected refrigerant, a mass and energy balance on the adiabatic mixing process can be used to determine the resulting state of the refrigerant in the compressor, which will be represented as state 3:

$$h_3 = \left(1 - R_{inj}\right)h_2 + R_{inj}h_8 \tag{2}$$

For convenience, the injection mass flow rate ratio, R_{inj}, is defined as the ratio of the injection mass flow rate, $\dot{m}_{inj}$, to the mass flow rate entering the compressor, $\dot{m}_s$:

$$R_{inj} = \dot{m}_{inj} / \dot{m}_s \tag{3}$$

This ratio is defined relative to the suction mass flow rate because it is assumed that injection will have a negligible impact on the volumetric efficiency or mass flow rate entering the compressor. The injection mass flow rate ratio must be specified by the model user, if injection flow rates are available from the compressor manufacturer, or can be varied over a range of values to study the impact on system performance. Following the mixing process, the refrigerant continues to be compressed and Equation (2) is used to calculate the resulting discharge state from the compressor.

Using an isentropic compressor efficiency and an adiabatic, isobaric mixing process to model the compressor with or without injection not only simplifies the model considerably, but also enables the model to predict load stand performance with two-phase injection. In addition, the following assumptions are proposed:

1. Steady-state, steady flow conditions.
2. One-dimensional flow.
3. The compressor can be modeled using an isentropic efficiency.
4. Any injection processes can be modeled as adiabatic, isobaric mixing processes.
5. The pressure drop through pipes is negligible.
6. The refrigerant exits the condenser as saturated liquid.
7. Compared to the heat transfer between the condenser and the heat sink, the heat transfer between the pipes and the ambient is negligible.
8. The throttling devices are isenthalpic, with no work or heat transfer.
9. Kinetic and potential energy changes are small relative to changes in enthalpy and can be disregarded.

The load stand model was implemented using Engineering Equation Solver (Klein, 2009). It requires the user to specify the condensing and evaporating temperatures, degree of superheat at the compressor inlet and subcooling at the condenser at the condenser outlet, compressor power input, mass flow rate and isentropic efficiency, and heating or cooling capacity. The compressor manufacturer typically provides all of these parameters on the performance sheet. Making the assumptions mentioned above, the model then will calculate the thermodynamic properties at each state, the mass flow rate through each line of the load stand, and heat transfer rate in the condenser.

4. RESULTS

The model is intended to provide guidance during experiments and predict the impact of injected refrigerant quality and the number of injection ports on load stand performance. To illustrate how the model serves this purpose, results are presented for the load stand operating with a commercially available compressor, a Copeland ZP44K3E scroll. This compressor is not designed to operate with injection, but it is assumed that the incorporation of injection ports would not impact the isentropic efficiency significantly. While this assumption and the assumption that the injection process occurs instantaneously likely introduce significant error into the compressor model, this study is focused on predicting load stand performance, not compressor performance. Because the compressor does not actually contain injection ports and the injection mass flow rate depends on port design, this flow rate is varied in order to study its impact on the allowed range of injection states, as discussed in the following sections.

4.1 States of the Hot Gas Bypass Two-Phase Injection Cycle

Figure 2 shows the state points of a hot gas bypass cycle operating with two-phase injection on a pressure-enthalpy diagram. In this diagram, state 1 represents the inlet to the compressor and state 2 represents the conditions in the compressor as the refrigerant reaches the injection pressure. An instantaneous mixing process then occurs at the injection pressure, with the two-phase injection state represented by state 8 and the resulting mixture represented by state 3. Next, the refrigerant is compressed from state 3 to state 4, the compressor discharge state. After exiting the compressor, the refrigerant expands through a set of EEVs to a slightly lower pressure (state 9) that is fixed by the condensing water temperature. The cooling water for the load stand will be provided by a dedicated chiller with both a chilled water circuit and a tempered water circuit, allowing the water temperature to be varied from approximately -5°C to 120°C. Therefore, the condensing temperature and pressure can be closely controlled by adjusting the chiller setpoint.

