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PERFORMANCE STUDY OF A VARIABLE SPEED COMPRESSOR WITH SPECIAL ATTENTION TO SUPERCHARGING EFFECT

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Abstract

Compressor performance is influenced by many factors. Besides the commonly considered factors such as suction gas heating, leakage, compressed gas re-expansion, gas pulsations, and back flow, supercharging effect on performance is also studied. Since the compressor used in this study operates in a wide speed range, from 1100 rpm up to 9000 rpm, how those factors affect the compressor performance under different speeds is quantitatively studied. However, it must be noted that the compressor studied was a research compressor (not a production compressor) and all simulation data is therefore to be interpreted on a relative scale.

Introduction

Tremendous improvements of system performance have been achieved in the field of refrigeration and air-conditioning during the past few decades. Energy conservation has been a very important research topic and is still the focus of the researchers in this field. Besides the legal requirement established by the United States government, the efficiency of refrigeration products is a very important parameter to the customers.

In the area of residential air-conditioners and household refrigerators, ON/OFF control is commonly used as a capacity control system. This technique is cheap and simple to implement. However, its disadvantage is the unavoidable reduction of seasonal energy efficiency ratio (SEER) due to the losses caused by the frequent ON/OFF cycling during actual changing load conditions [1].

At the emergence of variable frequency inverters, variable speed compressors became a reality. Therefore, the capacity of refrigerators and air-conditioners can also be controlled by the speed of the compressors. This new technique has attracted a growing interest for several reasons [2]. (1) In most applications, a variable speed controlled refrigeration system is more efficient than the system using ON/OFF or stepped capacity control. (2) A variable speed controlled system is capable of controlling its capacity in a wide range, and is able to adjust compressor output closely according to actual changing load conditions.

The performance of a variable speed rolling piston rotary compressor, which is shown in Figure 1, is investigated in this paper. Controlled by a frequency converter, this compressor can operate in the speed range from 1100 rpm to 9000 rpm. Compressor performance has been analyzed by many researchers [1, 3-5], but most of them assume that the compressor runs at the constant conventional speed. The intent of this paper is to study the effects of various factors on the compressor under different speeds, especially the speed dependent factor such as supercharging. Thus the performance of the compressor can be improved.

Overview of the Compressor Simulation

There are a number of complicated phenomena occurring inside the compressor during each cycle of operation: unsteady flow through the suction and discharge passages, thermodynamic processes inside the compressor cylinder, heat transfer in the compressor system, valve dynamic motion, and gas pulsations. Mathematical models are necessary to describe those phenomena. These models, consisting of a group of nonlinear ordinary and partial differential equations, can not be solved analytically because of their complexities and the coupling among them. Therefore, to predict the unknown variables associated with the operation of compressors, computer simulation is the only accurate and reliable tool.

In order to study the performance of the variable speed compressor, a complete computer simulation model was built. The primary features of the simulation include: a one-dimensional gas dynamic model for the intake, the first-law thermodynamic model for the cylinder, and the linear acoustic model for the discharge. The gas pulsation models of the intake and discharge are coupled to the cylinder thermodynamic model by the boundary conditions at the suction and discharge ports. Detailed description of the mathematical models can be found in reference [6]. For brevity, only the predicted results about the compressor performance are presented in this paper.

Performance Criterion

The performance of a compressor is not a uniquely defined term. Coefficient of performance (COP) and energy efficiency ratio (EER) are widely used in industry to evaluate the performance of compressors. By definition [7], COP is the ratio of the heat removed or cooling produced in Btu/hr to the work needed in Btu/hr, and EER is the ratio of the heat removed or cooling produced to the work required in Watt. Pandeya and Soedel [3] considered both of them too general to describe the performance of compressors, since what COP or EER represents is not the sole performance of a compressor but the performance of a whole refrigeration system consisting of an evaporator, a condenser, an expansion device, and a compressor, even though the compressor is the most important part and dominates the value of COP or EER.

