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SURFACE VEHICLE RECOMMENDED PRACTICE

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Guide to the Application and Use of Passenger Car Air-Conditioning Compressor Face Seals

1. ***Scope***—This SAE Recommended Practice is intended as a guide in the usage of mechanical face seals for the passenger car air-conditioning compressor application. Included in this guide is a compilation of present practices; for example, a description of various type seals, material combinations, design data, tolerances, drawing format, qualification testing, inspection information, and quality control data. The terminology used is recommended to promote uniformity in seal nomenclature.

2. ***References***

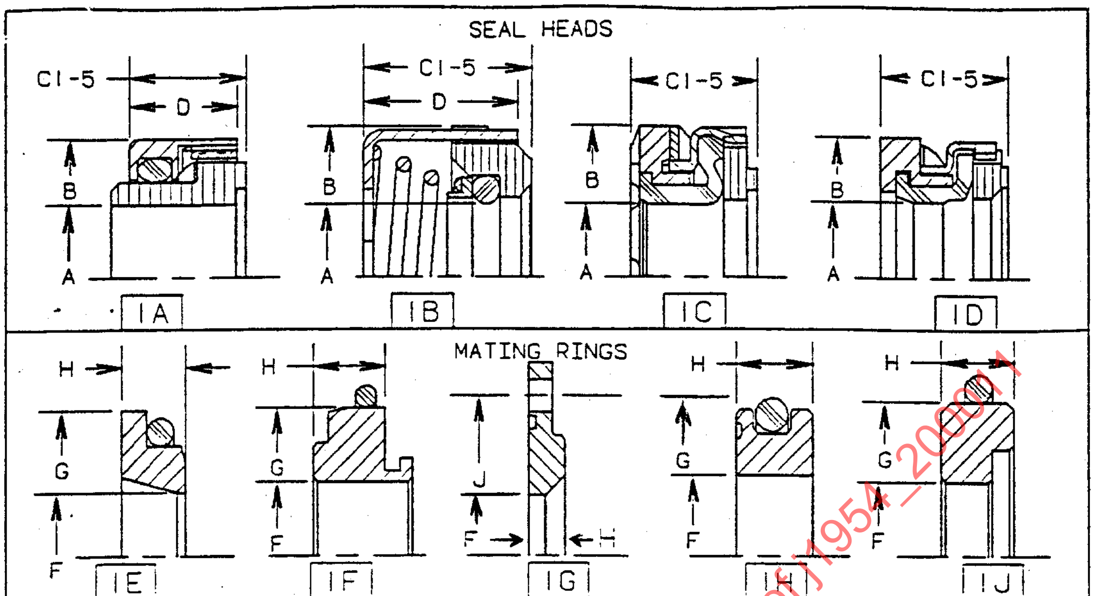
2.1 **Applicable Publication**—The following publication forms a part of this specification to the extent specified herein.

2.1.1 ANSI PUBLICATION—Available from ANSI, 11 West 42nd Street, New York, NY 10036-8002.

ANSI B46.1

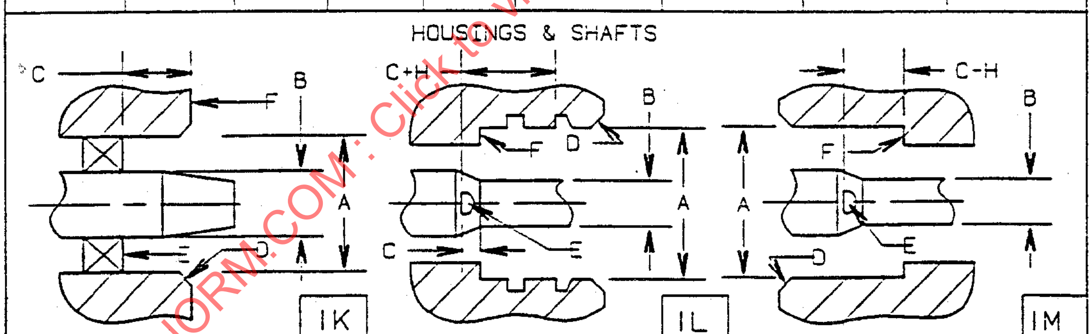
3. ***Seal and Mating Ring Types***—The mechanical face seal assemblies utilized in passenger car air-conditioning compressors consist of a seal head and a mating ring. Although many variations of face seal designs exist, two basic concepts are predominantly applied with respect to both seal head and mating rings.

3.1 **Seal Head**—The two basic design classifications are pusher and nonpusher. By definition, pusher seals are mechanical seals employing a secondary sealing element (such as O-rings, V-rings, U-cups, wedges, etc.) that are pushed along the respective sealing surfaces while the primary sealing function at the faces is being performed. A typical pusher seal is shown in Figure 1B. Nonpusher seals shown in Figures 1A, 1C, and 1D employ a bellows or diaphragm as a secondary sealing element. Axial displacement of the seal components caused by wear at the seal faces and/or shaft movement causes sliding or “pushing” of the secondary seals in the pusher seals. This same displacement in nonpusher seals is taken up by flexing of the bellows. The selection of seal head design is based primarily on seal environmental conditions.



