System and Component Design of Directly Driven Reciprocating Compressors with Modelica

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Abstract

Directly driven reciprocating compressors have the potential to be used at large scale for refrigerant compression in domestic refrigerators and freezers and to improve the energy efficiency of these cooling devices considerably. Compared to their well-established conventional counterparts with rotational motor and rotation/translation transducer they offer advantages such as (I) higher efficiency due to the absence of gear friction losses, (II) broad-range modulation of the compressor's mass flow rate, e.g. by control of the piston stroke and (III) a simpler mechanical structure.

With typical strokes of approximately 15…20 *mm* and piston peak forces up to approximately 150 *N*, electrodynamic and electromagnetic linear motors are predestined to be used as direct drives for those compressors when operated under resonance conditions with one or more springs. Appropriate motor and system configurations are currently developed and evaluated at Dresden University of Technology. Highly nonlinear compression forces, strong interactions between multiple physical domains and a large number of design parameters to be chosen properly are challenges during system and component design that call for the usage of appropriate models and efficient simulation approaches within the design process. With its multi-domain paradigm, Modelica is excellently suited for the model-based design of directly driven refrigerant compressors, their power supply and control prior to detailed design, manufacture and test of prototypes.

This paper is intended (I) to present an overview of the above compressor technology, (II) to enlighten the beneficial use of Modelica for the design of heterogenous systems such as the above compressors and (III) to illustrate the design of electro-magneto-mechanical converters by means of lumped magnetic network models and the Modelica Magnetic library.

Keywords: linear compressor; system design; magnetic network; electrodynamic actuator

1 Directly driven reciprocating compressors for refrigeration

1.1 Principle structure and operation

The piston of a directly driven reciprocating compressor is connected to the armature of its driving linear motor without any additional motion transducer or gear element (Fig. 1). This results in greatly reduced friction compared to conventional reciprocating compressors with rotational motor and rotation/translation transducer and hence in a higher overall efficiency of the compressor. In conventional compressors of domestic refrigerators, the motion transducer accounts for approximately 60 % of the total friction losses of the compressor.

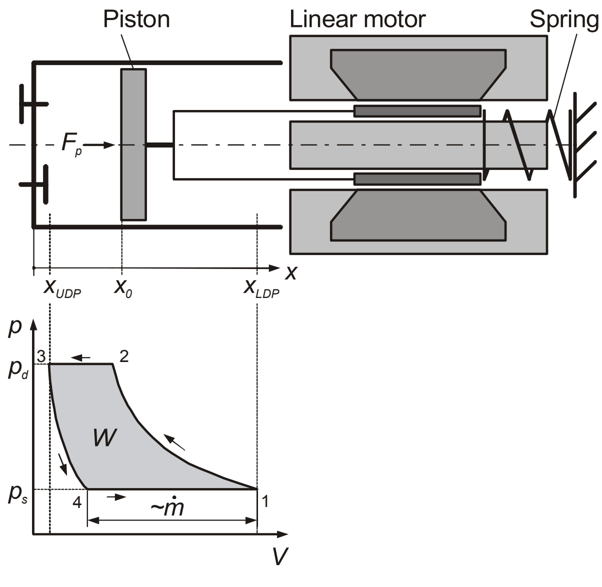

Fig. 1 Principle structure of a directly driven reciprocating compressor (not scaled) and idealised indicator diagram $p(V)$

For reasons of efficiency and size of the linear motor, directly driven reciprocating compressors are normally operated at or close to their mechanical resonance frequency. The mass of the armature/piston assembly in conjunction with at least one helical or diaphragm spring forms a mechanical resonant system. If diaphragm springs are used, they also guide the armature in radial direction. It must be noted that the compressor load, i.e. the suction pressure p_s and discharge pressure p_d for a particular piston stroke $h = x_{LDP} - x_{UDP}$ influences the resulting net resonance frequency due to its properties of a nonlinear gas spring and hence the overall efficiency at operating frequency. The influence of this gas spring must be taken into account during system design (section 2).

As for conventional reciprocating compressors, an idealised compression cycle of one stage of a compressor consists of four phases (numbers according to idealised indicator diagram of Fig. 1):

- $1\rightarrow2$ isentropic compression,
- $2\rightarrow3$ isobaric discharge,
- $3\rightarrow4$ isentropic re-expansion of the residual gas,
- $4\rightarrow1$ isobaric suction of gas for the next cycle.

