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**BLOWER NOISE AND SOLUTION:  
AN INTRODUCTION IN THE A.W. CONVEL BLOWER**

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**INTRODUCTION**

The paper describes the sources of noise in high speed positive displacement blowers and relates them to the processes involved, with the conclusion that some of these sources are inherent in all conventional types.

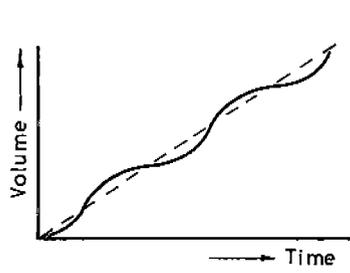
From a theoretical conception of an ideal process, the stages of translation of theory to practice are given.

A requirement existed for a blower of 0.24 Kg/sec (413 cfm) air flow for a low noise two stroke engine project. This set the very difficult target of having to design the mechanism of the new blower to fit the same space as the existing Roots type blower, run at the same speed and offer a substantial reduction of noise. Compromises from perfection had to be made but test results given show this new type of blower to give the reduction in noise necessary for the 80 dBA two stroke engine of the 1980's.

**1. NOISE IN THE COMPRESSION PROCESS**

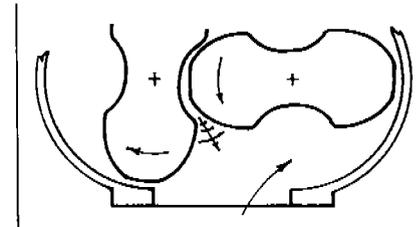
The problem is to take in, compress and deliver air without making noise. This offers an interesting challenge to the designer. Particularly so when one realises that the object is to carry out something of the order of 200 complete thermo dynamic cycles per second and that the audible frequency range is 20 to 40,000 cycles per second. One tenth of a thermodynamic cycle would be equivalent to a frequency of 2,000 Hertz and so disturbances of this order are very important. The main sources of noise need describing and are shown diagrammatically in Figure 1. Noise is a repetition of a change of pressure and so as air velocity and pressure are interchangeable, pressure being proportional to velocity squared, so a fluctuation in air velocity as shown by Figure 1a is a source of noise and should be avoided.

**Flow Rate  
Fig. 1a.**



**Implosions  
-Explosions  
-Squish**

**Fig. 1b.**



To stop and start flow every cycle at least creates a noise of the frequency of operation of the cycle, usually it excites a lot of other noise too, so the target is a constant and continuous velocity at inlet and delivery. To create uniform velocity flow the mechanism causing flow should give a linear change of volume with time, not as is the usual, have a working volume which tends to follow a sine curve.

Secondly, as depicted by Figure 1b, explosions or implosions of pockets of air create a lot of noise, not only by exciting the natural frequency of the space in which they occur, but also all other spaces that they meet in their travels before they are damped out. It takes very little energy to make a chamber resonate if the energy is supplied at regular intervals at some harmonic of, or in tune with the noise waves.

**Vortex  
Fig. 1c.**



Thirdly, there is turbulence. Eddies, as shown in Figure 1c, contain air moving at greatly differing velocities and so they are in effect pressure disturbances and make noise. They are the major source of noise in the centrifugal blower, and build up to vast proportions when tip speeds go up, say to the level of the blower of a jet engine. However, in the context of the positive displacement compressor the importance of this noise source lies in that this is one way in which air leakage

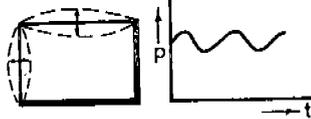
makes noise. You have only to crack open a valve on an air bottle to discover that a very small leakage gap can make a very big noise when the pressure drop is high.

Transmitted Pulse Fig. 1d.



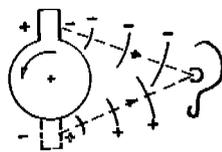
An important feature of leakage is the rate of change of leakage. Figure 1d shows that a varying pressure on one side of a leakage gap produces a similar but smaller change on the other side. This is one way in which delivery pressure fluctuations are transmitted to the intake side. Such delivery pulses can also cause a mechanical change of shape of the structure as represented by the buckling panels of a box in Figure 1e.

Panel Pulsing Fig. 1e.



Lastly, there is the noise caused by rate of change of shape and depicted in Figure 1f. This is of great importance and yet is rarely recognised as a source of noise. It can readily be understood when it is a change of surface shape resulting from mechanical vibration, such as a loudspeaker diaphragm but it must be recognised as also resulting from the rotation of any non concentric form.

