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## Properties of Refrigerant Affect Compressor Design

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### ABSTRACT

The paper examines selected thermodynamic properties of commonly used refrigerants and how they may affect design of a compressor. Among those properties are volumetric capacity, system pressure difference, system compression ratio, isentropic coefficient of performance, gas density, temperature of discharge gas, velocity of sound etc. The is made on the scale of evaporating temperatures from  $-40\text{ }^{\circ}\text{C}$  to  $30\text{ }^{\circ}\text{C}$ , and condensing temperature  $40.5\text{ }^{\circ}\text{C}$ . The temperature of gas entering suction port is assumed constant and equal to  $32.2\text{ }^{\circ}\text{C}$ .

### 1. INTRODUCTION

The usual procedure of verification that an existing compressor can satisfactorily run with a new (or an alternative) refrigerant is lengthy and expensive testing. Such tests are performed on the calorimeter, in the sound lab, and on a life-test rig, etc. Nevertheless, by using thermodynamic properties of a refrigerant one can predict, or at least to estimate, how an existing compressor will perform with a new refrigerant. Better yet, one can use thermodynamic properties of a refrigerant to design new compressor from the scratch. The NIST Standard Reference Database 32 better known as REFPROP is an excellent tool for such a comparison.

### 2. VOLUMETRIC CAPACITY

The volumetric capacity of a refrigerant determines necessary displaced volume per one revolution of a positive displacement compressor. The thermal power, in kilo-Joule per hour a compressor has to deliver can be expressed as

$$Q = \rho \cdot V \cdot \Delta h \cdot 60 \cdot n \quad (1)$$

Where

$Q$  is required thermal power [kJ/hr]

$\rho$  is density of gas [ $\text{kg}/\text{m}^3$ ]

$V$  is volume displaced during one revolution [ $\text{m}^3/\text{rev}$ ]

$\Delta h$  is change in enthalpy (either in the evaporator or in the condenser; see Fig. 1) [kJ/kg]

$60$  is constant [min/hr]

$n$  is number of revolutions per minute [rev/min]

By rearranging variables in equation (1) into two groups, we get

$$Q_{\text{VOL}} = \rho \cdot \Delta h \quad (2)$$

$$V_{\text{HR}} = V \cdot 60 \cdot n \quad (3)$$

Where

$Q_{\text{VOL}}$  is volumetric capacity [kJ/ $\text{m}^3$ ]

$V_{\text{HR}}$  is volume displaced per hour [ $\text{m}^3/\text{hr}$ ]

