Numerical Simulation of Flow in Rotor-Casing Gap of an Rotary Piston Flow Meter

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Abstract: In the theoretical assessment of accuracy of rotary piston meters, the leakage flow in the rotor-casing gap is of essential importance. The present work aimed at detailed investigation of this flow field employing realistic numerical flow simulation. The resulting flow displayed significant differences to the simple assumption.

Keywords: Rotary flow meter, Flowrate, Numerical flow simulation, Leakage flow, Unsteady flow

1. Introduction

Rotary piston flow meters (RPFM) have been demonstrated as stable and reliable flow meters, having a large operating range. Their favourable properties render them suitable for usage as reference meters. They have been studied in the past fairly extensively and the theory of their operation is thought to be well understood. However, although recent experimental work resulted in the development of highly accurate rotary piston meters, many of the physical effects that influence their accuracy are difficult to explain and need further investigations.

One of the first detailed flow field investigations carried out in rotary piston flow meters using theoretical tools and experimental work was presented by Vollheim [1]. Further information about the theory of operation of RPFMs, offered by Masri and Kaiser, can be found in [2]. Some of the mechanisms of leakage generation as well as new methods to improve the accuracy of RPFMs have been studied by Dijstelbergen and van der Beek in [3]. The pulsating flow downstream of a typical RPFM has been investigated by a highly accurate ultrasound flow meter by Nath and Löber [4]. All of the above publications offer valuable information about the operational behaviour of the RPFMs, but each of them considered only a subset of the known effects influencing the accuracy of the meters. The aim of the present work was to unify the known simplified theories of operation in order to create a solid basis for further investigation of higher order flow effects in the RPFMs. The present authors extended the above theory to a comprehensive theoretically derived uncertainty that incorporated many aspects of the error sources [5].

In the present work, the unsteady compressible flow in a gap between the rotor and the outside casing of an rotary piston flow meter was numerically simulated. The only simplifying assumption was that the flow was two-dimensional. At this time, one piston was simulated since only the gap flow was of interest. The configuration was based on the rotary piston meter G400 DN100 manufactured by Elster GmbH. The piston rotated at prescribed angular velocities of 100 and 217 rad/s, the later corresponding to the maximum volumetric flow. Two forms of the tip of the piston were investigated: the usual design with sealing strip and a smooth piston. The corresponding shapes of the piston tip as well as the local grid are shown in Fig. 2 below. The flow was simulated using the commercial program Star CCM+, the grid was generated using the program Gridgen.

2. Principle of Operation

The basic theory of operation can be explained by observing the schematic picture of a typical rotary piston flow meter (RPFM) taken from [4].

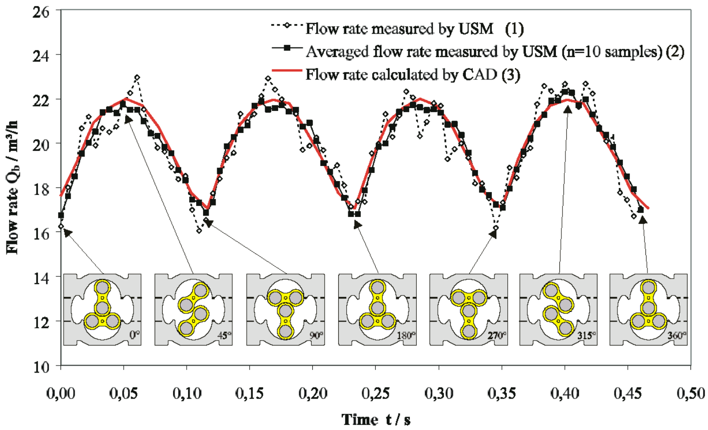

Fig. 1 Flow rate downstream of a RPFM as a function of time, from [4].

As shown in fig. 1, the rotating pistons form with the surrounding housing four volumes that are being expulsed into the pipe downstream, generating four pulses per one complete rotation of a piston. Since the size of the volumes is uniquely defined by the geometry of the meter, the volumetric flow should correlate directly with the rotational speed of the pistons. There is, however, an additional gas flow through the inevitable gaps between the pistons and the housing and between the two pistons, leading to leakage flow not included in the four volumes. The difference between the indicated flow rate (given by the meter under investigation, RPFM) and the true total flow rate (obtained by some reference meter) is the deviation ε. According to the above theory, ε due to leakage should always be negative. Assuming an experimental setup in which the RPFM is followed by a reference meter, the deviation is computed from:

$$\varepsilon = \frac{Q_{RPFM}\dfrac{P_{RPFM}}{P_{ref}}\dfrac{T_{ref}}{T_{RPFM}} - Q_{ref}}{Q_{ref}} \qquad (1)$$

In equation (1), P denotes the absolute static pressure and T the temperature, measured at a proper location at the corresponding meter. Some of the effects mentioned above could not be explained by the simple theories of operation given in the past in open literature. In particular, the equation of the dependence of the relative error of the meter on either the flow rate or the mass flow given by, for example, Dijstelbergen and van der Beek [1], displayed the correct tendencies but lacked the explicit consideration of the effects measured by several other investigators.

