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Extending ORNL HPDM Capabilities for Design and Optimization of New Refrigerant Blends – FY19 1st Quarter (Regular) Milestone Report: Development of Heat Pump System Models (Space Heating) for HFO Blends

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OAK RIDGE NATIONAL LABORATORY
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Extending ORNL HPDM Capabilities for Design and Optimization of New Refrigerant Blends – FY19 1st Quarterly Milestone Report: Development of Heat Pump System Models (Space Heating) for HFO Blends

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Development of Heat Pump System Models (Space Heating) for HFO Blends
(Regular Milestone)

Executive Summary

We recently published numerous heat pump system models at the HPDM website, in the category of "Space Heating". The published system configurations include single-stage heat pumps using TXV for expansion or a short-tube fixed-orifice coupled with a suction line accumulator, a two-stage heat pump system putting two compressors in series, single stage heat pump using two parallel compressors (tandem), vapor injection systems with a vapor injection compressor coupled with an intermediate economizer or a flash tank. These configurations should cover all the residential heat pump products on the market. Additionally, To accelerate the refrigerant transition and make thousands of existing compressor maps reusable by the new refrigerants, ORNL developed a mathematical approach to scale typical vapor injection compressor maps, developed from one baseline refrigerant to be used by the new HFO refrigerants/blends.

Heat pump system models and examples, capable of new refrigerants, released online

At the official website of the latest DOE/ORNL Heat Pump Design Model, https://hpdmflex.ornl.gov/hpdm/wizard/welcome.php, ORNL released numerous heat pump system models for space heating, where they are listed in the category of "Space Heating". The published system configurations include single-stage heat pumps using TXV for expansion or a short-tube fixed-orifice coupled with a suction line accumulator, a two-stage heat pump system putting two compressors in series, single state heat pump using two parallel compressors (tandem), vapor injection systems with a vapor injection compressor coupled with an intermediate economizer or a flash tank. Users can select fin-and-tube coils or microchannel heat exchangers for indoor and outdoor heat exchangers. Within the systems, the compressor models can simulate variable-speed compressors, vapor injection compressors, and tandem (two parallel) compressors. These system configurations cover cover all the residential heat pumps on the market. All the heat exchanger models use segment-to-segment modelling approach, and first-principle-based heat transfer and pressure drop correlations at the segment level, which can be applied for the new HFO refrigerants/blends. The compressor models accept AHRI 10-coefficient compressor maps. The compressor models use an internal scaling approach to convert the maps for other alternative refrigerants, assuming that the compressor has the same volumetric and isentropic efficiencies, as a function of the suction and discharge pressures.

It should be noted that some recent HFO refrigerants/blends cause higher discharge temperature than R-410A and R-22 in heat pump applications, which could burn the compressor lubricant. Manufacturers are more likely to adopt vapor injection compressors to decrease the compressor discharge temperature for the HFO refrigerants/blends. To prepare the equipment design tool, ORNL created a mathematical approach to scale typical vapor injection compressor maps, developed from one baseline refrigerant to be used by the new HFO refrigerants/blends. The approach is introduced in the Appendix.

Schematics of the heat pump system examples are illustrated below.

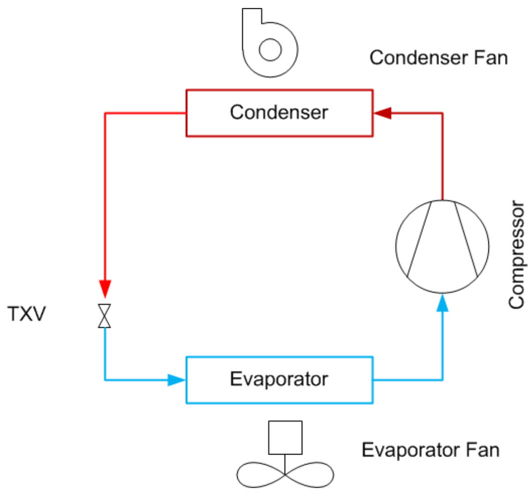

Figure 1: Single-stage heat pump using TXV

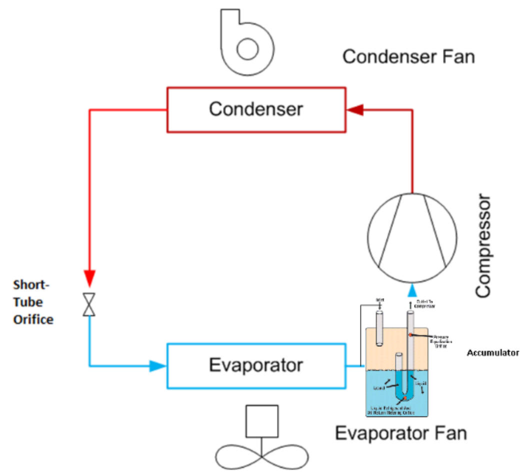

Figure 2: Single-stage heat pump using short-tube orifice and suction line accumulator

