1990

The Utilization of Contemporary Engineering Tools during the Development of Reliable Discharge Lines

J. F. Quesada
United Technologies Carrier Corporation

THE UTILIZATION OF CONTEMPORARY ENGINEERING TOOLS DURING THE DEVELOPMENT OF RELIABLE DISCHARGE LINES

John F. Quesada
Product Engineer

Carlyle Compressor Company
Carrier Corporation
Syracuse, New York 13221

ABSTRACT

The development engineer is constantly striving to increase the compressor resistance to abusive conditions in order to improve long term reliability. In internally suspended hermetic compressors, the discharge line design can have a significant impact on product life. Purely analytical techniques to evaluate discharge line stress levels have not proven adequate, especially during transient starts. The use of both conventional and advanced engineering tools is necessary in identifying potential weaknesses and improving the discharge line design.

This paper will outline the methodology for evaluating discharge line stress under field generated liquid slugging conditions. Emphasis will be on the tools and techniques used to evaluate possible options and verify the final design. Visualization and measurement of the line dynamics using high speed photography will be discussed. The development of laboratory bench tests that simulate the actual compressor operating conditions in the system played a key role in the analysis. This approach facilitated the verification of the appropriate design response in a timely manner. Design tools, including computer aided engineering and finite element analysis were beneficial.

Our findings demonstrate higher line displacements than thought possible during transient liquid slugging. The test results as well as the approach will be thoroughly discussed.

INTRODUCTION

Today's modern consumer has grown accustomed to reliable, trouble-free products. Manufacturers highlight these characteristics in product advertisements to make their goods appealing to the customer; for example, automobile advertisements emphasizing quality and customer satisfaction. Carrier Corporation is committed to achieving the highest level of customer satisfaction in all of its products. Since the compressor accounts for 40% of the reliability of the total air-conditioning system, it deserves much attention. Carrier's compressor field experience indicates that liquid refrigerant handling is the dominant failure cause. Poor field charging practices play a role here, as service technicians tend to overcharge; after all, charge can leak out, but it can't leak in! The present field overcharging tendency is further exacerbated by the 1992 federal energy standards, which mandate a 10.0 minimum seasonal energy efficiency ratio (S.E.E.R.). These higher efficiency systems will typically require a 25% increase in the refrigerant charge level. Clearly, increased resistance to transient liquid abuse (slugging) is required.

TECHNICAL OVERVIEW

Increasing compressor resistance to slugging is not an easy task for a multitude of reasons. First, the lack of sufficient reference data on this topic makes each liquid abuse analysis its own research project. Also, the variation of slugging severity requires a large quantity of test vehicles in order to obtain statistical confidence in the results. Moreover, the parameters involved; suction and discharge pressure, ambient temperature, piston top dead center (TDC), refrigerant charge level and soak time are not well understood with regard to their role in slug severity. The development of engineering techniques which may be used to handle these types of situations is the focus of this paper.

One approach to increasing resistance to slugging is to "beef-up" the overstressed components. This is acceptable for simple, linearly loaded structural components such as straps, brackets and bolts, where calibrated increases in strength can be predicted and verified quickly. Typically, "beefed-up" modifications evolve through a "cut and try" technique. "Cut and try" focuses efforts in laboratory tests on a single instrumented test compressor. Data is obtained and analyzed **sequentially**, under the guidelines of the following methodology: 1) Instrument the area in question (problem area, failing part) with strain gages, pressure transducers, or other appropriate measurement devices. 2) Run the instrumented compressor on the actual laboratory test which sites the problem, and record baseline data. 3) Develop a response (solution) based on the data, and re-test the compressor with the response. If the new data is promising, proceed to the following step. Otherwise, reiterate this step.

4) Qualify the design response with a large sample size. 5) Release the design as a success, or as a compromise offering an improvement over the present design.

CONTEMPORARY ENGINEERING TOOLS METHODOLOGY

Strengthening a design in a "cut and try" fashion is acceptable provided the component is single-functional with linear loading through a simple forcing function. Dynamic components, which undergo non-linear loading, transient shock loads, are linked to the suspension or otherwise reacting to compressor run speed should be developed through parallel testing in efforts to utilize all available engineering tools efficiently. A thorough understanding of the problem mechanism is acquired through the utilization of contemporary engineering tools. Specifically, this requires the integration of laboratory testing, computer aided engineering (CAE), and research analysis. The contemporary tools are divided into the three groups listed below.

