1988

# Performance and Gas Pulsations When Pumping Different Cases with the Same Compressor

J. Kim
*Purdue University*

W. Soedel
*Purdue University*

Kim, J. and Soedel, W., "Performance and Gas Pulsations When Pumping Different Cases with the Same Compressor" (1988). *International Compressor Engineering Conference.* Paper 625.
https://docs.lib.purdue.edu/icec/625

# PERFORMANCE AND GAS PULSATIONS WHEN PUMPING DIFFERENT GASES WITH THE SAME COMPRESSOR

by

J. Kim and W. Soedel
Ray W. Herrick Laboratories
School of Mechanical Engineering
Purdue University

## SUMMARY

It is illustrated what happens when a research compressor of fractional horsepower size is used to pump two types of refrigerant, R-12 and R-22, and also air and helium. A computer simulation program is used with identical suction conditions and identical discharge pressure conditions to investigate changes caused by switching gases. Comparisons of cylinder pressure, suction and discharge pressure pulsations and valve motions were made.

## INTRODUCTION

An issue that surfaces regularly is the question if a compressor that is designed for one gas can be utilized to pump another gas without changes in basic design. In the following, it is illustrated what happens when a research compressor of fractional horsepower size is used to pump two types of refrigerant, R-12 and R-22, and also air and helium. The demonstration is a purely theoretical one. A computer program is used that is able to handle a one cylinder compressor with flexible reed valves and suction and discharge gas pulsation. Identical suction conditions in terms of pressure and temperature are assumed and identical discharge pressure conditions. Since the compressor is running at 3600 rpm, compression and expansion is taken as adiabatic for all cases. Changes that were observed are explained by investigating approximate scaling laws in terms of specific volume.

It should be noted that the problem of lubrication when different gases are used is not addressed here.

## SIMULATION

The simulation model is in its concepts based on references [1, 2, 3], but contains significant extensions that allow for more accurate results. However, the purpose of this paper is not to discuss the model, but to focus on changes caused by switching gases. The input data is summarized in Table 1. Calculated specific volumes at suction and discharge conditions are given in Table 2. Note that the discharge temperatures are different because of the adiabatic assumption. The high temperatures for air and helium indicate immediately that cooling would have to be improved in a practical application. Figs. 1 - 3 show the comparisons between R-12 and R-22 in terms of cylinder pressure, and suction and discharge pressure pulsations. Figs. 4 - 6 show the same comparisons between air and helium. While the differences between R-12 and R-22 are not dramatic, the differences to air and helium are.

Volume flow transfer functions of the discharge and cavity systems are shown in Figs. 7 and 8. Table 3 shows a comparison of the thermodynamic efficiencies and a coefficient of performance, $C_p$, which is defined as mass flow rate divided by input power. [$lb_m$/Kw·hr].

## SCALING

If we examine some key equations which are influenced by gas properties, we can explain much of the observed behavior. They are

1. Polytropic process of the cylinder

$$\frac{P_c(t)}{P_o} = \left[\frac{m_c(t)}{\rho_o V(t)}\right]^k = \left[\frac{\rho(t)}{\rho_o}\right]^k \tag{1}$$

where $m_c$ and $\rho_o$ are cylinder mass and density of the gas. Subscript o means "at initial condition," and k is the adiabatic constant.

2. Mass flow through valve port:

$$\dot{m}_v = P_u A_v \sqrt{\frac{2kg_c}{(k-1)RT_u}} \sqrt{\left(\frac{P_d}{P_u}\right)^{2/k} - \left(\frac{P_d}{P_u}\right)^{k+1/k}} \tag{2}$$

where, $P_u$ and $P_d$ are upstream and downstream pressures of the valve, R is the gas constant, $T_u$ is the upstream temperature.

