INSTALLATION

OPERATION

MAINTENANCE

Technical Systems DIVISION — RAE CORPORATION
POST OFFICE BOX 309 PRYOR, OKLAHOMA 74362 TELEPHONE 918/825-0720

NOTICE:

ALL TSI EQUIPMENT

MUST BE STARTED AND SERVICED

BY QUALIFIED REFRIGERATION AND

AIR CONDITIONING

SERVICE PERSONNEL

TEMPERATURE PRESSURE CHART

Deg. F	R—12	R—22	R—500	R—502	Deg. F	R—12	R—22	R—500	R—502
-100	*27.0	*25.0	-	*23.3	30	28.5	54.9	36.0	65.4
- 95	*26.4	*24.1	-	*22.1	32	30.1	57.5	37.9	68.2
- 90	*25.7	*23.0	-	*20.7	34	31.7	60.1	39.9	71.1
- 85	*25.0	*21.7	-	*19.0	36	33.4	62.8	41.9	74.1
- 80	*24.1	*20.2	-	*17.1	38	35.2	65.6	43.9	77.1
- 75	*23.0	*18.5	-	*15.0	40	37.0	68.5	46.1	80.2
- 70	*21.8	*16.6	-	*12.6	45	41.7	76.0	51.6	88.3
- 65	*20.5	*14.4	-	*10.0	50	46.7	84.0	57.6	96.9
- 60	*19.0	*12.0	-	*7.0	55	52.0	92.6	63.9	106.0
- 55	*17.3	*9.2	-	*3.6	60	57.7	101.6	70.6	115.6
- 50	*15.4	*6.2	-	0.0	65	63.8	111.2	77.8	125.8
- 45	*13.3	*2.7	-	2.1	70	70.2	121.4	85.4	136.6
- 40	*11.0	.05	*7.6	4.3	75	77.0	132.2	93.5	148.0
- 35	*8.4	2.6	*4.6	6.7	80	84.2	143.6	102.0	159.9
- 30	*5.5	4.9	*1.2	9.4	85	91.8	155.7	111.0	172.5
- 28	*4.3	5.9	0.1	10.5	90	99.8	168.4	120.6	185.8
- 26	*1.6	6.9	0.9	11.7	95	108.3	181.8	130.6	199.7
- 24	*0.3	7.9	1.6	13.0	100	117.2	195.9	141.2	214.4
- 22	0.6	9.0	2.4	14.2	105	126.6	210.8	152.4	229.7
- 20	1.3	10.1	3.2	15.5	110	136.4	226.4	164.1	245.8
- 18	2.1	11.3	4.1	16.9	115	146.8	242.7	176.5	262.6
- 16	2.8	12.5	5.0	18.3	120	157.7	259.9	189.4	280.3
- 14	3.7	13.8	5.9	19.7	125	169.1	277.9	203.0	298.7
- 12	4.5	15.1	6.8	21.2	130	181.0	296.8	217.2	318.0
- 10	5.4	16.5	7.8	22.8	135	193.5	316.6	232.1	338.1
- 8	6.3	17.9	8.8	24.4	140	206.6	337.3	247.7	359.1
- 6	7.2	19.3	9.9	26.0	145	220.3	358.9	264.1	381.1
- 4	8.2	20.8	11.0	27.7	150	234.6	381.5	281.1	403.9
- 2	9.2	22.4	12.1	29.4	155	249.5	405.1	298.9	427.8
0	10.2	24.0	13.3	31.2	160	265.1	429.8	317.4	452.6
2	11.2	25.6	14.5	33.1					
4	12.3	27.3	15.7	35.0					
6	13.5	29.1	17.0	37.0					
8	14.6	30.9	18.4	39.0					
10	15.8	32.8	19.7	41.1					
12	17.1	34.7	21.2	43.2					
14	18.4	36.7	22.6	45.5					
16	19.7	38.7	24.1	47.7					
18	21.0	40.9	25.7	50.1					
20	22.4	43.0	27.3	52.5					
22	23.9	45.3	28.9	54.9					
24	25.4	47.6	30.6	57.4					
26	26.9	49.9	32.4	60.0					
28		52.4	34.2	62.7					

*INCHES VACCUM

TABLE OF CONTENTS

INSTALLATION, OPERATION, & MAINTENANCE INSTRUCTIONS
TSI AIR COOLED CONDENSERS, CONDENSING UNITS & PACKAGED CHILLERS

A. RECEIVING AND INSPECTION

1. Immediately upon receiving shipment, equipment should be inspected for evidence of any damage received in transit. If shipping damage has occurred, a claim should be made with the transportation company, and the local TSI representative should be advised of nature of damage.

2. Equipment should be inspected for compliance with original order acknowledgment (equipment model numbers, voltages, etc.).

B. RIGGING

1. Equipment is equipped with lifting eyes or sling points. These should be used to prevent structural damage.

2. Equipment should be lifted in a near level condition to prevent undue stress on structural members.

3. See enclosed diagram for lifting instructions.

C. MOUNTING

1. Equipment should be mounted on a smooth, hard, level surface.

2. Mounting surface should be rigid, and provisions should be made to prevent noise transmission (structural) to surrounding areas.

3. Air cooled equipment should not be installed under low structural overhangs which can cause condenser air recirculation or restriction.

4. Adequate Area (approx. 1 unit width) must be provided around equipment for unrestricted air flow and service. Two units side by side should have a minimum of 1½ unit width between them.

5. Care should be taken to prevent air from other sources from entering condenser if this air is at an elevated temperature.

6. Indoor design equipment must be installed in a protected enclosure.

D. PIPING

1. All piping must be in accordance with applicable local and state codes.

2. Refrigerant piping (split systems) should be designed and installed in accordance with recommended practices as outlined in ARI or ASHRE piping guide. (Engineering Guide #1 for air conditioning systems.)

3. Water piping (Chillers) should be designed and installed to meet application requirements. Provisions must be made to prevent freezing if design ambient temperatures dictate.

4. Compressor shipping hold down bolts (spring mounted compressors) should be loosened until compressor swings free on its isolators before piping is made-up. Line isolators or loops must be used with spring mounted compressors.

D. PIPING (Continued)

5. When piping is completed, a thorough leak test should be performed before evacuation start-up. Do not exceed 150 psig test pressure on low side of system.

E. WIRING

1. All local and state codes must be strictly adhered to and good electrical practices should be followed to achieve the best installation possible.

2. Power wiring to equipment must be adequately sized for minimum ampacity as shown on unit nameplate. A disconnect should be located adjacent to unit for both safety and servicing purposes.

3. Equipment wiring diagram should be examined and thoroughly understood before field wiring connections are made.

4. Power supply should be checked to be certain that supply voltage agrees with equipment nameplate. Serious damage to compressors and motors can occur if improper voltage is applied.

5. When wiring is completed, fan motors should be checked for proper rotation. All fan motors of multiple fan equipment have been factory wired to operate with same rotation. If rotation is found to be incorrect, reverse two of three leads on main incoming power.

F. SYSTEM EVACUATION

1. With refrigerant piping completed and leak tested, equipment is ready to evacuate. Do not use compressor to evacuate system. A quality vacuum pump capable of 350 micron vacuum is necessary for adequate and dependable system vacuum. Moisture in a re-

TROUBLE SHOOTING CHART

PROBLEM	POSSIBLE CAUSES	POSSIBLE CORRECTIVE STEPS
Low Suction Pressure (Continued)	f) Condensing temperature too low g) Compressor will not unload h) Insufficient water flow	f) Check means for regulating condensing temperature g) See Corrective Steps for failure of compressor to unload h) Adjust gpm.
Compressor will not unload or load up	a) Defective capacity control b) Unloader mechanism defective c) Faulty thermostat stage or broken capillary tube d) Stages not set for application	a) Replace b) Replace c) Replace d) Reset thermostat setting to fit application
Compressor Loading-Unloading Intervals too short	a) Erratic water thermostat b) Insufficient water flow	a) Replace b) Adjust gpm.
Little or no oil pressure	a) Clogged suction oil strainer b) Excessive liquid in crankcase c) Oil pressure gauge defective d) Low-oil pressure safety switch defective e) Worn oil pump f) Oil pump reversing gear stuck in wrong position g) Worn bearings h) Low oil level i) Loose fitting on oil lines j) Pump housing gasket leaks k) Flooding of refrigerant into crankcase	a) Clean b) Check crankcase heater. Reset expansion valve for higher superheat. Check liquid line solenoid valve operation c) Repair or replace. Keep valve closed except when taking reading d) Replace e) Replace f) Reverse direction of compressor rotation g) Replace compressor h) Add oil i) Check and tighten system j) Replace gasket k) Adjust thermal expansion valve
Compressor loses oil	a) Lack of refrigerant *b) Velocity in risers too low *c) Oil trapped in line d) Excessive compression ring blow-by	a) Check for leaks and repair. Add refrigerant b) Check riser sizes c) Check pitch of lines and refrigerant velocities d) Replace compressor
Motor overload relays or circuit breakers open	a) Low voltage during high load conditions b) Defective or grounded wiring in motor or power circuits c) Loose power wiring d) High condensing temperature e) Power line fault causing unbalanced voltage f) High ambient temperature around the overload relay g) Failure of second starter to pull in on part-winding start system	a) Check supply voltage for excessive line drop b) Replace compressor-motor c) Check all connections and tighten d) See Corrective Steps for high discharge pressure e) Check supply voltage. Notify power company. Do not start until fault is corrected f) Provide ventilation to reduce heat g) Repair or replace starter or time delay mechanism
Compressor thermal protector switch open	a) Operating beyond design conditions b) Discharge valve partially shut c) Blown valve plate gasket	a) Add facilities so that conditions are within allowable limits b) Open valve c) Replace gasket
Freeze protection opens	a) Thermostat set too low b) Low water flow c) Low suction pressure	a) Reset to 40 F or above b) Adjust gpm. c) See "Low Suction Pressure"