SEAL ASSEMBLY FIG'S 1A, 1B, 1C, & 1D	MATING RINGS FIG'S 1E, 1F, 1G, 1H & 1J	APPLICATION PASSENGER OR NON PASS	A MIN./MAX. BORE I.D.	B MAX. OUTSIDE DIA.	C1 MIN./MAX. FREE LENGTH	C2 MAX. SOLID LENGTH	C3 MIN. OPERATING LENGTH	C4 NOM. OPERATING LENGTH

C5 MAX. OPERATING LENGTH	D MAX. SHELL LENGTH	E MAX. LOAD AT C3	E2 NOM. LOAD AT C4	E3 MIN. LOAD AT C5	F MIN./MAX. MATING RING ASS'Y I.D.	G MIN./MAX. MATING RING ASS'Y O.D.	H MIN./MAX. MATING RING ASS'Y THICKNESS	J MIN./MAX. MATING RING BOLT CIRCLE



A SEAL HOUSING BORE DIA.	B SHAFT DIA.	C SEAL CAVITY LENGTH	D SEAL BORE LEAD IN CHAMFER	SURFACE ROUGHNESS A	SURFACE ROUGHNESS B	SHAFT END PLAY	CONCEN-TRICITY A-B	SQUARE-NESS B-E	SQUARE-NESS B-F

FLUID COMPOSITION__________
R.P.M.__________MIN.______MAX.
TEMPERATURE______MIN.______MAX.
PRESSURE________MIN.______MAX.
OIL VISCOSITY__________
REMARKS__________

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FIGURE 1—DRAWING FORMAT

Independent of classification, the seal head is the rotating element of the seal assembly and is comprised of the following components:

a. Primary Ring (see Item 1, Figure 2) which is in rubbing contact with the stationary mating ring (see Item 5, Figure 2). This interface forms the primary seal.
b. Secondary Seal (see Item 2, Figure 2) as defined in 3.1.
c. Spring (see Item 3, Figure 2) which provides mechanical load to the primary faces.
d. Hardware consisting of retaining and drive devices (see Item 4, Figure 2) which provide reactionary support for the spring and transmit shaft rotation drive to the primary ring. Secondary seal support function is also provided by these components. On some nonpusher seal designs, seal head drive is provided via a positive interference of the elastomeric secondary seal element with the compressor shaft and hardware components.

