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A REVIEW OF MODELS FOR PREDICTING INSTANTANEOUS HEAT EXCHANGE BETWEEN THE GAS AND CYLINDER IN RECIPROCATING COMPRESSORS

ABSTRACT: It is an accepted fact that heat transfer between the gas and the metal surfaces delineating the compression space of a reciprocating piston compressor (cylinder wall and cover, and piston) has significant effect upon the compressor performance and the temperature of the valves. This has given rise to a number of theoretical, computational, and, to a lesser extent, experimental studies aimed at explaining and quantifying the relevant mechanisms. However, major part of the research effort has been spent in the field of hermetic refrigeration compressors, which is quite understandable given the vast number of such machines in use. On the other hand, the heat transfer models developed are not necessarily useful for predicting the cylinder heat transfer in small, air-cooled, air compressors used on commercial vehicles for braking and other auxiliary purposes.
Surveyed in the paper are several heat transfer models from the open literature that could be suitable for use within performance prediction software for small air compressors. One promising model was implemented in such a program in order to be compared to the data. Based on the comparison with limited data available to the authors, this simple unsteady model is apparently capable of predicting the cylinder heat transfer with an acceptable accuracy. However, more measurement data are required before a fully qualified statement as to its general utility can be made. The authors hope to obtain these on the test rig for small air compressors in the Engine Laboratory of the Faculty of Engineering of Kragujevac, which is currently being brought into operation.

KEYWORDS: Reciprocating Compressor, Cylinder, Heat Transfer Model.

INTRODUCTION

Within the working cycle of a reciprocating compressor, the work being imparted to the gas enclosed in the cylinder by the moving piston in the compression phase brings about increases in both the gas pressure and temperature. Conversely, in the back-expansion phase of the process, the respective values of the above two gas state variables diminish. Since the time constants of the relevant thermal processes in the gas and the metal parts delineating the compression chamber of the machine are vastly different, there are temporal and spatial temperature differences between them that give rise to heat exchange. Additionally, the gas stream entering the cylinder in the suction phase mixes vigorously with the cylinder gas, giving rise to an intensified heat exchange. It is thus to be expected that these processes influence the machine performance.

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Studying the effects of the heat exchange in and around the compressor cylinder by purely experimental methods is not only time-consuming and expensive, but in some aspects also hardly feasible. Therefore, mathematical modelling of the relevant phenomena can provide valuable insight into the physical mechanisms involved, and help thus the designer in quantifying their effects upon the thermodynamic performance of the compressor, and to its reliability and service life as well.

In the classical reciprocating compressor literature (e.g. [13]), heat exchange between the gas and the cylinder wall was considered to have an important effect upon the global compressor performance. This opinion was repeated in the study by Adair et al. [1], who claimed that heat transfer may account for as much as 10 to 20% decrease in the compressor thermodynamic and volumetric performances. Since prior to their work the heat transfer correlations used to calculate the reciprocating compressor performance came from the field of internal combustion engines, the authors also established their own correlation, customized for the compressor cylinders. Prakash and Singh [39] also stressed the importance of considering cylinder heat transfer in a refrigerating compressor model.

In 1980, Brok et. al. [6] expressed doubt as to the strong influence of the cylinder heat transfer upon the machine performance claimed by the previous authors. Based on their own calculations, the thermodynamic and volumetric efficiencies could be reduced by about 2.5% (4% in the worst case) as a consequence of heat transfer, and concluded that the inclusion of heat transfer models into the code for predicting compressor performance is rather doubtful. Essentially the same opinion can be found in another contemporary paper [23], the machine simulated being a refrigerating compressor.

In 1998, Shiva Prasad [42], quoting previous work by Gerlach and Berry [14] and his own extensive research, made a strong case for studying the cylinder heat transfer effects both at the fundamental and the application levels (i.e. in performance prediction tools) in order to help compressor users in understanding this problem. A strong influence of the model used to calculate the above effects in a performance prediction program upon the global performance figures of a labyrinth-piston compressor (LABY[5], see e.g. [49]) was demonstrated in a 1996 paper by Ninkovic [32]. Two heat transfer correlations ([30] and [5]) were compared in a subsequent paper [33], exhibiting again the importance of accounting for the gas-wall heat exchange in the cylinder model.
In the past twenty years, the reciprocating compressor research community has seen an upsurge in the activities (theoretical, CFD, and experimental) aimed at gaining more insight into the effects of heat exchange in small refrigeration compressors. There was also a joint research project under the auspices of the European Forum for Reciprocating Compressors (henceforth EFRC) that dealt with this phenomenon in the field of large process compressors [2].