A different performance criterion was proposed for compressors by Pandeya and Soedel [3], which is called Efficiency of Performance and is defined as the product of volumetric efficiency, compression efficiency, mechanical efficiency, and motor efficiency, such as:

$$\eta = \eta_v \, \eta_c \, \eta_f \, \eta_m \tag{1}$$

The above relation was also used by Ozu and Itami [4]. However η is called by them a synthesis efficiency or overall efficiency and the rest of the terms are defined as:

$$\eta_v = \text{volumetric efficiency} = \frac{\text{actually discharged gas mass}}{\text{theoretically discharged gas mass}}$$

$$\eta_c = \text{compression efficiency} = \frac{\text{theoretical gas compression work}}{\text{actual gas compression work}}$$

$$\eta_f = \text{mechanical efficiency} = \frac{\text{actual gas compression work}}{\text{motor shaft output}}$$

$$\eta_m = \text{motor efficiency} = \frac{\text{motor shaft output}}{\text{motor electrical input}}$$

This performance criterion will be also used in this study, except that the "compression efficiency" is replaced by a different term called "cylinder process efficiency". On the pressure-volume indicator diagram, the above so called "compression work" is identified as the enclosed area by the suction, closed compression, and discharge processes. Energy loss occurs not only during the closed compression process, but also during the suction and the discharge processes. Therefore the terminology "cylinder process efficiency" is more appropriate here than the "compression efficiency".

Mass Flow Gain, Losses, and Volumetric Efficiency

Volumetric efficiency is proportional to the actual mass delivered by the compressor per cycle and is defined as the ratio of the actually delivered mass flow rate to the ideally delivered mass flow rate. What cause the actual discharged gas mass to be less than the ideally discharged mass are suction gas heating, leakages from high pressure chamber to low pressure chamber, compressed gas re-expansion from the clearance volumes, and back flow through discharge valves. On the other hand, the supercharging effect of the suction line increases the volumetric efficiency when the suction pipe is tuned properly. All effects having influence on the volumetric efficiency will be studied in this section.

Mass Flow Gain due to Supercharging

Wave dynamics in the intake can significantly influence the compressor volumetric efficiency. A properly design suction line can cause the suction port pressure in the downstream to be higher than the nominal suction pressure in the upstream, and therefore supercharge the compressor cylinder. The supercharging phenomenon was studied in reference [8] in detail. When the suction chamber of the compressor is supercharged, an extra amount of gas is delivered into the cylinder. If only the supercharging effect is considered, the actual mass flow rate delivered by the compressor is greater than the ideal mass flow rate. Therefore, the actual volumetric efficiency can be more than 100 %.

The supercharging volumetric efficiency gain under different speeds is shown in Figure 2. The maximum supercharging effect occurs when the following simplified relation $N_{rpm} = 15\frac{a_0}{L}$ is satisfied [9], where N_{rpm} is the compressor speed in rpm, a_0 is the speed of sound, and L is the length of the suction pipe.

Suction Gas Heating Loss

When the gas picks up heat before it is enclosed in the cylinder, it expands and therefore causes less amount of gas being delivered into the cylinder. Suction gas heating occurs in two regions of the compressor system [10]. The first place is the suction pipe. The heat transfer rate between the suction pipe and the gas flowing inside it was included and studied in the unsteady compressible flow model [8]. The second region of suction gas heating is the suction volume of the cylinder. As refrigerant gas is pumped into the suction chamber, it is further warmed up by the hotter cylinder walls until the suction port is closed.

To see the effect of suction gas heating on the mass flow rate through the system, the computer simulation is first run without considering the heat transfer terms in both the suction pipe unsteady flow model and the cylinder thermodynamic model, then the simulation is performed by including the heat transfer terms. The difference in the amount of refrigerant mass delivered under those two cases gives us the mass flow loss due to heat transfer. The volumetric efficiency loss due to suction gas heating is plotted versus speed in Figure 2.

Other Mass Flow Losses

Leakages, re-expansion of the gas in the clearance volume, and back flow also cause less refrigerant to be delivered by the compressor. Detailed studies on them can be found in reference [6]. For brevity, only simulation results are shown Figure 2.

Overall Volumetric Efficiency

The ideal mass flow rate of the compressor is a function of the cylinder volume and the compressor running conditions, and therefore is known. The volumetric efficiency is just the ratio between the actual mass flow rate, obtained from the computer simulation, and the ideal mass flow rate. The volumetric efficiency corresponding to different speeds is plotted in Figure 6.

Energy Consumed during the Cylinder Processes and Cylinder Process Efficiency

Energy losses during the cylinder processes can be identified by comparing the ideal pressure-volume indicator diagram with the actual PV diagram of the cylinder processes. The actual PV diagram can be obtained either from experiment or from computer simulation. To illustrate the energy losses during each cycle, an actual PV diagram generated by the computer simulation is superimposed on the corresponding ideal PV diagram in Figure 3. All the factors causing the actual PV diagram to deviate from the ideal one are analyzed in this section.

