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# Improving Lifetime of Domestic Compressor Subjected To Repetitive Internal Stresses

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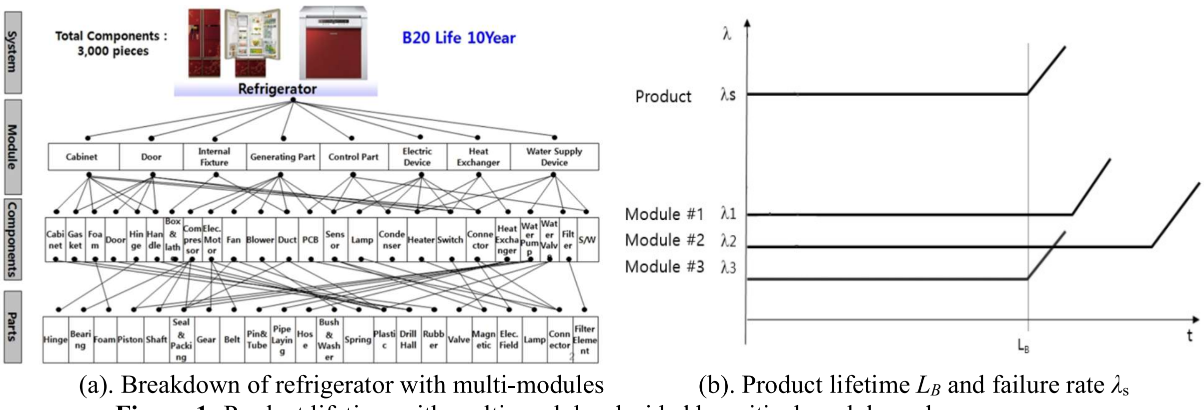
## ABSTRACT

To enhance the lifetime of a mechanical system such as compressor, parametric accelerated life testing (ALT) as a systematic reliability method is proposed. It consists of (1) a parametric ALT plan formed on BX lifetime, (2) load examination for ALT, (3) a customized parametric ALTs with design alternatives, and (4) assessment of the compressor design to secure the objective BX lifetime is fulfilled. As an experiment instance, newly designed reciprocating compressors for French-door refrigerators returned from the marketplace were investigated. The refrigerators had been making disagreeable noise and vibrations, creating the consumer to ask for replacing their refrigerators. As the vibration level of the problematic refrigerators was recorded in an anechoic chamber, the result was 0.35g. Upon closer inspection of the refrigerator, the noise originated from the reciprocating compressor where there was interference between the compressor upper shell and the stator frame. To reproduce the compressor problem from the marketplace, a parametric ALT was performed. The failure mode during the ALTs for the compressor was alike to those of the failed refrigerators from the field. As a corrective action, the stator frame in the compressor system was redesigned to increase the minimum gap between the compressor upper shell and the stator frame. During the second ALT, there were no issues. After parameter ALTs were used to develop corrective action plans, the lifetime of the compressor was reassured to have a B1 life 10 years.

## 1. INTRODUCTION

A refrigerator supplies chilled air from the evaporator to both the freezer and refrigerator sections where frozen or fresh food is stocked. It utilizes a vapor-compression refrigeration cycle (Sonntag, 2007). A refrigerator is composed of approximately 3,000 components, including cabinet, doors, sensors and controls, internal fixtures (shelves and drawers), generating parts (motor or compressor), heat exchanger (condenser and evaporator), a water supply device, and other components. A refrigerator's lifetime is anticipated to be no less than B20 life 10 years. The product lifetime is determined by module #3 such as compressor, which has design faults (Figure 1).




(a). Breakdown of refrigerator with multi-modules (b). Product lifetime $L_B$ and failure rate $\lambda_s$