Small, fast-running air compressors are somewhat different from the bulk of machines whose heat transfer mechanisms and their effects upon the system performance have been studied in the past. In comparison with e.g. hermetic refrigerating compressors, they operate over wide speed ranges, have much higher pressure ratios, and have a very different thermal contact with the surrounding air. As Muellner and Bielmeier [31] and Stouffs et al. [43] point out, these design and operational features can have serious impact upon the performance and reliability of this class of machines, and should therefore be studied in detail.

Therefore, the question nowadays does not seem to be whether cylinder heat transfer ought to be considered in the performance prediction programs, but rather how it is to be modelled. It is with a view to this that the present paper has been written; its second purpose is to serve as a first step toward own research in this area.

MODELLING OF THE CYLINDER THERMODYNAMICS

For the subsequent analysis the compressor cylinder shall be abstracted to a zero-dimensional, variable-volume, open thermodynamical system. The volume change is effected by a periodically moving piston; the latter may be driven by a connecting rod, or by a piston rod, as in e.g. a double-acting cylinder. The cylinder exchanges mass with the surroundings mainly through the suction and delivery valves; additional (small) mass transfer may be possible through an imperfectly sealed piston-wall gap (piston blow-by) and/or the piston rod sealing. Heat exchange can take place at all surfaces that are in contact with the gas contained in the volume, i.e. at the cylinder cover, cylinder wall, the piston, and the piston rod, if the latter is available. Note that although the cylinder model is zero-dimensional with respect to the gas state, it allows for the existence of metal surfaces with different temperatures; the latter refers to e.g. the cylinder cover and cylinder wall temperatures not being equal.
In the differential form, the law of mass conservation for the cylinder gas reads:

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[5] *LABY is a registered trade mark of Burckhardt Compression Ltd., Winterthur, Switzerland*

$$\frac{dm_g}{dt} = \sum_{ports} \dot{m} \quad (1)$$

wherein the subscript g refers to the gas trapped in the cylinder, and *ports* (see [16]) are surfaces at the control volume where gas flows into or out of the cylinder, i.e. the suction and delivery valves, and the leakage locations. Adopting the same formalism as in Eq. (1) above, the energy balance of the cylinder gas can be expressed as:

$$\frac{dU_g}{dt} + \sum_{ports} \dot{m} \cdot h_t = \frac{\delta Q}{dt} - \frac{\delta W_p}{dt} \quad (2)$$

wherein U_g stands for the gas internal energy, h_t for the total (stagnation) enthalpy, Q for the amount of heat exchanged, and W_p for the piston work. The inflowing mass flow rates and heat added to the gas are considered positive, whereas the piston work comes with a negative sign.

In order to integrate Eq. (2) above for the internal energy, all terms to the right of the latter must be known as functions of time. As it happens, only the piston work term is defined in a simple manner in terms of the compressor geometry and the crankshaft kinematics. In the absence of leakages, the enthalpy flows are determined in a complex way by the characteristics of the valves and their interactions with the cylinder and its attachments; this is the subject of a companion paper by the same authors at this Conference [36].

Although cylinder heat transfer has been an actively pursued research subject in the past decades, there are no comprehensive reviews of the state of the art. The paper of Shiva Prasad [42], in the author's own words, does not represent a thorough and exhaustive review, and a recent one by Ribas et al. [41] deals with the thermal design of a class of machines, which is a broader issue than the gas-wall heat exchange. There is also a short review of the models applicable to a related area of Stirling engines in [22], but it is 25 year old. Finally, the dissertation of Lekic [28] also contains a review of heat transfer models from the standpoint of the gas springs, which also bear resemblance to reciprocating compressors.

This state of affairs prompted the present authors to undertake a modest attempt toward providing at least a part of the lacking information.

