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ABSTRACT

To ensure the reliability of the newly designed reciprocating compressor for a refrigerator, new methodologies were investigated, including the accelerated life test (ALT) using the new definitions of failure rates and B_1 life figures. Applying conservation of mass and energy, loads such as the pressure difference between the condensing and evaporator pressure were determined in the Mollier diagram. An accelerated life test setup--a simplified vapor-compression cycle--was fabricated. The failure mode of the ALT results was the compressor locking due to the fracture of the suction reed valve, which was identical to that of the suction reed valve in the marketplace. The failure rate and B_1 life of the compressor were 3.63 percent per year and 1.5 years, respectively. The root cause of the fracture was the impact of the valve plate in the suction port. A corrective action plan included the redesign of the suction reed valve and the valve plate modified by rounding the corners and introducing a brushing process. The failure rate and B_1 life of the compressor now can be guaranteed as 0.06 percent per year and 12.9 years, respectively. These methodologies are effective in the reliability design of the product.

1. INTRODUCTION

The basic function of a refrigerator is to store fresh and/or frozen foods. Refrigeration is the thermodynamic process of moving heat energy from the evaporator to the condenser in a vapor compression cycle. This cycle consists of four important components—the compressor, condenser, capillary tube and evaporator. For a refrigerator, a reciprocating compressor provides enough pressure so the refrigerant can be condensed in the condenser. Most compressor companies are making every effort to develop more efficient high-volumetric system. At the same time, pressure for cost reduction of the product leads companies to seek cheaper and more reliable parts. As attempts are made to provide a newly designed compressor, reliability should be evaluated by a proper testing method.

Product liability law now requires manufacturers to design products more stringently in the Europe Union and the United States. Products with minor design flaws may result in recalls and loss of brand name value. Preventing massive recalls is an important factor in the product development process of design, production, shipping and field testing. Conventional methods, such as product inspection, rarely identify reliability problems occurring in marketplace. Designing for maximum reliability requires extensive reliability testing at each development step. As a result, the cost of quality assurance and appraisal can increase significantly.

In reliability testing, most global companies focus on the accelerating life test (ALT). This method can help shorten the product development cycles, costs less money, and clarify diverse design faults. However, there are some caveats of using ALT: Any failures after ALT may not represent those occurring under field conditions. This problem usually arises because of the inconsistency of the direction and magnitude of the load, such as force or pressure in system dynamics. Moreover, the number of test samples and the test times are insufficient to uncover infrequent failure modes. ALT should be performed with sufficient samples and test time, and ALT equipment can and should be designed to match product loads.

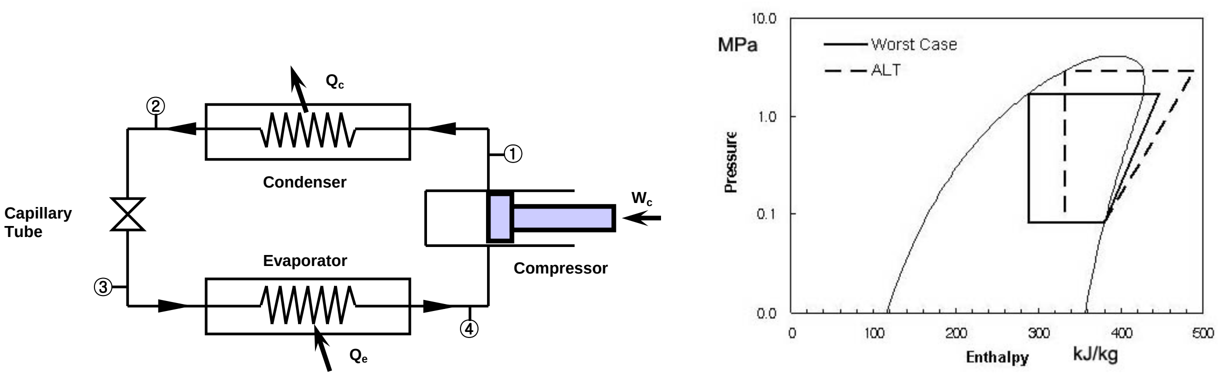
When a rotary compressor was abnormally locking in 1987, global companies, such as GE and Matsushita, experienced massive recalls of the compressor. As the oil sludge in the refrigeration system blocked the capillary tube, the cooling capacity of the refrigerator decreased. In the compressor development process, reproducing this failure mode and preventing the blocking of this tube were very important to the reliability of the refrigerator. However, reliability testing methods such as the ALT, was used at that time.

