Thermal Modelling for the Motor in Semi-hermetic Screw Refrigeration Compressor under Part-load Conditions

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ABSTRACT

Semi-hermetic screw refrigeration compressor is mostly driven by a three phase asynchronous induction motor with its squirrel cage rotor attached to the shaft of the compressor helical male rotor. The motor is cooled by low temperature refrigerant coming from evaporator before it is inspired into compressor suction port. In order to investigate cooling effects of the motor and performance of the compressor under part-load conditions, a lumped parameter model describing heat transfer characteristics inside the motor and a part-load model of screw refrigeration compressor are developed. Heating effects at compressor suction end are taken into account and nodes representing refrigerant inside the motor cooling passages are modeled separately. Due to the decreasing refrigerant mass flow rate and motor efficiency, temperature rise of refrigerant through the motor increases and stator winding temperature curve goes up when the compressor load drops by suction volume variation.

1. INTRODUCTION

Development of high efficiency double-side asymmetric profile, maturity of rapid precision machining technology and introduction of oil injection make twin screw refrigeration compressor have various advantages, such as stable operation, high efficiency, reliable and compact structure. Semi-hermetic screw refrigeration compressor is widely utilized in industrial and commercial refrigeration systems. In the semi-hermetic arrangement, the squirrel cage rotor of three phase asynchronous induction motor is directly attached to the shaft of compressor helical male rotor, refrigerant gas flows through the motor and absorbs heat dissipation due to motor losses before it is inspired into compressor suction port, refrigerant and oil leakage through the shaft seal occurring in the open-type arrangement can be avoided in addition to more compact compositions.

Temperature rise is an important parameter to evaluate performance of the motor, it is directly related to the motor lifetime, output power and efficiency. With increasing requirements for energy efficiency and cost reduction, as well as the imperative to fully exploit new topologies and materials, temperature estimate inside the motor is increasingly essential (Boglietti et al., 2009). Besides, semi-hermetic screw refrigeration compressor is frequently operated under part-load conditions to meet the load variation caused by different seasons and applications. When the compressor is under part-load conditions, mass flow rate of refrigerant through the motor decreases, so does the motor efficiency because its output power deviates from the rated one, then the motor may not be cooled sufficiently and the superheat degree of refrigerant at the compressor suction port would rise, adversely affecting the reliable continued operation and performance of the semi-hermetic screw refrigeration compressor. Consequently, it is of great importance to study heat transfer characteristics inside the motor and investigate their influences on performance of the semi-hermetic screw refrigeration compressor under part-load conditions, which can be helpful for motor selection, arrangement of cooling passages and improvement of part-load efficiency.

2. THERMAL ANALYSIS OF THE MOTOR

In order to make sure sufficient motor cooling effect and minimize temperature rise and pressure drop of refrigerant through the motor, coolant channels inside the motor should be arranged according to distribution of motor losses. Losses inside the three phase asynchronous induction motor can be divided into iron losses, ohmic losses and stray load losses by the induced mechanism. Therefore there are three flow domains for motor cooling in the semi-hermetic screw refrigeration compressor shown in Figure 1, stator cooling ducts, inner gap and rotor cooling holes.

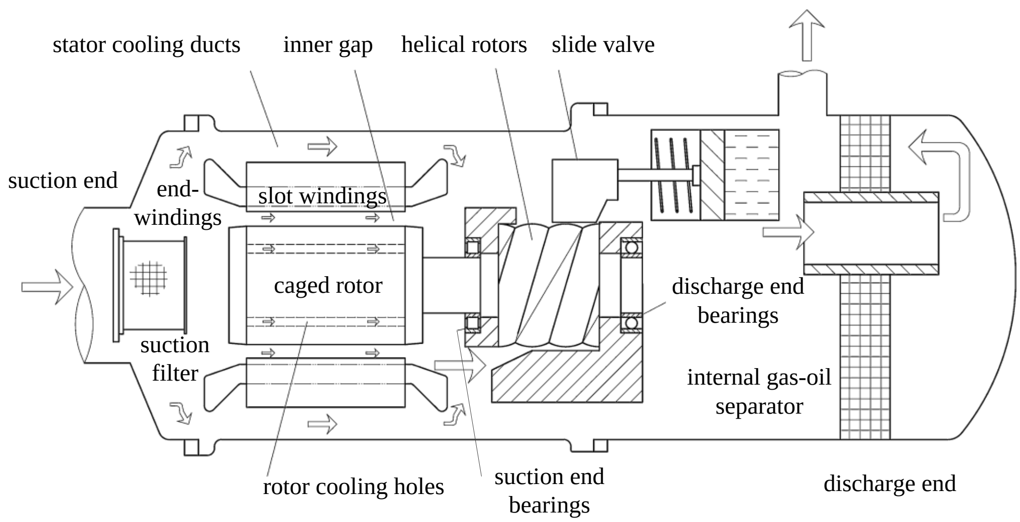


Figure 1: Schematic of semi-hermetic screw refrigeration compressor (exaggerated inner gap)