Rate of Change of Projected Shape Fig. 1f.



So having laid down the features for analysis, existing types of blower may be examined to see whether they have been just designed, without recognition of noise requirements or are inherently unsuitable for future requirements.

## 2. PROBLEMS WITH EXISTING BLOWERS

The requirement of a smooth continuous air flow writes off practically every known design of positive displacement compressor. It is bad enough that the intake velocity is usually linear plus a super-imposed sine curve, but it is worse on the delivery side. With the Roots blower air is allowed to blow back, in order to carry out the compression. With machines having internal compression it is usual to have a period of no air delivery while the compression is taking place. The best that has

been done is to make a number of cycles overlap, as in the sliding vane type having a large number of vanes. This type is good also in that it does not present a rapid change of shape to the air passages. The Roots and screw types are bad in this respect in that the meshing of the rotors presents a face to the air which is in effect a rapid buckling diaphragm. When looking at a pair of rotors rotating slowly one sees part of the total surface coming towards you and parts going away, changing all the time. These movements are comparable to the ripples which make the noise on a loud speaker diaphragm, so it can readily be understood that a pair of meshing rotors is an intense noise maker.

Most types of blower have mechanisms which create pockets in which air is carried back from the high pressure side to the intake side of the rotors. These create minor explosions and so noise. Designers do try to minimise such pockets and expand the air that they contain slowly before opening it to the intake space, but when one considers the operating speed any disturbance taking only part of a cycle is fast enough to create noise.

For blowers having a working clearance between parts the leakage does not differ greatly between the various types, clearances depend upon the same machining tolerance of about 1 thou per inch and the length of leakage round the working volume is surprisingly constant for machines of economic proportions. However, the leakage does depend upon the pressure and so any process taking place in the machine which causes the delivery pressure to be exceeded will make unnecessary noise. In the Roots blower the back flow almost doubles the pressure on the lobe and on its clearance gap. This surge of leakage creates a pressure wave in addition to increased leakage hiss.

Applying this analysis, it appears that no existing type of machine is going to offer the basic requirements for low noise. The sliding vane type is best but unfortunately this type involves reciprocating motion which limits the speed and so it does not compete well on the grounds of size, cost and life, with the screw or Roots type. It is concluded that a better machine is needed, so the next question is can it be designed.

## 3. AN IDEAL COMPRESSION PROCESS

Having determined what is wanted, an ideal compression process can be synthesised from the basic requirements. Figure 2a shows the first requirement, a pipe with air travelling at a steady velocity which is somehow compressed at the change of pipe section to be delivered at the same velocity along a smaller pipe.

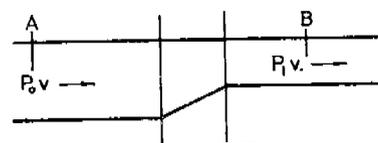
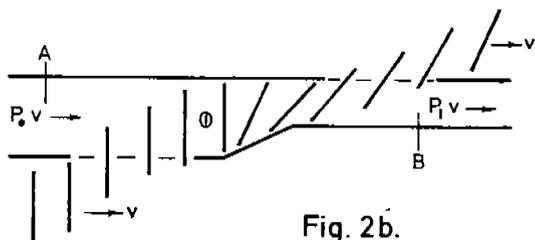


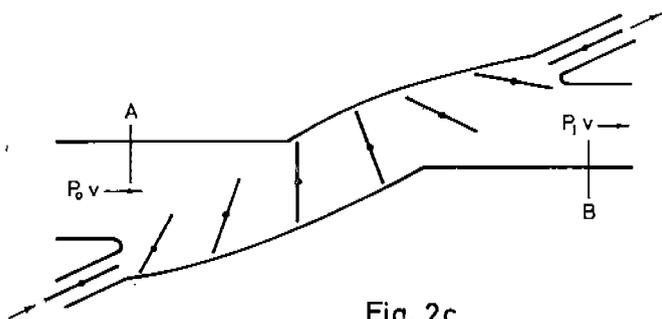
Fig. 2a.