**Figure 1:** Product lifetime with multi-modules decided by critical module such as compressor

To improve the efficiency of a French-door refrigerator, it was fitted with a newly designed, high efficiency compressor. A compressor expands the refrigerant pressure in a vapor-compression refrigeration cycle. In the process, compressor systems are subjected to repeated stress due to internal pressure loads. If there is a design fault in the structure that causes unexpected vibrations when dynamic loads are applied, the compressor start making noises before it reaches its expected lifetime. In such case, consumers may request to replace their refrigerators.
Robust design methods, including statistical design of experiments (DOE) (Montgomery, 2013) and the Taguchi approach (Taguchi, 1978), were evolved to conduct optimal design for mechanical systems. Because numerous design parameters for a mechanical structure have to be examined, the Taguchi product array demands an enormous experimental design over a range of conditions to ensure a reliable of mechanical system. To resolve rare product failures in the field, another engineering approach includes strength of materials and the finite element method (FEM) (Hrennikoff, 1941). Numerous engineers believe uncommon product failures can be evaluated by: (1) mathematical modeling utilizing Newtonian methods, (2) estimating the time response of the system for dynamic loads and discovering the product stress, (3) utilizing the rain-flow counting method for von Mises stress, and (4) approximating system damage which uses the Palmgren-Miner's rule. However, utilizing an analytical methodology that can create a closed-form, precise consequence often requires inducing numerous presumptions that are not capable of recognizing multi-module product failures – fatigue – due to design defects that occurs from atomic level.
Most global manufacturers have evolved a reliability testing method such as parametric ALT. This methodology can help clarify diverse design faults within shortening test cycles. However, there are some things to be considered in using ALT. Any failures after ALT may not represent those occurring under field states. This problem usually occurs because of the inconsistency of the direction and magnitude of the load and no straightforward failure model. Furthermore, the number of test samples and the test times are inadequate to expose rare failure modes under accelerated conditions. ALT equipment also must be properly designed to match product loads.
This study develops a new methodology as parametric ALT for improving the lifetime of domestic compressor subjected to repeated internal stresses. As product lifetime is targeted, parametric ALT for mission time can identify the design faults of a product such as vibration at a specific frequency from the induced pressure loads and modify them. The lifetime of a newly designed compressor might be improved. It includes: (1) a parametric ALT plan formed on product BX lifetime, (2) a load examination for ALT, (3) a customized parametric ALTs with the design alternatives, and (4) an assessment of whether the design(s) of the product fulfill the objective BX lifetime. As an experiment instance, newly designed reciprocating compressor in French-door refrigerators returned from the marketplace will be investigated.

# 2. PARAMETRIC ALT FOR A COMPRESSOR

## 2.1 Index of product lifetime such as BX life

A suitable index of product lifetime, such as BX life, needs to be chosen for running parametric ALT. BX life is expressed as the time at which $X$ percent of the items in a population will have been unsuccessful. For instance, if a product has B20 life 10 years, then 20% of the population will have failed during 10 years of working time. 'BX life Y years' helps to correctly decide the cumulative failure rate of a product and its lifetime replying to market circumstances. The meantime to failure (MTTF) as the reverse of the failure rate cannot represent the lifetime, because it means around B60 life and its failure rate until lifetime is too large. BX life reflects the designer's feelings more adequately than MTTF.

## 2.2 Putting an Whole Parametric ALT Plan

Reliability can be explained as the ability of a system to work under stated conditions for a specified period of time. It can be clarified by a diagram named a "bathtub curve" that consists of three regions (Klutke, 2015). First, there is a decreasing failure rate in the premature life of the product ($\beta < 1$). Then, there is a constant failure rate ($\beta = 1$) in its middle. Eventually, there is an increasing failure rate to the end of the product's life ($\beta > 1$). If a product pursuits the bathtub curve, it may have difficulties succeeding from the marketplace because of the lofty failure rates and shortened lifetimes due to design defects. Consequently, manufacturers incur financial losses for the whole product life cycle. They can definitely enhance the design of a product by putting reliability targets for products to (1) lessen early failures, (2) decrease random failures during the product working time, and (3) increase product lifetime. As the design of a mechanical product is improved, the failure rate of the product from the marketplace decreases and its lifetime increases. For such circumstances, the conventional bathtub curve can be changed to a linear line with the shape parameter β (Figure 2).

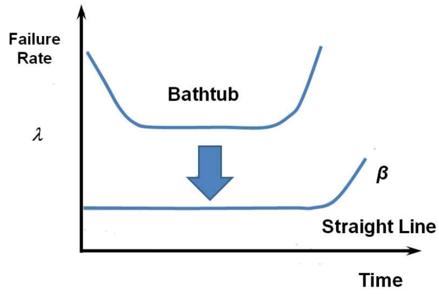



**Figure 2:** Bathtub curve and straight line with slope $\beta$

Because the linear line with low failure rate pursuits the exponential distribution, the reliability of the mechanical product might be quantified from the multiplication of product lifetime $L_B$ and failure rate $\lambda$ as follows:

$$R(L_{BX}) = 1 - F(L_{BX}) = e^{-\lambda L_{BX}} \cong 1 - \lambda L_B \tag{1}$$

where $R(\cdot)$ is reliability function, $F(\cdot)$ is cumulative distribution function, $\lambda$ is failure rate, and $L_B$ is BX lifetime

A refrigerator compressor is a mechanical device that increases the refrigerant pressure by reducing the cylinder volume through crank mechanisms. To keep the intended function, the product for objective BX lifetime should be designed to properly operate under the range of conditions subjected to it by consumers. After targeting the product lifetime $L_B$, the engineer should recognize design faults through parametric ALT and alter them.

In targeting the lifetime of a compressor in refrigerator, there is 1) a new module, 2) a modified module, and 3) a similar module where there is no alternation to the earlier design on the basis of market data. The compressor discussed in section 2.4 can be considered as an altered module because consumers want to enhance the energy efficiency and compressor parts is redesigned. To reply to customer request, a compressor lifetime might be put to B1 life 10 years.

