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Study on the operation variables by the static characteristic experiment and analysis calculation of gas-injected scroll compressor

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Abstract
Gas-injected scroll compressors have recently attracted attention in vapor compression systems, especially in high temperature lift applications, due to the advantages they offer. Modeling the steady state of a gas-injected scroll compressor is of considerable importance in considering the behavior of the compressor's characteristic parameters. Semi-empirical models are proposed primarily for standard scroll compressors to accurately calculate compressor performance without knowledge of the compression process or geometry. Verification of the model accuracy based on experimental data of previous studies has been carried out.

Keywords: Scroll compressor, semi-empirical, calculation model, gas-injected

Introduction

Scroll compressors are widely used in residential heat pumps and air conditioners due to their advantages such as low noise and high efficiency (Duprez *et al*., 2007 ; Cuevas and Lebrun, 2009) [4, 2].

Further research is needed to improve the performance of scroll compressor.

Among them, refrigerant injection technology has attracted greater attention in high temperature lift applications due to inherent advantages such as increase of system capacity (Wang *et al*., 2008) [13], inherent adjustment of system capacity trough of injected refrigerant mass flow rate variation (Wang *et al*., 2008) [13], and reduction of refrigerant temperature in compressor discharge with associated enlargement of compressor operating range.

However, despite the advantages offered by this kind of compressor, the optimal control strategy (Heo *et al*) as well as the optimal structure (Ma and Zhao, Wang,) are still under investigation.

In this context, the development of a mathematical model of a scroll compressor with steam injection can be helpful to support the analysis of system configuration or control.

Many studies have been carried out on modeling of steam jet scroll compressor.

Indeed, many studies available in the open literature deserve attention to the simulation of steam jet scroll compressors, but unlike the modeling of standard scroll compressors where three distinct modeling techniques (geometric, semi-empirical, and empirical modeling) are mainly proposed, the geometric models for steam jet scroll compressors are mainly presented with few exceptions, as discussed below (Dardenne *et al*., 2015) [3].

Winandy and Lebrun (2002) [14] developed a semi-empirical model of a fixed-speed scroll compressor with a refrigerant injection working with R22. The semi-empirical model was able to calculate the refrigerant mass flow rate, the compressor emf and the refrigerant temperature in the compressor discharge.

Ma and Chai (2004) [8] developed and validated a thermodynamic model of the compression process inside a fixed-speed scroll compressor with a steam jet working with R22 (Ma *et al*. 2004) [8].

(2003) data. The model was used to study the influence of refrigerant injection pressure on compressor operating parameters as a function of evaporation temperature.

Wang *et al*. (2008) developed and validated a deterministic model of a fixed-speed scroll compressor with a steam jet working with R22The sensitivity analysis of the model carried out, leading to the conclusion that the heat transfer between the scroll cover and the refrigerant and the was backpressure space configuration have very little influence and can be neglected in the modeling process. Finally, the concept of continuous "adiabatic throttling + isobaric mixing" for describing the refrigerant injection process in scroll compressor was introduced.

Qiao *et al*. (2015a) [9] developed a transient and intensive model of a fixed-speed scroll with a gas injection working with R410A.

The compression process is simulated by introducing a polytropic process, whose index is estimated based on the experimental results and the model requires eight curve fitting constants to simulate the compressor.

All the previous papers provide valuable models of gas-injected scroll compressor.

The choice of modeling method for scroll compressor depends largely on the level of detail required and the specific purpose of the research or design process.

The authors validate the accuracy of the model using experimental observations on the operation of a compressor with and without gas injection. The modeling approach in this study is intended to be applicable to fixed-speed and variable-speed compressors for operation with and without gas injection.

The proposed model requires relatively few empirical parameters to reduce the need for experimental testing.

2. Modelling and methods

In this study, the modeling method requires several inputs to describe the operating conditions of the compressor.

It is assumed that the thermodynamic states of the refrigerant are known at the inlet and inlet, and additional thermodynamic properties can be evaluated in the range of operation.

Using inputs and empirically derived model parameters, the method estimates the following quantities:

On the other hand, it was recognized that the following assumptions are necessary for the modeling based on the theory of gas-injected scroll compressor:

- Heating through the intake duct wall of the compressor during refrigerant intake is neglected.
- The cooling due to heat loss through the compressor shell during the discharge of the refrigerant is neglected.
- The effect of compressor lubricant is neglected.
- The effect of gravity is neglected.