HEAT TRANSFER MODELLING

Given the working principle of reciprocating compressors, heat exchange between the gas and the surrounding walls proceeds in an unsteady manner, chiefly by the mechanism of forced convection. The rate of heat exchange is customarily calculated by the classical Newton formula:

$$\frac{\delta Q}{dt} = \alpha \cdot A_w \cdot \left(T_w - T_g\right) \quad (3)$$

wherein α represents the local convection coefficient, and T_w and T_g stand for the wall and gas temperature, respectively. Note that all quantities at the RHS of Eq. (3) can be instantaneous values. Models based on the above equation differ in the way the local convection coefficient is related to the gas state in the cylinder, compressor design, operating point, wall properties, etc. The same concept still constitutes the basis of heat transfer modelling in the field of internal combustion engines [46].

In steady pipe flow, the heat exchange rate of Eq. (3) varies in phase with the gas-wall temperature difference, which results in the so-called quasi-steady approximation of the cylinder heat transfer process. It has been known for quite a long time that this is not necessarily the case if the flow oscillates, which led to the development of unsteady flow heat transfer models.

Modelling of the instantaneous heat exchange in a compressor cylinder can also be approached from other directions, such as e.g. by performing a spatial discretization of the cylinder, which do not need an explicit formula like Eq. (3) above. This approach is referred to as numerical methods in the present paper.

Quasi-Steady Models

The models in this group have their origin in the work of Nusselt [37], which concerned heat transfer in the IC engine cylinders. The departing point is the hypothesized analogy with heat transfer in steady, fully developed, turbulent flow in a pipe. Nusselt arrived at the following formula for the local convection coefficient:

$$\alpha = 0.0278 \cdot p_g^{\frac{2}{3}} \cdot T_g^{\frac{1}{3}} \cdot (1 + 0.38 \cdot c_m) \left[\frac{BTU}{h \cdot ft^2 \cdot {}^{\circ}R}\right] \quad (4)$$

with c_m representing the mean piston speed. Eichelberg [10] proposed a similar formula

$$\alpha = 0.0565 \cdot p_g^{\frac{1}{2}} \cdot T_g^{\frac{1}{2}} \cdot c_m^{\frac{1}{3}} \left[\frac{BTU}{h \cdot ft^2 \cdot {}^{\circ}R}\right] \tag{5}$$

which was followed by several other expressions of similar form by other authors. If the gas state variables are taken as functions of time, one obtains the instantaneous value of the local convection coefficient.

One of the best known correlations for modelling heat transfer in IC engines was developed by Woschni [48]. In keeping with the standard practice in the study of convective heat transfer, it was formulated in terms of the Nusselt and Reynolds numbers:

$$Nu = a \cdot Re^b \tag{6}$$

with $a = 0.035$ and $b = 0.8$. The characteristic geometric parameters, flow velocity, and gas properties that make up the Nusselt and Reynolds numbers are defined so as to correspond to the problem at hand, i.e. there will be no combustion term if the model is used for calculating a reciprocating compressor. For example, Annand [4] recommends $a = 0.76$ and $b = 0.64 \pm 0.10$ for two-stroke engines, calculating Nu and Re with instantaneous gas properties, cylinder diameter as the characteristic length, and mean piston speed.

Adair et al. [1] carried out measurements on a running compressor and calculated the instantaneous heat flux from a fast response surface thermocouple mounted in the cylinder head. They used the data thus obtained to arrive at the following correlation:

$$Nu = 0.053 \cdot Re^{0.8} \cdot Pr^{0.6} \tag{7}$$

In order to account for the agitated gas flow in the suction phase of the cycle, the characteristic gas velocity is calculated by multiplying the crankshaft angular velocity ω by a two-branch, raised cosine function of the crankshaft angle φ:

$$w_g = \begin{matrix} 2 \cdot \omega \cdot [1.04 + cos(2\varphi)] \;\; for \; \frac{3\pi}{2} < \varphi < \frac{\pi}{2} \\ 1 \cdot \omega \cdot [1.04 + cos(2\varphi)] \;\; for \; \frac{\pi}{2} < \varphi < \frac{3\pi}{2} \end{matrix} \tag{8}$$

The upper branch velocity of Eq. (8) is referred to as swirl velocity, a term borrowed from the IC engine terminology, where it describes an intentionally created circular motion of the air admitted into the cylinder. It is frequently used in the field of compressor heat transfer modelling in spite of the fact that such gas motion is hardly possible in a compressor cylinder.

