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Scroll Compressor Design Criteria for Residential Air Conditioning and Heat Pump Applications, Part I: Mechanics

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SCROLL COMPRESSOR DESIGN CRITERIA FOR RESIDENTIAL
AIR CONDITIONING AND HEAT PUMP APPLICATIONS
PART I: MECHANICS

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ABSTRACT

The application of scroll compressor technology to the residential air conditioniong and heat pump markets has generated a variety of compressor design approaches to address special system requirements. This paper will review both special design needs of this application and unique scroll design concepts used to address these needs. Design concept discussions will include:

- Scroll compliance - radial and axial
- High pressure versus low pressure shell
- Bearing systems

Scroll compressor technology, in comparison to other positive displacement types, will be outlined in terms of energy efficiency, noise and vibration, durability, and field application. Both inherent design advantages and design challenges will be reviewed for scroll compressors.

INTRODUCTION

A scroll concept was identified early in the twentieth century as a steam engine expander [1], and shortly after was recognized as having potential as a gas compressor [2] [3]. Until recently, both technology and motivation for high compressor performance were not sufficient to warrant engineering resources to develop the design as a production product. Even today, some view the scroll as an object of curiosity with limited potential to compete with existing positive displacement compressor technologies. It is the objective of this paper to demonstrate the scroll compressor to be the clearly superior design technology of the future. Included in this review is a discussion of both scroll technology and design criteria necessary to meet the rugged demands of a high performance, high durability air conditioning and heat pump compressor.

Being a relatively new technology for serious development, engineers have approached scroll design from many directions with the result being interesting technical differences of opinion as to the best design approach. One example of this is the decision to use either a low pressure or high pressure compressor housing. Reciprocating compressors generally employ a low pressure housing, while fixed vane rotary compressors generally operate with a high pressure housing. Scroll compressors with either a high or low pressure housing are both in production today.

THE SCROLL ADVANTAGE

Regardless of the design approach, scroll compressors have inherent advantages over comparable reciprocating and rotary technologies. Most important of these is efficiency where the scroll potential is approximately 10 percent better than competing technologies. Figure 1, which illustrates a breakdown of compressor energy loss for each technology, shows the primary difference between scroll and other technologies to be the near zero valve and gas flow loss. In general, the scroll compressor is more efficient due to a nearly continuous flow process with the absence of valve loss, excellent sealing potential, and enhanced thermal efficiency.

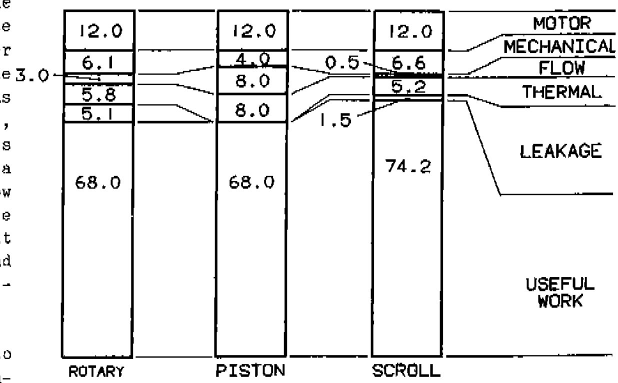

FIGURE I

COMPRESSOR LOSS ANALYSIS (ARI-1990 PROJECTED)

In addition to inherently high mechanical efficiency, scroll compressor efficiency is further enhanced in an air conditioning and heat pump system due to the compressor's near 100 percent volumetric efficiency at all operating conditions. Figure 2 illustrates system cooling and heating capacity when both scroll and reciprocating

