Reduction of the suction losses through reed valves in hermetic reciprocating compressors using a magnet coil

J Hopfgartner¹, S Posch¹, B Zuber¹, R Almbauer¹, K Krischan¹ and S Stangl²

¹ Graz University of Technology, 8010 Graz, Austria
² Secop Austria GmbH, 8280 Fürstenfeld, Austria

hopfgartner@ivt.tugraz.at

Abstract. Reed valves are widely used in hermetic reciprocating compressors and are responsible for a large part of the thermodynamic losses. Especially, the suction valve, which is opened nearly during the whole suction stroke, has a big potential for improvement. Usually, suction valves are opened only by vacuum created by the moving piston and should be closed before the compression stroke starts to avoid a reversed mass-flow through the valve. Therefore, the valves are prestressed, which results on the other hand in a higher flow resistance.

In this work, a suction valve is investigated, which is not closed by the preload of the valve but by an electromagnetic coil located in the suction muffler neck. Shortly before the piston reaches its bottom dead centre, voltage is applied to the coil and a magnetic force is generated which pulls the valve shut. Thereby, the flow resistance through the valve can be reduced by changing the preload on the reed valve because it is no longer needed to close the valve. The investigation of this adapted valve and the electromagnetic coil is firstly done by numerical simulations including fluid structure interactions of the reed valves of a reciprocating compressor and secondly by experiments made on a calorimeter test bench.

1. Introduction

Reed valves play a key role in hermetic reciprocating compressors used for domestic refrigerating appliances. Beneath simplicity, their high reliability and low costs are the main reasons why reed valves are so extensively used in hermetic reciprocating compressors. Reed valves open and close depending on the pressure difference between cylinder and suction/discharge muffler, thus reed valves are self-actuating. The piston, moving from the top dead centre to the bottom dead centre, decreases the pressure in the cylinder and creates a vacuum, which forces the suction valve to open. When the forces due to the vacuum become bigger than the valve preload and the oil "stiction" forces, the suction valve starts to open. In reality, the suction valve in a hermetic compressor opens and closes several times during one suction stroke depending on the operating conditions of the compressor and the parameters of the reed valve. It may happen that the suction valve has not closed yet, when the piston starts to move upward again and thus gas can flow back from the cylinder into the suction muffler. To avoid this reversed mass-flow, the preload must force the valve to close exactly at the end of the gas movement into the cylinder.

Many different studies, which investigate the suction and discharge losses inside reciprocating compressors, were done in the past using either experiments, simulations or both. [1] measured the cylinder pressure and valve lift curves of a reciprocating compressor simultaneously to investigate the...
losses. Special attention was paid to the opening and closing delay of the valves as well as back-flow effects. [2] analysed the characteristic of valve dynamics and introduced a 1D model, used for the prediction of the valve performance of different designs of the reed valve. The authors showed a big potential for improvement of the suction valve which can mainly be attributed to a timely closing leading to a reduction of the gas back-flow through the suction port. Similar results can be observed in the work of [3]. The authors changed the rotational speed of the compressor while measuring the lift of the suction valve. Since the movement of the valve is mainly a function of its resonant frequency, the closure time varies related to the bottom dead centre of the compressor at different rotational speeds. This results in different gas back-flow rates and in a fluctuation of the volumetric efficiency. In the work of [4] the influence of different suction valve parameters on the efficiency of the reciprocating compressor was investigated. It was shown, that a reduction of the valve spring preload results in a significant reduction of the indicated power consumption during the suction phase. Nevertheless, a slightly higher back-flow rate of the gas could be observed.

Summarizing, it can be pointed out, that a perfectly designed valve should open as fast as possible, stay open at a maximum lift and close at the right time with low velocities to enlarge the durability. In order to use the big advantage of a lowered preload of the suction valve, this work investigates a compressor with a suction valve, which is negatively pre-stressed. This means, the valve is slightly open at equalized pressure. To close the suction valve in time, an electromagnetic coil which is located in the suction muffler neck is used instead. The coil starts to pull the suction valve in order to shut shortly before the piston reaches its bottom dead centre. The basic idea using electromagnetic forces to close reed valves was already published by [5].