Although the above heat transfer model was calibrated with the data acquired in a reciprocating compressor, representing thus the first compressor heat transfer model (other authors used the IC engine models), the authors admit a poor prediction accuracy of the measured heat flux. Brok et al. [6] modified the velocity formula of Eq. (8) by halving the cosine term in both branches, and achieved a better agreement with the data. Recktenwald et al. [40] compared the original model of Adair et al. with a finite-difference model of the compression space. The latter produced heat flux values comparable to the data recorded in motored engines, whereas the former were at least one order of magnitude lower, which led the authors to advice against implementing the model of Adair et al. in the performance prediction software. Nevertheless, it seems to have been used in a number of such programs [12]. On the other hand, there were also programs that disregarded the cylinder heat transfer [23], or considered the latter as additional processes taking place before and after an otherwise isentropic cylinder [47].

Todescat et al. [45] compared the heat transfer models of Adair et al. [1], Annand [4], and a modification of the latter which consisted of multiplying the results by three (the same factor was established in [33] when comparing the correlations of [5] and [30]), with the data measured on a hermetic refrigeration compressor. None of the models was able to match the entire data set: while the model of Adair et al. produced the best agreement in terms of power consumption, it performed worst on the mass flow rate. The pattern was reversed in the case of the modified Annand model. On the other hand, in the study of Lee et al. [25] the model of Annand [4] delivered the highest values of the convection coefficient among several models, including the ones by Adair et al. [1] and Eichelberg [10].

In 1984, Liu and Zhou [30] reported on an experimental investigation of the radial and axial temperature distributions in a refrigerating reciprocating compressor. Using the data acquired, they also performed a least squares parameter fitting of the Adair model (Eq. (7) and (8)), arriving at the following result:

$$Nu = 0.75 \cdot Re^{0.8} \cdot Pr^{0.6} \tag{9}$$

$$w_g = \begin{array}{ll} 2 \cdot \omega \cdot [1.04 + 0.45 \cdot cos(2\varphi)] & for\ \frac{\pi}{2} < \varphi < \frac{3\pi}{2} \\ 1 \cdot \omega \cdot [1.04 + 0.5 \cdot cos(2\varphi)] & for\ \frac{3\pi}{2} < \varphi < \frac{\pi}{2} \end{array}$$

In effect, the constant multiplier in the Nusselt number formula in their model is approximately 15 times larger than in the original one, and the characteristic velocity, which they refer to as swirl-squish velocity, is close to the modification made by Brok et al. [6]. Note that the crankshaft angle origin in [30] is the top dead center (TDC). Fagotti at al. [11], following the work of Todescat et al. [45], evaluated five heat transfer models, including [1], [4], [6] and [30], by comparing the simulation results with the global performance figures of a refrigerating compressor. While they concluded that the model of Liu and Zhou [30] was the best one in terms of the accuracy of temperature prediction, they decided in favour of the model of Annand [4] for inclusion into their performance prediction software on account of better results of the latter with a parameter specific for their application.

The model of Eq. (9) was built into a comprehensive compressor simulation program of Sulzer-Burckhardt Ltd. (now Burckhardt Compression) and its performance assessed by comparing the global performance results with the measured data for different machine types (lubricated ring process compressors and LABY) compressing various gases. Initially, parameter studies were performed with the original form of Eq. (9), which indicated strong influence of the heat transfer upon the global results [32]. A subsequent study [33] revealed a critical dependence of the simulation results upon the heat transfer model in the case of labyrinth piston compressors (the two-branch curve of [30] for the characteristic gas velocity was replaced by a single smooth curve in these tests). It was found that the model of Annand [4] is not suitable for the latter machine type. However, results obtained with the program with the heat transfer model of Liu and Zhou [30] were found to be generally satisfactory, including also an extreme case of the last stage of an ethylene hyper-compressor with a delivery pressure of 2900 bar [34].