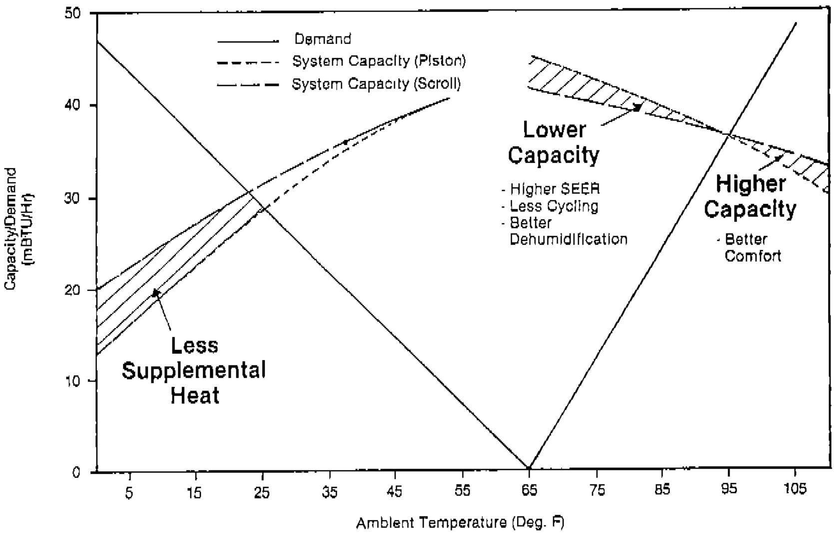

Figure 2

System Capacity vs. Residence Demand

compressors are used to satisfy a typical three-ton residential requirement. For comparison, both compressors yield the same system capacity at the 95°F ambient rating point. However, at the 82°F point used for rating the system energy efficiency ratio (SEER), the scroll compressor capacity is sufficient to meet the cooling requirement but less than the comparable reciprocating compressor. This reduced scroll system capacity results in a reduced temperature differential across the system condenser and evaporator coils which, in turn, lowers the compressor's compression ratio and increases both the compressor and system energy efficiency ratio. Depending on the system efficiency, the net effect of this capacity unloading benefit is approximately 2 to 4 percent. Thus, a scroll and a reciprocating compressor may have identical efficiencies at a specific rating point, but the scroll will still average a system efficiency 3 percent better than the reciprocating compressor.

During the heating mode of operation, the higher heating capacity obtained with the scroll compressor relative to the reciprocating design results in higher overall system efficiency due to a reduction in the use of electrical resistance makeup heat. Figure 2 shows the scroll capacity to be 10 to 20 percent higher than the comparable reciprocating compressor over most heating operation.

The near continuous flow process important to scroll compressor efficiency also results in inherently low noise due to low gas pulsation. The scroll compression and discharge processes occur over approximately 800 degrees of crankshaft rotation. By comparison, the rotary and reciprocating compressors have 360- and 180-degree processes respectively. Also, the abrupt valve impact noise present with reciprocating and rotary compressors is replaced by the continuous rolling and sliding motion of the scroll vanes. Finally, scroll compressor vibration is very low due to the smoother flow process and potential for near perfect balancing of all dynamic shaking forces. This results in minimal rotational and lateral vibration components.

Inherent high durability is an additional scroll advantage precipitated by the compressor's smooth flow process and a minimal number of moving parts. For a typical scroll compressor, the only moving parts consist of an eccentric shaft, an orbiting scroll, an oldham coupling, and where radial compliance is present, an additional coupling to provide variable eccentricity. These components, when used in a fully compliant design, have a high tolerance for both system contaminants and liquid flooding. Both laboratory contaminant tests and field application tests for scroll compressors have demonstrated the compressor's ability to pass typical system contaminants with only minor markings of the scroll flanks and no significant loss of performance. This assumes a compressor suction path which is capable of separating larger, heavier particles by either gravitational, inertial, or filtering methods. In other words, particles up to 0.25mm in size have been found to cause minimal scroll damage whereas larger particles may cause performance deterioration and possible scroll vane breakage. Low side (low pressure shell) compressors with indirect suction processes effectively filter larger particles from the scroll vanes resulting in exceptional contaminant tolerance relative to both reciprocating and rotary technologies.