The compressor is a key component of the vapor-compression system. To ensure the reliability of newly designed compressors, new methodologies for ALT -- the theoretical background of a new failure rate and *Bx* life--were introduced (Ryu, 2005a). The procedures for the compressor reliability design can be summarized as (1) load analysis of the refrigeration cycle; (2) fabrication of ALT equipment; and (3) performance of several ALT tests to predict reliabilities. For the loads of the compressor, it is effective to use the traditional thermodynamic cycle model (Woo, 1992) and refrigerant property at each state (Downing, 1974). ALT equipment can be fabricated on the basis of load analysis.
In this paper, we investigate the reliability of the design of the compressor and suggest new methodologies for ALT. First, we analyzed the pressure conditions in a vapor-compression cycles using principles of conservation of mass and energy. After a sequence of reliability testing, including ALT, we evaluated the compressor reliability with the new failure rate and *Bx* life. Finally, we prove the effectiveness of these methodologies for main component reliability design.

2. ANALYSIS

2.1 Load Analysis of Compressor

A domestic-refrigerator consists of a compressor, a condenser, a capillary tube and an evaporator. The vapor-compression refrigerator cycle receives work from the compressor and transfers heat energy from the evaporator to the condenser. The capillary tube is used to reduce the high pressure of the refrigerant in the condenser to the low pressure in the evaporator. In a refrigerator design, it is important to determine both the condensing pressure, P_c and evaporating pressure, P_e. These pressures depend on ambient conditions and heat exchanger capacity in the design stage of the system (Figure 1).



(a) A vapor-compression cycle (b) P-*h* diagram

Figure 1: Schematic diagram for a vapor compression system

The mass flow rate of refrigerant in a compressor can be modeled as

$$\dot{m} = PD \times \frac{\eta_v}{v_{suc}} \tag{1}$$

where *PD* is the volume flow rate, η_v is the volumetric efficiency, and ν_{suc} is the specific volume. The mass flow rate of refrigerant in a capillary tube can be modeled as (Whitesel, 1957)

$$\dot{m}_c = A\left[\frac{-\int_{P_2}^{P_3} \rho \, dP}{\frac{2}{D} f_m (L_3 - L_2) + \ln\left(\frac{\rho_2}{\rho_3}\right)}\right]^{0.5} \tag{2}$$

where *A* is the cross area of the capillary tube, ρ is the refrigerant density, f_m is the mean friction coefficient, $L_3 - L_2$ is the capillary length of the two-phase interval, η_v is the volumetric efficiency, and ν is the specific volume.
Applying conservation of mass, the mass flow rate can be determined as,

$$\dot{m} = \dot{m}_c \tag{3}$$

The heat transfer in the condenser is given by:

$$Q_c = \dot{m}(h_1 - h_2) = (T_c - T_o)/R_c$$

(4)

The heat transfer in the evaporator is:

$$Q_e = \dot{m}(h_4 - h_3) = (T_i - T_e)/R_e \tag{5}$$

When nonlinear equations (3), (4) and (5) are solved simultaneously, the mass flow rate, $\dot{m}$; evaporator temperature, T_e; and condenser temperature, T_c can be determined. Since the saturation pressure, P_{sat} is a the function of temperature, the evaporator pressure, P_e (or condenser pressure P_c), can be obtained as:

$$P_e = f(T_e) \tag{6}$$

The stress (or life) of the compressor depends on the pressure difference suction pressure, P_{suc}, and discharge pressure, P_{dis}. That is,