Dynamic Loss in the Suction Process or "Wire Drawing" Loss

From the PV diagram, it is known that the energy loss during the actual suction process is due to the fact that the actual cylinder pressure in the suction process is lower than the ideal suction pressure, or the nominal suction pressure. Since there is no suction valve in the rolling piston rotary compressor, the under pressure is caused by the inertia of the gas. In other words, the refrigerant gas can not fill the suction volume fast enough during some portion of the suction process because of the dynamic effect.

The "wire drawing" loss under different conditions is plotted in Figure 4. The ratio of the "wire drawing" power loss to the total power required for the cylinder processes is shown in Figure 5.

Effect of Supercharging on the Energy Consumption

In the ideal compression process, the cylinder gas is compressed from the nominal suction pressure to the nominal discharge pressure. However, when supercharging occurs, the start pressure of the closed compression process is no longer the nominal suction pressure, but has a higher value. Even though we assume that the actual compression process following the supercharged suction process is adiabatic, like the ideal compression process, the work done by the piston during the actual compression process is still larger than that during the ideal one, as shown by the area 'abcda' on the PV diagram of Figure 3. However, this amount of extra energy is neither lost nor wasted, but is utilized to compress the extra amount of gas pumped into the cylinder during the suction process due to the supercharging effect. The better the suction line is tuned, the higher the start pressure of the compression process is, and the more extra amount of energy is needed for compression.

The extra energy required during the actual closed compression process (the area of 'abcda' on Figure 3) due to the supercharging effect is plotted versus the compressor speed in Figure 4, and its ratio to the total power for the cylinder processes is shown in Figure 5.

Effect of Heat Transfer on Power Consumption

The actual closed compression process is not adiabatic. Because of the temperature difference between the compressor cylinder walls and the cylinder gas, heat exchange actually accompanies the whole compression process. Therefore, the actual closed compression process behaves more like a polytropic process than an adiabatic process. To calculate the energy loss due to heat transfer, the computer simulation is first run by including both the supercharging effect and the heat transfer terms, then it is run without considering the heat transfer. The difference between the energy consumption during the two different closed compression processes is the energy loss due to the heat transfer, which is plotted versus the compressor speed in Figure 4. Its ratio to the total power needed for the cylinder processes is shown in Figure 5.

Effect of Leakages on Power Consumption

When the refrigerant gas leaks from the compression chamber to the suction chamber, the pressure in the compression chamber is relieved as compared with the case of no leakages. The consequence of the leakages is that less compression power is required for the lesser amount of gas in the compression chamber. However, the more important but negative effect of leakage on the compressor is its corresponding volumetric efficiency loss. The overall effect of leakage on the performance of the compressor is negative, which will be shown later on.

The power reduction due to leakages is plotted in Figure 4, and its ratio to the total power needed for the cylinder processes is shown in Figure 5.

Discharge Valve Loss

In the ideal discharge process, the cylinder gas is dumped out of the cylinder quasi-statically, and there is no pressure differential across the discharge valve. However, in the actual discharge process, the cylinder pressure is always higher than the nominal discharge pressure in order to overcome the flow resistance which is a function of the inertia and the spring resistance of the valve. Because of the restriction of the discharge valve, the compressor has to do an extra amount of work to push the gas out of the cylinder. This extra amount of work is called the discharge valve loss and is illustrated in Figure 3. The energy loss due to the discharge valves can only be solved numerically. The numerical results are plotted in Figures 4 and 5.

Discharge Gas Pulsation Loss

When the discharge valve loss was studied previously, the back pressure seen by each of the two discharge valves was assumed to be the constant reference discharge pressure. However, as the cylinder gas is periodically dumped into the discharge manifolds, the pressure in the discharge cavities will oscillate. The gas pulsations in the discharge manifolds were modeled and explained in reference [11]. The presence of gas pulsations causes the back pressures seen by the discharge valves to be temporarily higher than the nominal discharge pressure [12]. Therefore, the compressor has to work even harder to push the gas out of the way.

The energy loss due to gas pulsations can only be accurately obtained by computer simulation. The computer simulation program is first run by assuming that the valve back pressures are the same as the nominal discharge pressure, then it is run by including the gas pulsation effect. The difference in energy consumption during the two different discharge processes gives the energy loss due to the discharge gas pulsations, as plotted in Figure 4. The ratio of the discharge gas pulsations power loss to the total power of the cylinder processes is shown in Figure 5.

