1986

New Internal Capacity Control for Reciprocating Compressors

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NEW INTERNAL CAPACITY CONTROL FOR RECIPROCATING COMPRESSORS

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ABSTRACT

For several applications of reciprocating compressors, the instantaneous matching of the working fluid flow rate with load variations is needed, the compressor flow rate being variable thanks to external or internal parameters and characteristics.

The performances of both compressor and prime mover strictly depend on the capacity control system characteristics, that, in their turn, are deeply affected by the operating conditions of the overall plant itself. The suitability of such capacity control devices shall be evaluated referring to the overall efficiency variations.

As far as these design choices are concerned, the advantages of the internal control devices have readily been demonstrated, both theoretically and experimentally.

A basically new concept of mass flow rate control device has been developed, so called N.I.C.C.(new internal capacity control), suitable for reciprocating compressors and capable of matching every load variation thanks to instantaneous, continous variations of the compressor flow rate.

Both theoretical and experimental analysis of the system have been carried out, whose results showed good agreement and a wide adaptability of such a new variation delivery device, basically advantageous in comparison with the conventional ones.

SYMBOLS

- C Stroke
- D Diameter of the cylinders
- L Connecting rod length
- m Ratio between specific heats
- r Wheel radius
- V Overall work volume
- Vo Dead volume
- Vs Sweep Volume
- X Axial movement abscissa
- Y Displacement of the tooth profile
- α Phase shift angle
- β Single piston phase shift angle
- γ Arctg (X/Y)
- δ Loss factors
- η_ϕ Leakage coefficient
- ρ Compression ratio
- ϕ Crank angle
- λ_τ Reduction coefficient for walls heating
- λ_v Charge coefficient
- μ Dead volume parameter (Vo/V)

INTRODUCTION

The adjustment of the flow rate of the reciprocating compressors is a requirement which becomes particularly important for powerful machine. While machine performance must often be adjusted to the requirements of the circuit, sometimes the compressor itself operates as the adjusting element by determining the operating conditions for the entire system / 1;2;3 /.

The capacity control at different operating conditions can be obtained using various systems, but in any case it's more appreciable the more vast and continuous the range is and the lower the variation in efficiency which accompanies it.

It is well known that the systems for capacity control in reciprocating compressors can be subdivided into systems inside and outside the machine / 4; 5 /. Obviously in addition to the mechanical and thermodynamic characteristics of the compressor, the choice depends on the following:

a) Plant operating conditions such as:
   - performance to be adjusted and fast operation requirements
   - degree of capacity control required and fraction of operating time at reduced flow rate
   - effect of the variation in efficiency;

b) Characteristics of the machine which concern the capacity control;

c) Cost of the system used.

This study presents a functional compressor design solution (New Internal Capacity Control - N.I.C.C.), which to permits vary its geometry and adapt it to the required operating conditions.

In addition the results are presented of the experiments performed on the prototype, comparing them with those obtained using other capacity control systems applied to the same compressor.

1 COMPRESSOR SCHEMATIC

Almost all traditional systems leave the geometry of the machine unchanged, while the N.I.C.C. here proposed is based on a compressor design which permits to vary continually all the geometric parameters which define the work cycle.

Fig.1 shows the schematic of the compressor, while Fig.2 details the kinematic of the coupled pistons / 2 /.

As shown in the scheme, the total work volume consists of two cylinders and in every instant the space available for the gas is determined by the relative position of the moving pistons.

The crank mechanisms of the two pistons are kinematically connected with the helical parallel coupling in which the idle wheel can move axially.

The respective axial slippages of two helical sections determine the reciprocal rotations of the generators. In the first approximation valid for small shifts, the following expression is possible to write (Fig.3):

$X = Y \ tg\,\gamma$ and: $Y = r \ \sin\beta$

from which: $X = r \ \sin\beta \,/\, tg\,\gamma$

This relationship connects the axial slippage X to the resulting phase displacement angle. Because of the opposing inclination of the serrated sections, the coupling of the two shafts is dephased by an angle $\alpha = 2\beta$. The variation of the total volume function introduced by the above-described kinematism is expressed by:

$$V=\pi D^2\left\{C+2L-C(\cos\Phi-\cos(\Phi-\alpha))/2-\sqrt{L^2-C^2 sen^2\Phi/4}-\sqrt{L^2-C^2 sen^2(\Phi-\alpha)/4}\right\}/4+Vo$$

and is represented in Fig.4 plotted versus the crank angle Φ, for various values of α.