In reality, the compression process differs from the above idealised thermodynamic changes of states due to (I) complex heat exchange between the gas and the cylinder assembly, (II) pressure drop and pressure pulsations in the valve assemblies and pipes and (III) deviations of the properties of refrigerants (real gas behaviour) from those of ideal gas as assumed in the above listing of compression cycle phases [1]. The area within the closed pressure-volume curve $p(V)$ of Fig. 1 represents the work per cycle that is exerted on the gaseous refrigerant.

Whereas the stroke of a conventional compressor is constant due to its crankshaft mechanism, the stroke of a directly driven reciprocating compressor can be varied. Thus it is possible to adapt the mass flow rate $\dot{m}$ of directly driven reciprocating compressors in refrigerators to varying thermal operation conditions of the cooling device: $\dot{m}$ is directly proportional to the suction length $l_{suc} = x_{LDP} - x_4$.

1.2 Usage for refrigeration

A successful attempt to utilise an oscillating electromagnetic motor for the direct propulsion of a reciprocating compressor's piston is already reported for the year 1908 [2]. Those attempts continued throughout the last century and resulted in a number of different designs of directly driven reciprocating compressors for different applications [2]. While some of these compressors are experimental prototypes up to now, others are already commercially utilised, e.g. in small air compressors, in small mobile refrigerators for medical and recreational use [3] and in Stirling and pulse tube cryo-coolers.

For refrigerant compressors of domestic refrigerators and freezers however, problems with collisions between the free moving piston and the cylinder head and valve assemblies at varying suction and discharge pressures prevented from the broad utilisation of this concept in the past (beside durability of springs used for resonance operation). To a great extent, these collision problems were due to premature controller realisations. However, situation changed during the last years due to the broad availability of low-cost microcontrollers and power electronic components. This enables now for the realisation of powerful yet cost-effective nonlinear controllers for stroke control and hence for save compressor operation at varying compression loads. For that reason, increased research and design efforts have been made during the last years for the industrial utilisation of directly driven refrigerant compressors. As a result, the Korean manufacturer LG Electronics, Inc. offers domestic refrigerators with directly driven compressors sized for the Korean market since 2004 [4] [5]. The linear motor of these compressors is based on a design of the US company Sunpower, Inc. [6].

1.3 Research objectives and sample application

The current research on directly driven reciprocating compressors for refrigerators at Dresden University of Technology focuses on:

1. analysis of the state of the art of motors for directly driven reciprocating refrigerant compressors and their control,
2. development and evaluation of principle motor designs for a chosen sample application,
3. conceptual control design for compressor operation under varying thermal loads (i.e. cooling powers) for the sample application,
4. establishment of a design methodology for directly driven reciprocating compressors based on modelling and simulation,
5. detailed design, prototype manufacture and test of the linear motor found as optimum under 2. for the sample application.

This sample application is specified as follows:

- domestic one-compartment refrigerator,
- refrigerant: R600 (n-Butane),
- continuous compressor operation, i.e. no intermittent operation for temperature control as in conventional refrigerators,

- specified cooling power for minimum/ nominal/ maximum thermal load: $\dot{Q}_0 = 5/25/50\,W$ with accordingly specified evaporator and condenser temperatures and refrigerant superheat,
- one-stage, one-sided compressor design, i.e. only one compression chamber at one side of the piston,
- compressor operation at mains frequency, i.e. at 50 *Hz* or 60 *Hz*,
- adaption to varying cooling power requirements by means of control of the piston's suction length (due to projected continuous operation at constant frequency).

2 Simulation-based system design

2.1 Compressor specification

One of the most important tasks during initial system design is the specification of the compressor's main parameters stroke and compression force versus position from given specifications of the refrigerator's vapour compression process to be realised. This is done by calculation of refrigerant mass flow rates, enthalpy differences and compression work per cycle for specified thermal load scenarios by means of the pressure-enthalpy diagram of the used refrigerant. For the sample design process described in this paper, refrigerant data from an external software package [7] has been used so far for this task. In the future, however, Modelica [8] libraries that are related to thermo-dynamic systems and refrigerant properties such as the commercial AirConditioning Library™ [9] and the free Modelica.Media library can be used advantageously, assumed that these libraries are available and contain data for the required refrigerants, respectively. Doing so would incorporate yet another physical domain and model components from its respective Modelica libraries into the resulting system model and would make model-based design of directly driven reciprocating compressors for refrigerators with Modelica even more seamless.