The authors therefore decided to extend the theoretical frame work used for the calculation of the leakage losses that are thought to be the main source of the measurement errors. Defining the leakage flow Q_l by $Q_{tot} = Q_{RPFM} + Q_l$, and assuming that $Q_{ref} = Q_{tot}$, results in

$$\varepsilon = \left(1 - \frac{Q_l}{Q_{tot}}\right)\frac{P_{RPFM}}{P_{ref}}\frac{T_{ref}}{T_{RPFM}} - 1 \qquad (2)$$

3. Theoretical Determination of Leakage Flow

It has been shown by several investigators [1, 2, 3] that the leakage flow could be computed by assuming simple two-dimensional flow between two plates, one of them moving with the tangential velocity of the piston $u_w = r \cdot \omega$, where r is the radius of the piston and ω is the angular velocity of the piston. This type of flow is called the "Couette flow" and is described in much more detail in, for example, Schlichting [6]. It shoul be remembered, however, that the simple theory in [6] assumes steady flow, whereas the flow in the RPFM is highly unsteady. Therefore, the theory strictly applies only to some representative flow variables obtained by averaging over some time period T much longer than the period of the flow fluctuation. A typical frequency of the fluctuations at maximum flow rate would be approximately 200 Hz. The mean flow velocity through the gap is therefore given by

$$\bar{u}_G = \tfrac{1}{2} u_w(t) - \frac{H(t)^2}{12\mu} \cdot \left(\frac{\partial p}{\partial x}(t) \right) \qquad (3)$$

with H being the height of the gap and μ the dynamic coefficient of gas viscosity. Most of the variables are unsteady, as their values fluctuate periodically with time. The main task now is to evaluate the pressure drop across the meter, computed from the gradient of the static pressure,

$$\Delta p = \Delta x \cdot \frac{\partial p}{\partial x} \quad , \quad \Delta x = b_L$$

where b_L is the width of the gap.

4. Numerical Simulation

In the present study, the flow in the gap between the piston and the casing was investigated by numerically simulating the corresponding flow field. As the entire RPFM was extremely difficult to simulate due to the pistons moving relatively to each other at varying velocity, it was decided to assume a simplified model including only one piston and the entire casing. However, the missing second piston would result in an open channel flow below the remaining piston, giving totally unrealistic results. Therefore, the passage below the piston was partially blocked by an infinitely thin vertical wall (2) that sealed the the opening against a second semicircular wall firmly attached to the piston (1). The model geometry is shown in figure 2.

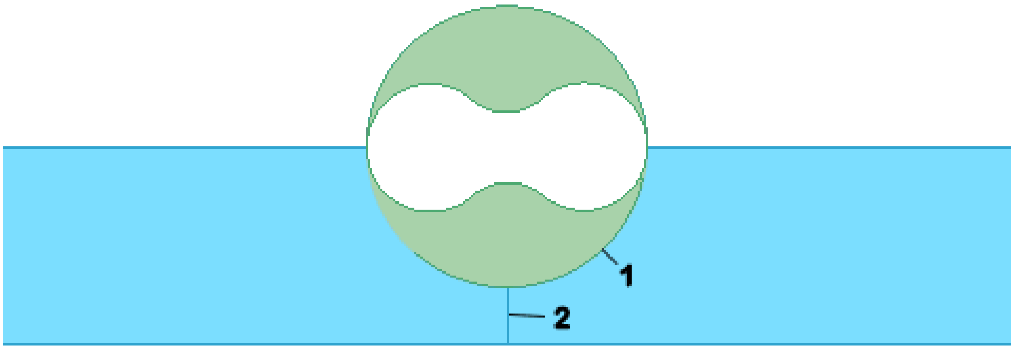

Fig. 2 Geometric configuration employed in the present work.

