1988

Steady State Characteristics of Failures of a Household Refrigerator

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Abstract

One of the problems facing manufacturers of household appliances is the report of the failure of the appliance during the warranty period. One manufacturer of household refrigerators reported that 50% of all parts replaced under warranty were still in working condition. This paper reports on the possibility of developing a diagnostic system to help resolve this problem. The paper discusses the following:

1. some of the common failures

2. the instrumentation placed on a refrigerator to see if such failures could be detected and to determine if different failures could be distinguished

3. the experimental results of several forced failures are presented and discussed

DIAGNOSTIC DES PANNES D'UN REFRIGERATEUR MENAGER.

RESUME : L'un des problèmes auxquels doivent faire face les constructeurs d'appareils ménagers est le compte-rendu des pannes de l'appareil au cours de la période de garantie. Un constructeur de réfrigérateurs ménagers a déclaré que 50 % de l'ensemble des pièces remplacées sous garantie étaient encore en état de marche. Ce rapport rend compte de la possibilité de mettre au point un système de diagnostic pour aider à résoudre ce problème. Il examine les poins suivants :

1. quelques-unes des pannes courantes ;

2. les instruments équipant le réfrigérateur pour voir si ces pannes pourraient être décelées et pour déterminer si différentes pannes pourraient être distinguées ;

3. les résultats expérimentaux de plusieurs pannes forcées sont présentés et discutés.

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1. INTRODUCTION

One manufacturer of household refrigerators[1] reported that a large percentage of all parts returned under warranty were still in working order. This is a tremendous cost to both the manufacturer and the consumer. The main thrust of this work was to help develop a diagnostic system for a household refrigerator to eliminate this problem. Beside helping to eliminate the replacement of good parts, such a system could warn the owner of pending failures or conditions, which if corrected could prevent larger, more costly failures; and eliminate any defective parts before the equipment leaves the manufacturer's facilities. In order to design such a system, it was necessary to identify the most common failures and their characteristic effects on the household refrigerator; identify the least number of sensors needed to locate and isolate these failures; and use, if possible, sensors that do not require breaking into the sealed system. With these objectives in mind, the research had the following tasks:

1. Identify the most significant failures.
2. Instrument and test a refrigerator for the purpose of observing the characteristics of each failure.
3. Define the characteristics of each failure by analyzing the results.
4. Determine which sensors give the most information about the failures.

By studying failure reports and in private communication with appliance manufacturers, the following failures were identified and investigated: a partially blocked capillary tube, heat exchanger fan failures, leaks in the compressor caused by a crack in the piston or the suction valve, frost on the evaporator, and charging failures. This report will present the characteristics from three of these failures: a blocked capillary tube, a failed evaporator fan, and a leaking compressor.

2. INSTRUMENTATION AND TEST PROCEDURE

The major objective of instrumenting the refrigerator was to determine the least number of transducers needed to locate and identify the most failures. With this objective in mind, the refrigerator was instrumented with a large number of transducers. There were two major reasons for this approach: first, the large number helped define and isolate each failure; and second, it provided a wide range of locations and types of transducers from which the most effective were identified. The most effective transducers were defined as those transducers that revealed the most information about each failure.

Figure 1 shows the locations and types of transducers used on the refrigerant loop, while Figure 2 shows the locations of the thermocouples used to measure the cabinet temperatures. The instrumentation consisted of 66 thermocouples, 5 pressure transducers, 3 venturi flow meters, and 1 watt meter. A thermocouple was placed on every bend along the length of the evaporator tube, and along every other bend along the length of the condenser tube making a total of 32 thermocouples on the condenser and 16 thermocouples on the evaporator. A venturi type of flow meter was selected instead of other types of flow meters for two major reasons: small size and good pressure recovery. A small flow meter has the advantage of direct installation into the refrigerant line without extra tubing. Since changes in pressures can change the operation of the refrigerator, good pressure recovery was necessary.