A) Laboratory Tools

In the laboratory, two categories of testing exist. First, several compressors will be instrumented fully; with strain gages, pressure transducers, TDC markers, etc. Strain gages are applied in such a way as to determine hoop stress, pressure transients, and bending. These fully instrumented compressors are then run a number of times on the specific reliability test, and data is collected for analysis.

Secondly, laboratory development of a bench test is conducted concurrently with the system reliability testing to simulate it in a more controlled and timely environment. When optimized, the control of parameters on this bench test leads to a level of test repeatability difficult to achieve in the actual system reliability testing. With the ability to vary the bench test parameters easily, one can quickly understand the conditions that generate the worst case loads. It is important to mention that the compressor must be instrumented exactly the same as in the actual reliability testing in order to correlate the data and compare tests. Moreover, the bench test allows the engineer to observe phenomena that are impossible to view in the system tests. Since the bench test used in our analysis was not a sealed refrigeration-cycle system, it was possible to view the dynamics of the internal components using high-speed photography. High-speed motion pictures played back on an analytic projector with variable frame advance rates revealed excessive (up to 1" total) discharge line motion This approach played a key role in the development of a more robust discharge line.

B) Design Tools

Computer aided engineering plays an important part in theory development and design confirmation. This consists of several categories: 1. Finite element analysis in a static model, to determine stresses under given displacements; 2. Finite element dynamic models to determine dynamic stresses, natural frequency and modeshapes; 3. Spectrum analysis for the empirical determination of assembly or component vibratory modes and; 4. Computer aided design (CAD) layouts for determining maximum displacements permissible inside the shell and clearances between parts.

C) Research Tools

Finally, in the research department, usage of conventional and scanning electron microscopy (SEM) in the analysis of fractures facilitated diagnosing the events leading to failure. Fatigue and vibration testing are possible with the use of electro-hydraulic closed loop load cells which can subject components to a variety of conditions; such as finite displacements used to generate load vs. deflection curves, and frequency sweeps to determine component natural frequencies and modeshapes, which are useful in understanding component response to forcing functions.

Summary of Contemporary Tools

Successful integration of the preceding tools in the design analysis will lead to an understanding of the failure mechanism, as well as an appropriate design response. The response can then be qualified with a large sample of compressors on the actual system reliability test with a high level of confidence. The following table encapsulates the tools.

LAB TOOLS	• INSTUMENTED COMPRESSORS • BENCH TEST MODEL • HIGH SPEED PHOTOGRAPHY WITH ANALYTIC PROJECTOR
DESIGN TOOLS	• FINITE ELEMENT ANALYSIS • DIGITAL SPECTRUM ANALYSIS • COMPUTER AIDED DESIGN
RESEARCH TOOLS	• MICROSCOPY • ELECTRO-HYDRAULIC LOAD CELL

BACKGROUND

All Carrier-applied compressors are required to withstand a minimum number of liquid-flooded start cycles at a specified refrigerant charge level. The compressor must withstand these transient start-ups and subsequently pass performance testing. Teardown analysis upon test completion must verify that the compressor is undamaged. Preliminary design flooded start testing yielded discharge line fractures. Visual analysis revealed fractures at the junction between the sleeve which exits the shell and the discharge line, an intuitively high stress area. A study of these compressor assemblies revealed shiny wear marks on the line and motor, providing evidence of relative motion and impact between the two parts. Figures (1) & (2) depict the specific locations of the marks, as well as the fracture site.

