1972

Design of a New Reciprocating Compressor Line

A. R. Worster
Ingersoll-Rand Company

Follow this and additional works at: http://docs.lib.purdue.edu/icec


This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
The design of reciprocating compressors is found in few textbooks. It is a specialized field in which most of the know-how is in the hands of compressor manufacturers. This is not so strange when the facts are considered. Unlike consumer products, the need for change in design is relatively small when adequate designs to meet customers' needs have been established. Good reciprocating units have been available for many years and have been produced by a number of reliable manufacturers. Consequently, there have been relatively few completely new reciprocating units developed in recent years as the overall requirements have been changing very slowly.

Most of the design effort on compressors has been slanted toward better utilization of existing designs. Many cases can be cited where lines have been extended by minor design changes. In some instances, this results in heavier loadings of existing parts. This is entirely practical if the original design was conservative. In other cases, the combination of components are altered to meet specific unusual customer requirements. This approach is realistic as long as there is no major change in application or use of compressors. However, there has been a real shift in installation and operating costs. This has brought about pressure to consider developing new concepts in the design and application of reciprocating compressors. This provides the designer with a unique opportunity to start from scratch and completely re-evaluate the need for a modern air compressor plant. This paper will describe the process used to arrive at a design which meets the needs of the user, as well as the manufacturer of air compressors.

Design is much more than good geometric layout, calculation of stresses, evaluation of codes, or consideration of mechanics of operation. In fact, it will be shown that definition of objectives and evaluation of what the ultimate user really needs is, in many ways, the most difficult task of the designer.

The first step involves analyzing customers needs. Most of this data comes from opinions and considered judgments rather than facts. Consequently, it is essential to obtain a wide cross-section of opinion from personnel who are working closely with customers. This permits optimizing the relative importance of the design parameters. These often conflict with each other. Important items to be evaluated include the following:

- Efficiency
- Reliability
- Space
- Foundations
- Cooling
- Drivers
- Noise Level
- Safety
- First Cost
- Pressures
- Sizes

The primary input for all of the above comes from Marketing and Sales personnel. The latter are in direct contact with customers and have first-hand knowledge of what is happening in the field. In the case study, we drew together selected personnel from all over a wide cross-section of the world. In fact, we had each representative prepare, in advance, his analysis of the requirements he would recommend in building a completely new line of air compressors. This encouraged independent opinions. A panel was then asked to evaluate all of the input to obtain a possible consensus.

The personnel involved were restricted only in the sizes to be considered. We gave them 75-150 HP as the range to be evaluated for plant air service.

Design personnel participated in the panel discussions. Some of the thinking and output follows.
EFFICIENCY

Years ago, efficiency of the compressor was of major importance. Power costs were high and labor costs were low. At one time, a compressor with under 19 BHP/100 at 100 psig had a major selling advantage over units which were less efficient. This is still true today, even though the need for high efficiency has been de-emphasized. Somewhat less efficient units are acceptable provided the efficiency sacrifice results in other cost savings that more than offset added power costs. In fact, some rotary positive machines are being sold for plant air service with as high as 24 BHP/100 which should not be acceptable to the man paying the power bill.

The consensus in this study was that although efficiency was not of paramount importance, we should design in the range of 20 BHP/100. This would avoid the problems of high horsepower and high heat rejection. Temperatures would be controlled which improves safety.

RELIABILITY

Reliability was of less importance years ago than today. Each plant had highly competent, well trained mechanics who took great pride in servicing the mechanical equipment. Most plants had standby units. There was relatively little movement of the labor force and personnel became completely familiar with all aspects of the equipment. It was possible to continuously adjust machinery and make major overhauls at a relatively low cost. In fact, a unit with adjustable bearings, crossheads, etc. seemed to be preferred. This has completely changed. Reliability, with minimum service, is the name of the game. It is essential to have any servicing made simple. Adjustable parts are a thing of the past. Servicing must be infrequent, quick and easy. Operating personnel demand an absolute minimum of maintenance. They are primarily concerned in having a continuous trouble-free supply of plant air. Reliability is critically important in any new design.