The portion of the refrigerant that passes through the condenser exits as a saturated liquid at state 5. With one injection line in use, a portion of the condensed refrigerant will be diverted to the injection line while the rest expands through a set of EEVs to the evaporating pressure (state 7). The refrigerant at state 7 mixes with the stream of refrigerant that bypassed the condenser and expanded directly to the evaporating pressure (state 11). The mass flow rate of each stream is adjusted until the mixing process results in the desired compressor suction conditions at state 1.

A similar process is used to establish the desired injection conditions. A stream of the condensed refrigerant at state 5 is expanded to the injection pressure (state 6) using EEVs, while another stream bypasses the condenser and expands directly to the same pressure (state 10). The division of refrigerant flow between these two streams is adjusted such that mixing them together results in the desired injection conditions.

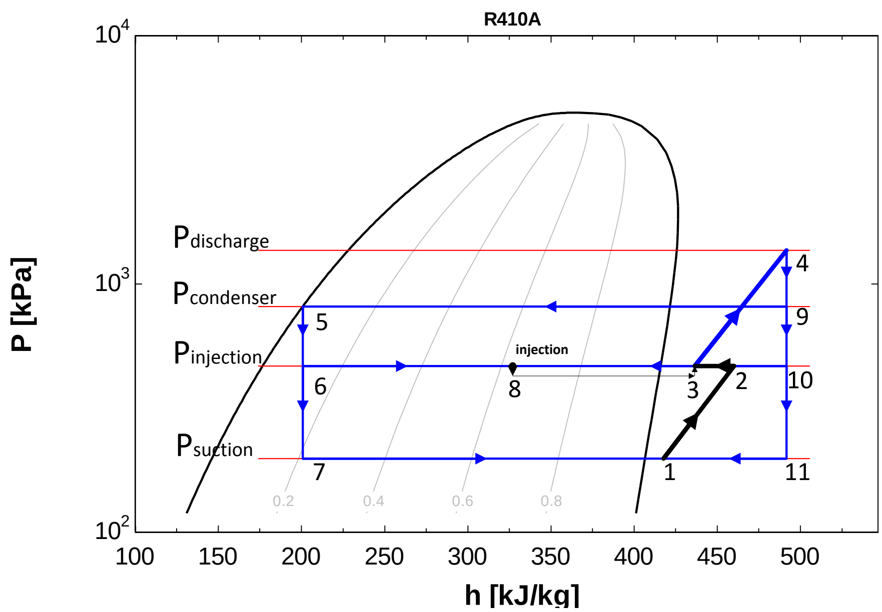


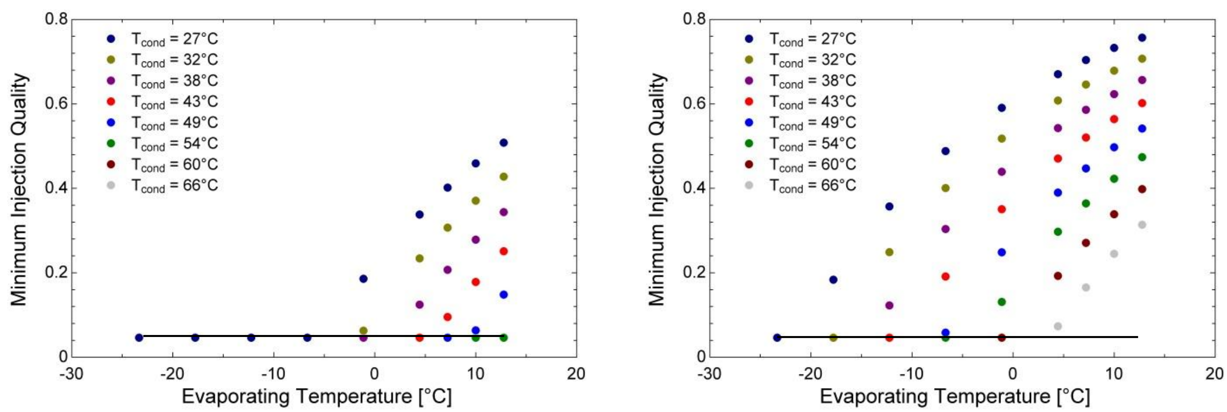
Figure 2: P-h diagram of load stand model with one injection port

