WARM LIQUID DEFROSTING TECHNOLOGY FOR SUPERMARKET DISPLAY CASES

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Abstract

A theoretical analysis of a warm liquid defrost (WLD) concept for supermarket display case defrosting is presented for a supermarket frozen food system consisting of 15 cases on a compressor rack operating at -30ºF (-34.4 °C) evaporation temperature. Compared to normal (non-defrost) system operation, compressor isentropic efficiency was seen to drop somewhat during defrost based on the analysis assumptions used depending on the number of cases defrosting simultaneously. With one case of fifteen in defrost system capacity was reduced by about 7%, however the computed compressor COP reduction was only about 1% due to suction gas desuperheating effects. An accumulator/liquid trap apparatus is proposed to collect any excess liquid during defrost and return it to the high-side receiver. In comparison, with electric defrost (one case in fifteen defrosting) the net efficiency penalty is about 10%.

1. Introduction

Low temperature display cases require periodic coil defrosting. There are many defrosting technologies for low temperature supermarket display cases. The most common methods are electric defrost and hot gas or cool gas defrosting. Since hot gas defrosting could result in severe thermal shocks to the case evaporators, the saturated vapor (or 'cool gas') defrosting method was derived, and the level of thermal shock was reduced.

This paper discusses the thermodynamic potential and operational implications of an alternative defrost method – warm liquid defrosting (WLD). The WLD concept (Mei, et al, 2001) uses warm liquid from the system condenser to defrost the evaporator coils. Mei, et al, (2002) described the concept in some detail including its advantages as compared to the electric and gas defrost methods. Among the advantages cited is the potential for a simpler overall system design eliminating hot gas or saturated vapor lines and some of the associated valves.

Responses to the earlier paper noted two major concerns. One of the concerns was that there could be a thermodynamic problem associated with the WLD approach, namely that there is a chance that not all the liquid used for coil defrosting and subsequently expanded into the suction line will be evaporated by the superheated vapor from the operating display cases under all conditions. The other concern is that at very low liquid temperature (below about 50-60°F, or 10-15 °C) there is not enough heat available in the condenser liquid to effectively defrost the evaporators. Addressing the second concern, it is possible to configure the system controls to maintain the condensing pressure at or above a minimum level necessary to provide warm enough liquid to

effectively defrost the case coils. The system efficiency and refrigeration capacity would be somewhat adversely affected because the system head pressure could not be allowed to float below the minimum level.

The primary purpose of this paper is to address the liquid handling issue and the thermodynamic viability of the WLD concept. As part of this effort, researchers at the Environmental Protection Agency (EPA) have conducted testing of a small-scale refrigeration system with the WLD concept incorporated. Gage and Kazachki (2002a) discusses some of the results of this testing. The EPA lab system includes four display cases (two reach-in and two open) connected to a small multiplexed compressor rack. This testing has helped to show that the WLD approach can effectively defrost the case evaporators when the condensing temperature is 90 °F (32.2 °C) or higher under the tested evaporating temperature (–30 °F or –34.4 °C). However, since only three cases were operating in refrigeration mode for each defrosting case there was never enough superheated suction vapor generated to completely evaporate the liquid exiting the defrosting case (even at a 105 °F (40.6 °C) condensing temperature) in the experimental setup. Additional heat was added to assure complete evaporation of the liquid during test operation. Based on the laboratory test data, the EPA investigators estimated that a lineup of 15-19 cases would be needed to generate enough vapor to evaporate the liquid from one defrosting case (Gage and Kazachki, 2002b).

To supplement the EPA test work a thermodynamic analysis is presented here to show that the WLD concept is a viable evaporator defrosting technology. A simple system design modification to deal with situations where the liquid returned from defrosting cases cannot be fully evaporated is also presented and discussed.

2. Thermodynamic Analysis

For this analysis a system consisting of a 15 reach-in glass door case lineup (each with a load of about 4000 Btu/h (1.17 kW) based on average for the reach-in door cases tested by EPA) with a multiplex compressor rack system using R404A is assumed. A schematic of the system is found in Figure 1. Figure 2 provides a detail schematic of the suction accumulator/liquid trap provided to deal with excess liquid. The following operating conditions were used for each scenario examined in this analysis:

-30 ºF (-34.4 °C) saturated evaporator temperature,
vapor leaving the case is saturated,
suction vapor temperature is 34.7 ºF (1.5 °C) for the baseline scenarios,
liquid to cases (from system receiver) is subcooled 9 ºF (5 °C),
liquid exiting case evaporators during defrost is held at 51 ºF (10.6 °C),
liquid from defrosting evaporators is throttled to –30 ºF (-34.4 °C),
cold liquid/vapor mixture from defrosting cases mixes with 34.7 ºF (1.5 °C) suction vapor from other cases in accumulator/liquid trap, and
any excess liquid in the liquid trap is drained back to the main receiver after defrost terminates.

