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DEVELOPMENT OF THE NON-LUBRICATED FOUR-STAGE COMPRESSOR COMPRESSING UP TO 24.52MPa

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ABSTRACT

The non-lubricated four-stage compressor which can compress up to 24.52 MPa has been developed. This newly developed compressor has high durability and efficiency in comparison with conventional non-lubricated compressors. For the purpose of improving durability, new technologies are adopted in this development. The plunger piston without piston rings is applied at the high pressure stages where the discharge pressure rises more than 4.90 MPa. And the new piston ring being composed of particular anti-wear materials, which are suitable for sliding movement in a non-lubricated condition, is applied at the other stages. Also the new grease, which has a good heat resistance and little scattering property, is selected for the sliding parts of crank mechanism. And then, the use of the original Labyrinth grooves on the plunger piston surface of the high pressure stages keeps a leakage of each stage at a minimum, and the use of the double acting mechanism at the first stage brings about the improvement of volumetric efficiency. Furthermore, the new Scotch Yoke mechanism with built-in springs makes low vibration and small-sized compressor. In this paper, we report the contents of the aforementioned developments in detail.

INTRODUCTION

Recently, a number of non-lubricated compressors have been developed. They are being used in many fields, such as food industry, petrochemical industry, medical industry and the other industries. They have been used for recovery, compressing, transfer of various gases, synthesis and charge of Hydrocarbons which are key materials for the industries. Because they do not have lubricant mist which makes carbon sludge owing to heavy oxidation at a high temperature and high pressure condition, as compared with compressors using a lubricating oil. The newly developed four-stage compressor is good for compressing Oxygen, inert gases and Hydrocarbons because of the above mentioned advantage. Also, as the compressor has realized to compress up to 24.52 MPa, it can be in use for Gas-assisted Injection Molding systems which requires high pressure. Further, the compressor is available for recovery system of Sulfur Hexafluoride, which is decided to reduce worldwide by Framework Convention on Climate Change-COP3 in Kyoto, Japan. The appearance of the compressor is shown in Figure 1.

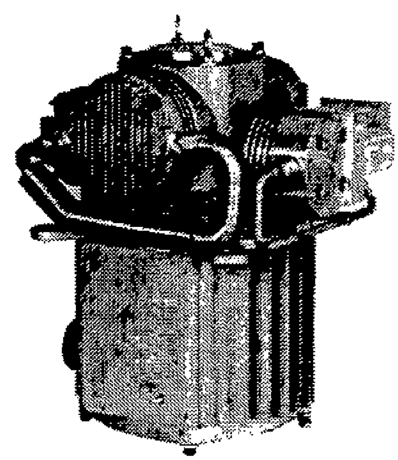

Figure 1. The four-stage compressor

THE IMPROVEMENT IN DURABILITY

The Compressing Mechanism of High Pressure Stages

Durability of non-lubricated compressors depends on anti-wear ability of the piston rings, and not only anti-wear ability but also sufficient strength of the ring is required at a high pressure condition. In case of the high pressure stages more than 4.90 MPa, many piston rings are needed in order to reduce the load per ring. Concretely the third and the fourth stage, where the discharge pressure rises to 4.90 MPa, need more than five piston rings on account of the load reduction, and the use of piston rings is not practical because of the complex structure. Therefore the plunger piston without piston rings is adopted at the third and fourth stage.

The seal mechanism with the plunger piston has advantages as mentioned below.

a. Since there is no piston ring in the contact area with gas, the discharge gas has not wear dust. So the high purity of discharge gas can be realized.

b. A sealing part does not wear, because the plunger piston does not make contact with cylinder during compressing gas.

c. Many kinds of gases can be compressed without any restriction.

Figure 2 shows schematic of the compressor with the plunger piston.