3. Pressure response of a cavity connected to an anechoic pipe:

$$\frac{P_{in}}{Q_{in}} = \frac{\left[\dfrac{\rho_o C_o}{S_{an}}\right]}{1 + \dfrac{j\omega V_o}{S_{an} C_o}} \tag{3}$$

where, $S_{an}$ is cross sectional area of the anechoic pipe, $C_o$ is speed of sound, $V_o$ is volume of the cavity, and $\omega$ is frequency of the gas pulsation. It should be noted that equation (3) is for a very simple gas manifold for the purpose of qualitative interpretation. The simulated model in this paper has a more complicated gas model including expansion chambers.

4. Speed of sound:

$$C_o = \sqrt{kg_c RT} \tag{4}$$

Differentiating both sides of equation (1) and by rearranging it,

$$\frac{\dot{P}_c(t)}{\dot{\rho}(t)} = \left(\frac{k}{\rho_o}\right) P_o \cdot \left[\frac{\rho(t)}{\rho_o}\right]^{k-1} \tag{5}$$

$\dfrac{\dot{P}_c(t)}{\dot{\rho}(t)}$ can be interpreted as the sensitivity of the cylinder pressure change to the density change of the cylinder gas. The sensitivity is approximately proportional to $\dfrac{1}{\rho_o} = v$, specific volume of the gas, because k and $\dfrac{\rho(t)}{\rho_o}$ are not as different from gas to gas as is v. It tells us that helium has the largest sensitivity by far. For example, the cylinder pressure of the helium compressor adjusts to changing mass density much faster than R-12 or R-12, if we compare Figure 1 and Figure 4.

Examining equation (2), it can be seen that approximately

$$\dot{m}_v \propto \frac{1}{\sqrt{R}} \tag{6}$$

Since $(m_c)_o \propto \dfrac{1}{R}$

$$\left|\frac{\dot{m}_v}{(m_c)_o}\right| \propto \sqrt{R} \propto \sqrt{v} \tag{7}$$

Therefore, the relative valve mass flow is larger for a larger gas constant, which means a larger specific volume under identical pressure conditions. This means the valve port area can be smaller for helium than for R-12, for example.

5. From equation (4), it is approximately true that

$$C_0 \propto \sqrt{R} \propto \sqrt{v}$$

Then, from equation 3,

$$\frac{P_{in}}{Q_{in}} \propto \rho_0 C_0 \propto \frac{1}{v}\sqrt{v} = \frac{1}{\sqrt{v}} \text{ or } \frac{1}{\sqrt{R}} \qquad (8)$$

Because the volume flow is not much different from gas to gas, equation (8) tells us that the amplitudes of pulsation pressures in the discharge or suction cavity are smaller for lighter gases.

It should be noted that "proportional," or "$\propto$" does not mean an exact mathematical proportionality in the above discussions. Rather, it is interpreted as "approximately proportional," or "has a correlation factor near to 1."

## CONCLUSION

Valve designs that show some restrictions for R-12 and R-22 are ample for air and helium. Valve stops may have to be adjusted. Suction and discharge muffler designs have to be different for the refrigerants as compared to air and helium. The thermodynamic efficiency comparison includes losses due to valve action and gas pulsations. The comparison utilizing a coefficient of performance for the compressor alone has some meaning in terms of refrigeration, but needs further interpretation. It has to be remembered that the suction pressure and temperature and the discharge pressure were held constant in this study.

## ACKNOWLEDGMENT

Support by Necchi, S.p.A. for related work is greatly acknowledged.

## *REFERENCES*

1. W. Soedel, "Gas Pulsations in Compressor and Engine Manifolds," Short Course Text Book of Purdue Compressor Technology Conference, Ray W. Herrick Laboratories, Purdue University, 1978.
2. W. Soedel, "Introduction to Computer Simulations of Positive Displacement Type Compressors," Short Course Text Book of Purdue Compressor Technology Conference, Ray W. Herrick Laboratories, Purdue University, 1972.
3. J.F. Hamilton, "Extensions of Mathematical Modeling of Positive Displacement Type Compressors," Short Course Text Book of Purdue Compressor Technology Conference, Ray W. Herrick Laboratories, Purdue University, 1974.