Recently, a novel quasi-steady cylinder heat transfer model was developed in a joint EFRC project [2], [50]. The basic hypothesis of the model is that the local heat flux densities $\dot{q}_i$, i.e. the amounts of heat exchanged between the gas and the compression space walls per unit area and unit time, are related to suitably chosen reference enthalpy densities $\dot{h}_{ref,i}$ weighted by the respective Stanton numbers St_i:

$$\dot{q}_i = St_i \cdot \dot{h}_{ref,i} = St_i \cdot c_p \cdot \rho_{ref,i} \cdot w_{ref,i} \cdot \Delta T_i \tag{10}$$

with the index *i* running over all heat exchange processes in the physical sense, and ΔT_i representing the respective gas-wall temperature differences. Since Stanton number is effectively a modified Nusselt number [17] and only instantaneous temperature differences are involved, this model is here reviewed in the quasi-steady category.

The authors identify four physical processes in the cylinder that generate enthalpy fluxes to be used in the model: suction gas inflow, compression, compressed gas outflow, and back expansion, but the compression and expansion processes are treated as the same heat transfer mechanism. However, the index *i* in Eq. (9a) still runs from 1 to 4 because the model equation for compression/expansion is expressed in terms of two Stanton numbers. In order to be able to calculate the component heat flux densities, one must know the corresponding Stanton numbers. Although not clearly stated in the paper [2], it appears that a set of four Stanton numbers must be known for each heat transfer surface in the cylinder in order to calculate all contributing heat fluxes. The values of the individual Stanton numbers are obtained by minimizing (in the least squares sense) the deviation between the model and a known heat flux over a machine period. The latter was obtained from CFD simulations of the compressors investigated in the study at several operating points. This is effectively the only possible choice, for measuring all component heat fluxes in a running machine is practically not feasible.

Very good agreement was obtained between the heat fluxes computed by the model as implemented in the performance prediction programs that were also developed in the project, and the results obtained from the CFD simulations. However, no comparisons with measured heat fluxes and/or global performance figures seem to have been published in the open literature; and the heat fluxes calculated seem to be rather low in magnitude. The good agreement of the respective pressure traces, as presented in [2], lets one expect accurate global results too, but with a view to the differences in the global performance figures for very similar indicator diagrams observed in e.g. [40], [33], a comparison with measured data could have shed some more light onto this interesting phenomenon.

Unsteady Models

The models reviewed in the previous group are all based on the assumption of steady flow, which does not hold for reciprocating machinery, be it piston compressors, IC or Stirling engines, or gas springs. In his paper of 1963, Annand [4] reviewed several publications containing measurement results obtained in IC engines (including the one by Nusselt [37]), which report on the observation that the heat flux does not follow the gas-wall temperature difference, having even significant values when the latter is zero. This would call for an infinite convection coefficient in Eq. (2), invalidating thus the quasi-steady modelling approach. However, Annand did not take into account these effects in his model.

The same phenomenon was observed in another two types of reciprocating machinery, namely in gas springs and Stirling engines. They have been the subject of intensive theoretical and experimental research in the past decades, which generated a large amount of valuable data on cylinder heat transfer ([22], [29]). Although these two machines are not quite the same as reciprocating compressors, for the suction and discharge phases of the compressor do not exist in gas springs and Stirling engines, and the flow past the piston is minimized in compressors, there are enough similarities from the standpoint of cylinder heat transfer to study the results obtained in these areas.

Since the unsteady heat transfer research in the above mentioned fields proceeded mostly in parallel in the past three decades, and given the amount of information available, only the most salient results from both fields that have a bearing on heat transfer modelling in reciprocating compressors shall be reviewed here. For more details the interested reader is referred to Torregrosa et al. [46] for IC engines, and Kornhauser et al. [22] and Lekic [29] for Stirling engines and gas springs, respectively.