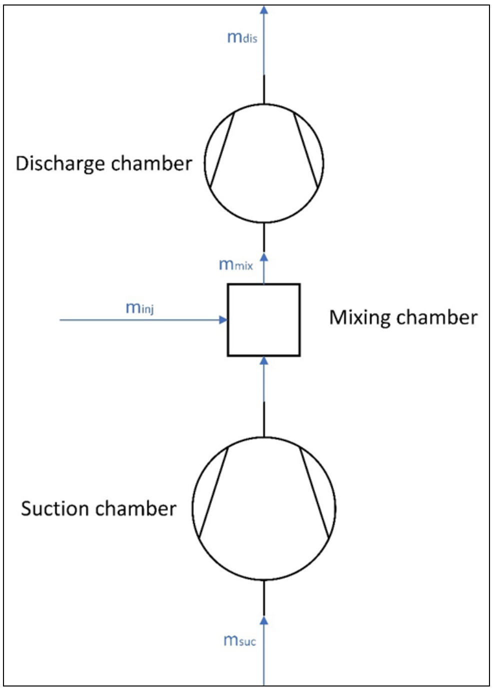

Fig 1: Simplified schematic diagram of a gas-injected scroll compressor

For modelling based on the theory of a gas-injected scroll compressor, the following assumptions are considered:

- Suction gas heating is neglected.
- The heat loss through the compressor shell during refrigerant discharge is neglected.
- The effect of compressor lubricant is neglected.
- The influence of gravity is neglected.

Although previous studies argued that the mixing process of the injected refrigerant and the compressed one is strictly an isobaric mixing process after throttling, we approximated it as an isobaric mixing process without throttling, so the process analysis for modelling consists of the following three steps.

- First polytropic compression of the suction refrigerant up to the injection pressure
- Isobaric mixing of compressed refrigerants and injection refrigerants
- Second polytropic compression of the mixed refrigerant up to the discharge pressure

2.1 Modeling of suction mass flow rate

To model the volumetric flow rate of scroll compressors, the approach proposed by Hjortland and Crawford was adopted here to model the suction flow rate of compressors with steam injection (Hjortland *et al*., 2024) [5].

To measure the deviation of the actual suction process from the ideal case, a compressor performance metric known as

volumetric efficiency can be used.
According to the results of previous studies, it is assumed that the volume displacement of the compressor depends mainly on the pressure ratio of the compressor.
Therefore, the compressor suction mass flow rate follows Eq. (1).

$$m_{suc} = \eta_v \rho_{suc} N V_{suc} \quad (1)$$

$$\eta_v = k_v \eta_{v,ref} \quad (2)$$

As for the volumetric efficiency at rated frequency ($\eta_{v,ref}$), it can be considered as the volumetric efficiency of a compressor with fixed frequency, which is considered to be in linear relationship with the ratio of discharge pressure to suction pressure (Sun *et al*., 2018) [11], and is expressed as follows:

$$\eta_{v,ref} = a_1 + a_2 \frac{p_{inj}}{p_{suc}} \quad (3)$$

$$k_v = \left(\frac{N}{N_{ref}}\right)^{a_3} \quad (4)$$

Here m_{suc} represents the suction mass flow rate, η_v represents the volumetric efficiency at the corresponding frequency, N represents the rotation speed of orbiting scroll (rps), V_{suc} represents the volumetric displacement per revolution of the suction chamber (m^3/r), v_{suc} represents the specific volume of suction flow (m^3/kg), P_{suc} represents the suction pressure respectively.
By combining Eq. (1)-(4), an explicit equation of suction mass flow rate based on the theory can be obtained as expressed in Eq. (5).

$$m_{suc} = \rho_{suc} N V_{suc} \left(a_1 + a_2 \frac{p_{inj}}{p_{suc}}\right)\left(\frac{N}{N_{ref}}\right)^{a_3} \quad (5)$$

$a_1 \sim a_3$ are directly regressed by the experimental data.

2.2 Modeling of injection mass flow rate
According to the mass conservation in the virtual mixing chamber, the explicit equation of injection mass flow rate based on the theory is derived from Eq. (6) (Sun *et al*., 2018) [11].

$$m_{mix} = m_{suc} + m_{inj} \quad (6)$$

$$m_{mix} = \rho_{mix} N V_{inj} \left(b_1 + b_2 \frac{p_{dis}}{p_{inj}}\right)\left(\frac{N}{N_{ref}}\right)^{b_3} \quad (7)$$

And the mass flow rate of the mixed refrigerant is expressed as in Eq. (7) by fitting it to the derivation of Eq. (5).