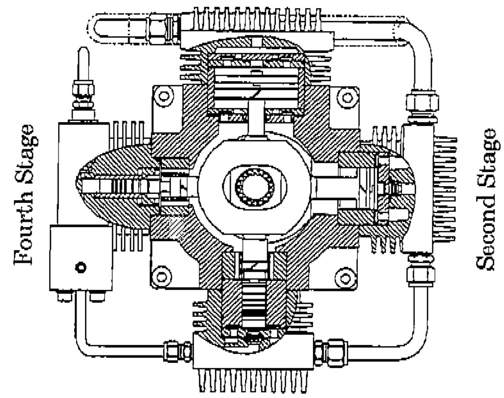

Figure 2. Schematic of the compressor

The New Piston Ring for Non-Lubricated Condition

The new piston ring had been developed to improve the durability of the compressor. Piston ring material needs high anti-wear ability and good sealing ability with the anodic oxide coating aluminum cylinder, which is used for the reason of its lightweight and good thermal conductivity. Kinds of ring materials are chosen from plastics such as polytetrafluoroethylene (PTFE), polyetheretherketone (PEEK) and polyimide (PI), and they are tested by Ring on Disk wear test. The combination of PTFE ring and the anodic oxide coating aluminum cylinder shows good performance in wear of the piston ring.

However, the PTFE ring material is not sufficient for our criteria. More investigation is done in a way of changing fillers added to base resin PTFE. New piston ring materials shown in Table 1 is tested by Ring on Disk wear test. PTFE C has the best anti-wear ability of them as shown in Figure 3

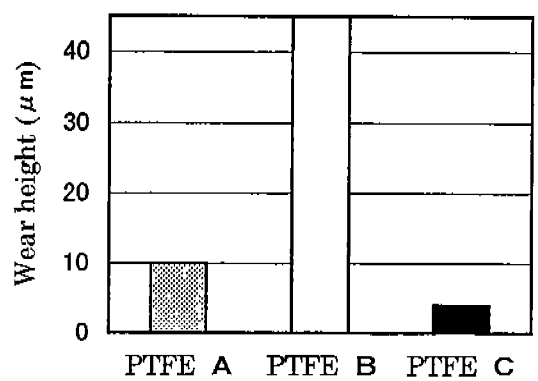

Figure 3. Wear test for PTFE piston ring material

Table 1. Tested PTFE ring materials

Material	Filler
PTFE A	Organic α
PTFE B	Organic β
PTFE C	Carbon, inorganic γ metallic δ

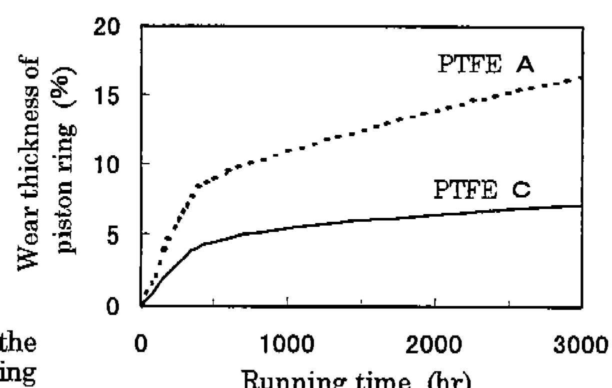

Figure 4. Result of durability test

Figure 4 shows the result of the durability test of the compressors using PTFE A or C as piston ring materials. PTFE C ring shows less wearing. In consequence, PTFE C has been adopted for the new piston ring material.

Development of the New Grease

The new grease used for sliding surface of the crank has been developed. Durability and less scattering are needed for the grease used. Because the grease which had been once investigated as a candidate showed degradation after long operation, the new grease has to be developed. Concretely, base oil and thickener of grease have been reevaluated. Kinds of grease listed in Table 2 are evaluated by Falex Pin-V block wear test.

Table 2. Evaluated grease and results of tests

Grease	Base oil	Thickener, Additive	Result of Falex wear test: Wear quantity	Appearance	Friction factor	Judgement	Durability test
A	Synthetic	Benton	—	—	+	NG	NG
B	Synthetic	Soap	—	—	+	NG	(No test)
C	Silicone	Unknown	—	—	—	NG	(No test)
D	Synthetic	Urea	+	+	+	OK	NG
E	Synthetic	Soap, EP	+	++	+	OK	Fine
F	Fluoride	PTFE	+	+	+	OK	NG

++ : excellent + : good — : poor

Grease D, E and F are selected from the result of Falex wear test. Furthermore as a result of the durability test, grease E which contains extra additives shows the best result in performance, such as appearance and wear of bearings. Consequently, the reliability of sliding part has been achieved by the development of the new grease.

