1992

Optimum Height and Bore of Rotary Compressor for Obtaining High EER

T. Maekawa
Daikin Industries

N. Kato
Daikin Industries

OPTIMUM HEIGHT AND BORE OF ROTARY COMPRESSOR FOR OBTAINING HIGH EER

by

TAKASHI MAEKAWA[1], NOBUYUKI KATO[2]
[1] Engineer, Small Compressor Division, DAIKIN INDUSTRIES.LTD,
[2] Chief Engineer, Small Compressor Division, DAIKIN INDUSTRIES.LTD,
1000-2 OHTANI OKAMOTO-CHO KUSATSU JAPAN

ABSTRACT

In order to minimize the tolerance of theoretical analysis, we conducted the test with many kinds of prototypes to obtain actual performance data. We made 19 different kinds of rolling piston type rotary compressors with identical piston dispracement and obtained actual EER. Cyliner height H and bore D are thought to be dominant factors for determining the rotary compressor performance We, therfore, conducted the test varying H and D, and adjusted the test data using multiple regulation analysis to investgate what really determines the performance. The result showed that the smaller the product of H and D, the higher the efficiency. If the product of H and D is constant, large D results in a small increase of eccentricity and the reliability increases. However, large D results in a large diameter compressor.

INTRODUCTION

Recently in Japan, small size scroll compressors (approx.1/2 to 1.5KW) have made their apperance in the field of residential air conditioners. However, owing to their advantages and the development of improvements, rotary compressors are still in use. In addsion, the recent restriction on SEER has lead the research into a higher EER compressor.

The capacity of a compressor follows its suction volume which is determined by some of the major dimensions such as eccentricity ε, cylinder height H and cylinder bore D. Therefore, a compressor of same suction volume can be designed with various combinations of these dimensions.The power reqired for gas compression and the friction force on the sliding face vary by these combinations and accordingly the mechanical efficiency also varies.

Therefore, the determination of these dimensions is essential for obtaining higher efficiency. It is well known that the simulation based on kinematic equations, the equations of motion of the moving element, using the repetition method enables us to predict the performance and the efficiency over a wide range of dimensional variation. Though there are some investigations on optimum combinations, they are mainly based on only mechanical friction.(*1)

In addition, some of the recent investigations, in order to be more practical, deal with performance in relation not only to mechanical friction but also to the overall factors of leakage, overheat, pressure losses and motor loss. Thus the simulasion accuracy has become higher.(*2)

However, we found that these investgations are not yet sufficient for performance prediction with higher accuracy. Therefore, we investigated how these dimensions influence the performance by testing 19 different kinds of prototypes, We conducted the test with these same suction volume compressors and confirmed the total efficiency and its tendency. In order to obtain only the influences of these dimensions. we minimized the tolerance of machining component and the error of measurement during the test. For the evaluation of performance, we also used several other ways than EER for expressing efficiency. For processing data with some errors, we used the multiple regulation analysis for investigating how these dimensions affect performance.

DETERMINATION OF DIMENSIONS FOR PROTOTYPE

For our experiment we made totally 19 compressors with same suction volume. (Vsuc=51.3cc/rev) Sixteen of them have different combinations of dimensions , and another 3 of them have same combination of dimensions of No.7 but different shaft diameters. The specification of these prototypes are shown in Table 1. Unless the relasion ($Rin-\varepsilon > drshaft/2$) is satisfied. a shaft and a roller cannot be assembled.

Where Rin ;Inner radius of roller, drshaft ;diameter of rear shaft

As a result.there were some prototypes with small rear shaft diameter and crank pin diameter (roller inner diameter). The purpose of prototype No.18, No.19 and No.20 which have different shaft diameters is for investigating the infuluence of rear shafts and pins.

With regards to the Table 1, the prototypes shown in horizontal lines have same cylinder Bore D ,those shown in vertical lines have same cylinder height H, and those shown in diagonal lines have same eccentricity ε. For simplifying the fabrication of prototypes, the following component dimensions are made the same;

- The thickness of brade (4.7mm)
- Brade spring
- Casig (shell height ,radius)
- Front shaft diameter(ϕ 28)
- , Top radius of the brade
- , Motor (including rotor weigt baranser)
- , Discharge valve,valve stopper

A crank pin and rollar were used to adjust the centrifugal force. A spacing piece was inserted to adjust the spring load. Oil was charged to the upper level of cylinder. The compressor components were machined with high accuracy to maintain every clearance within a small tolerance (actual result was within$\pm 5\mu$).