$$\Delta P = P_{dis} - P_{suc} \cong P_c - P_e \tag{7}$$

For medium stress, the life-stress model (LS model) (McPherson, J.W., 1989) can be modified as

$$T_f = A(S)^{-n} \exp\frac{E_a}{kT} = A(\Delta P)^{-n} \exp\frac{E_a}{kT} \tag{8}$$

where A is constant, T_f is the time to failure, k is Boltzman's constant, E is the activation energy, T is the absolute temperature and n is the quotient. The acceleration factor (AF) is given by:

$$AF = \left(\frac{S_1}{S_0}\right)^n = \left(\frac{\Delta P_1}{\Delta P_0}\right)^n \left[\frac{E_a}{k}\left(\frac{1}{T_0} - \frac{1}{T_1}\right)\right] \tag{9}$$

where S_1 (or ΔP_1) is medium stress (or pressure difference), and S_0 (or ΔP_0) is normal stress.

2.2 Theoretical Background of A New Definition of B_x Life for Accelerated Life Test

In a traditional textbook (Lee, 2003), the characteristic life is defined as:

$$\eta^\beta \equiv \frac{\sum t_i^\beta}{r} \cong \frac{n \cdot h^\beta}{r} \tag{10}$$

where β is the shape parameter in a Weibull distribution. As a product (or part) reliability improves, there are usually no failures in the test. It is often impossible to evaluate the characteristic life in Equation (11). When the failed sample number is below four, it follows the Poisson distribution (Ryu, 2005b). For a 60 percent confidence level, the characteristic life can be redefined as

$$\eta^\beta \cong \frac{1}{r+1} \cdot n \cdot h^\beta \tag{11}$$

To introduce the B_X life in the Weibull distribution, the characteristic life can be modified as

$$L_B^\beta \cong x \cdot \eta^\beta = \frac{x}{r+1} \cdot n \cdot h^\beta \tag{12}$$

where L_B = B_X life and x = 0.01X, on the condition that $x \le 0.2$. To assess the B_X life with about a 60 percent confidence level, the number of test samples can be derived by rearranging in Equation (12):

$$n \cong \frac{1}{x} \cdot (r+1) \cdot \left(\frac{L_B}{h}\right)^\beta \tag{13}$$

This equation is valid on the condition that the durability target, $h^* = h/L_B \ge 1$

3. EXPERRIMENT

The suction reed valves of the compressors in the marketplace were cracking and fracturing, as shown in Figure 2. Because the compressor was locking up, the customer would ask to have the refrigerator replaced. To solve the problem, it was very important to reproduce the failure mode of the suction reed valve. ALT is used.

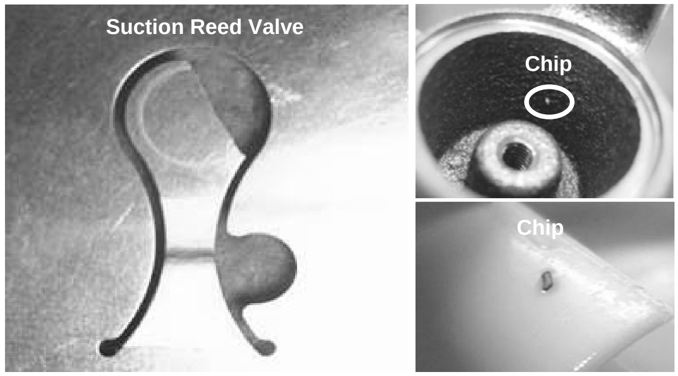


Figure 2: The fracture of the compressor suction reed valve in the marketplace

The ALT equipment was a simplified vapor-compression cycle and fabricated as shown in Figure 3. It consisted of an evaporator, compressor, condenser, and capillary tube. The inlet to the condenser section was at the top and the condenser outlet was at the bottom. At the condenser inlet, a quick coupling and the high-side pressure gauge pressure were installed. The 10-gram XH9 refrigerator dryer at the condenser inlet was mounted vertically. A thermal switch is attached to the condenser tubing at the top of the condenser coil to control the condenser fan. The evaporator inlet is at the bottom. At a location near the evaporator outlet, a quick coupling and pressure gauge were installed to enable access to the low side for evacuation and refrigerant charging.