The evaporation temperature, subcooling amount, and compressor suction temperature conditions above are taken from a test run at EPA for a 90 ºF (32.2 °C) condensing condition in March 2001 (Gage and Kazachki, 2002b).

EPA's data indicate that about 2.5 lb (1.13 kg) of ice must be removed each time one of the reach-in type cases in their lab system is defrosted. Their tests of the WLD concept at 90 °F (32.2 °C) condensing show that the liquid temperature difference (between the liquid entering and exiting the case evaporator during the defrost period) ranges from about 30 - 50 °F (17 – 28 °C). Thus, the minimum energy available in the liquid refrigerant for defrosting (based on the minimum temperature difference) is 10.7 Btu/lb (24.9 kJ/kg) for 90 ºF (32.2 °C) condensing operation. This is reduced for the lower condensing temperature scenarios. Refrigerant flow required for defrosting is determined by dividing the amount of heat required to melt 2.5 lbs (1.13 kg) of ice by the energy available in the defrost liquid stream. This is then multiplied by 1.5 to allow for heat needed to raise the ice to 32 ºF (0 °C), heat losses, etc. Results from the analyses are shown in Tables 1-3 (IP units) and A1-A3 (SI units) in Appendix A. Isentropic cycle efficiencies were calculated for comparison.

Table 1 – Baseline refrigeration system operation, no cases defrosting

	90 ºF saturated condensing temperature	80 ºF saturated condensing temperature	70 ºF saturated condensing temperature
Enthalpies, Btu/lb			
Liquid to TXV	39.61	35.97	32.39
Sat. vapor leaving cases	85.50	85.50	85.50
Compressor suction	98.36	98.36	98.36
Compressor discharge (isentropic compression)	121.01	119.38	117.93
Refrigeration load, Btu/h	60,000	60,000	60,000
Total refrigerant flow, lb/h	1308	1212	1130
Liquid temperature entering case expansion valve, ºF	81	71	61
Suction pressure, psig	10.1	10.1	10.1
Suction temperature, ºF	34.7	34.7	34.7
Discharge pressure, psig	205.3	174.1	148.1
Discharge temperature, ºF (isentropic compression)	168.5	157.8	148.1
Compressor power, Btu/h (isentropic compression)	29,626	25,476	22,114
Compressor COP (isentropic compression)	2.025	2.355	2.713

Table 2 – Refrigeration system operation during WLD defrost operation, one, two, or three cases defrosting, 90 ºF saturated condensing temperature

	60 min defrost one case	60 min defrost two cases	30 min defrost one case	30 min defrost three cases
Enthalpies, Btu/lb				
Liquid to TXV	39.61	39.61	39.61	39.61
Sat. vapor leaving cases	85.50	85.50	85.50	85.50
Defrost liq/vap mixture leaving cases	28.91	28.91	28.91	28.91
Compressor suction	95.60	92.68	93.06	85.50
Compressor discharge (isentropic compression)	117.55	113.78	114.26	104.5
Refrigeration load, Btu/h	56,000	52,000	56,000	48,000
Refrigerant flow, lb/hr				
Refrigerating cases	1221	1134	1221	1046
Defrosting cases	50	101	101	303
Total	1271	1235	1322	1349
Total through compressor	1271	1235	1322	1304[a]
Excess defrosting liquid trapped				
Weight, lb	-	-	-	22.3
Volume, ft^3	-	-	-	0.28
Liquid temperature entering case expansion valve, ºF	81	81	81	81
Suction pressure, psig	10.1	10.1	10.1	10.1
Suction temperature, ºF	21.1	6.5	8.4	-30.0
Discharge temperature, ºF (isentropic compression)	155.4	141.3	143.1	107.7
Compressor power, Btu/h (isentropic compression)	27,904	26,049	28,021	24,787
Compressor COP (isentropic compression)	2.007	1.996	1.999	1.937
COP reduction vs. baseline, %	0.9	1.5	1.4	4.4

[a] Suction line refrigerant is two-phase for this case. Liquid fraction is assumed to drain back to liquid receiver after defrost terminates and not to go through compressor.

