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N. Ishii
*Osaka Electro-Communication University*

S. Kawamura
*Osaka Electro-Communication University*

S. Yamamoto
*Matsushita Electric Ind. Co.*

K. Sawai
*Matsushita Electric Ind. Co.*

A. Hiwata
*Matsushita Electric Ind. Co.*

***See next page for additional authors***

**Authors**
N. Ishii, S. Kawamura, S. Yamamoto, K. Sawai, A. Hiwata, T. Nakomoto, H. Kawano, and T. K. Seng

# EFFICIENCY SIMULATIONS WITH CONSIDERATION OF HEAT LOSSES OF A R410A COMPACT SCROLL COMPRESSOR FOR ITS OPTIMAL PERFORMANCE

**Noriaki Ishii Susumu Kawamura**
Osaka Electro-Communication University, Neyagawa, Osaka 572, Japan
**Shuichi Yamamoto, Kiyoshi Sawai, Akira Hiwata and Tatsuya Nakamoto,**
Air Conditioning Research Laboratory, Matsushita Electric Ind. Co., Ltd. (Panasonic)
2-3-1-1, Nojihigashi, Kusatsu-City, Shiga 525-8520 , Japan.
**Hiroyuki Kawano**
Air Conditioning Devices Division, Matsushita Electric Ind. Co., Ltd. (Panasonic)
5-1-5,Sakuragaoka, Kusatsu-City, Shiga 525-8502, Japan.
**Ting Kian Seng**
Matsushita Compressor and Motor R&D Centre Sdn. Bhd., Lot 17, Jalan Puchong,
Taman Perindustrian Subang Utama, Seksyen 22, 40300 Shah Alam, Selangor, Malaysia

## ABSTRACT

The paper presents calculations for the resultant efficiency of a compact R410A scroll compressor for its optimal performance, where the effect of the heat transfer from the high temperature circumstances outside the compression mechanism into the suction and compression chambers upon the mechanical, volumetric and compression efficiencies was significantly addressed. The heat transfer in the scroll compressor has less effect upon the mechanical efficiency, while the heat transfer during suction process has some effect upon the volumetric efficiency, and the heat transfer during compression process has a small effect upon the compression efficiency, thus resulting in the resultant efficiency decrease due to heat transfer losses, by 2.3% at the cylinder volume of 11.4cc and by 5.1% at 2.5cc. However, optimum combination of major dimensions is not affected by the heat transfer losses.

## 1. INTRODUCTION

The suction volume of involute-type scroll compressors is determined on the basis of the major dimensions, such as the involute base circle radius, the scroll height, the scroll thickness and the cylinder diameter. It becomes clear that there are many combinations of the major dimensions that yield a scroll compressor with the same suction volume. It must be quite significant to examine which combination is the best in performance for the compressors. The frictional losses and the refrigerant gas leakage change, depending upon the combination of the major dimensions, thus significantly affecting upon the mechanical, volumetric and compression efficiencies. Based on such a concept, the computer simulations have been made to address the optimum combination of the major dimensions, yielding the highest performance in mechanical, volumetric and compression efficiencies, by Ishii *et al.* [1-5]. However, the effect of the heat losses from the high temperature circumstances outside the compression mechanism into the suction and compression chambers, upon the compressor efficiencies, has not been taken into considerations.

Ishii *et al.* [6,7] have first analyzed such an effect of the heat losses upon the compressor efficiencies of a compact rolling-piston type rotary compressor. In the present study, similar analysis was applied to compact R410A scroll compressors with the suction volume of 11.4cc and 2.5cc, where the scroll thickness was fixed at 3.0mm and the cylinder diameter at 67.54mm. The present paper assumes the flow pattern of the refrigerant gas in the compressed chamber to be a well-developed turbulent flow. Thereby the heat transfer through the whole walls surrounding the suction and compression chambers was calculated by the forced turbulent convection heat transfer theory for a flat plate, which is based on the Colburn's analogy. The present study introduces the basic theory to calculate the heat transfer coefficient, the heat transfer rate and the compressed pressure, first. Secondly the calculated results for the mechanical, volumetric and compression efficiencies, are presented, to address the effect of the heat transfer losses upon the resultant efficiency.