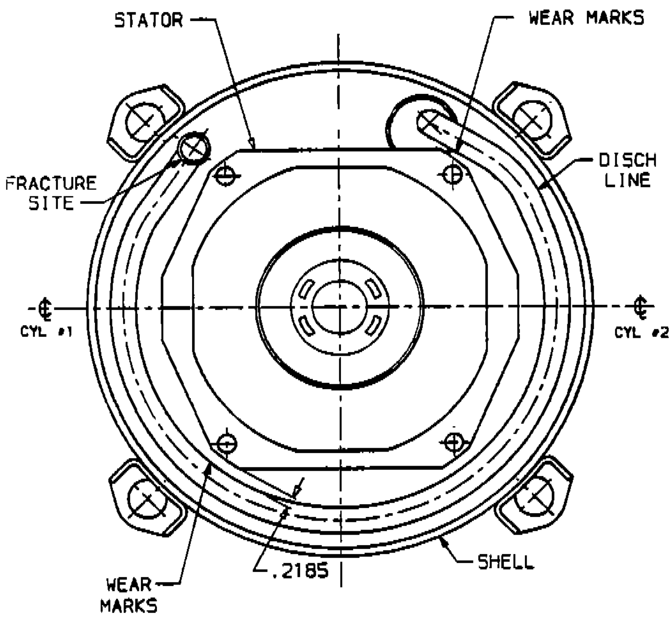

FIGURE 1. COMPRESSOR TOP VIEW

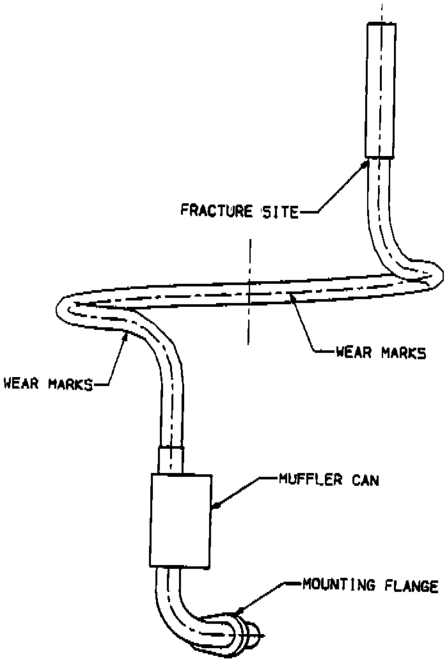

FIGURE 2. DISCHARGE LINE

The initial hypothesis of these failures was that the slugging reactions were extremely violent, thereby causing the internal assembly to move enough to physically contact the line and overstress it. Aware that discharge line alterations may adversely affect the torsional spring rate of the assembly, acoustics, and its natural frequency, it was clear this situation was to be analyzed to achieve an effective and timely solution.

DISCHARGE LINE DEVELOPMENT

The discharge line development will be discussed in two phases, due to the complexity of the problem and the amount of information gathered in the process. Again, it's vital to emphasize that all the tools available for use in the analysis be utilized in a **parallel** fashion for the most effective developments.

PHASE I

A study of the fractured parts under the microscope revealed high strain, low-cycle fatigue failure. A method devised by Carrier's research group helped to efficiently determine the location of maximum bending strain around the circumference of the tube with two strain gages applied 120° apart (see reference 1). Using this technique, a gaged discharge line was fabricated and installed in a compressor, along with a dynamic pressure transducer in the discharge plenum and a TDC indicator to reference the data to crank degrees. The compressor was run on the system flooded start test connected to a 14 channel video cassette recorder and cycled 20 times, in order to obtain more statistically accurate data than a single test offers. The data displayed two pressure spikes for every shaft revolution (expected) of a magnitude which was far too low to create problems with radial (hoop) stresses in the line. The TDC indicator revealed that the compressor was not up to full speed until approximately nine shaft revolutions. The highest pressure spikes occurred during the shaft speed ramp-up, whereas the highest strain in the line was typically during the first few revolutions. The location of maximum strain on the line as determined by the gages correlated well with the fracture initiation site viewed under the microscope. Figure (3) displays an oscilloscope plot of strain, TDC, and discharge plenum pressure.