Now, the required thermal power can be expressed as

$$Q = Q_{VOL} \cdot V_{HR} \tag{4}$$

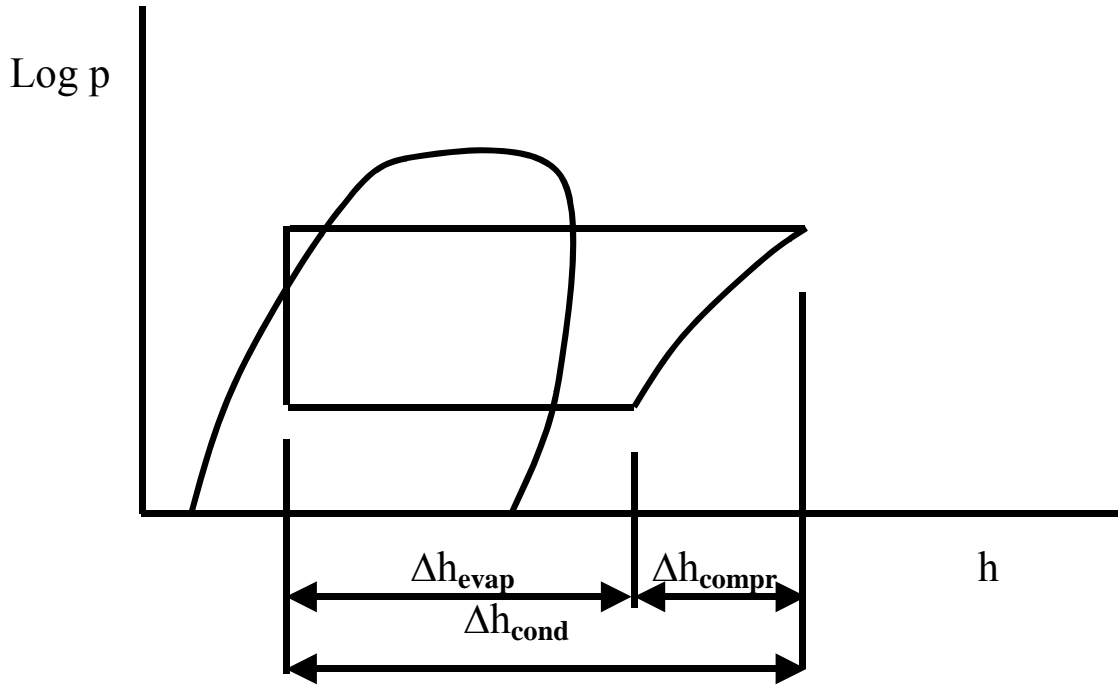


Fig. 1: Pressure – Enthalpy Diagram

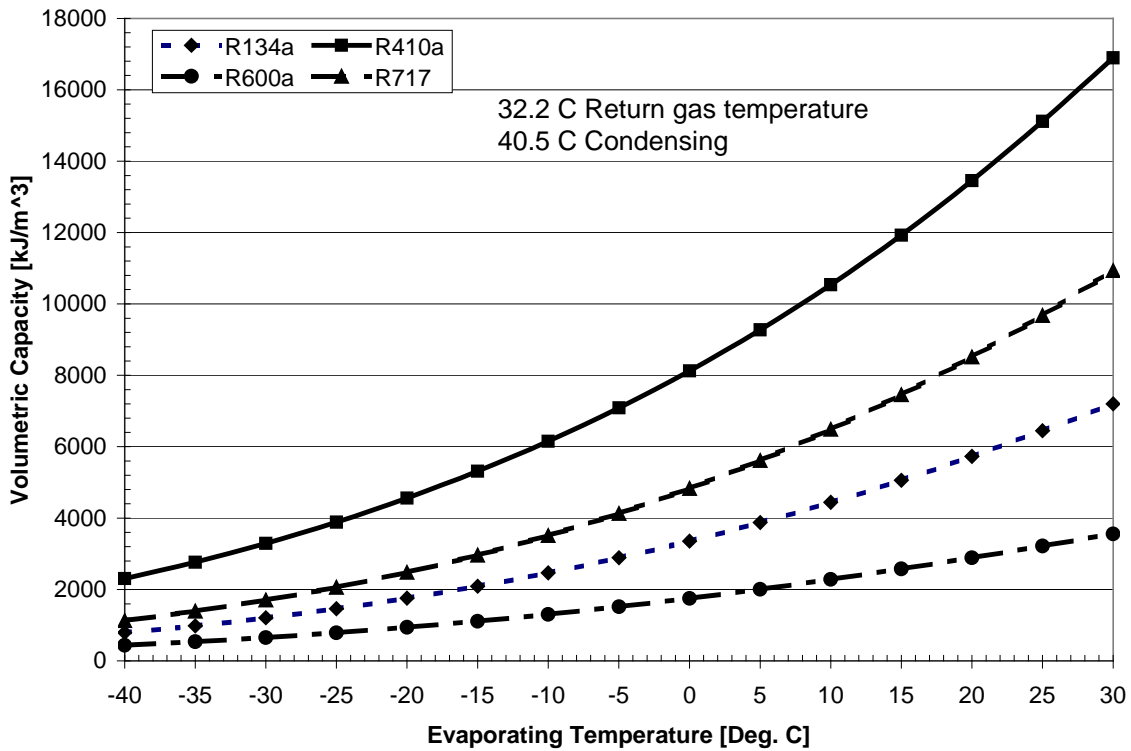


Fig. 2: Volumetric Capacity of R134a, R410a, R600a, R717

From equation (4), it is evident that the higher is the volumetric capacity  $Q_{VOL}$  the more thermal power we can get from the compressor with the displaced volume  $V_{HR}$ . In the other words, the compressor that uses refrigerant with low volumetric capacity has to be bigger. A bigger compressor is also more expensive compressor. Fig. 2. shows comparison of volumetric capacities of four refrigerants: R717 (ammonia), R134a, R410a and R600a (isobutene).

We can see in the Fig. 2 that the compressor running with isobutene (R600a) has to have approximately four times bigger displaced volume than the one running with refrigerant R410a.

### 3. CARNOT HEAT MULTIPLICATION FACTOR AND ISENTROPIC COP

The other important property of a compressor is its isentropic coefficient of performance, COP. The estimate of isentropic COP can be derived from the Fig. 1. It is equal to the change in enthalpy in the evaporator and in the condenser respectively divided by the change in enthalpy in the compressor. Obviously, numerical values of COP for cooling or refrigeration, and for heating are different. Those are

$$COP_{cool} = \frac{\Delta h_{evap}}{\Delta h_{compr}}, \quad \text{and} \quad COP_{heat} = \frac{\Delta h_{cond}}{\Delta h_{compr}} \quad (5)$$

Where

$COP_{cool}$  is isentropic coefficient of cooling (or refrigeration)