DEFINITIONS OF EACH EFFICIENCY

$$EER = Q / W0 \qquad \text{.............} \quad (1)$$

$$= G0 \cdot \Delta ie / W0 \qquad \text{.............} \quad (2)$$

$$= \frac{W1}{W0} \cdot \frac{W2}{W1} \cdot \frac{WGmax}{W2} \cdot \frac{G0 \cdot \Delta ie}{WGmax} \qquad \text{.......} \quad (3)$$

in addition.

w3 : minimum work per unit flow rate (1 Kg/h)

W3 = G0 * w3 : minimum work of actual flow rate

$$w3 = WGmax / Gmax \qquad \text{.......................} \quad (4)$$

$$\therefore \quad WGmax = Gmax \cdot w3 \qquad \text{.......................} \quad (5)$$

where:

- Q : actual capacity (Kcal/h)
- G0 : actual mass flow rate (Kg/h)
- Δie : specific enthalpy increase during evaporation process (Kcal/Kg)
- W0 : motor input (Watts)
- W1 : motor shaft output (Watts)
- W2 : P-v gas compression work per cycle
  = area (1' 2' 3' 4') · N/60 · g (Watts) (Fig.1)
- N : RPM of the motor (rpm)
- P : cylinder inner pressure (Kg/mm^2)
- Vc : cylinder volume (m^3)
- g = 9.8 (m/s^2)
- WGmax : minimum power of P-V diagram required to compress the maximum cylinder volume gas
  = area (1234) N · g/60
  $= (n/n-1) \cdot PsVc\,[(Pd/Ps)^{n-1/n} - 1]\,N \cdot g/60$
  n : average polytropic index which is to be applied to above euation n = 1.115 (for R22)

$$\therefore \quad EER = \frac{W1}{W0} \cdot \frac{W2}{W1} \cdot \frac{WGmax}{W2} \cdot \frac{G0 \cdot \Delta ie}{WGmax} \quad \cdots\cdots \quad (6)$$

$$= \frac{W1}{W0} \cdot \frac{W2}{W1} \cdot \frac{WGmax}{W2} \cdot \frac{G0 \cdot \Delta ie}{Gmax \cdot w3} \quad \cdots\cdots \quad (7)$$

$$= \eta mot \cdot \eta mec \cdot \eta cid \cdot \eta v \cdot \Delta ie / w3 \quad \cdots\cdots \quad (8)$$

Where the definition of each efficiency is as follows

- $\eta mot = W1/W0$; motor efficiency
- $\eta mec = W2/W1$; mechanical efficiency
- $\eta cid = WGmax/W2$; P-V indicator efficiency
- $\eta v = G0/Gmax$; volumetic efficiency

$\Delta ie/w3$: represents max EER
at the condition te5/tc55℃, R22, max EER=3.73
(in ASHRAE cond. max EER=4.14)

We further deffine another efficiencies as follows:

- $\eta c = \eta mec \cdot \eta cid \cdot \eta v$; compressor mechanical efficiency
- $\eta t = \eta mot \cdot \eta mec \cdot \eta cid \cdot \eta v$; total compressor efficiency
- $\eta d = \eta mec \cdot \eta cid$; indicator mechanical efficiency

$$EER = 3.73 \cdot \eta mot \cdot \eta mec \cdot \eta cid \cdot \eta v \quad \cdots\cdots \quad (9)$$

$$= 3.73 \cdot \eta mot \cdot \eta c \quad \cdots\cdots \quad (10)$$

$$= 3.73 \cdot \eta t \quad \cdots\cdots \quad (11)$$

While, for the mass flow rate
WG0 : minimum work per actualflow rate G0 (Kg/h) (=W3)

$$WG0 = w3 \cdot G0 \quad \cdots\cdots \quad (12)$$

$$= \frac{WGmax}{Gmax} \cdot G0 = WGmax \cdot \frac{G0}{Gmax} \quad \cdots\cdots \quad (13)$$

$$= WGmax \cdot \eta v \quad \cdots\cdots \quad (14)$$

$$\therefore \quad WGmax = WG0/\eta v = w3/\eta v \quad \cdots\cdots \quad (15)$$

