Analysis of the roto-cooler air-conditioning system

Yuan Mao Huang

Department of Mechanical Engineering, National Taiwan University, Taiwan,
Republic of China

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The roto-cooler rotary blade air cycle air-conditioning system is simpler and less expensive than conventional vapour compression air-conditioning systems. This study analyses the effects of the unsteady flow on the heat exchanger and the performance of the roto-cooler system. The air properties, effects of the heat exchanger length on the air properties, the cooling capacity, the thermal coefficient of performance and the total coefficient of performance of the system are studied. Numerical results of the specific operating conditions are compared with the available data. The heat exchanger length of the prototype can be reduced significantly without loss of system performance.

(Keywords: roto-cooler; air-conditioning system; elliptical stator; compressor)

Analyse du système de conditionnement d'air 'roto-cooler'

Le système de conditionnement d'air à cycle à air et à compresseur rotatif est plus simple et moins cher que les systèmes classiques à compression de vapeur. Dans ce rapport, on analyse les effets d'un écoulement invariable sur l'échangeur de chaleur et sur la performance du système. On étudie les propriétés de l'air, les effets sur celles-ci des dimensions de l'échangeur de chaleur, ainsi que la puissance de refroidissement, les coefficients de performance thermique et global. On compare les résultats numériques des conditions précises de fonctionnement avec les données disponibles. Les dimensions de l'échangeur peut être sensiblement diminuée sans perte de performance du système.

(Mots clés: conditionnement d'air; machine à air; compresseur rotatif; détendeur; conception; performance)

Although current vapour refrigeration air-conditioning systems have been well developed, they are complex, heavy and expensive. High-pressure lines and fittings are required. Compression and road vibration can cause leakage of the refrigerant through connections. The refrigerant can also be contaminated by moisture. The temperature of the evaporator should not be allowed to drop below 0°C, to avoid the water vapour freezing on it. The roto-cooler air-conditioning system is a simpler, less expensive and environmentally protective device, which may challenge current systems. The end view of the roto-cooler system is shown in Figure 1¹⁻³.

Edwards¹ used the assumptions of steady and incompressible flow to predict the effects of the geometries and operating conditions on the cooling capacity and coefficient of the performance of the roto-cooler system. In addition, Edwards failed to include an adequate analysis of the effectiveness of the heat exchanger, which was built conservatively. The purpose of this study is to include the effect of unsteady and compressible flow on the heat exchanger and the performance of the system, and to determine the required length of the heat exchanger.

Method of approach

As the roto-cooler configuration is complex, some assumptions used in the analysis are as follows:

1. The air is a perfect gas under the operating conditions.
2. The air in the compression and expansion sides of the rotor follows the polytropic process.
3. The properties of the air inside the blade segments are uniform.
4. Fluid dynamic effects at the blade tip and the blade slot interface are neglected.
5. No mass of air is accumulated at the inlet or outlet portions of the heat exchanger.
6. The cross-sectional area of the heat exchanger is constant.
7. The temperature and velocity of the air surrounding the heat exchanger remain constant.
8. The conductivity and the external heat transfer film coefficient of the heat exchanger are constant.
9. The heat exchanger is a straight duct except the inlet and outlet portions.
10. The change of potential energy is negligible.
11. Radiation from the heat exchanger is neglected.
12. The flow inside the heat exchanger is fully developed.