The computational grid consisted of two major parts, an annular region around the piston rotating with the piston, and a stationary grid for the remaining inflow and outflow piping as well as the chamber containing the pistons. Early in the study, it was decided to use an hybrid grid, consisting of structured regions in areas dominated by viscous effects, and the remaining unstructured regions. The rotating part was merged with the stationary one by so called sliding mesh boundary. Both grids are shown in figures 3 and 4.

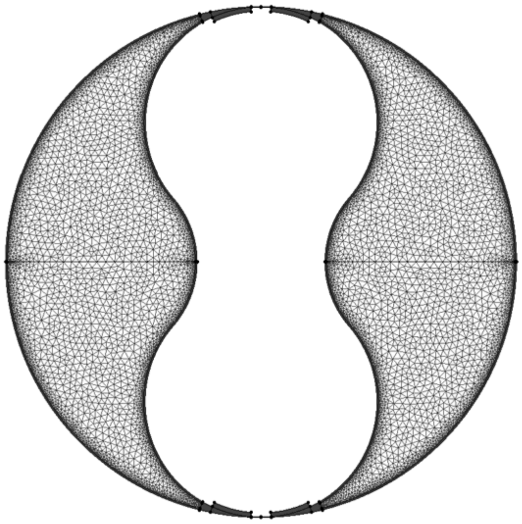

Fig. 3 Grid in the rotating part.

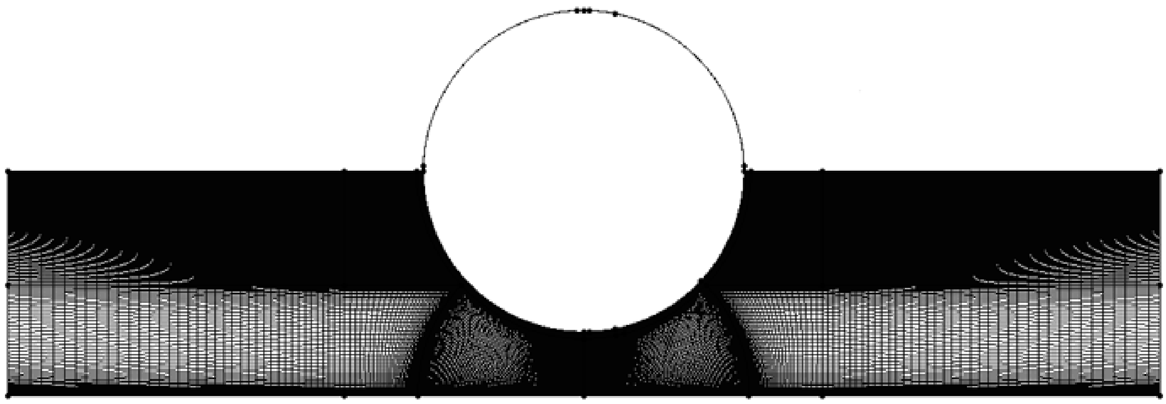

Fig. 4 Grid for the stationary parts.

There were two piston configurations, one with a seal and one smooth, see figure 5 for details. Previous simple computations indicated, that the simple smooth piston actually provided a better sealing properties than the one with the sealing groove. The grid consisted of approximatally 180,000 cells (smooth) and 200,000 cells (with groove) , respectively, arranged in more than 30 blocks, all in two dimensions.

The flow simulations employed a commercial CFD-program Star-CCM+ manufactured by adapco. Due to the relatively low Mach numbers in the flow field, the uncoupled incompressible versión was used. The algorithm selected presently was second order accurate in time and space. The flow was assumed to be viscous and turbulent, with the well known Nikuradse velocity profile at the inflow.

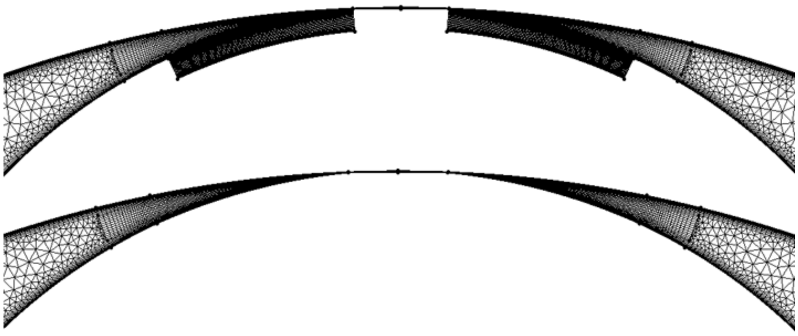

Fig. 5 Detail of grids at the piston tips.