| Gas | k | R<br>ft·lbf/°R·lbm | $P_s$<br>(psi) | $T_s$<br>(°F) | $P_d$<br>(psi) |
|---|---|---|---|---|---|
| R-12 | 1.18 | 12.77 | 20 | 150 | 190 |
| R-22 | 1.20 | 17.88 | 20 | 150 | 190 |
| Air | 1.40 | 53.34 | 20 | 150 | 190 |
| Helium | 1.67 | 386.25 | 20 | 150 | 190 |

Table 1. Input Data

| Gas | $T_d$<br>(°F) | $v_s$<br>(ft$^3$/lbm) | $v_d$<br>(ft$^3$/lbm) |
|---|---|---|---|
| R-12 | 315 | 7.38 | 0.35 |
| R-22 | 340 | 3.35 | 0.50 |
| Air | 586 | 10.18 | 2.04 |
| Helium | 893 | 73.72 | 19.10 |

Table 2. Specific Volumes

| Gas | $\eta_t$ | $C_p$<br>(lbm/kW-hr) |
|---|---|---|
| R-12 | 0.83 | 123.11 |
| R-22 | 0.86 | 87.55 |
| Air | 0.92 | 26.95 |
| Helium | 0.95 | 3.33 |

Table 3. Comparison of Thermodynamic Efficiencies and Performance Coefficient

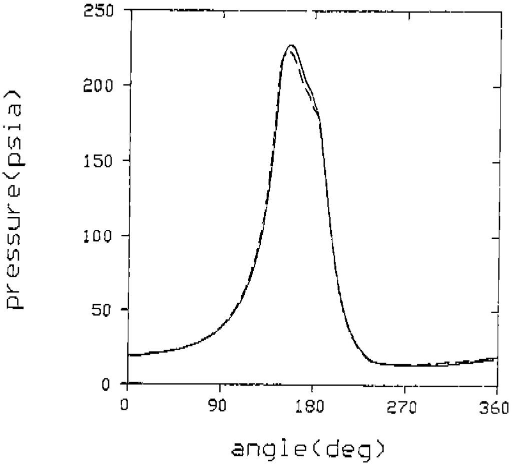