Nomenclature

A	Cross-sectional area of heat exchanger (m^2)	V	Blade segment volume (m^3)
a	Sonic speed (m s^{-1})	W	Work (N m)
c_p	Specific heat at constant pressure ($\text{J kg}^{-1} \text{K}^{-1}$)	W_s	All work except flowing work (N m)
c_v	Specific heat at constant volume ($\text{J kg}^{-1} \text{K}^{-1}$)	x	Distance from heat exchanger inlet (m)
D	Inside hydraulic diameter of heat exchanger (m)	z	Number of blades
e	Stored energy per unit mass (J kg^{-1})	<i>Greek letters</i>	
f	Friction factor	β	Coefficient of performance
h	Enthalpy per unit mass (J kg^{-1})	ρ	Density (kg m^{-3})
h_p	Potential energy per unit mass (J kg^{-1})	ω	Rotational speed (rad s^{-1})
i	Internal energy per unit mass (J kg^{-1})	<i>Subscripts</i>	
M	Mass (kg)	1, 2, 3, 4, 5	Condition in the blade segment volume
n	Polytropic process exponent constant	a	Ambient condition at the roto-cooler inlet
P	Power (J s^{-1})	c	Condition in blade segment or compression
p	Pressure (N m^{-2})	e	Condition at heat exchanger outlet or expansion
Q	Heat added (J)	ea	Condition surrounding heat exchanger
Q_c	Cooling capacity (J s^{-1})	f	Friction
q	Heat transfer rate per unit area ($\text{J s}^{-1} \text{m}^{-2}$)	io	Initial condition at heat exchanger inlet
Re	Reynolds number	th	Thermodynamic quantity
T	Temperature (K)	tt	Total quantity
t	Time (s)	<i>Superscripts</i>	
U	Overall heat transfer coefficient ($\text{J s}^{-1} \text{m}^{-2} \text{K}^{-1}$)	$\circ$	Time rate
u	Velocity (m s^{-1})

Figure 1 The roto-cooler air-conditioning system

Figure 1. Le système de conditionnement d'air rotatif

13. The specific heats are constant under the operating conditions.

14. The effect of water vapour condensation during the expansion process is neglected.

For counterclockwise rotation of the rotor, air at essentially atmospheric pressure and temperature is drawn from the inlet duct into the blade segment V_i as it expands, as shown in Figure 1. When the rear blade of segment V_i arrives at point A, the maximum blade segment volume V_1 is obtained, and the intake process is completed. The air inside the blade segment is assumed to be compressed polytropically with polytropic exponent n_{c1} to higher pressure and temperature until the leading blade of the blade segment V_2 arrives at point B. As long as the values of n_{c1} , the rotational speed and the

number of blades are given, the air properties inside the blade segment at any location which is a function of time can be determined. The work of compression for the polytropic process from V_1 to V_2 is⁴

$$W_{c1} = p_a V_1 \frac{1}{1 - n_{c1}} \left[\left(\frac{V_2}{V_1} \right)^{1 - n_{c1}} - 1 \right] \quad (1)$$

Most of the air is then pumped from the blade segment into the heat exchanger, where the air is cooled down by rejecting heat to the surrounding air. The work required is

$$W_{c2} = \int_{V_2}^{V_3} p \, dV \quad (2)$$

Therefore the total compression work is

$$W_c = W_{c1} + W_{c2} \quad (3)$$

Taking the compression blade segment as the control volume, the rate of the energy equation for the compression chamber is

$$\dot{Q} - \dot{W} + (h\dot{M})_{in} - (h\dot{M})_{out} + \left[\frac{1}{2} \dot{M} u^2 \right]_{in} - \left[\frac{1}{2} \dot{M} u^2 \right]_{out} + [h_p \dot{M}]_{in} - [h_p \dot{M}]_{out} = \frac{d}{dt} [M_c (i_c + \frac{1}{2} u_c^2 + h_p)] \quad (4)$$

As the volume of the blade segment varies while the air discharges from the blade segment into the heat

exchanger, the air properties flowing through the heat exchanger vary periodically. The flow is unsteady and is assumed to be one-dimensional. The continuity equation is

$$\frac{\partial \rho}{\partial t} + u \frac{\partial \rho}{\partial x} + \rho \frac{\partial u}{\partial x} = 0 \quad (5)$$

