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A Simplified Procedure for Designing Hermetic Compressors

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ABSTRACT

Research on the various aspects of hermetic compressor design and development has been going on for a long period of time. As old questions get answered, new ones keep coming up. So much work has been done that a beginner in the compressor design field could easily get confused, and might not even know where to start.

The purpose of this paper is to present a consolidated and simplified design approach that could be used as a guideline to get started with the very complex task of designing a hermetic compressor. Ideas that have proven successful over the years have been discussed. Particular attention has been given to design for performance. Critical problem areas and issues that influence the design tremendously have been discussed.

INTRODUCTION

Designing a compressor -- is it an art or a science? Most of us have wondered at one time or the other about the answer to this question. Volumes of research work done in the past decades [3, 6, 7] indicate that there is alot to be learned, investigated, and understood. And yet, as any experienced compressor designer would tell us, there are subtle mysteries involved in compressor designing that would require an experienced hand to unravel. Perhaps, one would have to admit that designing a compressor is both an art and a science. Indeed one could very well say that science itself becomes art when it is well understood and well practiced.

Although the title of the paper suggests that it is limited to hermetic compressors, it would be safe to say that the ideas expressed here could easily be applicable to most other types such as open and semi-hermetic air-conditioning and refrigerating compressors, and even air and gas compressors to some extent.

Ideally, a compressor designer should be a researcher (both a theoretician and an experimentalist), a practical hand, and an innovator -- all combined into one. Unfortunately (or fortunately for some of us), such a person does not exist. Most of us in the compressor field have had to or have to begin our careers with very little knowledge about compressors. What is most needed at those times is a simplified procedure that explains how to go about designing one. An attempt has been made in this paper to fulfill that need.

CONCEPT DEVELOPMENT

Generally, development of a product goes through two distinct phases: "Concept Development" and "Product Design". Figure 1 shows, in brief, a typical flowchart of the concept development phase. The four steps, that one must go through in this phase, are discussed below:

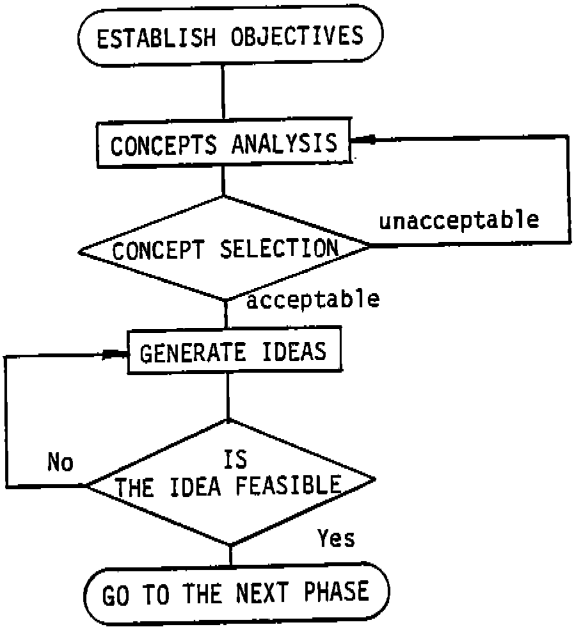

Figure 1: Concept Development

Establishment of Objectives: The first and perhaps the most important step from the management's point of view is the setting of proper objectives or the design criteria that the final product must achieve. Depending upon how realistic the objectives are, the ultimate design may either end up as a grand success story or in utter failure and frustration. Ironically, the compressor designer has least say at this stage of the game. The objectives generally are both technical and non-technical. Most important ones are: efficiency, capacity, acceptable noise level, reliability requirements, system requirements, cost, size, available development time, and available resources for the project.

The technical objectives are almost always dictated by ultimate consumer considerations. For instance, higher efficiency became a very important factor after the last energy crisis which generated a lot of enthusiasm for energy conservation. The non-technical objectives may depend a lot on the prevailing competitive market trends as well as on other business considerations of the company.

Concept Analysis: The second step in the concept development phase is the analysis of the basic compressor concepts in use at the time in the industry. For example, reciprocating, rotary, scroll, screw, Wankel, ROVAC, etc. This is more or less like a "fact finding mission" and must be handled by a committee of people knowledgeable in various aspects of the business -- both technical and non-technical. The designer's input is very minimal at this stage. In any case, one concept is selected, to be pursued for further development.