Cylinder Process Efficiency

The cylinder process efficiency or thermodynamic efficiency is the ratio of the work needed for the ideal cylinder processes to that of the actual cylinder processes, or the ratio of the enclosed area on the ideal PV diagram to that of the actual PV diagram. It is plotted versus the compressor speed in Figure 6.

Overall Efficiency of the Compressor

Neither the volumetric efficiency nor the cylinder process efficiency can be considered as an adequate judgment of the compressor performance. For instance, leakage reduces the volumetric efficiency but improves the cylinder process efficiency, while the supercharge effect increases the volumetric efficiency but lowers the cylinder process efficiency. Therefore, to see the net effects of the previously mentioned factors on the compressor performance, we need to study the influence of them on the product of the volumetric efficiency and the cylinder process efficiency, which is plotted in Figure 6.

As explained in the beginning, the overall efficiency of the compressor is obtained by multiplying the volumetric efficiency, the cylinder process efficiency, the mechanical efficiency, and the motor efficiency together. The first two efficiencies have been completely studied in this chapter, while the other two are considered as input parameters in this study.

Experimental Verification

To validate the simulation results, the compressor was run on a load stand. Both the mass flow rate and the power consumption were measured and compared with the predicted results, as shown in Figures 7 and 8. It is found that the computer model is satisfactory and reliable.

Discussions and Conclusions

Net Effect of Supercharging on the Compressor Performance

The volumetric efficiency gain is proportional to the degree of supercharging, which is explained in Figure 2. However, supercharging also causes the energy needed for the cylinder processes to increase and therefore reduces the cylinder process efficiency, as indicated in Figure 6.

To see the net effect of supercharging on the compressor performance, the ratio of the mass flow rate to the power required for the cylinder processes under only supercharging is plotted in Figure 9, and is compared with the ideal one, the actual one, and the actual one without supercharging. In the ideal case, one Joule of energy is required for the cylinder processes to deliver approximately 0.09 kg of refrigerant corresponding to the condition specified. But the actual cylinder processes are not so efficient. The net effect of supercharging on the ratio of the mass flow rate and the power required for the cylinder processes is negative, especially at high speeds. This means that supercharging degrades the performance of the compressor.

Significance of the Losses

Supercharging, heat transfer, compressed gas re-expansion, leakages, discharge valve, discharge gas pulsations, and back flow are all the factors influencing the compressor efficiency. All of them degrade the compressor performance. The next step is to identify the factors of greatest significance to the compressor efficiency so that we can start from the right direction to improve the compressor efficiency.

The product of volumetric efficiency and cylinder process efficiency is plotted in Figure 10, corresponding to many different cases. It has the lowest value when all the factors mentioned in the first paragraph are included. When the leakages are not considered, its value increases dramatically, which means that reducing the mass flow losses due to leakages will improve the compressor efficiency considerably. Figure 10 also shows that leakages and clearance volume are the most detrimental factors to the efficiency of the compressor, while discharge gas pulsations and back flow have very little effect; heat transfer and the discharge valve are the factors which fall in between. Supercharging does not reduce the efficiency very much under a certain speed, but its effect becomes greater at high speeds.

Conclusions

In this paper the performance of a variable speed compressor has been analyzed. The results are summarized as follows. (1) The supercharging effect increases the capacity of the compressor significantly, but reduces the overall efficiency of the compressor slightly. (2) Leakage is the dominant loss factor at low speeds, and compressed gas re-expansion is the dominant loss factor at high speeds. (3) The product of volumetric efficiency and cylinder process efficiency has its highest value in the speed range between 5500 rpm and 6500 rpm, consequently the compressor is most efficient in this speed range. (4) Generally speaking, the increase of speed will not reduce the overall efficiency of the compressor.