The refrigerator was placed in a psychometric room in order to keep the ambient temperature within ±1 F. The psychometric room was allowed to run until the room temperature was stable at 90 F. This temperature was chosen because it is a temperature used by the Association of Home Appliance Manufacturers (AHAM) to simulate a standard load on a refrigerator.[2] To further simulate loads, two 30 watt light bulbs were placed in the freezer and the fresh food cabinets. Another purpose of the 30 watt sources was prevent the compressor from cycling, so that the compressor was operating for the duration of each test. The refrigerator was turned off and allowed to come to thermodynamic equilibrium with the room temperature. The purpose of this was to give a common initial temperature for all of the components of the refrigerator. To allow sufficient time for the refrigerator components to come to steady state conditions, the duration of each test was 5 hours. The temperatures in the cabinets had much slower time responses than the refrigerant pressures, temperatures, and flow rate. These latter parameters were fairly constant over the last hour of the tests; and therefore were averaged over the last hour. These time averaged data were used to show the steady state characteristics of each test.

3. TEST RESULTS AND FAILURE CHARACTERISTICS

In order to compare data from different test runs operating under different failure conditions, all data were normalized as follows:

$$P^* = \frac{P}{P_n} \tag{1}$$

$$\dot{m}^* = \frac{\dot{m}}{\dot{m}_n} \tag{2}$$

$$\text{Ratio}^* = \frac{\text{Ratio}}{\text{Ratio}_n} \tag{3}$$

$$x^* = \frac{x}{L} \tag{4}$$

$$T^* = \frac{T - T_A}{T_B - T_A} \tag{5}$$

The subscript n denotes time averaged test data from the household refrigerator operating under normal conditions. P, T, and $\dot{m}$ are pressure, temperature and mass flow rate respectively. L is the total length of either the evaporator or the condenser, and x is the distance along the specified heat exchanger. The temperatures along the condenser from all of the tests were normalized using the same temperature range. The data from all of the tests fell within this range. Another similar temperature range was selected for the evaporator. T_B and T_A are the upper and lower temperatures of each range. The temperatures of the air before and after the evaporator were normalized using the same temperature range that is used to normalize the temperatures along the evaporator. "Ratio" is the pressure ratio of the discharge pressure over the suction pressure.

3.1 Normal conditions test (Normal)

Figures 3 and 4 are plots of the steady state temperatures vs. location for the evaporator and the condenser respectively. Also plotted on these graphs are the saturation temperatures for the average pressure of each heat exchanger. Figure 3 shows that over almost the entire length of the evaporator, boiling occurred; Figure 4 shows that the first 10% of the condenser is in the superheated region while condensation takes place over the remaining 90%.

3.2 Partially blocked capillary tube (Blocked Cap. Tube)

A blocked capillary tube is one problem which leads to the returning of good parts. A blocked capillary tube can cause the compressor to over heat because the refrigerant flow is slowed or stopped, and therefore can not cool the compressor effectively. The high temperatures in the compressor will cause the over ride switch to open up. When the repairman comes, he or she measures the resistance of the compressor winding, finds an open circuit, and assumes the compressor has failed. The repairman replaces the good compressor, evacuates the system, and installs a drier. This solves the problem, but at the expense of replacing a good compressor.

The blockage or restriction in the capillary tube can be caused by moisture freezing or dust and dirt particles being trapped in the tube. This restriction was simulated with a ball valve placed after the condenser and before the capillary tube. Figures 5 and 6 and Table 1 compare the results of the refrigerator with a partially blocked capillary tube against the refrigerator operating under normal conditions.

A lower than normal mass flow rate is a characteristic of this type of failure, see Table 2. A partially blocked capillary tube will increase the resistance to refrigerant flow, thus reducing the mass flow rate. A second characteristic of a blocked capillary tube is liquid refrigerant accumulating in the condenser, see Figure 6. The refrigerant condensed along the first third of the condenser length, dropped almost to the temperature of the surrounding air in a very short distance, and continued to subcool gradually for the remainder of the length. The low mass flow rate allowed more refrigerant to condense along a shorter than normal distance. The constriction caused this excess refrigerant to build up in the condenser. The discharge pressure was almost the same as the the discharge pressure under normal conditions see Table 1. One would have expected a larger than normal pressure due to the accumulation of liquid refrigerant.[3] The evaporator had a lower than normal pressure, see Table 1. The evaporator inlet temperature was above the saturation temperature; therefore the refrigerant entering the evaporator was superheated, see Figure 5. The low mass flow rate combined with the low pressure caused the refrigerant to completely evaporate in the capillary tube.

