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Reliability design and case study of a refrigerator compressor subjected to repetitive loads

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ARTICLE INFO

Article history:
Received 18 March 2008
Received in revised form
6 May 2008
Accepted 21 July 2008
Published online 31 July 2008

Keywords:
Refrigeration system
Compression system
Reciprocating compressor
Design
Component
Parameter
Capacity reduction

ABSTRACT

A newly designed crankshaft of a compressor for a side-by-side (SBS) refrigerator was studied. Using mass and energy conservation balances, a variety of compressor loads typically found in a refrigeration cycle were analyzed. The laboratory failure modes and mechanisms were compressor locking and crankshaft wear. These were similar to those of the failed samples in the field. Failure analysis, accelerated life testing (ALT), and corrective actions were used to identify the key reliability parameters. The design parameters of the crankshaft included the hole locations and the groove of the crankshaft used for oil lubrication, crankshaft hardness, and thrust washer interference. Based on the analysis and design changes, the B_1 life of the new design is now over ten years with a yearly failure rate of 0.01 percent. A five step procedure is recommended for parts design.

Compresseur frigorifique assujetti á des charges á répétition : conception vis-á-vis de la fiabilité et étude de cas

Mots clés : Systéme frigorifique ; Systéme á compression ; Compresseur á piston ; Conception ; Composant ; Paramétre ; Réduction de puissance

1. Introduction

Robust design techniques, including statistical design of experiment (SDE) and Taguchi methods (Taguchi, 1976; Taguchi and Tsai, 1992), have been developed by statisticians. Taguchi methods describe the robustness of the system for the evaluation and design improvements in product development, generally referred to as "quality engineering" (Barker, 1986) or "robust engineering" (Wilkins, 2000). For a wide variety of part dimensions, Taguchi's method uses design to avoid random "noise"

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0140-7007/$ – see front matter
doi:10.1016/j.ijrefrig.2008.07.006

Nomenclature

AF	acceleration factor
B_x	durability index
C1	oil lubrication region
C2	starting RPM
C3	crankshaft material
C4	thrust washer dimension, mm
e	effort
e_0	effort under normal stress conditions
e_1	effort under accelerated stress conditions
E_a	the activation energy
f	flow
$F(t)$	unreliability
F	force, kN
$g(x_{noise})$	the probability density function for the noise array
h	testing time (or cycles)
h^*	non-dimensional testing cycles, $h^* = h/L_B \geq 1$
i	current, A
k	Boltzmann's constant, 8.62×10^{-5} eV/deg
KCP	key control parameter
KNP	key noise parameter
$L(x_{con})$	the loss function for control factors
L_B	the target B_x life and $x = 0.01X$, on the condition that $x \leq 0.2$
n	the number of test samples
N1	pressure difference, MPa
ΔP	pressure difference, MPa
ΔP_1	pressure difference under accelerated stress conditions, MPa
ΔP_0	pressure difference under normal stress conditions, MPa
P_c	condenser pressure, kPa
P_e	evaporator pressure, kPa
P_{suc}	compressor suction pressure, kPa
P_{dis}	compressor discharge pressure, kPa
Q	volume flow rate, m^3/s
Q_c	heat transfer in the condenser, kW
Q_e	heat transfer in the evaporator, kW
r	failed numbers
S	stress
S_0	mechanical stress under normal stress conditions
S_1	mechanical stress under accelerated stress conditions
t_i	the test time for each sample
T	the absolute temperature, K
T_1	the absolute temperature under accelerated stress conditions, K
T_0	the absolute temperature under normal stress conditions, K
T_f	the time-to-failure
V	velocity, m/s
V	voltage, V
W_c	compressor power, kW
x	$x = 0.01X$, on condition that $x \leq 0.2$.
x_{con}	the control parameters (or factor)
x_{noise}	the noise parameters (or factor)
$Y(x_{con}, x_{noise})$	the simulation output for control and noise factors

Greek symbols

η	Characteristic life
ω	Angular velocity, rad/s

Superscripts

β	shape parameter in a Weibull distribution
n	the stress dependence, $n = -[\partial \ln(T_f)/\partial \ln(S)]_T$

Subscripts

0	normal stress conditions
1	accelerated stress conditions

that can cause failure and to determine the proper parameters and their levels (Kackar, 1985; Byrne and Taguchi, 1987).

Taguchi's approach (Phadke, 1989) employs two experimental arrays: one for the control array and the other for the noise array. Trials on a desired output are taken for every combination of control factors, x_{con}, and noise factors, x_{noise}. For a given x_{noise} in the control array, the trials generated by the noise array can be reduced to a signal-to-noise ratio that is related to the loss functions. When the loss function averages a measure of loss over the distribution of the noise factors, it can be redefined as (Welch, 1990)

$$L(x_{con}) = \int Y^2(x_{con}, x_{noise}) g(x_{noise}) dx_{noise} \quad (1)$$

Optimizing over the control factors, x_{con}, an engineer can find a design that minimizes this loss function.