Idea Generation: The next step in the concept development phase consists of generating ideas within the framework of the selected concept, that would meet the objectives established earlier. A very common approach used is brainstorming sessions where everyone knowledgeable in the field is invited to present ideas without any fear of being ridiculed. The ideas then need to be sorted out for their applicability, practicality, cost, etc. From this point on, the compressor designer's role becomes exceedingly important.

Feasibility Study: At this point, it becomes very important that only those ideas be considered that have considerable merit. A brief paper analysis of each worthwhile idea is done. This study should consist of preliminary sketches for physical design considerations; preliminary design calculations such as loads, capacity, etc.; and basic thermodynamic considerations for the soundness of the idea. The ideas that pass through this analysis should then be tested for feasibility using modified existing hardware. These tests should be very preliminary in nature. Their purpose is only to test the soundness or validity of the idea, and NOT to pass any judgement on the relative worth of the idea. However, at the end of this study, a general consensus can be drawn as to which particular idea should be developed further. Sometimes, more than one idea may emerge as candidates at this stage, in which case either all of them can be simultaneously developed or the most suitable one can be picked, depending upon the funds available for the development program.

PRODUCT DESIGN

The second phase of the product development program may well be called the product design phase. The flow-chart shown in figure 2 describes this phase graphically. Once a particular concept has been decided upon, the next step is to establish the preliminary design parameters such as swept volume, number of cylinders, bore/stroke ratio, etc. The following relationships could be used to obtain the total swept volume at this stage.

$$\text{Total swept volume} = \frac{\text{Capacity (BTUH)}}{60\ \rho_s N\ \eta_{vol}\ \Delta h} \qquad (1)$$

Where ρ_s = suction gas density

N = RPM of the motor

η_{vol} = volumetric efficiency

Δh = enthalpy increase during evaporation

Volumetric efficiency will have to be assumed at this point based on existing compressors of similar design. Generally, at this stage, most of these parameters are picked up based on existing designs and / or experience. The purpose is to simply establish some starting point or base design data.

<u>Design for Performance</u>: EER (energy efficiency ratio) or COP (coefficient of performance) and volumetric efficiency are typically used as the performance indicators for the compressors in the air-conditioning and refrigeration industry. EER is defined as the compressor capacity (BTU/H) divided by the power consumed in watts. The tests conducted during the feasibility study would give an idea of the base performance values. The next step is to figure out what design variables affect the performance most and then to attempt to optimize them for the best performance.