## 2. HEAT TRANSFER IN SCROLL COMPRESSORS

Given the involute base circle radius $r_b$, the scroll thickness t and the cylinder diameter D, the area of the crescent-shaped suction chamber, $S_0$, is given by the following expression:

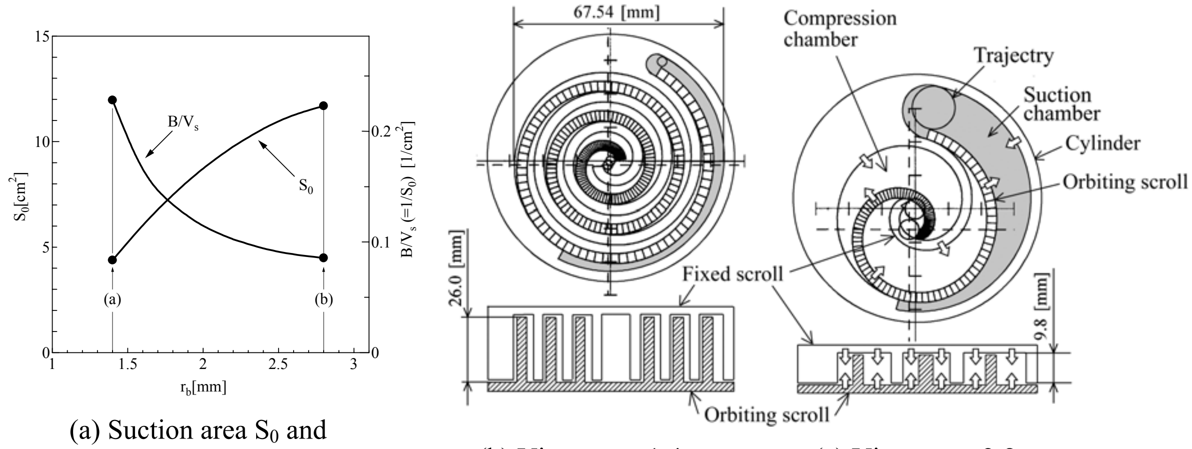
$$S_0 = 4\pi r_b^2 \left\{ \pi\left(\varphi_{oe} - \frac{3}{2}\pi\right) - \frac{t}{r_b}\left(\varphi_{oe} - 2\pi\right) - \frac{1}{2}\left(\frac{t}{r_b}\right)^2 \right\} \quad , \tag{1}$$

where the ending involute angle of the outer scrolls, $\phi_{oe}$, and the orbiting radius $r_0$ are given by

$$\phi_{oc} = \sqrt{\left(\frac{D - r_o}{2r_b}\right)^2 - 1} - \frac{t}{r_b} \quad , \qquad r_0 = \pi\, r_b - t \quad . \tag{2}$$

The suction area $S_0$ at D=67.54mm and t=3.0mm is shown in Fig.1a. It is quite natural that the larger the suction area $S_0$, the smaller the scroll height B for a given value of the suction volume $V_s$, as shown by the reduced scroll height $B/V_s$ in Fig.1a. Two representative examples at suction volume of 11.4cc are shown in Figs.1b at $r_b$=1.4mm and Fig.1c at $r_b$=2.8mm, where the suction area $S_0$ drastically increases, while the scroll height B drastically decreases.

As shown by white arrows in Fig.1c, the heat transfer into the suction and compression chambers occurs from the heated walls surrounding the chambers, that is, from the cylinder, the fixed and orbiting scrolls. If the heated wall temperature is given, the heat transfer into the suction and compression chambers can be calculated.




(a) Suction area $S_0$ and scroll height B.

(b) View at $r_b$=1.4mm.

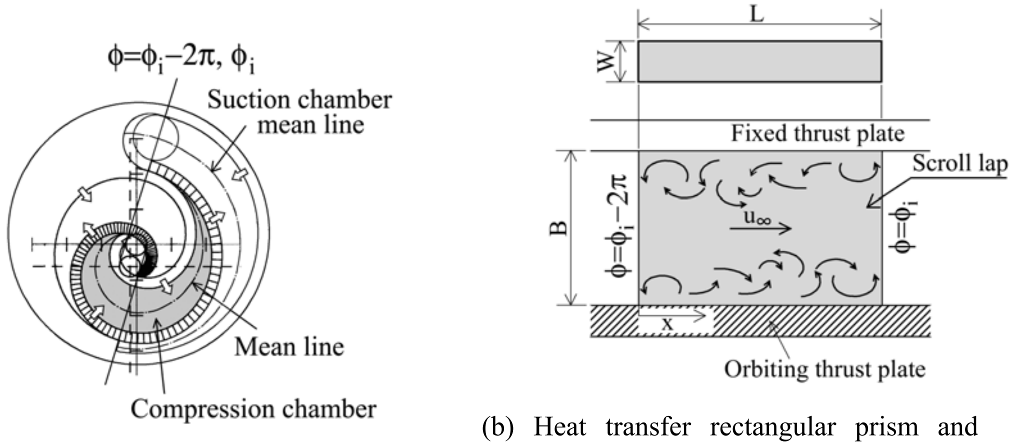
(c) View at $r_b$=2.8mm.

Fig.1 Scroll configurations and heat transfer.