2.1 Thermal equivalent circuit

Thermal equivalent circuit, the most common approach of the lumped parameter method, has become the main tool used by many researchers involved in thermal analysis of electrical machines (Okoro, 2005). As shown in Figure 2, the motor is geometrically divided into a series of lumped elements, each element is indicated by a thermal node storing thermal information like heat generation and thermal capacity and interconnecting to neighboring nodes through a linear mesh of thermal resistances, average temperature of the element is represented by the node temperature. The circular nodes represent motor solid components, whereas the square ones indicate refrigerant gas in coolant channels. Combining and solving ordinary differential energy conservation equations of all the thermal nodes, steady temperature distribution inside the motor can be obtained.

Heat generation is equivalent to current source in electrical circuit, including electrical losses, heating effects at compressor suction end, friction and gas-flow losses. Electrical losses are derived by electromagnetic analysis and then applied to the corresponding thermal nodes. Friction losses are caused by rotor rotation and mainly generated in inner gap, space around rotor end-rings and rotor cooling holes. Gas-flow losses are associated with tangential acceleration of refrigerant gas in inner gap and rotor cooling holes. Friction and gas-flow losses are significant only at high rotation speeds (higher than 20000 r min^{-1}) (Saari, 1998) while the rotation speed is usually lower than 6000 r min^{-1} for the motor in semi-hermetic screw refrigeration compressor, so they are ignored in this paper. Heating effects of oil back from suction end bearings, bearing losses and by-pass flow venting back through the by-pass port when the compressor is under part-load conditions by movement of a slide valve, are dependent on rotation speed, by-pass flow state, oil temperature and flow rate as

$$TS_{ref} = M_{oil}c_{oil}\Delta T_{oil} + 0.15\frac{F}{d}u\times10^{-5} + M_{bp,oil}c_{oil}\Delta T_{bp,oil} + M_{bp,g}\Delta h_{bp,g}\times10^{3} \qquad (1)$$

where c_{oil} represents oil specific heat capacity, M_{oil} and ΔT_{oil} indicate oil mass flow rate and temperature difference back from suction end bearings respectively, F is bearing load, d and u represent diameter and peripheral velocity at the circle connecting centers of bearing balls respectively, $M_{bp,oil}$ and $\Delta T_{bp,oil}$ denote mass flow rate and temperature difference of by-pass oil flow, while $M_{bp,g}$ and $h_{bp,g}$ denote mass flow rate and specific enthalpy of by-pass refrigerant gas respectively.

