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Design and Development of an Old Concept Using New Material to Produce an Air Compressor

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DESIGN AND DEVELOPMENT OF AN OLD CONCEPT
USING NEW MATERIALS TO PRODUCE AN AIR COMPRESSOR

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The WOB·L piston compressor[1] is designed and developed to meet a need for an economical and reliable high-pressure, fractional-horsepower compressor to produce oil-free air. The approach taken to provide the reciprocating motion is a compromise between a diaphragm compressor and an articulated piston-type compressor. This motion is embodied in the connecting rod-piston of the WOB·L compressor. The sealing element material and design for this arrangement is of prime importance. This element provides the sealing function and reliability required to produce a successful product.

Figure 1 is a photograph of our Model 607 WOB·L compressor, and the specific one we will be discussing. It is a 1/8th-horsepower compressor capable of 100 PSI operation continuously with a life approaching 10,000 hours.

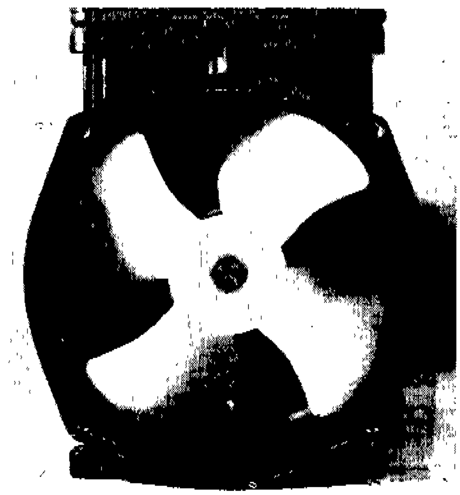


Figure 2 shows the unit with the front cover removed. As you notice, the unit is equipped with an axial flow fan in a circular cross section housing.[2] The fan is slip-fitted to the motor shaft and provides cooling air both to the motor and to the cylinder sleeve area.

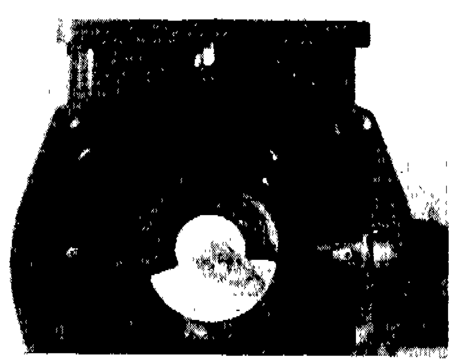
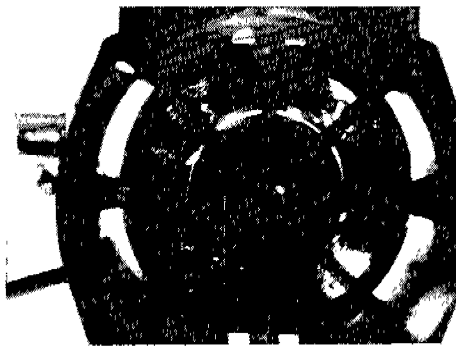


Figure 3 illustrates the general arrangement of the eccentric, eccentric bearing, connecting rod-piston, and the housing. The balanced eccentric produces the stroke, which is quite basic.



On Figure 4, notice the space around the outside diameter of the stator. The cooling air is driven through and around the stator, and diverted to the sleeve area giving a low-running temperature and increased overall efficiency.

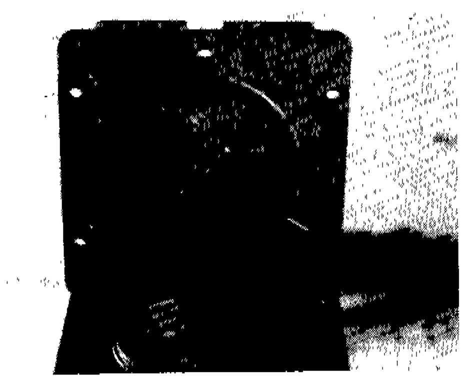
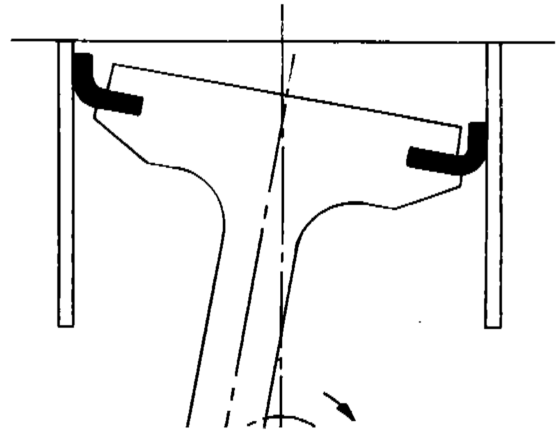