Acknowledgment

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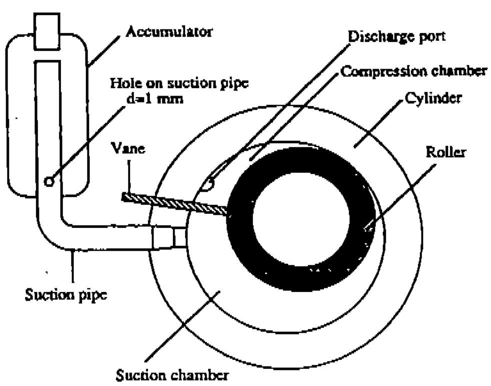

Figure 1 A rolling piston rotary compressor mechanism

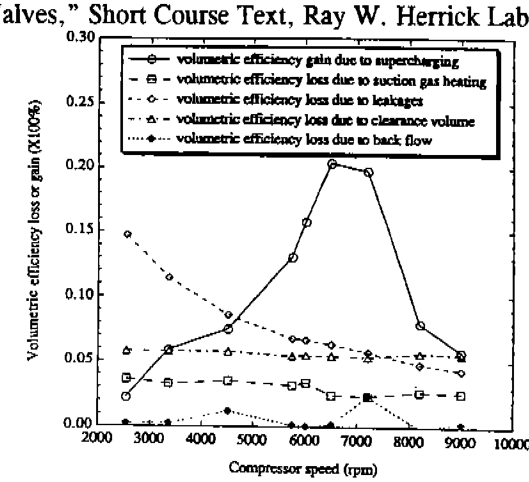

Figure 2 Volumetric efficiency gain and losses under different speeds (p_s=63.5 psia, T_s=45.5 F, p_d=242 psia)

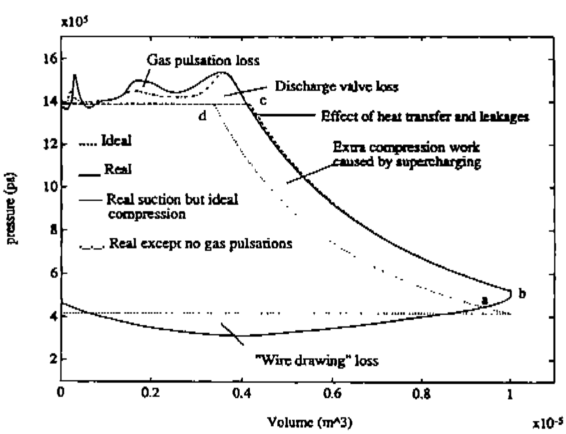

Figure 3 Energy losses on PV diagram (Ω=6550 rpm, p_s=418511.76 pa, T_s=42.9 F, p_d=1388604.1 pa)

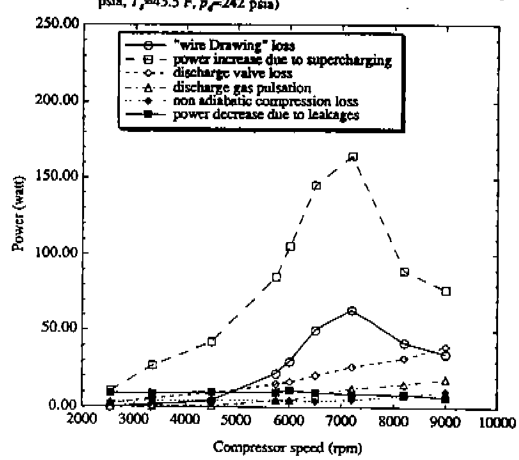

Figure 4 Power variation under different speeds (p_s=63.5 psia, T_s=45.5 F, p_d=242 psia)

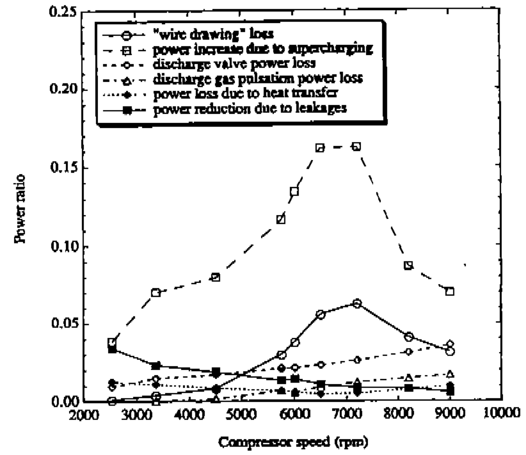

Figure 5 Ratios of the power variations to the total power needed for the cylinder processes (p_s=63.5 psia, T_s=45.5 F, p_d=242 psia)