*Remote condenser models

TROUBLE SHOOTING CHART

PROBLEM	POSSIBLE CAUSES	POSSIBLE CORRECTIVE STEPS
Compressor will not run	a) Main switch open Circuit breakers open b) Fuse blown c) Thermal overloads tripped or fuses blown d) Defective contactor or coil e) System shut down by safety devices f) No cooling required g) Liquid line solenoid will not open h) Motor electrical trouble i) Loose wiring	a) Close switch b) Check electrical circuits and motor winding for shorts or grounds. Investigate for possible overloading. Replace fuse or reset breakers after fault is corrected. c) Overloads are auto. reset. Check unit closely when unit comes back on line. d) Repair or replace e) Determine type and cause of shut-down and correct it before resetting safety switch. f) None. Wait until unit calls for cooling. g) Repair or replace coil h) Check motor for opens, short circuit, or burnout i) Check all wire junctions. Tighten all terminal screws
Compressor noisy or vibrating	a) Flooding of refrigerant into crankcase *b) Improper piping support on discharge or liquid line c) Worn compressor	a) Check setting of expansion valve b) Relocate, add, or remove hangers c) Replace
High Discharge Pressure	a) Condenser water insufficient or temperature too high b) Fouled condenser tubes (water-cooled condenser). Clogged spray nozzles (evaporative condenser). Dirty tube and fin surface (air cooled condenser) c) Non-condensibles in system *d) System overcharged with refrigerant e) Discharge shut off valve partially closed *f) Condenser undersized *g) High ambient conditions	a) Readjust water regulating valve. Investigate ways to increase water supply b) Clean c) Purge the non-condensibles d) Remove excess e) Open valve f) Check condenser rating tables against the operation g) Check condenser rating tables against the operation
Low Discharge Pressure	a) Faulty condenser temperature regulation b) Suction shut-off valve partially closed c) Insufficient refrigerant in system d) Low suction pressure e) Compressor operating unloaded *f) Condenser too large *g) Low ambient conditions	a) Check condenser control operation b) Open valve c) Check for leaks. Repair and add charge d) See Corrective Steps for low suction pressure below e) See Corrective Steps for failure of compressor to load up below f) Check condenser rating table against the operation g) Check condenser rating tables against the operation
High Suction Pressure	a) Excessive load b) Expansion valve overfeeding c) Compressor unloaders open	a) Reduce load or add additional equipment b) Check remote bulb. Regulate superheat c) See Corrective Steps below for failure of compressor to load up
Low Suction Pressure	a) Lack of refrigerant b) Evaporator dirty c) Clogged liquid line filter-drier d) Clogged suction line or compressor suction gas strainers e) Expansion valve malfunctioning	a) Check for leaks. Repair and add charge b) Clean chemically c) Replace cartridge(s) d) Clean strainers e) Check and reset for proper superheat Replace if necessary

*Remote condenser models

F. SYSTEM EVACUATION (Continued)

frigeration system can cause corrosion, expansion valve freeze-up and oil sludge.

2. Attach vacuum pump to both high and low side of system through compressor service valves and evacuate to 350 microns (all service valves*, hand valves, and solenoids must be open during evacuation). It is suggested that vacuum pump be run for a period of time after vacuum has been reached.

***Service valves are back seating valves and must be in mid-position to open to both sides of system.**

G. SYSTEM CHARGING (LESS FLOODED HEAD PRESSURE CONTROL)

With system wired, piped, and evacuated, unit is ready for refrigerant charging. All charging lines and manifolds must be purged with refrigerant vapor prior to admitting refrigerant into system to prevent contaminating system with noncondensibles.

1. Connect charging line to receiver outlet valve and admit "liquid" refrigerant into high side of system until flow stops due to pressure equalization between high side and drum pressure. Backseat outlet valve and disconnect charging line.

2. Connect charging line to compressor suction service valve and admit "vapor" into low side of system.

3. Energize equipment and continue to admit vapor into low side of system until liquid line sight glass clears, indicating a fully charged system (it may be necessary to defeat low pressure control on initial start to prevent nuisance trip until low side pressure is above cut out point of control).

H. SYSTEM CHARGING (WITH FLOODED HEAD PRESSURE CONTROL)

1. Initial charging is the same as outlined in Item G.

2. Do not adjust flood control valves. These are either factory preset or nonadjustable, depending on size.

3. Calculate required charge in pounds for the minimum ambient at which equipment is to operate (see catalog sheets TAC-279-8 Condenser, RCU 118-2 Refrigeration Condensing Unit, CU478-2 Air Cooled Condensing Unit, WC78-2 Chiller).

4. Set unloaders (if supplied) to load compressors to 100% while charging. Add additional charge to bring total charge up to required charge as calculated in 3 above. Additional charge to be added through low side (vapor) as outlined in Section G, Item 3.

I. START-UP

This is a continuation of "system charging" and must be performed before equipment can be left operating and unattended. This will involve checking and adjusting of all safety and operating controls (pressure and temperature controls have been set at the factory, however; it is still desirable to confirm that settings are correct and controls function properly). Do not attempt to function safety controls without some means of stopping compressor in event of extreme high or low pressure conditions that could damage the equipment. If controls fail to function at set points, determine cause and correct. Jumpering any safety control other than for testing purposes is dangerous to personnel and equipment, and nullifies equipment warranty.

1. High pressure control - connect a gauge to the compressor discharge service valve. Stop condenser air flow by stopping fans on air cooled equipment. Control should open immediately when discharge pressure reaches control set point.

2. Low pressure (Pump-down) Control - connect a gauge to the compressor suction service valve. Throttle receiver outlet valve to lower suction pressure at compressor. Compressor should pump-down and be de-energized when suction pressure reaches "cut-out" setting of control. Open receiver outlet valve and observe rise in pressure at compressor suction connection. Compressor should be energized when pressure reaches "cut-in" setting of control.

3. Oil pressure control - Jumper between terminals T1 and T2. Compressor should run approximately 90 seconds and cycle off. Remove jumper and reset control. Check operating oil pressure. This is the differential between oil pump discharge pressure and suction pressure and should be a minimum of 10 psig. Also, after several hours running time, check oil level in compressor. Proper level is approximately ¼ level on oil sight glass.

4. Temperature control (Air Cooled Water Chillers - This control is the main unit operating thermostat. The control cycles both the compressor and the unloader(s) in response to return water temperature. The sensing bulb is located in a well on the return water nozzle. This control has been factory set to maintain 45 ° F. leaving water temperature (based on

I. START-UP (Continued)

a 10° F. TD Design). The temperature control can be field adjusted. To calculate the proper set point, add the specified leaving water temperature and the design temperature difference of return to supply (TD). Temperature control adjustment should be made as follows:

EXAMPLE: Design conditions - 10° F. TD, 55 ° F. entering (return) water, 45° F. leaving (supply) water and 5 psi water side pressure drop at rated flow.

Step 1: Check proper flow thru cooler. Pressure gauges at the entering and leaving nozzles should be used. Adjust flow by balancing valves or throttling valves on discharge side of the cooler to corresponding pressure drop.

Step 2: The system water temperature should be at 60° F. or above to simulate a pull down for proper start-up and check out.

Step 3: Adjust temperature control dial to 55° F. This will produce 45° F. leaving water temperature at designed flow rate.

Step 4: Observe the suction pressure as the return water temperature continues to fall and the compressor(s) and/or unloader(s) are staged by the temperature control. The suction pressure at the compressor should not fall below approximately 58 psig during all stages of compressor/compressors operation.

The compressor(s) will eventually be staged off by the temperature control. When this occurs, the return water temperature must rise to the cut in set point of the temperature control (the dial setting) before the compressor(s) is staged on again.

Always consult equipment submittals for proper design conditions before adjusting the temperature control.