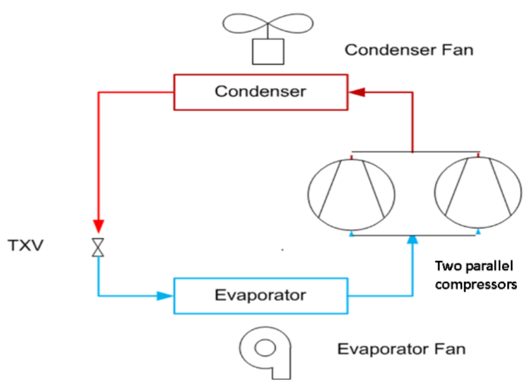

Figure 3: Single-stage heat pump using tandem compressors (two parallel compressors)

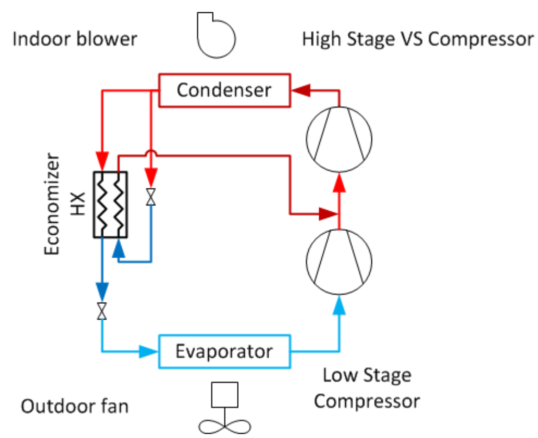

Figure 4: Two-stage heat pump putting two compressors in series and an intermediate economizer

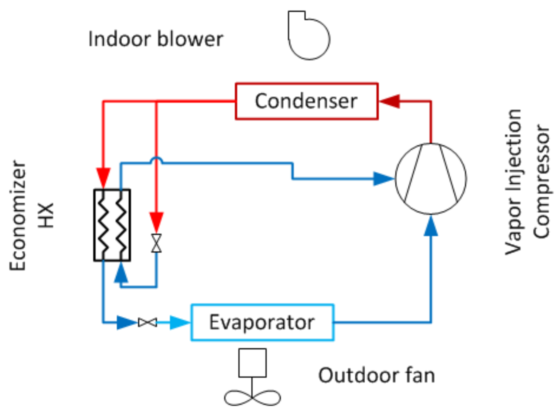

Figure 5: Heat pump using vapor injection compressor and intermediate economizer

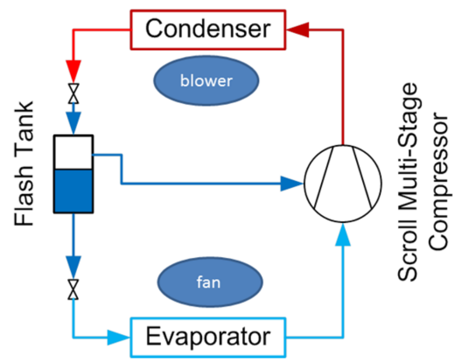

Figure 6: Heat pump using vapor injection compressor and intermediate flash tank

Appendix: A Mathematical Approach to Scale Maps of Vapor Injection Compressors for Alternative HFO Refrigerants/Blends

As shown in Figures 5 and 6, a vapor injection (VI) compressor has two inlet ports, i.e. the suction port connected to an evaporator, and the intermediate injection port connected to an economizer or flash tank, and it has one outlet port discharging vapor to an condenser. As thus, the refrigerant flow rate to the condenser is the sum of the flow rates of the evaporator and the intermediate economizer. To model the VI compressor, it is necessary to correlate the compressor power, evaporator and condenser mass flow rates as functions of the inlet and outlet pressures (saturation temperatures).

Compressor manufacturers provide maps predicting compressor performance values as functions of the compressor suction and discharge saturation temperaures as shown below.

$$X = C_1 + C_2 \cdot (t_S) + C_3 \cdot t_D + C_4 \cdot (t_S^2) + C_5 \cdot (t_S \cdot t_D) + C_6 \cdot (t_D^2) + C_7 \cdot (t_S^3) + C_8 \cdot (t_D \cdot t_S^2) + C_9 \cdot (t_S \cdot t_D^2) + C_{10} \cdot (t_D^3) \tag{1}$$

Where:

C_1 through C_{10} = Regression coefficients provided by the manufacturer
t_D = Discharge dew point temperature, °F, °C
t_S = Suction dew point temperature, °F, °C

X represents a compressor performance value, such as power [W], mass flow rate [lbm/hr], etc.