When a defrost occurs the system refrigeration load drops by 6.7% for each case in defrost. This leads to a reduction in compressor isentropic COP during defrost. However, desuperheating of the compressor suction gas by mixing vapor from the operating cases with the cold liquid/vapor mixture from the defrosting cases offsets this impact to some extent. The net impact is a reduction in compressor isentropic COP during defrost relative to that of the corresponding baseline, non-defrost operation for the set of scenarios examined. For those scenarios where only one of the fifteen cases is in defrost at any one time, the reduction is limited to about 1% for the assumptions used in this analysis.

Table 3 – Refrigeration system operation during WLD defrost operation, one or two cases defrosting, 70 & 80 ºF saturated condensing temperature

	80 ºF 60 min defrost one case	80 ºF 30 min defrost two cases	70 ºF 60 min defrost one case	70 ºF 30 min defrost two cases
Enthalpies, Btu/lb				
Liquid to TXV	35.97	35.97	32.39	32.39
Sat. vapor leaving cases	85.50	85.50	85.50	85.50
Defrost liq/vap mixture leaving cases	28.91	28.91	28.91	28.91
Compressor suction	93.96	85.50	89.45	85.50
Compressor discharge (isentropic compression)	113.86	103.15	106.73	101.85
Refrigeration load, Btu/h	56,000	52,000	56,000	52,000
Refrigerant flow, lb/hr				
Refrigerating cases	1131	1050	1055	979
Defrosting cases	77	306	155	621
Total	1208	1356	1210	1600
Total through compressor	1208	1310[a]	1210	1327[a]
Excess defrosting liquid trapped				
Weight, lb	-	23.1	-	136.5
Volume, ft^3	-	0.29	-	1.76
Liquid temperature entering case expansion valve, ºF	71	71	61	61
Suction pressure, psig	10.1	10.1	10.1	10.1
Suction temperature, ºF	12.9	-30.0	-9.8	-30.0
Discharge temperature, ºF (isentropic compression)	136.3	96.0	103.9	85.1
Compressor power, Btu/h (isentropic compression)	24,033	23,124	20,906	21,697
Compressor COP (isentropic compression)	2.330	2.249	2.679	2.397
COP reduction vs. baseline, %	1.1	4.5	1.3	11.7

[a] Suction line refrigerant is two-phase for this case. Liquid fraction is assumed to drain back to liquid receiver after defrost terminates and not to go through compressor.

By comparison, consider scenario 1 in Table 2 (first column) when using electric defrost. Data from EPA's baseline testing of the reach-in cases at 90 ºF (32.2 °C) condensing indicated that it took about 35 minutes to defrost these cases with a 1550 W electric heater (Gage and Kazachki, 2002b). Using this heater power and defrost time with one case defrosting on the 15-case lineup, the system COP penalty is about 10% compared to non-defrost operation.

The WLD penalty is also dependant upon the refrigerant flow through the defrosting evaporator(s). In the calculations in Tables 2 and 3 above, the required defrost flow is calculated assuming that 50% extra flow is needed above the minimum required to melt the accumulated frost, to heat the ice and evaporator from –30 ºF to 32 ºF (-34.4 to 0 °C), and to allow for miscellaneous heat losses. For the first scenario in Table 2, if the heat losses are assumed to be much greater and the required liquid flow in the defrosting plate were doubled to 100 lb/hr (45.4 kg/hr), the COP reduction would increase from 0.9% to about 1.4%. If it were doubled again to 200 lb/hr (90.7 kg/hr), the penalty would increase to about 2.3%. To minimize the defrost penalty with WLD, therefore, it will be necessary to minimize heat loss from the liquid flowing to the defrosting case.