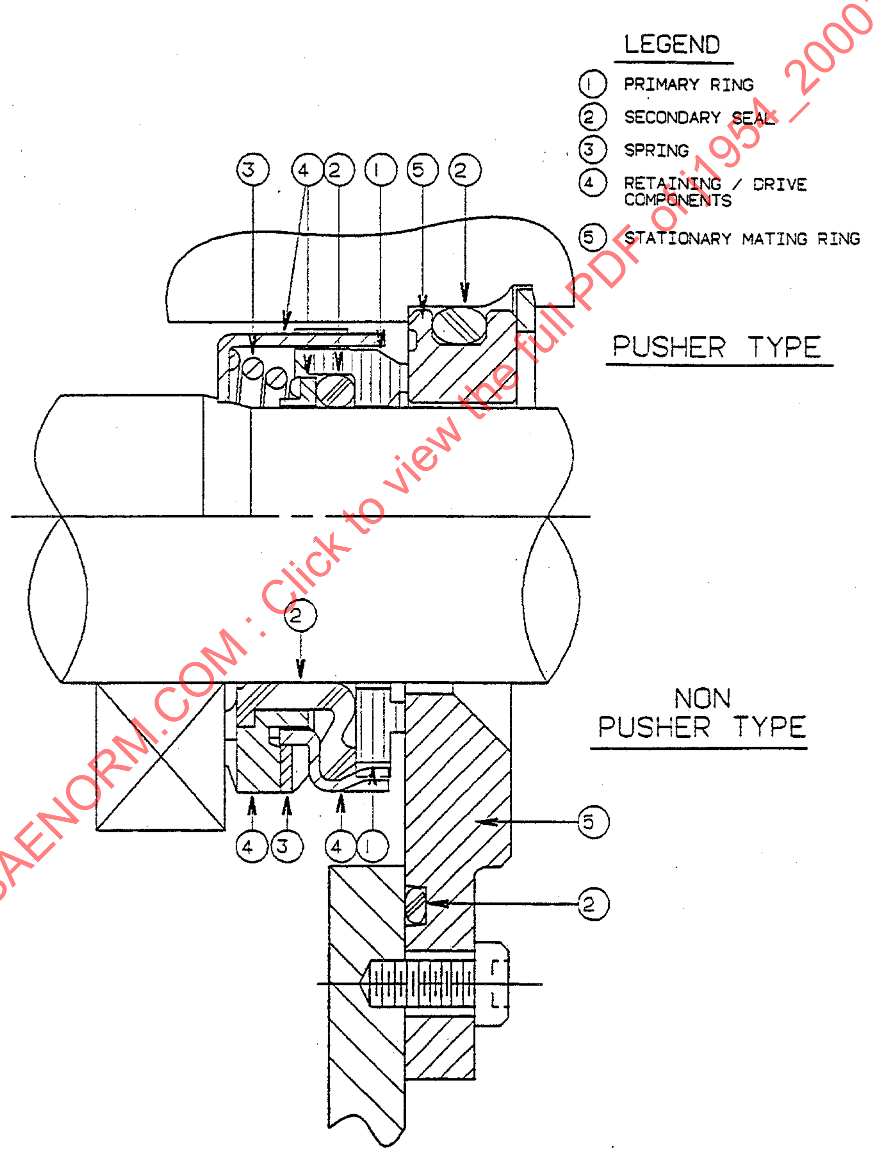

FIGURE 2—SEAL COMPONENTS

3.1.1 NONPUSHER SEAL—Nonpusher seals have been preferred for passenger car air-conditioning compressor seals where one or more of the following conditions exist:

a. Axial shaft movement with amplitudes greater than 0.203 mm (0.008 in).
b. Deposition of carbonized oil, foreign material, and/or corrosive products on the compressor shaft in sufficient magnitude to restrict the movement of the pusher-type secondary seal. This restriction prevents the seal from following shaft dynamics or prevents the primary faces from axial movement to accommodate seal face wear. The end result is excess leakage.
c. Distortion of the primary seal faces caused by excessive volume swell of secondary seal elastomer.
d. Excessive face loading due to pressure being sealed. Reduced seal balance is required to minimize heat generation in applications where standard balanced seals cannot be accommodated. Nonpusher seals, although classified as unbalanced, can be designed to have a balance of less than one without a stepped-shaft design normally required for a balanced seal. It should be noted that the balance of nonpusher seals will vary with the pressure sealed. The amount of balance variation is a function of the bellows design and materials. The term "seal balance" as used here is the ratio of hydraulic area closing the seal face to that of hydraulic area opening (Ac/Ao).

3.1.1.1 Advantages—In addition to the application conditions listed previously, nonpusher seals have the following advantages:

a. Normally do not require a positive mechanical drive with the compressor shaft
b. Are more tolerant of out-of-square mating ring face conditions

3.1.1.2 Disadvantages

a. Generally more expensive due to the bellows
b. Generally more difficult to install into compressor, due to the interference fit between shaft and drive portion of bellows

3.1.2 PUSHER SEALS—Recent passenger car compressor designs have substantially improved the seal environment, particularly with respect to temperature excursions and shaft end play. These improvements have permitted the use of pusher seals.

3.1.2.1 Advantages

a. Generally lower cost due to the O-ring
b. Generally reduced exposure of the elastomeric member to interfacial frictional heat
c. Closer primary ring-to-shaft concentricity

3.1.2.2 Disadvantages

a. Require positive shaft drive
b. More vulnerable to seal "hang-up"
c. More vulnerable to primary ring face distortion caused by secondary seal volume change

3.2 Mating Ring—Two basic mating ring designs are incorporated in air-conditioning compressor seals:

a. End-plate designs as shown in Figure 1G which are an integral part of the compressor
b. Separate secondary seal-mounted mating rings as shown in Figures 1E, 1F, 1H, and 1J

The selection of the mating ring design is based principally on compressor design, seal operating conditions, and mating ring material.

3.2.1 END PLATE DESIGNS—End plate designs are generally machined from fine-grained cast iron with a precision-lapped seal mating surface.

3.2.1.1 Advantages

a. Requires less axial space
b. Provides better frictional heat transfer capabilities

3.2.1.2 Disadvantages

a. Geometry generally limits material selection
b. Subject to face distortion due to bolt stresses. Distortion can be minimized by use of a clamping ring which eliminates localized bolt stresses.
c. More difficult to lap primary seal face due to size and shape

3.2.2 SEPARATE MATING RINGS—Separate mating rings are fabricated from various materials and are precision lapped.

3.2.2.1 Advantages

a. Due to relatively simple shape, rings can be produced from various materials such as cast iron, sintered iron, high alumina ceramics, etc.
b. Lower cost due to simpler configuration and smaller size
c. Less face distortion when installed
d. Easier to lap

3.2.2.2 Disadvantages

a. Reduced frictional heat dissipation
b. Secondary seal is often subjected to higher temperatures due to closer proximity to the primary seal faces
c. More potential for installation damage or misassembly of secondary seals

3.3 Secondary Seals—Three basic secondary seals are employed. The most universal is an O-ring. Flat gaskets and molded or lathe-cut rings are also incorporated in conjunction with end-plate designs and mating rings. In end-plate designs, the O-ring, molded or lathe-cut ring is preferred over flat gaskets for control of flatness and to improve heat transfer from the end plate.

O-ring configurations are also preferred for separate mating rings, again primarily for simplicity and cost.

4. Seal Materials—Environmental conditions dictate the type of material which should be used in a specific application. Seal materials can be fully evaluated only in terms of specific operating conditions and performance requirements. The following paragraphs describe primary seal ring materials, mating ring materials, secondary seals and the elastomeric compounds used to fabricate them, seal hardware, and springs.

4.1 Primary Ring—The primary ring material must be impervious to environmental pressure, display adequate friction/wear properties, have good thermal conductivity and remain stable under the temperature, pressure, and fluid conditions in a compressor.