Each loss of work is defined as follows

$$\left[\begin{array}{ll} \text{Motor loss} = W0-W1=(1-\eta_{mot})\cdot W0=\Delta W_{mot} & \text{----- (16)} \\ \text{Friction loss} = W1-W2=\Delta W_{mec} & \text{----- (17)} \\ \text{Compression loss} = W2-WG_{max}=\Delta W_c & \text{----- (18)} \\ \text{Leakage loss} = WG_{max}-WG0=\Delta W_L & \text{----- (19)} \end{array}\right.$$

Above definition is similar to that of shown in refference(*4)
From reference (*5) we assume that EER is defined as follows

$$EER=\frac{W1}{W0}\cdot\frac{W2}{W1}\cdot\frac{W3}{W2}\cdot\frac{G0\cdot\Delta ie}{W3} \quad \text{------- (20)}$$

$$=\eta_{mot}\cdot\eta_{mec}\cdot\eta_{comp}\cdot 3.73 \quad \text{------- (21)}$$

and then η_{comp} = W3/W2= Compression efficiency
In our definition $\eta_{comp}=\eta_{cid}\cdot\eta_v$ ------- (22)

In reference (*5), η_{comp} is defined without separating η_{cid} and η_v.
However we devided η_{comp} into two terms η_{cid} and η_v. And then ,in reference (*5), (total) compressor efficiency is defined as follows

$$\eta_t=\eta_{mot}\cdot\eta_{mec}\cdot\eta_{comp} \quad \text{------- (23)}$$

Both of two (*4,*5) did't explain cleary the relation between EER and above efficiencies. And equation (8) is sligtly different from eq.(26) of refference(*3).

TEST RESULTS

1. Factors which affect EER

We investigated the various variables (see note) affect the capacity Q, input W and EER by using multiple regulation analysis.(*6,*7) The tendency of the data of No.4 and No.12 are different from the others. We assumed that something were wrong and eliminated them from our evaluation. The data of No.18, No.19 and No.20 show no big difference between that of No.7. This is because the major dimensions of No.18,No.19 and No.20 are exactly the same with those of No.7 except for the shaft diameters. Therefor, we used 14 datas for the analysis.

Note : $\underline{H*D}$, $\underline{H*(D+2R_0)}$, $D, H, \varepsilon, D^2, D^3, H/D, (H/D)^2, D^2*H$
We found that the underlined variables are the most influential.

In oder to accurately measure the influence of major dimensions, motor efficiency was measured separately as a motor itself. As a result,

(1) The variable (H*D) and the force to pressurize gas (Fp) affect the EER and η_d much more than any other variables ($D, H, \varepsilon, D^2, D^3, H/D, (H/D)^2, (H/D)^3$ or D^2*H).
We obtained the following experimental equations values.

$$EER=1.208-1.953*H*D \quad \text{correlation factor } r=-0.74 \quad \text{---(24)}$$

$$\eta_d=1.07-0.00011*Fp \quad \text{" " } r=-0.85 \quad \text{---(25)}$$

(Fig.4 and Fig.5)

From the above equations, we find that the smaller the (H*D),the higher the EER.

Note : Correlation factor indicates how much it affects EER.
- r=±1 :It has perfect relation.
- ±0.8 :It has considerable relation.
- ±0.6 :It has some relation.
- ±0.5 :It may have some relation.

(2) The smaller the (H*D), the larger the capacity Q.
We obtained the following experimental equation.

$$Q = 1.10 - 0.0829 * H * (D + 2R0) \qquad r = -0.52 \quad --- \quad (26)$$

(Fig.6)

2. <u>Infuluence of clearance on EER.</u>

We conducted the test of prototype No.7 with 44 different clearances to investigate its influence on EER.(*10) The test results show that though the clearance is maintained within the tolerance of $\pm 5\,\mu$, this tolerance affects EER. By using the regulation analysis we obtain the following equation (27) and (28).