$COP_{heat}$  is isentropic coefficient of performance of heating

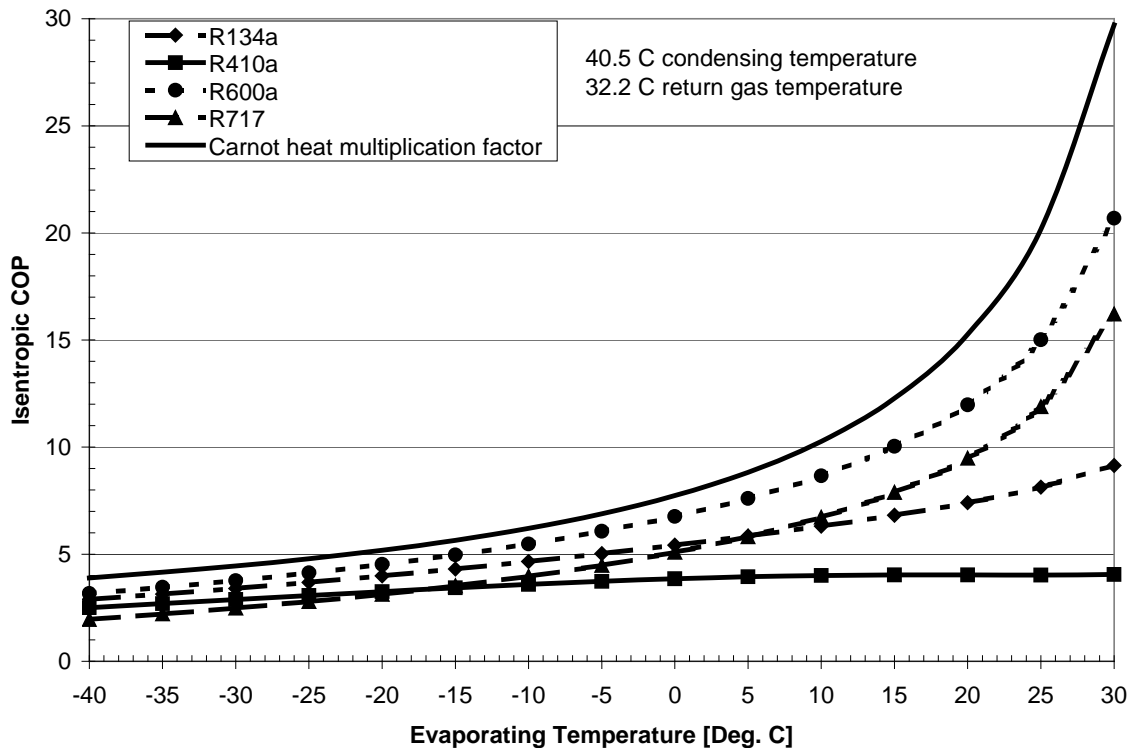


Fig. 3: Isentropic Coefficient of Performance

Carnot heat multiplication factor is actually a reciprocal of thermal efficiency of Carnot cycle. It is equal to

$$\text{CHMF} = \frac{T_{\text{cond}}}{T_{\text{cond}} - T_{\text{evap}}} \quad (6)$$

Where

CHMF is Carnot heat multiplication factor [ ]

$T_{\text{cond}}$  is condensing temperature [ $^{\circ}\text{K}$ ]

$T_{\text{evap}}$  is evaporating temperature [ $^{\circ}\text{K}$ ]

Carnot heat multiplication factor is theoretical maximum of isentropic COP, and it does not depend on the substance that undergoes changes in temperature. It is equal to 1 when evaporating temperature is absolute zero ( $T_{\text{evap}} = 0$ ), and it reaches infinity when temperatures of condensation and evaporation are equal ( $T_{\text{cond}} = T_{\text{evap}}$ ). The value of isentropic COP can also reach infinity if in the equation (5)  $\Delta h_{\text{compr}} \rightarrow 0$ . In the Fig. 3, we can see that COP of ammonia and isobutene follow the trend of CHMF while COP of R410a is almost flat for evaporating temperatures from  $-10^{\circ}\text{C}$  to  $30^{\circ}\text{C}$ .

#### 4. SYSTEM PRESSURE DIFFERENTIAL

The difference between condensing pressure and evaporating pressure is an important design parameter. This pressure differential generates forces acting on the crankshaft, and therefore affecting function of hydrodynamic bearings, shaft diameter etc. Fig. 4 shows system pressure differential of selected four refrigerants. Thus the compressor running with R410a refrigerant will require more robust parts, and therefore may become more expensive to manufacture than the compressor running with R600a refrigerant.