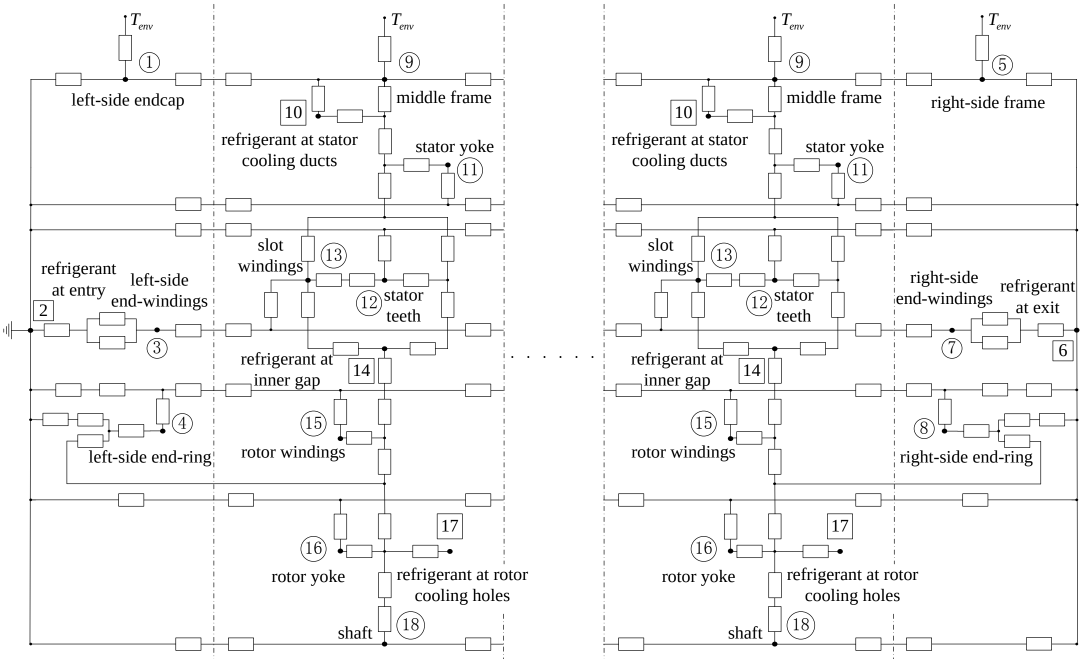


Figure 2: Thermal equivalent circuit of the motor in hermetic refrigeration compressor

Thermal storage represents the ability of storing heat energy and resisting temperature rise in a certain region, it is disabled for the thermal equivalent circuit when the semi-hermetic screw refrigeration compressor is under steady operation. Thermal resistance denotes heat transfer intensity between neighboring nodes. For the motor in semi-hermetic screw refrigeration compressor, only heat conduction and convection are considered, whereas heat radiation is so weak that it can be neglected (Boglietti et al., 2006). Conduction thermal resistance between motor solid components is equal to the path length η divided by product of the path area A and the material's thermal conductivity λ :

$$TR_{cond} = \frac{\eta}{\lambda A} \tag{2}$$

Convection thermal resistance is equal to one divided by product of the surface area A and convection heat-transfer coefficient α :

$$TR_{conv} = \frac{1}{\alpha A} \tag{3}$$

2.2 Modeling of refrigerant

In semi-hermetic screw refrigeration compressor, most of the heat generated from motor losses is removed by refrigerant gas due to the forced open cooling circuit, and there is a rather high temperature rise for refrigerant when it arrives at compressor suction end. The irreversible flow direction makes that temperature of downstream flow is always higher than that of upstream flow and refrigerant gas moves with heat from a node to the next one, heat does not flow simply from higher temperature nodes to lower temperature nodes as happening in solid components. Hence, heat exchange between refrigerant elements must be analyzed separately to get reliable calculated results of temperature distribution inside the motor with forced open cooling circuit. It is assumed that refrigerant through the motor is incompressible with constant properties and the heating of refrigerant gas is linear, then refrigerant temperature rise for the ith section in a flow passage can be written as

$$\Delta T = 2R_q \Phi_i \tag{4}$$

R_q is thermal resistance of refrigerant defined as

$$R_q = \frac{1}{2M_i c_p} \tag{5}$$

Φ_i is sum of heat absorbed in node i, M_i is mass flow rate of refrigerant gas passing node i. Then it can be concluded that: temperature rise at refrigerant node i relative to earth is equal to sum of two products, the first is $2R_q$ multiplied by heat absorbed by refrigerant before node i and the second is R_q multiplied by heat absorbed at node i :

$$T_i = \sum_{j=1}^{i-1}\left(2R_q\Phi_j\right) + R_q\Phi_i \tag{6}$$

3. PART-LOAD MODEL OF SCREW COMPRESSOR

The working process of screw refrigeration compressor under part-load conditions is a rather complicated variable mass thermodynamic process, where the effects of two-phase flow leakage, oil injection, heat transfer and capacity control have to be taken into account. The model is developed based on energy conservation in the working chamber

$$\mathrm{d}\left(mu\right) = \mathrm{d}E_i - \mathrm{d}E_o - \mathrm{d}Q + \mathrm{d}W \tag{7}$$

the item on the left side means increment of internal energy, the right four items in order are energy carried in by inflow, energy carried out by outflow, heat removed from refrigerant and power consumed in the working chamber. Continuity equation and other additional relations, such as heat-transfer calculation and refrigerant properties, are also employed. Then capacity control process has to be simulated to make up a complete model describing working process of screw refrigeration compressor under part-load conditions.