## 2.3 Failure Mechanics and Accelerated Testing

A compressor increases the refrigerant pressure in a vapor-compression refrigeration system. Thus, compressor systems are subjected to repeated stress due to internal pressure loads. If there is design fault in the structure that causes vibration where the loads are exerted, the compressor may suddenly fail over its anticipated lifetime. After recognizing the product failure by reliability test such as parametric ALT, an engineer can optimally design the compressor.

The most important issue in the reliability testing of compressor is how fast the potential failure mode can be attained. To do this, it is required to express a failure model and determine the associated coefficients. First of all, we can configure the life-stress (LS) model, which incorporates stresses and reaction parameters. This equation can explain various failures in the mechanical structure. Product failure originates from a micro-depletion (void) regardless of whether it is a mechanical or electronic system, the life-stress model might be expressed as the following processes.

When an electric magneto-motive force, $\xi$, is applied, the impurities in materials, generated by electronic movement, easily migrate because the barriers of junction energy are lessened. Linear transport processes can be expressed as follow:

$$J = LX \tag{2}$$

where $J$ is a flux vector, $X$ is defined as (driving or thermodynamic) force, and $L$ is a phenomenological transport coefficient which makes a connection between fluxes and forces.

For example, it is utilized for solid-state diffusion of impurities in silicon: 1) electro-migration-induced voiding; 2) build-up of chloride ions; 3) trapping of electrons or holes.

So the life-stress model might be expressed as (Grove, 1967):

$$J = \left[aC(x-a)\right]\cdot \exp\left[-\frac{q}{kT}\left[w - \frac{1}{2}a\xi\right]\right]\cdot v$$

[Density/Area]·[Jump Probability]·[Jump Frequency]

$$= -[a^2 v e^{-qw/kT}]\cosh\frac{qa\xi}{2kT}\frac{\partial C}{\partial x} + [2ave^{-qw/kT}]C\sinh\frac{qa\xi}{2kT}$$

$$\cong \Phi(x,t,T)\sinh(a\xi)\exp\left(-\frac{Q}{kT}\right)$$
$$= B\sinh(a\xi)\exp\left(-\frac{Q}{kT}\right) \quad . \tag{3a}$$

Contrastingly, a reaction process that is depending on speed might be defined as:

$$K = K^{+} - K^{-} = a\frac{kT}{h}e^{-\frac{\Delta E - \alpha S}{kT}} - a\frac{kT}{h}e^{-\frac{\Delta E + \alpha S}{kT}} = 2\frac{kT}{h}e^{-\frac{\Delta E}{kT}} \cdot \sinh\left(\frac{\alpha S}{kT}\right)$$
$$= B\sinh(aS)\exp\left(-\frac{\Delta E}{kT}\right) \tag{3b}$$

So the reaction rate, *K,* from Equations (3a) and (3b) can be outlined as:

$$K = B\sinh(aS)\exp\left(-\frac{E_a}{kT}\right) \tag{4}$$

where $B$ is constant

If the reaction rate in Equation (4) takes a reverse function, the generalized stress model can be obtained as

$$TF = A[\sinh(aS)]^{-1}\exp\left(\frac{E_a}{kT}\right) \tag{5}$$

The hyperbolic sine stress term in Equation (5) increases the stress as follows: (1) $S^{-1}$ initially in low stress effect, (2) $S^{-n}$ in medium stress effect, and (3) $(e^{as})^{-1}$ in high stress effect. Because accelerated testing will be performed in the medium range, Equation (5) is expressed as follows:

$$TF = A(S)^{-n}\exp\left(\frac{E_a}{kT}\right) \tag{6}$$

As the compressor stress is hard to be quantified in accelerated testing, we need to alter Equation (6). The stress of mechanical system comes from the acting force. Especially, we know that the compressor stress may obtain from the pressure difference between suction pressure, $P_{suc}$, and discharge pressure, $P_{dis}$ in the refrigerating cycle (Woo and O'Neal, 2006). For a compressor system, the time-to-failure, TF, can be expressed as

$$TF = A(S)^{-n}\exp\left(\frac{E_a}{kT}\right) = A(F)^{-\lambda}\exp\left(\frac{E_a}{kT}\right) = A(\Delta P)^{-\lambda}\exp\left(\frac{E_a}{kT}\right) \tag{7}$$