2 THE PROTOTYPE AND THE EXPERIMENTAL EQUIPMENT

A prototype has been built made in the Laboratory of Dipartimento di Energetica of Ancona University, in order to compare its performance during the adjustment phase with that performance obtained using traditional systems.

The performance of the machine has been measured by connecting it to the test system illustrated in Fig.6. This plant uses the traditional measuring system; in particular to measure the gas flow rate, an orifice meter and a differential water pressure gauge have been used in accordance with standard methods.

The pressures have been taken by piezoresistive trasducers connected with a multichannel amplifier and displayed on a four-track memory oscilloscope. For taking temperatures, T type thermocouples have been used.

To measure performances at costant delivery pressure and variable displacement, the stabilization tank (S1) and discharge valve (V1) have been used; this valve delivers the air flow necessary for maintaing the prescribed pressure in the tank.

The discharge is then conveyed inside the tank (S2) sized sufficiently to guarantee re-establishment of the physical parameters of the outlet gas.

3 THEORETICAL BEHAVIOR OF THE PROTOTYPE

The experimental phase has been preceded by a thorough theoretical investigation to better evaluate the actual behavior of the proposed system.

This investigation has been carried out by using the known relationships which describe the operation of the reciprocating compressors. The diagrams of Fig.5 schow the variation of the main geometric parameters of the prototype against the phase shift angle .

The decrease in the charge coefficient is due mainly to the presence of the dead space in which the residual gas stagnates and subsequently re-expands during the intake stroke thereby reducing the useful part.

Les influence have:

- the compressed gas leaks,
- the decrease of the gas density of the gas because of its heating by the cylinder walls and the throttling in the intake valve,
- the pressure increase inside the cylinder during the delivery phase which causes an increase in the residual mass in the dead space.

The influence of the loss factors on the charge coefficient is defined by the following empirical expression:

$$\lambda_v = \eta_\phi (1-\delta_1)\lambda_\tau \left\{1-\mu\left[\rho^{1/m}(1+(\delta_1+\delta_2)/m)/\lambda_\tau - 1\right]\right\}$$

By only taking into account the influence of the variation in the degree of dead space and leaving out the other loss factors , which were not in any case considered in this prototype and are also extraneous to the strictly theoretical approach / 1 /, is reduced to the following:

$$\lambda_v = 1 - \mu(\rho^{1/m} - 1)$$

In this particular case, the charge coefficient is a function of the dephasing introduced by the degree of dead volume and also by the ratio compression.

This function is represented in Fig. 7 which shows the strong influence the compression ratio has on the charge coefficient (λ_v) compared with the influence of the phase shift angle α. As for the volumetric flow rate plotted in relation to the compression ratio in

Fig. 8 , it obviously follows the same trend as the charge coefficient.

Finally, Fig. 9 shows the input power and the theoretical flow rate trends in relation to the phase shift angle and the compression ratio.

4 EXPERIMENTAL INVESTIGATION

With the above-described test plant, measurements have been made on the prototype according to the method indicated below.

A preliminary compressor test has been carried out with nil phase displacement. The test gave the performance values of the machine and the physical characteristics of the gas in order to define the behavior of the compressor with no adjustment present.

Subsequently other adjustment systems will be compared to this operating condition. Then the experimental measuring has been made with adjusting being effected by displacing the phase of the piston movement.

For each phase shift value and at different delivery pressures, the following have been measured:

- gas flow rate during delivery,
- electric power absorbed by the motor,
- gas temperature immediately downstream of the delivery valve,

The gas pressure has been taken in the four points indicated in Fig.6. while Fig. 10 shows the time progression as registered by the oscilloscope for the drawing pressure (P1), the gas pressure inside the compressor (Pi) and the delivery pressure measured immediately after the valve (Pv) and in the stabilization tank (P2).