SPACE

The availability of space for plant air systems, either in old plants or new, is at a premium. Building costs have risen sharply. In many cases, space simply isn't available. Plants are locked in by available property boundaries. There is a strong tendency toward designing more power in a cubic foot of space than was true in the past. This applies to all equipment, not just air compressors. Therefore, it was decided that the new design should concentrate on maximum utilization of space without sacrificing reliability or accessibility.

FOUNDATIONS

Earlier reciprocating compressors required massive foundations. There were two reasons. In many designs, the foundations were used to align the frame, running gear and cylinders. In all cases, the foundation was needed to absorb the large unbalanced forces. However, similar to space, the cost for foundations has risen sharply. Consequently, it was decided that the new design be based on minimum foundation requirement as a means of controlling installed cost. It would also make it easy for customers to install. Perfect balance did not seem mandatory, however, good balance seemed critical.

COOLING

For many years, water cooling seemed synonymous with heavy duty and air cooling seemed synonymous with light duty. There is, however, a growing need to provide heavy duty air cooled equipment due to our growing water shortage. In many cases, water is simply not available; in others, the quality of the available water is marginal. It was decided that the new design of reciprocating compressors must provide the possibility of both water and air cooling.

Most installations today are expected to continue with water cooling as a means of removing the heat of compression to help control the ambient temperatures in power plant installations. In some cases, where water is limited or of poor quality, closed water systems will allow rejecting the heat to the atmosphere rather than in the compressor room.

DRIVERS

The electric motor continues to be, by far, the most popular driver for compressors. However, to meet all needs, any compressor design must be suitable for engine and turbine as well. The preferred electric motor drive is direct-connected, however, coupled or belt driven should be possible for special applications. Integral motor design also minimizes space requirements and drive losses.

NOISE LEVEL

Recent Federal legislation under Walsh Healey defines today's limits on acceptable noise generation. However, looking to the future, it was decided to design for much lower noise levels. This would protect the user and prepare for possible future legislation. We know that quieter environments will be a must for future generations.

SAFETY

Similar to noise levels, safety in plant air systems
is becoming critical. Safety requirements must consider the total air plant system as well as the bare compressor. The recent surge in product liability suits emphasize this point. It was agreed that no compromise should be made in design which would increase the hazard in operating the plant air system. In fact, steps should be taken to reduce hazards.

FIRST COST

The need to design a low first cost machine was seen to be of critical importance to the marketing man. It is difficult to compete if the first cost is high. This would require engineering evaluations to compete. This is difficult and time-consuming. In some cases, the user refuses to take time to evaluate. At the same time, the cost can directly affect the efficiency, reliability, noise level and safety. It was finally agreed that, although first cost is important, it was essential to keep the design consistent with the operating needs of the plant.

PRESSURES

The review of the entire market showed clearly that most of the needs of plant air systems can be met by designing units for 125 psig maximum operating pressure. In limiting the design to this pressure level, we would be able to optimize the design to cover most of the plant air applications. It was decided to avoid over-design for higher pressures. This would avoid higher costs.

SIZES

The utilization of the new compressor line was evaluated. Most would be used with electric motor drive. It was recommended we size in accordance with available standard electric motors. This would mean four sizes - 75, 100, 125 and 150 HP. Selection of engine and turbine drive would simply be the nearest available engine or turbine to drive the compressor.

This part of the design process took many months. Numerous meetings were held. When fairly firm design objectives were established, the designers were then ready to go to the drawing boards. Earlier layouts would have meant decisions would have been made before the needs were clearly known.

Many layouts were made covering alternate solutions. Each was evaluated and the overall design began to take shape. Concurrently, studies of available performance data was analyzed, some by computer, and some manually. Part of this lengthy process, including results, follows.

RUNNING GEAR

A preliminary look made it easy to know that two stage compression was mandatory. This was dictated by the need for reasonable efficiency and low temperatures. The latter is essential for safety. The first question was then one of geometry of the two stages. Layouts of opposed cylinders as compared to either L or Y design made it clear that space could be minimized with either the L or Y design. Although some unbalance would be experienced, it could be minimized by keeping reciprocating weights to an absolute minimum. It was decided that the L design offered no disadvantages to the user.