5. Freeze protection control - Control is a pressure sensing, manual reset safety control. It responds to suction pressure and prevents circuit operation should suction pressure fall below control set point for a period in excess of 120 seconds. Control is factory set and sealed at a pressure selected for the fluid being cooled or application requirements. For water systems, the control is set at 54 psig. The fixed time delay (120 seconds) allows circuit to stabilize on start-up and normal pump-down operation. Control is sealed at the factory. This seal must be intact to maintain warranty on compressor and/or cooler failures. Should readjustment be necessary, contact the factory for authorization.

6. Thermal expansion valve - Check superheat setting and adjust if necessary. Normal superheat at compressor should be 8 to 12° F. (Suction Line Temperature at compressor minus saturated suction temperature corresponding to suction pressure at compressor.)

7. Hot Gas By-Pass (if supplied) - Connect gauge to compressor suction service valve. Throttle receiver outlet valve to lower suction pressure at compressor. Hot gas regulator should begin to open as suction pressure approaches design suction pressure. This should be done before unit has pulled down to design conditions (Engineering Guide #2 "Hot Gas Bypass").

J. SHUT DOWN

Equipment which will not be required to operate for a period of time should be secured by storing refrigerant charge in the receiver or condenser.

1. Front seat the receiver outlet valve. Set thermostat at a setting below system temperature to insure that liquid line solenoid is

Possible causes of control failures and recommended corrective action:

CAUSE	CORRECTION
A. Temperature Control Setting too low	Readjust to proper setting. Check catalog or submittal data.
B. Cooler flow incorrect	Balance flow to catalog or submittal requirements
C. Low refrigerant charge	Recharge to correct operating level - clear sight glass.
D. Restricted liquid line	Check valves and dryer
E. Thermal expansion valve adjustment	Adjust superheat to approximately 8 - 12° F. at the compressor.

EFFECTS OF UNBALANCED VOLTAGE ON MOTOR PERFORMANCE

Alternating-current polyphase motors shall operate successfully under running conditions at rated load when the voltage unbalance at the motor terminals does not exceed 1 percent. Performance will not necessarily be the same as when the motor is operating with a balanced voltage at the motor terminals.

A relatively small unbalance in voltage will cause a considerable increase in temperature rise. In the phase with the highest current, the percentage increase in temperature rise will be approximately two times the square of the percentage voltage unbalance. The increase in losses, and, consequently, the increase in average heating of the whole winding will be slightly lower than the winding with the highest current.

If nuisance trip outs or repeated trip outs of a motor are experienced and diagnosis of the motor shows no faults, phase unbalance is a likely cause.

To illustrate the severity of this condition, an approximate 3.5 percent voltage unbalance will cause an approximate 25 percent increase in temperature rise.

The percent of voltage unbalance is equal to 100 times the maximum voltage deviation from the average voltage divided by the average voltage.

EXAMPLE:

220 Volt Circuit
Phase 1, 2 = 217 V
2, 3 = 221V
3, 1 = 228V

$$\frac{217 + 221 + 228}{3} = 222V$$

$$228 - 222 = 6V$$

$$\frac{100 \times 6}{222} = 2.7\% \text{ Voltage Unbalance}$$

Percent Voltage Unbalance	Percent Current Rise	Percent Temperature Rise
3.5	29.4	25.0
3.0	25.2	18.0
2.5	21.0	12.5
2.0	16.75	8.0
1.5	12.5	4.5
1.0	8.0	2.0
0.5	3.8	0.5
0.0	0.0	0.0

REFERENCE NEMA STANDARDS MGI-12.45A and 14.34C-January, 1974

— NOTES —

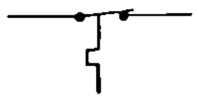
Wiring Diagram Symbols

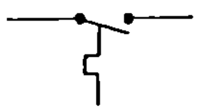

Time delay S.W.


Pressure Activated S.W.
Breaks on rise


Pressure activated S.W.
Makes on rise

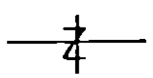

Flow Activated S.W.


Temperature Activated S.W.
Breaks on Rise

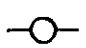

Temperature Activated S.W.
Makes on Rise

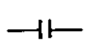

Terminal Block

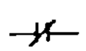

Relay


Solenoid

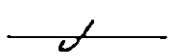

Wire NO's


Contactor or
Starter Coil


N.O. Contacts for Relay, Contactor, or Starter


N.C. Contacts for Relay, Contactor, or Starter


Fuse


Jumper

Factory Wiring

Field Wiring

Enclosure Perimeters

J. SHUT DOWN (Continued)

energized. Defeat the low pressure control and allow unit to pump down to a suction pressure of approximately 5 psig. It may be necessary to repeat pump-down as some refrigerant will remain in oil and will slowly boil off. When suction pressure holds at 5 psig, front seat suction service valve. Lock disconnect in off position. On chillers, water flow must be maintained during this procedure.

2. On units with coolers or water cooled condensers, special precautions must be taken to completely drain the vessels to prevent freezing if ambient should be below 32° F.

3. Inspect system for possible worn or faulty components and repair if required.

K. SYSTEM RESTART AFTER SHUT DOWN

1. Thorough leak test should be performed.

2. Coil(s) should be checked for dirt accumulation or obstruction and cleaned if necessary.

3. Refill water system and purge all air from system.

4. Energize crankcase heaters and allow a minimum of 12 hours operation before compressor restart.

5. Install gauges, start system and check for correct refrigerant charge, and proper system operation and balance.

L. MAINTENANCE

1. The system should be checked periodically. Use only the services of a qualified refrigeration mechanic for inspection and maintenance checks or service operations.

2. Water Treatment (Chillers and Water Cooled Condensing Units) - The water should be tested by a local testing agency and their recommendations adhered to.

3. Air Cooled Condenser - The inlet side of the condenser coil must be kept clean through a regular preventive maintenance program.

4. Compressor Oil Level - The compressor oil level should be checked periodically. If oil is needed, allow equipment to pump down to approximately 5 psig crankcase pressure. Place disconnect in off position and close suction and discharge service valves. Slowly remove oil fill plug from compressor crankcase. (Slight pressure is still in crankcase and care must be taken to prevent "blow out" of plug.) Add clean, dry oil (Suniso 3GS or equal) until oil is approximately mid-point in oil sight glass. Replace oil fill plug and open discharge and suction service valves. Restart compressor and check oil level after two hours of operation. (Loss of oil would suggest that a leak may be in system. Carefully inspect entire system for evidence of oil and repair as necessary.)

5. Periodically check all electrical connections for possible loose or corroded terminals. Repair as necessary.

IN WARRANTY REPLACEMENT PARTS

Should you require replacement parts for TSI equipment, they may be obtained through a manufacturer's representative or directly from the factory.

When requesting in warranty replacement parts, it is necessary to furnish the equipment model and serial number, TSI part number if possible, part description, and shipping information. Upon determination that the defective part is in warranty, a return authorization expires after 30 days and returned parts will no longer be accepted.

The Warranty Registration Card and the TSI Start-Up Report must both be completed and returned to the factory to validate the warranty.

Should you have any questions, call your representative or the factory.

BASIC WARRANTY

TECHNICAL SYSTEMS INCORPORATED (TSI) MAKES NO WARRANTY OF MERCHANTABILITY AND NO WARRANTY OF FITNESS FOR ANY PARTICULAR PURPOSE, NOR DOES IT MAKE ANY WARRANTY, EXPRESS OR IMPLIED, OF ANY NATURE WHATSOEVER WITH RESPECT TO PRODUCTS SOLD BY TSI OR THE USE THEREOF EXCEPT AS IS SPECIFICALLY SET FORTH ON THE FACE HEREOF, EVEN THOUGH IT MAY HAVE BEEN NEGLIGENT. TSI SHALL IN NO EVENT BE LIABLE FOR DIRECT, INDIRECT, SPECIAL, INCIDENTAL, CONSEQUENTIAL OR PENAL DAMAGES. TSI MAKES NO WARRANTY OF ANY KIND, EITHER EXPRESS OR IMPLIED, TO 'CONSUMERS' AS THAT TERM IS DEFINED IN SEC. 101 OF PUBLIC LAW 93-637, THE MAGNUSON-MOSS WARRANTY-FEDERAL TRADE COMMISSION IMPROVEMENT ACT.

Technical Systems Incorporated (TSI) warrants to the original Purchaser-User that products manufactured by TSI shall be free from defects in material and workmanship under normal use and service for a period of eighteen months from date of shipment from TSI plant or twelve months from date of start-up, whichever period first expires.

The obligation of TSI under this warranty is limited to TSI repairing or replacing, free of cost to Purchaser-User, F.O.B. factory, any part or parts that in the judgment of TSI show evidence of defect provided that upon TSI authorization the said part or parts be returned to TSI, transportation prepaid, for inspection and judgment. Under this warranty TSI assumes no responsibility for the expense of labor or materials necessary to remove a defective part or install repaired or new parts.

This warranty is issued only to the original Purchaser-User, is not transferable, applies only to a unit installed within the United States of America, its territories or possessions and Canada and is in lieu of all other warranties expressed or implied. TSI neither assumes or authorizes any other person to assume for TSI any liabilities not herein stated.