The diagrams of Figs.11 and 12 show the flow rate and power trends upon the variation of the phase shift angle and the delivery pressures. The accordance of the curve trends with corresponding theoretical curves in Figs. 8 and 9 should be noted.

The diagram of Fig. 13 shows the temperature trends of the gas coming out from the compressor in relation to the delivery pressure and the phase shift angle, at particular values of α and P2. The congruence of these diagrams with those related to the absorbed power confirms the reliability of the tests.

The diagram in Fig. 14 which deals with the specific work trend in relation to the phase shift angle and to delivery pressure, shows the low influenze of α compared to the influence of P2. Therefore it's easy to forecast now very good performance for N.I.C.C. proposed.

Overall efficiency is then traced for different delivery pressures plotted versus the volumetric flow rate in Fig.15. The comparative analysis of the diagrams in Figs 11, 12 and 15 gives a complete picture of the behavior of the machine during the adjustment phase.

When the measurements of the phase displacement adjustment have been completed, the machine has been returned to its initial condition ($\alpha = 0$) and compressor performance measurements have been made by adjusting the flow rate using the following systems:

-Throttling the intake
-Throttled by-passing
-Variation of speed

The measurements have been completed by following methods used previously, and efficiency values represents the absorbed power trends in relation to the flow rate, for three of tested control systems (see Fig. 16).

The comparison is even more clear if based on the variation of overall efficiency in relation to the volumetric flow rate as shown in Fig. 17 where the validity of the proposed system is demonstrated.

An analysis of the results permits affirming that the proposed delivery control makes it possible to obtain better efficiency values compared with that which obtainable by-passing and throttling the intake ,without obviously reaching the values which can be obtained with the decrease in speed.

5 CONCLUSIONS

The experimental investigation has shown a good agreement between the performance of the prototype and that predicted by the theoretical analysis.

This encouraging result suggests that the research should be continued in order to design a new model capable of reducing the loss factors and of obtaining higher performance.

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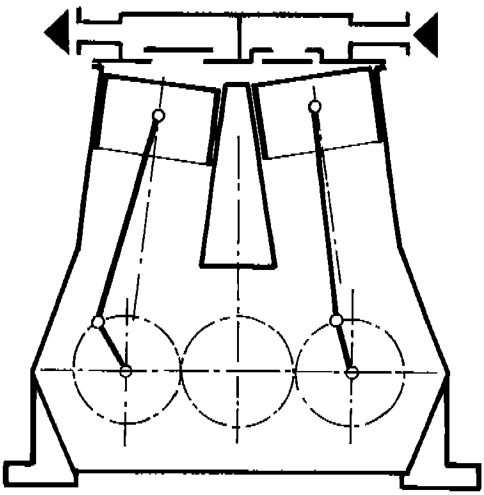

Fig. 1
Basic mechanical arrangement for reciprocating compressor with New Internal Capacity Control.

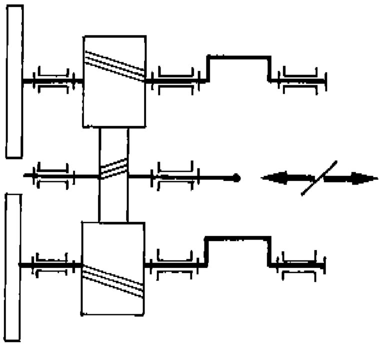

Fig. 2
Proposed mechanism scheme for variation of phase angle.

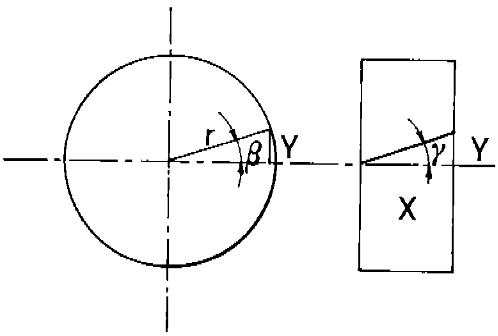

Fig. 3
Geometric scheme of phase shift angle generation.