4.1.1 RESIN BONDED GRAPHITE—Thermoset materials generally made from phenolic resin, graphite, and varying amounts of mineral fillers. Usually processed in the 149 to 315 °C temperature range.

4.1.1.1 Advantages

a. Good wear resistance
b. Readily molded to complex geometry and close tolerances
c. Good inherent low porosity
d. Low cost
e. Good handling characteristics (resists chips and cracks)

4.1.1.2 Disadvantages

a. Fair thermal stability
b. Lower maximum operating temperature
c. Fair thermal conductivity

4.1.2 CARBON-GRAPHITE—A manufactured product of carbon and graphite in a rigid, hard structure produced by firing at high temperatures usually in the range of 900 to 2000 °C. Carbon-graphite is inherently porous after firing and must be made impervious for use as a primary seal ring. Various materials are used to impregnate the carbon-graphite structure.

4.1.2.1 Advantages

a. Excellent temperature resistance and stability
b. Excellent compatibility with refrigerants and associated lubricants (impregnant must be properly chosen)
c. Excellent wear resistance
d. Good thermal conductivity
e. Low thermal distortion coefficient

4.1.2.2 Disadvantages

a. Difficult to mold complex shapes and maintain close tolerances
b. Fair handling characteristics (vulnerable to chips and cracks)
c. High cost

4.2 Mating Ring—The mating ring is usually a dissimilar material, and harder than the primary seal ring. The material choice depends upon operating conditions, configuration, costs, and performance requirements.

4.2.1 CAST IRON—Cast iron is used in various forms and grades. Microstructure and hardness are principal criteria for grade selection.

4.2.1.1 Advantages

a. Low cost for complex shapes
b. Excellent thermal shock resistance
c. Good wear resistance
d. Low thermal distortion coefficient

4.2.1.2 Disadvantages

a. Control of microstructure and casting defects critical to insure good performance
b. Fair corrosion resistance
c. Fair scratch resistance

4.2.2 CERAMIC—Ceramic materials used are hard, dense aluminum oxides (85 to 99% Al_2O_3 by weight) formed by compacting finely ground oxide powders with fluxing agents and inhibitors at high pressure. The formed part is then fired at temperatures of 1400 to 1800 °C. After firing, the part is composed mostly of pure alumina crystals of controlled size.

4.2.2.1 Advantages

a. Excellent wear resistance
b. Excellent dimensional stability
c. Excellent fluid compatibility
d. Low cost for simple shapes
e. Excellent scratch resistance

4.2.2.2 Disadvantages

a. Mechanical and thermal shock susceptibility
b. Difficult to mold complex shapes
c. Low thermal conductivity

4.3 Secondary Seal—Secondary seals are generally made from elastomers principally selected for compatibility with refrigerants and associated lubricants. Other vehicle environmental conditions should also be considered when selecting the secondary seal material.

4.3.1 NEOPRENE COMPOUNDS (CR)—This material's operating temperature range is –40 to 121 °C.

4.3.1.1 Advantages

a. Good processability
b. Fair oil resistance
c. Low cost
d. Low permeability coefficient

4.3.1.2 Disadvantages

a. Fair temperature resistance

4.3.2 NITRILE COMPOUNDS (NBR)—This material's operating temperature range is –40 to 121 °C.

4.3.2.1 Advantages

a. Good processability
b. Good oil resistance
c. Low cost
d. Low permeability coefficient

4.3.2.2 Disadvantages

a. Fair temperature resistance

4.3.3 FLUOROELASTOMER COMPOUNDS (FKM)—This material's operating temperature range is –32 to 204 °C.

4.3.3.1 Advantages

a. Excellent high temperature resistance
b. Excellent oil resistance
c. Low permeability coefficient

4.3.3.2 Disadvantages

a. Poor processability (in terms of shape factor)
b. High cost

4.4 Hardware—Hardware components are usually fabricated from low carbon steel or powdered metal. They are easily formed, readily available, and low in cost. They may require plating or other treatment to achieve in-process corrosion resistance.

4.5 Springs

4.5.1 COIL SPRINGS—Coil springs are compression springs wound from music wire or stainless steel wire. They are low in cost, and may require plating or other treatment to achieve in-process corrosion resistance.

4.5.2 WAVE SPRINGS—Wave springs are fabricated from carbon steel or stainless steel sheet stock. They are low to moderate cost, and may require plating or other treatment to achieve in-process corrosion protection.

5. Application Design Data—This section is to provide guidelines as to specific dimensions and conditions that may functionally affect the passenger car air-conditioning compressor face seal.

5.1 Flatness—Overall flatness of sealing surfaces is critical to maintain a refrigerant oil or gas tight seal.

5.1.1 PRIMARY SEAL RING AND MATING RINGS—Surface flatness for both should be two helium light bands maximum.

5.2 Surface Roughness—Surface roughness is a function of base material, grain size, structure, and method of finishing. Surface roughness is to be evaluated as agreed upon by the supplier and user for specific combinations of materials and for specific applications. See Table 1 for recommended surface finish roughness for seal rings.