4. Results

Four cases were considered: smooth piston at 100 rad/s (955 rpm), smooth piston at 217 rad/s (2072 rpm), piston with seal at 100 rad/s and piston with seal at 217 rad/s. The input values are summerized in table 1. In all four cases, the pressure difference across the gap were taken from experimental data.

The flow simulations were carried out for all 4 cases for at least 10 complete rotations of the piston.

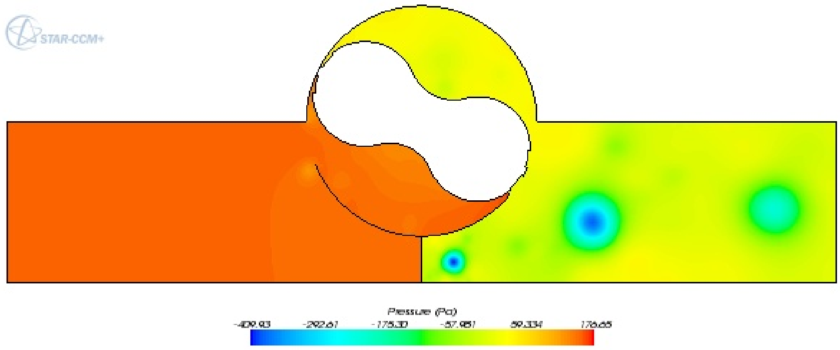

Fig. 6 Flow field of piston with seal, 100 rad/s

The corresponding pressure difference across the gap can be seen in figure 7.

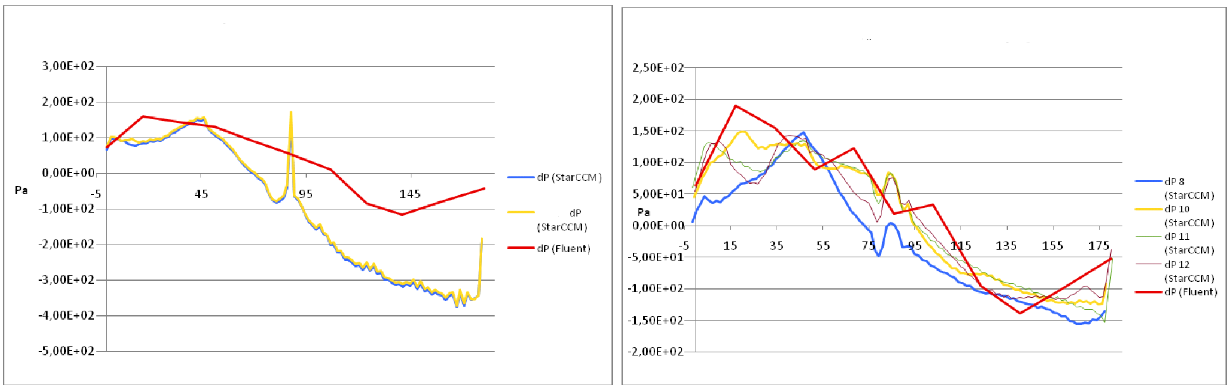

Fig. 7 Pressure difference for slotted piston (left) and smooth piston (right), 100 rad/s.

Here, the pressure difference is shown as a function of angular position of the piston. In the left picture, displayed for the slotted piston, a gradual increase of the absolute value of the pressure difference can be seen. The sudden pressure increase at approximately 90 degrees is due to a vortex trailing the opposite side of the piston, being overtaken by the piston tip under investigation. A similar sudden change of the pressure can be also seen in the right picture for the smooth piston.

At higher rotational speed, the average pressure difference is obviously larger. The pressure difference is displayed as a function of the angular position of the piston in figure 8.