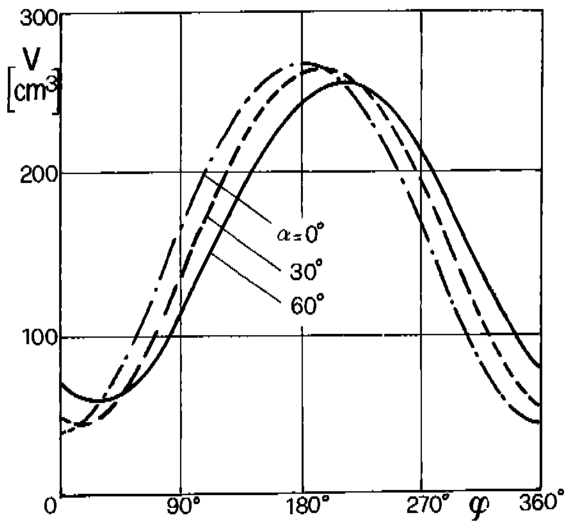

Fig. 4 - Overall work volume for various values of α

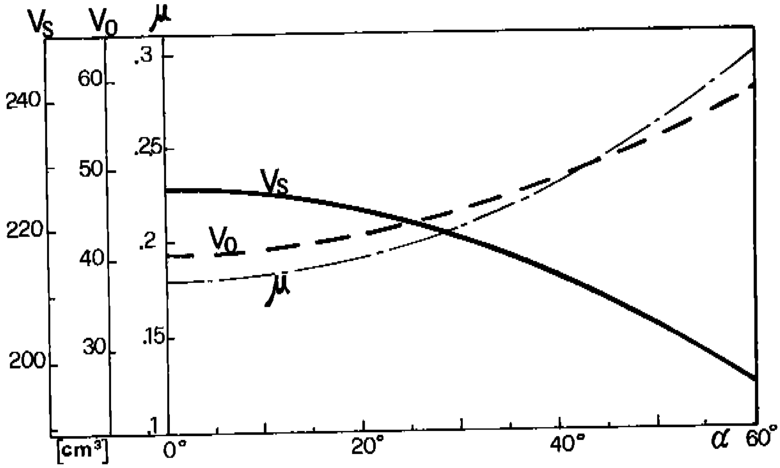

Fig. 5 - Variation of main geometric characteristics of the prototype versus phase shift angle

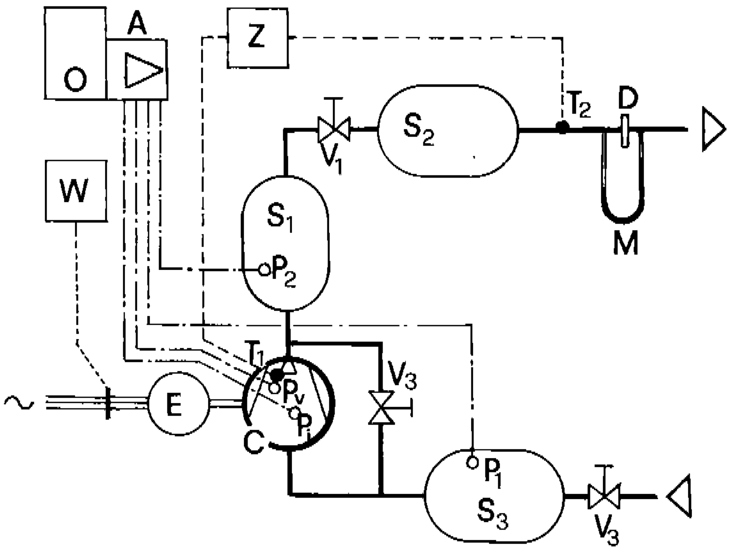

Fig. 6 - Schematic of test equipment: A, amplifier - C, compressor - D, orifice meter - E, electric motor - M, differential manometer with water column - O, oscilloscoper - P, pressure trasducers - S, tanks - T, termocouples - V, manual register valves - W, wattmeter - Z, displaiers