Here m_{inj} denotes the mass flow rate of the injected refrigerant and m_{mix} denotes the mass flow rate of the mixed refrigerant, respectively.
On the other hand, with the energy conservation for the mixing process, Eq. (8) holds and also the enthalpy of the mixed refrigerant, the suction refrigerant and the injected refrigerant can be considered as a linear function of the specific volume, with the same pressure and similar temperature as expressed in Eq. (9) (Sun *et al*., 2018) [11].

$$m_{mix} h_{mix} = m_{suc} h_{com} + m_{inj} h_{inj}$$

$$\begin{cases} h_{mix} = k_1 + \dfrac{k_2}{\rho_{mix}} \\ h_{com} = k_1 + \dfrac{k_2}{\rho_{com}} \\ h_{inj} = k_1 + \dfrac{k_2}{\rho_{inj}} \end{cases}$$

From Eq. (5), (8), and (9), the equation of the injection mass flow rate was derived based on the theory that the ratio of the specific volume of gas in the compression process is equal to the ratio of the volume of the compressed space as follows:

$$\begin{aligned} m_{inj} &= \rho_{inj}\left(\frac{m_{mix}}{\rho_{mix}} - \frac{m_{suc}}{\rho_{com}}\right) = \\ &= \rho_{inj}\left[N V_{inj}\left(b_1 + b_2 \frac{P_{dis}}{P_{inj}}\right)\left(\frac{N}{N_{ref}}\right)^{b_3} - \frac{V_{inj}}{V_{suc}} N V_{suc}\left(a_1 + a_2 \frac{P_{dis}}{P_{inj}}\right)\left(\frac{N}{N_{ref}}\right)^{a_3}\right] = \\ &= \rho_{inj} N V_{inj}\left[\left(b_1 + b_2 \frac{P_{dis}}{P_{inj}}\right)\left(\frac{N}{N_{ref}}\right)^{b_3} - \left(a_1 + a_2 \frac{P_{dis}}{P_{inj}}\right)\left(\frac{N}{N_{ref}}\right)^{a_3}\right] \end{aligned} \quad (10)$$

2.3 Modeling of total input power
To develop a semi-empirical model of total input power, Sun *et al*. (2018) [11] first proposed an explicit model based on the theory of total input power of a gas-injected scroll compressor.
Since the heat loss through the compressor shell during the scroll compressor operation was assumed to be negligible, from the previous study, the following equations are established:

$$W_{tot} = W_{suc} + W_{dis}$$

$$\begin{cases} W_{suc} = m_{suc} P_{suc} v_{suc} \left(\dfrac{n_{suc}}{n_{suc}-1}\right)\left[\left(\dfrac{P_{inj}}{P_{suc}}\right)^{\frac{n_{suc}-1}{n_{suc}}} - 1\right] \\ W_{dis} = m_{mix} P_{inj} v_{mix} \left(\dfrac{n_{inj}}{n_{inj}-1}\right)\left[\left(\dfrac{P_{dis}}{P_{inj}}\right)^{\frac{n_{inj}-1}{n_{inj}}} - 1\right] \end{cases}$$

Here, W_{tot}, W_{suc} and W_{dis} represent the total input power, the theoretical input power in the first compression process, and the theoretical input power in the second compression process, respectively.

On the other hand, n_{suc} and n_{inj} in Eq. (15) are polytropic exponents for the first and second compressions, respectively.

Therefore, Eq. (12) follows the following derivation:

$$\begin{cases} W_{suc}=c_1 m_{suc}P_{suc}v_{suc}\left(\left(\dfrac{P_{inj}}{P_{suc}}\right)^{c_1}-1\right) \\ W_{dis}=c_2 m_{mix}P_{mix}v_{mix}\left(\left(\dfrac{P_{dis}}{P_{inj}}\right)^{c_2}-1\right) \end{cases} \quad (13)$$

Consequently, the equation of power consumption is:

$$W_{tot}=c_1 m_{suc}P_{suc}v_{suc}\left(\left(\frac{P_{inj}}{P_{suc}}\right)^{c_1}-1\right)+c_2 m_{mix}P_{mix}v_{mix}\left(\left(\frac{P_{dis}}{P_{inj}}\right)^{c_2}-1\right) \quad (14)$$

2.3 Modeling of discharge temperature

An explicit equation of discharge temperature is developed using Eq. (15).