The investigation of this closing mechanism is firstly done by numerical simulations including fluid structure interactions of the reed valves and secondly by experiments made on a calorimeter test bench. For the measurements inside the calorimeter, stabilized operating conditions according to ASHRAE have been applied. Therefore, the R600a-compressor is operated at a fixed rotational speed of 3000 rpm. Simulations as well as measurements show a general potential for improvement of the COP between 0.4 to 1.7 %.

2. Experimental Setup
A conventional suction valve is a prestressed thin piece of steel, which is closed at equalized pressure. In this work, a reed valve with a negative preload, which is closed by a magnetic coil, is investigated. Therefore, a commercially available reciprocating compressor with a displacement of approximately 6 ccm is equipped with extensive measurement instrumentation and a magnetic coil.

Figure 1 shows schematically the magnetic arrangement and a CAD-drawing where the assembly situation of the magnetic coil and the suction valve can be seen. The coil consists of an iron core and a copper winding with approximately 200 turns. It is located above the suction muffler neck and its iron core feeds through the muffler neck to the reed valve. To close the magnetic circuit, an iron cramp is mounted which connects the iron core with the ferromagnetic parts fixing the valve.

**Figure 1.** Magnetic arrangement (a) and assembling situation of the electromagnetic coil (b).
2.1. Electromagnetic coil
To close the suction valve when the piston reaches its bottom dead centre, an electric current is applied to a magnetic coil. The resulting magnetic field closes the suction valve. After the valve has shut and the pressure in the cylinder has risen, the electromagnetic coil is demagnetized by applying a negative voltage to avoid a delay of the suction valve opening and thus an increasing of the suction losses.

To estimate the magnetic force acting between the iron core and the reed valve, the approach described in [6] is used. With the magnetomotive force \( \Theta = N \cdot i \), where \( N \) is the number of turns and \( i \) is the current, the magnetic flux through the coil \( \Phi_c \) is given with:

\[
\Phi_c = \frac{\Theta}{R}
\]  

(1)

Therein, \( R \) is the equivalent reluctance of the whole system consisting of the effective reluctance of the iron core and the iron cramp \( R_c \), the effective reluctance of the valve \( R_v \), the effective reluctance of the gas gap \( R_g \) (between valve and iron core) and the reluctance \( R_l \) which takes the leakage flux into consideration. The equivalent reluctance can finally be written as:

\[
R = R_c + R_v + \frac{R_1 \cdot R_g}{R_l + R_g} = \frac{l_c}{\mu_0 \cdot \mu_t \cdot A_c} + \frac{l_v}{\mu_0 \cdot \mu_t \cdot A_v} + \left( \frac{l_1}{\mu_0 \cdot A_l} \right) \cdot \left( \frac{\alpha \cdot x}{\mu_0 \cdot A_g} \right)
\]  

(2)

To simplify this approach, the relative permeabilities \( \mu_t \) are assumed to be constant. \( l \) and \( A \) are the effective lengths and effective areas of the particular reluctances. With the flux linkage of the coil \( \psi = N \cdot \Phi_c \) and Faraday's law of induction, the voltage \( u_L \) can be written as:

\[
u_L = \frac{d\psi}{dt}
\]  

(3)

Using the coil current \( i(x,\psi) \) expressed in terms of the flux linkage and the gas gap, the magnetic energy is:

\[
W_m = \int_0^\psi i(x,\psi) \, d\psi = \int_0^\psi R(x) \cdot \frac{\psi}{N^2} \, d\psi = \frac{\psi}{2 \cdot N^2} \cdot \psi^2 = \frac{N^2 \cdot i^2 \cdot \left( \frac{\alpha \cdot x}{\mu_0 \cdot A_g} + R_l \right)}{2 \cdot \left( (R_c + R_v + R_l) \cdot \frac{\alpha \cdot x}{\mu_0 \cdot A_g} + R_l \cdot \left( R_c + R_v \right) \right)}
\]  

(4)

With the derivative of the magnetic energy with respect to the gas gap, the magnetic force \( F_m \) is given by:

\[
F_m = \frac{\partial W_m}{\partial x} = -\frac{N^2 \cdot i^2 \cdot \frac{\alpha}{\mu_0 \cdot A_g} \cdot R_l^2}{2 \cdot \left( (R_c + R_v + R_l) \cdot \frac{\alpha \cdot x}{\mu_0 \cdot A_g} + R_l \cdot \left( R_c + R_v \right) \right)^2}
\]  

(5)