$$Q(Kcal/h) = B0 + B1(CR) + B2(CR)^2 + B3(CR)^3 + B4(CB) + B5(CB)^2 + B6(CB)^3 \quad --- \quad (27)$$

$$W(watts) = C0 + C1(CR) + C2(CR)^2 + C3(CR)^3 + C4(CB) + C5(CB)^2 + C6(CB)^3 \quad --- \quad (28)$$

where

Bn,Cn ;coefficient of regulation.
CR ;clearance between roller and cylinder height ceiling
CB ;clearance between vane and cylinder height ceiling

To evaluate the data on the same clearance , we compensated the EER of each compressor using these equations. In addtion, the all values of EER was divided by those of the prototype No.7 so that they can be evaluated on the dimensionless level. The test result of the relation of(H*D) and EER are shown in Fig.2 and the simulation results in Fig.3.

The tendency of the simulation results corresponds with that of the test results. However, there are some differences in absolute values. Therefore, it is not yet allowable to predict the performance only with simulation.

3. <u>Relation between volumetic efficiency and indicator mechanical efficiency.</u>

The larger the volumetic efficiency ηv, the larger the indicator mechanical efficiency ηd. We obtained the following experimental equation.

$$\eta' d = 0.5 + 0.48\,\eta v \qquad r = 0.55 \quad \text{-----------} \quad (29)$$

FUNDAMENTALS FOR DESIGNING COMPRESSOR WITH HIGHER EFFICIENCY

1) Under the condition of same (H*D), it is favorable to enlarge D, because the reliability improves due to the small increase of eccentricity ε and less blade friction.

However, the large D results in the large diameter of a compressor casing. Therefore, within the allowable limit of casing diameter, it is favorable to increase D. Fig.8 is a chart prepared for the compressor design. The chart is based on an experimental equation which gives the influence of (H*D) on EER.

2) In order to improve EER, it is necessary to improve either each efficiency (see note) or the product of these efficiencies. (Fig.9) (*8,*9)

Note: ηmot ; motor efficiency, ηmec ; mechanical efficiency
ηcid ; P-V indicator efficiency, ηv ; volumetic efficiency

In Fig.9, the compressor under study is fixed in a bolted shell and it has the same dimensions as the No7. And from the P-V diagram the indicator efficiencies ηcid and the mechanical efficienciy ηmec are calculated. At this time we use the motor efficiency ηmot which is gained separately in hot condition.

As rotational speed increase, ηmec decrease rapidly and ηv, ηcid , indicate a quadratic curve. Compressor mechanical efficiency; ηc is gained from power of all those three efficiencies. Fig.9 shows compressor mechanical efficiency; ηc has a maximum value at 60 Hz. If we wish to change the maximum point to another higher Hz , We must design the compressor which has another different each efficiencies. Of course, in the case of higher rotational speed , some especial efforts must be made. We should pay attension to valve,clearance , amount of oil, motor,so on.(*11,*12)

In oder to improve EER for an inverter controlled compressor, it is essential to design a comprssor to give the highest EER at the frequency most often used througout the whole year.

3) The smaller the (H*D) ,the higher the total efficiency. If (H*D) is maintained small, the force to pressurize gass (FP) becomes small and accordingly the journal loss becomes small.
Fig.10 shows the tendency of the other manufacturers' compressors. With the years, the compressor technology innovated. It shows that their tendency of the (H*D) is becoming smaller and the efficiency has increased. We assume that this is affected by the small (H*D).

4) As a result , we improve the efficiency of a 3/4 Hp size dual cylinder rotary compressor by reducing 20 % of H and achieved 2 % of EER increase. In addition to this, we added other improvement and increased EER. This compressor is now in production.

CONCLUSION

1. In oder to obtain high efficiency, it is essential to reduce the product of H and D.

2. However , small (H*D) results in large eccentricity ε and lowers the reliability. The allowable reliability limit needs to be investigated.

3. The total compressor efficiency ηt can be divided into motor efficiency ; ηmot, mechanical efficiency ; ηmec, indicator efficiency ; ηcid and volumetic efficiency; ηv.

   In oder to improve EER , it is necessary to either improve each efficiency or improve the product of these efficiencies.