TSI shall not be liable for any damage or delays occurring in transit, for any default or delays in performance caused by any contingency beyond its control including war, government restrictions or restraints, strikes, short or reduced supply of raw materials, fire, flood or other acts of God, nor for damage or loss of any products, refrigerant, property, loss of income or profit due to malfunctioning of said unit.

THE FOREGOING IS IN LIEU OF ALL OTHER WARRANTIES, EXPRESS OR IMPLIED, NOTWITHSTANDING THE PROVISION OF THE UNIFORM COMMERCIAL CODE, THE MAGNUSON-MOSS WARRANTY-FEDERAL TRADE COMMISSION IMPROVEMENT ACT, OR ANY OTHER STATUTORY OR COMMON LAW, FEDERAL OR STATE.

return of the liquid to the compressor at a rate which the compressor can safely tolerate.

Flooding typically can occur on heat pumps at the time the cycle is switched from cooling to heating, or from heating to cooling, and a **suction accumulator is mandatory on all heat pumps unless otherwise approved by the Copeland Application Engineering Department.**

Systems utilizing hot gas defrost are also subject to liquid flooding either at the start or termination of the hot gas cycle. Compressors on low superheat applications such as liquid chillers and low temperature display cases are susceptible to occasional flooding from improper refrigerant control. Truck applications experience extreme flooding conditions at start-up after long non-operating periods.

On two stage compressors the suction vapor is returned directly to the low stage cylinder without passing through the motor chamber, and **a suction accumulator should be used to protect the compressor valves from liquid slugging.**

Since each system will vary with respect to the total refrigerant charge and the method of refrigerant control, the actual need for an accumulator and the size required is to a large extent dictated by the individual system requirement. If flooding can occur, an accumulator must be provided with sufficient capacity to hold the maximum amount of refrigerant flooding which can occur at any one time, and this can be well over 50% of the total system charge in some cases. If accurate test data as to the amount of liquid floodback is not available, then 50% of the system charge normally can be used as a conservative design guide.

5. Oil Separators

Oil separators cannot cure oil return problems caused by system design, nor can they remedy liquid refrigerant control problems. However, in the event that system control problems cannot be remedied by other means, oil separators may be helpful in reducing the amount of oil circulated through the system, and can often make possible safe operation through critical periods until such time as system control can be returned to normal conditions. For example, on ultra low temperature applications or on flooded evaporators, oil return may be dependent on defrost periods, and the oil separator can help to maintain the oil level in the compressor during the period between defrosts.

WARNING – Extreme care must be taken in starting compressors for the first time after charging. At this time all of the oil and most of the refrigerant might be in the compressor, creating a condition which could cause compressor damage due to slugging. Directing a 500 watt heat lamp on the lower shell of the compressor for approximately fifteen minutes should be beneficial in eliminating this condition which might never reoccur.

1. **Minimize Refrigerant Charge**

The best compressor protection against all forms of liquid refrigerant problems is to keep the charge within the compressor limits listed in Table 1. Even if this is not possible, the charge should be kept as low as reasonably possible.

Use the smallest practical size tubing in condensers, evaporators, and connecting lines. Receivers should be as small as possible.

Charge with the minimum amount of refrigerant required for proper operation. Beware of bubbles showing in the sight glass caused by small liquid lines and low head pressures. This can lead to serious overcharging.

2. **Pumpdown Cycle**

The most positive and dependable means of properly controlling liquid refrigerant, particularly if the charge is large, is by means of a pumpdown cycle. By closing a liquid line solenoid valve, the refrigerant can be pumped into the condenser and receiver, and the compressor operation controlled by means of a low pressure control. The refrigerant can thus be isolated during periods when the compressor is not in operation, and migration to the compressor crankcase is prevented. A recycling type of pumpdown control is recommended to provide protection against possible refrigerant leakage through control devices during the off cycle. With the so-called one time pumpdown, or non-recycling type of control, sufficient leakage may occur during long off periods to endanger the compressor.

Although the pumpdown cycle is the best possible protection against migration, it will not protect against liquid flooding during operation.

If a pumpdown cycle is used, an approved start capacitor and relay must be used on all compressors having single phase PSC motors.

3. **Crankcase Heaters**

On some systems, operating requirements, cost, or customer preference may make the use of a pumpdown cycle undesirable, and crankcase heaters are frequently used to retard migration.

The function of a crankcase heater is to hold the oil in the compressor at a temperature higher than the coldest part of the system. Refrigerant entering the crankcase will then be vaporized and driven back into the suction line. However in order to avoid overheating and carbonizing of the oil, the wattage input of the crankcase heater must be limited, and in ambient temperatures approaching 0°F., or when exposed suction lines and cold winds impose an added load, the crankcase heater may be overpowered, and migration can still occur.

In no event should a compressor be started unless the crankcase heater has been energized for a period of at least 4 hours immediately prior thereto.

Crankcase heaters are effective in combating migration if conditions are not too severe, **but they will not remedy slugging or liquid floodback.**

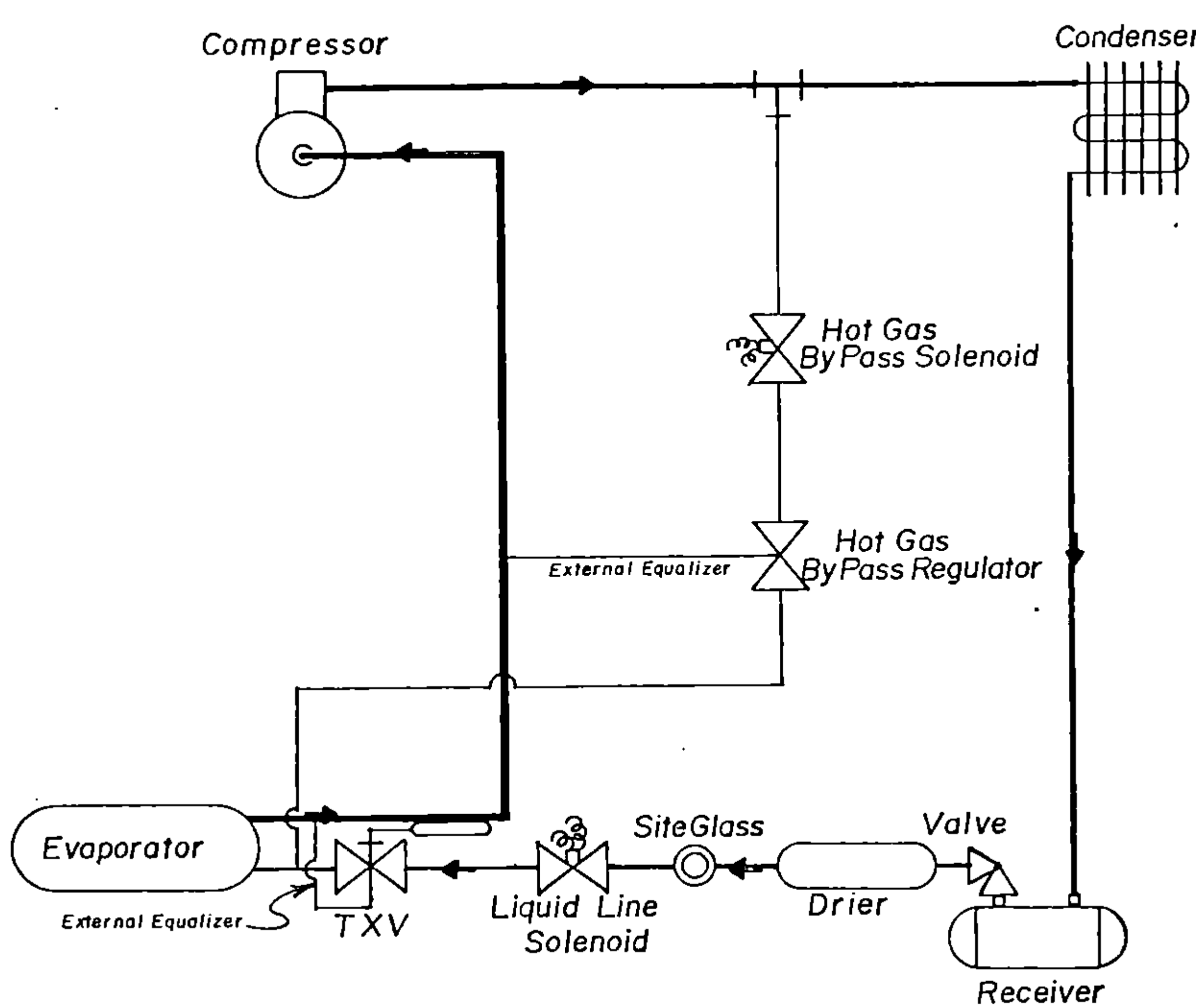
Crankcase heater elements will burn out if overheated, and historically crankcase heaters have not had a good reliability record in the field. Recent studies indicate that over voltage may be a major cause of crankcase heater failure. Traditionally, heaters have been specified at nominal 220 volt and 440 volt conditions, but modern transmission systems may operate at 10 to 15% above those voltages. Motors are designed to withstand wide swings in supply voltage without damage, but high voltage on the resistance element in a crankcase heater may cause excessive current, power input, and temperature.