TABLE 1—RECOMMENDED SURFACE ROUGHNESS FOR PRIMARY AND MATING SEAL RINGS

Roughness		Resin Bonded Graphite	Carbon Graphite	Cast Iron	Ceramics (Al_2O_3)
µm Ra	MIN	0.025	0.051	0.076	0.10
µm Ra	MAX	0.38	0.63	0.25	0.45

5.3 Waviness—Waviness should be 0.63 µm maximum for both sealing surfaces.

5.4 Shaft Related Reference Dimensions

5.4.1 SQUARENESS—Squareness of the face of the mating ring is to be within 0.05 mm FIM of shaft centerline.

5.4.2 SHAFT RUNOUT—Shaft runout is defined as twice the distance the center of the shaft is displaced from the center of rotation, expressed in FIM, and should not exceed 0.05 mm.

5.4.3 CONCENTRICITY—Concentricity is defined as the radial distance the geometric center of the mating ring is displaced from the geometric center of the shaft along the axis of rotation and should be held to within 0.13 mm FIM.

5.4.4 END PLAY—End Play is defined as total axial shaft movement and is a function of bearing selection and mating component fits. The maximum recommended end play is 0.2 mm.

5.5 Installation—Proper installation of seal components is necessary for the seal assembly to function properly.

5.5.1 LEAD-IN CHAMFER—A lead-in chamfer is required on the seal components, compressor housing bore, and the bearing shaft end for ease of seal installation and prevention of damage to the secondary seals. All corners should be blended smoothly and lubricants may be required for seal component assembly.

5.5.2 INSTALLATION TOOLS—Special seal component installation tools may be required to be used as lead-in chamfer or cover sharp edges (keyway, splines, etc.) to protect secondary seals from damage.

5.6 Proper performance of the primary and secondary sealing elements is dependent on proper contact with their respective sealing surfaces. Exposure of these sealing surfaces to contaminants from improper storage, handling, contaminated assembly lubricant, unclean installation tools or assembly areas may result in premature seal failure. Common sense “good housekeeping” practice is recommended to prevent the sealing surfaces from coming in contact with contaminants.

5.7 Seal Cavity Design—The seal cavity must be designed to provide adequate refrigerant-oil circulation for proper seal cooling and lubrication.

6. Drawing Designation—It is recommended that the standard SAE seal and housing drawing format be used. This format (Figure 1) is a composite of the engineering application, seal, and compressor housing dimensional data that is required to assure functional compatibility of the seal in a specific application. The format is intended as a guide and it is not required that it be followed precisely as shown. It is understood that standard engineering practices, as employed by some users, will not require that this amount of detailed information be shown on the print since it may be recorded elsewhere in their engineering standards. In those cases, it is recommended that the format and/or sketches be suitably altered to meet the user's requirements.

The seal user should only supply that portion of the engineering application and dimensional data that is necessary for the particular product requirements. The seal specification data should be furnished by the seal supplier in conjunction with the user. This data and information must be such that it is compatible with the engineering application data as supplied by the user.

7. Qualification Test—This test is conducted to determine the durability and performance characteristics of an air-conditioning compressor shaft seal in a functional compressor assembly.

7.1 Description of Equipment and Installation—The following equipment and system orientation is recommended (see Figures 3 and 4):