The most effective transducers for this failure were the high side and low side pressure transducers, the thermocouple at the inlet of the evaporator, and the thermocouples along the condenser. The pressure transducers indicated the large pressure difference between the high side and low side pressures, no change in the discharge pressure, and the decrease in the suction pressure. The thermocouple at the inlet of the evaporator indicated that the refrigerant entering the evaporator was superheated. The thermocouples along the condenser showed that a major portion of the condenser was subcooled.

3.3 Evaporator fan off (Evap. Fan Off)

Figures 7 and 8 and Table 1 compare the results with the evaporator fan turned off against the results of the normally operating refrigerator. The evaporator fan failure was simulated by switching off the evaporator fan.

With the evaporator fan turned off, the resistance to heat transfer increased due to a change from forced convection to free convection heat transfer along the outer wall of the evaporator. This increased resistance reduced the heat transfer to the evaporator and increased the average temperature difference between the fluid in the evaporator and the surroundings. The increased temperature difference was reflected by a decrease in the average temperature of the evaporator, see Figure 7. This decrease was also reflected by the decrease in the suction pressure. The reduction in suction pressure was propagated to the discharge pressure resulting in its decrease, see Table 1. The pressure ratio increased, although the pressures through out the system decreased, see Table 1. The air temperature at the top of the evaporator was lower than the temperature at the bottom, see Table 2. This characteristic was due to the fact that the evaporator was vertical. When the fan was on, the air flowed upward across the evaporator. This caused the temperature at the top to be lower than the temperature at the bottom. When the fan was off, however, the cooler air sank to the bottom of the evaporator, while the warm air rose; thus causing the temperature at the top of the evaporator to be warmer.

The most effective transducers used to determine an evaporator fan failure were the suction and discharge pressure transducers and the thermocouples used to measure the temperatures of the air around the evaporator. The pressure transducers indicated a drop of the pressures throughout the sealed system as well as indicating that the pressure ratio increased. The thermocouples showed that the air at the top of the evaporator was warmer than the air at the bottom.

3.4 Compressor Leak (Comp. Leak)

Leaks in the compressor valves and a cracked or worn piston are failures that have been investigated by D. Jankov.[4] Jankov has found that a leak in the discharge valve will cause the compressor to work harder. This is due to high pressure gas leaking back into the cylinder during the suction process. Jankov has also shown that it takes less time from start up to reach the discharge pressure due to the fact that higher pressure gas is in the cylinder at the beginning of the compression process. Jankov found that a leak in the suction valve reduces compressor work. During the compression cycle, some of the gas leaks through the suction valve into the compressor return line. This causes the return line to have a higher that normal pressure. Since the leak reduces the resistance to compression, the compressor works less. Jankov also indicated that it takes more time from start up to reach the discharge pressure. Piston leakage has the same effects as a suction valve failure. Jankov simulated the compressor leakage by puncturing the valves or piston. Pressure transducers were installed in the compressor to measure the pressures. Using pressure transducers, however, require opening the sealed system, this in turn could cause leaks. It was therefore desirable to find a diagnostic scheme which does not require opening the sealed system. It is with this in mind that further testing was done on compressor leakage. Since the compressor shell was not opened, a bypass line with a valve was installed to simulate the leakage. The bypass line passed refrigerant from the discharge line to the suction line. The line was instrumented with a valve to control the leakage and a differential pressure transducer to measure the mass flow rate of the refrigerant flowing through the valve. The bypass line simulated suction valve leakage and a cracked piston. To simulate the discharge valve leak, it would have been necessary to drill a hole in the valve, therefore this failure was not simulated in this research.

Figures 9 and 10, and Table 1 compare the results of a refrigerator with a leaking suction valve or a cracked piston against a normally operating refrigerator. A characteristic of a refrigerator with a leaking compressor is a higher than normal suction pressure and a lower than normal discharge pressure, see Table 1. Also the mass flow rate decreased, see Table 2.

The most effective transducers used to indicate a leaking compressor were the discharge and suction pressure transducers. They indicated a rise in the low side pressure as well as a decrease in the high side pressure, and thus a decrease in the pressure ratio.

3.5 Other Results

By examining tests at different ambient temperatures, it was found that the inlet temperatures to the compressor were almost the same as the air temperatures surrounding the compressor. Enough energy was transferred from the return line so that the refrigerant was in thermodynamic equilibrium with the surroundings as it entered the compressor.