These theoretical calculations show that the WLD defrost concept should work effectively with relatively modest impact on system efficiency during the defrost periods – considerably less negative impact than with electric defrost. However theory is one thing and practical application is another, which begs the question – Why choose WLD over electric defrost? One reason is that electric defrost is the least energy efficient of any defrost option. Another is that product temperature increase during defrost (for equal defrost cycle times) could be lower with WLD. Both these advantages must be balanced against the greater refrigeration piping complexity (i.e. cost) required on the compressor rack and at the cases. To fully deal with these questions requires detailed system design, evaluation and optimization efforts that are beyond the scope of this paper. It can be noted, however, that many supermarkets use gas defrost systems (hot gas or saturated vapor). Gas defrost systems also have more complex refrigerant piping than electric defrost (see Figure 3) and thermal shocks at the evaporator connections (due to wide temperature swings) have been identified as a cause behind refrigerant leaks (Gage and Troy, 1998). The thermal shock impact with WLD should be much less. Gas systems may be more complex than a WLD implementation as well since the latter does not require use of a third pipeline to the cases.

It must be noted that there are at least two aspects of the WLD defrost concept that may impose some additional limitations on normal system operation. As shown in Tables 2 and 3 there are situations where not enough heat is available in the suction gas from the operating cases to evaporate the liquid from defrosting cases – multiple cases defrosting, lower ambient temperatures (lower condensing temperature), defrost time 30 minutes instead of one hour, etc. In these cases the accumulator/liquid trap is assumed to collect the excess liquid during defrost and allow this liquid to be returned to the main receiver upon defrost termination. The next section discusses the operation of this device. It is possible that during the initial pressure equalization process between receiver and liquid trap some liquid may flash to vapor depending upon the degree of subcooling of the liquid in the receiver and the relative volumes of the receiver and liquid trap.

The second limitation, as noted briefly in the Introduction, is the requirement that the liquid in the system be above a minimum temperature limit, which for the –30 ºF (-34.4 °C) evaporating case lineup examined might be about 70 ºF (17.8 °C). This will

impose a minimum on the allowable discharge pressure, thus systems using WLD could suffer some efficiency and capacity penalty compared to systems that do not have this limitation on discharge pressure. In addition, for extremely cold ambient temperature operation where the condenser liquid may be subcooled to a temperature below the minimum required for defrost, some supplemental heating of the receiver liquid may be needed. The magnitude of this penalty will depend upon local climatic conditions and will be higher for colder regions than for warmer regions. With WLD, system controls modification would be necessary to maintain the minimum required liquid temperature in the receiver.

3. Design Modification

Figure 2 shows the design modification proposed to deal with excess defrost liquid accumulation. If there is not enough superheated vapor from the operating cases to evaporate the liquid from the defrosting cases expanded into the suction manifold, the excess liquid will be stored in the accumulator (or liquid trap). During the defrosting cycle, the suction line is routed through the accumulator. When the defrosting cycle is over, if there is enough liquid in the accumulator, a float valve will activate the three way valves and the solenoid valve so that the accumulator is isolated from the low-side pressure and connected to the receiver. If the accumulator is located above the liquid receiver, the excess liquid in the accumulator will drain back to the receiver by gravity. Otherwise, a pump or a hot gas line will be needed to pump or push the excess liquid back to the receiver. In any case, there will be no liquid throttling so that refrigeration capacity will not be affected by the defrosting cycle. However, when this very cold liquid is drained back to the receiver, the liquid temperature in the receiver could be reduced below the minimum level required for effective evaporator defrosting. The receiver liquid would have to be brought back to the minimum temperature before another defrost cycle could be initiated. However, this delay should not be very lengthy. Once the defrost is over, the cold liquid accumulated in the receiver should be replaced with liquid at normal operating temperature within a few minutes (about 1-8 minutes for the scenarios examined here).

4. Liquid Throttling Process

For both the hot gas and saturated vapor defrosting systems there is a liquid throttling process to reduce the high-side pressure by about 30 psi during the defrosting cycle. The process is necessary because the hot gas, or saturated vapor exiting the evaporator after defrosting the coil must be returned to the liquid manifold. Because the pressure drop across the evaporator is usually higher than that of the condenser, the two-phase refrigerant exiting the defrosting evaporators has lower pressure than that at the liquid manifold. The pressure throttling process is needed to reduce the pressure in the liquid manifold feeding the cases so that the two-phase fluid from the defrosting cases can be returned to the liquid manifold.