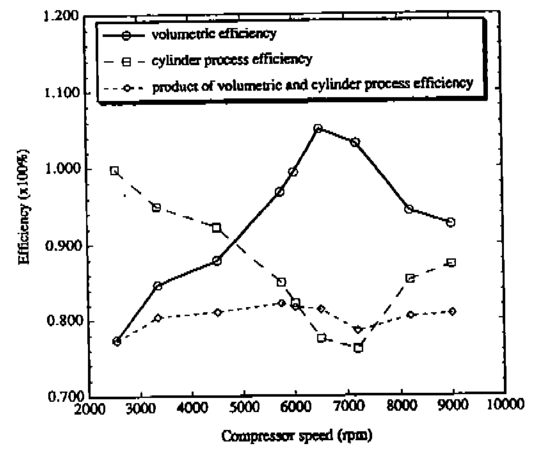

Figure 6 Comparison among volumetric efficiency, cylinder process efficiency, and their product (p_s=63.5 psia, T_s=45.5 F, p_d=242 psia)

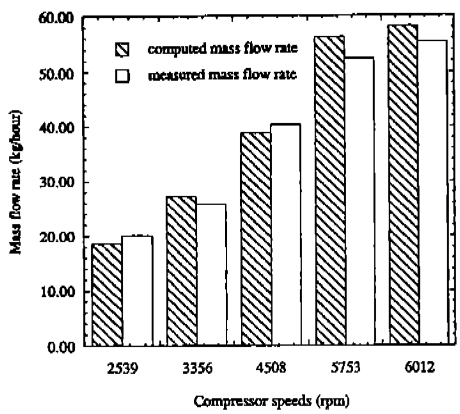

Figure 7 Mass flow rate comparison (p_s=63.5 psia, T_s=45.5 F, p_d=242 psia)

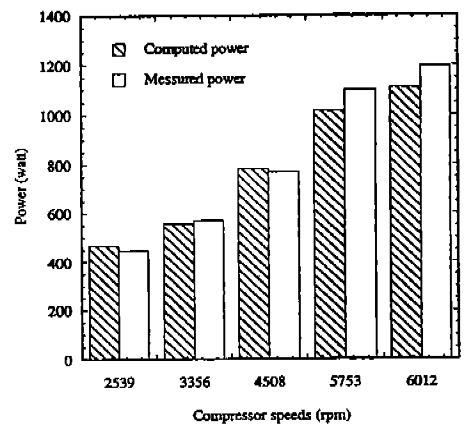

Figure 8 Total power consumption of the test compressor

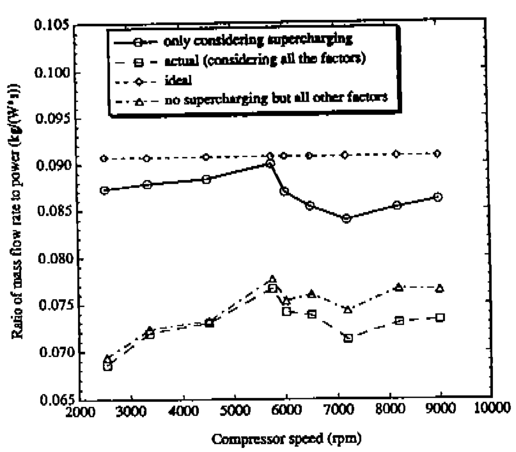

Figure 9 Ratio of mass flow rate to the corresponding power required for the cylinder processes (p_s=63.5 psia, T_s=45.5 F, p_d=242 psia)

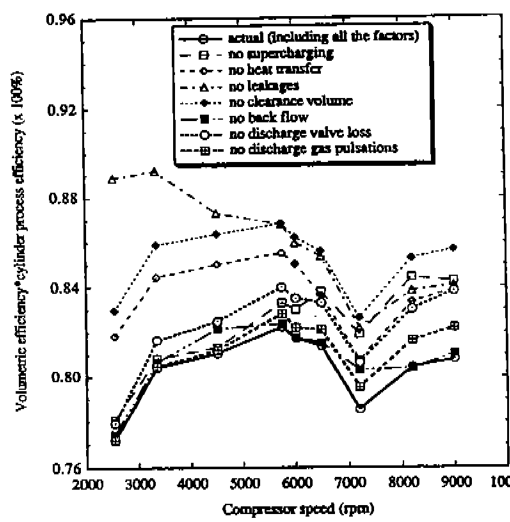

Figure 10 Product of volumetric efficiency and cylinder process efficiency under different situations (p_s=63.5 psia, T_s=45.5 F, p_d=242 psia)