## 3. BASIC THEORY

**Heat Transfer Coefficient and Heat Transfer Rate**

The orbiting forces the refrigerant gas in the suction and compression chambers to move relative to the fixed scroll and the cylinder. Since the refrigerant gas is sucked into the suction chamber and then compressed at a relatively high speed, one may assume that the heat transfer from the heated walls into the refrigerant gas can be treated as the forced convection heat transfer from the heated flat plate, as shown in Fig.2. Such an assumption is one unavoidable choice, since few studies have been presented regarding the flow patterns in the suction and compression chambers. The crescent-shaped compression chamber, shaded in Fig.2a, is extended to a rectangular prism, as shown in Fig.2b, where L by B is the equivalent scroll lap and L by W is the equivalent thrust plate. L represents the length of the mean line of crescent-shaped area, shown by chain line in Fig.2a, and W represents its mean width. From these flat plate surfaces, the heat transfers into the compressed gas. The mean velocity of compressed volume movement is represented by $u_\infty$ .




(a) Suction and compression chamber. (b) Heat transfer rectangular prism and well-developed turbulent flow pattern.

Fig.2 Equivalent heat transfer model of scroll compressor.

The heat transfer coefficient can be derived from the Colburn's analogy:

$$N_u = \frac{C_f}{2} R_e P_r^{\frac{1}{3}} , \tag{3}$$

which prescribes the relation between the skin frictional coefficient for the velocity boundary layer, $C_f$, and the Nusselt number for the thermal boundary layer, $N_u$. The Reynolds number is represented by $R_e$ and the Prandtl number is by $P_r$. $N_u$, $R_e$ and $P_r$ are respectively defined by

$$N_u \equiv \frac{\alpha x}{\lambda}, \quad R_e \equiv \frac{u_\infty x}{\nu}, \quad P_r \equiv \frac{\mu C_p}{\lambda}, \tag{4}$$

where $\alpha$ is the local heat transfer coefficient, $\lambda$ is the thermal conductivity, $\nu$ is the kinematic viscosity, $\mu$ is the viscosity coefficient and $c_p$ is the specific heat at constant pressure. As shown in Fig.2b, the distance from the left-hand side end of the scroll lap is represented by x.

The Nusselt number represents the heat quantity ratio of the heat transfer to the heat conduction. The Prandtl number represents the ratio of the kinematic viscosity to the thermal diffusivity defined by $\lambda/\rho_{cp}$ ($\rho$: specific mass), which takes nearly on a constant value for gas, independent of temperature and pressure. It has been well studied that the Colburn's analogy is valid for the fluid of the Prandtl number from 0.5 to 50, even if the fluid is under a well-developed turbulent flow.

If the refrigerant gas flow on the thrust plates is so well-developed turbulent that the Reynolds number is from $5\times10^5$ to $10^7$, the skin friction coefficient $C_f$ is empirically given by

$$C_f = 0.0592 R_e^{-\frac{1}{5}} . \qquad \text{(see Johnson \& Rubesin [8])} \tag{5}$$

Substituting this expression into the Colburn's analogy, the Nusselt number is given by

$$N_u = 0.0296 R_e^{\frac{4}{5}} P_r^{\frac{1}{3}} \quad \text{or} \quad \alpha = 0.0296\lambda\left(\frac{u_\infty}{\nu}\right)^{\frac{4}{5}} P_r^{\frac{1}{3}} x^{-\frac{1}{5}} . \tag{6}$$

The average heat transfer coefficient over the thrust plate with the length of L , $\overline{\alpha}$ , is derived as follows:

$$\overline{\alpha} = \frac{1}{L}\int_0^L \alpha dx = 0.037\frac{\lambda}{L} P_r^{\frac{1}{3}} R_{eL}^{\frac{4}{5}} , \tag{7}$$

where the mean Reynolds number $R_{eL}$ is defined by

$$R_{eL} \equiv \frac{u_\infty L}{\nu}. \tag{8}$$

The heat transfer from the heated solid surface into the fluid is essentially subject to the Newton's law of cooling. Representing the thrust plate temperature by $T_w$ and the compressed gas temperature by T, the heat transfer rate Q is given by

$$Q = \overline{\alpha}(T_w - T)S , \tag{9}$$

where S represents the heat transfer area.

**Compressed Gas Pressure**

The heat quantity which transfers into the compressed gas during a small time of dt, is given by Qdt. Therefore, the first law of thermodynamics gives the following equation:

$$Qdt = Gc_v dT + pdV , \tag{10}$$

where G is the gas mass, $c_v$ is the constant-volume specific heat, dT is the temperature increase, P is the pressure and dV is the volume increase. The right-hand side first term of (9) represents the internal energy increase. The equation of state is given by

$$PV = GRT , \tag{11}$$

from which the following relation can be derived.

$$GdT = \frac{1}{R}\left(VdP + PdV\right) - TdG , \tag{12}$$

where R is the gas constant.