2.2 Dynamic system simulation

Currently, both different principle operating and control concepts and principle motor designs are established and evaluated and the interactions between the major subsystems controller, power supply, linear drive and compression load are designed for the chosen sample application. The elaborated design method is strongly based on dynamic simulations and analysis of the results at steady-state oscillations. Compared to modelling and design methods based on mechanical and electrical impedances (e.g. [10]), the approach based on transient simulation enables for consideration of all important nonlinear effects in the $p(V)$ relationship and the derived piston force versus position relationship $F_P(x)$. Whereas description of the compression process with an equivalent, stroke dependent spring constant $c_{gas}(h)$ and an equivalent damping coefficient $d_{gas}(h)$ might work for double-sided compressors with one compression chamber at either side of the piston, this approach can not be used for modelling of one-sided compressors as is the case for the chosen sample application. This is due to the shift of the piston's average position x_{avg} away from its rest position x_0 in presence of unbalanced, one-sided compression forces (Fig. 4). This shift of average piston position in turn influences the required stroke that is necessary to realise a desired suction length. Thus it must be considered carefully during conceptual design.

Fig. 2 shows the graphical representation of a Modelica model that is used for analysis of fundamental system behaviour and for prediction and evaluation of motor performance to be expected for particular motor designs.

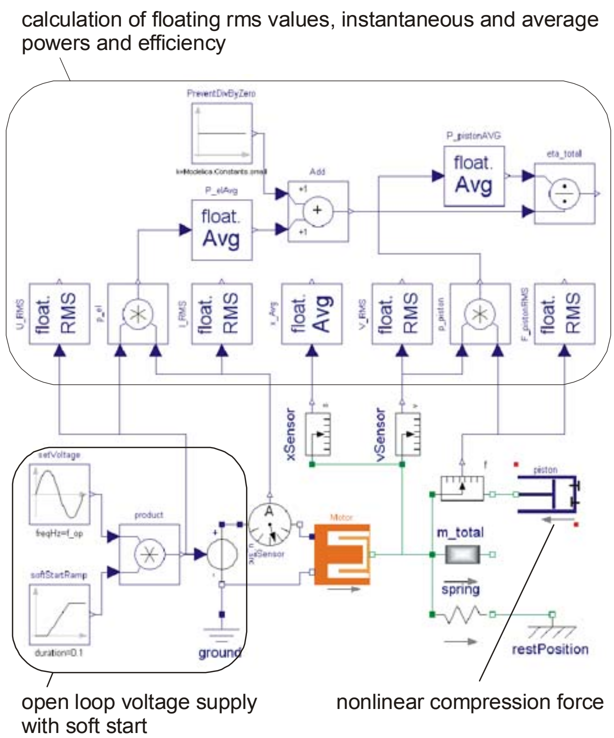

Fig. 2 Graphical representation of the Modelica system model that is used for analysis of fundamental system and motor behaviour

In this model, a sinusoidal voltage is fed to the motor model to be evaluated. This voltage will evoke a current i through the motor that causes a motor force F_M to be developed. This force acts against the inertia of the armature/piston assembly's mass m, against the spring(s) used for resonance operation (spring stiffness c) and against the compression force that acts upon the piston F_P with the above mentioned four idealised phases per cycle according to the force equilibrium

$$0 = F_M(i) + m\ddot{x} + c(x - x_0) + F_P(x, x_0, p_s, p_d, n) \quad (1)$$

In the above equation, x denotes the piston position, x_0 its rest position in non-energised state (no pressure difference acting on piston, no supply voltage), p_s and p_d the suction and discharge pressure, respectively, and n the isentropic exponent used for modelling of idealised compression and re-expansion forces according to the isentropic condition $px^n = const.$

For initial analyses of system behaviour, a simple model of a coarsely dimensioned electrodynamic linear motor was used in the system model depicted in Fig. 2. This motor is of moving coil type. The electro-magneto-mechanical energy conversion process is based on *Lorentz* forces and is described by

$$F_M = B\,l\,i \quad (2a)$$

$$u_i = B\,l\,\dot{x} \quad (2b)$$

with F_M being the motor force, u_i the induced counter electromotive force (back-emf), B the flux density imposed to the coil, l the total length of the coil wire and i and $\dot{x}$ current and armature velocity, respectively. In the graphical Modelica representation of the above motor equations shown in Fig. 3, B and l are combined into the motor constant c_m. Besides its back-emf, the electrical subsystem of the motor is made up of its coil resistance R and its inductance L.