The condenser outlet connected the evaporator outlet with the capillary tube. The compressor mounted on the rubber pads was connected to the condenser inlet and evaporator outlet. A fan and two 60 Watt lamps maintained the room temperature within the insulated (fiberglass) box. The thermal switch attached on the compressor top controlled the 30 *cfm* Suntronix axial fan, Model ST1238.

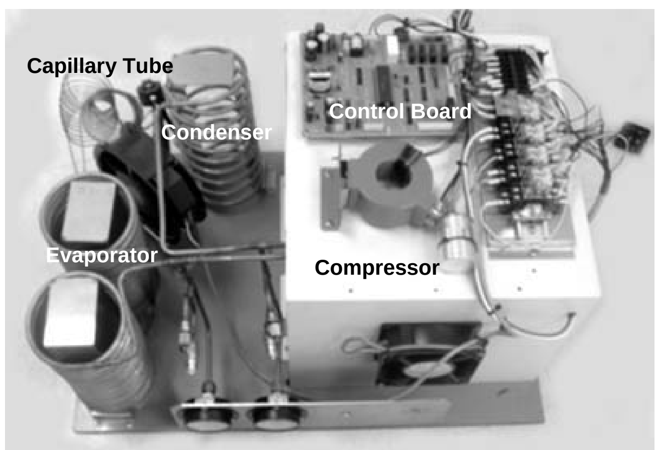


Figure 3: Equipment of accelerated life test

After the leak test, the equipment was evacuated from both high and low sides to 0.01 torr of the static capability of the vacuum pump. It was charged with refrigerant through the valve on the low side of the equipment to an instantaneous pressure of 30 kg/cm^2. The test conditions and test limits were set up on the control board. As the test began, the high-side and low-side pressures could be observed on the pressure gauge (or display monitor).

Generally, the operating temperature of the compressor is 0 ~ 50℃, 0 ~ 85 % relative humidity and 0.2 ~ 0.24G vibration. The normal operating cycles for one day are approximately twenty-two; the worst case is ninety-eight times. Under worst-case, the objective compressor cycles for ten years are 73,000 cycles (Table 1).

Table 1: Operating number of a reciprocating compressor

Item	Operating cycle (times)			
	1 day		10 years	
	Normal	Worst	Normal	Worst
Compressor	22	98	80,300	357,700

From the test data, the normal operating pressure is 13.0 kgf/cm^2 and *AF* design pressure is 30.0kgf/cm^2. Assume quotient n are 2, *AF* is approximately 5.3 in Equation (4).

Table 2: ALT conditions in a vapor compression cycles

System conditions		Worst case	ALT	AF
Pressure, kg/cm^2	High side	13.0	30.0	5.3
	Low side	0.0	0.0	
	ΔP	13	30	
Temp., ℃	Dome Temp.	90	120	3.9
		-		20.9

The test cycles and the numbers of sample (Ryu, 2005c) used in ALT were calculated as follows:

$$n \cong (r+1) \cdot \frac{1}{x} \cdot \left(\frac{L_B}{AF \cdot h} \right)^{\beta} \tag{14}$$

where *r* is the number of failures; *n* is the number of test sample; *x* is 0.01; *AF* is the acceleration factor; *h* is testing cycles; and L_B is the target B_X life.
Assuming the shape parameter β is 1.9, the test cycles and test sample numbers calculated in Equation (14) were 39,000 cycles and 20EA, respectively. The ALT was designed to assure a B_1 of ten years life with about a 60 percent level of confidence if no unit in it fails during 39,000 cycles (Ryu, 2005d).

4. RESULTS AND DISCUSSIONS

4.1 Validity of the accelerated life test and failure analysis

Figure 4 shows failed products in the field and after accelerated life tests. In the photo, the shape and location of the broken piece in the failed market products are similar to those in the ALT results.