The liquid line pressure, just before the expansion device of the operating cases, would therefore be reduced whenever any one of the evaporators in a case line up is defrosting with hot or cool gas. Thus, the case expansion valves would have to open wider to have the same amount of refrigerant flow to the evaporators as during non-

defrost operation. According to the orifice equations (Yuan 1967), when the orifice opening is larger, the high-side-to-low-side pressure differential will decrease, if constant fluid flow rate is maintained. The reduction of the pressure differential, proportional to the square of the orifice opening change, means less throttling, and thus higher suction temperature and reduced refrigeration capacity. This phenomenon of raised suction temperature during defrosting (suction temperature increase from -5ºF to 10ºF or –21 to –12 °C) was observed by a supermarket refrigeration maintenance person (Domitrovic 1999) at a Knoxville, TN area supermarket using a saturated vapor defrosting system. For the WLD design, liquid throttling is not needed. Also eliminated are the defrost gas lines and associated control valves for each evaporator.

5. Conclusions

The present study provides a theoretical analysis to show that the WLD defrost concept is a viable technology for supermarket display case defrosting. With the assumptions used and for the specific system and operating scenario examined (15 cases on a compressor rack operating at –30 ºF (-34.4 °C) evaporation temperature) compressor isentropic efficiency was seen to drop somewhat during defrost operation with the WLD approach compared to that during normal operation depending on the defrost flow rate and the number of cases defrosting simultaneously. With one case out of fifteen in defrost system capacity was reduced by about 7%, however the computed compressor COP reduction was only about 1% with the minimum defrost liquid flow rate assumption considered. If the defrost flow assumption is increased by a factor of four, the COP reduction was about 2.3%. Suction gas desuperheating effects from mixing vapor from the operating cases with the cold liquid/vapor mixture from the defrosting cases works to minimize the defrost efficiency penalty. By comparison, the estimated efficiency penalty if electric defrost is used (one case out of 15 defrosting) is about 10% compared to normal (non-defrosting) operation.

Concerns about liquid spilling over to the compressor can be solved with a liquid drain-back design. To assure reliable case defrosting with WLD, the system discharge pressure must be maintained at or above a minimum level so that the liquid from the condensers is warm enough. This may impose an efficiency and capacity penalty on the overall system in some areas because the discharge pressure will be unable to float to as low a level as it could with other defrost methods.

6. References

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7. Acknowledgements

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8. Appendix A – SI unit tabular data and results

Table A1 – Baseline refrigeration system operation, no cases defrosting

	32.2 °C saturated condensing temperature	26.7 °C saturated condensing temperature	21.1 °C saturated condensing temperature
Enthalpies, kJ/kg			
Liquid to TXV	92.07	83.61	75.06
Sat. vapor leaving cases	198.74	198.74	198.74
Compressor suction	228.63	228.63	228.63
Compressor discharge (isentropic compression)	281.28	277.49	274.12
Refrigeration load, kW	17.58	17.58	17.58
Total refrigerant flow, kg/h	593.3	549.8	512.6
Liquid temperature entering case expansion valve, ºC	27.2	21.7	16.1
Suction pressure, kPa	171	171	171
Suction temperature, ºC	1.5	1.5	1.5
Discharge pressure, kPa	1518	1303	1123
Discharge temperature, ºC (isentropic compression)	75.8	69.9	64.5
Compressor power, kW (isentropic compression)	8.68	7.47	6.48
Compressor COP (isentropic compression)	2.025	2.355	2.713

Table A2 – Refrigeration system operation during WLD defrost operation, one, two, or three cases defrosting, 32.2 ºC saturated condensing temperature

	60 min defrost one case	60 min defrost two cases	30 min defrost one case	30 min defrost three cases
Enthalpies, kJ/kg				
Liquid to TXV	92.07	92.07	92.07	92.07
Sat. vapor leaving cases	198.74	198.74	198.74	198.74
Defrost liq/vap mixture leaving cases	67.20	67.20	67.20	67.20
Compressor suction	222.22	215.43	216.31	198.74
Compressor discharge (isentropic compression)	273.24	264.48	265.59	242.90
Refrigeration load, kW	16.41	15.24	16.41	14.07
Refrigerant flow, kg/hr				
Refrigerating cases	553.8	514.4	553.8	474.5
Defrosting cases	22.7	45.8	25.8	137.4
Total	576.5	560.2	599.6	611.9
Total through compressor	576.5	560.2	599.6	591.5[a]
Excess defrosting liquid trapped				
Weight, kg	-	-	-	10.1
Volume, m^3	-	-	-	0.008
Liquid temperature entering case expansion valve, ºC	27.2	27.2	27.2	27.2
Suction pressure, kPa	171	171	171	171
Suction temperature, ºC	-6.1	-14.2	-13.1	-34.4
Discharge temperature, ºC (isentropic compression)	68.6	60.7	61.7	42.1
Compressor power, kW (isentropic compression)	8.18	7.63	8.21	7.26
Compressor COP (isentropic compression)	2.007	1.996	1.999	1.937
COP reduction vs. baseline, %	0.9	1.5	1.4	4.4