For VI compressors, manufacturers map the evaporator mass flow rate and compressor power in the same way as single-speed compressors without VI. Instead of of mapping codenser refrigerant flow rate, they provide a map representing the intermediate injection saturation temperature (TIJ) as a function of the suction and discharge saturation temperatures. At the standard condition of the compressor calirometer test to produce the ma[p, the condenser exit subcooling degree, the economizer exit superheat degree and the temperature difference between the economizer exit liquid temperutre and the injection saturation temperature are controlled at standard constants. Therefore, knowing the injeciton saturation temperature and the evaporator mass flow rate at the compressor mapping condition, the condenser refrigerant mass flow rate can be calculated via the energy balance in the economizer using the two equations below:

$$mr_{cond} = mr_{evap} + mr_{inj} \tag{2}$$

$$mr_{cond} * h_{cond,exit} = mr_{evap} * h_{ec,liq} + mr_{inj} * h_{ec,vap} \tag{3}$$

Where mr_{cond}, mr_{evap}, and mr_{inj} are the condenser, evaporator and injection mass flow rates, respectively, and mr_{evap} is given by the map, mr_{cond} and mr_{inj} are two unknowns. $h_{cond,exit}$ is the condenser exit liquid enthalpy, resulted by the discharge pressure and the standard subcooling degree. $h_{ec,liq}$ is the economizer exit liquid enthalpy, calculated by the discharge pressure, and the temperature difference between the exit liquld and the injection saturation temperature. $h_{ec,vap}$ is the economizer exit vapor enthalpy, led by the injection pressure and the standard economizer exit superheat degree. Using the two equations of (2) and (3), the two unknowns of mr_{cond} and mr_{inj} can be calculated.

VI compressor manufaturers and customers reported that the evaporator mass flow rate and compressor power maps are credable in a wide range of operation conditions. However, the map of injection saturation temperature is problematic. In real products using VI compressors, they don't necessarily maintain the same condenser and economizer exit liquid temperatures and exit superheat degree. Particularly, the injection saturation temperature are impacted by the upstream throttling and condenser subcooling degree in an actual system. The actual injection saturation temperature can be quite different from that predicted by the map. Consequently, the predicted condenser mass flow rate is not accurate if the acutal operation condtions are different from the standard conditions during development of the map.

The recent HFO refrigerants/blends, as replacements of R-22 and R-410A tend to cause higher compressor discharge temperatures, which could burn the compressor oil and impact the product reliability. For the coming refrigerant transition, manfaturers will more likely use VI compressors. All the VI compressor maps are empirical, and only useful for the original refrigerant. To accelarate the transition, we need to develop a method to make the thousands of existing VI compressor maps directly reuseable by the new refrigerants.

A new method should be developed to tackle two chanllenges: 1) Improve the condenser mass flow rate prediction accuracy when the actual injection saturation temperature is away from the mapped condtion; 2) Scale the VI compressor map from a baseline refrigerant to be used by an alternative new HFO refrigerants/blend.

In essense, the VI compression is a two-stage compression process, which has two separate chambers, low stage and high stage, both uses a common shaft rotating at one speed. Equations are given below to describe the VI two-stage compression process.

$$mr_{evap} = Volume_{disp,l} \times Speed_{rotation} \times Density_{suction} \times \eta_{vol,l} \tag{4}$$
$$mr_{cond} = Volume_{disp,h} \times Speed_{rotation} \times Density_{mixed} \times \eta_{vol,h} \tag{5}$$
$$Power_l = mr_{evap} \times (\mathrm{h}_{s,l} - h_{suction})/\eta_{isentropic,l} \tag{6}$$
$$Power_h = mr_{cond} \times (h_{s,h} - h_{mixed})/\eta_{isentropic,h} \tag{7}$$
$$Power = Power_l + Power_h \tag{8}$$

Considering energy conservation without heat loss to the enviroment:

$$h_{mixed} = (mr_{evap} * h_{dis,l} + mr_{inj} * h_{ec,vap})/mr_{cond} \tag{9}$$
$$h_{dis,l} = (mr_{evap} * h_{suction} + Power_l)/mr_{evap} \tag{10}$$
$$h_{dis,h} = (mr_{cond} * h_{mixed} + Power_h)/mr_{cond} \tag{11}$$