**Fig. 2b.**

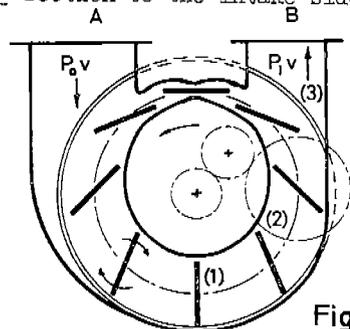
For it to be a positive displacement machine there must always be a barrier across the air flow. A series of barriers will divide the air flow into working volumes - and Figure 2b shows barriers taking the form of thin blades, sliding across the air flow to trap individual volumes at the entry to the compression section at 1. The blades must move at the speed of the air - and get through the compression section they tilt. The blades then travel down the delivery pipe, still at a constant velocity and somehow leave the flow.

The tilt can become a steady rotation as in Figure 2c.



**Fig. 2c**

What is then required is to get the blades back to the beginning of the cycle again. This can be done by bending the whole diagram into a circle as in Figure 2d. Air then enters at A is trapped 1, compressed at 2, enters the delivery at 3 and is discharged at B. The blades pass through a narrow sealing section to the intake side again.



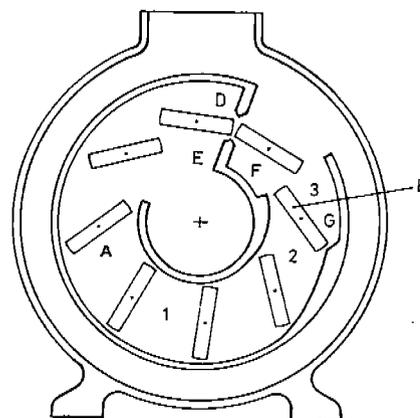
**Fig. 2d.**

All that is then required is mechanism to guide the blades.

Fortunately, geometry favours this motion and it is easily obtained, all this requires is a two to one reduction gear, between the blades and the rotor, their tips then describe a geometrical shape known as the Limacon of Pascal. This gives the unique

feature that the blades always point to the narrow slot, and so they feed themselves through it perfectly. On this theme, a nearly ideal basic compressor design could be a horse shoe shaped rectangular section air channel having three solid walls, and the open side closed by a rotating disc, which supports a number of half speed rotating blades. This mechanism is not novel, a Dr. Karl Rabe patented a similar design many years ago. So why did it not sweep the market as the ideal blower? The answer lies in the dynamics. With the air passing through the clearances and leaking back at least 150 m/s (500 ft/sec) the body of air must move forward at over 15 m/s (50 ft/sec) to get a useful volumetric efficiency, and 30 m/s (100 ft/sec) is a reasonable design feature. Bending this air stream to say a 8 cm (3.14") radius gives a running speed of 3600 rpm, which is reasonable, but the centrifugal loading becomes 1166 g. With this loading not only would the blades bend but the bearing loads would be so high that the friction losses would make the machine uneconomic. So this was the intrinsic fault that killed the synthesised perfect blower, and is the one which must be overcome to achieve a practical machine.

The solution to this problem is to keep the back plate still and rotate the casing. Figure 3 shows the change. With this design only the rotor is subject to the large centrifugal forces and with careful design these can be handled without compromising the working cycle. The blades now rotate in a stationary back plate which also supports the bearings of the rotor. This takes the form of rotating annular channel which is wide at 1



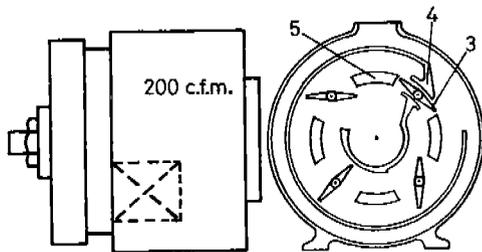
**Fig. 3 Parent Blower Form**

and narrows to B for compression as required by the elementary process of Figure 2. The channel is divided by a barrier DE which separates the intake from the delivery and in effect forms a rotating piston. The circular form of the rotor permits continuous intake of air from the centre and delivery from the circumference so that by putting it into a box a new type of compressor results.

Turning the machine about makes the compression space stationary and so the intake velocity head is lost but otherwise it does not lose the features of the ideal compressor which should offer a low noise

production. Now, instead of the blades moving with the air stream as 'effective pistons', it is the closed section of the rotor DE which acts as the effective piston - a piston moving at a constant velocity and so creating a steady inlet flow. The blades still trap the air at 1, hold it during compression 2, and introduce it to the delivery at 3. It is interesting to note that all the work is done by the rotor which is rigidly fixed to the shaft, with the result that the power is not transmitted through any mechanism. The blades are in fact no more than rotary valves and need only a light phasing gear train. Even in the trapped volume it is not the blades but the sloping sides of the rotor channel F & G which cause the compression.