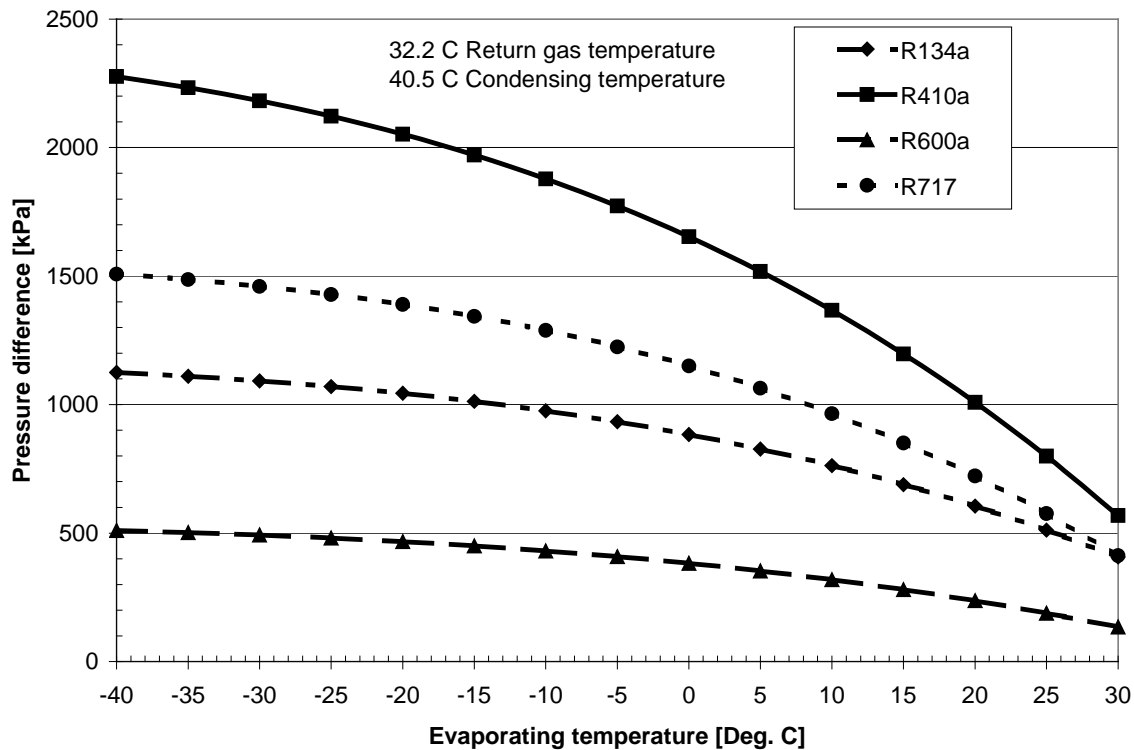


Fig. 4: System Pressure Differential

## 5. SYSTEM COMPRESSION RATIO

The compression ratio of a system is another important design parameter. It is commonly known that the volumetric efficiency of a compressor drops rather rapidly when the compressor runs with compression ratio above five. In the Fig.5 we can see that the refrigerant R410a has system compression ratio of five at the evaporating temperature  $-15^{\circ}\text{C}$ , while the refrigerant R134a has system compression ratio above five at evaporating temperature of  $-5^{\circ}\text{C}$ .

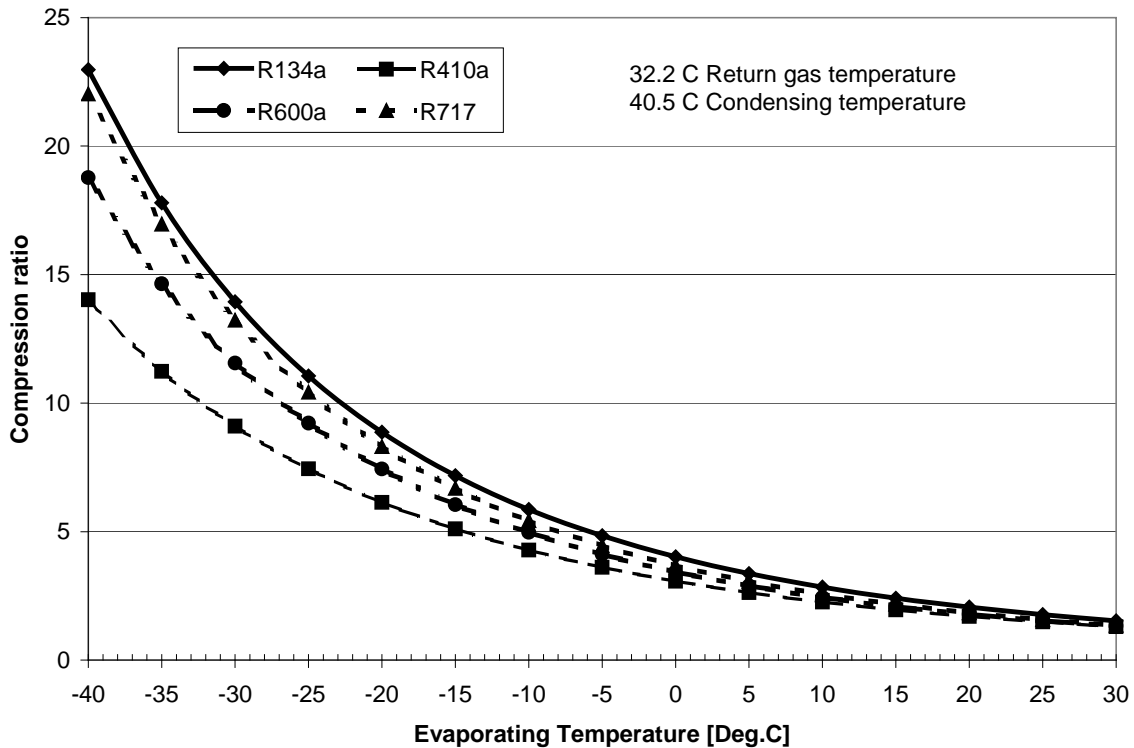


Fig. 5: System Compression Ratio

## 6. GAS DENSITY

The density of refrigerant gas plays important role in the design of self-acting (automatic) valves. The motion of a valve may have three phases. In the beginning of opening as well as before the closing, the flow of gas through the valve is controlled by the valve lift  $h$ . In this case, the valve lift satisfies the condition

$$h \leq \frac{d}{4} \quad (7)$$

If the lift of the valve  $h$  exceeds  $d/4$ , the flow through the valve is controlled by the port diameter  $d$ . In the third case when the pressure ratio across the valve exceeds its critical value, the flow of gas is choked, and the velocity of flow is constant. Generally, we may have to deal with two different velocities of flow, flow through the valve opening, and the flow through the valve-port. These velocities are not the same. In order to estimate these velocities, we can use an expression for the velocity of flow of an ideal gas

$$w = \sqrt{2 \cdot \frac{k}{k-1} \cdot \frac{p_i}{\rho_i} \cdot \left[ 1 - \left( \frac{p_o}{p_i} \right)^{\frac{k-1}{k}} \right]} + w_i^2 \quad (8)$$