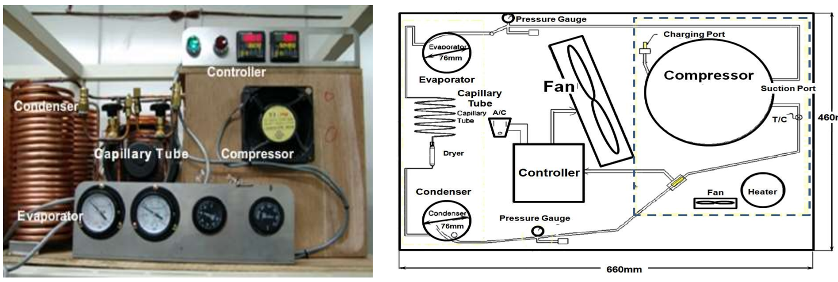
The accelerated factor (AF) from Equation (7) can be redefined as

$$AF = \left(\frac{S_1}{S_0}\right)^{n}\left[\frac{E_a}{k}\left(\frac{1}{T_0} - \frac{1}{T_1}\right)\right] = \left(\frac{F_1}{F_0}\right)^{\lambda}\left[\frac{E_a}{k}\left(\frac{1}{T_0} - \frac{1}{T_1}\right)\right] = \left(\frac{\Delta P_1}{\Delta P_0}\right)^{\lambda}\left[\frac{E_a}{k}\left(\frac{1}{T_0} - \frac{1}{T_1}\right)\right] \tag{8}$$

To get the mission cycle of a parametric ALT from the objective BX lifetime on the test plan, the sample size equation combined with acceleration factors might be derived as (Woo *et al*., 2020):

$$n \geq (r+1)\cdot\frac{1}{x}\cdot\left(\frac{L_{BX}^{*}}{AF\cdot h_a}\right)^{\beta} + r \tag{9}$$

The lifetime of a refrigerator compressor is targeted to have B1 life 10 years. If the objective number of life cycles $L_{BX}$ and $AF$ are given, the mission time can be acquired from Equation (9). The ALT equipment can then be used in accordance with the work of the refrigeration cycle (Figure 3).



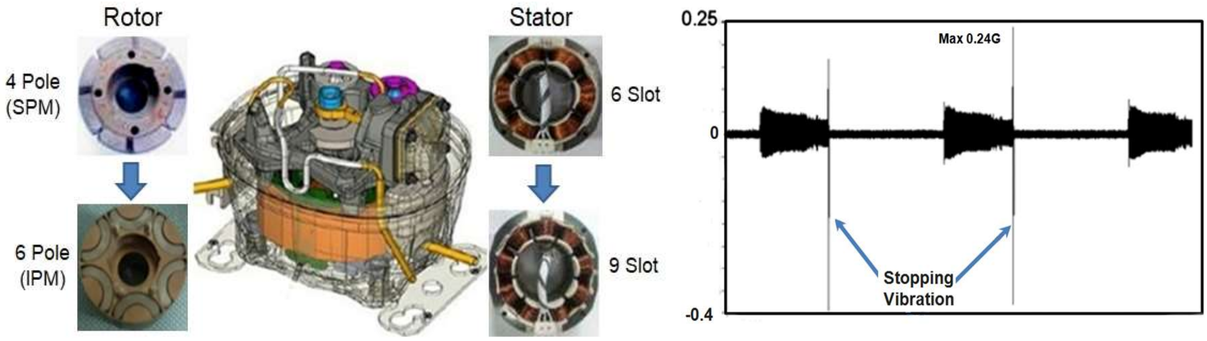

(a) Photograph (b) A drawing of the test system

**Figure 3:** Equipment for the compressor accelerated life tests

In parametric ALTs, the design flaws of the mechanical system can be recognized to fulfill the lifetime target (Woo *e t al*., 2008&2009&2010).

## 2.4 Case Study: Improving the noise of mechanical compressor subjected to repetitive pressure loading

To improve the energy efficiency of a refrigerator, a reciprocating compressor with newly designed rotor and stator was redesigned (Figure 4a). In the field, the refrigerators had been making noise, causing the consumer to request replacement of them (Figure 4b). When the vibration in the problematic refrigerator was measured, the peak vibration level shown in the chart was 0.35g. Because the vibration specifications requirements were for less than 0.2g, the refrigerators with the newly designed compressor did not meet the specifications. Upon closer inspection, we found interference marks on the compressor shell for the failed compressor. We suspected that these marks came from a possible design flaw in the compressor, i.e., the newly designed stator frame. When the compressor operates, the piston moved back and forth within the cylinder assembly that was directly connected to the stator frame. It also supported the piston-cylinder assembly. The cylinder assembly and the stator frame could interfere with the compressor shell because it was not properly designed with enough clearance. To decide the root cause(s) of the failure of this refrigerator experimentally, it was necessary to be reproduced and corrected by a proper ALT. So the compressor might be robustly designed to work under a wide span of consumer usage conditions (Figure 5).