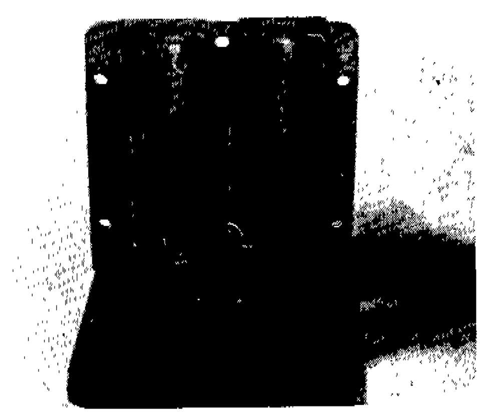
Figure 5 shows the top of the connecting rod-piston sleeve and valve plate with the intake valve. Notice the thin wall construction of the hard-coated teflon-impregnated aluminum cylinder sleeve.

A great deal of care must be taken when designing the sealing element (or cup)/ piston assembly. The cup material itself requires foremost consideration. The material must exhibit high flexural strength, low abrasive qualities to assure long cylinder sleeve life and high wear resistance to pressure-velocity loading (PV's approach 20,000). The material selected is a teflon-based compound.

"Thin" material provides good sealing ability due to the flexibility in the hinge area; cup stress concentration near the radiused portions of the piston are reduced; pressure-velocity wear, however, causes premature failures. "Thick" materials are prone to fatigue near the hinge area. The sealing ability is reduced due to the relative inflexibility. Upon selecting the proper thickness material, radial clearances across the bore must be carefully determined.



Although no piston side loading forces are present due to compression, a centrifugal force component can cause the piston to displace towards the "high" tilt side of the piston on the compression portion of the cycle. (Figure 6) This results in higher stresses on the cup on the "low" tilt side, causing rupture-type failures. Tests have shown that a broad range of cylinder sleeve finishes are tolerated before cup life or sealing ability is adversely affected.



On Figure 7, the valve plate is back in position and the cylinder head is shown. The exhaust valve is shown on the valve plate. Both valves are standard 0.003" thick Swedish stainless steel reed valves. The head contains an elastomer sealing element which divides and seals the two cavities-inlet and discharge. The longer boss is a developed inlet tube which, in conjunction with the inlet chamber volume, reduces the compressor sound level. The smaller boss is provided to allow a pressure relief valve, if desired.

Figure 8 is a photograph of the housing itself. It shows the sleeve shelf and general arrangement.

The various parts that comprise the compressor, then, are the:

1. Eccentric and bearing
2. Connecting rod piston and sealing element
3. Piston cylinder sleeve
4. "O" ring seal
5. Valve plate
6. Cylinder head and seal

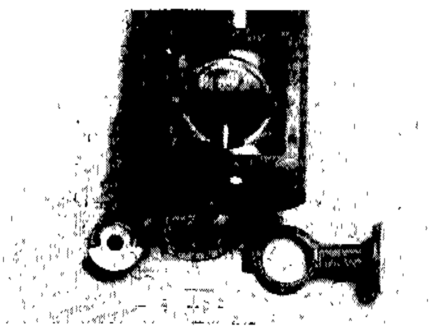


Figure 9. These are the significant parts that make up the vertical tolerance stack. They are: housing, eccentric, and connecting rod piston. All other parts as shown on Figure 8 are, of course, necessary to produce a compressor, but the three parts mentioned are the primary ones that determine the clearance volume.