A large number of experimental trials in Taguchi's product array may be required because the noise array is repeated for every trial in the control array. For a mechanical structure, numerous design parameters and their levels should be considered in Taguchi's robust design. More parameters and levels make it harder to predict the part life and failure rate from the testing results.

Based on the failure analysis of the failed product in the field, an alternative parameter study, ALT with a new concept, B_x, and sample size, can be considered (Woo and Pecht, 2008). Our proposed method involves the following five steps:

1.1. Loads analysis

As parameters and required input, Taguchi's methods use a robust design schematic comprising the input signal factor, control factors, noise factors and output response. Loads are classified as noise factors. As an example, the energy flow in

Table 1 – Effort and flow in the multi-port system

Refrigerator parts	Effort, $e(t)$	Flow, $f(t)$
Mechanical translation (draws, dispenser lever)	Force component, $F(t)$	Velocity component, $V(t)$
Mechanical rotation (door, cooling fan)	Torque component, $\tau(t)$	Angular velocity component, $V(t)$
Compressor	Pressure difference, $\Delta P(t)$	Volume flow rate, $Q(t)$
Electric (PCB, condenser)	Voltage, $V(t)$	Current, $i(t)$

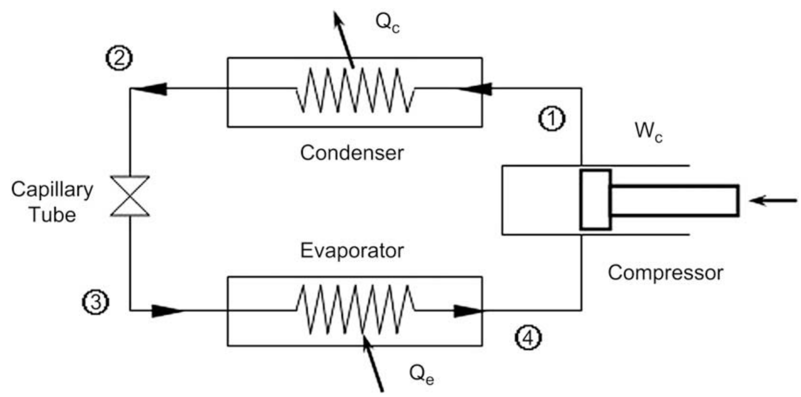


Fig. 1 – A vapor-compression refrigeration cycle.

the refrigeration system can be expressed as the product of an effort and a flow variable (Table 1) (Karnopp et al., 2000).

Refrigerator reliability problems in the field often occur when the components cannot endure the repetitive stresses due to internal or external forces over a specified period of time. The time-to-failure approach employs a generalized life-stress model (LS model) (McPherson, 1989) such as

$$T_f = A(S)^{-n} \exp\frac{E_a}{kT} = A(e)^{-n} \exp\frac{E_a}{kT} \quad (2)$$

And the repetitive stress can be expressed as the duty effect that carries the on/off cycles and shorten the part life (Ajiki et al., 1979). Under accelerated stress conditions, the acceleration factor (AF) can be described as

$$\mathrm{AF} = \left(\frac{S_1}{S_0}\right)^n \left[\frac{E_a}{k}\left(\frac{1}{T_0}-\frac{1}{T_1}\right)\right] = \left(\frac{e_1}{e_0}\right)^n \left[\frac{E_a}{k}\left(\frac{1}{T_0}-\frac{1}{T_1}\right)\right] \quad (3)$$

1.2. Derivation of B_x life and sample size

The characteristic life, η, in the Weilbull distribution by maximum likelihood estimation can be defined as

$$\eta^\beta \equiv \frac{\sum t_i^\beta}{r} \cong \frac{n \cdot h^\beta}{r} \quad (4)$$

As product (or part) reliability improves, there are usually no failures in the test. Thus, it is not appropriate to evaluate the characteristic life in Eq. (4). When the number of failed samples is below four, it follows the Poisson distribution (Ryu and Chang, 2005). At a sixty-percent confidence level, the characteristic life is defined as

$$\eta^\beta \cong \frac{1}{r+1} n \cdot h^\beta \quad (5)$$

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

$$L_B^\beta \cong x \cdot \eta^\beta = \frac{x}{r+1} n \cdot h^\beta \quad (6)$$

To assess the B_x life with about a sixty-percent confidence level, the number of test samples is derived from Eq. (6). That is,

$$n = \frac{1}{x}(r+1)\left(\frac{L_B}{\mathrm{AF} \cdot h}\right)^\beta = \frac{1}{x}(r+1)\left(\frac{1}{h^*}\right)^\beta \quad (7)$$

on the condition that the durability target, $h^* = (\mathrm{AF} \cdot h)/L_B \geq 1$.