(a) Reciprocating compressor with redesigned rotor and stator (b) A noised (or vibrated) refrigerator

**Figure 4:** Reciprocating compressor and its noise

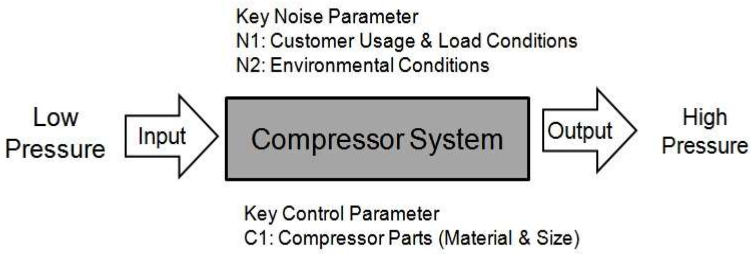
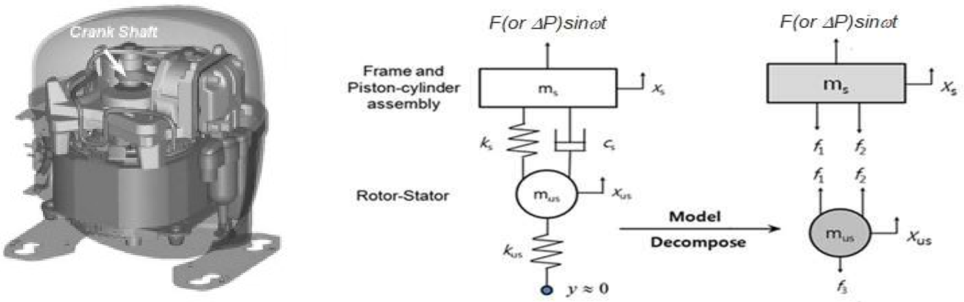



**Figure 5:** Parameter diagram of compressor system

To assess the ride quality of the new designed stator/rotor frame and piston-cylinder assembly, the most practical mathematical model for a vehicle suspension system installed on compressor shell was a quarter car model (Jazar, 2008). Although quarter car model was two degree of freedom (DOF) and four state variables, it served the purpose of determining the vehicle motion mounted on compressor. For all values of the forcing frequency, the presumed model of the vehicle consisted of the sprung mass and the un-sprung mass, separately. The sprung mass, $m_s$, represents 1/4 of the body of the vehicle, i.e., the piston-cylinder assembly and stator frame. The un-sprung mass, $m_{us}$, represents the wheel of the vehicle, i.e., the rotor-stator assembly. The cucial suspension was modeled as a spring $k_s$ and a damper $c_s$ in parallel, which attaches the un-sprung to the sprung mass. The suspension spring on compressor shell was modeled as a spring $k_{us}$ and represented the transfer of the road force to the un-sprung mass (Figure 6)




(a) Reciprocating compressors (b) A quarter car model of compressor assembly and its decomposition

**Figure 6:** Compressor assembly and its quarter car model

The governing differential equations of motion for the quarter car model can be expressed as:

$$m_s\ddot{x}_s + c_s(\dot{x}_s - \dot{x}_{us}) + k_s(x_s - x_{us}) = F(or\,\Delta P)\sin\omega t \tag{10}$$

$$m_{us}\ddot{x}_{us} + c_s(\dot{x}_{us} - \dot{x}_s) + (k_{us} + k_s)x_{us} - k_s x_s = 0 \tag{11}$$

So the above equations of motion can be briefly expressed as:

$$\begin{bmatrix} m_s & 0 \\ 0 & m_{us} \end{bmatrix}\begin{bmatrix} \ddot{x}_s \\ \ddot{x}_{us} \end{bmatrix} + \begin{bmatrix} c_s & -c_s \\ -c_s & c_s \end{bmatrix}\begin{bmatrix} \dot{x}_s \\ \dot{x}_{us} \end{bmatrix} + \begin{bmatrix} k_s & -k_s \\ -k_s & k_{us}+k_s \end{bmatrix}\begin{bmatrix} x_s \\ x_{us} \end{bmatrix} = \begin{bmatrix} F(or\,\Delta P)\sin\omega t \\ 0 \end{bmatrix} \tag{12}$$

As a result, Equation (12) can be represented in a matrix form

$$[M]\ddot{X} + [C]\dot{X} + [K]X = F(or\,\Delta P)e^{i\omega t} \tag{13}$$

When equation (13) was numerically integrated, we obtained the time response of the state variables (displacement, velocity, and acceleration) due to the compressor operation, i.e., excitation forces (or pressure difference) $F(or\,\Delta P)e^{i\omega t}$. And the force transmissibility (Q) for all values of the forcing frequency was defined as (Rao, 2017):

$$Q = \frac{F_T}{kY} = r^2\left[\frac{1+(2\zeta r)^2}{(1-r^2)^2+(2\zeta r)^2}\right]^{1/2} \approx \frac{r^2}{(r^2-1)} \quad \text{for small values of damping ratio} \tag{14}$$

The acceleration factor (AF) from Equation (8) is expressed as the ratio between the normal condition and the accelerated condition. When most accelerated testing is carried out in room temperature, AF can be defined as the multiplication of the excitation forces (or pressure difference) and force transmissibility Q. In other words,

$$AF = \left(\frac{S_1}{S_0}\right)^n = \left(\frac{F_1}{F_0}\right)^\lambda = \left(\frac{\Delta P_1}{\Delta P_0}\frac{F_T}{kY}\right)^\lambda = (R\times Q)^\lambda \approx \left(\frac{\Delta P_1}{\Delta P_0}\right)^\lambda \quad \text{for } r \ge 2 \tag{15}$$

For the worst case, the differential pressure anticipated in the compressor would be an operating pressure of 0.38MPa. For the accelerated testing, the differential pressure was increased to 1.35MPa. With a cumulative damage exponent, $\lambda$, of 2, the accelerated factor (AF) was roughly 12.6 utilizing Equation (13) (Table 1).