The relative simplicity of the WOB·L produces advantages many times unavailable in conventional compressors. The advantages include:

* low clearance volume and clearance volume variations
* low cylinder element operating temperatures
* increased efficiency
* ease of manufacture
* extended life
* low operating sound levels

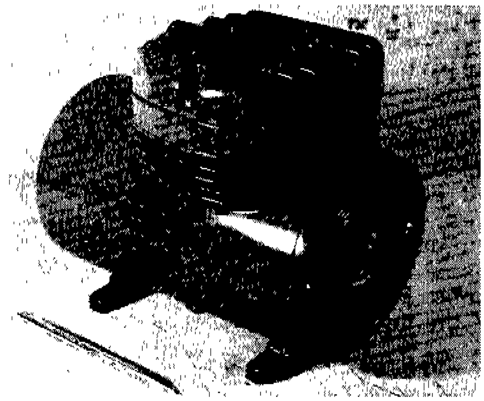


Figure 10 is a photograph of a diaphragm compressor of comparable size. The pressure is limited to approximately 50 PSI.

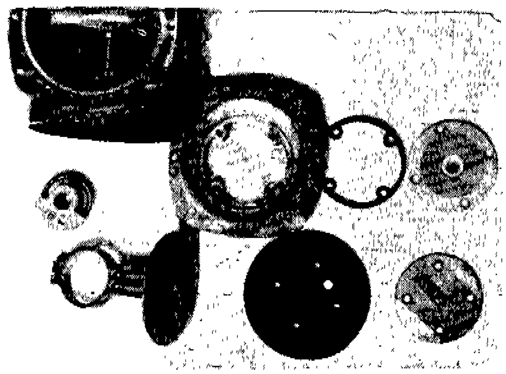


Figure 11 shows the parts of the diaphragm compressor. These parts determine the clearance volume and the resultant variation in the clearance volume from unit to unit. Compare these to the three primary parts of the WOB·L, as you can see, eight parts versus three!

We would like to discuss one of the WOB·L advantages in greater detail.

<u>Clearance Volume Variations</u>
The same WOB·L piston simplicity of design and construction also results in the important additional benefit of a significant reduction in clearance volume variations between units. This fact is borne out by both theoretical and production line results. The benefits from this reduction are threefold:

1) It allows the designer to "close in" the connecting rod-piston pan/valve plate clearance at connecting rod-piston top dead center. The designer is limited primarily by piston pan tilt and three major part tolerances. This results in a compressor whose volumetric efficiency is relatively high.

2) Since clearance variations are the major contributors to high pressure capacity variations, production line specifications can be more effectively established because capacity deviations from the mean are small. This results in a lower production line capacity reject level and improved control over production variables.
3) The eventual user of the WOB·L style compressor can expect lower capacity variations from unit to unit and production run to production run.

In order to predict the capacity variations due to clearance volume variations in the WOB·L piston series of compressors, the following analysis has been utilized. The analysis results are then related to production line results.

As an approximation to capacity, the following expression is offered:

$$Q = \left(V_t - \frac{P_d}{P_s} V_c\right) \frac{S}{1728} \quad ft^3/MIN. \qquad \text{(CFM)} \quad (1)$$

where

Q=capacity, volumetric flow in free air, ft^3/min.
V_t=unswept volume at bottom dead center, in^3.
V_c=unswept volume at top dead center, in^3.
P_d=discharge pressure, PSIA.
P_s=suction pressure, PSIA.
S=compressor speed, RPM.

Equation (1) is a representation of capacity for the WOB·L series of compressors which is a fairly close representation considering the expression implies:

1) Compression and expansion proceed in an isothermal manner.
2) The effects of sealing element "blow-by" are negligible.
3) Ideal valving efficiency.

In order to relate the changes in capacity to the variables affecting capacity, we can use the incremental chain law[3]:

$$\text{(CFM)} \quad (2)$$

where: $\Delta Q = \frac{\partial Q}{\partial V_T}\Delta V_T + \frac{\partial Q}{\partial V_c}\Delta V_c + \frac{\partial Q}{\partial P_d}\Delta P_d + \frac{\partial Q}{\partial P_s}\Delta P_s \; \frac{\partial Q}{\partial S}\Delta S + \Delta Q_M$ are the partial derivatives of capacity with respect to each variable affecting capacity. ΔQ_M is a term representing capacity measurement errors. The other Δ terms are:

ΔV_c, ΔV_t=volume changes due to part tolerances
ΔP_d, ΔP_s=discharge and inlet pressure variations from the head ports to compression chamber and uncorrected barometric pressure variations on P_s
Δ S=motor speed variations

Analysis shows that the largest contributor to ΔQ is $\frac{\partial Q}{\partial V_c}\Delta V_c$.