1.3. Reliability design and experiment

Based on the customer usage conditions, the normal range of operating conditions and cycles of the product (or parts) are investigated. Under the worst case, the objective number of cycles and the number of required test cycles can be obtained from Eq. (7). ALT equipment can then be conducted on the basis of load analysis. In ALT testing we can find the missing parameters in the design phase.

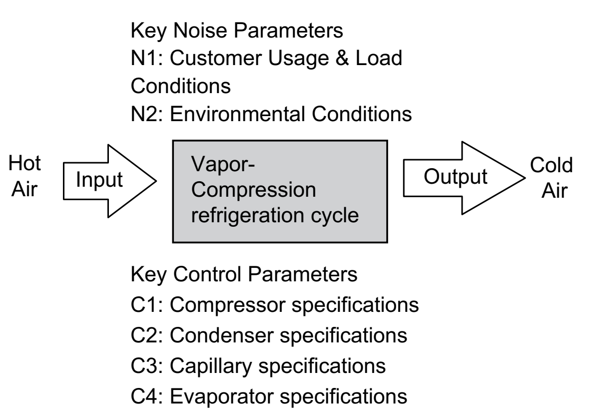


Fig. 2 – Parameter diagram of refrigeration cycle.

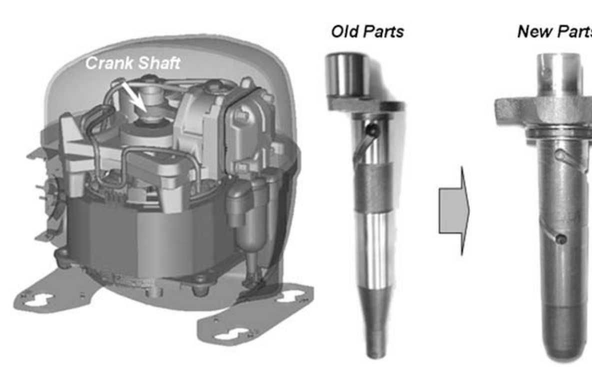


Fig. 3 – Redesigned compressor and crankshaft.

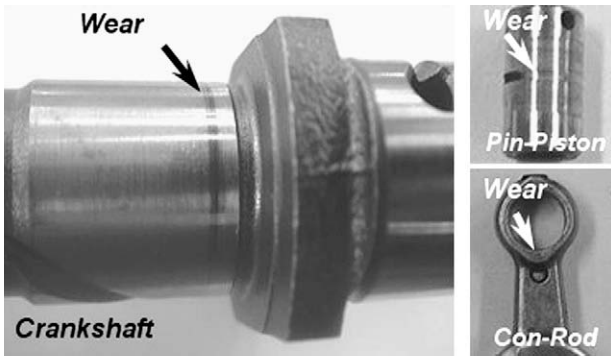


Fig. 4 – Some parts of a locked compressor in the marketplace after use.

1.4. *Parameter design with ALTs and corrective action plans*

The parameter design criterion of the newly designed samples can be more than the target life of B_x ten years. The B_x life of the samples in Eq. (6) can be redefined as:

$$B_X \cong \frac{h\cdot \mathrm{AF}}{L_B}\left(\frac{x\cdot n}{r+1}\right)^{\frac{1}{\beta}} \qquad (8)$$

From the field and a sample after accelerated life testing with corrective action plans, we can obtain the missing vital parameters of parts and their levels in the design phase.

1.5. *Life expectation*

With the improved design parameters, we can derive the expected B_x life of the final design samples.

2. Case study: reliability design of a redesigned crankshaft

A refrigerator consists of a compressor, a condenser, a capillary tube and an evaporator (Fig. 1). The refrigerant enters the compressor at a low pressure. It then leaves the compressor and enters the condenser at some elevated pressure, the refrigerant is condensed as heat is transferred to the surroundings. The refrigerant then leaves the condenser as a high-pressure liquid. The pressure of the liquid is decreased as it flows through the expansion valves, and as a result, some of the liquid flashes into cold vapor. The remaining liquid at a low pressure and temperature is vaporized in the evaporator as heat is transferred from the fresh/freezer compartment. This vapor then reenters the compressor (Sonntag and Borgnakke, 2007). The main function of the refrigerator is to provide cold air from the evaporator to the freezer and refrigerator compartments. Fig. 2 overviews the parameter diagram of refrigerator cycle.