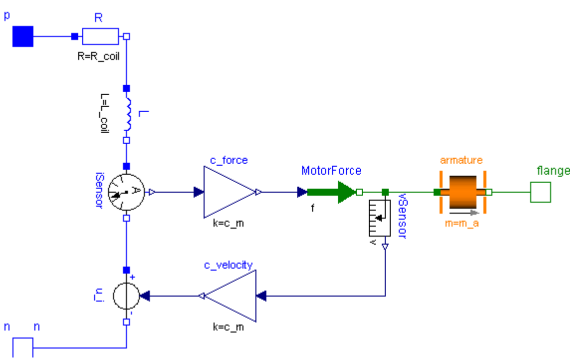

Fig. 3 Graphical Modelica representation of the simple model of a electrodynamic linear motor

It must be pointed out that the above initially used model of a moving coil type linear motor is not the optimum motor design with respect to efficiency, related power and other criteria to be expected. Neither is the used model a detailed one: Assumption of a constant motor constant c_m throughout the complete stroke range is a rather coarse approximation. Also, the motor's magnetic subsystem is not incorporated into the above model as must be done for design of a motor's magnetic components. However, the above simple model is well suited for initial analyses due to its simplicity and linearity. More detailed motor models for additional principle motor designs will be described in section 3.

Using *Kirchhoff*'s voltage law, the differential equation of the electrodynamic motor's electrical subsystem can be written as

$$u_{src} = L\dot{i} + R\,i + c_m\dot{x} \quad (3)$$

with u_{src} being the supply voltage. Equations 1 and 3 make up the system of nonlinear ordinary differential equations that describes the compressor's dynamic behaviour. Transient simulation of this system will reveal its behaviour with respect to time and enables for derivation of (mostly integral) quantities appropriate for performance evaluation of directly driven reciprocating compressors and their linear motors. For the herewith presented project, *Dymola* is used as simulation environment [11].

Fig. 4 shows exemplary results of a dynamic simulation for a particular motor design and compression load. These results have been obtained with the system model of Fig. 2. A slow initial increase of the supply voltage u_{src} prevents from strong overshoot of the piston oscillation at start-up and hence from piston collisions with the cylinder head and valve assemblies. The fact that the motor voltage u_{src}, the motor's internal back-emf u_i (induced at armature motion) and the voltage drop across the coil resistance $u_R = R{\cdot}i$ are nearly in phase at steady-state oscillation indicates that the piston oscillates close to its resonance frequency for this particular compression load.

The diagram in the middle of Fig. 4 shows instantaneous and floating root mean square (rms) values of the compression force F_P (index P for piston) and the motor force F_M, respectively. From an energetic point of view it is interesting to note that the motor force $F_{M\,rms}$ is smaller than the piston force $F_{P\,rms}$ although both motor armature and piston move with the same velocity due to its direct coupling. A detailed investigation shows that this is due to the fact that at piston force maximum (discharge phase) the piston velocity becomes zero (upper dead point position). Thinking of the system as being linear with harmonic oscillations would result in a phase angle between piston force and velocity close to 90° and hence in reactive components of the mechanical load. The analysis of both compression power and electrical input power with the presented nonlinear model though shows that the differences in the rms values of motor force and piston force are in accordance with proper physical behaviour (see efficiency calculation, equation 4).

From the bottom diagram of Fig. 4 can be seen that build-up of the piston oscillation from its rest position x_0 in presence of one-sided, i.e. unbalanced compression forces results in a shift of the piston's average position x_{avg} away from this rest position as discussed earlier. In the diagram, the resulting suction length l_{suc} for each cycle is shown, too.