THE ACCOMPLISHMENT OF HIGH EFFICIENCY

Optimization of the Labyrinth Seal

Labyrinth seals of contact free seal mechanism are generally used in reciprocating and rotary mechanism to prevent leakage of high-pressure fluid to low pressure. For high efficiency, it is necessary for Labyrinth seals to have small clearance and small pitch grooves. In such cases, it sometimes causes sudden pressure-rise or instability vibration of plunger piston. Therefore optimization of them is important in various conditions. In this development, the optimization of Labyrinth grooves on the plunger piston surface of the high-pressure stages is put in practice using Computational Fluid Dynamics(CFD). Basic equations of compressible fluid are equation of continuity and motion, shown as follows.

$$\frac{1}{\sqrt{g}}\frac{\partial}{\partial t}\left(\sqrt{g}\,\rho k\right)+\frac{\partial}{\partial x_j}\left(\rho u_j k-\frac{\mu_{eff}}{\sigma_k}\frac{\partial k}{\partial x_j}\right)=u_t(P+P_B)-\rho\varepsilon-\frac{2}{3}\left(u_t\frac{\partial u_i}{\partial x_i}+\rho k\right)\frac{\partial u_i}{\partial x_i}$$

$$\frac{1}{\sqrt{g}}\frac{\partial}{\partial t}\left(\sqrt{g}\,\rho\varepsilon\right)+\frac{\partial}{\partial x_j}\left(\rho u_j\varepsilon-\frac{\mu_{eff}}{\sigma_k}\frac{\partial\varepsilon}{\partial x_j}\right)=C_{\varepsilon1}\frac{\varepsilon}{k}\left[(P+C_{\varepsilon3}P_B)-\frac{2}{3}\left(\mu_t\frac{\partial u_i}{\partial x_i}+\rho k\right)\frac{\partial u_i}{\partial x_i}\right]-C_{\varepsilon2}\rho\frac{\varepsilon^2}{k}-C_{\varepsilon4}\rho\varepsilon\frac{\partial u_i}{\partial x_i}$$

Where

t : time

u_i : absolute fluid velocity compornent in direction x_i

u_j : u_j-u_{cj}, relative velocity between fluid and local coodinate frame that moves with velocity u_{cj}

$\sqrt{g}$: determinate of metric tensor

μ_{eff} : $\mu+\mu_t$, μ_t is the turbulent viscosity

s_i : momentum sourceco mprnents

ρ : dencity

P : $2s_{ij}\frac{\partial u_i}{\partial x_j}$

P_B : $-\frac{g_i}{\sigma_{h,t}}\frac{1}{\rho}\frac{\partial\rho}{\partial x_i}$

τ : stress tensor component

σ_k : empirical coefficient

$C_{\varepsilon i}$: empirical coefficient

And then it is calculated by considering eddy model making use of equations k-ε. Discretization of derivative equations is calculated using Finite Volume Method, and algorithm of analysis has been adopted SIMPLE method that is solved by implicit scheme.

Specification of the Labyrinth seal, which is calculated by CFD, are shown in Table 3. Generally, straight-through type is adopted for its workability and reliability. Boundary conditions are shown in Table 4. Numerical models have been made from each crank angle, which is divided one rotation into four parts. And then it changes pressure ratio and velocity of piston.

Table 3. Specification of Calculation models

Unit : mm

	No Groove	TYPE 1	TYPE 2	TYPE 3
Groove Width	—	0.3	1.0	1.0
Groove Depth	—	0.3	0.3	0.3
Groove Pitch	—	4.0	4.0	Irregular Pitch

Table 4. Boundary Conditions

Operating Fluid	Nitrogen
Density (Kg/m³)	41.44~187.1
Coefficient of Viscosity (Pa·S)	$22.4 \sim 26.3 \times 10^{-6}$
Specific heat (J/Kg·K)	1083~1196
Thermal Conductivity (W/m·K)	0.0341~0.0423
Pressure	
High (MPa)	7.55~19.71
Low (MPa)	0.98
Temperature	
High (K)	383
Low (K)	293
Velocity of Piston (m/s)	0~1.51

The calculation model is shown in Figure 5. This calculation model is set up pressure condition of INLET and OUTLET. In addition, the plunger piston surface has been given periodic velocity for wall function.