Substituting (11) into (9), the pressure increase dP is given by

$$dP = \frac{1}{V}\left\{(\kappa - 1)Qdt - \kappa pdV + RTdG\right\}, \tag{13}$$

where $\kappa$ represents the specific heat ratio. From (11), the temperature increase dT is given by

$$dT = \frac{1}{GR}\left(VdP + PdV\right) - \frac{T}{G}dG . \tag{14}$$

Therefore, pressure P(t+dt) and temperature T(t+dt) at time (t+dt) can be calculated from those at time t, as follows:

$$P(t + dt) = P(t) + dP \quad ; \quad T(t + dt) = T(t) + dT . \tag{15}$$

## 4. SPECIFICATIONS OF SCROLL COMPRESSOR FOR NUMERICAL CALCULATIONS

Table 1 gives the specifications of the R410A scroll compressors with the suction volume of 11.4cc and 2.5cc, and the compression volume ratio of 2.8. When the suction volume is 11.4cc, the cooling capacity of 3043kcal/h is achieved at the rated conditions of the average crankshaft speed 3498rpm, the suction temperature 10.5°C, the suction pressure 0.81MPa and the discharge pressure 2.46Mpa. The scroll thickness and the cylinder diameter were fundamentally kept at 3.0mm and 67.54mm, respectively. The clearance between the orbiting and fixed scroll was kept at 3μm for the axial and 6μm for the radial.

Table 2 gives the mechanical constants of the compressors. The crankshaft moment of inertia, $I_0$, was adjusted depending upon the necessary driving shaft power, and the orbiting scroll mass $m_o$ was upon the scroll height. The frictional coefficients at each pair of the moving compressor elements were kept at from 0.055 to 0.0013, measured by friction tests.

As listed up in Table 1, calculations were made for the involute base circle radius $r_b$ from 1.4 mm to 2.8mm. Therewith, the scroll height B changes from 26.0mm to 9.8mm, as shown in Figs.1b and 1c.

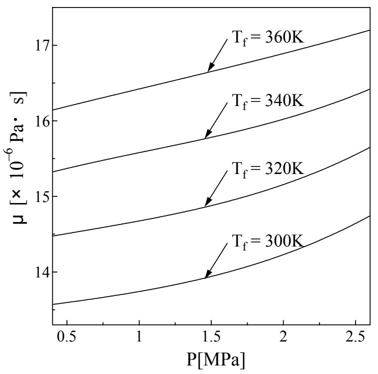
Regarding the heat transfer, the viscosity coefficient μ, the kinematic viscosity ν, the heat conductivity λ, the constant-pressure

Table1 Major specifications of scroll compressors

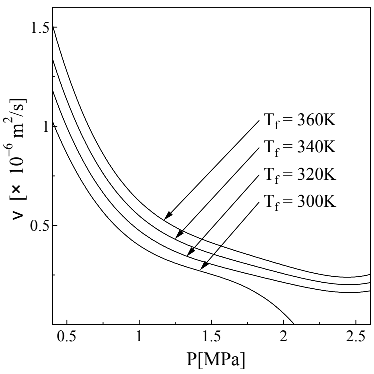
| | | | |
|---|---|---|---|
| Suction volume | Vs[cc] | 11.4 | 2.5 |
| Cooling capacity | [kcal/h] | 3043 | |
| Operationg speed | [rpm] | 3498 | |
| Involute base circle radius | $r_b$ [mm] | 1.4～2.8 | |
| Scroll height | B [mm] | 26.0～9.8 | 5.7～2.1 |
| Scroll thickness | t [mm] | 3.0 | |
| Cylinder disameter | D [mm] | 67.54 | |
| Volume ratio | | 2.80 | |
| Pressure ratio | | 3.04 | |
| Specific heat ratio | κ | 1.08 | |
| Suction temperature | $T_s$ [°C] | 10.5 | |
| Wall temp. discharge | $T_w$ [°C] | 95 | |
| Wall temp. suction | $T_w$ [°C] | 80 | |
| Suction pressure | $P_s$ [MPa] | 0.81 | |
| Discharge pressure | $P_d$ [MPa] | 2.46 | |
| Axial clearance | $\delta_a$ [μm] | 3.0 | |
| Radial clearance | $\delta_r$ [μm] | 6.0 | |