Suction volume variation, achieved by the slide valve motion, is still the main method widely employed for capacity control in the twin screw compressor. A by-pass port appears between the slide valve and slide stop when the compressor is under part-load conditions, and the normal compression process is postponed due to the by-pass process. In screw refrigeration compressor, oil is usually injected into the working chamber at the preliminary stage of compression soon after suction and the oil port position does not change with movement of the slide valve. Therefore the by-pass process partly overlaps with the oil injection process, and the flow through the by-pass port should be treated as two-phase flow since much oil has been injected into the working chamber. The mass flow rate of gas-oil mixture through the by-pass port can be expressed by (Lin, 2003)

$$\begin{cases} \dot{m} = \dfrac{Ca\left(SA\right)\sqrt{2\rho_l \Delta P}}{\left(1-x\right)\varphi + x\sqrt{\rho_l/\rho_g}} \\ \dot{m}_g = x\dot{m} \\ \dot{m}_l = \left(1-x\right)\dot{m} \\ \varphi = 1.48625 - 9.26541\left(\dfrac{\rho_g}{\rho_l}\right) + 44.6954\left(\dfrac{\rho_g}{\rho_l}\right)^2 - 60.6150\left(\dfrac{\rho_g}{\rho_l}\right)^3 \\ \qquad -5.12966\left(\dfrac{\rho_g}{\rho_l}\right)^4 - 26.5743\left(\dfrac{\rho_g}{\rho_l}\right)^5 \end{cases} \tag{8}$$

where *a* is the void fraction, *x* is the ratio of refrigerant in the gas-oil mixture, *C* is the flow coefficient.

4. RESULTS AND DISCUSSION

For the sake of model validation, experimental study on temperature distribution inside the motor and performance of semi-hermetic screw refrigeration compressor under part-load conditions by slide valve motion is implemented. Main parameters of the semi-hermetic screw refrigeration compressor are listed in Table 1. Several K-type thermocouples are immersed into stator windings; refrigerant thermodynamic states at motor entry and compressor suction port are measured.

Table 1 Main parameters of the semi-hermetic screw refrigeration compressor

Parameter	Value	Parameter	Value
Refrigerant	R22	Number of stator cooling ducts	5
Voltage (V)	380	Cross angle of each stator cooling duct (deg)	48
Supply frequency (HZ)	50	Stator outer radius (mm)	150
Number of motor poles	2	Frame inner radius (mm)	162.5
Rated power (kW)	67	Number of rotor cooling holes	3
Length of stator core (mm)	241	Radius of rotor cooling holes (mm)	4
Inner gap length (mm)	1.0	Compressor maximum displacement ($m^3 \cdot h^{-1}$)	237

Volumetric efficiency and power consumption of the compressor at different slide valve positions with an evaporation temperature of -5 ℃ and a condensation temperature of 35 ℃ are shown in Figure 3. The superheat degree on the suction line is 12 ℃. Figure 4 presents temperature curve of stator windings, temperature rise and pressure drop of refrigerant under part-load conditions. It can be observed that measured and calculated results are in good agreement with each other, indicating that the model developed in this paper is reasonable for thermal analysis of the motor in semi-hermetic screw refrigeration compressor under part-load conditions.

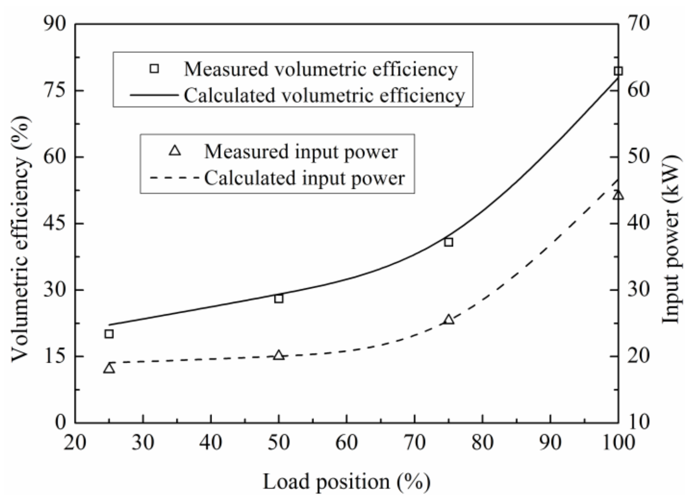
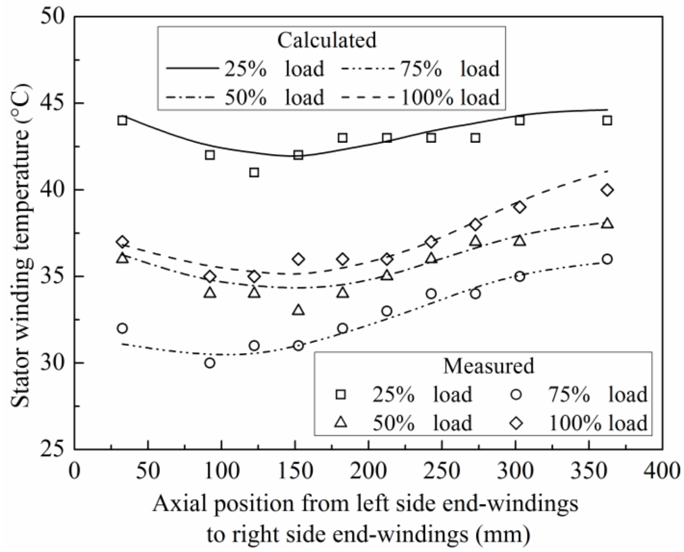