In this analysis, fluid force which cases instability vibration in reciprocating mechanism is only calculated by shearing stress of circumferential.

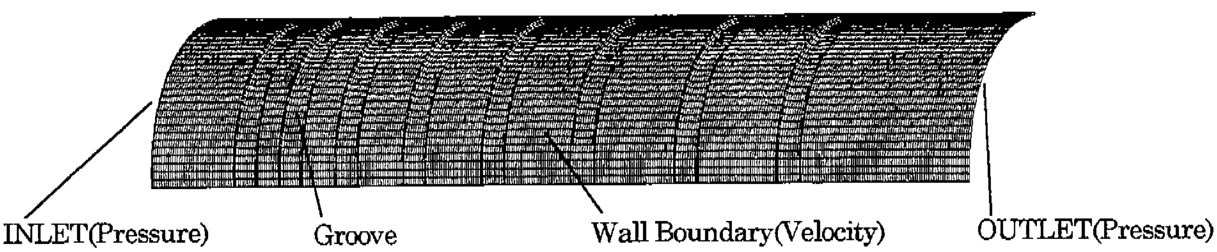

Figure 5. Calculation model

Figure 6 shows flow pattern of internal Labyrinth groove. In the Labyrinth groove, instability eddy generates at turbulent region. And the more main flow is prevented and channel resistance increases, the more seal affection is improved. Figure 7 shows the relation between specification of Labyrinth groove and velocity fluctuation. The horizontal axis indicates the distance from INLET to OUTLET. Consequently, All with grooves is disposed to make the velocity slow as compared with type of no groove. Type of with groove, setting 0.2 which is ratio of depth against width, has made a good result. In relation to groove pitch, the seal of irregular pitch is better than equational pitch. Then, the shearing stress which affects on each wall is at most 2×10^{-6} N/m² and is the level allowable.

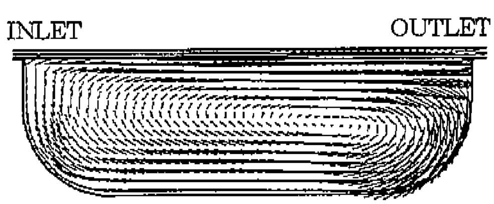

Figure 6. Flow Pattern in Labyrinth Groove

Leakage test with each type shown in Figure 8 is put on operation, and the result is shown in Figure 9. The result shows the leakage of type 3 is less than that of current type 1 by 9%. Also Figure 10 shows the gas flow rate of type 3 is more than that of type 1 by 10%, and its sealing ability is improved.