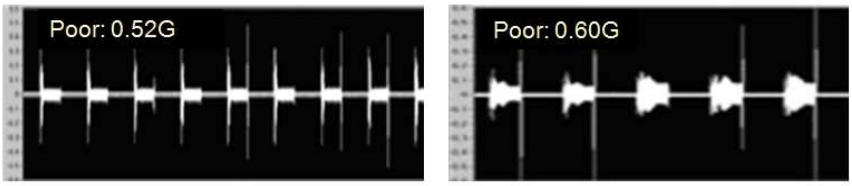
**Table 1:** ALT conditions in refrigeration cycle for refrigerant R600a

| System Conditions | | Worst Case | ALT | AF |
|---|---|---|---|---|
| Pressure (MPa) | High-side | 0.40 | 1.39 | 12.6 |
| | Low-side | 0.02 | 0.04 | |
| | ΔP | 0.38 | 1.35 | |

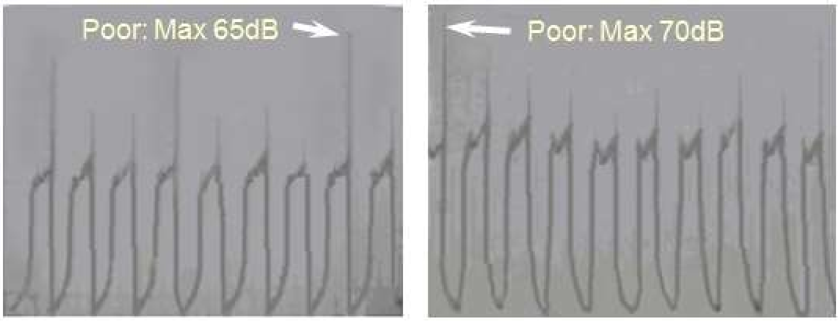
The lifetime target is put to have more than the B1 life 10 years. The reciprocating compressor works cycles on and off between 22 and 74 times per day. With product life cycles for 10 years, $L_B$, the compressor is expected to have up to 270,100 usage cycles. The assumed shape parameter $\beta$ was 2.0. The actual test cycles computed in Equation (9) were 21,400 cycles for a sample size of 100 units. If the parametric ALT for the compressor system fails less than once during 21,400 cycles, it would be reassured to have a B1 life 10 years with about a 60% level of confidence.

## 3. RESUTS AND DISCUSSION

Among one hundred we found two samples with noise and vibrations at 100 cycles in the 1st ALT (Figure 7). The failure sites from the marketplace and the first ALT happened in the upper shell of compressor. Figure 8 is a graphical analysis if the ALT results and market data show on a Weibull plot. For the shape parameter, $\beta$, the approximated value on the chart was 2.0. For the last design, the shape parameter was confirmed to be 1.9.




(a) Vibration of failed samples




(b) Noise of failed samples

**Figure 7:** The vibration and noise of failed compressors in 1st ALT

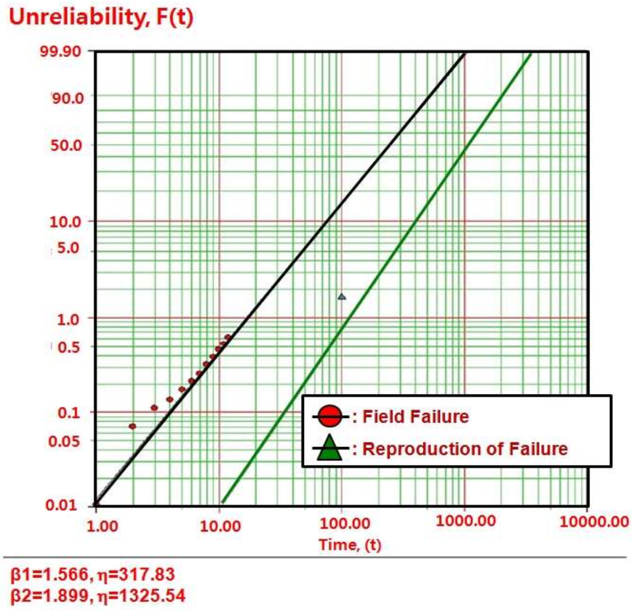
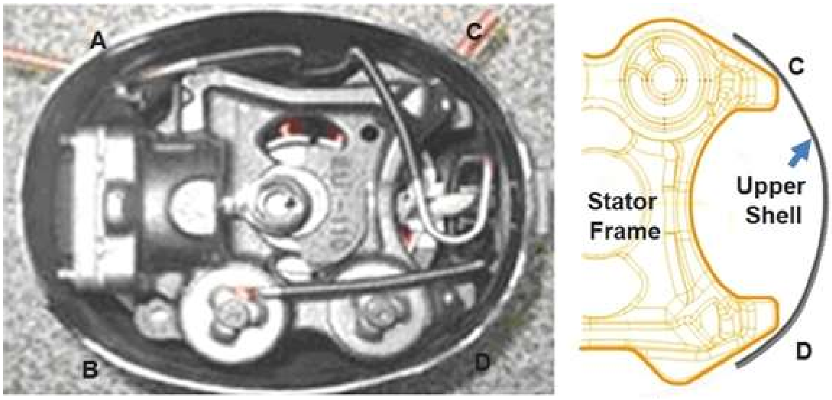