7.1.1 Complete production air-conditioning compressor assembly in which the shaft seal will operate.

7.1.2 Two-speed drive motor rated at 15 hp and 1750/3500 rpm.

7.1.3 Water-cooled condenser rated at 10 kW/h.

7.1.4 Discharge pressure controlled water valve (to control condenser cooling water flow).

7.1.5 Automotive evaporator and blower motor assembly.

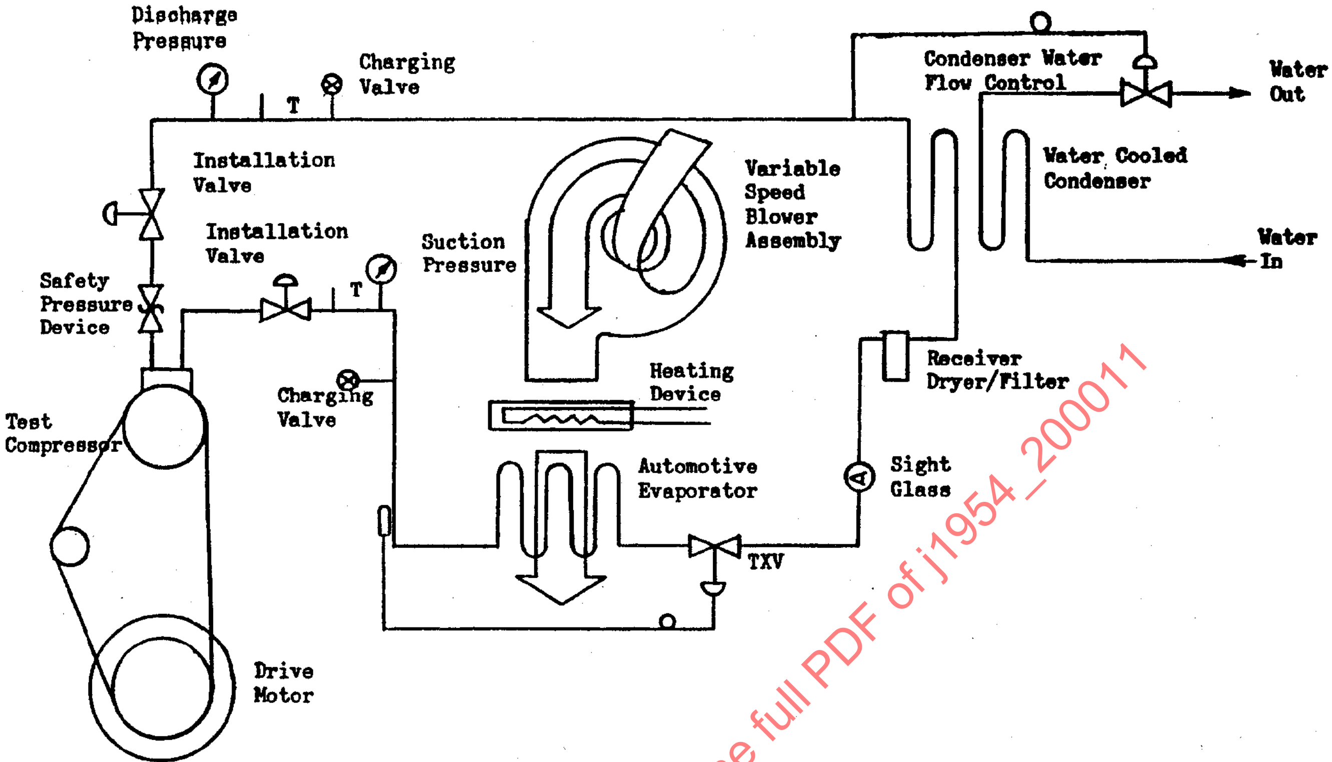

FIGURE 3—AUTOMOTIVE COMPRESSOR TEST LOOP

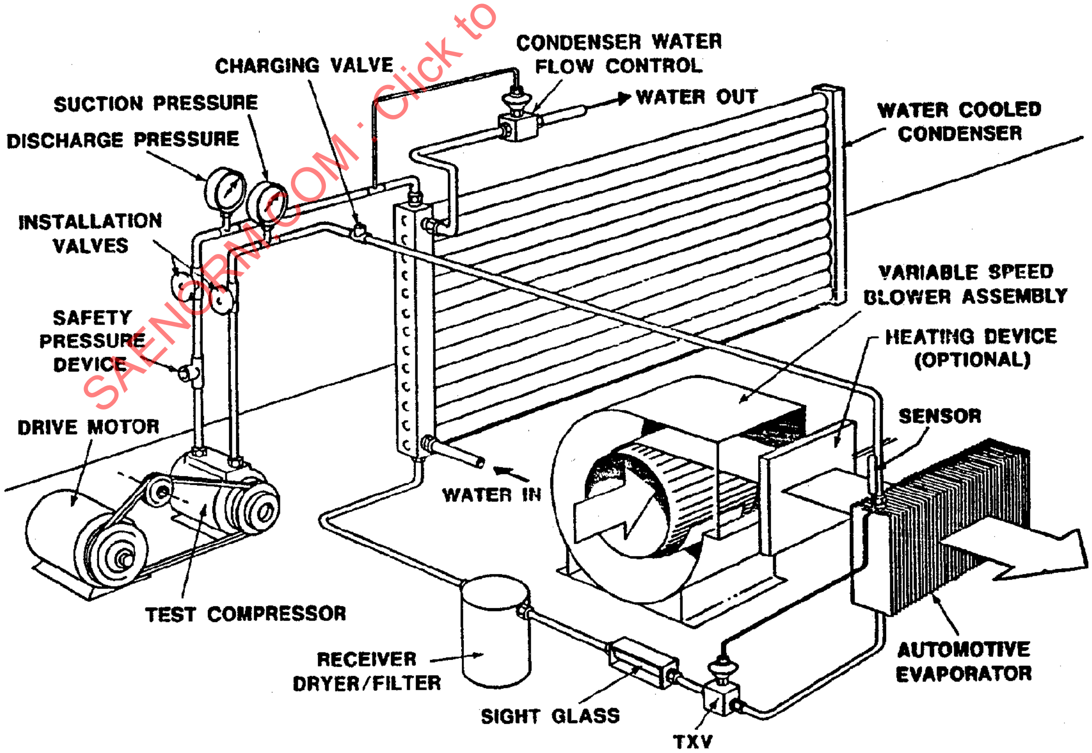

FIGURE 4—AUTOMOTIVE COMPRESSOR TEST LOOP

7.1.6 Thermostatic expansion valve (15°SH).

7.1.7 Valves (to add refrigerant charge).

7.1.8 12.8 V power supply (40 A minimum) for clutch and blower motor assembly.