In order to improve heater reliability, Copeland specifications for crankcase heaters have been changed to nominal voltage ratings of 120, 240, 480, and 600. These coincide with the standardized nominal service voltage specified by ARI and NEMA. The heaters listed in the following tables reflect these new voltage ratings.

4. **Suction Accumulators**

On systems where liquid flooding is apt to occur, a suction accumulator should be installed in the suction line. Basically the accumulator is a vessel which serves as a temporary storage container for liquid refrigerant which has flooded through the system, with a provision for metered

Hot Gas Bypass System



HOT GAS BYPASS

BYPASS EVAPORATOR INLET WITH DISTRIBUTOR

This method of application provides distinct advantages over the other methods, especially for unitary or field built-up units where the high and low side are close coupled. It is also applicable on systems with remote condensing units, especially when the evaporator is located below the condensing unit. The primary advantage of this method is that the system thermostatic expansion valve will respond to the increased superheat of the vapor leaving the evaporator and will provide the liquid required for desuperheating. Also the evaporator serves as an excellent mixing chamber for the bypassed hot gas and the liquid-vapor mixture from the expansion valve. This insures a dry vapor reaching the compressor suction. Oil return from the evaporator is also improved since the velocity in the evaporator is kept high by the hot gas.

VALVE/EQUIPMENT LOCATION & PIPING

When the evaporator is located below the condensing unit on a **remote** system, bypass to the evaporator inlet is still the best method of hot gas bypass to insure good oil return to the compressor. When this is done, the bypass valve and hot gas solenoid valve must be located at the condensing unit rather than at the evaporator section. This will insure obtaining rated capacity by the bypass valve at the conditions it was selected for. If the evaporator is above or on the same level as the condensing unit, this valve location will also eliminate the possibility of hot gas condensing in the long bypass line and running back into the compressor during the off-cycle.

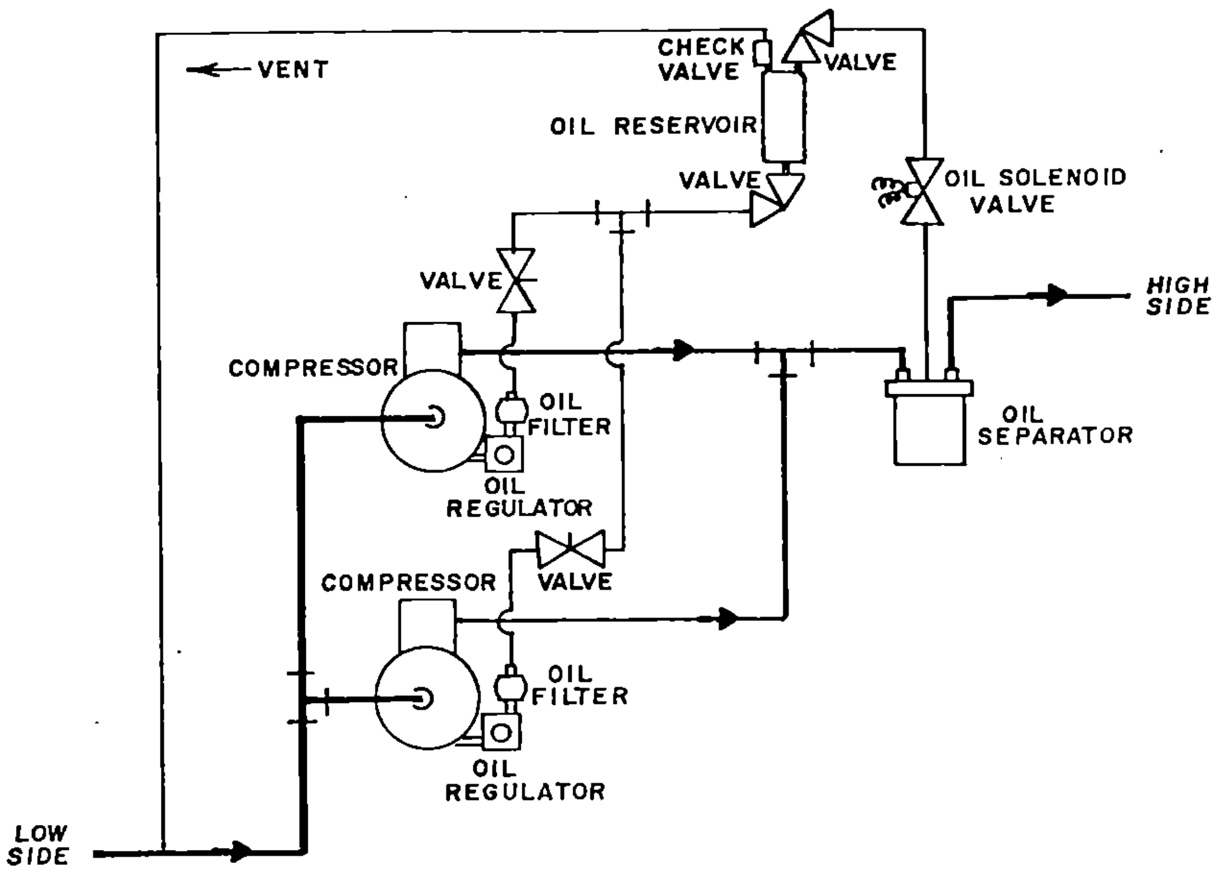
EXTERNALLY EQUALIZED BYPASS VALVES

Since the primary function of the H.G.B.P. is to maintain suction pressure, the compressor suction pressure is the **control** pressure and must be exerted on the underside of the valve diaphragm. When the H.G.B.P. is applied where there is an appreciable pressure drop between the valve outlet and the compressor suction, the externally equalized valve must be used. This is true because when the valve opens, a sudden rise in pressure occurs at the valve outlet. This creates a false **control** pressure, which would cause the internally equalized valve to close.

CAUTION:

Use only the approved side port distributor or adapter to make the hot gas connection.

Oil Control System



OIL CONTROL SYSTEM — CM SERIES UNITS

The CM Series condensing units and chillers are designed having two compressors on a common circuit. When using this method, it is necessary that oil equalization between each compressor be maintained to prevent loss of oil from either compressor.

The method of crankcase equalization used by TSI is the AC&R System consisting of an oil separator mounted in the common discharge line, an oil reservoir, and an oil regulating valve mounted on each compressor crankcase. In addition, a check valve having a 20 pound differential is installed in the vent line from the reservoir to the suction line.

With either or both compressors running, the oil separator will collect the oil vapor leaving the compressor(s) and return this oil to the reservoir. The reservoir will be at a pressure approximately 20 pounds above the compressor crankcase pressure. Oil from the reservoir is piped to the oil regulators which are mounted on the compressor crankcase. As crankcase oil level drops, the regulator will admit oil to the crankcase to maintain the proper operating level.

This method is approved by Copeland Corporation in accordance with Copeland Application Engineeering Bulletin No. AE-1235 dated 7-1-73.

HOW IT WORKS

A reserve of oil is necessary for the operation of the OIL CONTROL SYSTEM. The OIL RESERVOIR is the holding vessel for this stand-by oil. It has two sight glass ports on the shell to observe the oil level inside the vessel. Oil is fed into the OIL RESERVOIR by the OIL SEPARATOR.

It can be realized that high pressure from the high side returns with the oil from the OIL SEPARATOR to the RESERVOIR. In a period of time, enough pressure could build up to adversely affect the float and needle assembly in the OIL LEVEL REGULATOR. For protection, a vent line is installed from the top of the OIL RESERVOIR (a fitting is provided) back to the low pressure suction line. This line permits the pressure in the OIL RESERVOIR to be approximately the same as the pressure in the suction line and the crankcases of the compressors. Oil in the OIL RESERVOIR feeds down through 3/8" and 1/4" OD tubing and keeps the OIL LEVEL REGULATORS supplied with oil.

A REGULATING VALVE is mounted on the suction line vent connection on top of the OIL RESERVOIR and will maintain 20 lbs. pressure differential over the crankcase. This positive pressure will keep the oil line to the OIL LEVEL REGULATORS filled and ready.

The valve on the top of the OIL RESERVOIR automatically receives oil from the OIL SEPARATOR (open position). To add oil to the OIL RESERVOIR manually, close the valve and fill the OIL RESERVOIR through the 1/4" flare connection on the side of the valve. Open valve after filling.

Slugging can result in broken valves, blown head gaskets, broken connecting rods, and other major compressor damage.

Slugging frequently occurs on start-up when liquid refrigerant has migrated to the crankcase. On some units, because of the piping configuration or the location of components, liquid refrigerant can collect in the suction line or evaporator during the off cycle, returning to the compressor as solid liquid with extreme velocity on start-up. The velocity and weight of the liquid slug may be of sufficient magnitude to override any internal anti-slug protective devices of the compressor.