Table 2 – ALT conditions in a vapor-compression cycles

System conditions		Worst case	ALT	AF
Pressure, MPa	High side	1.07	1.96	3.36 (1)
	Low side	0.0	0.0	
	ΔP	1.07	1.96	
Temp., °C	Dome temp.	90	120	3.92 (2)
Total AF (= (1) × (2))		–		13.2

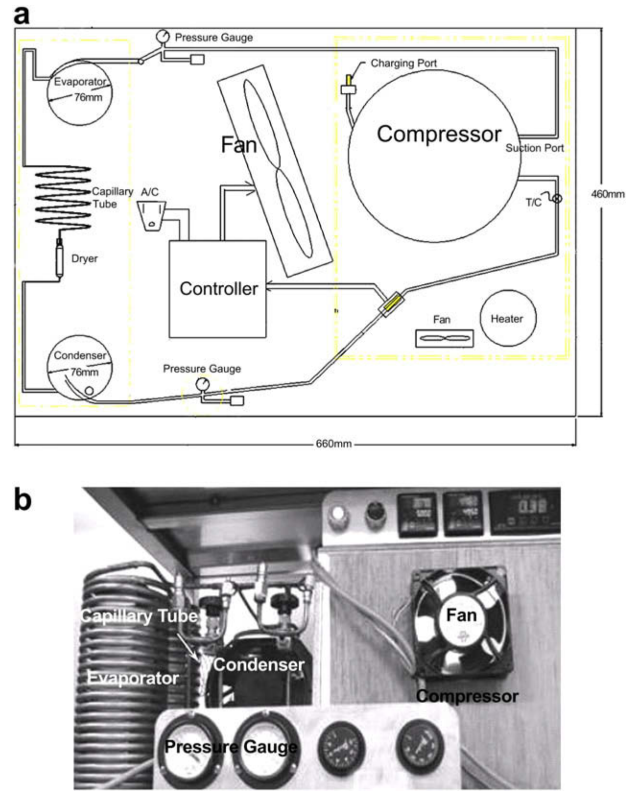


Fig. 5 – Equipment used in accelerated life testing. (a) A drawing of the test system. (b) Photograph.

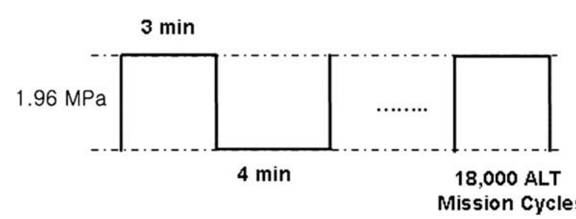


Fig. 6 – Duty cycles of repetitive pressure difference on the compressor.

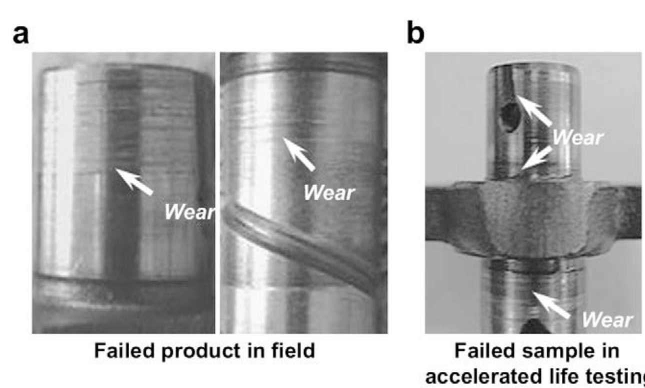


Fig. 7 – Failed product in field and ALT.

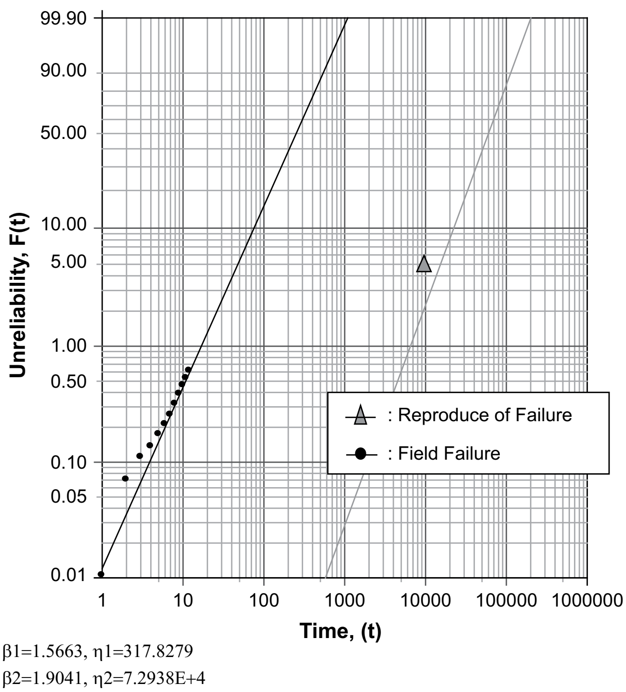


Fig. 8 – Field data and results of ALT on Weibull chart.