A relationship was found between the pressure ratio and the temperature ratio across the compressor. It was assumed that the refrigerant under went a polytropic compression process. The relationship is:

$$\frac{T_o}{T_i} = \left[\frac{P_o}{P_i}\right]^{\frac{n-1}{n}} \qquad (6)$$

It was experimentally found that n=1.0685.

4. EFFECTIVE TRANSDUCERS

There were two criteria in choosing the most effective transducers: (1) which ones identified the most failures and (2) which ones had minimal effects on the the refrigerator. The first criteria required that the transducer gave output that was unique to the type of failure. One of the objectives of the second criteria was to use transducers that did not require opening the sealed system. Looking at Table 1, it can be shown that the discharge and suction pressures combined with the pressure ratio are unique to the type of failure. For example, a high suction pressure and a low discharge pressure identifies compressor leakage. Another example, a low suction pressure combined with a discharge pressure that is the same as normal conditions indicates a partially blocked capillary tube.

The major problem of using pressure transducers is that it requires opening the sealed system. This can be avoided as follows:

1. A careful study of the temperature profiles of the condenser shows that for steady state operation, the refrigerant is always in the two phase region between the first 10% to 20% of its length. Thus the pressure inside this part of the condenser is the saturation pressure for the measured temperature.
2. Since the pressure drops in both the condenser and the evaporator are small, the pressures entering and leaving the compressor are the same as the saturation pressures in the evaporator and the condenser respectively.
3. By placing thermocouples at the inlet and the outlet of the compressor and using equation 6, the discharge pressure can be determined.

5. CONCLUSIONS

It was found that suction pressure, discharge pressure, and pressure ratio uniquely identified the failures listed. To avoid pressure transducers, three thermocouples can be used to find the suction and discharge pressures. A thermocouple placed between the first 10% to 20% of the condenser length will locate the saturation temperature, and hence the pressure, of the condenser. Thermocouples placed near the inlet and outlet of the compressor will give the temperature ratio across the compressor from which the pressure ratio can be determined using equation 6. Using the pressure ratio and the discharge pressure, the suction pressure and failure can be determined.

TABLE 1 - Normalized Pressures and Pressure Ratio of Failures

Failure	P^* Discharge	P^* Suction	Ratio*
Normal	1.00	1.00	1.00
Blocked Cap. Tube	0.99	0.51	1.91
Evap. Fan Off	0.86	0.66	1.30
Comp. Leak	0.91	1.63	0.56

TABLE 2 - Normalized Mass Flow Rates and Air Temperatures

Failure	$\dot{m}^*$	Normalized Air Temperatures Before Evap.	Normalized Air Temperatures After Evap.
Normal	1.00	0.36	0.33
Blocked Cap. Tube	0.39	0.91	0.87
Evap. Fan Off	0.54	0.24	0.54
Comp. Leak	0.63	0.67	0.64

6. REFERENCES

1. Personal Communication, Whirlpool Corporation
2. *An American National Standard: Household Refrigerators, Combination Refrigerator-Freezers, and Household Freezers*, Chicago: Association of Home Appliance Manufacturers, 1979.
3. *ASHRAE Handbook 1983 Equipment Volume*. Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1983.
4. D. Jankov, "Valve Failure Detection in Refrigeration Compressors", M.S. Thesis, School of Mechanical Engineering, Purdue University, August 1985.

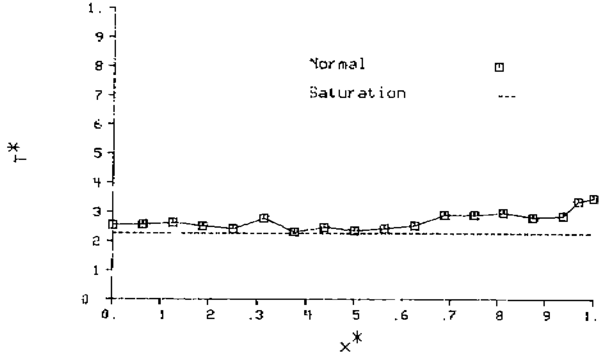

Figure 3. Normalized Steady State Temperatures Along Evaporator of a Household Refrigerator Operating Under Normal Conditions