4. Tripping of Oil Pressure Safety Control

One of the most common field complaints arising from a liquid flooding condition is that of a trip of the oil pressure safety control after a defrost period on a low temperature unit. The system design on many units allows refrigerant to condense in the evaporator and suction line during the defrost period, and on start-up this refrigerant floods back to the compressor crankcase, causing a loss of oil pressure and recurring trips of the oil pressure safety control.

One trip or a few trips of the oil pressure safety control may not result in serious damage to the compressor, but repeated short periods of operation without proper lubrication are almost certain to result in ultimate compressor failure. Trips of the oil pressure safety control under such circumstances are frequently viewed by the serviceman as nuisance trips, but it cannot be stressed too strongly that they are warning trips, indicating the compressor has been running without oil pressure for 2 minutes, and that prompt remedial action is required.

RECOMMENDED CORRECTIVE ACTION

The potential hazard to a refrigeration or air conditioning system is in almost direct proportion to the size of the refrigerant charge. It is difficult to determine the maximum safe refrigerant charge of any system without actually testing the system with its compressor and other major components. The compressor manufacturer can determine the maximum amount of liquid the compressor will tolerate in the crankcase without endangering the working parts, but has no way of knowing how much of the total system charge will actually be in the compressor under the most extreme conditions. The maximum amount of liquid a compressor can tolerate depends on its design, internal volume, and oil charge. Where liquid migration, flooding, or slugging can occur, corrective action should be taken, the type normally being dictated by the system design and the type of liquid problem.

Table 1 lists refrigerant charge limits for Copeland compressors used without some means of liquid refrigerant control. It is possible that due to system design, particularly on systems where large quantities of liquid can flood to the compressor on start-up, even though the system contains a refrigerant charge smaller than the limits listed in Table 1, some protective action may be required.

When liquid refrigerant control becomes a problem one or more of the following types of corrective action may be necessary.

TABLE 1
REFRIGERANT CHARGE LIMITS (R-22)

Motor-Compressor Model	Crankcase Heater or Pumpdown Required If Refrigerant Charge Exceeds
Copelametic	
K	4.0 LBS.
E	6.0
3	8.0
L	8.0
N	10.0
M	15.0
9	17.5
4	23.0
6	28.0
Copelaweld	
JR	1.75 LBS.
RR (2 models)	3.0
RR (4 models)	4.0
SR	8.0
CR	6.0
YR	10.0

One of the basic characteristics of a refrigerant and oil mixture in a sealed system is the fact that refrigerant is attracted by oil and will vaporize and migrate through the system to the compressor crankcase even though no pressure difference exists to cause the movement. On reaching the crankcase the refrigerant will condense into a liquid, and this migration will continue until the oil is saturated with liquid refrigerant. The amount of refrigerant that the oil will attract is primarily dependent on temperature, increasing rapidly as the temperature increases and approaching a maximum at normal room temperatures.

When the pressure on a saturated mixture of refrigerant and oil is suddenly reduced, as happens in the compressor crankcase on start-up, the amount of liquid refrigerant required to saturate the oil is drastically reduced, and the remainder of the liquid refrigerant flashes into vapor, causing violent boiling of the refrigerant and oil mixture. This causes the typical foaming often observed in the compressor crankcase at start-up, which can drive all of the oil out of the crankcase in less than a minute. (Not all foaming is the result of refrigerant in the crankcase — agitation of the oil will also cause some foaming.)

One condition that is somewhat surprising when first encountered by field personnel is the fact that the introduction of excessive liquid refrigerant into the compressor crankcase can cause a loss of oil pressure and a trip of the oil pressure safety control even though the level of the refrigerant and oil mixture may be observed high in the compressor crankcase sight glass. The high percentage of liquid refrigerant entering the crankcase not only reduces the lubricating quality of the oil, but on entering the oil pump intake may flash into vapor, blocking the entrance of adequate oil to maintain oil pump pressure, and this condition can continue until the percentage of refrigerant in the crankcase is reduced to a level which can be tolerated by the oil pump.

TYPICAL EFFECTS OF IMPROPER REFRIGERANT CONTROL

Liquid refrigerant problems can take several different forms, each having its own distinct characteristics.

1. Refrigerant Migration

Refrigerant migration is the term used to describe the accumulation of liquid refrigerant in the compressor crankcase during periods when the compressor is not operating. It can occur whenever the compressor becomes colder than the evaporator, since a pressure differential then exists to force refrigerant flow to the colder area. Although this type of migration is most pronounced in colder weather, it can also exist even at relatively high ambient temperatures with remote type condensing units for air conditioning and heat pump applications.

Anytime the system is shut down and is not operative for several hours, migration to the crankcase can occur regardless of pressure due to the attraction of the oil for refrigerant.

If excessive liquid refrigerant has migrated to the compressor crankcase, severe liquid slugging may occur on start-up, and frequently compressor damage such as broken valves, damaged pistons, bearing failures due to loss of oil from the crankcase, and bearing washout (refrigerant washing oil from the bearings) can occur.

2. Liquid Refrigerant Flooding

If an expansion valve should malfunction, or in the event of an evaporator fan failure or clogged air filters, liquid refrigerant may flood through the evaporator and return through the suction line to the compressor as liquid rather than vapor. During the running cycle, liquid flooding can cause excessive wear of the moving parts because of dilution of the oil, loss of oil pressure resulting in trips of the oil pressure safety control, and loss of oil from the crankcase. During the "off" cycle after running in this condition, migration of refrigerant to the crankcase can occur rapidly, resulting in liquid slugging when restarting.

3. Liquid Refrigerant Slugging

Liquid slugging is the term used to describe the passage of liquid refrigerant through the compressor suction and discharge valves. It is evidenced by a loud metallic clatter inside the compressor, possibly accompanied by extreme vibration of the compressor.

HOW IT WORKS (Continued)

The valve on the bottom of the OIL RESERVOIR is the distribution valve to the OIL LEVEL REGULATORS (open position). To remove oil from the OIL RESERVOIR, close the valve and use the 1/4" flare connection on the side of the valve to drain the oil out. Open valve after draining.

NEW SYSTEM START UP

On system start-up of a new parallel system, oil should be added to the OIL RESERVOIR to the upper sight glass port, NOT ABOVE IT. It is commonly accepted that in a new refrigeration system, some oil will be absorbed by the refrigerant as the system becomes balanced out. After two hours of operation, the OIL RESERVOIR, if necessary, should again be filled to the upper sight glass and also after two days, by which time the entire refrigeration system should be balanced out. Then the OIL RESERVOIR must be observed on each service call. No oil should be added again until the oil level falls below the lower sight glass port.

EXISTING SYSTEM START-UP

When installing this OIL CONTROL SYSTEM, on a parallel system that has been in operation for some time, the amount of oil should be added cautiously. With the efficiency of the new OIL SEPARATOR, the oil return could likely be sufficient to fill the OIL RESERVOIR. So fill the OIL RESERVOIR to the lower sight glass port only. Observe for one day. After the second day, if the oil level has not risen to the upper sight glass, add oil. If the oil level has risen above the upper sight glass port, remove the excess oil from the OIL RESERVOIR.

HEAD PRESSURE CONTROL

In order to achieve proper system operation, it is necessary that adequate discharge pressures be maintained to insure that the expansion valve will feed correctly to prevent low suction conditions. The expansion valve is sized to meet capacity requirements at a pressure differential between discharge pressure and design suction pressure. If discharge pressure is allowed to drop below a point which will not maintain this design differential, the suction will also drop due to "starving" of the evaporator. When this happens, nuisance tripping of low pressure control, or low evaporator temperature, will occur.

Normal system design will allow satisfactory operation with discharge pressure down to approximately 180# (95° F.).

On air cooled equipment, a number of methods can be used to maintain this minimum discharge pressure. The most common, and suitable for applications where ambient temperatures are not extreme, is fan cycling. When closer control, or extreme ambient differentials are encountered, condenser flooding would be preferred. Other methods are suitable; however, not offered due to design or component limitations.

On water cooled equipment, discharge pressures are much easier to control, and the most common would be by the use of condenser water regulating valves directly sensing discharge pressure. (See Piping Illustration, Figure 1 and 2).

Since systems are normally subjected to varying load requirements, and fluctuating ambient temperatures, it is not possible to design units to operate satisfactorily and with optimum efficiencies without some means of control of discharge pressures.