If the body force is neglected, the momentum equation can be written as

$$u^2 \frac{\partial \rho}{\partial x} + 2\rho u \frac{\partial u}{\partial x} + \frac{\partial p}{\partial x} + \frac{f \rho u^2}{2D} + u \frac{\partial \rho}{\partial t} + \rho \frac{\partial u}{\partial t} = 0 \quad (6)$$

where the friction factor, f , is a function of the surface condition of the heat exchanger and the Reynolds number of the flow⁵. Because the value of the friction factor is uncertain, an empirical formula is assumed⁶:

$$f = 0.316 Re^{-0.25} \quad (7)$$

As there is no shaft work, the energy equation is^{7,8}

$$\dot{Q} - \frac{\partial}{\partial x} \rho A u \left(e + \frac{p}{\rho} \right) dx = \frac{\partial}{\partial t} (\rho A e) dx \quad (8)$$

The overall heat transfer rate, $\dot{Q}$, is given by

$$\dot{Q} = \int q \pi D dx \quad (9)$$

where

$$q = U(T_{ca} - T) \quad (10)$$

and U is the overall heat transfer coefficient, which is assumed to be constant for the temperature range of interest⁹.

Non-dimensional variables are defined as follows:

$$\begin{aligned} \rho^* &= \frac{\rho}{\rho_a} \\ T^* &= \frac{(T - T_a)}{(T_{io} - T_a)} \\ t^* &= \frac{ta_a}{D} \\ u^* &= \frac{u}{a_a} \\ x^* &= \frac{x}{D} \end{aligned} \quad (11)$$

The method of characteristics is used with non-dimensional characteristics as

$$\lambda^* = \frac{dx^*}{dt^*} \quad (12)$$

The non-dimensional characteristic directions determined from Equations (5), (6) and (8) are

$$\lambda^* = u^* + a^* \quad (13)$$

$$\lambda^* = u^* - a^* \quad (14)$$

and

$$\lambda^* = u^* \quad (15)$$

The compatibility equations along the characteristic directions (Equations (13), (14) and (15) respectively) are

$$d\rho^* + A_1 du^* + A_2 dT^* + A_3 dt^* = 0 \quad (16)$$

$$d\rho^* + B_1 du^* + B_2 dT^* + B_3 dt^* = 0 \quad (17)$$

$$d\rho^* + C_2 dT^* + C_3 dt^* = 0 \quad (18)$$

where $A_1, A_2, A_3, B_1, B_2, B_3, C_2$ and C_3 are functions of the air properties and configuration of the system. These equations, which are non-linear ordinary differential equations, can be solved by the finite difference method. The air properties inside the heat exchanger at any location and any time and the heat out of the heat exchanger can then be determined¹⁰⁻¹⁴.

In the meantime, the mass left in the clearance volume V_1 which is bypassing the heat exchanger mixes with the cooled air from the outlet of the heat exchanger. The charging of the expansion blade segment is completed when the rear blade of the blade segment V_4 just passes the outlet port of the heat exchanger, C, as shown in Figure 1.

Taking the expansion blade segment as the control volume, the mass flowrate into the expansion chamber is

$$\dot{M} = u_c \rho_c A \quad (19)$$

Assuming that there is no heat transfer during the charging process, the energy equation during the intake process of the expansion blade segment is

$$-p_c dV_c + c_p T_c dM_c + \frac{u_c^2 dM_c}{2} = d \left(c_p T_c M_c + \frac{M_c u_c^2}{2} \right) \quad (20)$$

The expansion work when the air flows polytropically into the expansion blade segment from the outlet of the heat exchanger is

$$W_{e1} = \int_{V_1}^{V_4} p dV \quad (21)$$

The relatively cool high-pressure air inside the segment V_4 expands and lowers its temperature and pressure when the rotor rotates. A polytropic expansion with polytropic exponent n_{e3} occurs until the leading blade of the blade segment V_5 arrives at the roto-cooler outlet D, as shown in Figure 1. This expansion work is

$$W_{e2} = \frac{p_4 V_4}{1 - n_{e3}} \left[\left(\frac{V_4}{V_5} \right)^{n_{e3} - 1} - 1 \right] \quad (22)$$

The position of the final outlet of the roto-cooler is chosen to give the air pressure as close as possible to atmospheric pressure. The air is then pumped through

Table 1 Air properties obtained from the current analysis and from ref. 1
Tableau 1 Propriétés de l'air déterminées par la présente analyse et tirées de la référence 1

