Journal of Energy Storage xxx (2017) xxx–xxx

Contents lists available at ScienceDirect

Journal of Energy Storage

journal homepage: www.elsevier.com/locate/est

Heat transfer losses in reciprocating compressors with valve actuation for energy storage applications

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ARTICLE INFO

Article history:
Received 15 October 2016
Received in revised form 20 May 2017
Accepted 24 July 2017
Available online xxx

Keywords:
Exergetic loss
Irreversible heat transfer
Gas spring
Reciprocating compressor

ABSTRACT

Understanding the exergy losses stemming from heat transfer in compressors and expanders is important for many energy storage applications such as compressed air and pumped thermal storage. In order to obtain a better understanding of these losses, CFD simulations were performed for simple gas springs, for a gas spring with an internal grid to mimic valve flow, and for a reciprocating compressor with functioning inlet and outlet valves. The wall heat exchanges for these three cases were examined and compared. The model adopted has previously been validated for a simple gas spring using experimental data from literature. For the gas spring with an internal grid it was found that increased mixing leads to higher heat-transfer-induced hysteresis losses and (at high piston speeds) to a significant pressure loss. These two types of loss can be distinguished by undertaking adiabatic-wall calculations. For a compressor (i.e., with valve flows) heat transfer over the cycle depends very much on valve timing. For example, at 1500 rpm, when the delivery valve is opened at 7 bar the heat transfer coefficient for the initial stages of compression is similar to that for a simple gas spring, whereas for the same speed at 6 bar it is more than doubled.

1. Introduction

To achieve a reliable and sustainable energy supply the percentage of electrical energy from renewable sources will increase strongly within the next years. Since many of these sources (particularly wind and solar energy) do not provide continuous supplies and since the demand for power varies over the day and over the year, electricity storage is seen as increasingly important for a sustainable energy system.

Aside from their common use in internal combustion engines, reciprocating compressors and expanders have a wide range of applications in energy storage devices and energy systems in general. They are potentially important, for instance, for compressed air energy storage (CAES), and pumped thermal energy storage (PTES), and they may also be used in combined heat and power (CHP) [1], heat pumps [2] and Stirling engines for solar applications [3]. In the case of PTES (which is the focus of the current study), previous work has shown that compression and expansion losses have a major impact on the overall (round-trip) efficiency, essentially because the complete charge-discharge process involves two compressions and two expansions [4].

The main sources of loss in reciprocating compressors and expanders are likely to be valve pressure losses and heat transfer irreversibility. Whereas pressure losses may be reduced by maximizing the valve open area and optimising the valve timing, trends for heat-transfer-related loss are less obvious. To better understand heat transfer effects, various researchers have examined the processes occurring in a gas spring [5–7] – i.e., a reciprocating piston within a cylinder, but without any valves. This enables examination of heat transfer loss independently of losses incurred by the valve flows, but in a real compressor the behaviour will be considerably different since mass is exchanged during every cycle, and the incoming flow generates turbulence, thereby affecting rates of heat transfer. The incoming and outgoing gas also transports energy to and from the cylinder.

This paper examines the heat transfer losses in a reciprocating device by means of CFD simulations. In the first step a gas spring without any valves was studied. Simulations were carried out for a wide range of piston speeds and for two geometries with two different gases (air and helium) and these were used for the validation of the model. Helium was used since experimental data for the validation of the model is available from the literature [7,5]

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http://dx.doi.org/10.1016/j.est.2017.07.024
2352-152X/

and air because an experimental setup using air is currently under development, as described by Mathie et al. [6]. In a second step, mixing inside the cylinder was increased by inserting a grid. This was intended to mimic the turbulence generated by valve flows. These simulations are described in greater detail elsewhere [8] but are summarised here for completeness. In the last step the CFD model was adapted to include valve flows so that differences between heat transfer for a gas spring and a real compressor with mass through-flow could be examined.

The simulations with valves require substantial computing time to obtain converged solutions and only three operating conditions have so far been computed. Although this is not sufficient to give a full picture of exergetic losses (this being the ultimate aim), the results obtained highlight some important qualitative differences between gas springs and real compressors (or expanders) and provide valuable information for the design of on-going simulations.

2. Computational method and model validation

Simulations were performed using the 'coldEngineFoam' solver of OpenFOAM [9], version 2.3.0, which is a solver for 'cold' (non-combustion) flow in internal combustion engines. It includes mesh motion for the moving piston and uses the PIMPLE transient solver for incompressible flow. PIMPLE is a combination of the PISO (Pressure implicit with splitting of operator) and SIMPLE (Semi-Implicit Method for Pressure Linked Equations) algorithms. PISO solves velocity-pressure coupling for each time step by predicting the velocity field from an initial guess of pressure and then correcting pressure and velocity to satisfy mass conservation. The correction step is then repeated. In the PIMPLE algorithm an outer correction loop can be used which iterates over the same time step using the last value as input for the next iteration, while under-relaxing variables between these outer iterations.