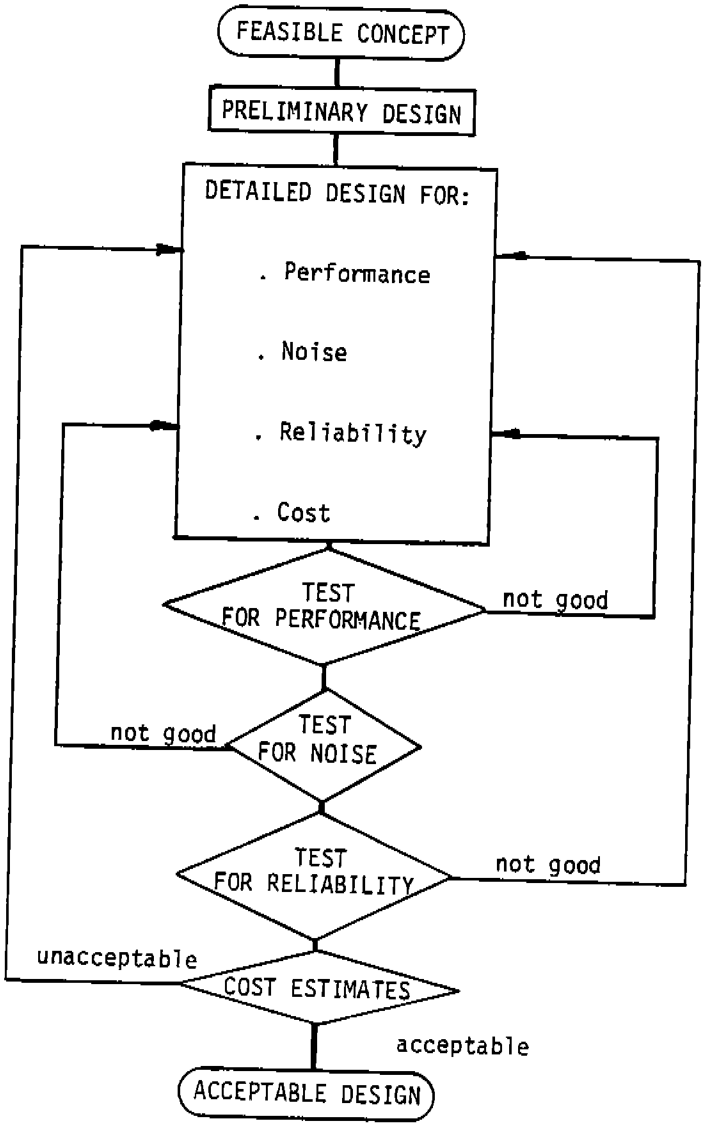

Figure 2: Product Design Phase

Thermodynamic Analysis: Using the energy and mass balance approach from reference [1], one can write:

$$EER = 3.413 \frac{(m - \Delta m)\ \Delta h}{(m - \Delta m)\ \Delta H \ + \Delta W} \qquad (2)$$

Where $m = V\rho$ = ideal mass flow per cycle

Δm = mass flow loss per cycle

Δh = enthalpy rise during evaporation

ΔH = ideal enthalpy rise during compression

ΔW = energy loss per cycle

One can easily see that Δh and ΔH are totally dependent on the operating conditions and refrigerant properties. m depends on the displacement and the operating conditions. Thus, Δm and ΔW are responsible for the degradation in EER. Thus, at ASRET conditions i.e. 45°F evaporator, 130°F condenser, 95°F return gas, and 15°F subcooling, one obtains 16.8 BTU/WH as the optimum value of EER using R-22 refrigerant. By changing the refrigerant, however, this optimum will also change, subject to the limiting value given by the Carnot cycle, which for the same operating temperatures would be:

$$\text{Carnot Cycle EER at ARI} = 3.413 \frac{460 + 45}{(460 + 130) - (460 + 45)} = 20.3 \text{ BTU/WH}$$