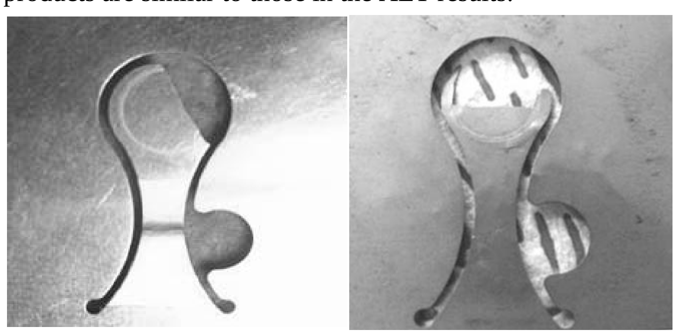


Figure 4: Failure of suction reed valve in marketplace

Figure 5 represents the graphical analysis of the ALT results and market data on a Weibull plot. For the shape parameter, the guess value in the previous ALT and the final value obtained on the Weibull plot was 1.9, respectively. We conclude that these ALT methodologies were valid for reproducing the market failure. (1) The location and shape of the fractured suction reed valve of market and ALT results were similar for both the market and test pieces. (2) On the Weilbull plot, the shape parameters of the ALT results, β_1 and market data, β_2, were similar.

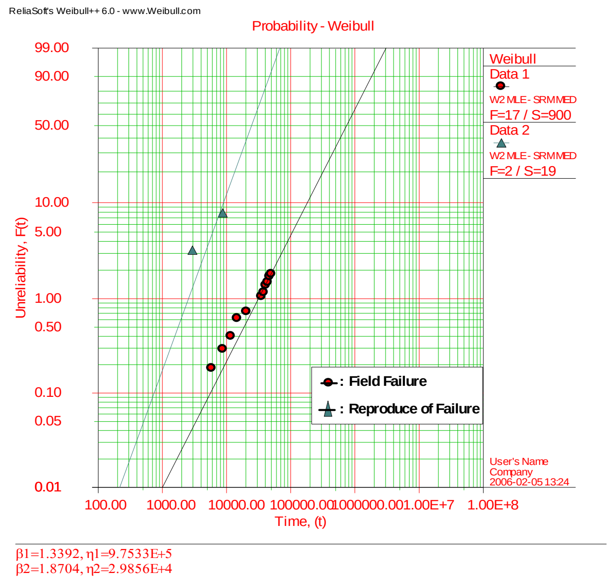


Figure 5: Field data and results of accelerated life test on weibull chart

The fracture of the suction reed valve comes from its weak structure. It (1) had an overlap with the valve plate; (2) used a weak material (7C); and (3) the valve plate had a sharp edge (Figure 6). When the suction reed valve impacted the valve plate continually, it fractured easily. The dominant failure mode of the compressor was leaking and locking due to the cracking and fracturing of the suction reed valve.

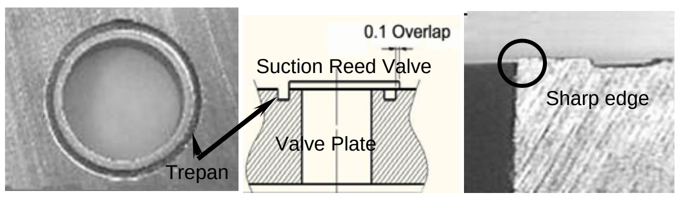
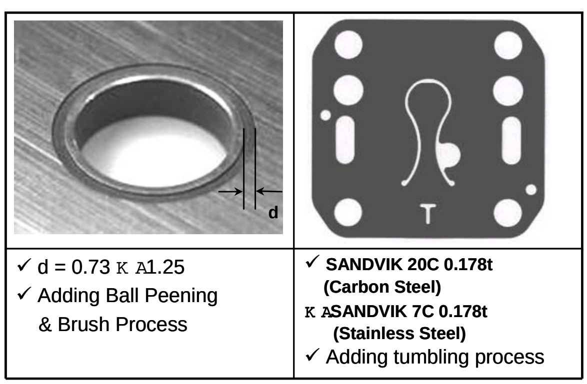


Figure 6: Structure of suction reed and valve plate

4.2 Corrective action plan and life prediction

Figure 7 shows the redesigned suction reed valve and the valve plate. They control the refrigerant gas in the process of suction and compression. The suction reed valve required high bending/impact fatigue properties. The modified design points were: (1) increasing the trespan size of the valve plate, *d*, from 0.73 mm to 1.25 mm (2) changing the material property from 20C to 7C; (3) adding a ball peening and tumbling process (Figure. 4).