7.1.9 Cycle counter and running time meter.

7.1.10 Valves for installing and removing the test compressor from the refrigerant loop.

7.1.11 Gauges to measure system pressures.

7.1.12 Thermocouples for measuring system temperatures.

7.1.13 Liquid line sight glass.

7.1.14 Refrigerant hoses and clamps (production parts preferred).

7.1.15 Evacuation pump.

7.1.16 Halogen leak detector capable of measuring 7.1 g/year refrigerant leakage per year and 7.1 g/year calibrated leak master.

7.1.17 Scale accurate to 1.0 mg to determine seal oil weepage.

7.1.18 Scale accurate to 1.77 g, avdp., to determine weight of compressor and oil.

7.1.19 An appropriate pressure relief device must be added to the refrigerant system if not integral to the compressor. All water and refrigerant lines shall have a burst pressure at least two and one half times the operating pressure and conform to all reasonable safe operating practices.

7.1.20 Heating device for evaporator.

7.1.21 Liquid line receiver/dryer and filter.

7.2 Procedure—The following procedural outline is provided as a guide; obviously, this procedure should be modified to be compatible with the user's established standard engineering practice.

7.2.1 INITIAL OPERATING CONDITIONS—The air-conditioning compressor assembly should be prepared for test by the following procedure.

7.2.1.1 Drain the oil from the compressor assembly and record the quantity.

7.2.1.2 Weigh the compressor assembly and record.

7.2.1.3 Replace the oil charge and install the compressor assembly into the refrigerant loop.

7.2.1.4 A new felt of known weight is to be inserted at the external side of the seal assembly to absorb oil weepage.

7.2.1.5 Evacuate the system to a minimum of 50 mm of Mercury vacuum for 30 min. Check for leaks as evidenced by vacuum decay.

7.2.1.6 Belt tension should be set to manufacturer's specifications.

7.2.1.7 Weigh in 0.91 kg of refrigerant to the discharge side service valve.

7.2.1.8 With Halogen leak detector calibrated to 14.18 g/year refrigerant leak rate, check all connections for leaks. Measure and record leakage rate at the shaft seal. See Section 9 for recommended seal cavity leakage apparatus and procedure.

7.2.1.9 Operate compressor at 2300 rpm with suction and discharge pressure as appropriate for the refrigerant. Adjust fan speed and/or heat load to the evaporator assembly to obtain desired suction pressure after the discharge pressure is established by adjusting the water control. DO NOT ADD CHARGE TO DISCHARGE SIDE WHILE THE COMPRESSOR IS OPERATING.

7.2.1.10 Add refrigerant vapor to the system through the suction charging valve until the flow in the sight glass just becomes bubble-free.

7.2.1.11 Repeat step 7.2.1.9 if needed.

7.2.1.12 Operate the compressor at the previously mentioned conditions for 2 h for break-in.

7.2.2 SUBSEQUENT TEST INSTALLATION—Subsequent test compressor installations may be made without discharging the entire refrigerant loop by utilizing the following procedure:

7.2.2.1 Stop the compressor rotation and allow the suction and discharge pressure to equalize.

7.2.2.2 Close the suction and discharge installation valves.

7.2.2.3 Loosen the discharge line connection slightly to allow the refrigerant to bleed off slowly without losing the oil charge.

7.2.2.4 Remove the compressor assembly from the refrigerant loop.

7.2.2.5 Weigh the compressor assembly at end of test and record the weight.

7.2.2.6 Drain the oil from the tested compressor assembly and record the amount.

7.2.2.7 Drain the oil from the new test compressor assembly.

7.2.2.8 Weigh the new compressor assembly and add oil charge to match the amount recorded in step 7.2.2.5.

7.2.2.9 Install the compressor assembly into the refrigerant loop.

7.2.2.10 Evacuate the compressor assembly to a minimum of 50 µm of Mercury vacuum for 30 min. Check for leaks as evidenced by vacuum decay.

7.2.2.11 Open the suction and discharge installation valves.

7.2.2.12 Repeat steps 7.2.1.4 through 7.2.1.6.

7.2.2.13 Skip 7.2.1.7.

7.2.2.14 Repeat steps 7.2.1.8 through 7.2.1.12.

7.2.3 TEST DURATION AND CONDITIONS—The air-conditioning compressor assembly should be run under the following conditions commensurate with the user's standards:

7.2.3.1 Adjust compressor speed to 4600 rpm ± 50 rpm.

7.2.3.2 Discharge pressure should be maintained as appropriate for the refrigerant.

7.2.3.3 Suction pressure should be maintained at as appropriate for the refrigerant.

7.2.3.4 Discharge temperature should be monitored and must not exceed 121 °C.

7.2.3.5 When stable conditions are maintained, cycle the compressor clutch at a 10 s on and 5 s off interval. This cycle is to be repeated for the test duration.

7.2.3.6 At 24 h intervals, discontinue the clutch cycling for a sufficient time to insure that the initial pressure and temperature parameters are maintained.

7.2.3.7 *Test Duration*—500 h.