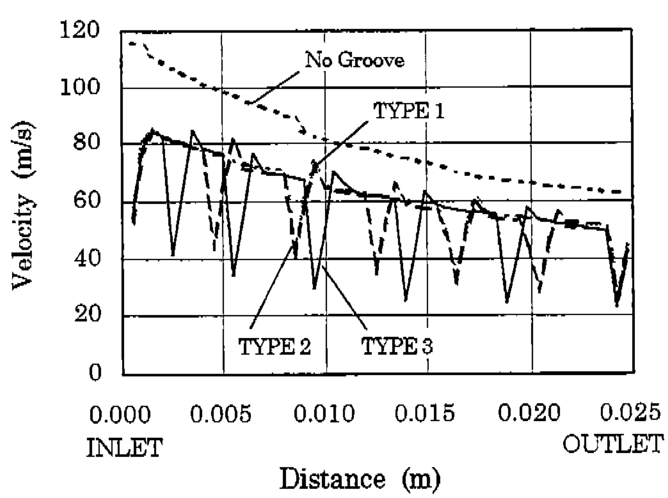

Figure 7. Velocity Fluctuation from INLET to OUTLE

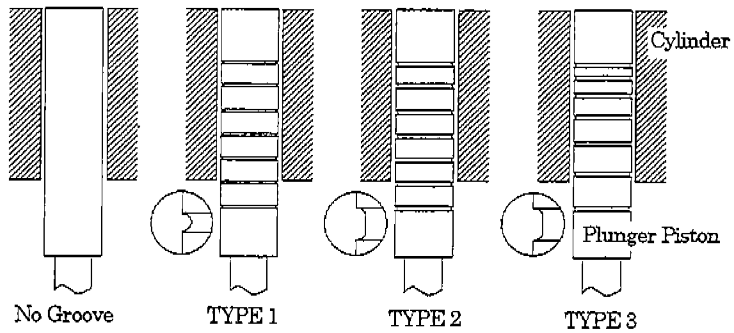

Figure 8. Groove Pattern

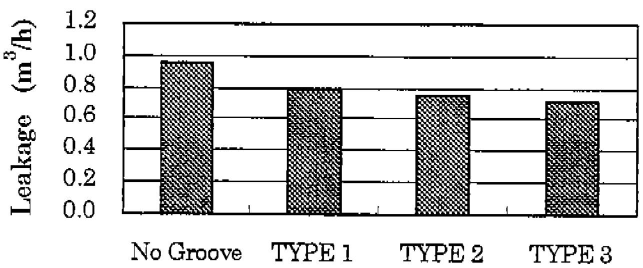

Figure 9. Result of Leakage Test

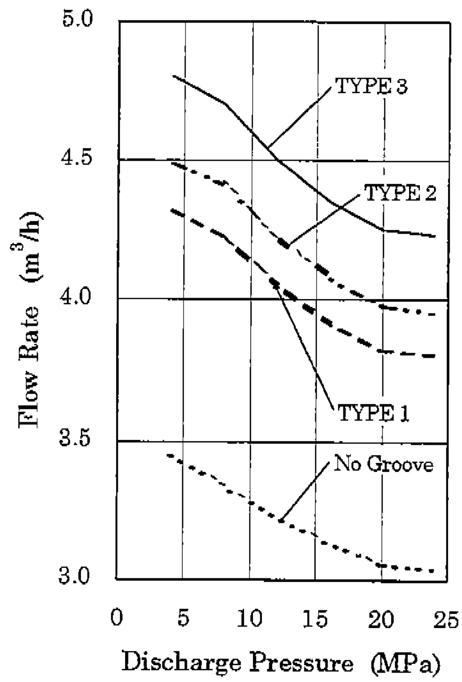

Figure 10. Result of Gas Flow Rate Test

The Double Acting Mechanism at the First Stage

In addition to optimization of the labyrinth seal, the double acting mechanism, which is another means for high efficiency, has been adopted at the first stage. The use of the double acting is effective for increasing gas flow rate without expanding cylinder diameter at the first stage. Theoretically, the double acting mechanism makes the flow rate up to 112%, compared with no double acting. The state of post-stage and pre-stage is given by the following equations.

<post-stage> $$P_{po} = P_d (V_{c1} / V_{po})^{\kappa} \qquad (1)$$
$$V_{po} = V_{c1} + V_o (1 - \sin\theta) / 2 \qquad (2)$$
<pre-stage> $$P_{pr} = P_s \left[(V_{c2} + V_o) / V_{pr} \right]^{\kappa} \qquad (3)$$
$$V_{pr} = V_{c2} + V_o (1 + \sin\theta) / 2 \qquad (4)$$

where

P_d : discharge pressure of post-stage
P_{po} : internal pressure of post-stage
V_{po} : internal volume of post-stage
V_{c1} : clearance volume of post-stage
V_o : piston displacement
P_s : suction pressure of pre-stage
P_{pr} : internal pressure of pre-stage
V_{pr} : internal volume of pre-stage
V_{c2} : clearance volume of pre-stage
κ : specific heat ratio

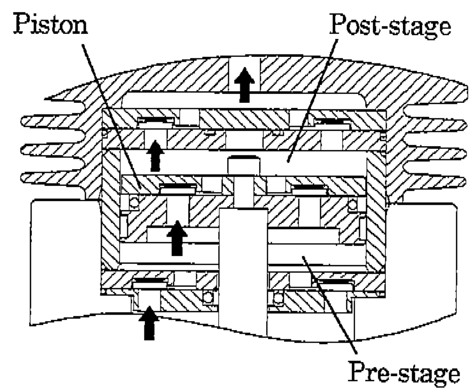

Figure 11. Structure of the first stage

When the double acting is adopted, the suction gas is taken in the post-stage if $P_{po} = P_{pr}$. Therefore, the pressure P_{po} is the suction pressure of the first stage. Quantity of intake Q_1 with double acting and Q_2 without double acting are obtained from equations (5) and (6).

$$Q_1 = V_o (P_{po} / P_{at}) \qquad (5)$$
$$Q_2 = V_o (P_s / P_{at}) \qquad (6)$$

where

P_{at} : atmospheric pressure

From above equations, the quantity ratio of intake (Q_1/Q_2) becomes 1.12, and the flow rate would increase by 12%. Figure 11 shows structure of the first stage and Table 5 shows the result of the operating test.