(A) Valve plate and (B) Suction reed valve

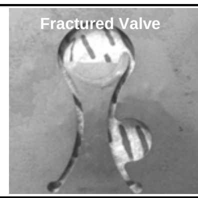
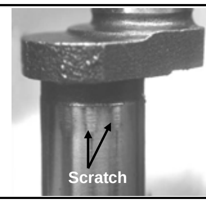
FIGURE 7. REDESIGNED SUCTION REED VALVE AND VALVE PLATE

Table 3 shows the improved design of the suction reed valve based on the ALT results. One sample in the first ALT (n=20) failed in 8,687 cycles. The confirmed value, β, in Figure 5 was 1.9. The shapes and locations of the failure in samples from the first ALT and the marketplace were identical (see Figure 4). This ALT methodology was very effective in reproducing the fracture of the suction reed valve that was claimed in the marketplace. The modified design points were (1) the trespan size of the valve plate, *d*, from 0.73 mm to 1.25 mm; (2) the material property from 20C to 7C; and (3) the added ball peening and tumbling process. However, the calculated life of the newly designed compressor did not reach the target life of B_1 ten years.

For the second ALT, the test sample number calculated in Equation (14) was 30EA and the mission cycles were 32,000. Three samples failed in 17,000 cycles. However, the calculated life of the newly designed compressor still did not reach the target life of B_1 of ten years. From the failure analysis, the root cause of the failed compressor was attributed to a scratch on the crank shaft. The modified design points required heat treating the surface of the crank shaft. To improve compressor reliability, a second ALT was implemented with the modified design.

For the third ALT, the test cycles and test sample number calculated in Equation (14) were 23,000 cycles and 60EA. In the third ALT results, it was found that the compressor did not crack and fracture until 29,000 cycles of testing.

Table 3: Results of the ALTs results

	ALT (1)	ALT (2)	ALT (3)
	Initial Design	Design Improvement (1)	Design Improvement (2)
In 23,000 Cycles, Crack & Fracture of Blade is less than 1.	✓ **8,687 cycles: 1/20 (5.0 %)** ✓ **8,687 cycles: 19/20 Suspension**	✓**17,000 cycles: 3/30 (10.0 %)** ✓**17,000 cycles: 27/30 Suspension**	✓ **23,000 Cycles: 30/30 OK** ✓ **29,000 Cycles: 30/30 OK**
Suction Reed Valve Structure			-
Material & Spec.	✓ **SANDVIK 20C 0.178t (Carbon Steel)** ⇨ **SANDVIK 7C 0.178t (Stainless Steel)**	✓ **FCD500 + No Heat Treatment** ⇨ **FCD500 + Heat Treatment**	✓ **SANDVIK 20C 0.178t (Carbon Steel)** ✓ **FCD500 + Heat Treatment**