7.3 Data to be Recorded—To provide meaningful test data, the following minimum data should be recorded:

7.3.1 BEFORE TEST—The following data should be obtained and recorded prior to compressor assembly and testing as a baseline for wear data:

7.3.1.1 *Primary Seal Ring*

a. Material (type and code)
b. Surface roughness (by lot capability)
c. Surface flatness
d. Surface waviness (optional)
e. Face height

7.3.1.2 *Mating Ring*

a. Material (type and code)
b. Surface roughness
c. Surface flatness
d. Surface waviness (optional)
e. Hardness (optional)

7.3.1.3 *Seal and Housing Assembly*

a. Seal head operating length
b. Seal load at operating length (on an oiled shaft)
c. Test schedule, refrigerant, and vacuum leakage rate
d. Shaft speed
e. Squareness of mating ring to shaft
f. Shaft runout
g. Shaft end play

7.3.2 DURING TEST—The following data should be recorded during the test sequence at the stabilized condition:

a. Compressor rpm
b. System pressures
c. System temperatures

7.3.3 AFTER TEST—The following data or observations should be noted and recorded after completion of the test sequence prior to removal of the seal from the compressor assembly:

a. Felt weight
b. Refrigerant leakage rate at seal housing area
c. If the refrigerant leakage observed is out of specification, carefully locate the leak source

7.3.4 AFTER TEST—The following data or observations should be noted and recorded after completion of the test sequence:

7.3.4.1 Seal and Housing Assembly

a. Seal load at operating length
b. Shaft end play
c. Shaft runout
d. Contamination

7.3.4.2 Primary Seal Ring

a. Wear pattern and visual characteristics (blisters, voids, cracks, drive wear, etc.)
b. Surface roughness
c. Surface flatness
d. Surface waviness (optional)
e. Face height
f. Calculate face wear (initial minus final face height measurement)

7.3.4.3 Mating Ring

a. Wear pattern and visual characteristics (thermal distress, carbonized oil, etc.)
b. Surface roughness
c. Surface flatness
d. Surface waviness (optional)

7.3.4.4 Spring

a. Spring load
b. Inspect for wear

7.3.4.5 "O" Ring/Bellows Conditions

a. Time-temperature effect (heat aging)
b. Abrasion
c. Blisters
d. Assembly damage

8. Inspection and Quality Control Data—The following is presented as an inspection guide and outlines general quality control equipment and procedures. These guidelines should be reviewed and modified to be commensurate with the supplier's and user's standard inspection and quality control procedures:

8.1 Concentricity and Squareness Relationships—Concentricity and squareness relationships defined in accordance with referenced dimensions given in Figures 1K, 1L, and 1M are:

a. Concentricity between Bore A and Shaft B (FIM)
b. Squareness between Shaft B and Surface E (FIM)
c. Squareness between Shaft B and Surface F (FIM)
d. Shaft endplay

Measurement of these relationships shall be made with dial indicators having accuracy within 0.0025 mm. The use of precision collets and/or expanding mandrels will greatly facilitate measurement of these relationships.

Measurement of concentricity between A and B and squareness between B and F shall be made with bearing and shaft assembly installed in the compressor housing. Measurement of squareness between B and E may be made independent of other compressor components. See References 5 for recommended limits of concentricity and squareness.

8.2 Seal Load Determinations—Seal load determinations are to be performed on spring scale apparatus having the capability of measuring operating length to within 0.02 mm and loads to within 0.4 N. Seal loads are to be determined by compressing the seal to its solid length, C2, and reading the seal loads in the direction from solid length toward the free length. Repeat the procedure until two consecutive readings are within 2.2 N.

Solid length is defined as the dimension at which the seal load is 2.5 times nominal load specified at the nominal operating length.

Maximum load is defined as the load measured at the minimum operating length, C3. Nominal load is defined as the load measured at nominal operating length, C4. The minimum load is defined as the load measured at the maximum operating length, C5. Recommended load tolerance at nominal operating length is ±20%.

8.3 Operating Length Variation—Operating length variation is defined as dimension (C5-C3) given in Figure 1 (Figures 1A, 1B, 1C, and 1D). Recommended operating length variations are 0.61 mm maximum for Figures 1A, 1B, 1C, and 1D.

NOTE—Dimension C3 should exceed dimension C2 by 0.51 mm minimum. To determine dimension C2, compress the seal from its free length to a point where the load reaches 2.5 times the nominal load specified at the nominal operating length, C4.

8.4 Overall Flatness—It is recommended that overall flatness measurements of reflective primary seal face be made through the use of an optical flat and helium monochromatic light source (half wavelength = 0.3 µm). Optical flat reference surface shall be flat to within 1/10 a helium light band. See 5.1 for flatness recommendations. In the case of nonreflective seal faces, stylus type equipment shall be used to determine flatness. See 8.6.

8.5 Surface Roughness—Surface roughness of seal faces shall be measured with stylus type surface profiling equipment having either linear or circumferential trace capability. The equipment employed shall be in compliance with ANSI B46.1, Section 4, 1978. Stylus radius shall be 0.013 mm and a minimum stroke of 3.2 mm shall be used. Stylus load shall be sufficient to maintain contact with the surface without surface destruction and shall not exceed 2.5 g. Wavelength cutoff shall be 0.76 mm. It is recommended that a data reading be taken at three different locations and averaged. See Table 1 for surface roughness limits of various seal face materials.