ROTATIVE AND PISTON SPEEDS

Another step was the selection of rotative and piston speeds. Rotative and piston speed selection is most critical. They both play a major part in the final geometry of the unit. They are major factors in the final cost of product. These must be optimized to result in the smallest package and the most economical use of materials. From extensive test programs and computer analysis programs, we know that high rotative speeds considered alone tend to result in low efficiency. This is partly due to inertia effects on the valves. In addition, there are increased losses from pressure drop through valves due to high velocities and mechanical friction. High piston speeds make adequate valving for good efficiency extremely difficult.

It was decided to select 880 RPM induction motor speed which was a little higher than previous speeds for the same class of equipment. This resulted in reducing the size of the final package and, at the same time, allowed the use of an economical electric motor. Tests showed that efficiency would not be seriously affected. However, piston speed was selected on a conservative basis at 733 feet per minute. This is somewhat lower than some units of past designs including large process units. It would allow conservative valve speeds and result in a higher efficiency.

UNBALANCED FORCES

The unbalanced forces in an L design are directly proportional to the reciprocating weights. The primary forces can be completely balanced. However, the secondary forces are always present unless elaborate, expensive balancers are added to the system. These are not desirable or necessary. Our designers evaluated each component that affected unbalanced forces. They were challenged to keep these to an absolute minimum and, at the same time, use conservative stresses in these parts. The result was a radical reduction in reciprocating weights as compared to past practice resulting in very low unbalanced secondary forces.
UNLOADING

The design objective was to keep the new unit as simple and fool-proof as possible. The use of three-step unloading as compared to earlier consideration of five-step control was considered acceptable for all users on these sizes of units. This simplified control is easier to maintain and results in adequate control of line pressure.

COOLING

It was decided to have the basic design with an option for either water cooling or air cooling depending upon the user's needs. The uniqueness of the design allows changing to either style of cooling with the same basic cylinders and frame and gear.

A major forward step was to incorporate an aftercooler as standard equipment. Past practice had been to supply a bare compressor and recommend the use of aftercoolers. The result was that, in many cases, the user did not recognize the clear-cut need for aftercooling to improve plant operation and, more importantly, to improve safety. By incorporating the aftercooler in the basic design, it becomes automatic. The installation cost is also kept to a minimum.

LUBRICATION

The frame and gear lubrication was selected as a simple pressure feed system.

The question of cylinder lubrication was a more difficult decision. Considerable study was made of normal oil lubrication compared to minilube compared to non-lube. From the standpoint of air plant system contamination, either the non-lube or the minilube has obvious advantages. Unfortunately, the state of the art shows clearly that normal oil lubricated cylinders are dramatically more reliable and free from frequent maintenance. In addition, normal lubrication permits designing with minimum reciprocating weights.

NOISE

The Walsh Healey act dictates today that noise levels shall not exceed 90 dBA. It is our belief that this may be reduced in the near future. In fact, the earlier drafts of the act included an 85 dBA requirement. This was changed as most existing designs would not meet the lower requirement. However, the lower figure does result in more comfort and a better environment. Therefore, our design objective was 85 dBA.

Analysis of what makes noise in a compressor showed that much of the higher noise level is due to air noise. Careful intake silencer design and the use of large generous air passages with a minimum of interconnecting piping were expected to produce low noise. The designers used more generous passageways and lower velocities than past experience would have indicated. The end result proves this was effective.

Concurrent with the design phase described above, much component test work was performed in the Development Lab. Some was done on existing compressors, some by static loading of models using strain gages, and some by destruction testing. Many components were pretested before the first prototype was built. By using the design time effectively, there were a minimum of changes after the prototype was tested.

The final result of the new compressor design can be seen from Figure 1.

THE AIR CUBE MODEL LLE
LOCATION OF MAJOR COMPONENTS

Figure 1 Right Front View of Air Cube

Significant changes from conventional arrangement were made. These were needed to accomplish all of the design objectives for this completely new air plant.

CRANKSHAFT AND MAIN BEARINGS

The crankshaft is forged steel and the main bearings are double row, spherical roller bearings. The forged shaft is used to obtain consistent counterweighting. At one point, we considered castings but discarded this as a poor economy move at the expense of safety and reliability. The anti-friction main bearings minimize friction.

CONNECTING RODS AND BEARINGS

The connecting rods are conventional forged steel
design with drilling for lubricating both ends. The big end bearing has a babbitted surface with low bearing pressures. The small end bearings are bronze. Both bearings are precision type with no adjustments needed.