4.2 Impact of Suction Conditions on Minimum Achievable Injection Quality

Because one of the objectives of the load stand is to evaluate the impact of two-phase injection on compressor performance, it must be able to control the quality of the refrigerant in the injection line. Ideally, the injected refrigerant could be a saturated liquid, SLVM, saturated vapor, or superheated vapor. However, if the refrigerant exiting the condenser at state 5 is saturated liquid, Figure 2 indicates that the refrigerant in the injection line at state 6 will always have a quality greater than zero. Saturated liquid injection can only be achieved by subcooling the refrigerant exiting the condenser and injecting the refrigerant at state 6 without adding any of the bypassed vapor (state 10). For the purposes of this paper, the minimum quality of refrigerant that can be injected will be analyzed for a system without subcooling, but the same analysis applies to systems with subcooling.

The minimum injection quality will depend on both the injection pressure and the injection mass flow rate. While these two parameters are typically linked by the design of the compressor, with higher injection pressures driving higher mass flow rates, the model does not contain a detailed compressor sub-model. Therefore, the injection pressure and injection mass flow rate must be specified or varied over the range of conditions of interest. In addition, the minimum injection quality will depend on the load stand operating conditions. In order to evaluate the impact of operating conditions on minimum injection quality, the model is exercised with evaporating temperatures from -40°C to 7°C and a range of condensing temperatures from 30°C to 50°C, for a total of 33 conditions. The refrigerant is specified to be superheated 11°C at the compressor inlet and saturated liquid at the condenser exit. The compressor isentropic efficiency and the mass flow rate into the compressor at each operating condition are obtained from the manufacturer-provided performance sheet. Finally, the intermediate pressure is fixed at 1000 kPa and the injection mass flow rate ratio is varied from 0.2 to 0.4 to study the impact on minimum quality.

The minimum injection quality can be achieved under two possible conditions. The most obvious method to minimize quality is to specify that the mass flow rate of the bypassed vapor flowing from state 9 to state 10, $\dot{m}_{9-10}$, is zero. In this situation, the minimum quality will be equal to the quality of the two-phase mixture that results from expanding the refrigerant exiting the condenser to the injection pressure. However, because the model allows the used to specify any injection mass flow ratio, it is also possible for the injection process to result in such a dramatic cooling effect that the refrigerant exits the compressor with a lower enthalpy than at its inlet. In this case, it is impossible for the load stand to reestablish the desired suction state because even the bypassed refrigerant would require heating to achieve the necessary superheat. For the purposes of this paper, this situation will be referred to as "overcooling" of the compressor. EES circumvents this problem by specifying that the mass flow rate of condensed refrigerant supplied to the compressor suction state, or the refrigerant flowing from state 5 to state 7 in Figure 2, $\dot{m}_{57}$, must be negative. Although the negative mass flow rate satisfies the mass and energy balances on the system, this situation is physically impossible. Therefore, it is specified that $\dot{m}_{57}$ must be greater than or equal to zero. Note that when $\dot{m}_{57}$ is equal to zero, the low quality of the injected refrigerant will result in compressor discharge conditions that match the suction conditions.



(a) (b)

Figure 3: The dependence of minimum quality on inlet conditions when refrigerant is injected at 1000 kPa with a mass flow rate ratio of: (a) 0.2; (b) 0.4

The minimum injection quality that is physically achievable under both of the previously discussed conditions is plotted in Figure 3 for multiple discharge pressures when the injection pressure and condensing temperature are held constant. The results are plotted as a function of evaporating pressure and temperature, which are defined in the absence of an evaporator as the suction pressure and saturation temperature at the suction pressure, respectively. Because the condensing temperature is held constant regardless of suction or discharge pressure, the state of the refrigerant at the condenser exit remains constant and fixes the state of the two-phase refrigerant supplied to the injection line (state 6 in Figure 2). If this refrigerant is injected directly into the compressor without adding any bypassed refrigerant, then the injection quality is represented by the horizontal line in Figure 3.