**Figure 8:** Market data and 1st ALT on Weibull chart

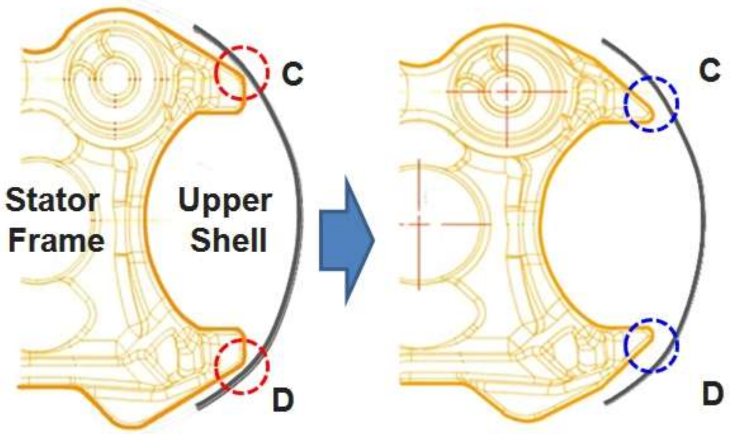



(a) Stator frame mounted in compressor

| A | B | C | D |
|---|---|---|---|
| 8.5 | 9.1 | 3.2 | 2.9 |

(b) Gap between frame and upper shell (Spec.: 6mm↑)
**Figure 9:** Structure of noised compressor in field

As seen in Figure 7 and Figure 9, the noise and vibration in the first ALT originated from the interference between the stator frame and the compressor upper shell. The gap between the frame and the shell was calculated to be approximately 2.9 mm even though the design specification was 6mm.
To prevent the noise and vibration of the compressor as action plan, the stator frame in compressor was redesigned and the gap increased from 2.9 mm to 7.5 mm (Figure 10). As the design flaw was rectified, the parametric design basis of the newly designed samples was shown to have more than the lifetime target - B1 life 10 years. The affirmed value, $\beta$, on the Weibull chart was 1.9. For the second ALT with sample size 100 units, the real test cycles in Equation (9) were 21,400. In the second ALT, there were no design issues in the compressor system until the test was performed to 25,000 cycles. We thus deduced that the altered design parameters obtained from 1st ALT were successful.




(a) Modified stator frame mounted in compressor

| | A | B | C | D |
|---|---|---|---|---|
| Old Part | 8.5 | 9.1 | 3.2 | 2.9 |
| New Part | 8.3 | 8.8 | 7.7 | 7.5 |

(b) Final gap between frame and upper shell (Spec.: 6mm↑)
**Figure 10:** Improved structure of noised compressor in field

Table 2 abridges the ALT results. With the altered design parameters, the last samples of the compressor system were reassured to have the lifetime target – B1 life 10 years.

**Table 2:** Results of ALTs

| **Parametric ALT** | **1st ALT** | **2nd ALT** |
|---|---|---|
| | Initial Design | Final Design |
| In 21,400 cycles, there is no noise and vibration | 100 cycles: 2/100 noise | 21,400 cycles: 100/100 OK<br>25,000 cycles: 100/100 OK |
| Photo | <Compressor><br> | - |

## 4. CONCLUSION

We suggested the reliability methodologies that can find and modify the design problems of mechanical system such as the noise of a newly designed reciprocating compressor in French-door refrigerator. It consists of the inspection of the failed products from the marketplace, a parametric ALT plan formed on BX lifetime, load examination, parametric ALTs with design alternatives, and assessment if the objective BX lifetime is fulfilled. As an experiment instance, newly designed compressor returned from field was studied.