Scroll compressor tolerance to liquid refrigerant flooding is unlimited in nearly all real system applications. The practical aspect of this is the elimination of the accumulator and crankcase heaters normally used with reciprocating and rotary compressors. Tests in "worst case" laboratory systems and field application units have demonstrated a minimal scroll reaction to liquid refrigerant during both flooded start (overnight liquid migration to the compressor) and heat pump defrost testing. This efficient liquid handling characteristic is best demonstrated by listening to the consecutive startups of a scroll and reciprocating compressor with the same liquid refrigerant charge contained in the compressor shell. Typically, the reciprocating compressor will make a loud noise due to valve operation and hydraulic pressures (2000 psi) in the cylinders. On the other hand, the scroll noise increase will be only slightly noticeable.

The relatively slow compression process of the scroll contributes to its liquid handling ability. This provides ample time for any liquid to leak to lower pressure zones before pocket pressures become excessive. Also, the use of a compliant design (discussed later) allows sealing surfaces to separate, enhancing the leakage. Yet another factor in high scroll liquid tolerance is the relative insensitivity of the scroll vane components to the marginal lubrication occurring during liquid passage. The rolling sliding motion of the scroll vanes requires minimal lubrication and can survive extended periods with liquid refrigerant removal of oil from the contacting surfaces.

In summary, the scroll compressor has many inherent advantages relative to competing technologies when applied in residential air conditioning and heat pump applications. In addition to high efficiency potential, both noise levels and durability can be excellent with proper scroll compressor design. Application simplifications such as the elimination of crankcase heaters and accumulators can further enhance system efficiency as well as provide an overall applied cost which is competitive with other technologies.

SCROLL TECHNOLOGY

Because the scroll compressor is unique relative to other positive displacement types, specific scroll technology areas should be understood prior to selecting a preferred design approach. In this section, key scroll technologies will be discussed.

Axial Compliance

Controlling leakage between the vane tip and the baseplate is the most critical sealing requirement in the scroll compressor. The potential leakage perimeter can typically be several times that of the flank contacts. Usual methods of controlling this leakage are tip seals and biasing the scroll members together using gas or spring forces.

Tip Seals: This is a fairly common method relying on a floating element placed in a groove machined into the vane tip. See Figure 3. Advantages include simple construction and the use of conventional machining tolerances to control vane height and tip clearances.

Design of a reliable and durable seal has been a long-term challenge for scroll designers. The simplest designs include strips of thermoplastics or engineering resins, but are typically challenged by premature wear. Also, stiffness of the seal strip can cause binding in the close-radius inner wrap, affecting sealing in the most critical region. The use of a laminated metal seal is occasionally a solution to these problems. The laminated construction provides flexibility while the metal is chosen for wear resistance.

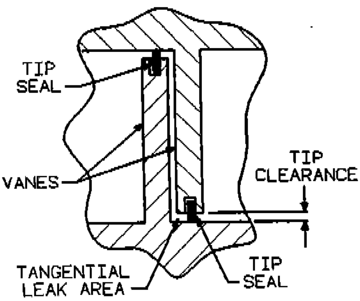

FIGURE 3
TIP SEALS

Problems inherent to any tip seal design centers around groove formation and scroll component size. Tip seal grooves are usually formed by an end milling process. This requires lengthy machining time and the use of small, fragile cutting tools. Also, the extra vane width required to carry the seal results in increased scroll size and weight, increasing both cost and critical bearing loads.

An inherent leak also requires attention. The area formed between the seal edge, sidewall, vane tip, and baseplate allows a tangential flow of gas. Control of this area is typically reflected in tighter control of tip clearance tolerances, reducing one of the advantages of this method.

Gas Loading of Orbiting Scroll

This technique uses gas pressure underneath the orbiting scroll to load it against the fixed scroll. The vanes are formed to very closely matching heights (less than 0.01 mm) and leakage is controlled through very small tip-to-base clearances or contact. See Figure 4. This has the advantage of more straightforward vane geometry and reduced part size, but requires tight vane height tolerances and special sealing of the pressure balancing cavity. Biasing pressure may be provided by discharge gas or by a combination of discharge gas and an intermediate pressure tapped from the compression pockets through vents in the orbiting scroll baseplate.