However, Figure 3 also shows that bypassed refrigerant must be added to the injection line in many cases to raise the quality of the refrigerant and avoid "overcooling" the compressor. In this case, the injected refrigerant quality is specified such that the compressor discharge enthalpy will match the suction enthalpy, and thus the bypassed refrigerant will reestablish the suction state without mixing in any condensed refrigerant ($\dot{m}_{57}$ would equal zero). The figure reveals that this quality depends strongly on evaporating temperature. Assuming that the isentropic efficiency does not change significantly, decreasing the evaporating temperature while holding discharge pressure constant will result in a greater increase in enthalpy during the compression process. Therefore, a lower quality refrigerant can be injected at low evaporating temperatures without violating the requirement that the compressor discharge enthalpy must be greater than or equal to the suction enthalpy. However, as the evaporating temperature increases and thereby decreases the pressure rise across the compressor, the cooling effect of the refrigerant injection becomes more pronounced and the minimum quality is limited by the need to avoid "overcooling" the compressor.

The transition between these limiting cases can be seen most clearly in Figure 3(a). Note that the transition between a quality limited by the enthalpy of the condensed refrigerant and a quality limited by the need to avoid "overcooling" occurs at lower evaporating pressures as the discharge pressure decreases. This is due to the fact that decreasing the discharge pressure also decreases the enthalpy change across the compressor, which makes it easier to "overcool" the compressor.

The injection mass flow fraction also impacts the minimum injection quality. When the injection mass flow rate is 20% of the suction mass flow rate, Figure 3(a) shows that the minimum quality is generally fixed by the enthalpy at the condenser exit. Only at higher evaporating temperatures does the second constraint become important, requiring the addition of bypassed flow to the injection line to avoid "overcooling" the compressor. When the injection mass flow rate increases to 40% of the suction mass flow rate, the minimum quality is limited by the cycle's ability to reestablish the suction state at almost all of the test conditions. Therefore, greater restrictions apply to compressors that operate with larger injection mass flow rates.

4.3 Impact of Injection Conditions on Minimum Achievable Injection Quality

To study the impact of injection pressure and injection mass fraction on the minimum achievable injection quality, the model is exercised at a suction pressure of 640 kPa, a discharge pressure of 2290 kPa, and with 11°C superheat and 0°C subcooling.

Figure 4 shows the value of minimum quality as a function of the injection pressure. As with an actual economized cycle, the minimum quality of the refrigerant that can be injected on the compressor test stand generally depends on both the injected mass flow rate and the injection pressure. However, for small injection mass fractions, the minimum injection quality does not depend upon the injection flow rate because refrigerant can be expanded directly from the condenser exit and injected without mixing in any bypassed vapor. Therefore, the quality is only limited by the enthalpy of the refrigerant exiting the condenser. For the case plotted in Figure 4, this situation occurs when the injection mass fraction is 1%, 5%, 10% or 20%. Because the injection quality is calculated based on the assumption of an isenthalpic expansion process and longer expansion processes generate more vapor, the minimum quality increases when the refrigerant is injected at lower pressures.

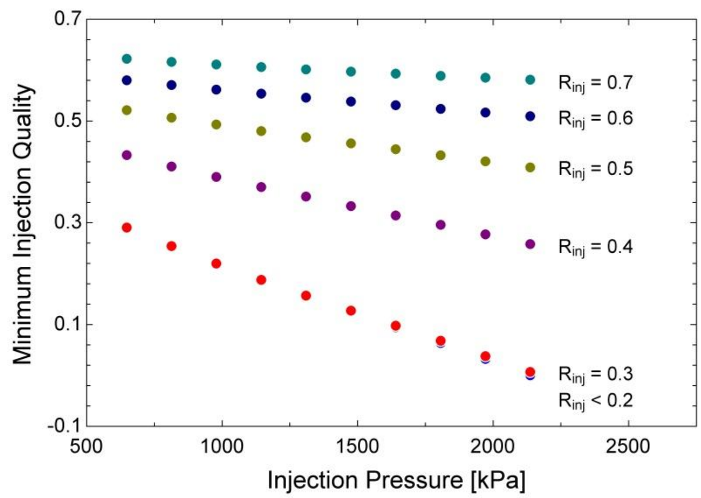


Figure 4: Minimum quality versus injection pressure (640 kPa suction pressure with 11°C superheat, 2290 kPa discharge pressure, and 35°C condensing temperature with 0°C subcooling)

As the injected mass fraction increases, the minimum injection quality is limited by the "overcooling" scenario discussed previously. In order to avoid an enthalpy at the compressor exit that is less than the suction enthalpy, bypassed refrigerant must be added to the injection line. At lower injection mass fractions, adding a small amount of bypassed vapor increases the injection quality dramatically; the effect of mixing in additional vapor becomes less pronounced as the injection mass fractions increase. In all cases, the minimum achievable quality again increases as the injection pressure drops due to the increasing quality of the refrigerant supplied from the condenser. However, the impact of injection pressure becomes less pronounced at high injection mass fractions because the bypassed refrigerant composes a larger portion of the flow.