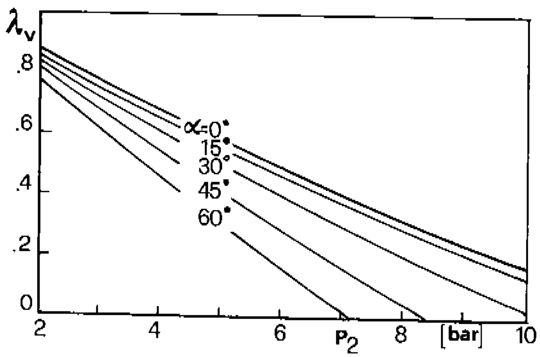

Fig. 7 - Charge coefficient trends plotted versus the delivery pressure at different phase shift angle

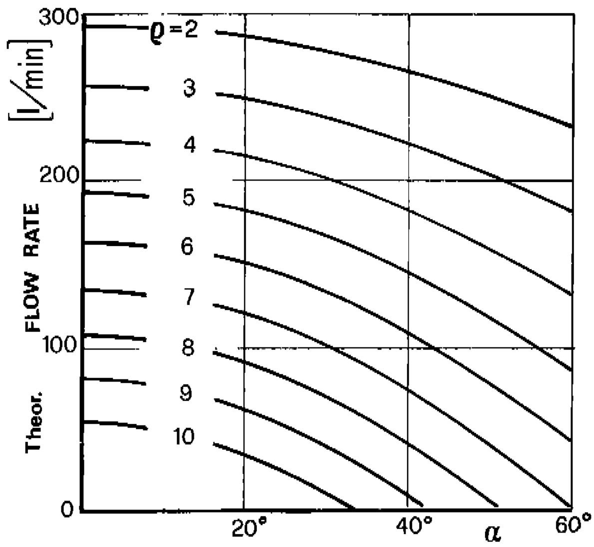

Fig. 8 - Theoretical flow rate against the phase shift angle for various value of compression ratio

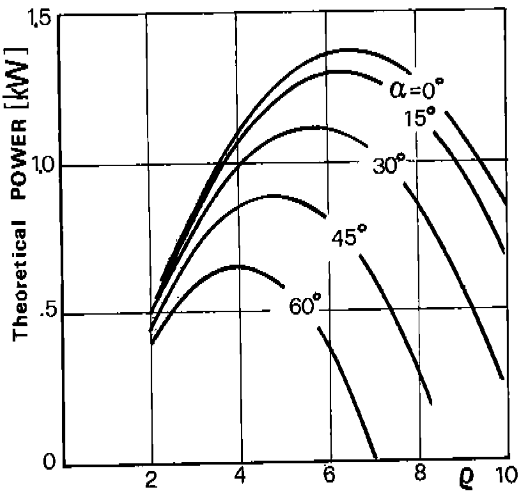

Fig. 9 - Theoretical power supplied plotted versus the compression ratio for various values of α