Where $Volume_{disp,l}$ and $Volume_{disp,h}$ are the given displacement volumes of the low and high stages. $Speed_{rotation}$ is the shaft rotational speed. $Density_{suction}$ is the suction vapor density. $Density_{mixed}$ is the vapor density of vapor at the injeciton port mixed with the vapor exiting the low stage. $\eta_{vol,l}$ and $\eta_{vol,h}$ are volumetric efficiencies of the low stage and high stage, respectively. $h_{suction}$ is the suction vapor enthalpy. $\mathrm{h}_{s,l}$ is the enthalpy determined by the suction entropy and injection pressure. h_{mixed} is the mixed vapor enthalpy. $h_{s,h}$ is the enthalpy dtermined by the mixed vapor entropy and discharge pressure. $\eta_{isentropic,l}$ and $\eta_{isentropic,h}$ are the isentropic efficiencies of the low stage and high stage. $Power_l$ and $Power_h$ are the powers consumed by the low and high stages, respectively. $Power$ is the total compressor power, predicted by the VI compressor map. $h_{dis,l}$ is the discharge vapor enthalpy out of the low stage, $h_{dis,h}$ is the discharge vapor enthalpy at the compressor exit, i.e. high stage. At the standard map condition, the compressor maps as functions of the suction and discharge saturation temperatures present mr_{evap}, mr_{cond}, $Power$, and injection saturation temperature, and the resultant enthalpy and entropy at the suction, injection and discharge pressures. One more constrait is needed to balance the equations and unknowns, for which assumes $\eta_{isentropic,l} = \eta_{isentropic,h}$. By this way, the four efficiency values of $\eta_{vol,l}$, $\eta_{vol,h}$, $\eta_{isentropic,l}$ and $\eta_{isentropic,h}$ can be reduced from the compressor maps at the standard map condition.

In the case that the actual injection pressure differed from the injection pressure predicted by the map at the suction and discharge pressures, it is assumed that the volumetric and isentropic efficienies of the low stage are only impacted by its suction and actual injection pressure. In one flow direction, the low stage can only see its inlet and outlet. The required discharge pressure, i.e. "mapped" discharge pressure leading to the actual injection pressure, can be back-calculated using the compressor map to match the actual injection pressure. Using the actual suction, injection and the "mapped" discharge pressures, the efficiencies of $\eta_{vol,l}$ and $\eta_{isentropic,l}$ at the low stage can be determined. Additionally, a high stage volmetric efficiency, $\eta_{vol,h,1}$, is calculated at the mapped discharge pressure.

It is assumed that the difference in the injection pressure only impacts the compression process in the high stage. The difference in mixing, over or under-compression due to the varied injection pressure all occur in the high stage chamber. Using the actual injection pressure and discharge pressure, the required suction pressure, i.e. "mapped" suction pressure leading to the actual injection pressure, can be back-calculated to match the actual injection pressure. Another high stage volmetric efficiency of $\eta_{vol,h,2}$ is calculated at the mapped suction pressure. The average of $\eta_{vol,h,1}$ and $\eta_{vol,h,2}$, i.e. $\eta_{vol,h} = (\eta_{vol,h,1} + \eta_{vol,h,2})/2$, is adopted as the volumetric efficiency of the high stage, to account for the impact due to the injection pressure differed from the map condition.

It reported that the VI compresosr power prediction is not impacted by the injection pressure variation. With the low stage isentropic efficiency of $\eta_{isentropic,l}$ calculated at the actual suction, injection and the "mapped" discharge pressures, the high stage isentropic efficiency of $\eta_{isentropic,h}$ can be calculated from the compressor power at the actual suction and discharge pressures.

To scale the compressor map from a baseline refrigerant to be used by an alternative HFO refrigerant/blend. It is assumed that refrigerants have similar working pressures and densities, i.e. the drop-in replacements using the HFO refrigerants/blends, the efficiency values of $\eta_{vol,l}$, $\eta_{vol,h}$, $\eta_{isentropic,l}$ and $\eta_{isentropic,h}$ hold constant at the same suction, injection and discharge pressures. Based on the assumption, for a alternative refrigerant, with the suction and discharge pressures known, using properties and compressor maps of a baseline refrigerant, e.g. R-22 or R-410A, equations from (2) to (11), the $\eta_{vol,l}$, $\eta_{vol,h}$, $\eta_{isentropic,l}$ and $\eta_{isentropic,h}$ can be calculated for the baseline refrigerant. Next, keeping the same efficiency values, and using properties of the alternative refrigerant, the compressor power, evaporator and condenser mass flow rates can be predicted.