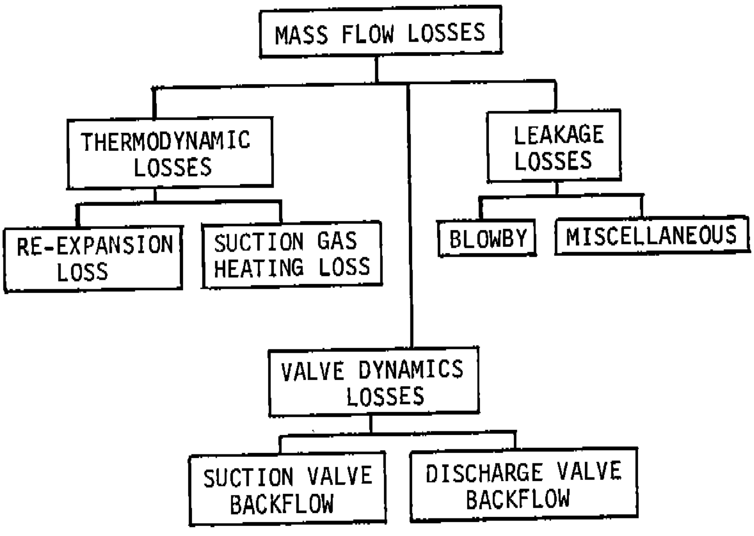

Figure 3: Mass Flow Losses

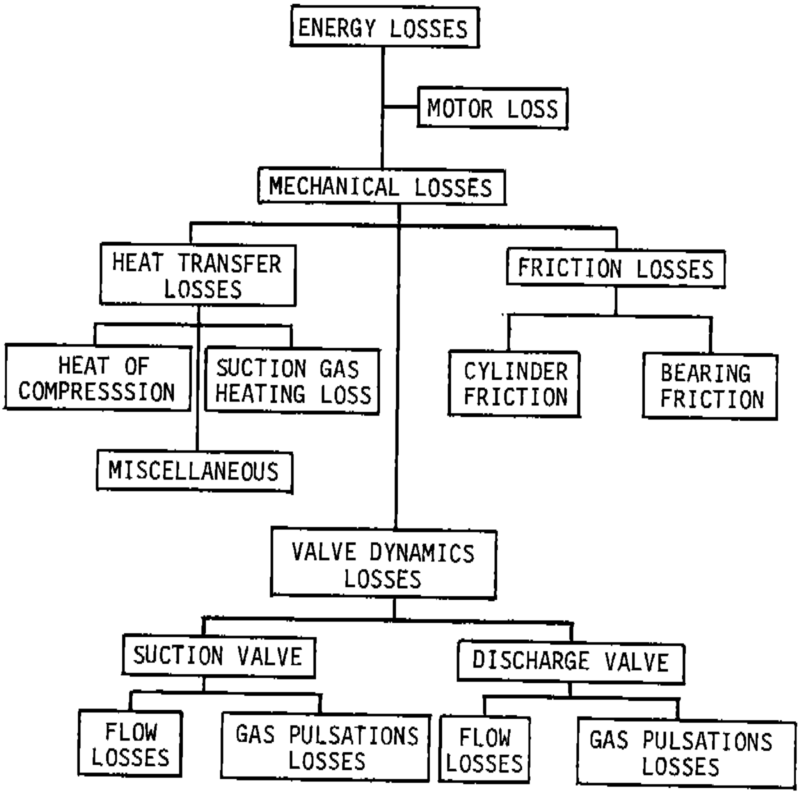

Figure 4: Energy Losses

The difference between the optimum value of 16.8 BTU/WH for R-22 and 20.3 BTU/WH for Carnot cycle is only due to the "real fluid" properties of the refrigerant R-22. In the real world, EER is way below 20.3 or even 16.8. In order to optimize the performance, therefore, one has to minimize the mass flow and energy losses as far as practically possible. Figures 3 and 4 show these losses graphically.

The leakage and friction losses depend very much on the particular design and geometry of the compressor and are also somewhat interdependent on each other. Motor loss depends on the type (and cost) of the motor used. All the remaining losses can be grouped under two broad categories: valve losses, and heat transfer losses.

Valve Design Procedure: Suction and discharge valves produce energy loss in the form of under-pressure and over-pressure respectively (see figure 5). Closely related with these and perhaps conflicting in nature are the suction and discharge valve backflow losses. By far, these two offer the biggest opportunity in performance improvement via optimized valve design. Valve dynamics also affects the gas pulsations. Subject of valve design has been vastly covered by researchers, and an excellent source of information and bibliography is