In the field and the 1st ALT, the compressors were noisy and vibrated because of a design flaw – there was interference between the stator frame and the compressor upper shell. By reshaping the stator frame in the compressor, the problematic compressor system was corrected. As a result of this altered design, there were no issues in the 2nd ALT. Consequently, the newly designed compressors were expected to have the lifetime requirement of B1 life 10 years. Through the inspection of returned products in field, load examination, and parametric ALTs with design alternations, we knew that the study of the missing design parameters of compressor system was successful in having more reliable parts with notably longer life.

This systematic reliability method might be relevant to other mechanical systems such as airplane, automobiles, refrigerators, washing machines, construction equipment, etc. To utilize this systematic reliability method, engineers should comprehend why mechanical products such as compressor fail. That is, if there is a design flaw in the structure that causes insufficient stiffness (or strength) when the loads are exerted, the product will fail during its lifetime. Engineers also need to recognize the load attributes of a product and failure model that can model in an atomic size through quantum mechanics. Finally, the parametric ALT can be conducted to meet the required number of mission cycles under accelerated conditions so that engineers will recognize the design problems of a mechanical system such as compressor and modify them.

## NOMENCLATURE

| | |
|---|---|
| $a$ | distance between (silicon) atoms |
| $BX$ | time which will be $X$ percent of the accumulated failure rate, durability index |
| $E_a$ | activation energy, 0.56 eV |
| $F(t)$ | unreliability |
| $h$ | testing cycles (or cycles) |
| $k$ | Boltzmann's constant, $8.62 \times 10^{-5}$ eVdeg-1 |
| $L_B$ | target BX life and $x = 0.01$X, on the condition that $x \le 0.2$ |

| | |
|---|---|
| $n$ | number of test samples |
| $\Delta P$ | pressure difference, MPa |
| $P$ | pressure, MPa |
| $r$ | failed numbers |
| $r$ | frequency ratio ($=\omega/\omega_n$) |
| $S$ | stress |
| $t_i$ | test time for each sample |
| $T$ | absolute temperature, K |
| $X$ | accumulated failure rate, % |
| $x$ | $x = 0.01 \cdot X$, on condition that $x \leq 0.2$ |

**Greek symbols**

| | |
|---|---|
| $\eta$ | characteristic life |
| $\lambda$ | cumulative damage exponent in Palmgren-Miner's rule |
| $\zeta$ | damping ratio ($=c/c_c$) |
| $\xi$ | applied field |
| $\alpha$ | confidence level |

**Superscripts**

| | |
|---|---|
| $\beta$ | shape parameter in a Weibull distribution |
| $n$ | stress dependence, $n = -\left[\dfrac{\partial \ln(T_f)}{\partial \ln(S)}\right]_T$ |

**Subscripts**

| | |
|---|---|
| 0 | normal stress conditions |
| 1 | accelerated stress conditions |
| $a$ | actual |

## REFERENCES


Grove, A. (1967). *Physics and Technology of Semiconductor Device*. 1st ed., Wiley International Edition.

Hrennikoff, A. (1941). Solution of problems of elasticity by the framework method, *Journal of Applied Mechanics*, 8, 169–175.

Jazar, R. (2008). *Vehicle Dynamics: Theory and Application*. Springer, pp. 931-975.

Klutke, G., Kiessler, P.C., Wortman, M.A. (2015). A critical look at the bathtub curve, *IEEE Transactions on Reliability*, *52 (1)*, 125–129.

Montgomery, D. (2013). *Design and Analysis of Experiments*, 8th Ed., Hoboken, NJ: John Wiley.

Rao, S. S. (2017). *Mechanical Vibration*. 6th ed., New York: Pearson Prentice Hall.

Sonntag, R.E., Borgnakke, C. (2007). *Introduction to Engineering Thermodynamics*. York, PA: John Wiley & Sons, Inc,.

Taguchi, G. (1978). Off-line and on-line quality control systems. In: Proceedings of the International Conference on Quality Control, Tokyo, Japan

Woo, S., Pecht, M., O'Neal, D. (2020). Reliability design and case study of the domestic compressor subjected to repetitive internal stresses, *Reliability Engineering & System Safety*, *193(C)*, 106604.

Woo, S., Pecht, M. (2008). Failure analysis and redesign of a helix upper dispenser, *Engineering Failure Analysis*, 15(4), 642-53.

Woo, S., O'Neal, D., Pecht, M. (2009). Design of a hinge kit system in a Kimchi refrigerator receiving repetitive stresses, *Engineering Failure Analysis*, 16 (5), 1655–1665.

Woo, S., O'Neal, D., Pecht, M. (2010). Failure Analysis and Redesign of the Evaporator Tubing in a Kimchi Refrigerator, *Engineering Failure Analysis*, 17 (2), 369–379.

Woo, S., O'Neal, D. (2006). Reliability Design of the Newly Designed Reciprocating Compressor for a Refrigerator, *Proceedings of International Compressor Engineering Conference*.