Where

- w is velocity of gas at the outlet [m/s]
- k is adiabatic exponent [ ]
- p<sub>i</sub> is pressure at the inlet [Pa]
- ρ<sub>i</sub> is density of gas at the inlet [kg/m<sup>3</sup>]
- p<sub>o</sub> is pressure at the outlet [Pa]
- w<sub>i</sub> is velocity of gas at the inlet [m/s]

If the flow at the outlet cross-section reaches critical values, pressure p<sub>o</sub> = p<sub>k</sub> and density of gas ρ<sub>o</sub> = ρ<sub>k</sub>, the outlet velocity will become independent of the pressure difference p<sub>i</sub> - p<sub>o</sub>, and its vale will be constant and equal to

$$w = w_k = \sqrt{2 \cdot \frac{k}{k-1} \cdot \frac{p_k}{\rho_k}} = c \tag{9}$$

Where

- w<sub>k</sub> is critical velocity [m/s]
- p<sub>k</sub> is critical pressure [Pa]
- ρ<sub>k</sub> is critical density [kg/m<sup>3</sup>]
- c is velocity of sound [m/s]

The total force that acts on the valve consists of three effects. It comes from the pressure difference p<sub>i</sub> - p<sub>o</sub>, and it is proportional to

$$F_p \propto A_p \cdot (p_i - p_o) \tag{10}$$

Where

- F<sub>p</sub> is force due to pressure difference [N]
- A<sub>p</sub> is effective pressure area [m<sup>2</sup>]

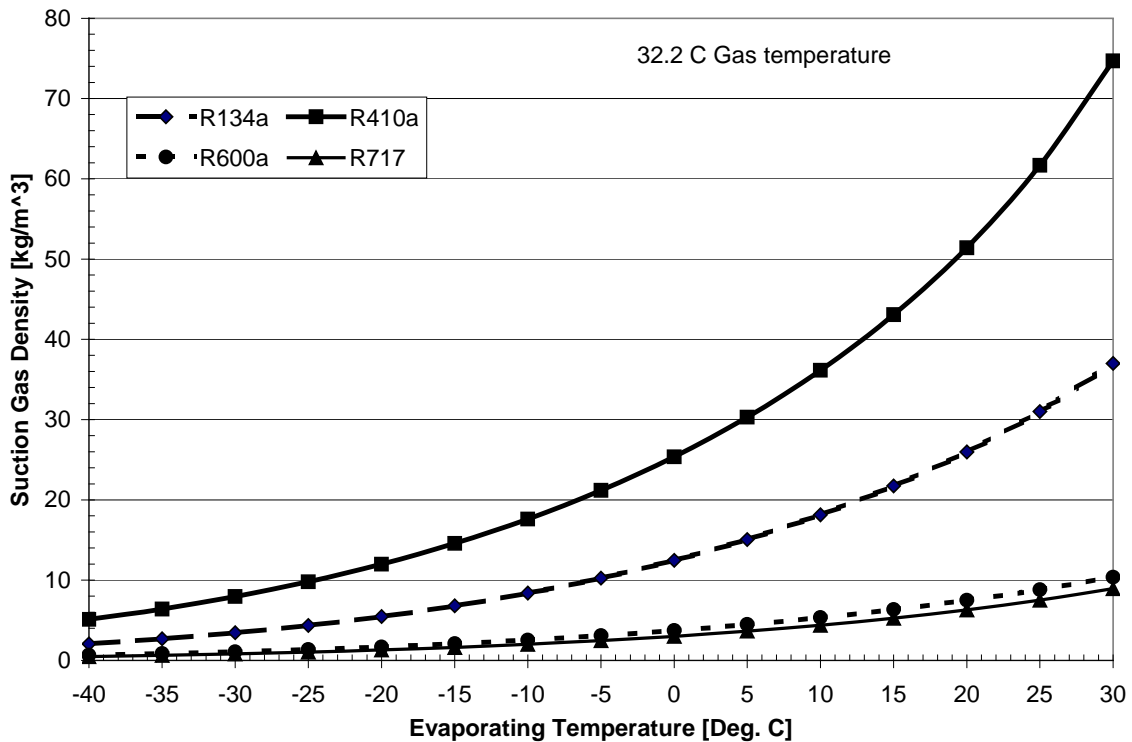


Fig. 6: Suction Gas Density

From the stagnation pressure

$$F_s \propto A_s \cdot \frac{1}{2} \cdot \rho \cdot w^2 \quad (11)$$

Where

$F_s$  is force due to stagnation pressure [N]  
 $A_s$  is effective pressure area exposed to stagnation pressure [m<sup>2</sup>]

And the force due to change in momentum of flowing gas

$$F_m \propto A_m \cdot \rho \cdot w^2 \quad (12)$$

Where

$F_m$  is force due to change in momentum (direction of flow) [N]  
 $A_m$  is effective momentum area [m<sup>2</sup>]