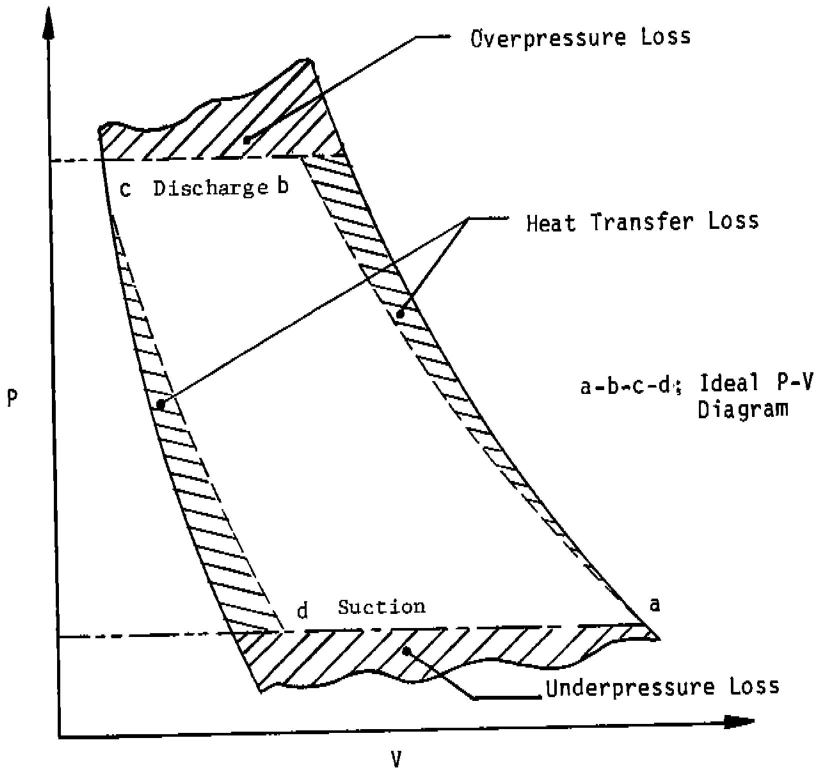

Figure 5: Typical P-V Diagram of a Compressor.

reference [3]. For a beginning engineer, however, some rules of thumb will be mentioned here.

1. Valve characteristics such as preset, overlap, and stiffness increase the energy loss and reduce the backflow when they are increased. Increased inertia of the valve generally results in increase of both energy loss as well as backflow. Increase or decrease in valve stop height could affect either way depending on whether it has been optimized or not.

2. Flow restrictions increase energy as well as backflow losses.

3. Gas inertia (pulsations) could affect either way, depending on how the system has been tuned acoustically.

As far as the gas pulsations loss is concerned, typically it increases with increase in instantaneous mass flow rate through the ports, decreases with increase in plenum volume and area of plenum exit. Of course, the muffler design and acoustic tuning affect it the most, in both ways.

Knowing the swept volume from equation (1) and the design conditions, which are usually the most important rating conditions, one can create a theoretical P-V diagram similar to figure (5) which can then give the net volume during the suction and discharge processes. Then, using kinematic equation for volume for the driving mechanism being used [5], one can calculate the discharge time and suction time and thus the suction and discharge port areas.

$$A_s = V_s / t_s M C_s \qquad (3)$$
$$A_d = V_d / t_d M C_d \qquad (4)$$

Where:

A_s, A_d = suction and discharge port areas

t_s, t_d = suction and discharge times

V_s, V_d = net suction and discharge volumes per cylinder

C_s, C_d = sonic velocity in the medium at suction and discharge conditions

M = Mach number

Thus, $A_s/A_d = V_s\, t_d\, C_d / V_d\, t_s\, C_s$ (5)

The importance of relationship (5) becomes obvious when one realizes that the total space available for both suction and discharge port areas is limited and depends upon the cylinder geometry and the type of valve selected. Of course, this is where the ingenuity of the designer becomes important. In any case, knowing the valve and cylinder geometry, the available space for the port areas can be distributed into A_s and A_d according to the equation (5). Next, the stop height is found by dividing the port area by the corresponding perimeter, assuming that the valve moves straight up and down.

At this point, one needs to determine the stiffness of the valve. Whereas on one hand, one would like to have very flexible valves to reduce the overpressure and underpressure, on the other hand, it is desirable to have very stiff valves to reduce the backflow loss. As it turns out, in most cases, it is the reliability issue that dictates the final outcome. In any case, for a first design one can assume roughly 10% overpressure and / or underpressure. Thus

$$K_s = 0.1\ P_s A_s / X_s \qquad (6)$$
$$K_d = 0.1\ P_d A_d / X_d \qquad (7)$$

Where:

K_s, K_d = Suction and discharge valve spring stiffness

P_s, P_d = Suction and discharge pressures

A_s, A_d = Suction and discharge port areas

X_s, X_d = Suction and discharge valve stop heights

Next step is to select the material, which is generally high speed spring steel; although stainless steel can also be used. Knowing the material properties one can determine the thickness that would give the desired stiffness.

Problem of Heat Transfer: Heat transfer in a hermetic compressor is a very complicated phenomenon. Short of a complete heat transfer model, it is almost impossible to quantify the various losses. However, a qualitative understanding of the various heat transfer paths and their effects on performance of the compressor should be of considerable help.

Basically, there are three sources of heat in a hermetic or semi-hermetic compressors: motor losses (Q_m), mechanical frictional (Q_f), and viscous friction (heat of compression, Q_c). Similarly, there are three sinks: suction gas (Q_s), discharge gas (Q_d), and environment (Q_e).