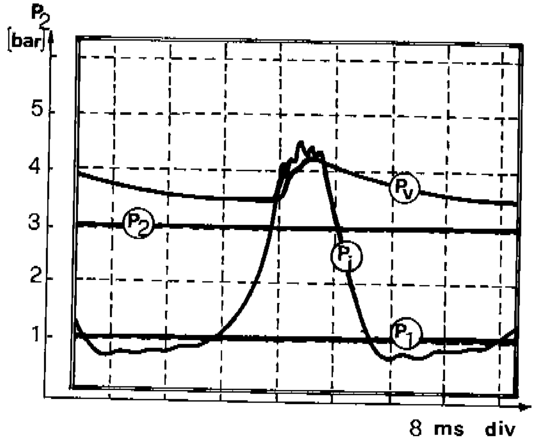

Fig. 10 - Pressure variations as showed on the screen of oscilloscope

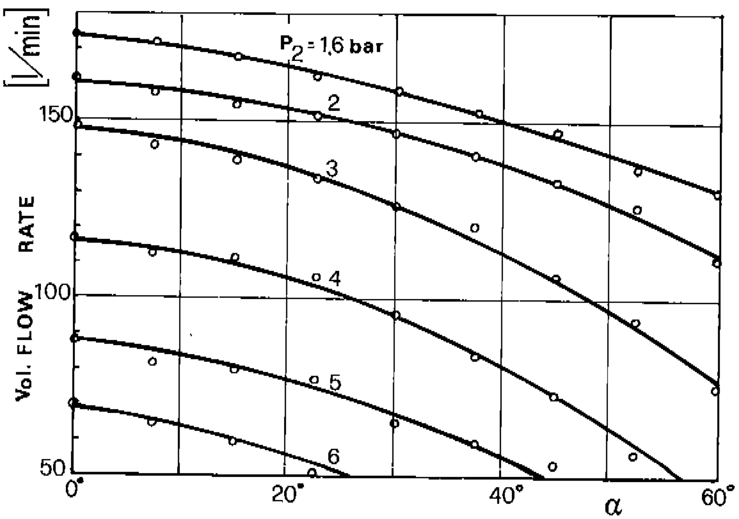

Fig. 11 - Adjustment of volumetric flow rate at different delivery pressures

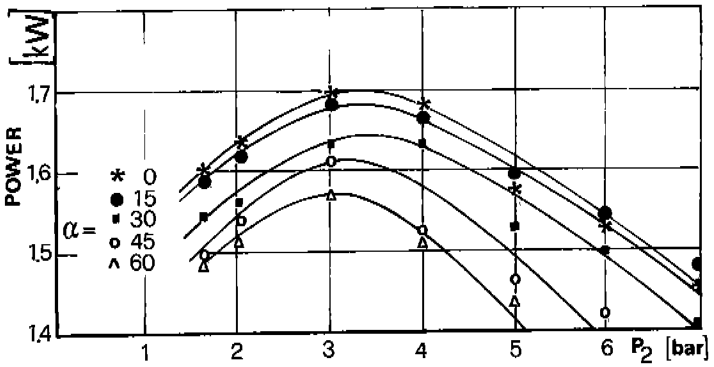

Fig. 12 - Absorbed power plotted versus the delivery pressure at different phase shift angle

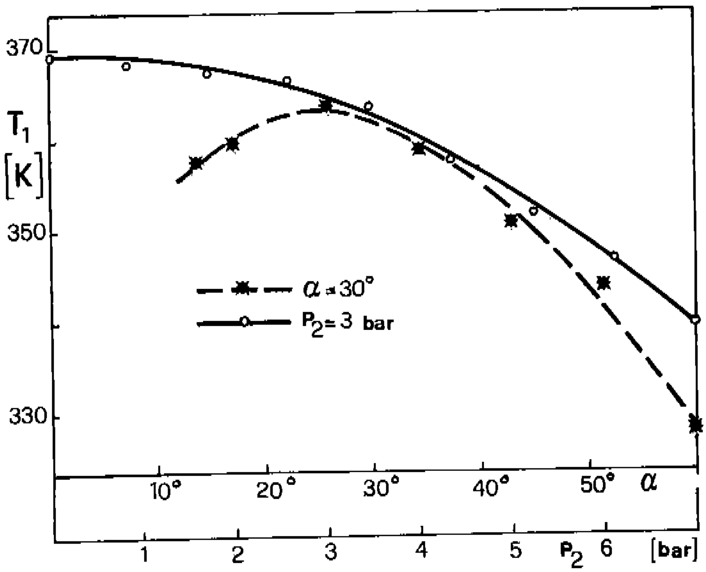

Fig. 13 - Temperature trends of the gas coming out from the compressor plotted versus P2 and α