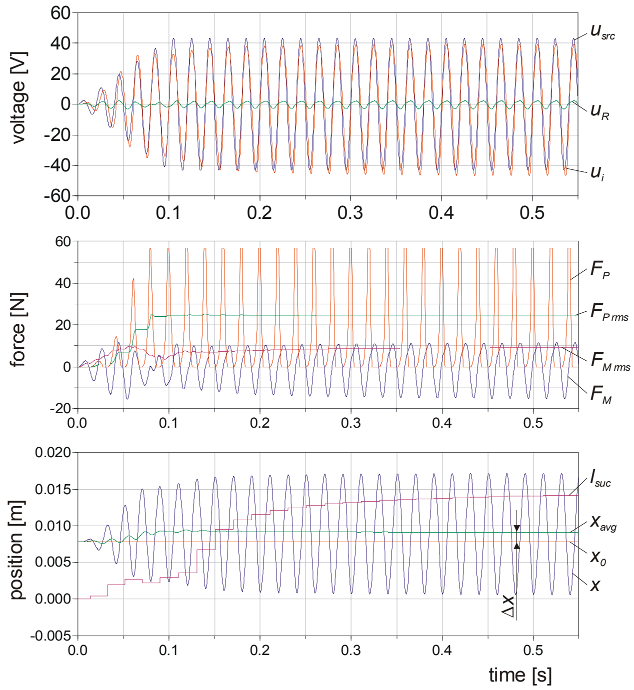

Fig. 4 Selected simulated voltages, forces and positions of an exemplary motor design and compression load

In Fig. 5, the piston force versus position $F_P(x)$ for the simulated compressor start-up shown in Fig. 4 is depicted. Note that the piston's upper and lower dead point positions and suction lengths are different for each cycle prior to achievement of steady-state oscillations.

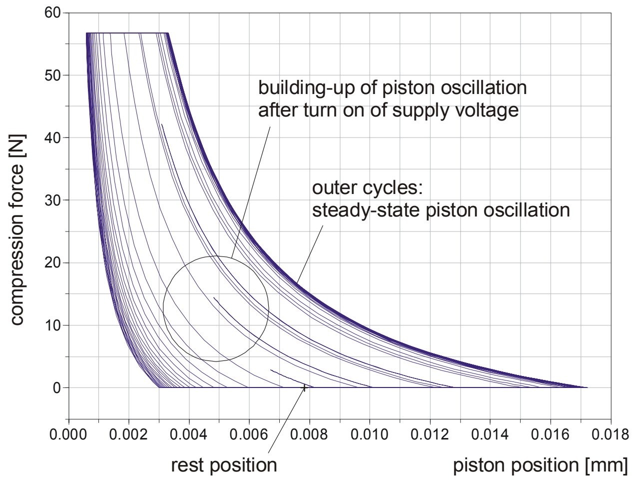

Fig. 5 Simulated piston force vs. position for the compressor start-up shown in Fig. 4

2.3 Performance evaluation

The most important aspects of system behaviour and motor performance that are analysed with the above system model are:

- effect of varying compression loads (suction and discharge pressure, stroke) on net resonance frequency and mechanical and electrical phase relations,
- influence of electrical subsystem on net resonance frequency,
- overall efficiency,
- electrical power factor,
- quality factor of motor (section 3.3).

The above mentioned dependency of the system's mechanical resonance frequency on the motor's electrical parameters coil inductance L and resistance R can be shown with an analysis of the corresponding, but simplified linear 3rd order system in the frequency domain, too. However, the important impact of nonlinear compression forces on system behaviour can not be considered with such an analysis.

The Overall efficiency η of the compressor can conveniently be calculated from floating average values per cycle of compression power and electrical input power:

$$\eta = \frac{P_{c\,avg}}{P_{el\,avg}} = \frac{\frac{1}{T}\int_{t-T}^{t} F_P(\tau)v(\tau)d\tau}{\frac{1}{T}\int_{t-T}^{t} i(\tau)u(\tau)d\tau} \qquad (4)$$

with T being the period of one cycle, F_P the compression force acting on the piston, v piston velocity and i and u motor current and voltage, respectively. At present, only ohmic losses of the coil are considered for calculation of efficiencies. Hence, the efficiency values shown below are too high compared to reality. Additional losses such as mechanical friction, friction due to gas leakage and ferromagnetic losses in the motor are not yet considered in the above model. This was done intentionally in order to keep the model behaviour straightforward during initial analyses. However, model components for consideration of the above loss mechanisms are currently added to the developed system and motor models and their suitability for system and motor design is tested.

Characteristic values for evaluation of the principle motor designs currently under development are briefly discussed in section 3.3.

2.4 Control concepts

Modulation of the compressor capacity, i.e. adaption of the refrigerant mass flow rate to the requested cooling power of the refrigerator, is possible by one or more of the following means:

- variation of suction length and associated stroke,
- variation of operating frequency and
- intermittent operation with varying duty cycle (as for conventional compressors with crankshaft mechanism and constant speed).