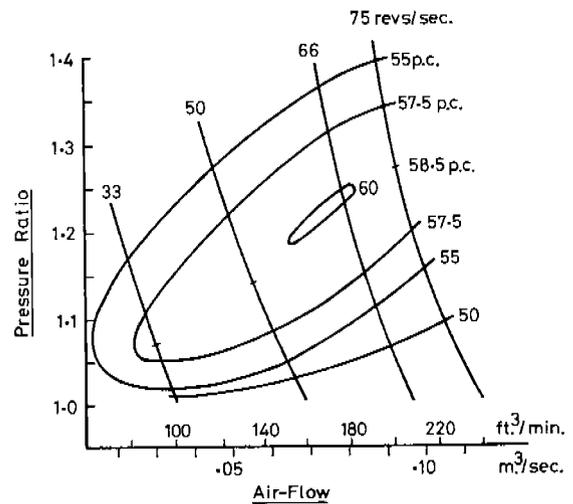
So the question remains, will this now work? Or, is there some other hidden snag? No snag, but the problem of turning theory into economic practice. A whole family of blowers can be designed from this parent form, the problem becomes that of choosing the design with the greatest potential for satisfying the specific requirement. The first version, which is shown in Figure 4 is now chiefly of historic interest but it shows the first of the simplifying modifications that were made.



**Fig. 4. B.I.C.E.R.A. Blade Type Blower**

These were to reduce the number of blades by fitting alternate spacing blocks, and easing the tolerances on the gear train by making the rotating seal on the rotor 4 span from the spacing blocks, 5 to the hub. This meant that it never had to seal on the blade tips and so there was no accurate phasing to be maintained. In practice this latter feature proved to be of value only during assembly as gear quality could not be reduced without loss of life and increased noise.

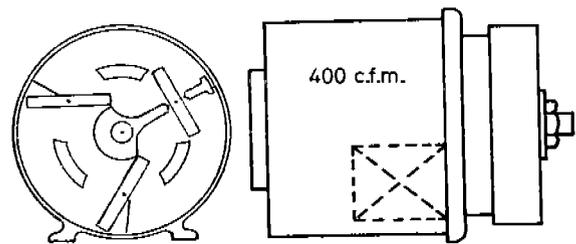
After a short period of development the performance was quite good for a low pressure blower and the characteristic curve is shown in Figure 5.



**Fig. 5. Overall Adiabatic Efficiency Characteristics of Blade Type Blower**

#### 4. THE CONVEL VERSION OF THE BLADE TYPE BLOWER

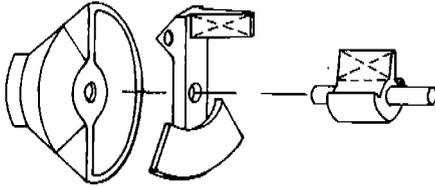
In 1970 Sir W.G. Armstrong Whitworth instigated a programme of noise research on two-stroke cycle engines. Work already done at the Institute of Sound and Vibration Research had shown that the two-stroke engine was not particularly noisy in those aspects of noise that were difficult to control except in that the Roots type blower was an additional source of noise. The study of sources of blower noise given in this paper led to the conclusion that the Blade type offered most potential for a quiet compressor. However, two-stroke engines use large volumes of air and size became a major problem. It was also desirable to eliminate the pockets of air carried over between the blades and the channel. A solution was found when it was discovered that a design of blade could be chosen which would make a seal directly on the outer casing, as shown diagrammatically in Figure 6.



**Fig. 6. Convel Blower**

This design has been called the CONVEL Blower because of its design for constant air velocity. In this design the choice of blade thickness is based on the arc over which the blade tilts during its air sealing period. These thick blades are shaped to have a constant radial thickness and as the lobe passes over them they form a perfect continuity of clearance with the spacing blocks, so avoiding any 'carry over' volume.

With this new design not only was the circular space round the outside of the rotor drum brought into the swept volume, but also because the blades had a greater diameter and thickness, the length could also be increased and gave nearly 100 per cent increase in throughput for a given diameter of casing. Other limitations then required further consideration.