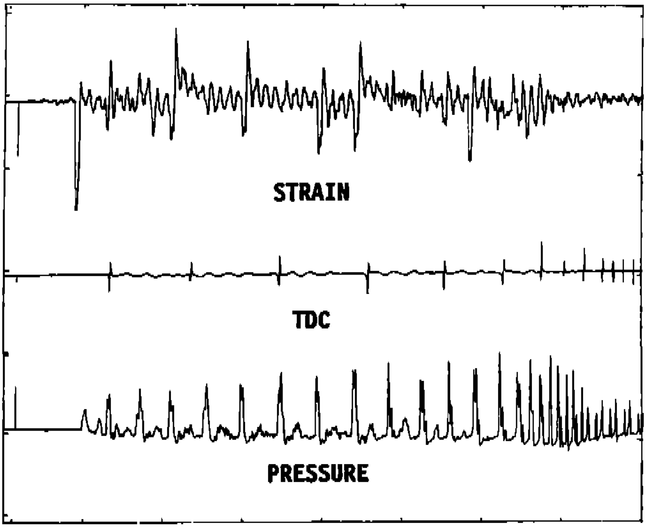

FIGURE 3.

An additional compressor was instrumented with accelerometers in an effort to obtain assembly x, y, and z translation as well as rotations about each of these axes. This was accomplished through the use of six accelerometers, three on each side of the crankcase. Each cluster of three accelerometers was arranged 90° apart, so as to obtain acceleration in each axis. One cluster was located at the base of the crankcase, and the other at the top to create the moment arm necessary for determining rotation. Data was gathered from repeated tests and was analyzed on a four-channel oscilloscope using an integration program. Double integration determined the actual displacements of the assembly. Figure (4) reveals an accelerometer signal with both integrations performed on a portion of the waveform. These crankcase displacements were transposed to local discharge line coordinates at the mounting location of the line to the crankcase. They could then be utilized in a static finite element analysis model which would input mounting flange displacements and torsion at the flange, and yield the stress at the failure site. However, these displacements were found to be less than .100" (contrary to the original belief) yielding stress under 10,000 psi on the line. CAD was then used to determine the maximum displacements and rotations the assembly could undergo before physical contact ("bottoming-out") was achieved. Even in this case, stress levels in the line, although close to the elastic limit, were far too low to be the cause of the low-cycle fatigue fractures being generated.

A static bench test was developed to further satisfy the conclusions being drawn, and to create new horizons for future testing. The static bench test consisted of the compressor mounted to a 1/2" steel plate with 3" diameter holes located in key areas of the shell for viewing the assembly and the discharge line. Four large steel bars were threaded into the stator 90° apart and passed through the shell via the holes. The shell was bolted together, so as to allow for ease of modification. A strain gaged line exactly the same as the one used in the system tests was installed in the bench test. The gages were then monitored while physically exercising the assembly to its limits in every direction. Here again, only strain levels within 0.25% could be generated.

Before moving on to the second phase of the investigation, let's summarize the findings thus far. Pressure in the discharge plenum was below a harmful threshold, and did not correlate well to the line strain. Crankcase displacements could not create large enough strain values to be the cause of the fractures. Maximum strains recorded typically occurred during the first few shaft revolutions.

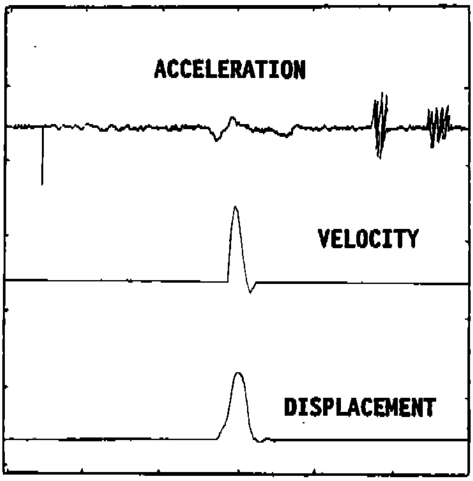

FIGURE 4.

<u>PHASE II</u>

The focus now was on the discharge line dynamics, as it was apparent from the previous tests that the line dynamics must play a major role in the problem. In the laboratory, a compressor was set up with a cut-a-way top shell to facilitate striking the line with a force transducer hammer while measuring the line response at various locations with a low mass accelerometer. Through the use of a real-time spectrum analyzer, the natural frequency of the line was found. The line natural frequency was also determined when fluid filled, to replicate being full of liquid in the system, and was found to be 10 Hz. lower. The finite element model was utilized to determine the modeshapes of the line analytically, along with an empirical determination using the bench test machine. This was accomplished by exciting the bench test assembly at the line natural frequency on the load cell, and observing the line displacement with the aid of a strobe light. When the line was filled with the liquid and driven at its "wet" natural frequency, the modeshape revealed the same line-to-stator contact as evidenced in the failure teardown analyses. It was now clear that the (fluid-filled) line dynamics under liquid slugging was the cause of the fractures.