REFERENCES

1. Ishii,N.,et al.,
"Optimum Combinasion of Dimensions for High Mecanical Efficiency of a Rolling-Piston Rotary Compressor,"
Proc.Purdue Int. Compressor Engineering Conference,
July.,1990 ,Vol.2,pp.418-424.

2. Yanagisawa ,T.,
"The behavior of roling piston and Mechanical Loss in Rotary Compressor,"
Trans.Japan Soc.Mech.Engs.(in Japanese),Series C,
Vol.48,No.435,1980,pp 1854-1861.

3. Prakash Pandeya,Werner Soedel
"A GENERALIZED APPROACH TOWARDS COMPRESSOR PERFORMANCE ANALYSIS"
Proc.Purdue Int. Compressor Engineering Conference,
July.,1978 ,pp.135-143.

4. Matszaka,T.,Nagatomo,S.,
"ROLLING PISTON TYPE ROTARY COMPRESSOR PERFORMANCE ANALYSIS."
Proc.Purdue Int. Compressor Engineering Conference,
July.,1982 ,pp.149-158. Toshiba Major Appliance Produce Eng.Labo.

5. KEIJU SAKAINO,et.al.,
"SOME APROACHES TOWARDS A HIGH EFFICIENT ROTARY COMPRESSOR,"
Proc.Purdue Int.Compressor Engineering Conference,
July.,1984 ,pp.315-322. Mitsubish Electric Corporation

6. MICRO SYSTEMS Co.Ltd."Elementaly Statistical Analysis Program,"1985

7. Okuno Tadakazu,et.al.,"Multivariate Data Analysis in a Technology,"
Printed in Japan by Nicka Giren,1986

8. Iida,S.,et al.,
"Efficiency Improvement in Rotary Compressor,"
Mitsbishi Hevy Ind.Technical Report,Vol. 20,No.3,1983-5, pp29-37

9 L.Sakitani,J.Koiwa,T.Maekawa,
"Performance Evaluation of Hermetic Refrigeration Compressor Using Torque Mesurement Method". Proc.Purdue Int.Compressor Engineering Conference,
Aug.,1986,VOL.1 ,pp.228-241. Daikin Industries.LTD.

10. Janghee Lee,Tae sik Min
"Performance analysis of rolling piston type rotary compressor"
Proc.Purdue Int. Compressor Engineering Conference,
July.,1988,Vol.1,pp.154-162. Daewoo Electronics Co.,Ltd.

11. Hiroshi Iwata,Akio Sakazume,and Kenziroh Yokoyama
"Loss Analysis of Rolling-Piston Type Speed Controlled Rotary Compressor"
Trans.Japan Soc.Mech.Engs.(in Japanese),Series B,
Vol.51,No.465,June,1985,pp 1736-1741. Hitachi,Ltd.

12. B.Sakurai,Dr.J.F.Hamilton
"THE PREDICTION OF FRICTIONAL LOSSES IN VARIABLE-SPEED ROTARY COMPRESSORS"
Proc.Purdue Int. Compressor Engineering Conference,
July.,1984,Vol.1,pp.331-338. Toshiba Corpration and Purdue University