The design of the magnetic coil is based on the requirement, that the “negative” preload and the magnetic force acting on the valve should be equal to the preload of the standard reed valve. With a valve deflection of 0.4 mm in its neutral position, a magnetic force of 0.212 N is needed. With the values stated in Table 1, the required number of turns can be estimated based on equation (5):

\[
N_{\text{min}} = \frac{2 \cdot F_m \cdot \mu_0 \cdot A_g \cdot R_g \cdot \left( R_c + R_v + R_l \right) + R_l \cdot \left( R_c + R_v \right)}{\alpha i_{\text{max}} \cdot R_l} \approx 190
\]  

(6)
Table 1. Basic assumptions for the design of the electromagnetic coil.

| value | unit     | value | unit     |
|-------|----------|-------|----------|
| $x$   | 0.4 mm   | $A_g$ | 3.14 mm² |
| $I_c$ | 50 mm    | $A_c$ | 3.14 mm² |
| $I_l$ | 20 mm    | $A_c$ | 0.5 mm²  |
| $I_l$ | 0.4 mm   | $A_l$ | 7.1 mm²  |
| $\alpha$ | 2 | $\mu_0$ | 1.26 e-6 Vs/Am |
| $i_{max}$ | 2.3 A | $\mu_{rc}$ | 1000 |
| $F_{req}$ | 0.212 N | $\mu_{rv}$ | 850 |

It should be noted at this point, that equation (6) is influenced by many different estimated values with a high level of uncertainty, since the magnetic flux in the real application cannot be predicted very well. With this rough estimate and taking several constraints due to the lack of space in the region of the cylinder head into consideration, the parameters for the investigated electromagnetic coil can be determined (Table 2).

Table 2. Parameters of the investigated electromagnetic coil.

| number of turns | unit   | 190 up to 205 |
|-----------------|--------|---------------|
| length of the core | mm    | 31.8          |
| iron core diameter | mm   | 2.5           |
| outer diameter of the coil | mm  | 9.3           |
| diameter of the wire | mm   | 0.5           |

2.2. Settings for the electric circuit

In order to reduce the suction valve losses, the magnetic coil should close the suction valve nearby the bottom dead centre of the piston. Therefore, the position of the piston on the one hand, and the delay between applying voltage to the coil and the completely closing of the valve on the other, must be known. To control the magnetic coil, a full bridge circuit is used which has two control inputs to change the direction of the supply voltage.

2.2.1. Determination of the bottom dead centre. For the determination of the bottom dead centre, a sensor based on the Hall-effect is used. It detects a permanent magnet integrated in the crank shaft and generates a periodic signal with a significant peak when the magnet passes the sensor. To determine the delay between the peak of the signal and the exact position of the bottom dead centre, the movement of the piston at different pressure ratios is measured with a Laser-Doppler vibrometer in a pretest.

2.2.2. Response behaviour of the closing mechanism. The closing behaviour of the adapted suction valve (0.41 mm opened in neutral position) was investigated on a dummy compressor without a piston. The closing time of the valve was measured by a Laser-Doppler vibrometer which was directed at the backside of the valve (middle of the valve plate) while it was shut by the magnetic coil. Figure 2 shows a diagram of the measured valve lifts over time for different coil supply voltages from 2 up to 8 V. The delay between applying voltage to the coil and closing of the reed valve varies between 0.95 and 1.55 ms.
2.3. **Further measurement instrumentation (temperature, pressure)**

Beside the measurement instrumentation necessary for controlling the electromagnetic coil, thermocouples and pressure transducers are installed in the compressor. Several thermocouples distributed on the compressor shell and inside the compressor are used to investigate the influence of the waste heat from the coil on the thermal behaviour of the compressor. The pressure sensors are installed in the suction muffler neck and cylinder head. Figure 3 shows pictures of the installed measurement instrumentation and the electromagnetic coil.

![Figure 3. Electromagnetic coil assembled in the cylinder head (left). Assembling situation of pressure transducers, thermocouples and electromagnetic coil (right).](image_url)

3. **Simulation Model**

Numerical simulation in this work is conducted using the commercial CFD software package Fluent from ANSYS. The simulation domain, which can be seen in Figure 4, includes suction muffler, suction and discharge valve, cylinder, cylinder head and discharge line. The mesh size of the domain varies between 1.9 and 5.2 million cells depending on the position of the piston and the reed valves. The movement of the piston and the reed valves requires a dynamic mesh which is enabled by using the remeshing, layering and smoothing techniques provided by the software. While the forced piston movement can be modelled easily, the movement of the reed valves must be modelled with a more complex fluid structure interaction which is described in a section below.