Additional research work with the bench test on the load cell yielded a load deflection curve for the discharge line. Based on this curve, loads in excess of 60 lbs. were required to induce the strain levels being measured at the fracture site. The cause of the generation of these loads was what needed to be determined in order to cope with them effectively. What seemed most relevant in this entire investigation to this point was the viewing of the discharge line modeshape on the load cell. This revealed the exact motions the line must undertake when the assembly is slugged. To actually view the line under the liquid slugging test seemed to be a nearly impossible feat, however, replicating the test with a dynamic bench test would simulate system conditions.

Efforts were placed on "slugging" the bench model on a pump-up stand. The strain gaged line, discharge pressure transducer and TDC marker were applied to correlate this data with actual system test data. An incompressible fluid of comparable viscosity to the slug was chosen as the working fluid for the simulation. Suction and discharge plenums (including the discharge line) were provided with a means for filling with the fluid as well as several bleed valves for eliminating all trapped gas pockets. The crank angle was indexed, and the cylinders were filled with the fluid. Over 35 iterations were completed, varying crank angle, quantity of fluid in the cylinders and in the plenums and discharge line. The highest discharge line strain occurred with both suction and discharge plenums full of fluid, the discharge line filled with fluid to the level of the muffler can, the lowermost (#1) piston at bottom dead center (BDC) with its cylinder half-full of fluid, and the #2 cylinder full of fluid. Partial filling of the #1 cylinder allowed the shaft to respond to motor torque quickly, whereas complete filling lead to extremely slow shaft speed at startup and correspondingly low discharge pressures and line strains.

There was now a tool for investigating line dynamics. The "bench slug" could easily be repeated consistently, and components were accessible for open view. High-speed 16 mm. photography was utilized to capture the line dynamics on film during the bench slug. These motion pictures were then played back on an analytic projector which featured variable frame rates, including frame-by-frame advance. This allowed for the determination of the line displacement amplitude and frequency. The frequency correlated well with the fluid filled line natural frequency. The amplitudes measured far exceeded those witnessed during the resonant dwell on the load cell. Violent interference of the line against the stator could be seen. Different camera angles revealed the modeshape of this displacement to be identical to both that viewed on the load cell and derived from the FEA model. A view of the 90° elbow directly above the muffler can revealed motion in-line with the elbow. This motion here was always the first displacement witnessed in the films.

A strain gage torsional bridge was then installed on the line near the mounting flange to measure the torsional reaction at the flange as the fluid moved through the elbow above it. This line was then cut off directly beyond the elbow, so as to allow free motion of this portion of the discharge line. When slugged on the bench test, torsional values of 45 ft-lbs. were computed using the gage measurements. Furthermore, viewing the "free-body" elbow on film emphasized the magnitude of the torsional reaction at the flange, created by this fluid momentum transfer through the elbow. It was now concluded that as the fluid column changed direction through the elbow at calculated speeds of nearly 200 mph, the energy (momentum) it transferred drove the line into its first mode.

SOLUTION DEVELOPMENT

With the problem fully understood, a solution could be developed and easily verified on the bench test. A multi-functional team was formed including product engineering, corporate research, and manufacturing engineering to effectively filter the ideas to insure smooth implementation of the final design response. The team decided that a simple added component(s) would be most timely, if it could prove effective. Efforts were then placed on limiting the displacements of the existing design with a mechanical stop. The finite element model revealed the antinode location for the first mode; the optimum location for a line restraining device. An instrumented restraint, which consisted of a bolt with its shank squared-off for gage installations and a clamshell clamp welded to the head, was used to measure the horizontal and vertical loads the line developed here when restrained. Calculations were performed to develop the parameters of the design based on the instrumented restraint loads. The critical strength region of the design was found to be the weldment between the restraint and its mounting surface. Specifically, the highest stress is developed from the horizontal force bending the weld. The finite element model revealed near elimination of first mode dynamics, as well as low stress in the new configuration when subjected to full assembly translations and rotations provided the line was free to slide radially inside the restraint under torsional loads.