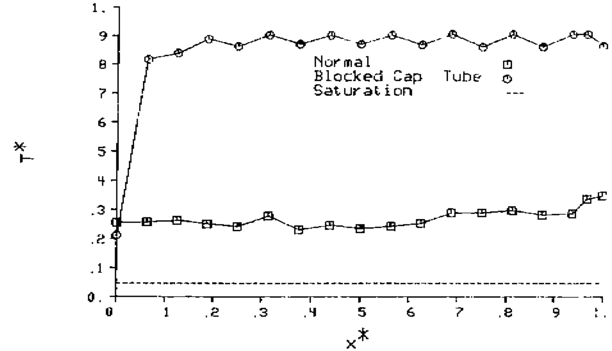

Figure 5. Normalized Steady State Temperatures Along Evaporator of the Failure Due to a Partially Blocked Capillary Tube vs. Normal Conditions

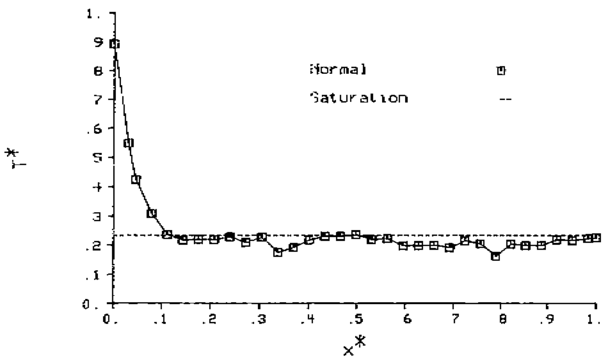

Figure 4. Normalized Steady State Temperatures Along Condenser of a Household Refrigerator Operating Under Normal Conditions

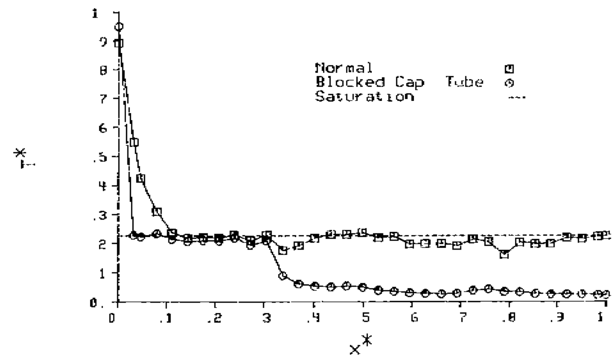

Figure 6. Normalized Steady State Temperatures Along Condenser of the Failure Due to a Partially Blocked Capillary Tube vs. Normal Conditions