Figure 1 Cylinder pressures
——— R-12,- - - - R-22

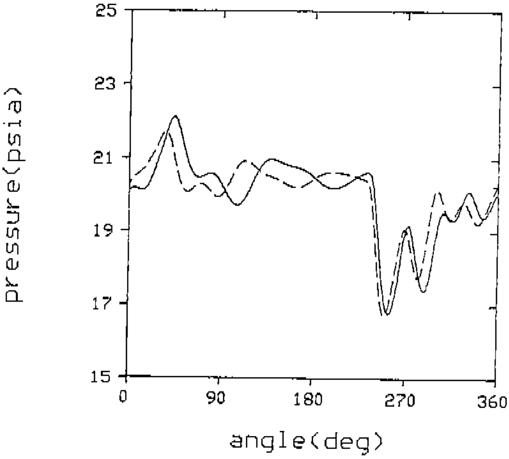

Figure 2 Suction pressures
——— R-12, - - - R-22

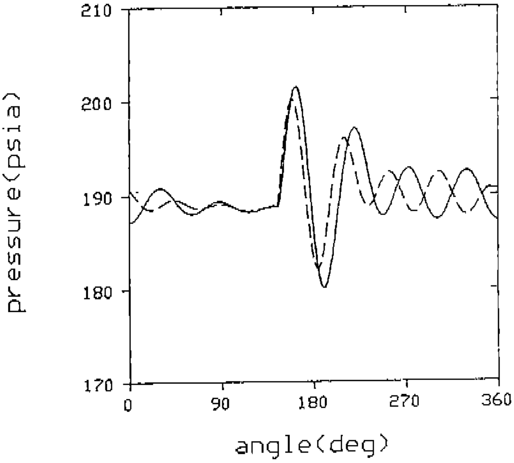

Figure 3 Discharge pressures
——— R-12, --- R-22

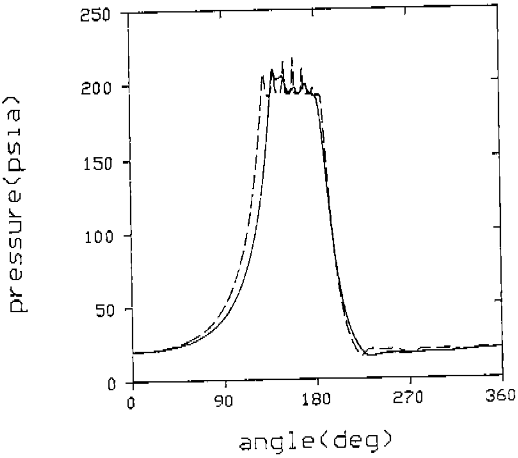

Figure 4 Cylinder pressures
——· air, --- helium

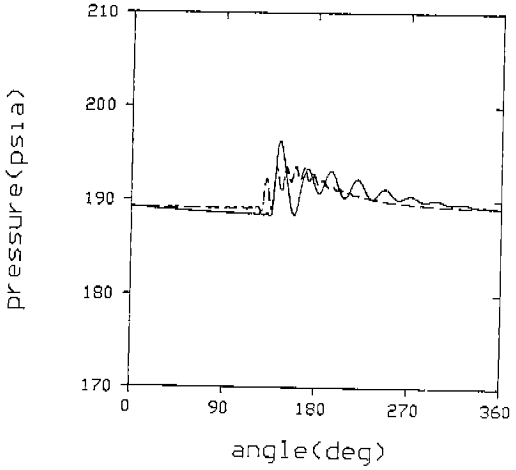

Figure 6 Discharge pressures
—— air, ---- helium

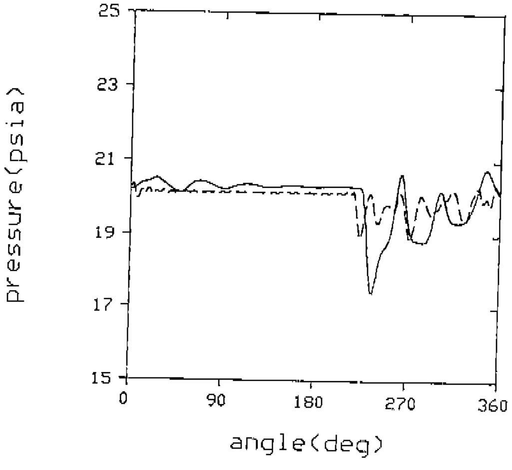

Figure 5 Suction pressures
—— air, ---- helium

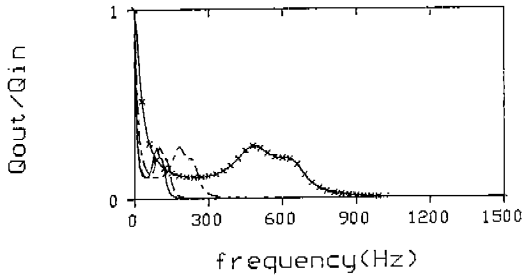

Figure 7 Volume flow transfer function of the discharge cavity
——— R-12, - - - - R-22, — - - —air, —×— helium

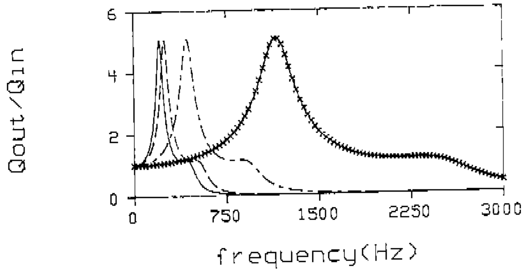

Figure 8 Volume flow transfer function of the suction cavity,
——— R-12, - - - R-22, — - - — air, —×— helium