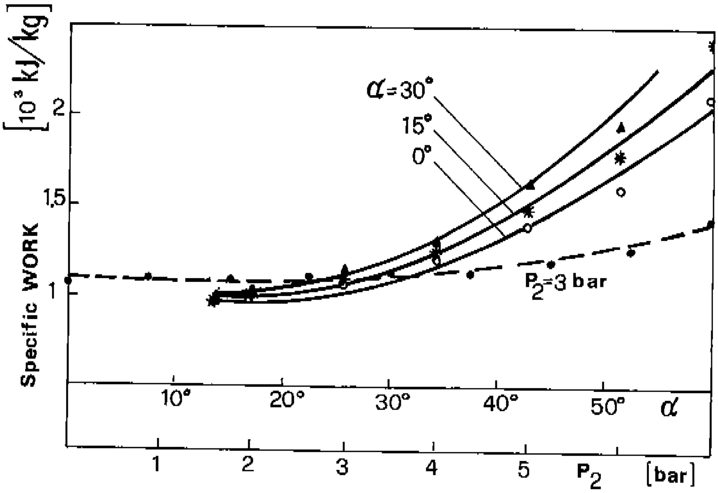

Fig. 14 - Specific work plotted against the phase shift angle and delivery pressure

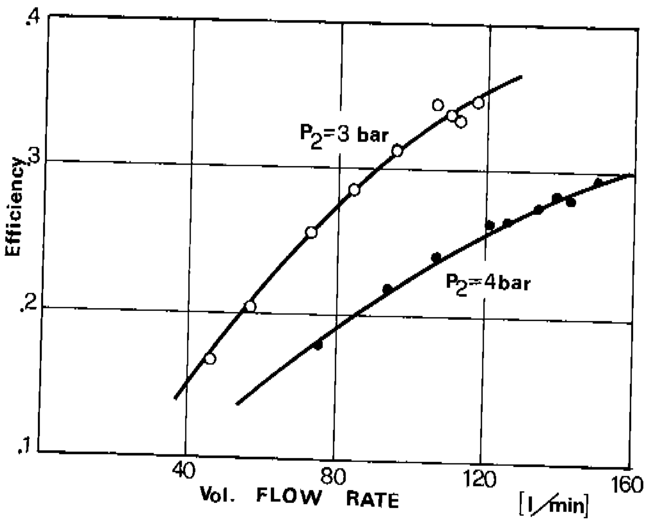

Fig. 15 - Overall efficiency plotted versus the volumetric flow rate at two different P2 values

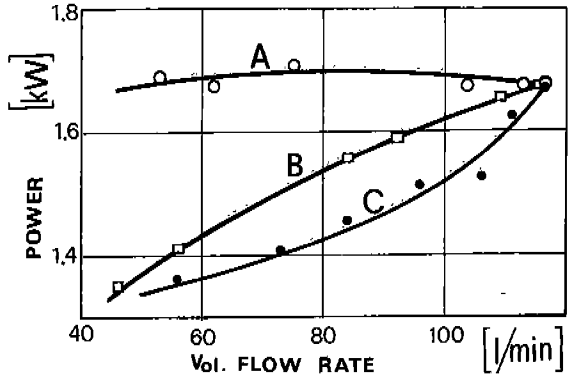

Fig. 16 - Absorbed power trends in relation to the volumetric flow rate: A, by-passing - B, throttling the intake - C, N.I.C.C.

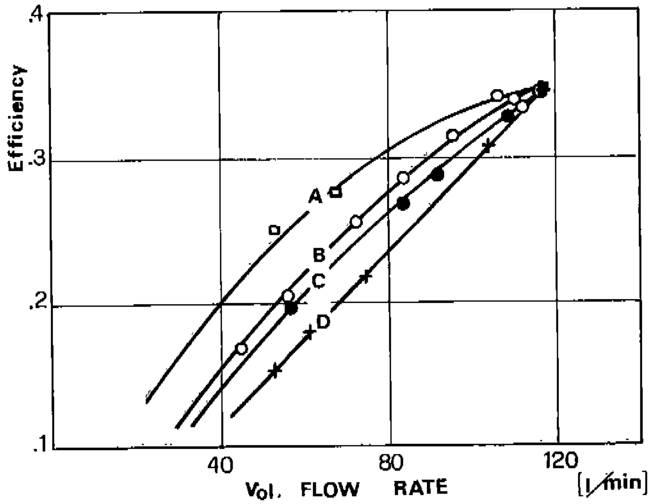

Fig. 17 - Overall efficiency plotted versus the volumetric flow rate: A, variation of speed B, N.I.C.C. - C, throttling the intake - D, throttled by-passing