**Fig. 7. 'Anchor' Type Rotor**

The flow areas had to be increased until the rotor was reduced from a drum containing a channel and ports, down to a rotating bar, as shown in Figure 7 which supports at one end the outer lobe which does most of the work and the other the sealing plate which closes the end of the trapped volume and acts as a rotary valve to determine the timing of the trapped volume. To divide the intake and delivery air flows required a rotating air duct which feeds the inlet side of the rotor whilst the air from the delivery side occupies the space round the outside of the duct. The inner lobe and hub complete the rotor assembly. The hub surface on which the blades seal can be an arc of a circle but is slightly eccentric. The smaller the radius of this arc the greater the air flow but a compromise has to be chosen to leave enough metal for balancing. Each component is designed to be in dynamic balance on its own so that internal stresses will not cause distortion.

The resulting improvement in size and shape is shown by comparing Figures 4 and 6 in which the throughput is shown to be doubled for an increase in length of only about 40 per cent. There is still potential for improvement in throughput per unit bulk by increase of speed or increase in blade length. Only by tests can the practical limits of output be found. So far the blade length to diameter ratio has been kept at about 1.2. The centrifugal force on the outer lobe is probably the major limitation to increasing speed. Fortunately the depth of section increases with blade diameter and so with larger blades a longer lobe can be used without reduction in running speed.

The specific requirement of this machine was to be interchangeable with the Roots type blower of the standard Chrysler TS3 rocker opposed piston engine. The particular design compromise chosen was influenced by this fact which dictated the running speed, mounting flange, and delivery and intake duct positions.

The photographs at the end of the paper show this blower. Figure 8 shows the size to be comparable to that of a Roots type blower. In the exploded view

in Figure 9 it can be seen that a tapered 'I' beam was chosen for the lobe. This is bolted to the rotor arm right through the counter balance weight so that a good stretch length was obtained for the bolt, the adaptor plate - below the rotor - was specific to the engine and not an essential part of the blower.

Normally the length of a blower is of small importance compared with its diameter but in this case the overhang from an end flange mounting sets a length restriction to the design and for this reason the blades were not, as would be usual, overhung from the gear case as this would require a wide bearing spacing. Instead needle roller bearings were fitted in their free ends and supported on a ring attached to the ends of the spacer blocks. The blades were in steel and the spacers and back plate were made in cast iron, the rest of the blower being in Aluminium. This use of iron made the blower a little heavier than the Roots blower which was all aluminium.

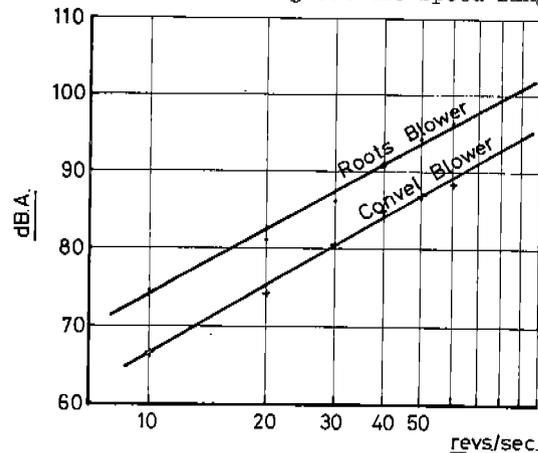
### 5. PROTOTYPE CONVEL BLOWER

The specification was:-

- Shape - Cylindrical 10 inches dia. (26 cm) 12½ inches long (32 cm).
- Weight 52 lbs (23.7 kg) Roots type 48 lbs (22.1 kg).
- Blades 3⅝ inches (86 mm) dia. 4 inches (101 mm) Long 0.475 inches (12 mm) thick.
- Pitch circle diameter 5.92 inches (150 mm).
- Nominal Swept Volume (3.375 - .475) x 4 x 5.92 x = 216 cuin. (3.54 l).
- Max. Speed 5,000 rpm. (83 rps).
- Target output 413 cfm (186 l/s) at 6 psi (0.4 bar) at 4,350 rpm (73.5 rps) and Power 18 hp. (13.5 KW).

### 6. PERFORMANCE

For a comparative noise test the blower was driven by a variable speed electric motor. Readings were taken at 3 feet (0.9 m) from the open intake with the delivery air ducted away. Figure 10 shows a reduction of 7 dBA throughout the speed range.