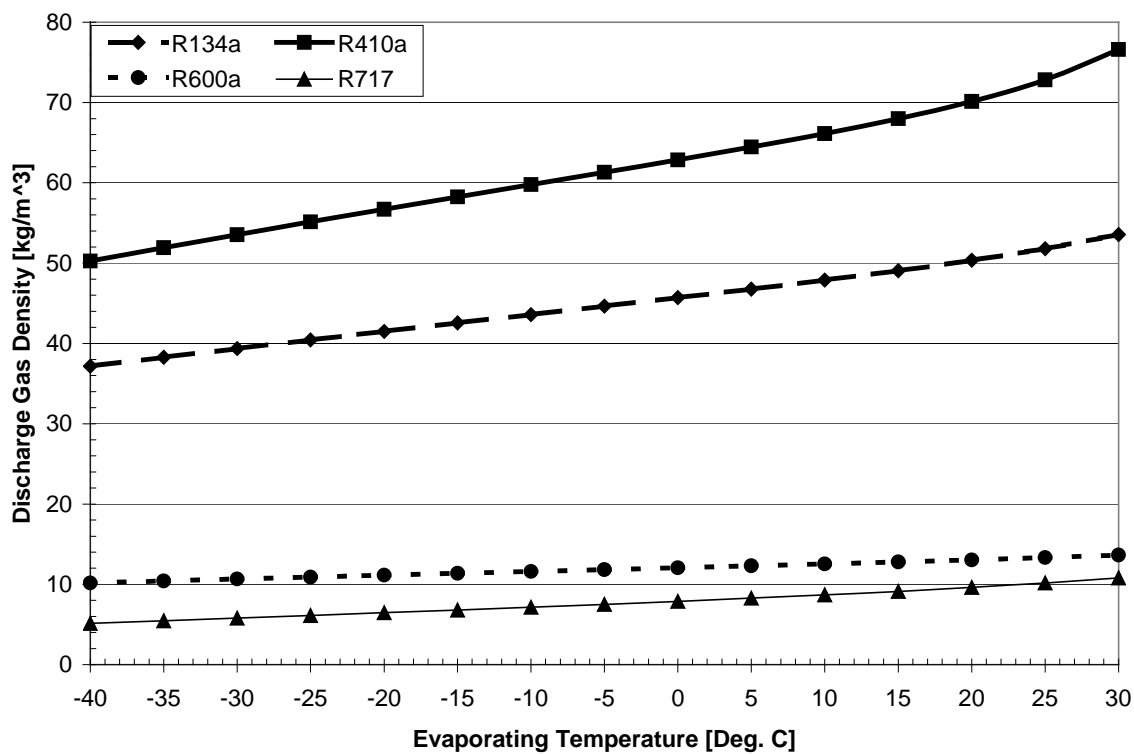


Fig. 7: Discharge Gas Density

Each affective force area ( $A_p$ ,  $A_s$  and  $A_m$ ) may be non-linear function of the lift of the valve. Further details can be found in (Bukac, 2002). Fig. 6 and Fig. 7 show density of suction and discharge gas respectively.

The assumption of ideal gas may be misleading. If we substitute the square of the velocity of gas flow from equation (8) into equations (11) and (12) we could conclude that the forces acting on the valve are independent from the density of gas, which as the experience shows is not true.

## 7. DISCHARGE GAS TEMPERATURE

Every lubricant that is used in the refrigerating or air-conditioning systems has maximum allowable temperature beyond which it either brakes or even burns and creates hard deposits. Therefore, the temperature of discharge gas is freon type refrigerants have reasonably rising discharge gas temperature with respect to ammonia.