Table 5. Result of the operating test

Structure	No Double Acting	Double Acting
Flow Rate (m^3/h)	4.3	4.8

THE NEW SCOTCH YOKE MECHANISM

In order to make small-sized compressor, the new Scotch Yoke mechanism has been adopted in the crank and each cylinder has been arranged crosswise. A Scotch Yoke mechanism generally has a yoke and cross-slider made of steels. In this development, the yoke is made of Aluminum alloy to make it light. As a result of that, the lightweight Yoke brings optimum dynamic balancing, and low vibration is accomplished. Schematic of the crank is shown in Figure 12.

Each piston is fixed to the yoke oppositely, and the load moves alternately by the change pressure in the cylinder. The yoke reciprocates on right and left with the revolution of crank pin, when the cross-slider slides up and down. The cage with needle rollers is set between the yoke and the cross-slider in order to reduce frictional resistance. Since the thermal expansion coefficient of the Yoke is different from that of the cross-slider, a gap increases as the temperature rises in the operation. And the cross-slider does not slide properly by the abnormal behavior of the cage. To solve this problem, the newly developed Scotch Yoke mechanism with built-in springs, which gives the cage pre-load at all times, has been adopted.

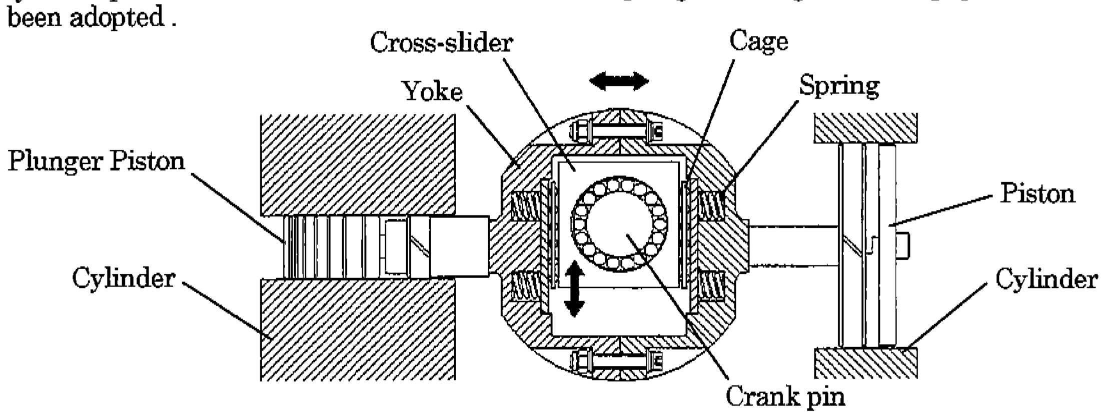

Figure 12. Schematic of the crank

CONCLUSION

The non-lubricated four-stage compressor has been developed, which is available for compressing Oxygen, inert gases, Hydrocarbons and Sulfur Hexafluoride etc. The compressor, which can compress up to 24.52MPa, would have the great future because of being possible to use for various purposes. The results of this development are summarized as follows.

1. The durability of the compressor has been much improved by the use of the plunger piston at the high pressure stages more than 4.90 MPa and the development of the new PTFE piston ring at the other stages.
2. The new grease with a good heat resistance and little scattering property, which contains extra additives, has been developed for the crank.
3. The high efficiency has been realized by the optimization of the Labyrinth grooves existing on the plunger piston surface and the use of the double acting mechanism at the first stage.
4. The cross-arrangement of cylinders and the developed Scotch Yoke mechanism with built-in springs have brought low vibration, lightweight and small-size to the compressor.

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