**Fig. 10. Noise Reduction of Convel Blower**

This included an unnecessary degree of gear noise caused partly by a small eccentricity of the gears which had simple machine cut soft teeth - the idlers being in bronze and rest steel. Cladding the gear cover removed this noise for demonstration on the quiet engine. This had already been given noise reduction treatment at I.S.V.R. and with the Convel blower fitted, noise levels of 96 to 97 dBA were obtained, which is equivalent to about 83 dBA in a vehicle on a drive past test. Normal conversation could be maintained in the engine test cell at full speed and load.

## 7. DISCUSSION

Testing to date has shown that the Convel offers a big step forward in the design of positive displacement blowers. It has proved reliable over about 100 hours of engine running and is as effective as the Roots blower in supplying the engines air requirement.

The Roots blower has always been the smallest positive displacement machine for a given swept volume and to compete so closely in ratio of size to displacement shows great promise for commercial production. Very often Roots blowers have to be derated in speed to make the noise acceptable and when this is done the size has to be increased to achieve the same throughput. Under these conditions the Convel will be very much smaller than the Roots blower. Roots blower noise has been shown to increase almost linearly on a base of log tip speed x number of lobes per rotor - and the size to have practically no effect on noise. This means that to save 7 dBA the Roots blower has to be reduced in speed by a factor of 2.3 and increased in size by a linear factor of 1.32 which multiplies the bulk, weight and cost by 2.3. This would make the Roots blower weigh 51 kg (110 lb) against the Convel at 23.7 kg (52 lb).

The Convel design is not quite as convenient as the Roots for cutting rotor length to suit specific throughputs and speeds from a given rotor diameter. However, five throughputs could be obtained by having two different blade lengths and doubling up blowers back to back or front to front within a single casing. For larger machines it is attractive to have two rotors on a single shaft as it eliminates end thrust on the rotor bearings and the symmetry makes balancing easier. It also doubles the throughput at less than double the cost against nearly four times the cost if the single rotor design is scaled up in size.

### Two Stage

With rotors mounted back to back they can be coupled in series to give two stages of compression with direct internal transfer of air from one side of the rotor to the other. This offers the potential of full filling the difficult requirements of reaching pressures of the order of 20 lb/in<sup>2</sup> (1.35 bar) without excessive noise.

## 8. CONCLUSIONS

Existing designs of rotary positive displacement

blowers do not satisfy the requirements for low noise production.

The Convel design, which has been based on the elimination of noise producing processes, has shown itself to be practical machine and offers a substantial reduction in noise for a size very little greater than the Roots type.

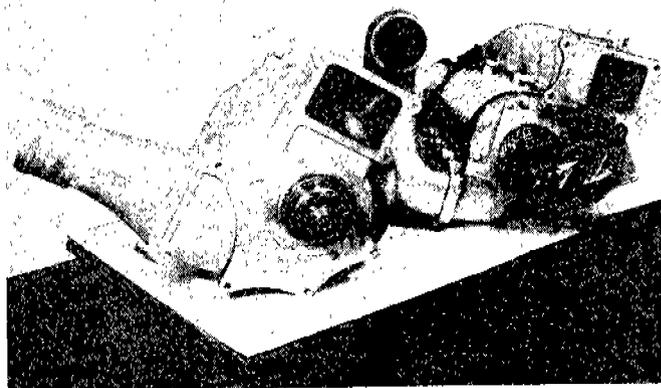


Fig.8. Convel Blower Comparison

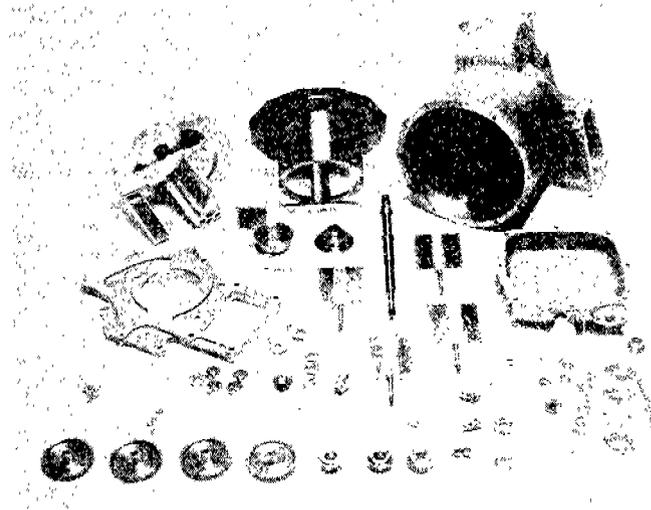


Fig.9. Exploded View of Parts of Convel Blower