Table2 Major mechanical constants of scroll compressors

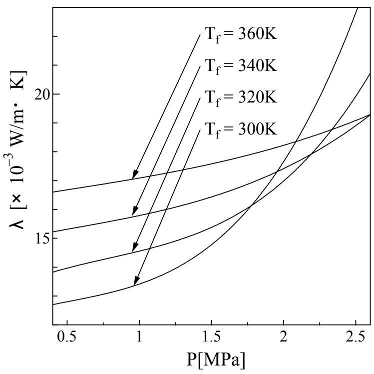
| | | | |
|---|---|---|---|
| Suction volume | Vs[cc] | 11.4 | 2.5 |
| Moment of Inertia of rrankshaft | $I_0$ [kg.m$^2$] | 0.139～0.147 | 0.106～0.108 |
| Orbiting scroll mass | ms [kg] | 0.116～0.177 | 0.106～0.112 |
| Oldham ring mass | $m_0$ [kg] | 0.037 | |
| Crankshaft radius | $r_Q$ [mm] | 8.0 | |
| Crankpin radius | $r_S$ [mm] | 8.0 | |
| Fric. coef. at oldham ring | | 0.055 | |
| Fric. coef. at thrust bearing | | 0.055 | |
| Fric. coef. at crankpin | | 0.011 | |
| Fric. coef. at crank journal | | 0.011 | |
| Fric. coef. at ball bearing | | 0.0013 | |



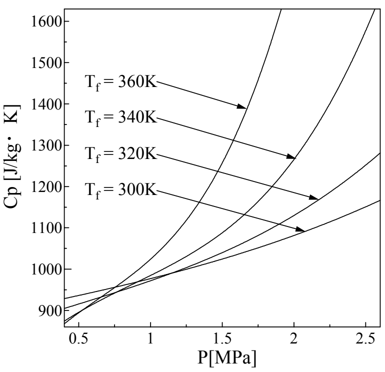

(a) Viscosity coefficient μ.



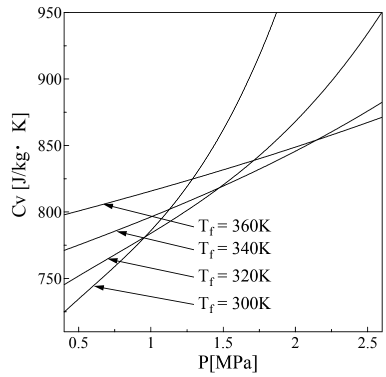

(b) Kinematic viscosity ν.




(c) Heat conductivity λ.




(d) Constant-pressure specific heat $C_p$.




(e) Constant-volume specific heat $C_v$.

Fig.3 Specifications for heat transfer.

and constant-volume specific heats $c_p$ and $c_v$, are shown in Fig.3, which vary depending upon the pressure P and the temperature of the thermal film near the heated walls, $T_f$, which is given by the average of the heated wall temperature $T_w$ and the compressed gas temperature T. The thrust plate temperature $T_w$ was carefully measured to address the wall temperature which linearly increases from 80$^o$C at the suction port to 95$^o$C at the discharge port, as listed in Table 1.