A capillary tube controls the flow in the refrigeration system and drops the high pressure of the refrigerant in the condenser to the low pressure in the evaporator. In a refrigeration cycle design, it is necessary to determine both the condensing pressure, P_c and the evaporating pressure, P_e. These pressures depend on ambient conditions, customer usage conditions, and heat exchanger capacity in the initial design stage.

Fig. 3 shows a redesigned crankshaft developed to reduce noise and improve energy efficiency of compressors in side-by-side (SBS) refrigerators. For these applications, the compressor needs to be designed robustly to operate under a wide range of customer usage conditions.

In SBS units sold it was found that the crankshafts of some compressors were locking. Locking refers to the inability of the electric stator to rotate the crankshaft, due to a failure of one more components within the compressor (see Fig. 4) under a range of unknown customer usage conditions. Field data indicated that the damaged products may have had a design flaw – oil lubrication problems. Due to this design flaw, the repetitive loads could create undue wear on the crankshaft and cause the compressor to lock.

For the theoretical single-stage cycle (ASHRAE Handbook, 1997), the stress of the compressor depends on the pressure difference suction pressure, P_{suc}, and discharge pressure, P_{dis} (Woo and O'Neal, 2006). That is,

$$\Delta P = P_{dis} - P_{suc} \cong P_c - P_e \tag{9}$$

By repeating the on and off cycles, the life of compressor shortens. The oil lubrication then relieves the stressful wear and extends the compressor life.

Because the stress of the compressor depends on the pressure difference of the refrigerator cycle, the life-stress model in Eq. (2) can be modified as

$$T_f = A(\Delta P)^{-n} \exp\frac{E_a}{kT} \tag{10}$$

The acceleration factor (AF) can be derived as

$$AF = \left(\frac{\Delta P_1}{\Delta P_0}\right)^n \left[\frac{E_a}{k}\left(\frac{1}{T_0} - \frac{1}{T_1}\right)\right] \tag{11}$$

3. Experiment

The normal ranges of operating conditions for the compressor were 0–50 °C ambient temperatures, 0–85 percent relative humidity and 0.2–0.24 G vibration. The normal number of operating cycles for one day was approximately ten; the worst

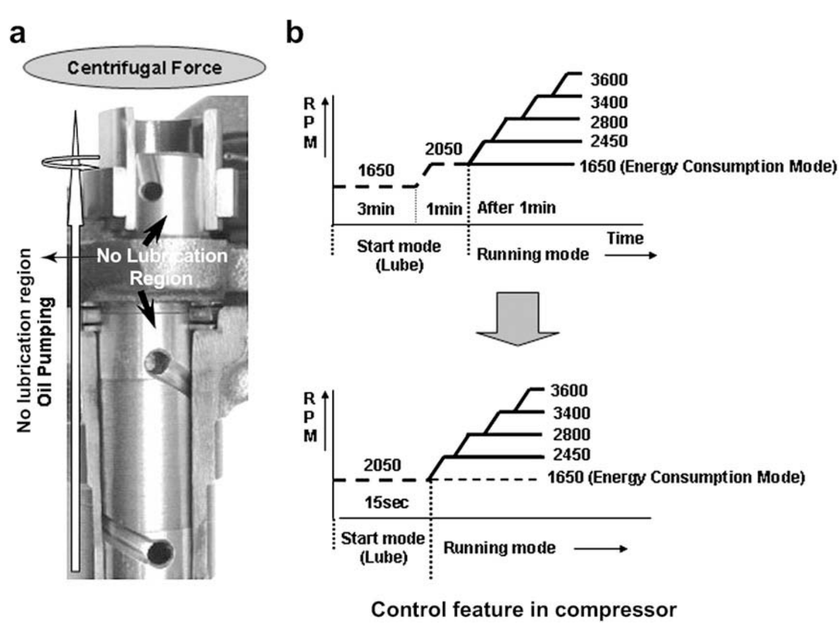


Fig. 9 – No lubrication region in crankshaft and low starting RPM (1650 RPM).