Water Regulating Valves

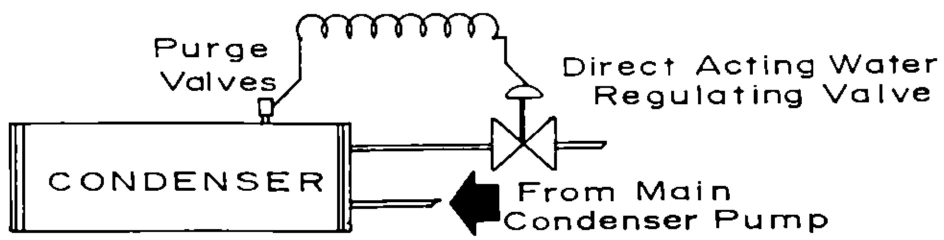


Fig. 1

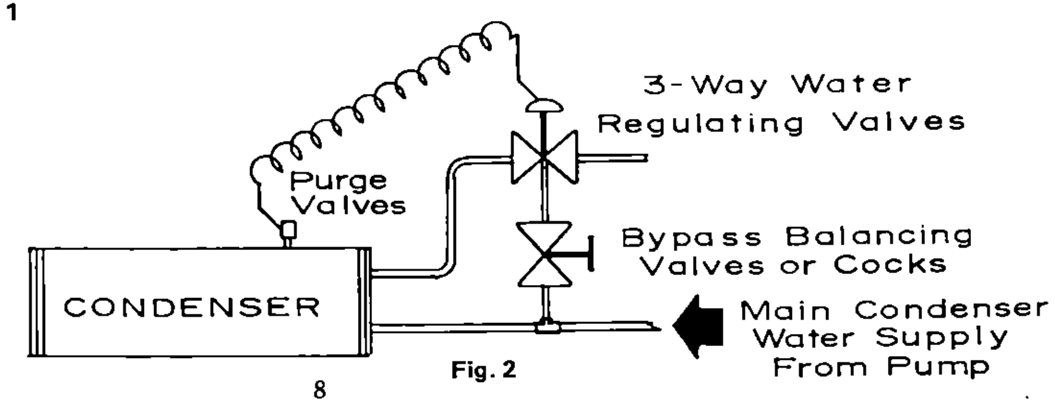


Fig. 2

Copeland®

ALL TSI UNITS ARE CATALOGED BASED ON COPELAND COMPRESSORS
— OTHERS CAN BE OBTAINED UPON REQUEST —

COMPRESSOR OVERHEATING
Todays Most Serious Field Problem

It is ironic that in an industry whose product is cooling and refrigeration, the most serious field problems arise from overheating the compressor. It is not only possible, but probable, that the great majority of compressor failures on low temperature systems originate in compressor overheating, and field experience indicates the great majority of those failures can be eliminated if discharge gas temperatures are reduced to a reasonable level.

Obviously it would be to everyone's benefit if it were possible to design a compressor that could live with the most demanding conditions with no concern about overheating. There are approaches in compressor design which can minimize the problems, such as good exterior heat transfer surface; direct metal to metal contact between the motor and the compressor housing; minimal heat transfer between the low pressure and high pressure areas of the compressor; low volumetric clearance; good motor design; and oil coolers. The fact remains that we are dealing with some physical laws of nature and at extreme operating conditions, the temperatures created may be beyond the capabilities of the best refrigeration compressor.

Why then do we continue to follow practices that make inevitable high failure rates? Certainly no responsible engineer in the industry knowingly designs for unsafe conditions. The reasons strangely enough appear to be as much psychological as physical.

Many engineers, servicemen, and users simply refuse to accept the fact that there are limits to a compressor's operation. They have been conditioned by past experience with conservative safety factors and product improvement to always expect that in some miraculous fashion a change of compressors will solve all their problems. They always cite the one compressor that survives, and ignore the five that have failed. These are probably the same people who refuse to accept the fact that the world's energy resources are reaching the bottom part of the barrel. Some people can cope with problems only by refusing to acknowledge them.

A second part of the problem may be that our educational and training programs are out of date. Twenty years ago liquid refrigerant slugging and liquid floodback were very serious problems. Compressor valves and bearings were more vulnerable, and system designs were not always the best. For twenty years every educational and training program has hammered at the dangers of liquid refrigerant and the need for proper superheat, and few words have been said about the dangers of high return gas temperatures. In effect we have tried to brainwash a whole generation of service personnel to worry only about liquid floodback.

Possibly because of the old fears of liquid refrigerant, we as compressor manufacturers and in general the industry have been slow in recognizing and identifying high temperature failures. At cylinder temperatures of 315°F to 325°F, the lubricating film is literally evaporated off the cylinder walls like water on a hot griddle, but modern refrigeration oils are so resistant to breakdown, carbon is not formed on the valve plate. As a result many high temperature failures are misdiagnosed as liquid failures because the analyst doesn't know the application.

LIQUID REFRIGERANT CONTROL IN REFRIGERATION AND AIR CONDITIONING SYSTEMS

One of the major causes of compressor failure is damage caused by liquid refrigerant entering the compressor crankcase in excessive quantities. Since improper control of liquid refrigerant can often cause a loss of lubrication in the compressor, most such compressor failures have been classified as lubrication failures, and many people fail to realize that the problem actually originates with the refrigerant.

A well designed, efficient compressor for refrigeration, air conditioning and heat pump duty is primarily a vapor pump designed to handle a reasonable quantity of liquid refrigerant and oil. To design and build a pump to handle more liquid would require a serious compromise in one or more of the following: size, weight, capacity, efficiency and noise, not to mention the all important factor, cost.

Regardless of design there are limits to the amount of liquid a compressor can handle, and these limits depend on factors such as internal volume of the crankcase, oil charge, type of system and controls, and normal operating conditions. Proper control of liquid refrigerant is an application problem, and is largely beyond the control of the compressor manufacturer.

The potential hazard increases with the size of the refrigerant charge and usually the cause of damage can be traced to one or more of the following:

1. Excessive refrigerant charge.
2. Frosted evaporator.
3. Dirty or plugged evaporator filters.
4. Failure of evaporator fan or fan motor.
5. Incorrect capillary tubes.
6. Incorrect selection or adjustment of expansion valves.
7. Refrigerant migration.

REFRIGERANT – OIL RELATIONSHIP

In order to correctly analyze system malfunctions, and to determine if a system is properly protected, a clear understanding of the basic refrigerant – oil relationship is essential.

In refrigeration compressors, oil and refrigerant mix continuously. Refrigeration oils are soluble in liquid refrigerant, and at normal room temperatures they will mix completely. Since oil must pass through the compressor cylinders to provide lubrication, a small amount of oil is always circulating with the refrigerant. Oil and refrigerant vapor do not mix readily, and the oil can be properly circulated through the system only if gas velocities are high enough to sweep the oil along.

If velocities are not sufficiently high, oil will tend to lie on the bottom of the evaporator tubing, decreasing heat transfer and possibly causing a shortage of oil in the compressor. As evaporating temperatures are lowered, this problem becomes more critical since the viscosity of oil increases with a decrease in temperature. For these reasons, proper design of piping is essential for satisfactory oil return.

The desuperheating valves were installed within three to six feet of the compressor suction valve with the suction line insulated from the expansion valve to the compressor inlet. Initially ½ ton valves were tried, but better performance was obtained with nominal one ton valves with compressors in the 7½ H.P. to 25 H.P. range. Several alternate expansion valve charges were tried but satisfactory performance was obtained only with the following:

Sporlan GR-1-L1
Alco LCL2E-IE.

If suction lines can be insulated, desuperheating valves are not necessary, but desuperheating valves do offer a means of reducing temperatures on systems where no other approach is possible.

Liquid to Suction Heat Exchangers

Liquid to suction heat exchangers in refrigeration systems are useful in raising the return gas temperature to prevent frosting or condensation, to evaporate any liquid droplets in the vapor stream, and to subcool the liquid to prevent flash gas in the liquid line.

However there appears to be some misunderstanding in the industry regarding any capacity increase as the result of a heat exchanger. To the extent that heat absorbed by the vapor from the liquid refrigerant displaces heat that might be picked up from the ambient or other non-refrigerated spaces there is a capacity increase, and therefore on systems with uninsulated suction lines there is normally an increase.

The mere transfer of heat from liquid to suction does not in itself add any significant capacity or efficiency to the system. While the enthalpy of the liquid will be decreased, thus increasing the enthalpy change per pound in the evaporator, the warmer vapor has a higher specific volume (cubic feet per pound) so the pumping capacity of the compressor will be decreased. These two factors, the decreased enthalpy of the liquid and the higher specific volume of the vapor, largely cancel out the effect of each individually, so the net effect is small, and if the liquid line loses any temperature in transit, it is doubtful if any increase is measurable.

With insulated suction lines the heat exchanger not only loses its capacity benefit, but may actually be a threat if it raises the temperature of the return gas excessively. In particular, liquid to suction heat exchangers in the machine room at the compressor may actually contribute to compressor failure if they elevate the temperature of the return gas beyond an acceptable level.

Summary

Compressor overheating has become and remains a major field problem primarily because few people recognize or understand the pattern of failure. In todays large sophisticated systems, lower return gas temperatures are essential, and it appears the most direct way of accomplishing this is by insulating suction lines. While this will increase the initial system cost, field experience indicates the reduction in compressor failure rates can quickly repay the added investment.

On existing systems where insulating lines is not feasible, desuperheating expansion valves have been proven to be an effective means of obtaining lower discharge temperatures.