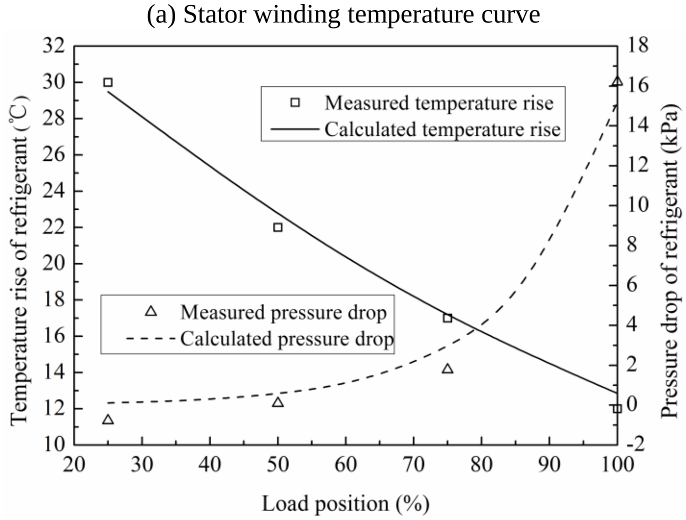
Figure 3: Performance of the compressor by suction volume variation

For windings in stator slots, the axial thermal conductivity is almost the same as it is for copper, while the radial thermal conductivity is far lower because of the presence of different insulation layers. As a result, most of the heat transformed from stator copper losses is removed from stator stack area to end-windings, turning end-windings into the hottest points inside the motor. However, it can be relieved to some extent for end-windings closed to motor entry by the prior cooling with the refrigerant just coming from suction pipe. Consequently, the temperature curve of stator windings along the axial direction should be declining firstly from end-windings closed to motor entry (denoted as left side) and then rising to the highest point at end-windings closed to compressor suction port (denoted as right side).

As the compressor load drops, both refrigerant mass flow rate and power consumption decrease (Figure 3), but the decrement rate of power consumption becomes smaller at lower load positions due to the larger proportion of irreversible losses (Chen et al., 2011). Moreover, motor efficiency will decline because the output power of the motor is much smaller than the rated power. As a result, stator winding temperature increases and temperature difference between the left side and right side end-windings decreases as shown in Figure 4 (a). Due to the larger power consumption when the compressor is fully loaded, stator winding temperatures are higher than that under 75% and 50% load positions. Heating effects of the oil and refrigerant gas venting back from the by-pass port, combined with the less mass flow rate, make the temperature rise of refrigerant increase sharply and suction pressure drop go down while the compressor loads down as shown in Figure 4 (b).



(a) Stator winding temperature curve



(b) Temperature rise and pressure drop of refrigerant

Figure 4: Measured and calculated results inside the motor by suction volume variation

5. CONCLUSIONS

Based on the lumped parameter method, thermal equivalent circuit of the motor in semi-hermetic screw refrigeration compressor has been developed, where heating effects at compressor suction end are taken into account and refrigerant inside the motor cooling passages is modeled separately. The part-load model of screw refrigeration compressor has been also established. Combining the validated models, temperature distribution inside the motor and performance of the semi-hermetic screw refrigeration compressor under part-load conditions can be calculated. The comprehensive model developed in this paper can be used for motor selection, cooling passage arrangement and efficiency improvement of semi-hermetic screw refrigeration compressor under part-load conditions in the following work. When the semi-hermetic screw refrigeration compressor load drops by slide valve motion, both the stator winding temperature and temperature rise of refrigerant through the motor increase, the compressor should not run under too low load positions.

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