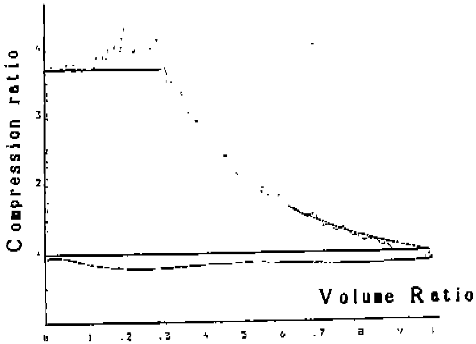

Fig. 1 P－V Diagram

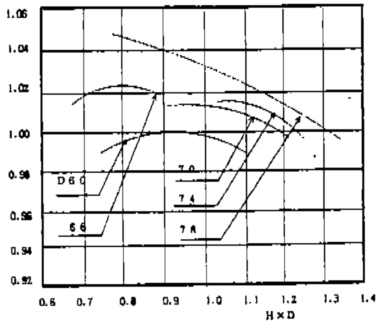

Fig. 3 Analytical Result

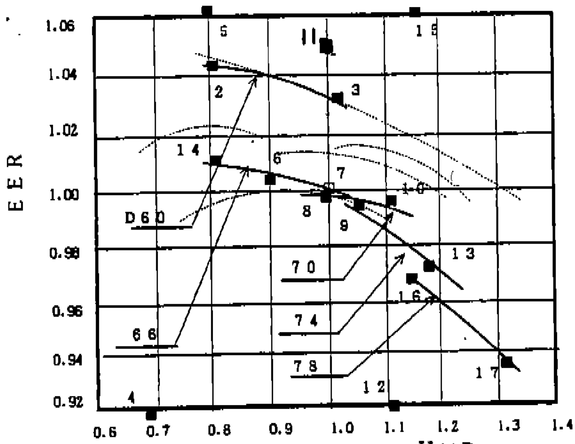

Fig. 2 TEST Result

TABLE 1. SPECIFICATION OF TEST COMPRESSOR
Main Front shaft dia Φ28, Crank pin dia Φ40

		1	2	3	4	5	6	7	8	9	10
	H / D	35	40	41.73	44.16	45	46.89	50	52.8	52.84	55.83
1	56							(☆1)			
2	60					☆2					☆3
3	66	☆4	☆5			☆6		★7			
4	70						☆8	☆9	☆10		
5	74				☆11			☆12		☆13	
6	78	☆14		☆15				☆16			☆17

☆18 Ria shaft dia Φ22, Crank pin dia Φ40
☆19 Ria shaft dia Φ20, Crank pin dia Φ40
☆20 Ria shaft dia Φ20, Crank pin dia Φ34

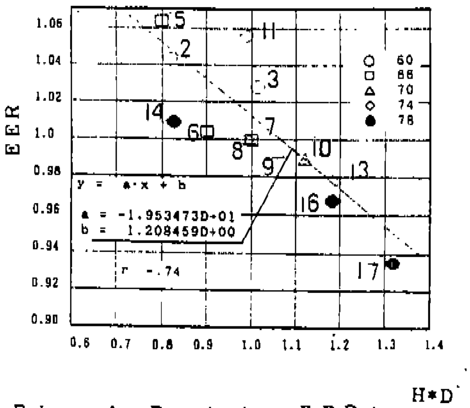

Fig. 4 Regulation EER vs. H*D

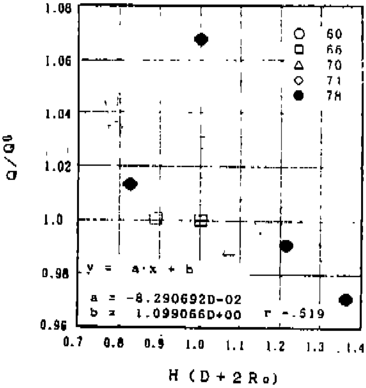

Fig. 6 Q vs. H (D+2Ro)