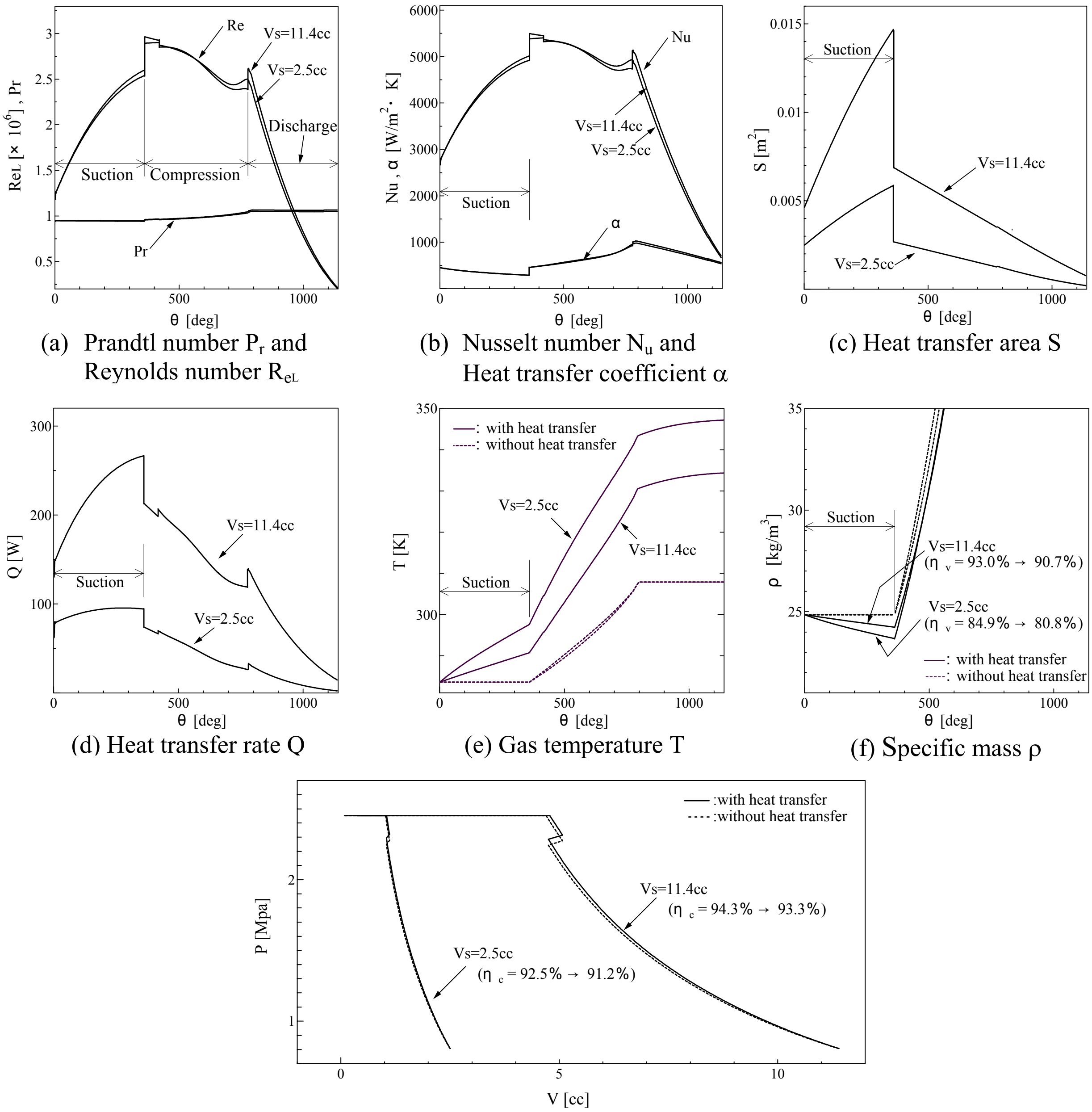
## CALCULATED RESULTS

A representative example, calculated for the involute radius of 1.8mm and the scroll height of 15.2mm at 11.4cc and of 3.3mm at 2.5cc, is shown in Fig.4, where major factors related to the heat transfer, resulting specific mass and pressure of the refrigerant gas are presented. The abscissa is the orbiting angle θ for suction, compression and discharge processes, as indicated in Fig.4a.

The Prandtl number $P_r$ is from 0.9 to 1.0 and the Reynolds number $R_{eL}$ is sufficiently larger than $5\times10^5$, except for near the end of compression. Therefore, the Colburn's analogy given by (3) and the skin friction coefficient $C_f$ given by (5) are well valid. Two different cylinder volume compressors has the same scroll configuration, thus resulting in less difference in $P_r$ and $R_e$, and also in the Nusselt number $N_u$ and the average heat transfer coefficient $\alpha$, given by (6), as shown in Fig.4b.

Of quite significance is in the suction process. Though the heat transfer coefficient $\alpha$ is comparatively small, the heat transfer area S rapidly increases, as shown in Fig.4c. Thereby, as shown in Fig.4d, the heat transfer rate Q becomes larger, thus increasing the gas temperature T, as shown in Fig.4e, and thus lowering the specific mass of the gas, ρ, as shown in Fig.4f, where the dotted line represents those without heat transfer. As a result, the volumetric efficiency $\eta_v$ decreases by 2.3% (93.0% to 90.7%) at 11.4cc, and seriously by 4.1% (84.9% to 80.8%) at 2.5cc, due to the heat losses.

On the other hand, due to the heat transfer during the compression process, the compressed gas pressure P slightly increases than the pressure without heat transfer, as shown in Fig.4g, where the P-V diagram is




(a) Prandtl number $P_r$ and Reynolds number $R_{eL}$

(b) Nusselt number $N_u$ and Heat transfer coefficient $\alpha$

(c) Heat transfer area S

(d) Heat transfer rate Q

(e) Gas temperature T

(f) Specific mass $\rho$

(g) Gas Pressure P

Fig.4 Temperature and pressure calculated for the involute base circle radius $r_b$ of 1.8mm and the scroll height B of 15.2mm at 11.4cc and of 3.3mm at 2.5cc.

presented. As a result, the compression efficiency $\eta_c$ decreases by 1.0% (94.3% to 93.3) at 11.4cc, and by 1.3% (92.5% to 91.2%) at 2.5cc.

Similar calculations were made for a number of combinations of the involute radius and the scroll height, to address the effect of the heat transfer losses upon the mechanical, volumetric and compression efficiencies. The equation of motion of the crankshaft was numerically calculated first, to find the crankshaft speed fluctuation ratio $\alpha$ and Mechanical efficiency $\eta_m$, as shown in Fig.5, where the abscissa is the involute radius $r_b$. The heat transfer losses have no effect upon the crankshaft rotation and the mechanical efficiency although it was a natural result.