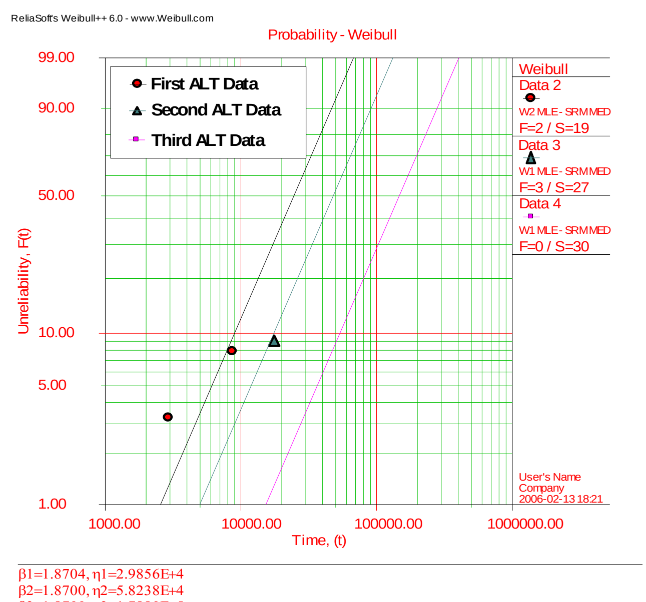


Figure 8: Result of ALTs plotted in weibull chart

The B_x life of the compressor was calculated as:

$$B_x \cong \frac{h \cdot AF}{L_B} \cdot \left(\frac{x \cdot n}{r+1}\right)^{\frac{1}{\beta}} \tag{15}$$

The B_1 lives of the compressor in the first and second ALTs were 1.5 and 2.5 years, respectively. Thus, the B_1 life of the newly designed compressor is 1.3 times that of the current one. Table 4 shows the results obtained from the third ALT. The B_1 life of the redesigned compressor using Equation (15) and Table 3 was 12.9 years. When the design of the current compressor was compared with that of the new one, the B_1 life expanded about 4.7 times, from 1.5 years to 12.9 years. It was found that the design improvements of redesigning the suction reed valve and reinforcing the crank shaft were very effective in expanding the reliability of the newly designed compressor.

Table 4: Results obtained by the third ALT

Factor	AF	β	h	r	L_B	n
Values	20.9	1.9	29,000	0	357,700	60

Assuming normal and maximum pressure differences are 13 and 30 kgf/cm^2, the quotient n is 2.0. The time to failure of the compressor in Equation (3) can be modeled as follows:

$$T_f \cong A \cdot (\Delta P)^{-2.0} \left(\frac{E_a}{kT}\right) \tag{16}$$

where E_a is 0.56 eV and k is 8.62 x 10^{-5} (eV/deg). This is a well-known value in calculating the acceleration factor of the chemical reaction.

5. CONCLUSIONS

To improve the reliability of the newly designed compressor in refrigerators, we have examined the failure mode and predicted the life of the compressor using the accelerated life test (ALT). The following conclusions were obtained:

(1) The suction reed valve in the marketplace was experiencing frequent failures in the field due to fracturing. The root causes of the failed suction reed valve in the refrigerator were (1) an overlap with the valve plate; (2) a weak material (7C); and (3) the sharp edge of the valve plate.
(2) To ensure the reliability of the newly designed compressor, Design improvements, such as redesigning the suction reed valve and heat treating the crank shaft, was implemented. The ALT methodology, using new concepts such as sample size, failure rate and Bx life, were very effective to expand the compressor reliability.
(3) In the third ALT, the failure rate and B_1 life of the redesigned compressor based were 0.05 percent per year and 14.2 years, respectively.

REFERENCES

Ryu, D. S., Chang, S. W., 2005, Novel Concept for Reliability Technology, *Microelectronics Reliability*, pp. 611-622.
Woo, S. W., 1992, A Simplified Cycle Simulation Model for the Vapor-Compression Refrigeration System Utilizing R-134a," *Master thesis*.
Domanski, P., 1986, Modeling of Heat Pump Charged with a Non-Azeotropic Mixture, NBS Report No. 1218.
Fisher, S. K., Rice, C. K., 1983, The Oak Ridge Heat Pump Model, ORNL/CON-80/R1.
Downing, R. C., 1974, Refrigerant Equation, *ASHRAE Transaction*, Vol.80, Part II, pp. 158-169.
Whitesel, W. A., 1957, Capillary Two-Phase Flow, Part II, Refrigerating Engineering, vol. 65, p. 35.
McPherson, J.W., 1989, Accelerated Testing, *Packaging and Electronic Materials Handbook*, ASM International, vol. 1, p.888, 892.
Lee, S. Y., 2003, *Reliability Engineering*, Seoul, Hyung Seol, pp 9-15.