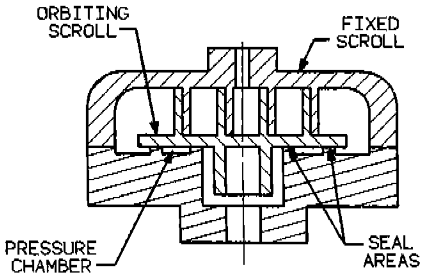

FIGURE 4
GAS LOADED ORBITING SCROLL

A characteristic disadvantage of this design is related to the pressure-induced loads required to maintain sealing. It is well-known [4] [5] that in addition to the normal thrust load, a tipping moment generated by the tangential drive load acting normal to the distance between the drive bearing and mid-vane must be resisted. See Figure 5. In rigidly supported orbiting scroll designs, this is accomplished by sizing the thrust bearing diameter large enough to allow the hydrodynamic thurst reaction to be offset sufficiently from the resultant of the axial gas force to generate a restoring moment. See Figure 6.

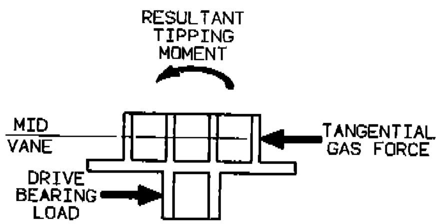

FIGURE 5
ORBITING SCROLL TIPPING MOMENT

In the axially compliant, or "floating" orbiting scroll, the upward thrust load is essentially hydrostatic in that its resultant is always on center. The only way to generate a restoring moment is to overpressure the upward thrust load, inducing a substantial load on the vane tips. The thrust reaction of the overpres-ure is capable of acting off center to generate the restoring moment. See Figure 7. The penalty for this approach lies in the sometimes heavy tip load (usually when other loads are heavy as well). This induces unnecessary friction, leading to a loss of efficiency, and high unit loading of scroll surfaces which can lead to excessive wear or even galling of scroll vane tips.

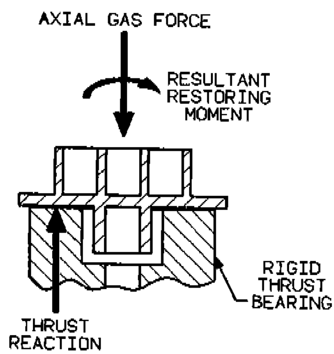

FIGURE 6
ORBITING SCROLL THRUST REACTION

Gas Loading of Fixed Scroll

This approach is similar in principle to gas loading of the orbiting scroll, except that the fixed scroll is the pressure loaded member. The orbiting scroll is supported by a rigid thrust bearing. The fixed scroll is suspended by means which are very stiff radially, yet flexible axially. One very successful approach has been to attach it to steel straps, or "leaf springs" as shown in Figure 8.

This design shares the advantages of the compliant orbiting scroll but has sealing and tolerance requirements which are equally difficult to attain. However, an additional advantage lies in the ability to virtually eliminate the friction loss resulting from overloading to generate a restoring moment.

Since the orbiting scroll is rigidly supported, its restoring moment is generated in the manner shown in Figure 6. Any generation of a similar moment in the fixed scroll is avoided by placing the radial supports at the level of the center of the vanes so that the mechanical reaction is essentially in line with the internal gas load. Since the tipping moment is balanced with no need for additional gas loads, only enough tip load to maintain sealing against a small moment caused by the inherently off center axial gas force is induced. Frictional loads and losses are substantially reduced from the case of the upwardly loaded orbiting scroll.