For the coupling of pressure and velocity, the pressure based coupled solver is applied. Turbulence is considered with the realizable k-ε model using the standard wall function for the near-wall treatment.
3.1. Boundary conditions

The numerical simulations in this work are based on the operating conditions according to ASHRAE (-23 °C / 55 °C). At the inlet of the suction muffler, a pressure inlet boundary condition has been set with a pressure of 62770 Pa and a temperature of 323 K. At the outlet of the discharge line, a pressure outlet boundary of 786000 Pa has been set. The temperature boundary conditions used in the simulations are based on the measurements of [7] and are shown in Table 3.

| Component            | Temperature [K] |
|----------------------|-----------------|
| suction muffler      | 332.1           |
| valve plate          | 360.0           |
| cylinder             | 357.2           |
| cylinder head        | 362.7           |
| discharge line       | 352.0           |

3.2. Modelling of the reed valve movement

Fluid structure interactions (FSI) must be taken into consideration, when reed valves are modelled. Therefore, subroutines are hooked to Fluent via so called User Defined Functions (UDF) which includes the oil stiction effect based on [8].

3.2.1. Discharge valve. In order to reduce the simulation effort, the discharge valve is assumed to be a parallel moving flat plate (see Figure 4). The equation of motion of this spring-mass system, including the preload and the oil stiction effect during the opening, is solved by using an explicit Runge-Kutta method of fourth order:

\[ \ddot{x} = \frac{1}{m^*} \left( F_{\text{pre}} + F_{\text{gas}} - c \cdot x - d \cdot \dot{x} - \frac{c_V}{x^3} \cdot \dot{x} \cdot \frac{c_V}{x^2} \right) \]  

(7)

3.2.2. Suction valve. A more accurate model must be used for the suction valve, since the suction losses are strongly influenced by the deflection and motion of the reed valve. Therefore, a Finite Element Approach is used to solve the nonlinear dynamic structure analysis problem. Forces due to
the oil stiction effect, valve preload, electromagnetic forces as well as forces due to contact with surrounding parts are considered. The electromagnetic force is assumed to act in the middle of the valve plate. The contact of the reed valve with surrounding parts is modelled based on a penalty-method neglecting friction. The time integration of the equation of motion shown in equation (8) is based on a Newmark algorithm coupled applying a Newton-Raphson iteration and its resulting linear equation system is solved with Cholesky decomposition using sparse matrices. Damping is considered with the so called Rayleigh-damping. Further details concerning the structure analysis can be found in [9].

\[
(a_0 \cdot M + a_1 \cdot D + t^{\Delta t} K^{(i)}) \cdot \Delta u^{(i)} = RHS
\]

\[
RHS = t^{\Delta t} r + M \cdot \left[ a_0 \cdot (t^u - t^{\Delta t} u^{(i)}) + a_2 \cdot t^\dot{u} + a_3 \cdot t^{\ddot{u}} \right]
+ D \cdot \left[ a_4 \cdot (t^u - t^{\Delta t} u^{(i)}) + a_2 \cdot t^\dot{u} + a_3 \cdot t^{\ddot{u}} \right] - t^{\Delta t} f^{(i)} + t^{\Delta t} \dot{f}^{(i)}_{pen}
\]

The parameters \(a_0 - a_5\) from the upper equation can be written as:

\[
\begin{align*}
a_0 &= \frac{1}{\alpha \cdot \Delta t^2}; & a_3 &= \frac{\delta}{\alpha \cdot \Delta t}; & a_2 &= \frac{1}{\alpha \cdot \Delta t}; \\
a_1 &= \frac{1}{2 \cdot \alpha} - 1; & a_4 &= \frac{\delta}{\alpha} - 1; & a_5 &= \frac{\Delta t}{2} \cdot \left( \frac{\delta}{\alpha} - 2 \right)
\end{align*}
\]

3.3. Electromagnetic force

The electrical circuit of the magnetic coil can be compared with a resistor-inductor circuit shown in Figure 5. When a voltage \(u\) is applied to the circuit, the inductor stores a certain amount of energy which takes time. Therefore, the voltage and the current of the inductor cannot change instantaneously. For such circuits, the temporal profile of the current is well known and can be given with:

\[
i(t) = \frac{u}{R} \cdot \left( 1 - e^{-\frac{t}{\tau}} \right)
\]