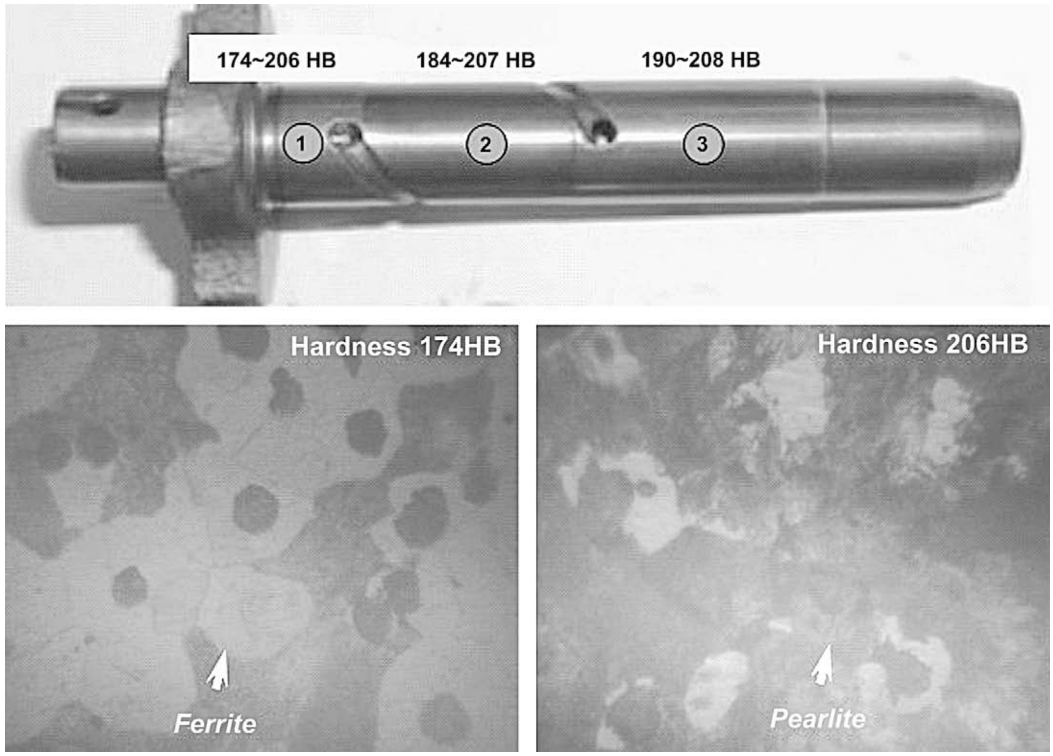


Fig. 10 – A large variation of hardness (FCD450) in crankshaft.

case was twenty-four. Under the worst case, the objective compressor cycles for ten years would be 87,600 cycles.

From the ASHRAE Handbook test data (1997), the normal pressure was 1.07 MPa at 42 °C and the compressor dome temperature was 90 °C. It was measured after T type thermo ouple pierced into the top compressor. For the accelerated testing, the acceleration factor (AF) for pressure at 1.96 MPa was 3.37 and for the compressor with a 120 °C dome temperature was 3.92 with a quotient, n, of 2. The total AF was approximately 13.2 (Table 2).

The parameter design criterion of the newly designed compressor can be more than the target life of B_1 ten years. Assuming the shape parameter β was 1.9 and x was 0.01, the test cycles and test sample numbers calculated in Eq. (7) were 18,000 cycles and 30 units, respectively. The ALT was designed to ensure a B_1 of ten years life with about a sixty-percent level of confidence that it would fail less than once during 18,000 cycles.

Fig. 5 shows the ALT equipment used for the life testing in the laboratory. Fig. 6 shows the duty cycles for the repetitive pressure difference, ΔP.

For the ALT experiments, a simplified vapor-compression refrigeration system was fabricated. It consisted of an evaporator, compressor, condenser, and capillary tube. The inlet to the condenser section was at the top and the condenser outlet

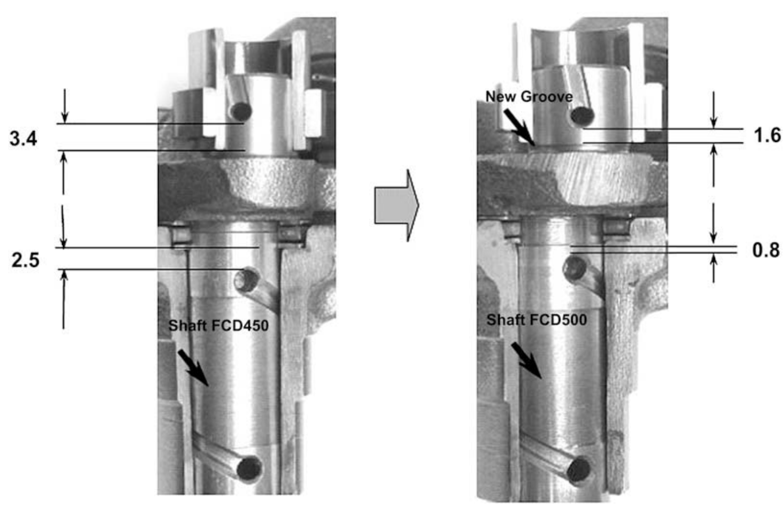


Fig. 11 – Redesigned crankshaft in first ALT.