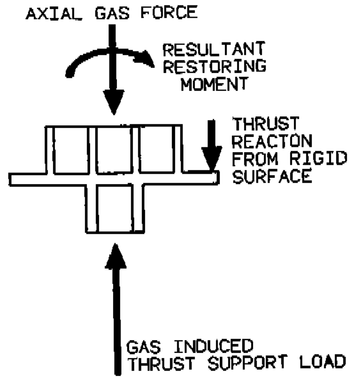

FIGURE 7
FLOATING SCROLL THRUST REACTION

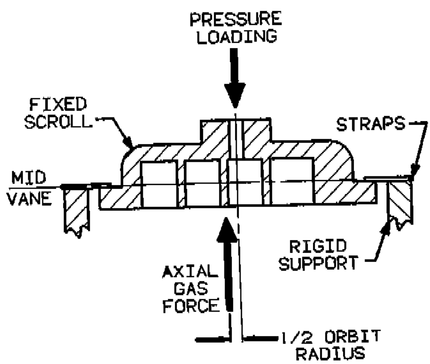

FIGURE 8
GAS LOADED FIXED SCROLL

Radial Compliance

The term "radial compliance" refers to the ability of the orbiting scroll to seek its own orbit path (as defined by the scroll vane geometry) in order to maintain flank contact and to move out of the way of foreign material such as solid contaminants or liquid refrigerant.

This freedom of motion is usually accomplished by connecting the bearing which drives the orbiting scroll to the crankshaft by a pivoting or sliding kinematic pair such that the motion allowed the orbiting scroll is essentially radial. The centrifugal force generated by the orbiting scroll mass can be used to counter the radial gas force and to provide a flank sealing load. There are generally two methods which may be used to modify this sealing load.

A fairly straightforward approach adds a counterweight to the drive bearing member. The force generated may be used to either add to or reduce the flank sealing load. An increased load is desired when the scroll mass is inadequate to generate sufficient sealing force. A decreased load may be used to reduce friction losses or wear when the scroll mass is larger. Caution should be taken to prevent instability inherent to this design when used to reduce flank loads. In this case the center of mass of the scroll and counterweight assembly is very close to the center of rotation. Often small deflections of the orbiting scroll move the center of mass inward, or even over center, reducing the centrifugal force and causing the flank load to be overcome by the radial gas force. Intermittant, or even complete unloading can result, causing noise and loss of performance mance. A common remedy is to provide some sort of mechanical stop to limit radial travel.

Modifying the transmission angle of the kinematic pair in the drive mechanism may be used to control the flank load without any inherent stability problems. The product of the tangential gas force and the cotangent of the transmission angle provides a radial force which is independent of centrifugal effects and which will vary roughly with the radial gas force. This is illustrated in Figure 9. Increasing or decreasing the transmission angle (from 90 degrees) will tend to respectively increase or decrease flank load.

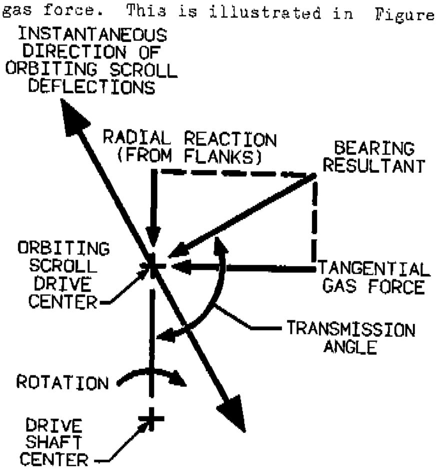

FIGURE 9
RADIAL COMPLIANCE WITH MODIFIED FLANK LOADING

Controlling flank clearances in non-compliant, so-called "fixed crank throw," designs is a formidable task. In this case the orbit path is independent of vane geometry. Flank clearances can only be controlled through a combination of tight machining tolerances and close assembly alignment. Minimum flank clearances in this case can be achieved by aligning the fixed scroll during assembly while motoring the orbiting scroll, and by allowing clearances in the drive (journal) bearing to provide a form of limited compliance. This is occasionally risky, since intermittent flank contact can produce significant impact loads on the oldham coupling (or other anti-rotation device) and any solid contaminant larger than the minimum allowed flank clearance can cause serious scroll damage. Any error in alignment or normal wear in main bearings can also cause flank impact.