In equation (10), \(u\) is the voltage applied to the coil and \(\tau = L(x)/R\) is the characteristic time constant with the electrical resistance \(R\) and the inductance \(L(x)\). With this expression of the current, equation (5) can finally be written as:

\[
F_m = -\frac{N^2 \cdot \alpha}{\mu_0 \cdot A_g \cdot R_1^2} \cdot \frac{u^2 \cdot \left( 1 - e^{-\frac{t \cdot L(x)}{R}} \right)^2}{2 \cdot \left( (R_c + R_v + R_1) \cdot R_g + R_4 \cdot (R_c + R_v) \right)^2}
\]

Since equation (11) contains several parameters which cannot be measured or predicted with high accuracy, the reluctance of the leakage flux \(R_1\) as well as the parameter \(\alpha\) are fitted so that the simulated closing of the suction valve meets the measured closing behaviour shown in Figure 2.

![Figure 5. Resistor-inductor circuit.](image-url)
4. Results

To validate the calculated reed valve motion, measurements from [10] are used. Therein, the valve lift measurements for a standard compressor of the same type as it is investigated in the present paper were carried out by using the Laser Doppler vibrometer method. A comparison between the simulated and the measured valve lift can be seen in Figure 6. The agreement between simulations and measurements using the standard suction valve is satisfying and therefore, the models of the reed valves are expected to be well suited for simulations with the adapted suction valve too.

Figure 7 shows the simulated pressure-volume diagram over 360 degree crank angle using the standard suction valve. On the right hand side the low pressure range is enlarged.

![Figure 6. Comparison between simulated and measured valve lift of the suction and discharge valve.](image_url)

![Figure 7. Simulated pressure-volume diagram using the standard suction valve.](image_url)

Results of the simulation using the adapted suction valve and the electromagnetic coil are shown in Figure 8. On the left side the pressure-volume diagram enlarged at low pressures and on the right side a diagram of the valve lift can be seen. For better comparison between the two investigated cases, the simulations using the standard suction valve are drawn in additionally. The reduced preload of the adapted suction valve results besides a slightly earlier opening in a significantly increased maximum valve lift. No significant differences could be seen in the movement of the discharge valve.

![Figure 8. Simulated pressure-volume diagram and suction valve lift using the adapted valve with the electromagnetic coil.](image_url)
Table 4 shows the calculated values of the indicated power consumption, the suction loss, the mass-flow rate and the reversed mass-flow of the suction gas at the end of the suction stroke. The reversed mass flow is given in percent related to the mass-flow rate.

**Table 4. Comparison of calculated values between standard and adapted suction valve.**

| valve         | indicated power [W] | suction loss [W] | mass-flow rate [g/s] | reversed mass-flow [%] | @ CA [°C] |
|--------------|----------------------|------------------|----------------------|------------------------|-----------|
| standard     | 40.76                | 0.84             | 0.283                | -0.26                  | 188.1 – 201.4 |
| adapted      | 40.49                | 0.60             | 0.282                | -1.60                  | 192.8 – 202.5 |

To verify the results of the numerical simulations, several measurements are carried out at steady-state conditions in a standardized calorimeter. The calorimeter was running at operating conditions according to the ASHRAE cycle with an ambient temperature of 32 °C, an evaporation temperature of -23.3 °C and a condensation temperature of 55 °C. Additionally to this operating condition, further measurements changing the condensation temperature to 45 °C were carried out. To get measurements which can be compared as good as possible, the same compressor was used for all measurements (only with different suction valves).

The electromagnetic coil is controlled with a full bridge circuit using two control inputs to demagnetize the coil. Voltage is applied to the coil approximately 1 ms before the piston reaches its bottom dead centre. After 3 ms the circuit is opened again. Results of previous investigations show, that the electric power consumption of the electromagnetic coil, operated at 8 V and a duty cycle ratio of 15 %, is about 2.1 W. This leads to increased temperatures inside the hermetic shell of approximately 2.5 K. Based on measurements investigating cooled compressors, a decreasing of about 0.25 % of the COP must be expected due to the waste heat of the electromagnetic coil.