Location	Temperature (°C)		Pressure (kPa)	
	Current	Ref. 1	Current	Ref. 1
Inlet of roto-cooler system	27.2	27.2	101.3	101.3
Complete compression	113.7	113.9	303.4	300.0
Compression blade segment before air flows completely into heat exchanger	113.7–100.8	—	303.4–266.1	—
Heat exchanger inlet	110.8–100.6	—	294.4–266.1	—
Heat exchanger outlet	32.7	32.2–33.3	259.2	—
Expansion blade segment before complete intake process	113.3–36.6	—	259.2	—

Table 2 Performance of the roto-cooler system obtained from the current analysis and from ref. 1
Tableau 2 Performance du système 'roto-cooler', déterminée par la présente analyse et tirée de la référence 1

	Current	Ref. 1
Compression power input (kJ h ⁻¹)	25 120	—
Expansion power output (kJ h ⁻¹)	19 614	—
Thermal power (kJ h ⁻¹)	5506	—
Friction power (kJ h ⁻¹)	9392	9284
Total power (kJ h ⁻¹)	15 004	—
Cooling capacity (kJ h ⁻¹)	5971	5686
Heat transfer rate out of the compression side of the stator (kJ h ⁻¹)	19 191	—
Heat transfer rate into the expansion side of the stator (kJ h ⁻¹)	8249	—
Net heat transfer rate out of the stator (kJ h ⁻¹)	10 942	—
Heat transfer rate out of the heat exchanger (kJ h ⁻¹)	9568	—
Thermal coefficient of performance	1.09	—
Total coefficient of performance	0.40	0.42–0.45

Figure 2 The variation range of the air temperature inside the heat exchanger
Figure 2 La gamme de températures de l'air à l'intérieur de l'échangeur de chaleur

the outlet duct into the space being cooled. Thus the work from the air to the blade is close to zero and it may be neglected. Therefore, the total expansion work is

$$W_e = W_{e1} + W_{e2} \tag{23}$$

The thermodynamic work per blade segment required to operate the roto-cooler system is

$$W_{th} = W_c - W_e \tag{24}$$

Figure 3 The variation range of the air pressure inside the heat exchanger
Figure 3 La gamme de pressions de l'air à l'intérieur de l'échangeur de chaleur

The thermodynamic power requirement, P_{th} , is

$$P_{th} = W_{th} \frac{z\omega}{2\pi} \tag{25}$$

where z is the number of blades and ω is the rotational speed. The cooling capacity of the system is defined as

$$\dot{Q}_c = \dot{M}c_p(T_1 - T_5) \tag{26}$$

where T_1 and T_5 are the inlet temperature and the final outlet temperature of the system respectively. The thermal coefficient of performance is defined as

$$\beta_{th} = \frac{\dot{Q}_c}{P_{th}} \tag{27}$$

If the friction power of the roto-cooler system, P_f , is included, the total coefficient of performance can be determined:

$$\beta_{tt} = \frac{\dot{Q}_c}{P_{th} + P_f} \tag{28}$$

Results

The geometry and operating conditions used are as follows: stator width 0.1524 m; length of stator major axis and stator minor axis 0.1524 m and 0.1168 m respectively; blade thickness 0.0476 m; rotor–stator clearance $0.000\,076\,2 \pm 0.000\,025\,4$ m; ambient air temperature and pressure 27.22°C and 101.3 kPa respectively; and relative humidity 25%. The materials of the stator, rotor and blades are vespel and all the friction coefficient

Figure 4 The variation range of the air velocity inside the heat exchanger

Figure 4 *La gamme de vitesses de l'air à l'intérieur de l'échangeur de chaleur*

Figure 5 Effect of the heat exchanger length on the air pressure at the final outlet of the system

Figure 5 *Effet de la longueur de l'échangeur de chaleur sur la pression de l'air à la sortie définitive du système*

Figure 6 Effect of the heat exchanger length on the air temperature at the final outlet of the system