Thus,

$$Q_m + Q_f + Q_c = Q_s + Q_d + Q_e \qquad (8)$$

Typically, in low side (where compressor housing is at suction or low pressure) compressors, Q_d is negligible; while in high side compressors, Q_s is negligible. The effect of a higher value of Q_s is to heat the suction gas, thereby lowering its density with consequent loss in mass flow rate and capacity. At the same time, work of compression also increases. Therefore, suction gas heating must be avoided as much as practically possible in order to achieve higher efficiency.

Unfortunately, in lowside compressors, the only places that the heat generated can go to are either the environment or the suction gas (Q_d=0). The other alternative, of course is to reduce Q_m, Q_f, and Q_c as much as possible. However, in a high side compressor (Q_s=0), the heat goes to the discharge gas which only increases the load on the condenser. Of course, motor efficiency goes down as a result of operating at higher temperatures.

<u>Design for Reduced Noise</u>: There are two types of noise sources:

1. Sources that generate compressor noise:

   - Suction and discharge valves
   - Mechanical unbalance
   - Torque pulsations
   - Poor crankcase mounting system
   - Gas space resonance
   - Shell vibrations

2. Sources that generate little compressor noise, but a lot of system noise:

   - Suction gas pulsations
   - Discharge gas pulsations
   - Poor compressor mounting system

Noise generated by valves is mostly the result of high valve impact velocity, which is a direct result of higher stiffness and larger valve stop height and undesirable natural frequencies. Thus, to reduce the valve generated noise, one should try to reduce either the stiffness, or the stop height, or both. It is obvious to any designer that mechanical unbalance should be avoided as far as practically possible. Torque pulsations depend very much on variables such as number of cylinders, compressor kinematics, etc. and may be beyond the designer's reach at this point. Crankcase mounting is very important in flexibly mounted compressors (typically those which have high amplitudes of vibration). The object is to isolate the mechanical vibration of the driving gear and the crankcase from the housing as much as possible. Question of gas space resonance and shell design is better left to experts rather than a beginning compressor designer.

The suction and discharge gas pulsations increase the evaporator and condenser coil resonance respectively and hence should be reduced as much as possible. Effective muffler design is essential to achieve this goal. This problem too needs expert attention, or a lot of trial and error and patience (!)

Design for Reliability: Weakest links in hermetic compressors typically are the valves and the bearings, although under severe operating conditions numerous other parts can fail. Therefore, special attention must be given to designing valves and the lubrication system.

Valves can fail due to overstressing caused either by higher order bending modes or high velocity impact or excessive deflection resulting from liquid slugging. Considerable amount of published work exists in the area of valve design [3, 6, 7, 8]. Private business houses typically have simulation programs and finite element analysis programs tailored to their specific needs. Given below are a few rules of thumb for reducing valve failures:

Reduce valve stop height
Increase valve support area (overlap)
Reduce port area
Increase material thickness
Increase allowable stress by changing the material.

Unfortunately, all except the last alternative will result in reduced efficiency.

Bearing failure occurs either due to lack of lubrication or due to excessive loading, or sometimes even due to poor bearing material. It is the author's opinion that key to a reliable bearing design lies in a good lubrication system characterized by sufficient lubricant circulation to carry the frictional heat away and a good venting scheme to avoid vapor - lock problems. Of course, excessive operating conditions can wipe out even the best designed bearings.

At this point one should also check other critical parts, such as shaft, connecting rods, valve plates, and crankcase for stresses and deflection in critical areas. The design may have to be beefed up in some cases.

Prototype Testing: The designer should now have sufficient information to put together a complete set of drawings to help cost out and build the prototype. The next step is to conduct the prototype testing to see if all the original objectives have been met. If it is a typical design, it would take several iterations before the final acceptable design will emerge.

CONCLUDING REMARKS

An effort was made in this paper to detail a step-by-step compressor design procedure for the benefit of a typical beginning compressor engineer/designer. It should be well understood that this paper only scratches the surface of enormous amount of work that has been done in the field of hermetic compressor design. In fact, emphasis has been on simplicity rather than on thoroughness.

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