Table 5 shows the differences between the reference measurements and the measurements with the adapted valve. The values stated in this table represent the percentage changes which are based on the reference measurements with the standard suction valve. For both ASHRAE cycles, a decreasing of the power consumption of approximately 0.7 % can be seen which results from the reduction of the suction losses. As it can be seen in the results of the numerical simulation, the adapted suction valve leads to a completely different dynamic behaviour during the suction stroke. The decreased cooling capacity of the measurement with a condensation temperature of 55 °C, can be explained due to the reversed mass-flow through the suction port because the suction valve could not be closed in time. When changing the condensation temperature to 45 °C, it seems that the reversed mass-flow has decreased.

**Table 5. Effect of the adapted suction valve in percent related to the reference measurements.**

| ASHRAE cycle     | COP   | cooling capacity | power consumption |
|------------------|-------|------------------|-------------------|
| -23 °C / 45 °C   | +1.72 | +0.98            | -0.73             |
| -23 °C / 55 °C   | +0.39 | -0.32            | -0.71             |

5. Conclusion

The reduction of the suction losses using a negative prestressed suction valve in conjunction with an electromagnetic coil was investigated. Numerical simulations at operating conditions according to the ASHRAE cycle (-23 °C/55 °C) showed that the indicated power consumption decreases by 0.68 %. The reversed mass-flow rate through the suction port increases from 0.26 to 1.6 % and thus the overall mass-flow rate decreases by 0.31 %. Measurements confirmed the results from the simulation (decrease of the power consumption by 0.71 %; decrease of the cooling capacity by 0.32 %) and showed an increased COP of 0.39 %. Additionally, a second operating condition was investigated by measurements which showed an increased COP of 1.72 %.

At this point it should be mentioned, that the power consumption of the electromagnetic coil should normally also be taken into consideration when comparing the COP, but here the effects of the adapted suction valve were in the centre of attention.
Summarizing it can be stated, that the adapted suction valve with the electromagnetic coil shows a general improvement of the performance of the compressor. Nevertheless, the effect depends on the operating conditions and therefore, the suction valve should be completely redesigned.

Nomenclature
Roman symbols

| Symbol | Description                   |
|--------|-------------------------------|
| $A$    | area                          |
| $c$    | stiffness                     |
| $c_t$  | capillary coefficient oil stiction |
| $c_v$  | viscous coefficient oil stiction |
| $d$    | damping coefficient           |
| $D$    | damping matrix                |
| $f$    | vector of nodal point forces  |
| $F_{\text{gas}}$ | resulting gas force         |
| $F_{\text{m}}$ | electromagnetic force     |
| $f_{\text{pen}}$ | force vector due to contact |
| $F_{\text{pre}}$ | preload                   |

| Symbol | Description                   |
|--------|-------------------------------|
| $R$    | resistance                    |
| $\mathcal{R}$ | reluctance               |
| $t$    | time                          |
| $u$    | voltage                       |
| $u$    | displacement vector           |
| $\mathbf{u}$ | velocity vector          |
| $\mathbf{\ddot{u}}$ | acceleration vector       |
| $x$    | gas gap, displacement        |
| $\dot{x}$ | velocity                   |
| $\ddot{x}$ | acceleration                |

Greek symbols

| Symbol | Description                   |
|--------|-------------------------------|
| $i$    | current                       |
| $\alpha$ | correction factor, Newmark param. |
| $\mathbf{F}$ | force vector due to contact |
| $\mathbf{r}$ | vector of externally applied loads |
| $\mathbf{F}$ | force vector due to contact |
| $\mathbf{F}$ | force vector due to contact |
| $\mathbf{F}$ | force vector due to contact |
| $\mathbf{F}$ | force vector due to contact |

| Symbol | Description                   |
|--------|-------------------------------|
| $K$    | stiffness matrix              |
| $l$    | length                        |
| $L$    | inductance                    |
| $m^{*}$ | equivalent mass               |
| $M$    | mass matrix                   |
| $N$    | number of turns               |
| $\mathbf{r}$ | vector of externally applied loads |
| $\delta$ | Newmark parameter             |
| $\Theta$ | magnetomotive force          |
| $\mu_0$ | vacuum permeability          |
| $\mu_r$ | relative permeability        |
| $\tau$ | characteristic time constant  |
| $\Phi$ | magnetic flux                 |
| $\psi$ | flux linkage                  |

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