Here T_{mix} is derived from Eq. (16).

$$T_{dis}=T_{mix}\left(\frac{P_{dis}}{P_{inj}}\right)^{\frac{n_{inj}-1}{n_{inj}}} \quad (15)$$

$$\begin{cases} h_{mix}=k_3+k_4T_{mix} \\ h_{com}=k_3+k_4T_{com} \\ h_{inj}=k_3+k_4T_{inj} \end{cases} \quad (16)$$

Equation (15) is expressed as follows:

$$T_{dis}=\frac{m_{suc}T_{com}+m_{inj}T_{inj}}{m_{mix}}\left(\frac{P_{dis}}{P_{inj}}\right)^{\frac{n_{inj}-1}{n_{inj}}} \quad (17)$$

On the other hand, T_{com} in Eq. (17) is developed by the preceding theory as follows:

$$T_{com}=T_{suc}\left(\frac{P_{inj}}{P_{suc}}\right)^{\frac{n_{suc}-1}{n_{suc}}}$$

Consequently, the explicit equation of the discharge temperature is expressed by Eq. (19).

$$T_{dis}=\frac{m_{suc}T_{suc}\left(\dfrac{P_{inj}}{P_{suc}}\right)^{d_1}+m_{inj}T_{inj}}{m_{suc}+m_{inj}}\left(\frac{P_{dis}}{P_{inj}}\right)^{d_2}$$

3. Results and discussion

3.1 Verification of model accuracy

The gas-injected scroll compressor used in the tests works with R410A refrigerant and POE lubricant, the intake volume of the compressor is 5.3 m^3/h at 50 Hz (Dardenne *et al*., 2015) [3]; the mass flow rate was measured by Coriolis flowmeter, and the pressure and temperature of the refrigerant were measured by a pressure gauge and a temperature sensor (Pt100) (Dardenne *et al*., 2015); the thermodynamic characteristics of the refrigerant (specific volume and enthalpy) based on the measured pressure and temperature are calculated through REROP 9.1 (Lemmon *et al*., 2013).

The coefficients in 4 equations are regressed using the least squares method as Eq. (20), and all experimental data (63 experimental data) presented by Dardenne *et al*. are used to regress the coefficients.

$$E_{\min}^2=\min\left[\sum_{i=1}^{M}\left(Y_{cal,i}-Y_{exp,i}\right)^2\right]$$

where E_{min}^2 is the minimum sum of squared error, $Y_{cal,i}$ represents the calculated mass flow rate, total input power and discharge temperature, and $Y_{exp,i}$ represents the corresponding experimental data.

M is the number of experimental points.

The regression explicit equations are used to compare the predicted and experimental values for the main characteristics of the scroll compressor.

Comparison of predicted values and experimental values is shown in Fig. 2-5.

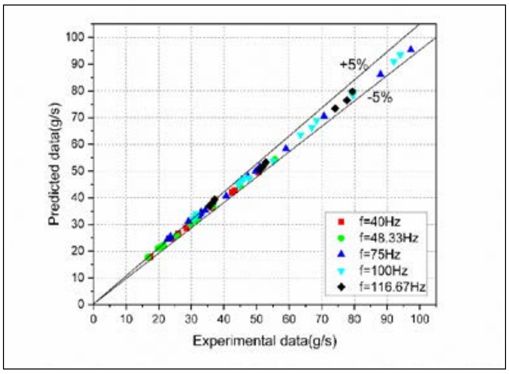

Fig 2: Predicted and experimental suction mass flow rates.

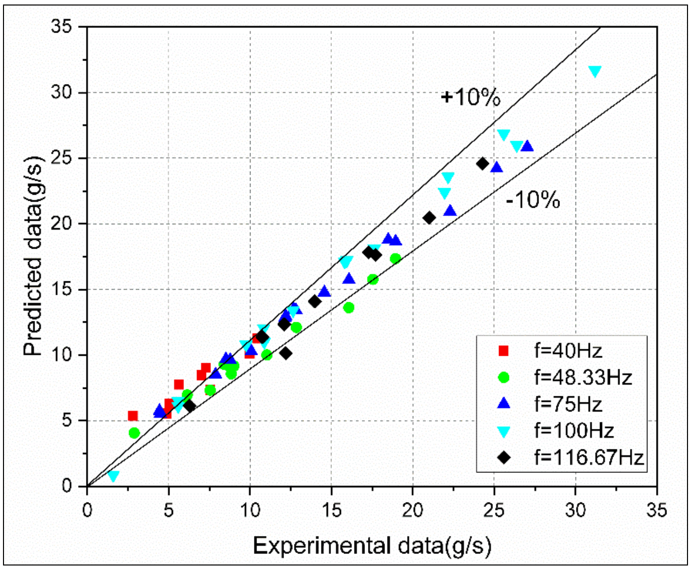

Fig 3: Predicted and experimental injection mass flow rates.