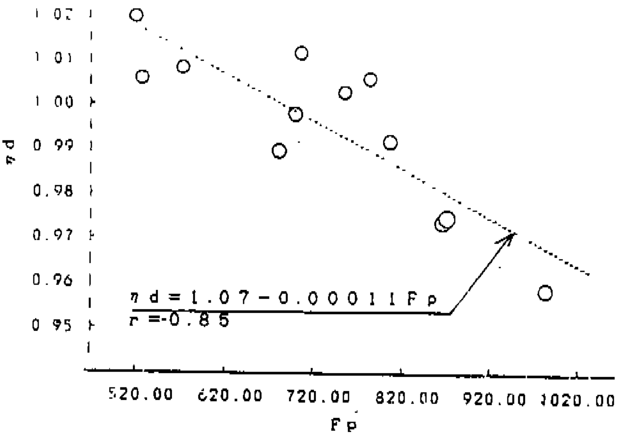

Fig. 5 Regulation ηd vs. Fp Kg

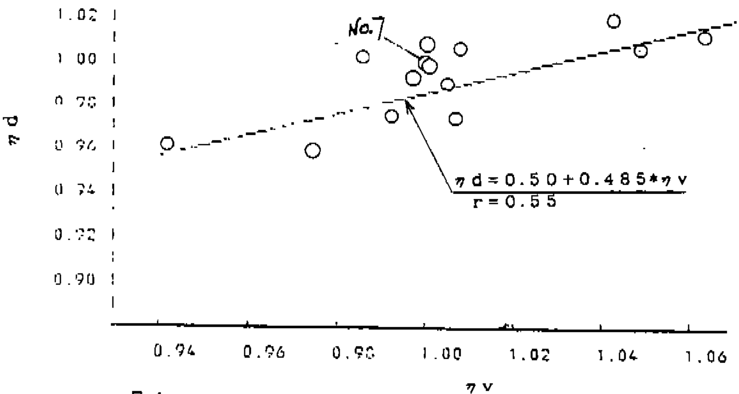

Fig. 7 ηd vs. ηv

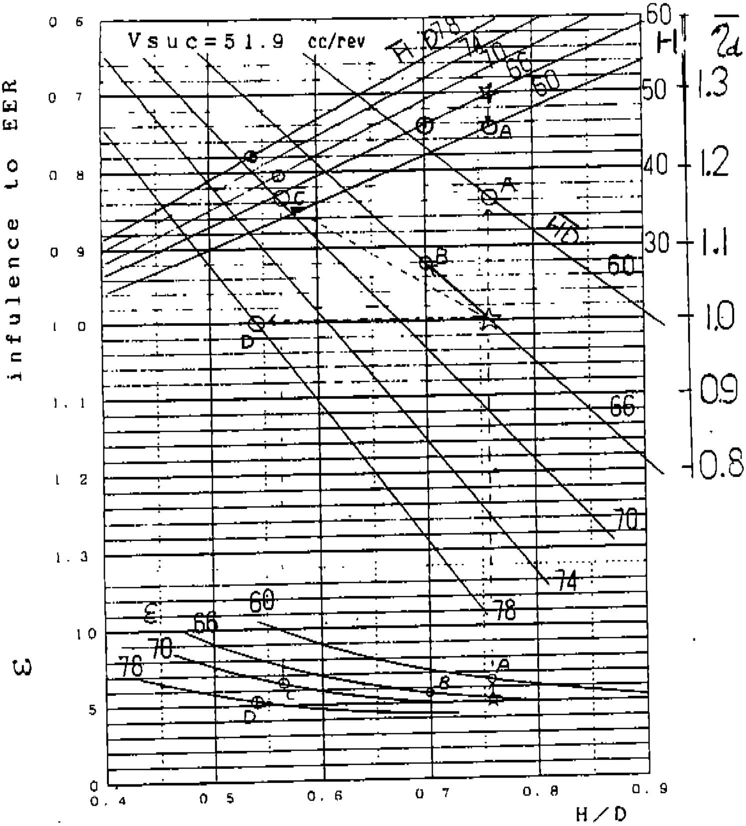

Fig. 8 Diagram for Design

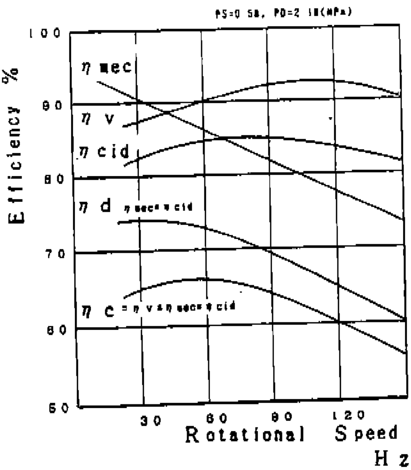

Fig. 9 Efficiency vs. Rotational Speed

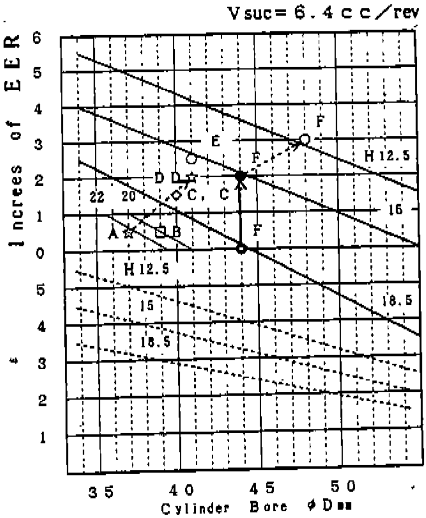

Fig. 10 Transition of factor