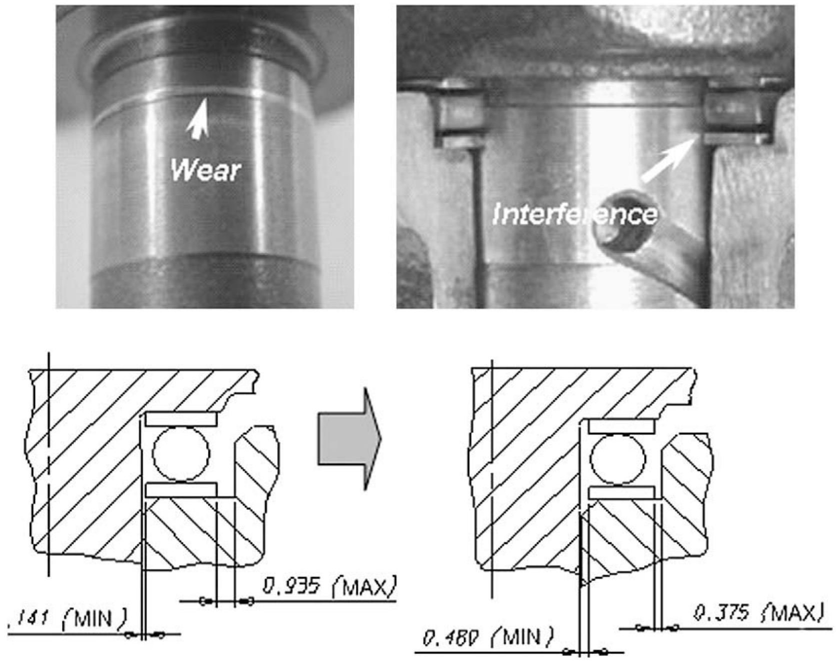


Fig. 12 – Redesigned crankshaft in second ALT.

was at the bottom. The condenser inlet was constructed with quick coupling and had a high-side pressure gauge. A ten-gram refrigerator dryer was installed vertically at the condenser inlet. A thermal switch was attached to the condenser tubing at the top of the condenser coil to control the condenser fan. The evaporator inlet was at the bottom. At a location near the evaporator outlet, pressure gauges were installed to enable access to the low side for evacuation and refrigerant charging.

The condenser outlet was connected to the evaporator outlet with a capillary tube. The compressor was mounted on rubber pads and was connected to the condenser inlet and evaporator outlet. A fan and two 60 Watt lamps maintained the room temperature within an insulated (fiberglass) box. A thermal switch attached on the compressor top controlled a 51 m^3/h axial fan.

4. Parametric ALTs with corrective action plans

Fig. 7 shows the crankshaft of a locked-up compressor from the field and a sample from the accelerated life testing. In the photo, the shape and location of the parts in the failed product from the field were similar to those in the ALT results. Fig. 8 represents the graphical analysis of the ALT results and field data on a Weibull plot. For the shape parameter, the estimated value in the previous ALT was 1.9.

It was concluded that the methodologies used were valid in pinpointing the weaknesses in the original design of the units sold in the market because (1) the location and shape of the locking crankshaft from both the field and ALT were similar; and (2) on the Weibull, the shape parameters of the ALT results, β_1 and market data, β_2, are very similar.

When both the locked compressors from the field and the ALT compressor were cut apart, severe wear was found in

Table 3 – Vital parameters based on the marketplace data and ALTs

CTQ			Parameters	Unit
Wear, locking	KNP	N1	Pressure difference	MPa
	KCP	C1	Oil lubrication region	–
		C2	Starting RPM	–
		C3	Crankshaft material	–
		C4	Thrust washer dimension	mm

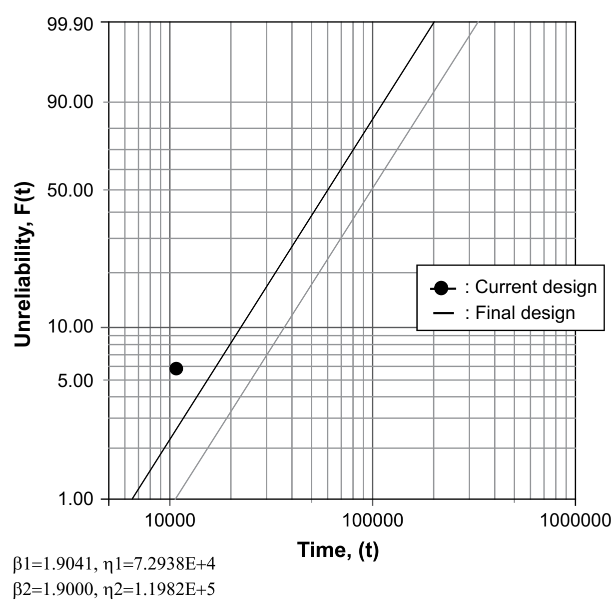
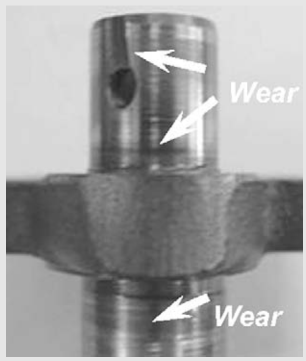