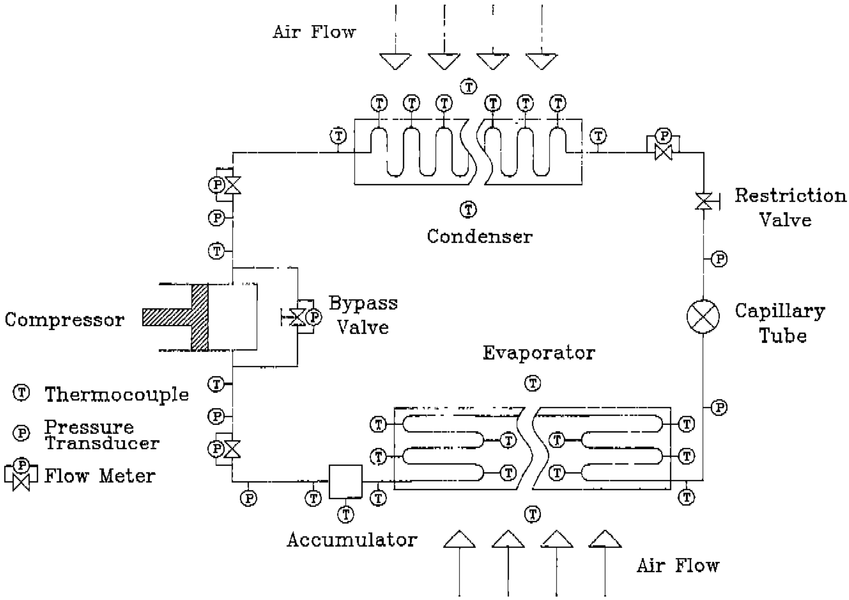

Figure 1 - Schematic of Transducers on Refrigeration Line

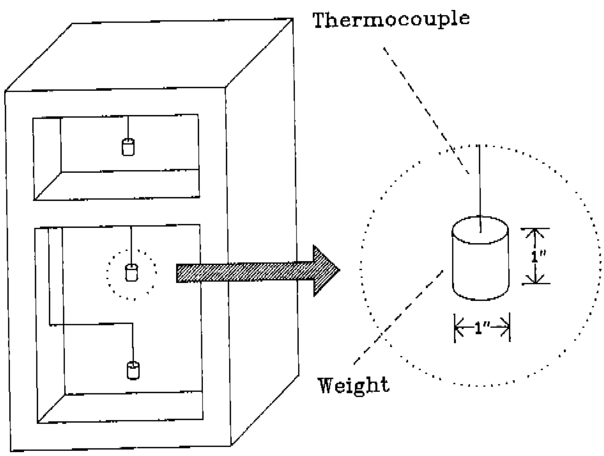

Figure 2 - Location of Weighted Thermocouples

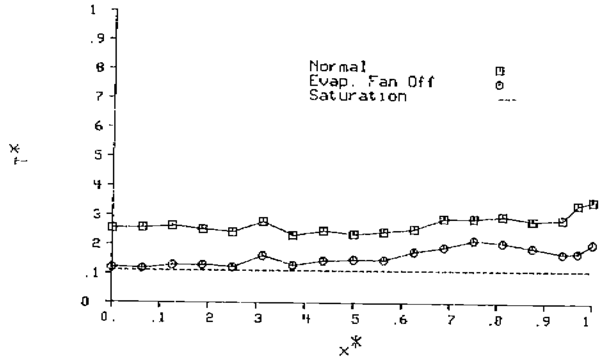

Figure 7. Normalized Steady State Temperatures Along Evaporator of the Evaporator Fan Failure vs. Normal Conditions

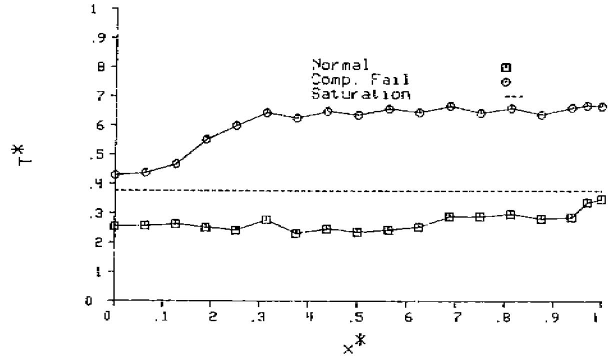

Figure 9. Normalized Steady State Temperatures Along Evaporator of the Failure Due to Compressor Leakage vs. Normal Conditions

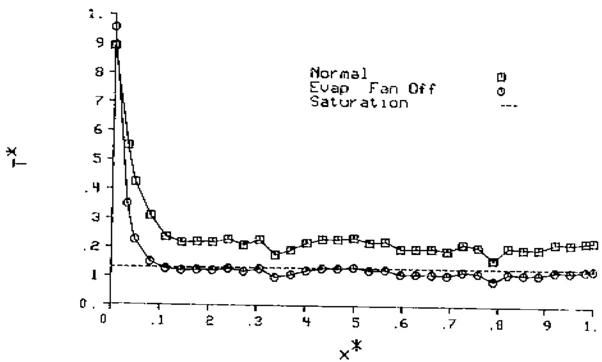

Figure 8. Normalized Steady State Temperatures Along Condenser of the Evaporator Fan Failure vs. Normal Conditions

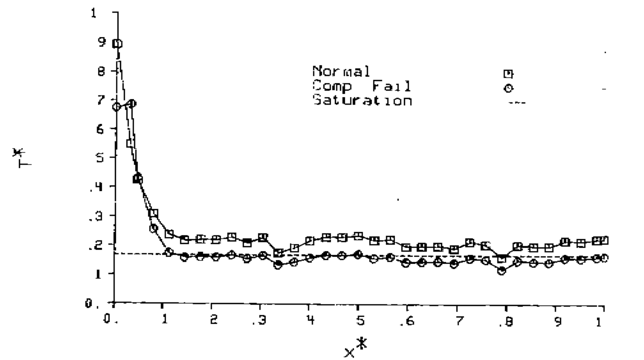

Figure 10. Normalized Steady State Temperatures Along Condenser of the Failure Due to Compressor Leakage vs. Normal Conditions