Bearing Layout

In conventional piston and rotary technology, it is a fairly easy matter to straddle the compression device with two main or stationary bearings connected to a rigid frame. With this approach, the main bearing loads are similar and are not much larger than one-half to three-quarters of the average compression load. This arrangement generally provides support for an overhung rotor, but may not provide stability against rotor deflection from electrical and centrifugal forces.

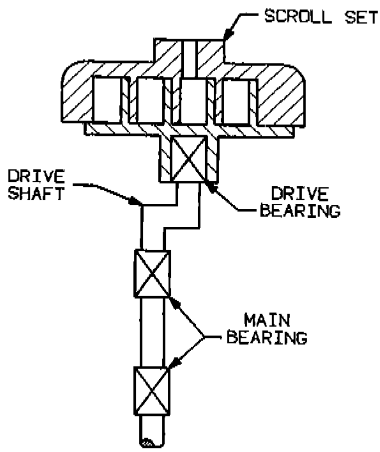

FIGURE 10
SCROLL COMPRESSOR DRIVE ARRANGEMENT

In the scroll compressor, the scroll vanes occupy the central region of the assembly and prohibit running the crankshaft through to either side. This results in an overhung drive arrangement where both main bearings are on one side of the compressor. See Figure 10. Special problems relating to bearing loads, angular alignment, and oil supply are posed.

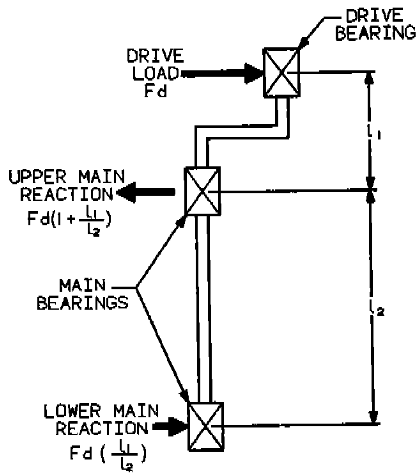

FIGURE 11
SCROLL SHAFT SCHEMATIC

Referring to the shaft schematic of Figure 11, note that the minimum load carried by the main bearing closest to the scrolls is equal to the drive load for extremely large main bearing separation. More realistically, the load will be somewhat greater. Since higher loads require larger bearings which represent increased drag and efficiency loss, it is desirable to control loads by making the bearing spacing large and bringing the bearing nearest the compressor close to the drive bearing.

Due to the cantilevered nature of the loading, the bearing reactions are opposite in direction to each other. If journal bearings are used, the shaft will take on an angular misalignment limited by the journal bearing clearances. It becomes even more desirable to maximize the main bearing spacing to control this condition.

Motor air gap control is another condition making a wide bearing spacing desirable. Minimum air gap variation provides more consistent starting torque and minimum electrically induced bearing loads. Controlling shaft angular misalignment also controls the air gap.

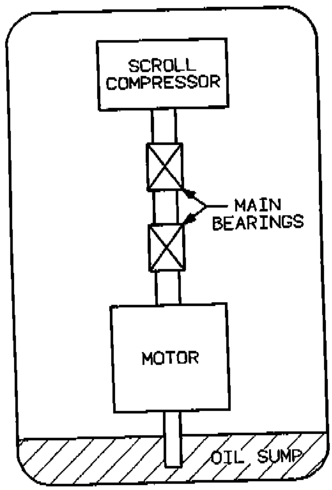

FIGURE 12
SCROLL LAYOUT - OVERHUNG MOTOR

Many scroll compressors have been built using the concept of a single bearing housing with both main bearings between the scrolls and motor. See Figure 12. While more conventional in layout and apparently cost effective, this approach requires design concessions in the form of oversized bearings, limited air gap control, counterbored rotors, and excess overall height.

A design approach which is somewhat less conventional but considerably more effective is to place the two main bearings on either side of the motor. See Figure 13. While this involves the apparent extra effort and expense of a secondary bearing housing, the benefits of minimum bearing size, minimum angular misalignment, rigid air gap control, elimination of rotor counterbore, and reduction in height provide maximum efficiency at a cost which is, at most, equivalent to that of other designs.