Since continuous operation at constant frequency is projected for the chosen sample application, only suction length control is applicable. Suction length can not directly be measured without large effort, therefore measures are taken to estimate the suction length of each cycle from the piston stroke and to realise a stroke control instead.

By now, different control concepts and appropriate controllers have been developed, implemented in Modelica and successfully tested with dynamic simulations at system level. The following control concepts are currently under closer investigation:

- stroke control only without control of top dead center position (guarantees optimum motor efficiency since no DC bias voltage is superimposed on the controlled AC supply voltage),
- combined stroke control and control of top dead center position (guarantees minimum clearance volume and related thermo-dynamic losses but increases ohmic losses in the motor due to a DC bias voltage to be superimposed on the controlled AC supply voltage),
- various additional control concepts that do not require measured or estimated information of the piston stroke, but information on top dead center position only (e.g. obtained from a proximity sensor).

The performance of the above control concepts with respect to overall and motor efficiency, electrical power factor and phase relations is currently investigated with the developed system models. The stroke controllers are mostly of proportional and integral type (PI). For the test of principle voltage supply concepts, different models have been developed and are currently tested (e.g. thyristor and triac control for simple and cost-effective mains operation).

2.5 Simulation-based design

The above mentioned controller models for stroke control are not only used for simulation and test of concepts for capacity modulation of the compressor, but also for simulation-based selection of fundamental design parameters. This is because stable piston oscillations with constant stroke can be guaranteed during dynamic simulation with closed loop stroke control, but not during dynamic simulation of large strokes with open loop voltage supply only due to the nonlinear compression forces.

A good example for the simulation-based selection of a design parameter is the stiffness of the spring(s) needed for resonant piston oscillation (Fig. 6).

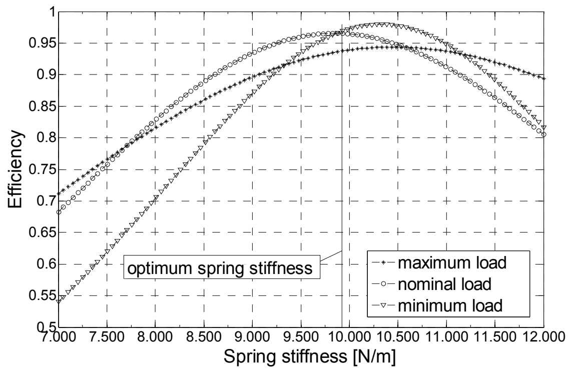

Fig. 6 Simulation-based selection of optimum spring stiffness obtained with parameter sweeps for this design parameter (only ohmic losses in the motor coil considered for efficiency calculation, see text)

The optimal required spring stiffness c_{spring} for a particular design can not be obtained otherwise with the same certainty. This is because of the properties of the compressed gas that resemble that of a strongly nonlinear gas spring (section 2.2). Its stroke-, position-, and pressure-dependent stiffness c_{gas} adds to the mechanical spring stiffness and affects the net resonance frequency f_{res} of the mechanical subsystem with the moving mass m:

$$f_{res} = \frac{1}{2\pi}\sqrt{\frac{c_{spring} + c_{gas}}{m}} \tag{5}$$

Each efficiency value shown in Fig. 8 was obtained from a steady-state efficiency calculation at the end of respective dynamic simulations during parameter sweeps for the spring stiffness c_{spring}. Stroke control was implemented in the Modelica model during these simulations, so the desired stroke length for each particular compression load was assured.

Notice that the calculated efficiency values are too high compared to reality since only ohmic losses in the motor coil were considered during these simulations as described in section 2.3.

3 Magnetic converter design

3.1 Design Methodology

An appropriate approach for the motor design for directly driven reciprocating compressors can roughly be outlined as follows:

1. Specification of mechanical motor parameters (stroke, rms force, average power) from compression data (idealised indicator diagram, compression work per cycle) for specified compression load scenarios (nominal and maximum load),
2. Specification of additional motor parameters, e.g. max. ambient temperature within compressor capsule, volume, refrigerator supply voltage and frequency,
3. Selection of promising principle motor designs, coarse dimensioning of these principle solutions (geometry and material of magnetic components, coil and wire data) by means of lumped magnetic networks,
4. evaluation of the found principle solutions by means of characteristic factors (section 3.3) and dynamic simulation at system level (sections 2.2, 2.3), selection of the optimum solution,
5. fine dimensioning of the selected motor's magnetic design with Finite Element Analysis (FEA),
6. detailed design, manufacture and test of prototypes.