Fig. 13 – Results of ALT plotted in Weibull chart.

Table 4 – Results of ALT

	1st ALT	2nd ALT	3rd ALT
	Initial design	Second design	Third design
In 18,000 cycles, locking is less than 1.	✓ 10,504 cycles: 2/30 Locking ✓ 18,000 cycles: 28/30 OK	✓ 18,000 cycles: 2/30 wear ✓ 18,000 cycles: 28/30 OK	✓ 18,000 cycles: 30/30 OK ✓ 20,000 cycles: 30/30 OK –
Crank Shaft structure			
Material and Spec	✓ FCD450 → FCD450 ✓ One New Groove ✓ Location modification of oil supply holes	✓ Modification of washer dimension	

regions of the crankshaft where there was no lubrication – the friction area between shaft and connecting rod, and also the friction area between crankshaft and block. The locking of the compressor resulted from several design problems. There was (1) no oil lubrication in some regions of the crankshaft (Fig. 9a); (2) a low starting RPM (1650 RPM) (Fig. 9b); and, (3) a crankshaft made from material with a wide range of hardness (FCD450) (Fig. 10).

The vital parameters in the design phase of the ALT were the lack of an oil lubrication region, low starting RPM, and weak crankshaft material. These compressor design flaws may cause the compressor to lock up suddenly when subjected to repetitive loads.

The parameter design criterion of the newly designed samples was more than the target life, B_1, of ten years. The confirmed values β on Weibull chart was 1.9. When the second ALT and third ALT proceeded, the recalculated test cycles and sample size in Eq. (9) were 18,000 and 30 units, respectively. Based on the B_1 life of ten years, the first, second, and third ALTs were performed to obtain the design parameters and proper levels. The compressor failure in the first ALT was due the compressor locking. In the second ALT, it was due to interference between the crankshaft and a thrust washer. During the third ALT, no problems were found with the compressor.

To improve the lubrication problems in the crankshaft, it was redesigned as the relocated lubrication holes, new groove and new shaft material FCD500 (Fig. 11). To avoid the wear between crankshaft and washer, the minimum clearance was increased from 0.141 mm to 0.480 mm (Fig. 12). With these modified design parameters, the SBS refrigerators can operate in the process of on and off repetitively with a B_1 life of ten years life. Table 3 shows the vital parameters confirmed from a tailored set of ALTs and results of the ALTs.

The modified design parameters, with the corrective action plans, included (1) the modification of the oil lubrication region, C1 (Fig. 11); (2) increasing the starting RPM, C2, from 1650 to 2050; (3) changing the crankshaft material, C3, from FCD450 to FCD500; and (4) modifying the thrust washer dimension, C4, (see Fig. 12).

Table 4 provides a summary of the ALT results. Fig. 13 show the results of ALT plotted in a Weibull chart. With the improved design parameters, the B_1 life of the samples in the first, second and third ALTs lengthens from 3.8 years to over 10.0 years.

5. Conclusions

To improve the performance life of the parts in SBS refrigerators, we have suggested five steps. A refrigerator compressor was studied as a case. The failure modes and mechanisms for locking of the compressor in the field were identified. Important design parameters were studied and improvements evaluated using accelerated life testing. The following general conclusions are:

(1) Based on the products returned from the field and results of the ALTs, compressor locking occurred in the crankshaft. The key design parameters of the failed compressor in the SBS refrigerator were a lack of lubrication in a region of the crankshaft and variation in the hardness in the shaft material.
(2) Based on the ALTs, interference between the crankshaft and thrust washer was identified as a problem. The additional key design parameter of the compressor was the modification of the thrust washer dimensions. After a sequence of ALTs, key design changes were identified. The yearly failure rate and B_1 life of the compressor, based on the results of ALT, were over 0.01 percent and 10 years, respectively.

(3) The study of missing parameter in the design phase, through the inspection of the failed product in the field, load analysis, and ALTs, was very effective in redesigning a more robust compressor with significantly longer life.

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