The leakage flow through the axial and radial clearances between the orbiting and fixed scrolls can be calculated with adequate accuracy, by assuming an incompressible and viscous turbulent flow [4]. Calculated results of the leakage mass through the axial and radial clearances to the suction chamber, $\Delta G_a$ and $\Delta G_r$, and its total $\Delta G$ during compression, are presented in Figs.6a, 6b and 6c, respectively. They do not exhibit a

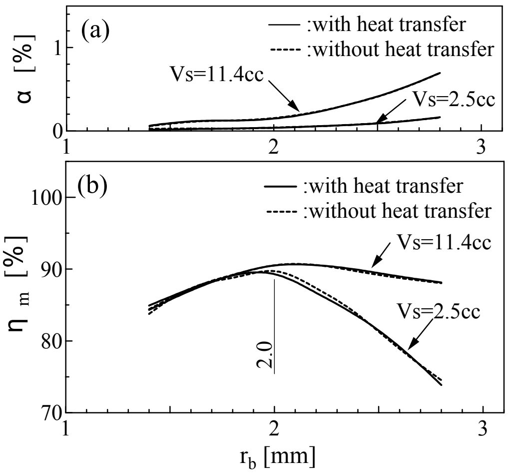


Fig.5 Crankshaft speed fluctuation ratio α and Mechanical efficiency $\eta_m$.

meaningful difference from those without heat transfer, represented by the dotted line. Nevertheless, the big difference has arisen in the volumetric efficiency $\eta_v$, as shown in Fig.6d. This was naturally caused by the heat transfer losses during the suction process, as explained by the specific mass ρin Fig.4f. A volumetric efficiency decrease, due to the heat transfer losses, was 2.34% at the cylinder volume of 11.4cc, and its fall became larger with increasing the cylinder volume, for instance, up to 4.01% at 2.5cc.

The compression efficiency $\eta_c$ is presented in Fig.7. A fall of efficiency due to the heat transfer losses during compression was unexpectedly small, 1.06% at 11.4cc and 1.28% at 2.5cc.

As a result, calculations gave the resultant efficiency of mechanical, volumetric and compression ones, η, as shown in Fig.8. The decrease in resultant efficiency, due to heat transfer losses, is 2.3% at 11.4cc and 5.1% at 2.5cc, each at optimum involute radius. The optimum value of the resultant efficiency was 78.9% at 11.4cc and 66.8% at 2.5cc. The optimum volume of the involute radius $r_b$ is 2.3mm at 11.4cc and 2.0mm at 2.5cc, which are not affected by the heat transfer losses.

## CONCLUSIONS

The crank-shaft speed fluctuation ratio and the resultant efficiency of the compact R410A scroll compressors with the suction volume of 11.4cc and 2.5cc, the scroll thickness of 3.0mm and the cylinder diameter of 67.54mm, were numerically calculated at an average rotating speed of 3498rpm, where the heat transfer from the heated walls of the suction and compression chambers into the refrigerant gas was taken into consideration.

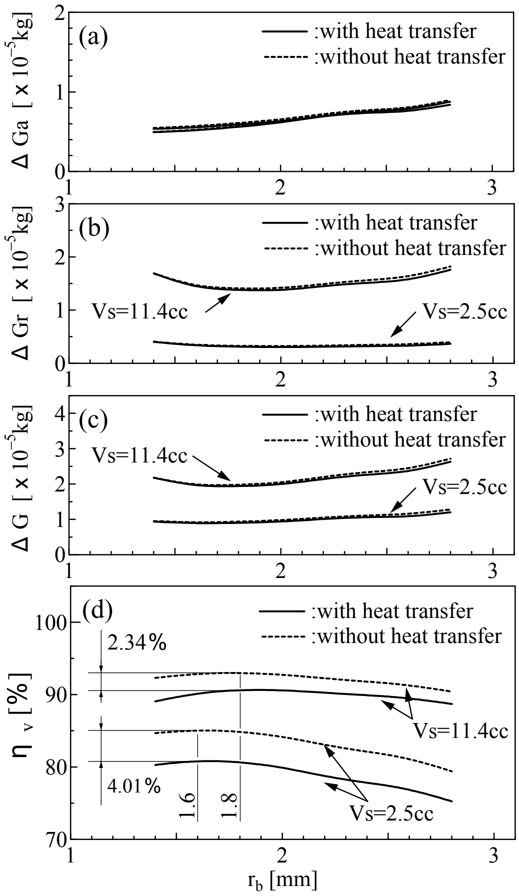


Fig.6 Refrigerant gas leakages through axial and radial clearances and volumetric efficiency $\eta_v$.