Whereas the use of FEA is valuable for optimisation of a particular motor's magnetic design, it is in most cases not suited for efficient coarse dimensioning of principle motor designs and for usage in extensive dynamic simulations due to the high time effort for model pre-processing and due to its high computational effort, respectively. Instead, lumped magnetic networks should be used for initial coarse designs. For the motor design within this project, the Modelica Magnetic library developed by the author is used for the implementation of magnetic network models of the motors [12]. Through its usage within this design project, the library is extended, e.g. with models for electrodynamic linear actuators, and its model components are enhanced. It is planned to submit a revised version of this library as soon as possible.

3.2 Principle motor designs

Different principle motor designs are already known for directly driven reciprocating compressors. They are a subset of the broad spectrum of different types and designs of electrodynamic and electromagnetic linear drives. In general, numerous classification criteria are popular for classification of electro-magneto-mechanical converters. For example, based on the underlying physical principle one can distinguish between:

- electrodynamic linear drives based on *Lorentz* forces [13],
- electromagnetic or reluctance drives based on surface forces between ferromagnetic component(s) and adjacent air gap(s) [14] and
- linear drives that utilise both of the above working principles for force generation.

Depending on the moving component, classification between the following motor categories is popular:

- moving coil (mostly for electrodynamic drives),
- moving magnet (electrodynamic and/or reluctance forces) and
- moving iron (reluctance drives).

Location of stator and armature components is a classification criterion, too.

At present, known motor designs are analysed and additional principle solutions (a promising subset of motors from the above categories) are developed. All found solutions are evaluated. The required coarse dimensioning for the sample application is mostly done with the above mentioned lumped magnetic network models. Despite this efficient design approach, coarse dimensioning of motors based on different working principles is a rather time-consuming process. Nevertheless, it is a prerequisite for optimum motor design for a particular application.

Although the above mentioned evaluation is not yet completed, it currently appears that the motor concept realised by Sunpower, Inc. [6] and already utilised in a commercial domestic refrigerator of LG Electronics, Inc. [4] [5] is an advantageous one. The principle structure of this motor is shown in Fig. 7 together with the magnetic field lines of the permanent magnetic flux in neutral position. Latter were obtained from FEA for the shown exemplary design. Operation of this motor is based on the superposition of permanent magnetic flux and electromagnetic coil flux in the respective air gap sections and results in a constant thrust over the complete stroke range. This thrust is proportional to the current. Hence motor operation can be described with equations 2a and 2b as for a purely electrodynamic linear motor. Based on that perception, the creators of this motor promote the treatment of the permanent magnet's magnetomotive force as equivalent currents at the two end planes of the permanent magnet's hollow cylinder [15]. Whereas this design approach enables for easy calculation of motor forces, it does not account for the permanent magnetic flux through the stator at off-center armature positions and for related saturation effects in the ferromagnetic stator components properly.

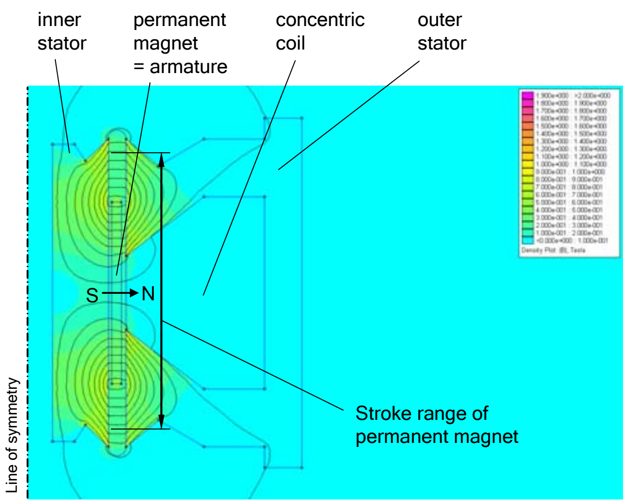

Fig. 7 Structure of a moving magnet motor based on [6] and permanent magnetic flux without stator current obtained by FEA

Unlike with the above modelling approach, the stator flux due to the permanent magnet at off-center armature positions is treated properly with the lumped magnetic network model shown in Fig. 8. It was created with model components from the Modelica Magnetic library [12]. The depicted reluctance elements represent the respective air gap and permanent magnet regions at either of the two pole regions. The motor force F_M is developed according to

$$F_M = -\frac{1}{2}\sum_{i=1}^{n_{linear}} \Phi_i^2 \frac{dR_{m\,i}}{dx} \qquad (6)$$

with n_{linear} denoting the number of linear reluctance elements, Φ_i the magnetic flux through each respective reluctance and $R_{m\,i}/dx$ the derivative of each reluctance with respect to armature position x [14].