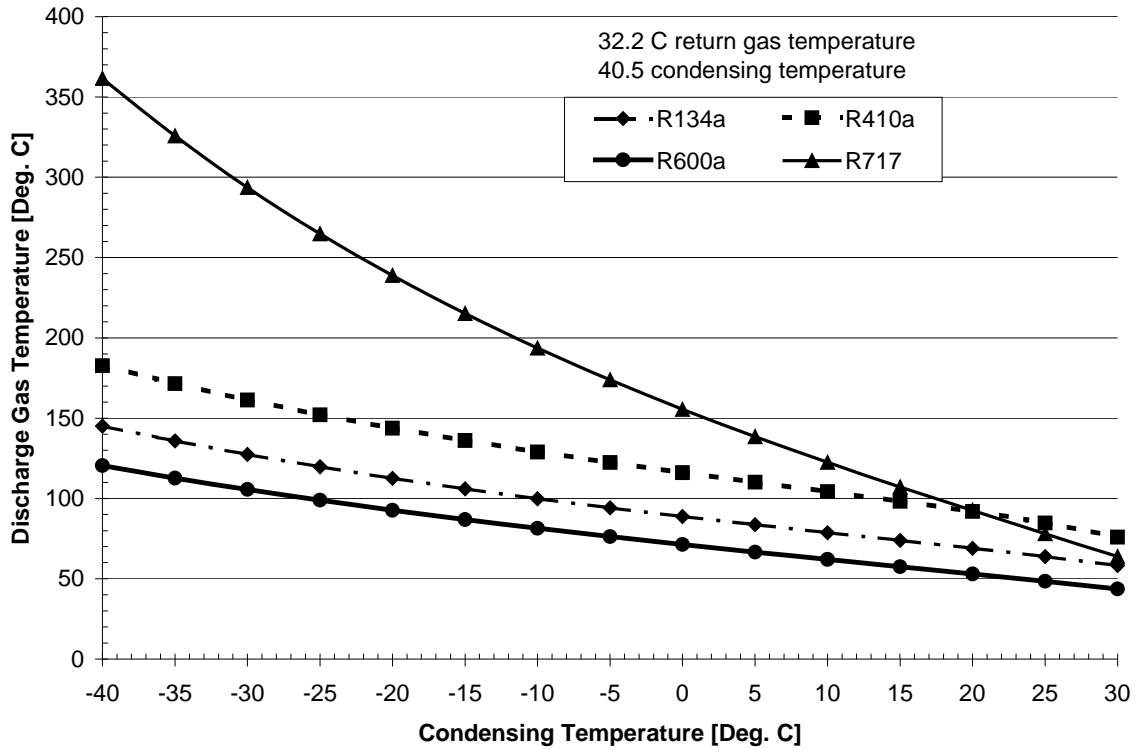


Fig. 8: Discharge Gas Temperature

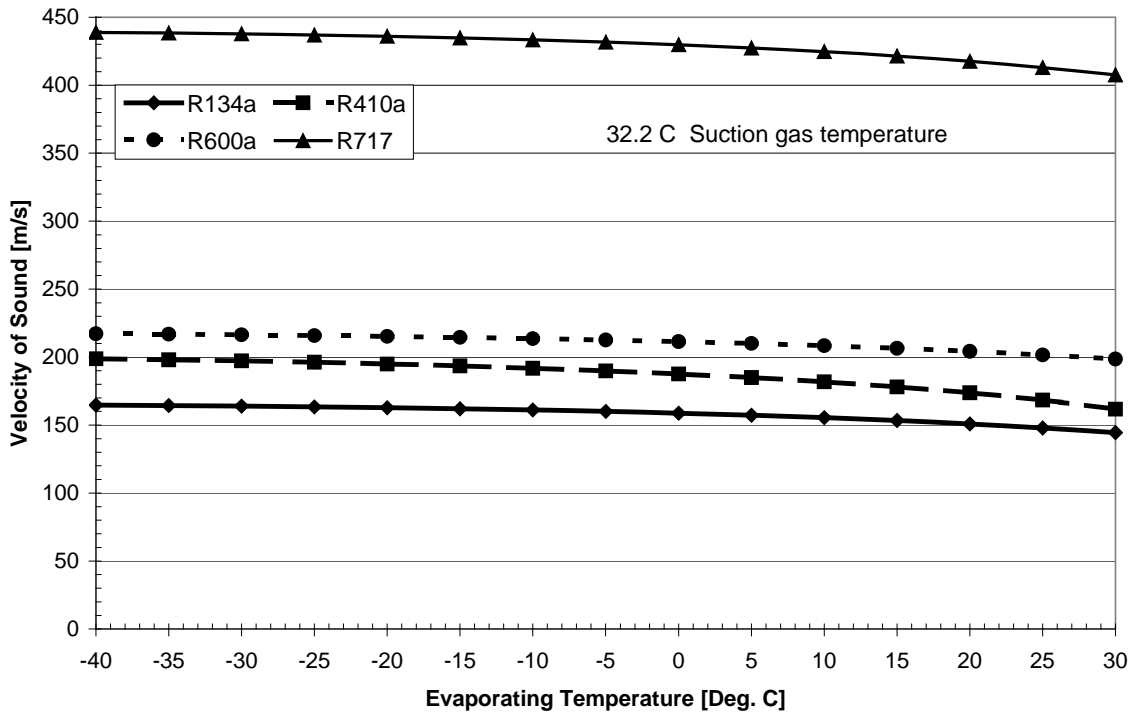


Fig. 9: Velocity of Sound, Suction Side

## 8. VELOCITY OF SOUND

The knowledge of velocity of sound is important for design of suction and discharge mufflers. For example, in order to attenuate certain frequency  $f$  in the suction or discharge side a quarter-wave resonator or an expansion chamber has to have length (Bukac, 2006)

$$L = \frac{4 \cdot c}{f} \quad (12)$$

Where

L is the length [m]  
 C is velocity of sound [m/s]  
 f is frequency [Hz]

Fig. 9 shows velocity of sound on the suction side, and Fig. 10 shows velocity of sound on discharge side. We can see, the mufflers that are designed for ammonia compressors could not be used in freon type ones. The freon-type refrigerants have only small difference in the velocity of sound on suction and discharge side of compressor. Thus, suction mufflers will need a little, if any, adjustment in their dimensions to attenuate required range of frequencies.