[a] Suction line refrigerant is two-phase for this case. Liquid fraction is assumed to drain back to liquid receiver after defrost terminates and not to go through compressor.

Table A3 – Refrigeration system operation during WLD defrost operation, one or two cases defrosting, 21.1 & 26.7 ºC saturated condensing temperature

	26.7 ºC 60 min defrost one case	26.7 ºC 30 min defrost two cases	21.1 ºC 60 min defrost one case	21.1 ºC 30 min defrost two cases
Enthalpies, kJ/kg				
Liquid to TXV	83.61	83.61	75.29	75.29
Sat. vapor leaving cases	198.74	198.74	198.74	198.74
Defrost liq/vap mixture leaving cases	67.20	67.20	67.20	67.20
Compressor suction	218.40	198.74	89.45	198.74
Compressor discharge (isentropic compression)	264.66	239.70	248.09	236.74
Refrigeration load, kW	16.41	15.24	16.41	15.24
Refrigerant flow, kg/hr				
Refrigerating cases	513.0	476.3	478.5	444.1
Defrosting cases	34.9	138.8	70.3	281.7
Total	547.9	615.1	548.8	725.8
Total through compressor	547.9	594.2[a]	548.8	601.9[a]
Excess defrosting liquid trapped				
Weight, kg	-	10.5	-	61.9
Volume, m^3	-	0.008	-	0.05
Liquid temperature entering case expansion valve, ºC	21.7	21.7	16.1	16.1
Suction pressure, kPa	171	171	171	171
Suction temperature, ºC	-10.6	-34.4	-23.2	-34.4
Discharge temperature, ºC (isentropic compression)	57.9	35.6	39.9	29.5
Compressor power, kW (isentropic compression)	7.04	6.78	6.13	6.36
Compressor COP (isentropic compression)	2.330	2.249	2.679	2.397
COP reduction vs. baseline, %	1.1	4.5	1.3	11.7

[a] Suction line refrigerant is two-phase for this case. Liquid fraction is assumed to drain back to liquid receiver after defrost terminates and not to go through compressor.