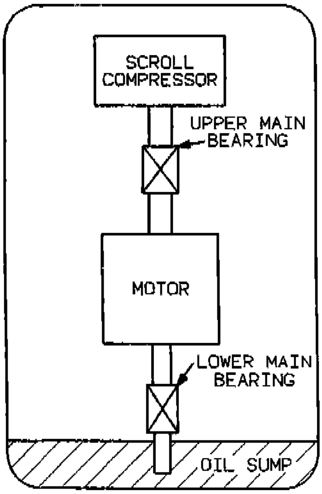

FIGURE 13
SCROLL LAYOUT - STRADDLED MOTOR

Gas Management

A key aspect for efficiency in any hermetic compressor design is control of suction gas preheating. Heating of suction gas represents a loss in refrigerant mass flow with no corresponding drop in power requirement. In the ideal case, no preheating would exist at all. In typical air conditioning or heat pump applications, each 5 F increase in suction temperature represents about a one percent drop in mass flow and efficiency.

Referring to Figure 14, the two methods used to restrict suction gas preheating are to maximize heat transfer to the ambient environment and to minimize re-circulation of compression and discharge heat.

The greatest heat source in the hermetic compressor which cannot be avoided is the electric motor. In a typical three-ton application, it alone can raise suction gas temperature by 20 F. An extremely effective method to reduce this loss is to press the stator into a compressor shell. The shell acts as an extended heat transfer surface and around 20 percent of the preheat due to the motor can be avoided.

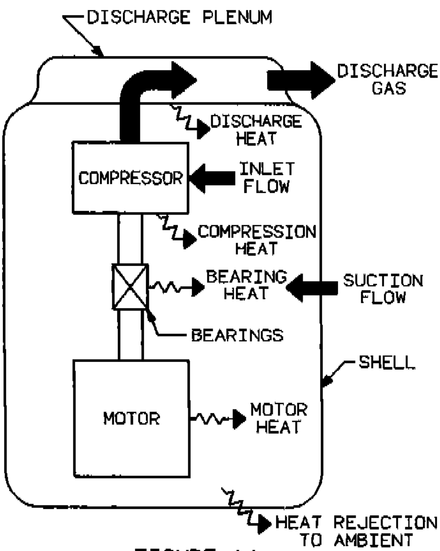

FIGURE 14
COMPRESSOR HEAT FLOW

Heat from bearing losses and the compression process are transferred to the suction gas from the bearing housing and scroll surfaces. Compression heat is especially undesirable since it does not represent a fixed loss but is only recirculation of heat which ideally should leave the compressor in the discharge gas. Any reduction in compression work due to modification of the polytropic process is small compared to the resulting loss in capacity. A solution is to isolate the suction gas flow, allowing only as much gas to circulate in the mid-shell region as necessary to cool the motor and bearings. Maintaining elevated mid-shell temperatures both decreases heat transfer from scroll housing surfaces and increases heat transfer to the ambient environment.

Heat transfer from the discharge plenum to the suction gas is just as undesirable since it is also a form of compression heat, and in theory totally avoidable. Were it not for the internal discharge plenum being an extremely effective and inexpensive means of reducing the discharge pressure pulse, it would be best to direct the gas immediately out of the shell using a short tube. A more practical approach isolates both suction and mid-shell gas from discharge heat with a pocket of more or less quiescent gas placed between the discharge plenum and the rest of the compressor assembly.

The scroll compressor is especially suited to all these methods of heat energy control. Its well-balanced operation permits the use of a rigid assembly which includes a pressed motor. The extreme liquid tolerance allows a very highly directed suction path which can also occasionally guide a slug of liquid refrigerant directly to the compressor inlet. The potential for damage to the suction or discharge valves of conventional piston and rotary compressors makes such a high degree of suction isolation very difficult and often requires the use of suction accumulators or other protection.

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