CROSSHEADS AND CROSSHEAD GUIDES

The crosshead is unique. Special surface materials have been avoided by designing in a hydrodynamic wedge to insure adequate oil film. The guide itself is separable from the frame. This can be rotated for double life if wear ever occurs and can also be easily replaced if necessary. No adjustments are needed.

PISTONS AND PISTON RODS

The conventional long piston would have been much too heavy to meet the low inertia load requirement. The wafer piston reduces weight and also reduces piston rod stress. The unique connection between the rod and the low pressure aluminum piston provides for thermal expansion. The high pressure piston is cast iron.

CYLINDERS

The square shape provides large pulsation chambers immediately adjacent to the cylinder bore. This reduces pulsation and, consequently, is considered a major factor in the low noise level of the overall unit, as well as eliminating undue stresses on the valves associated with conventional cylinders and critical pipe lengths.

VALVES

Valves were selected from our previous knowledge and years of experience with the patented channel valve. They provide efficient operation, and are quiet due to the air cushioning. This, again, helps to reduce noise level.

PROTECTIVE DEVICES

Figure 2 shows the solid state Tendamatic panel which was specifically designed for this compressor unit. It protects the compressor and thus the customer from major problems encountered with compressors. Most operating problems are a result of inadequate attention to maintenance procedures and supply of lubricant in the frame and gear and cylinders.

INTERCOOLER AND AFTERCOOLER

The intercooler and aftercooler, water-cooled, are of a somewhat traditional tube and fin type proven in larger units up to 1250 HP. The assembly and the flow-through are simplified in this compressor to minimize space.

SOLID-STATE 'TENDAMATIC' CONTROL

Monitors - Supervises - Protects

INTERCOOLER

AIR PRESSURE

OIL PRESSURE

DISCHARGE

AIR PRESSURE

LOAD INDICATOR

Figure 2

PERFORMANCE AT-A-GLANCE

UNBALANCED FORCES

Figure 3 shows the magnitude of the unbalanced forces experienced with this unit compared to other units in the same horsepower range. This provides for a major breakthrough in installation costs. The user is able to install a complete air plant in minimum time with little or no foundation.

MAXIMUM UNBALANCED FORCES

in various types of reciprocating compressors

Figure 3
DESIGN DETAILS

Table I shows dimensions and performance data on the 100 HP size of this new line. This unit delivers 535 CFM at 100 psig. You will note that all components are generously sized causing no major breakthroughs in stress levels. All components are conservatively loaded. No exotic materials are used.

<table>
<thead>
<tr>
<th>Nominal Rating</th>
<th>100 HP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power at 100 PSIG</td>
<td>105.8 HP</td>
</tr>
<tr>
<td>Capacity</td>
<td>535 ACFM</td>
</tr>
<tr>
<td>RPM</td>
<td>880</td>
</tr>
</tbody>
</table>
| Stroke           | 5"
| Piston Speed     | 733 |
| Crankshaft Diameter at M. B. | 3-1/2" |
| Con Rod Bearing Diameter | 3-1/2" |
| Con Rod Bearing Length | 2-1/8" |
| Length of Con Rod | 11-3/4" |
| Crosshead Pin Diameter | 1-3/4" |
| Crosshead Pin Bushing Length | 2-1/4" |
| Diameter of Crosshead | 6-1/4" |
| Length of Crosshead | 5-1/2" |
| Piston Rod Diameter | 1-1/4" |
| L. P. Piston Diameter | 13-1/2" |
| H. P. Piston Diameter | 8-1/4" |
| L. P. Valve Air Speed | 5870 |
| H. P. Valve Air Speed | 5820 |
| Weight           | 6600 Pounds |

The major trend toward unattended minimum maintenance installations mandates high reliability as a prime design consideration. Units are expected to, and have, run for thousands of hours with only lubrication system maintenance.

The important considerations of reliability and easy maintenance can be seen throughout the design.

RESULT

The final result is a new generation of air compressors. Emphasis has been placed on low first cost, low installation cost and low power cost. Even more emphasis has been placed on low servicing cost, quiet operation and safety. This design is setting new standards for the industry.