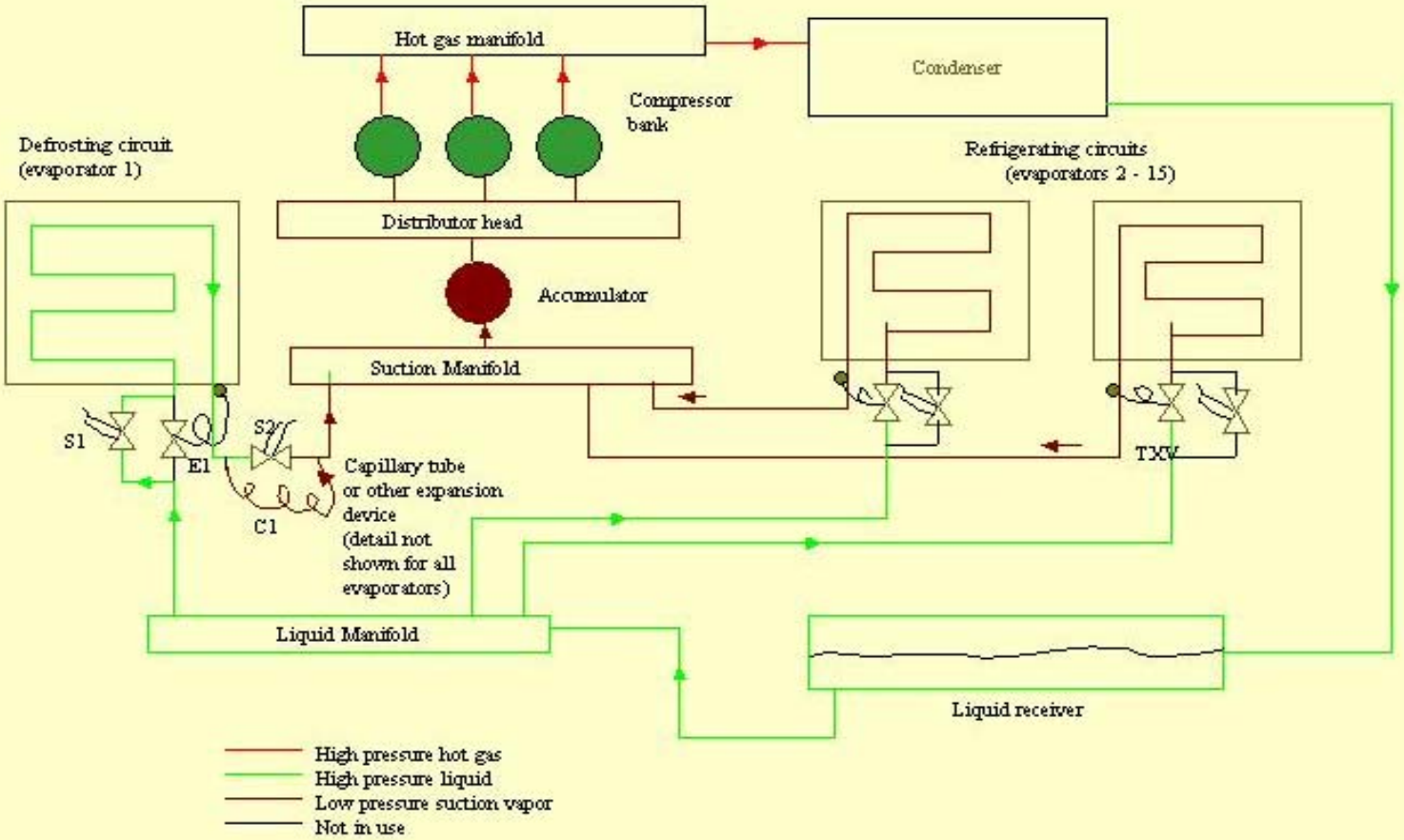

Figure 1. Warm Liquid defrosting schematic.

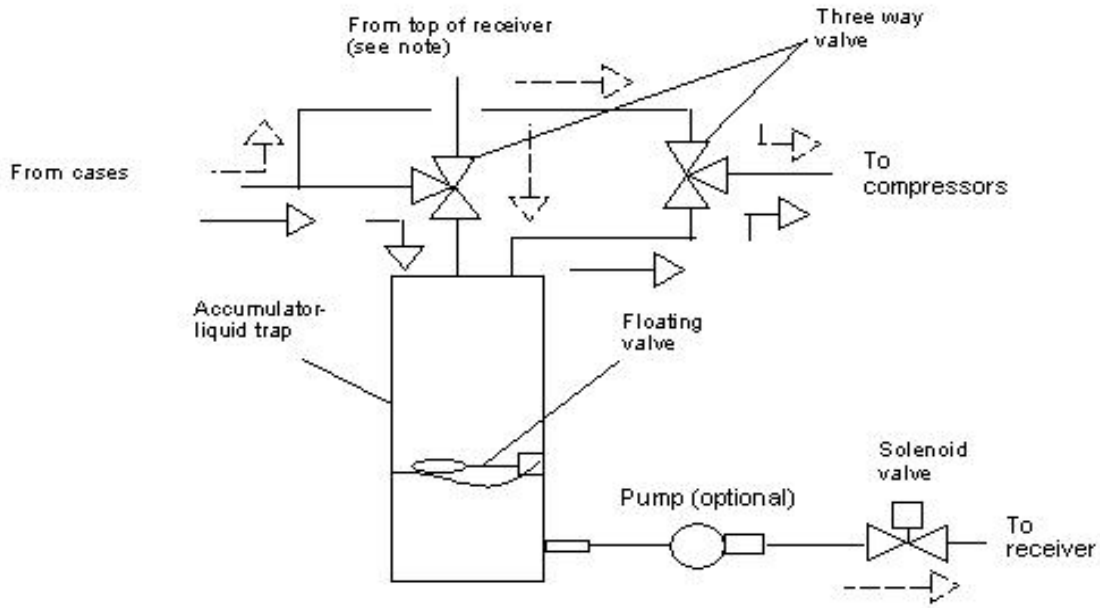

Fig. 2. Liquid return design

Normal and Defrost operation flow direction

Liquid return operation flow direction

Notes:
1. If three way valve is connected to hot gas line, no pump will be needed.
2. If liquid trap is located above receiver, gravity drain will return liquid from liquid trap to receiver