Figure 6 *Effet de la longueur de l'échangeur de chaleur sur la température de l'air à la sortie définitive du système*

of the material is 0.11. The thermal conductivities of air and the heat exchanger used for analysis are $0.0099 \text{ J s}^{-1} \text{ m}^{-1} \text{ K}^{-1}$ and $117.49 \text{ J s}^{-1} \text{ m}^{-1} \text{ K}^{-1}$ respectively. The inside diameter, wall thickness and length of the heat exchanger are 19.1 mm, 1.55 mm and 7.62 m respectively. The values of n_{c1} and n_{c3} are determined to be 1.30 and 1.35 respectively, based on a comparison of the calculated results with the experimental data¹. It is assumed that the effects of water vapour condensation during expansion and the reduction in air volume after condensation are neglected. The current calculated results and the data obtained from Edwards¹ are

Figure 7 Effect of the heat exchanger length on the cooling capacity of the system

Figure 7 *Effet de la longueur de l'échangeur de chaleur sur la puissance de refroidissement du système*

Figure 8 Effect of the heat exchanger length on the thermal coefficient of performance and the total coefficient of performance of the system

Figure 8 *Effet de la longueur de l'échangeur de chaleur sur les coefficients de performance thermique et global du système*

summarized in *Tables 1* and *2* for comparison.

The variation ranges of the air temperature, pressure and velocity inside the heat exchanger versus the location along the heat exchanger are shown in *Figures 2, 3 and 4* respectively. The effects of the heat exchanger length on the air pressure and temperature at the final outlet of the system are shown in *Figures 5 and 6* respectively. Also, the effects of the heat exchanger length on the cooling capacity, the thermal coefficient of performance and the total coefficient of performance of the system are shown in *Figures 7 and 8* respectively.

Discussion

The calculated variation ranges of the air velocity, pressure and temperature in the heat exchanger may be as large as 129, 10 and 3% of the respective mean values. As the air properties change during the periodic motion of the blades, the effects of unsteady flow exist.

The calculated temperature and pressure of the air at various locations and the performance of the system are compared and show good agreement with the data obtained by Edwards¹. Comparing the calculated heat transfer rate out of the heat exchanger from Equation (9), 9568 kJ h^{-1} , with the difference of the air enthalpy at the inlet and the outlet of the heat exchanger, 9513 kJ h^{-1} , shows that the calculated result is correct.

Various lengths of the heat exchanger up to 7.62 m are investigated. The temperature and velocity of the air do

not change appreciably for heat exchangers longer than 5.08 m. The temperature profile along the heat exchanger is comparable with that of the experimental model. It also shows that the heat out of the heat exchanger, the cooling capacity, the thermal coefficient of performance and the total coefficient of performance of the roto-cooler system do not increase appreciably for heat exchangers longer than 5.08 m. The thermal coefficient of performance begins to decrease gradually for heat exchangers longer than 6.35 m. As the length of the heat exchanger increases, the air pressure at the heat exchanger outlet decreases because of the friction effect. Therefore the recovered work of expansion decreases and the thermal power required increases. The air temperature at the exit of the roto-cooler system is -15.2°C . Further analysis is worthwhile to determine the water vapour condensation and reduction of the air volume during the expansion process. Atmospheric air can be used to mix with this cold air to reduce the effect of water vapour condensation.

Conclusion

The prototype roto-cooler system delivers the required design cooling capacity, but the coefficient of performance of the system is low. The calculated results prove that the prototype heat exchanger was designed conservatively. Friction losses are a large fraction of the power required to drive the roto-cooler system. Mechanical friction and compression losses for the roto-cooler may be reduced by design improvements.

State point data of the prototype obtained from experiments are used to determine the best polytropic exponents in the analytical model. The results are limited to conditions similar to those used in the tests of the prototype. However, the performance is not sensitive to the choice of these exponents for the range of interest. Therefore, the mathematical model could be used to predict performance of the system with a wide range of operation conditions. A heat exchanger 5.08 m long,

which is 2.54 m shorter than that of the prototype, can be used without loss of system performance.

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