Figure 4 also shows that there is a relatively large region of qualities that cannot be achieved regardless of the intermediate pressure or the injected mass flow rate. There are two options for modifying the cycle operation to achieve these conditions. Figure 5 shows the results of increasing the subcooling at the condenser exit from 0°C to 5°C, which only noticeably impacts the results at lower injection mass fractions. However, the these lower injection fractions are the more practical range for applications, and thus increasing subcooling is expected to be an important goal for load stand operation. In fact, for the case shown in Figure 5, the limit on injection quality is removed for low injection mass fractions operating at high injection pressures, which is why the two points on the bottom right-hand side of the graph are not shown.

Another option for decreasing the minimum injection quality is to decrease the condensing temperature. However, this approach has the downside of limiting the injection pressures that can be tested. For example, Figure 6 shows the results of the model operating under the same conditions as in Figure 4, but with the condensing temperature lowered from 35°C to 24°C. The graphs are shown with the same scale on the x-axis to emphasize the lower limit on maximum injection pressure with a lower condensing temperature. However, the load stand can now achieve significantly lower minimum qualities at low injection mass fractions.

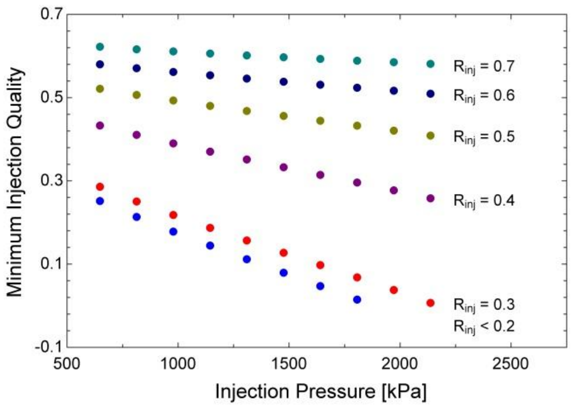


Figure 5: Minimum quality versus injection pressure (640 kPa suction pressure with 11°C superheat, 2290 kPa discharge pressure, and 35°C condensing temperature with 5°C subcooling)

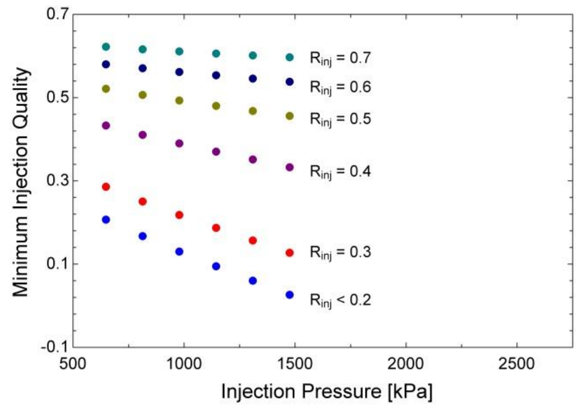


Figure 6: Minimum quality versus injection pressure (640 kPa suction pressure with 11°C superheat, 2290 kPa discharge pressure, and 24°C condensing temperature with 0°C subcooling)

5. CONCLUSIONS AND FUTURE WORK

A model has been developed to predict the performance of a hot gas bypass test stand capable of supplying two-phase refrigerant for injection at two intermediate pressures. Because the model will be used to predict load stand performance both with and without injection, the compressor performance is modeled using a user-specified isentropic efficiency and assuming that the injection process can be treated as an instantaneous, adiabatic mixing process. Model results were presented for a commercially available R-410A compressor that is designed to operate without injection under the assumption that an injection port could be added without changing the compressor's isentropic efficiency. The model predicts the state and mass flow rate of the refrigerant at each point in the system and thus will be useful for setting the valves in the system to achieve the desired operating conditions.