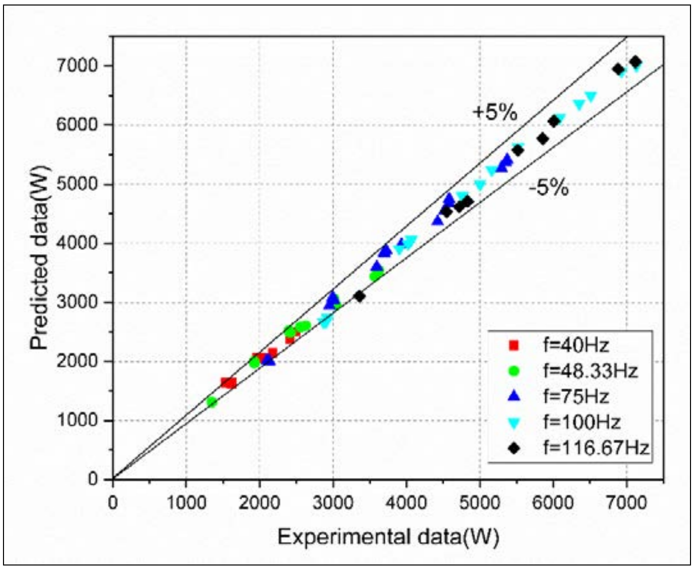

Fig 4: Predicted and experimental total input power.

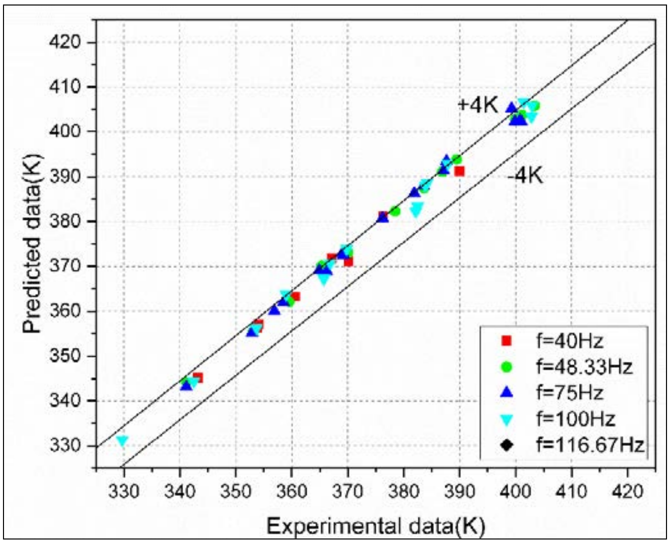

Fig 5: Predicted and experimental discharge temperature.

The calculated results of the mass flow rate can describe 87.1% of the experimental data within a deviation of ±5% the predicted suction mass flow rate and 82.1% of the experimental data within a deviation of ±10% of injection mass flow rate. The calculated results of the total input power show that the predicted input power can describe

90.1% of the experimental data within a deviation of ±5%. The calculated results of the discharge temperature show that the predicted discharge temperature can describe 89.6% of the experimental data within a deviation of ±4 K.

3.2 Comparison of proposed model with previous model

To evaluate the improvement of the proposed model over previous studies, the model of Sun *et al.* which was originally developed an explicit calculation model for the working process of a variable speed scroll compressor with vapor injection was used.

For objective comparison of the proposed model and the previous ones, we used the experimental data by Dardenne's experimental data (Dardenne *et al*., 2015) [3] as we already mentioned, and we applied the same regression method (using interpolation function of MATLAB) to determine the coefficients of the explicit equation. As a result, we obtained calculations that were of significant difference from those of Sun *et al*.

Table 1 shows the comparison of model accuracy.

Table 1: Comparison results of present and Sun s' model.

Main parameter	Model type	
	Sun 's model	**Present model**
Suction mass flow rate	±5%, 85.7%	±5%, 87.1%
Injection mass flow rate	±10%, 73.0%	±10%, 82.1%
Total input power	±5%, 89.5%	±5%, 90.1%
Discharge temperature	±5K, 71.4%	±4K, 89.6%

We have demonstrated a partial (to some extent) improvement in model accuracy.

However, this paper did not reflect the heating process of the refrigerant through the duct wall during the refrigerant suction and did not consider the heat loss through the shell of compressor during the discharge.

4 Conclusion

In this paper, a semi-empirical computational model based on experimental data was developed that can accurately predict the performance parameters without a deep knowledge of the geometry of the gas-injected scroll compressor.

The calculated results showed that the suction mass flow rate, the injection mass flow rate, total input power and discharge temperature were ±5%, 87.1%; ±10%, 82.1%; ±5%, 90.1%; ±4K and 89.6%, respectively, which showed significant improvement.

The proposed computational model is considered suitable for application with relatively high accuracy.

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