Compressor rating data is a continuing source of confusion in the industry. In order to provide a common basis for comparison, compressors are rated and capacity data is published at common conditions. In the case of Copeland low temperature compressors, the rating data is presented with return gas temperature of 65°F. This temperature was selected many years ago, probably for convenience of testing as much as any other factor, and at that time might have represented an acceptable operating condition for most systems. Over the years the trend to larger horsepower compressors, the demand for lower evaporating temperatures, and in particular the increasing use of multiple compressors on a single system have tremendously increased the stresses on the compressor.

Users still interpret the published rating data with 65°F return gas as a recommendation that the compressor should be operated at that condition. In the case of smaller, single compressor systems operating at more moderate suction pressures, 65°F may still be acceptable, but on larger, more sophisticated equipment, lower return gas temperatures are required if the compressor is to be maintained within acceptable temperature limits.

Refrigeration is not an exact science. System operation can be drastically affected by many variables in the field installation and operation, and since each party would like to believe that all problems are the fault of someone else, unfortunately there are widely varying opinions on every aspect of system performance. It is easy to understand the difficulty of the final user in itelligently evaluating and specifying the reliability that he might truly desire. It is easy to understand the difficulty of the design engineer in maintaining conservative design factors based on only his good judgment when the analysis of failures is cloudy, and he is subject to the tremendous cost pressures of a fiercely price competitive industry.

At a meeting of service managers from several large commercial refrigeration manufacturers, it was unanimously agreed that compressor overheating is the greatest single field problem existent today, and the one on which the least headway was being made in solving the problem at the user level. If we are to make any major reduction in failure rates due to overheating, it is essential that there is widespread understanding of the problem in the industry.

For those that refuse to accept the fact that there is a problem, little can be done. The encouraging side of the picture is the fact that for those that are willing to make the effort, low temperature overheating failures can be stopped dead.

Temperature Limits

Most refrigeration oils will start to break down or carbonize at temperatures of 350°F. Tests in a contaminant free atmosphere may indicate a reasonable tolerance for even higher temperatures, but the real world is inhabited by a multitude of systems that have varying degrees of contaminants such as air and moisture.

Extreme ring and piston wear can occur at cylinder temperatures of 310-330°F with little oil carbonization. There is growing evidence that modern refrigeration oils have been so highly refined to obtain good solubility and high breakdown temperatures that the oil is unable to maintain a lubricating film at high temperatures.

Field experience in general would indicate for good long life characteristics, piston, ring, and valve port temperatures should be maintained below 300°F. Normally discharge line temperatures within 6 inches of the compressor outlet will be from 50°F to 75°F cooler than cylinder and piston temperatures, depending on the compressor design and the refrigerant mass flow. Therefore as a general rule, 275°F discharge line temperatures represent a certain failure temperature condition, 250°F is usually a danger level, and 225°F and below are desirable for reasonable life expectancy.

There are different opinions in the industry on oil sump temperatures. The viscosity of the common refrigeration oils decreases rapidly as the temperature increases, and becomes dangerously low at temperatures of 200°F and above. At high temperatures the characteristics of the lubricant are critical, additives may be necessary, and bearings must be capable of withstanding the extreme temperatures. Lower temperatures are generally much more conducive to long life.

Low Temperature R-502 Applications

Few users realize the critical nature of single stage low temperature refrigeration. The supermarket manager demands brick hard −20°F ice cream; the blast freezer operator demands tunnel temperatures of −40°F for faster processing; the ice cream distributor demands −25°F truck body temperatures; each thinking only of satisfying his temperature demand, and none of them realizing their demands are an almost certain death sentence for the compressor. The normal low temperature evaporating limit for single stage compressors is −40°F, and although this can be extended to intermittent periods of operation down to −50°F, operation below −50°F (0 psig with R-502) creates discharge temperatures which can destroy the compressor.

In a similar fashion, the contractor or original equipment manufacturer who for reasons of convenience or expediency cycles the fan providing compressor cooling, or locates the compressor so that adequate cooling air cannot impinge directly on the compressor may unwittingly be creating the same critical temperature conditions. At low temperature operating conditions the decreasing density of the refrigerant vapor and the heating effect of high compression ratios combine to create high discharge temperatures which cannot be controlled by refrigerant cooling alone. The additional heat transfer gained from a direct air blast on the compressor is absolutely essential for compressor survival, and any decrease in the recommended airflow or loss of direct impingement on the compressor can lead directly to excessive cylinder temperatures.

At todays demanding temperature conditions, a low temperature compressor may be operating right on the edge of its survival limits. The lower the evaporating temperature and the higher the condensing temperature the more critical the discharge temperature becomes. The only way to insure reasonable discharge temperatures at extreme conditions is by means of very low return gas temperature.

Table 1 illustrates some typical internal temperatures on low temperature compressors with 65°F return gas temperature, a common field condition. The cylinder temperatures have been calculated based on a 75°F increase in the return gas temperature after entering the compressor and prior to entering the cylinder on the suction stroke. The 75°F temperature increase is typical of that found in laboratory testing, and is caused by heat transfer from the motor and compressor body. The final column in the table is the return gas temperature entering the compressor that would be necessary to maintain internal cylinder temperatures below 300°F.

Undoubtedly with operating wear, the temperature conditions become even more severe. Table 1 data is calculated based on laboratory testing, but individual installations vary, and the goal should be to maintain temperatures on the discharge line six inches from the compressor at 225°F or less.

Due to the danger of freezing in underground trenches and sweating in the machine room, low return gas temperatures cannot be achieved in a typical supermarket with uninsulated suction lines. The increasing incidence of low temperature failures from overheating indicates some major change in design approach is required to improve system reliability.

TABLE 1

TYPICAL R-502 CYLINDER DISCHARGE GAS TEMPERATURES

Saturated Evaporating Temperature at Compressor Suction Pressure	Saturated Condensing Temperature at Compressor Discharge Pressure	Typical Return Gas Temp.	Cylinder Discharge Temperature	Return Gas Temperature Necessary to Limit Discharge Temp. to 300°F
−40°F	130°F	65°F	340°F	20°F
−40°F	120°F	65°F	335°F	35°F
−40°F	110°F	65°F	320°F	45°F
−25°F	130°F	65°F	320°F	45°F
−25°F	120°F	65°F	310°F	55°F
−25°F	110°F	65°F	300°F	65°F

One large mid-western chain decided to attack this problem in the early 1970's with spectacular results. They set the expansion valves for low superheat control, completely insulated the suction lines to the compressor service valves, provided cool air ventilation for the machine room, with the result that compressor failures have been almost completely eliminated.

The need for completely insulating suction lines on low temperature systems 7½ H.P. in size and larger seems clearly indicated, and is highly recommended to improve compressor life and reliability.

Medium Temperature R-22 Applications

An equally critical temperature condition can occur on medium temperature R-22 systems with evaporating temperatures below 10°F. Because of its poor temperature characteristics, R-22 is no longer used as a low temperature refrigerant, and if evaporating temperatures below 10°F are to be expected, R-502 or R-12 are preferable. Unfortunately many medium temperature R-22 systems designed for nominal evaporating temperatures of 5°F or 10°F wind up operating at suction pressures equivalent to evaporating temperatures of −10°F or lower, and such installations can develop severe problems. Note that it is the suction pressure at the compressor which is critical, not the case evaporating temperature, since in many cases the critical threat is caused by pressure drop between the case and the compressor.

Table 2 illustrates some typical internal temperatures on medium temperature applications with R-22. As in the case of R-502, the cylinder temperatures have been calculated based on a 75°F increase in the return gas temperature after entering the compressor and prior to entering the cylinder.

Desuperheating Expansion Valves

On existing systems with uninsulated lines where it may not be possible to change the system operating conditions, the only means of reducing discharge temperatures to an acceptable level may be with a desuperheating expansion valve.

Extensive field testing on problem installations revealed that discharge temperatures could be reduced almost degree for degree by reducing return gas temperatures. Repeatedly we found systems operating with 250°F to 260°F discharge line temperatures and 60°F return gas temperatures could be modified with a desuperheating expansion valve to obtain discharge line temperatures below 225°F with return gas temperatures of approximately 30°F.

TABLE 2

TYPICAL R-22 CYLINDER DISCHARGE GAS TEMPERATURES

Saturated Evaporating Temperature at Compressor Suction Pressure	Saturated Condensing Temperature at Compressor Discharge Pressure	Typical Return Gas Temp.	Cylinder Discharge Temperature	Return Gas Temperature Necessary to Limit Discharge Temp. to 300°F
−10°F	130°F	65°F	365°F	5°F
−10°F	120°F	65°F	350°F	25°F
−10°F	110°F	65°F	340°F	35°F
0°F	130°F	65°F	335°F	25°F
0°F	120°F	65°F	320°F	45°F
0°F	110°F	65°F	310°F	50°F
10°F	130°F	65°F	320°F	45°F
10°F	120°F	65°F	300°F	65°F
10°F	110°F	65°F	290°F	65°F