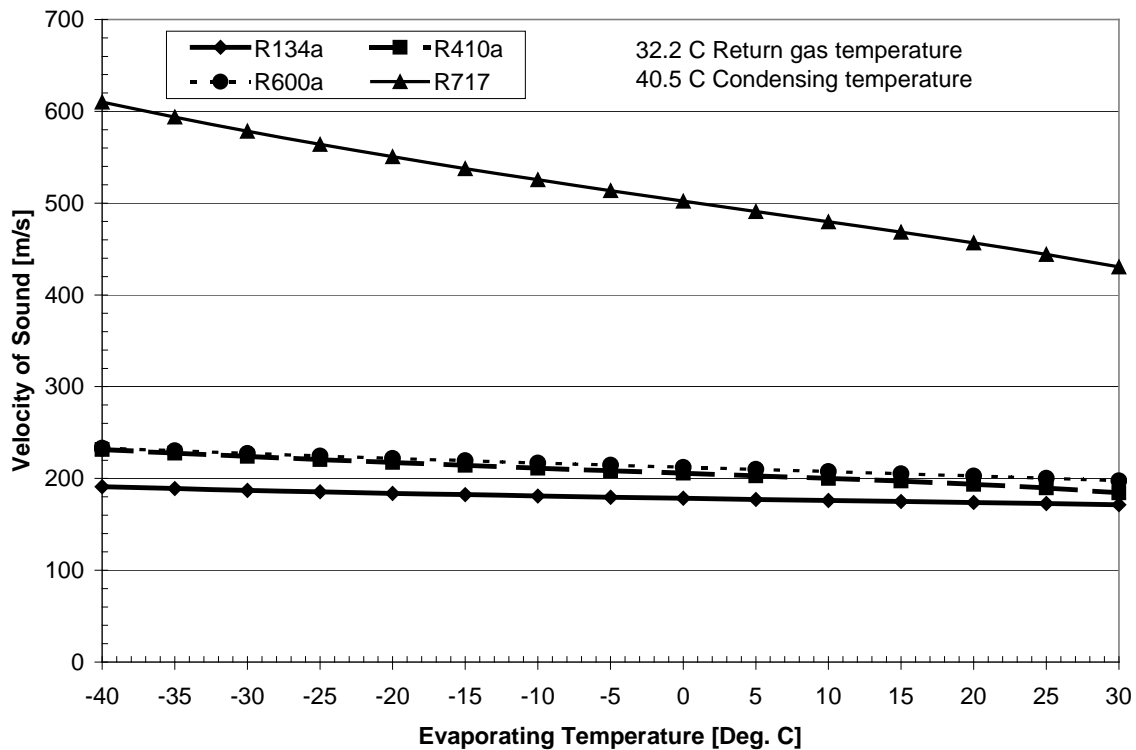


Fig. 10: Velocity of Sound, Discharge Side

## 9. SUCTION PRESSURE

Fig. 11 shows suction (evaporating) pressure that corresponds to evaporating temperature. We can see the highest suction pressure has refrigerant R410a and the lowest one the refrigerant R600a.

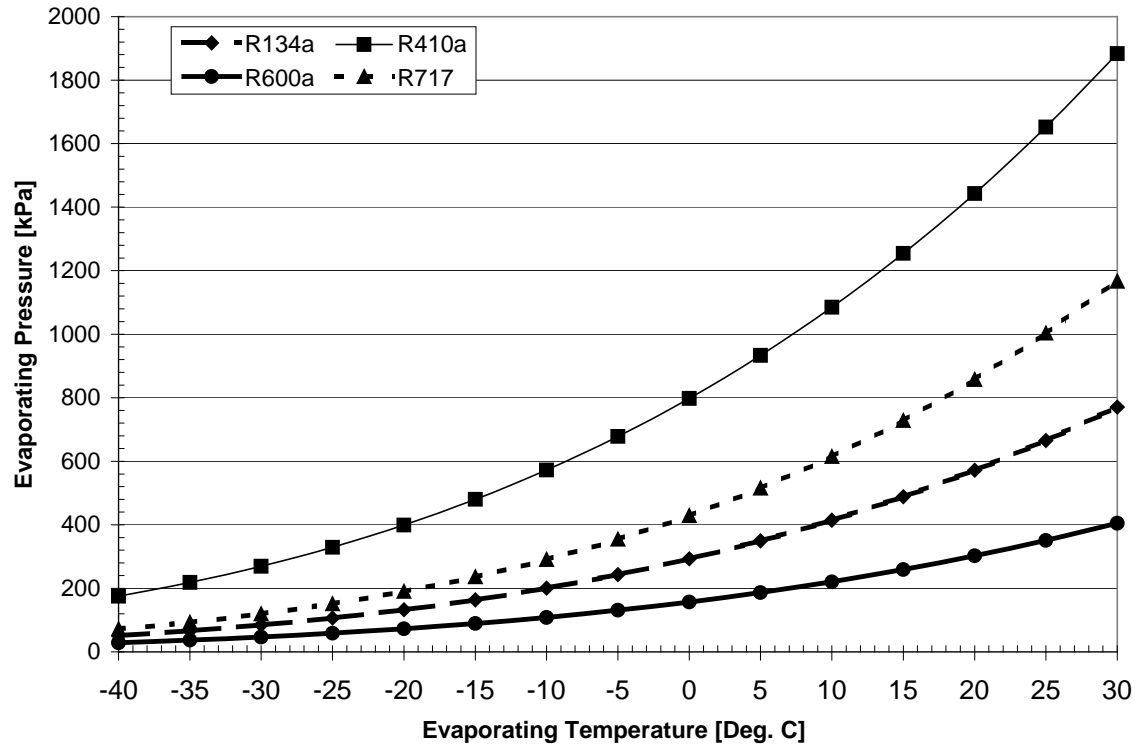


Fig. 11: Evaporating Pressure

## 10. CONCLUSION

A brief comparison of properties of four selected refrigerants shows how the properties of the refrigerant affect compressor design. The comparison shows that the refrigerant R410a has high volumetric capacity, high system pressure difference, and low isentropic coefficient of performance. While the refrigerant R600a has smallest volumetric capacity, low system pressure difference, and high isentropic coefficient of performance. It is always beneficial to know thermodynamic properties of refrigerant before any decision of its application are made.

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