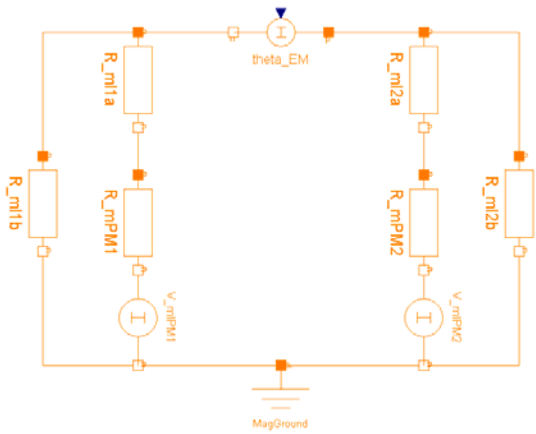

Fig. 8 Simple lumped magnetic network model of the moving magnet motor created with elements from the Magnetic library [12]

In the linear model of Fig. 8, nonlinear ferromagnetic reluctance elements of inner and outer stator components are omitted for reasons of initial simplicity. Also, the depicted model is only a stationary one. It was used for initial analyses of magnetic flux distributions at different armature positions and motor currents. The electro-magnetic energy conversion that is important for dynamic operation is not considered in this model. However, both nonlinear ferromagnetic reluctance elements and a dynamic model of the stator coil that describes the coupling between the motor's electrical and its magnetic domain can easily be added to the motor model from standard components of the Modelica Magnetic library (see examples in this library). Latter enhancement simply requires replacement of the coil's stationary magnetomotive force θ_{EM} of Fig. 8 with a dynamic model of a coil. The resulting dynamic motor model can then directly be used for dynamic system simulation, e.g. with the model shown in Fig. 2. This has already been done for magnetic network models of different moving coil motor designs and will be done for the additional motor designs currently under development, too.

3.3 Motor evaluation

For comparison of the found principle motor designs, the performance to be expected is currently evaluated by means of the developed motor models. Quantitative factors for this motor evaluation are:

- motor efficiency for specified minimum/ nominal/ maximum compression load,
- quality factor $E = F_M^2 / P_{el}$ as important measure for the sensitivity of electro-magneto-mechanical converters (F_M motor force, P_{el} electrical input power) [16],
- mass of selected components (permanent magnet, copper, iron) as measure for the material cost to be expected,
- related power, i.e. power per volume and
- motor constant c_m for calculation of thrust from motor current.

Additional qualitative criteria like the presence of parasitic radial magnetic forces or durability of the design (e.g. important for flexible wires to moving coils) shall be considered in this evaluation, too. This evaluation will be based on guidelines provided in [17].

4 Summary and outlook

The proper design of directly driven reciprocating compressors and their respective electrodynamic and electromagnetic motors is challenging due to nonlinear compression forces, strong interactions between the different subsystems and complex phase relations and resonance phenomena at varying compression loads. Modelica is excellently suited for consistent modelling of those heterogenous systems and the developed models can advantageously be used for simulation-based system and component design.

Development of compressor operation and control principles as well as of appropriate motor designs is currently done for a chosen sample application, namely for a refrigerant compressor of a domestic refrigerator. The elaborated design approach based on dynamic system simulation and analysis of integral quantities at steady-state piston oscillations proved to be well-suited for the rather complex system design. Although the time effort for initial model development is relatively high, the results obtained from dynamic simulations are extremely valuable for the search of appropriate system and component configurations and for understanding of the complex dynamic behaviour of directly driven reciprocating compressors. With the developed models, the ongoing simulation-based evaluation of different principle motor designs for the chosen sample application will be completed prior to the intended test of first experimental motor prototypes.

Evaluation of the suitability of different motor principles for the chosen sample application requires the coarse dimensioning of respective motors. The usage of lumped magnetic network models built with the Modelica Magnetic library is an efficient means for this rather laborious design task.

It is worth to note that the developed model-based design approach for directly driven reciprocating compressors is not restricted to refrigerant compressors but can be adapted to different fields of applications, too.

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