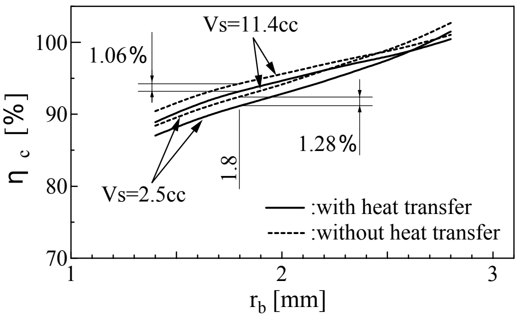


Fig.7 Compression efficiency $\eta_c$.

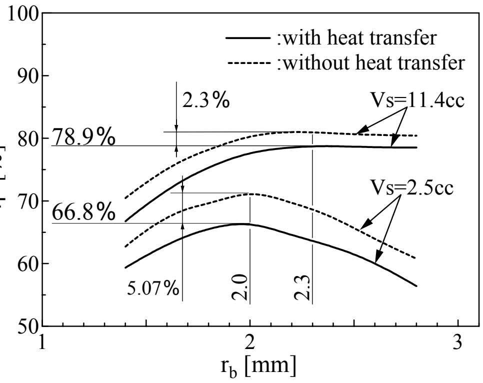


Fig.8 Resultant efficiency η.

The heat transfer into the suction gas decreases the specific mass of the suction gas, thus lowering the volumetric efficiency by about 2% at 11.4cc and by about 4% at 2.5cc, while the heat transfer into the compressed gas increases its pressure, thus lowering the compression efficiency by about 1.0% at both the suction volumes. The mechanical efficiency and the crankshaft rotation with low fluctuation are not affected by the heat transfer losses. As a result, the resultant efficiency takes it maximum value of 78.9% at 11.4cc, smaller by 2.3% than without heat transfer, and 66.8% at 2.5cc, smaller by 5.07% than without heat transfer. The optimum combination of the involute radius and the scroll height is not affected by the heat transfer losses.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## REFERENCES


(1) Ishii, N., Yamamoto, S., Muramatsu, S., Yamamura, M. and Takahashi, M.: Optimum Combination of Parameters for High Mechanical Efficiency of a Scroll Compressor, Proceedings of 1992 International Compressor Engineering Conference at Purdue, West Lafayette, Indiana, USA, pp. 118a1-118a8, 1992-7.

(2) Ishii, N., Yamamura, M., Muramatsu, S., Yamada, S. and Takahashi, M.: A Study on High Mechanical Efficiency of a Scroll Compressor with Fixed Cylinder Diameter, Proceedings of 1994 International Compressor Engineering Conference at Purdue, West Lafayette, Indiana, USA, Vol. 2, pp. 677-682, 1994-7.

(3) Ishii, N., Sakai, M., Sano, K., Yamamoto S. and Otokura T.: A Fundamental Optimum Design for High Mechanical and Volume Efficiency of Compact Scroll Compressors, Proceedings of 1996 International Compressor Engineering Conference at Purdue, West Lafayette, Indiana, USA, Vol.II, pp.639-644, 1996-7.

(4) Ishii, N., Bird, K., Sano K., Oono, M., Iwamura S. and Otokura T.: Refrigerant Leakage Flow Evaluation for Scroll Compressors, Proceedings of 1996 International Compressor Engineering Conference at Purdue, West Lafayette, Indiana, USA, Vol. II, pp.633-638, 1996-7.

(5) Ishii, N., Yamamoto, S., Sano, K., Sawai, K., Hiwata, A., Nakamoto, T. and Kawano, H.: Efficiency Simulations of a Compact $CO_2$ Scroll Compressor and its Comparison with same cooling capacity R410A Scroll Compressor, Proceedings of 2002 International Compressor Engineering Conference at Purdue, 2002-7.

(6) Ishii, N., Morita, N., Kurimoto, M., Yamamoto, S. and Sano, K.: Calculations for Compression Efficiency Caused by Heat Transfer in Compact Rotary Compressors, Proceedings of 2000 International Compressor Engineering Conference at Purdue, West Lafayette, Indiana, USA, Vol. I, pp.467-474, 2000-7.

(7) Ishii, N., Morita, N., Ono, M., Yamamoto, S. and Sano, K.: Net Efficiency Simulations of Compact Rotary Compressors for Its Optimal Performance, Proceedings of 2000 International Compressor Engineering Conference at Purdue, West Lafayette, Indiana, USA, Vol. I, pp.475-482, 2000-7.

(8) Johnson, H.A & Rubesin, M.W., Trans. ASME, 71, 1948, p.499.