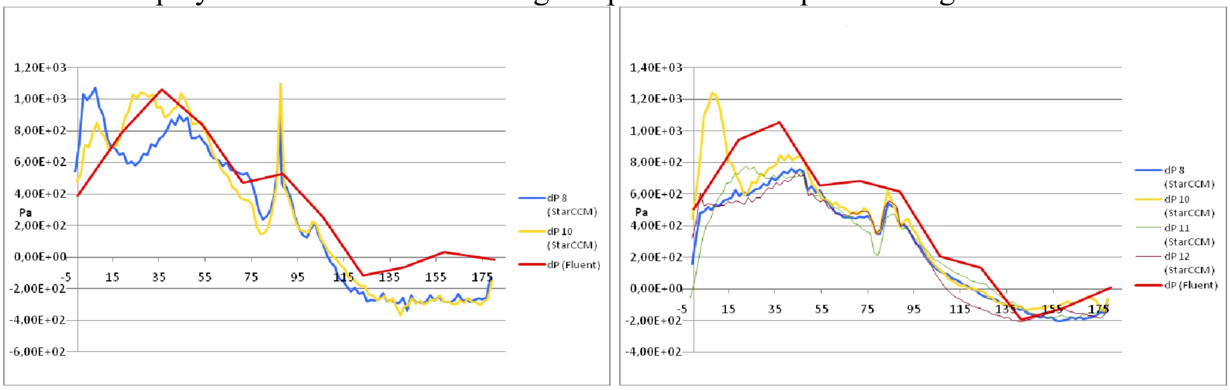

Fig. 8 Pressure difference for slotted piston (left) and smooth piston (right), 217 rad/s.

Qualitatively, the diagrams in figure 8 are the same as those in figure 7. Due to the higher rotational speed, a steady state condition corresponding to the well known Couette flow was established in the case of the slotted piston at approximately 120 degrees. After that time (and location), the pressure difference remained the same. In the case of the smooth piston, this condition was not reached until the piston completed its half turn pass the casing.

The physical conditions as well as the resulting mass flow are summarized in Table 1 for all four cases considered in the present work.

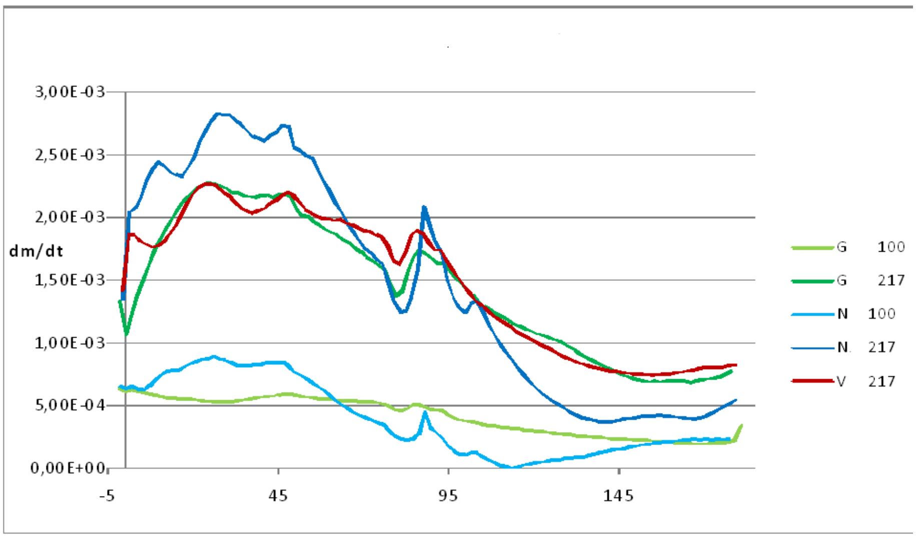

Fig. 9 Mass flow across the gap for all four cases.

The momentary mass flow through the gap for all four cases is compared in figure 9. The smooth piston is denoted as “G”, the piston with sealing slot as “N”. The curve “V” represents a case with a semicircular chamber in front of the intake. During the steady state operation of the slotted piston, the mass flow, corresponding to the Couette flow, is smaller than the still developing mass flow during the unsteady mode of operation. This is, however, only a small portion of the entire travel past the casing. At the full rotational speed at the maximum volumetric flow, the entire mass flow obtained by integrating the curves in figure 9, is large for the piston with sealing slot than for the smooth one, making the smooth solution a better one. At the half speed, the slotted piston is slightly better than the smooth one.

6. Conclusions

The numerical simulation of the flow across the gap between the piston and the casing in a rotary piston flow meter revealed that the flow in this region is mainly unsteady. Contrary to the assumption of fully developed Couette flow, the flow needed almost the entire half rotation (90 degrees) to reach a steady state that corresponded to the Couette flow. Before reaching this condition, the mass flow was significantly larger than for the fully developed viscous flow. The type of flow during this time was similar to intake flow in a circular pipe.

Geometry	Rot. Speed (rad/s)	Pressure diff. (Pa)	Mass flow (kg/s)
smooth	100	151	0.0748198
smooth	217	960	0.24383
seal	100	151	0.0712313
seal	217	960	0.26867

Table. 1 Summary of the four cases considered presently.

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