Using expression (1) and performing the operation indicated by equation (2):

$$\Delta Q_{V_c} = -\left(\frac{S P_d}{P_s}\right)\Delta V_c$$

To compare various types of reciprocating compressors, we can define a variation index:

If the compressors operate at the same rotational speed, and if they are referenced to the same discharge and inlet pressure, then the ratio of WOB·L capacity variations to diaphragm capacity variations can be approximated by the variations in their respective clearance volumes, or:

$$\text{variation index} = \frac{\Delta Q_{V_c \, WOB\cdot L}}{\Delta Q_{V_c \, DIAPHRAGM}} \cong \frac{\Delta V_{c \, WOB\cdot L}}{\Delta V_{c \, DIAPHRAGM}}$$

A production line random sampling of 71, 1/8th-horsepower diaphragm compressors and 42, 1/8th-horsepower WOB·L piston compressors by the Quality Control Department from September of 1976 to February of 1978 showed a capacity variation ratio of 0.28 (the ratio of capacity variations for a WOB·L to a seven-part tolerance diaphragm compressor standard deviations). Analytical results for the same two units resulted in a variation index of 0.21.

Analysis results for several other types of reciprocating compressors, realized variation ratios ranging from 0.2 to 0.45.

<u>Reliability</u>
An important contributor to the ultimate life of the WOB·L compressor becomes the teflon cup.

Test results show the typical wear pattern for this type of sealing element. Initial transfer of teflon material to the sleeve micro-grooves cause a rapid cup thickness reduction of several thousandths of an inch during the initial operation. During this time period, there is an increase in sealing ability and a reduction in cup/sleeve friction level. Following this initial "wear-in" period, there is an extended period of time where normal reduction in cup thickness is roughly proportional to the pressure-velocity product. Towards the end of the useful life of the compressor, the cup loses sealing ability and piston radial clearances allow sleeve/piston cup shelf interference.

Cup Wear
Model 607 Wob-I Comp.

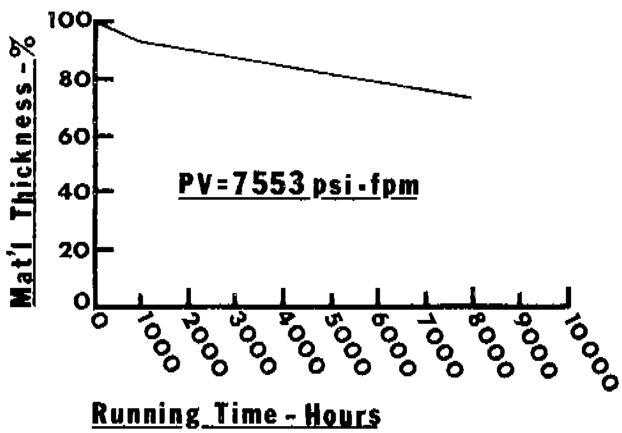


Figure 12 shows the cup wear rate for a Model 607 WOB·L compressor presently on life test operating at 80 PSI. Published wear factors (K)* for our cup material riding on an anodized aluminum surface (12-16 in. RMS) are in the 2 to 3 x 10^{-10} $\frac{in^3/minute}{lb/ft/hour}$ range.

Following the initial wear-in period, we are actually achieving a wear factor of 1 x 10^{-10} $\frac{in^3/minute}{lb/ft/hour}$.

* wear factor $K = \frac{R}{PVT}$

R=radial wear, in.
P=pressure, PSI
V=velocity, FPM
T=time, hours

Conclusions

The design and construction simplicity of the WOB·L piston style compressor has resulted in an economical and reliable compressor whose performance is dependant on several key parts.

This simplicity has produced several side benefits, one of which is the ease it lends itself to straight-forward analysis techniques. This provides Engineering, Manufacturing, and Quality Control personnel with a tool to further improve the overall success of the compressor. For example, investigations into statistical design techniques has started with the possible step improvement in capacity.

REFERENCES

1) Droege-Bell, U. S. Patent #3,961,868 and #3,961,869.

2) Chardon, C. C., and Roy, I. J., "Packing the Maximum Fan in the Minimum Space", Machine Design, September, 1973.

3) Burington, R. S., "Handbook of Mathematical Tables and Formulas", Handbook Publishers, Inc., 1956.