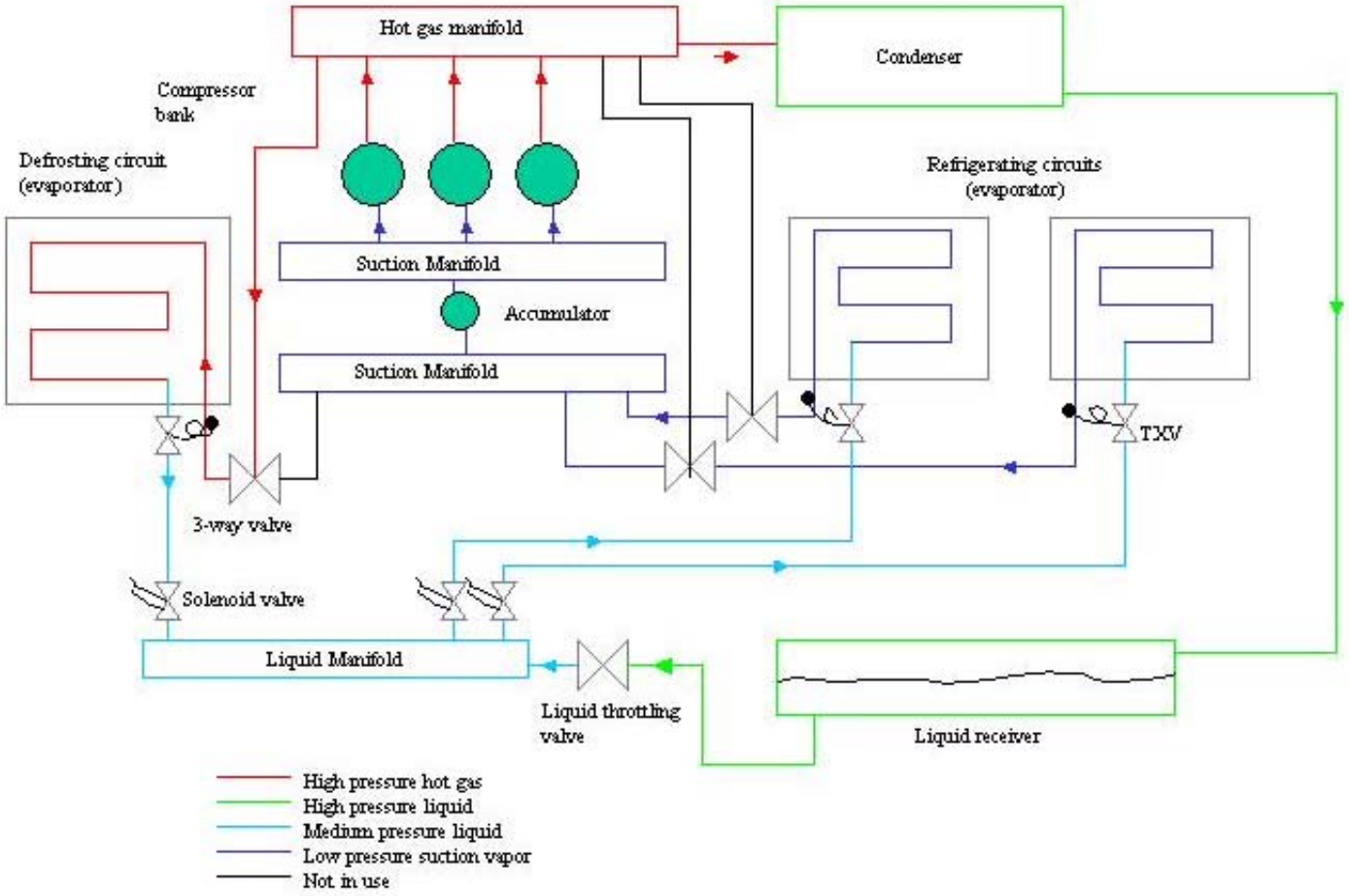

Figure 3. Hot gas defrosting schematic.