The model also was used to identify the range of injection conditions that can achieved with the load stand and to investigate the impact of different factors on the minimum achievable injection quality. At the lower injection mass flow rates typical of economized systems, the injection quality is generally limited by the enthalpy at the condenser exit. However, it was demonstrated that lower injection qualities can be achieved by increasing the degree of subcooling at the condenser exit or by decreasing the condensing temperature. A second limit on minimum injection quality was observed with large mass flow rates of two-phase refrigerant injection; if the compressor discharge enthalpy falls below the suction enthalpy due to the cooling effect of injection, then the load stand cannot reestablish the suction conditions without an auxiliary heater. Therefore, the model provides a valuable tool for determining which injection conditions can be tested on the load stand. In addition, it will provide insight into how to control the injection flow rates and injection pressures in a very complicated system.

In the future, the load stand model will be validated against experiments performed on compressors both with and without injection. It is hoped that the validated model will reduce the time required to establish steady-state conditions during each test and will provide insight into the settings required to achieve each test condition, such as the necessary condensing temperature. Ultimately, this will enable the load stand to be used for studies on the impact of two-phase refrigerant injection over a wide range of injection pressures and injection qualities, thus contributing to our understanding of economized system performance.

NOMENCLATURE

η_s	isentropic efficiency	(–)
h	enthalpy	(kJ/kg)
$\dot{m}$	mass flow rate	(kg/s)
R_{inj}	injection mass flow ratio	(–)

Subscripts

inj	injection
s	suction

REFERENCES

Baek, C., Lee, E., Kang, H., and Kim, Y. (2008). Experimental study on the heating performance of a CO_2 heat pump with gas injection. In *Proceedings of International Refrigeration and Air Conditioning Conference,* page 872.

Cabello, R., E.Torrella, R.Llopis, and D.Sa´nchez. (2010). Comparative evaluation of the intermediate systems employed in two-stage refrigeration cycles driven by compound compressors. *Energy,* 35:1274--1280.

Cho, H. and Kim, Y. (2000). Experimental study on an inverter-driven scroll compressor with an injection system. In *Proceedings of International Compressor Engineering Conference,* pages 785--790.

Dutta, A. K., Yanagisawa, T., and Fukuta, M. (1996). A study on compression characteristic of wet vapor refrigerant. In *Proceedings of International Compressor Engineering Conference,* page 1112.

Heo, J., Jeong, M. W., and Kim, Y. (2010a). Effects of flash tank vapor injection on the heating performance of an inverter-driven heat pump for cold regions. *Int. J. Refrigeration*, 33:848--855.

Jung, D., Kim, H., and Kim, O. (1999). A study on the performance of multi-stage heat pumps using mixtures. *Int. J. Refrigeration,* 22:402--413.

Liu, F., Huang, H., Ma, Y., and Zhuang, R. (2008). An experimental study on the heat pump water heater system with refrigerant injection. In *Proceedings of International Refrigeration and Air Conditioning Conference,* pages 2211--2216.

Liu, F., Tan, J., and Xiong, J. (2010). Experimental investigation on the two-stage compression heat pump water heater system with refrigerant injection. In *Proceedings of International Refrigeration and Air Conditioning Conference,* pages 12--15.

Liu, Z. and Soedel, W. (1994). An investigation of compressor slugging problems. In *Proceedings of International Compressor Engineering conference,* pages 433--440.

Liu, Z. and Soedel, W. (1995). A mathematical model for simulating liquid and vapor two-phase compression processes and investigating slugging problems in compressors. *HVACR Research,* 1(2):99--109.

Mathison, M. M., Braun, J. E., and Groll, E. A. (2011). Performance limit for economized cycle with continuous refrigerant injection. *Int. J. Refrigeration*, 34(1): 234--242.

Ozaki, K., A.Yabe, Kitazawa, M., and Kobayashi, T. (1992). Evaporation of liquid particles in two-phase compression process. In *Proceedings* of *International Refrigeration and Air Conditioning Conference,* page 288.

Ozaki, K., Endo, N., A.Yabe, and Kobayashi, T. (1990). Basic study on high performance heat pump systems accompanying two-phase compression price. In *Proceedings of International Compressor Engineering Conference,* page 704.