The first generation restraint clamped directly to the line without an isolation bushing. This configuration reduced line strains effectively, however, radial motion of the line through the restraint developed through stop-start testing wore the assembly loose enough to create radiated sound problems. An isolation bushing was developed to withstand temperatures in excess of 300°F as well as maintaining high impact strength, wear resistance, ductility, and energy absorption. Engineering elastomers were unacceptable due to the combination of their low shear strength, high coefficient of friction, high percent volume change in refrigerant and oil, and low temperature and creep resistance. An engineering thermoplastic was selected with success after experimentation with various polymers primarily for the correct balance between material stability (in refrigerant and oil at high temperatures) and ductility. The design utilizing a polymer bushing was verified on the system flooded start test with a large sample size. The design parameters and geometry are reviewed in figure (5).

NOMENCLATURE FOR FIGURE 5:

F_v = 110 LBS. VERTICAL FORCE INDUCED UPON CLAMP DURING SLUGGING.
F_h = 60 LBS. HORIZONTAL FORCE INDUCED UPON CLAMP DURING SLUGGING.
WELD SECTION PROPERTIES: $I=4.116E^{-}4 IN^4$. $A=.142 IN^2$. $c=.150$ IN.

$$\sigma_b = \frac{Mc}{I}$$

$$\sigma_b = \frac{60LB*1.3IN*.15IN}{4.116E^{-}4\ IN^4}$$

$$\sigma_b = 28{,}425\ PSI.$$

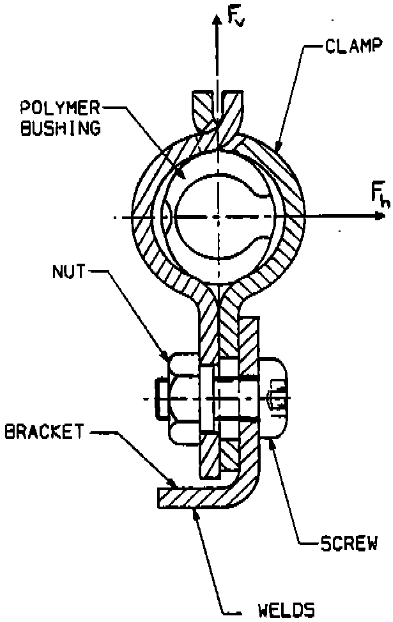

FIGURE 5.
RESTRAINT DESIGN

CONCLUSION

The utilization of contemporary engineering tools successfully identified "pipe whip" resulting from fluid momentum change as the discharge line failure cause. The dynamic bench test, once optimized, became more useful than the actual system slug test as up to ten iterations could be completed in one workshift compared to one iteration on the system test. Furthermore, slug severity was controlled by the established bench slug parameters, making it a more repeatable test than the actual system test. High speed photography made visualization of the line dynamics possible. Confirmation of the solution was performed on an actual system test with an instrumented compressor. Qualification of the design with a large sample size was achieved.

REFERENCES

1. Laub, J.S., "AN ANALYSIS OF PIPE FAILURE CRITERIA UNDER ARBITRARY OPERATING DISPLACEMENTS", Proceedings of the 1990 International Compressor Engineering Conference (PURDUE), July, 1990.

2. Schrank, D.A., Lewis, R.G., Vaccaro, J.P., "THE DESIGN AND ANALYSIS OF A RELIEF PLATE TO REDUCE CYLINDER PRESSURES IN A RECIPROCATING REFRIGERANT COMPRESSOR UNDER LIQUID SLUGGING CONDITIONS" Proceedings of the 1988 International Compressor Engineering Conference (PURDUE), July, 1988.

3. Boresi, A.P., Sidebottom, O.M., Seely, F.B., Smith, J.O. ADVANCED MECHANICS OF MATERIALS, Third Edition, 1978, John Wiley & Sons, Inc.