The model used in this study assumes perfect gas relations with constant heat capacity c_p, constant dynamic viscosity μ and constant Prandtl number. Reynolds-averaged simulations (RAS) based on a k-epsilon model are used for modelling turbulence. This is suitable for flows with small pressure gradients and no separation [10] and is less computationally intense than other models [11]. Note that although the solver is appropriate only for *incompressible* flow, this does not restrict it to constant density, which of course varies with the position of the piston. Mach numbers in the modelled domain are however typically less than 0.06 (when the Mach number is lower than 0.3 compressibility effects are negligible [8]), so local changes in pressure spread through the whole volume rapidly such that a new pressure with only small gradients is reached in each time step [12]. An incompressible solver like coldEngineFoam is therefore most suitable, although other reciprocating compression studies have been undertaken with compressible solvers [13].

The gas spring simulations were first checked for reliability: the CFD results were compared to theory for adiabatic and isothermal compression, and for pressure differences and axial velocity variations between the piston and cylinder head. Agreement was found to be very good, as detailed in Ref. [8]. Fig. 1 for instance shows pressure-volume (p–V) curves for different piston speeds. At high speed (1500 rpm) the forward and backward curves are very close together, indicating near-reversible processes. (The lost work is given by the area enclosed between the curves and is very small.) Compression and expansion are nearly adiabatic because there is insufficient time for heat transfer and consequently the curves lie close to the isentropic p–V curve, as shown in the figure (black symbols). For very low piston speed (0.01 rpm) the backward and forward curves are again nearly coincident, but this time lie close to the isothermal process (grey symbols), reflecting the fact that

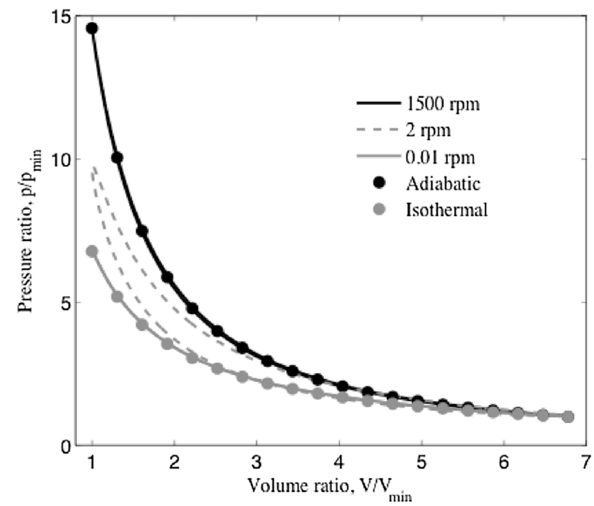

Fig. 1. Computed p–V curves for different piston speeds at compression ratio $r_v = 6.8$. The isothermal and adiabatic (isentropic) relations (pV = const. and pV^γ = const.) are also shown.

the gas is almost in thermal equilibrium with the cylinder walls. At intermediate speeds (e.g., 2 rpm) heat transfer occurs across significant temperature differences and the resulting irreversibility manifests itself as lost work and hence a difference between the forward and backward p–V curves.

The CFD results were further compared to experimental data available in the literature. Fig. 2 for instance shows the pressure against crank angle for two different piston speeds (note 0° corresponds to top dead centre, TDC). CFD results are compared with experimental data from [5] and generally show good agreement. No tuning was applied to achieve the agreement shown, here or elsewhere.

2.1. Wall heat fluxes

Wall heat fluxes are computed directly from

$$q_w = -k\frac{\partial T}{\partial n} \tag{1}$$

where k is the gas thermal conductivity and n is the distance normal to the wall. Negative values correspond to heat loss from the gas. For gas springs (for which the gas constitutes a closed

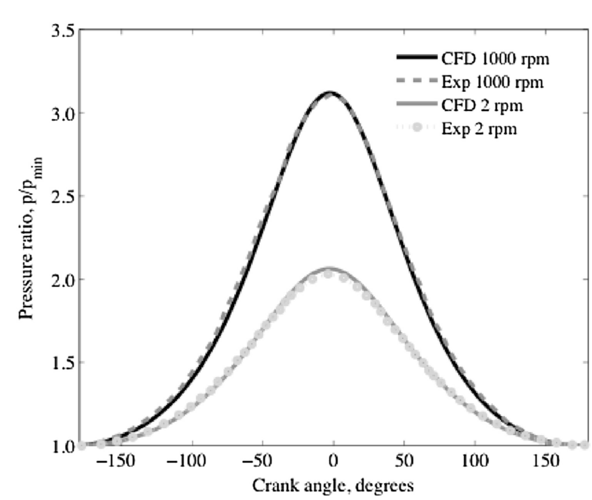

Fig. 2. Pressure vs. crank angle (CA) for 1000 rpm and 2 rpm. Experimental values and geometry are taken from [5]. The working fluid is helium.

Please cite this article in press as: C. Willich, A.J. White, Heat transfer losses in reciprocating compressors with valve actuation for energy storage applications, J. Energy Storage (2017), http://dx.doi.org/10.1016/j.est.2017.07.024