Pfriem [38] was the first to present a theoretical analysis of engine heat transfer under the conditions of pulsating pressure. According to his analysis, heat flux at the wall leads the gas temperature far from the wall by $\pi/4$, hence his conclusion that the use of Newton's formula would be inappropriate in this case. Pfriem then introduced complex-valued heat flux and temperature, and by dividing the former by the latter, obtained complex convection coefficient and, by extension, the complex Nusselt number. Utilizing these results he was able to show that at low pulsation frequencies and small hydraulic diameters the real part of the complex heat transfer coefficient was dominant, indicating thus quasi-steady heat exchange. Conversely, the above mentioned phase difference of $\pi/4$ would occur at high pulsation frequencies in large cylinders. Referring to Fig. 1 below, the results of a gas spring CFD study reported by Lekic and Kok [27] clearly demonstrate the correctness of the Pfriem's analysis.

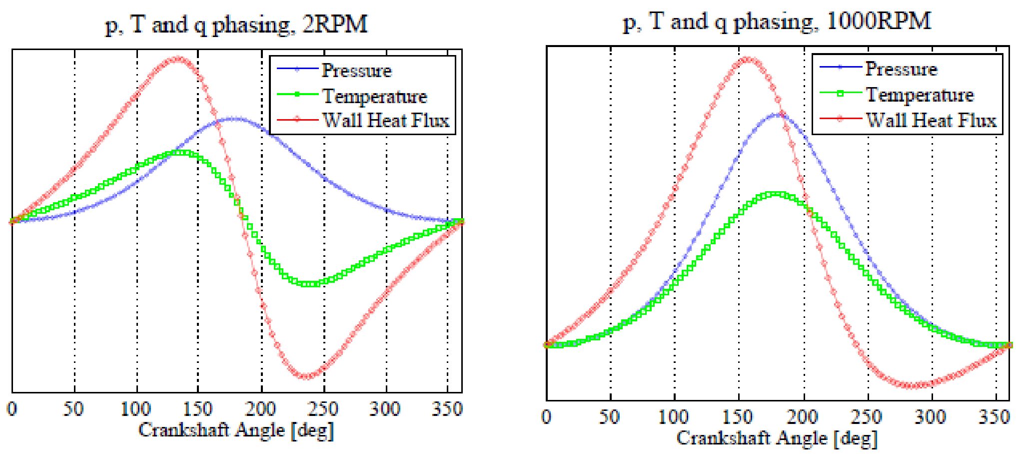

Figure 1 *Heat transfer in a gas spring at low and high rotational speeds (from [27])*

In an attempt to analyze losses due to heat transfer in gas springs, Lee [26] developed a theoretical model of heat transfer for small-amplitude pressure fluctuations in a one-dimensional geometry without turbulence. His model was used to predict gas spring performance, e.g. [20], but it did not account for turbulence generated in the cylinder by the inflowing gas; one can expect that the latter effect is much stronger in reciprocating compressors. They found out that the model predicted the loss at the cylinder volume ratio of 2.0 fairly well, but underpredicted (with an increasing trend) at larger volume ratios. The measurement data used for the investigation were acquired on a test rig built by Kornhauser for his PhD research [19], in which he also considered the turbulence in a model he developed by extending the one of Pfriem [38]. However, both models produced relatively small values of phase difference between the temperature difference and the heat flux. A later model of Cantelmi et al. [7] displayed an improved performance in this regard, but the need for further research remained [44].

In spite of the efforts of the NASA to further the basic research on cylinder heat transfer, the test rig of Kornhauser [19] remained practically the only one in the "public domain" until a sophisticated one was recently built at Twente University [29]. Comparing the cylinder heat transfer results obtained by means of a CFD simulation with the data of [19], Lekic and Kok [27] obtained a very good agreement in the low-speed, laminar flow, operation, whereas the opposite was true in the high-speed, turbulent flow, range. The reason for the latter was thought to lie in the way the instantaneous heat flux was calculated from the experimental data collected by Kornhauser [19]: since he did not measure the gas-wall heat flux, the latter had to be obtained from the bulk gas temperature, which in turn was extracted from the indicator diagram in terms of pressure and cylinder volume. Knowing that at high speeds the heat flux is not in phase with the temperature difference, the heat flux data thus obtained did not compare well with the CFD results. In his later work, however, Lekic [29] achieved good agreement between his DNS simulations and the data of [19], localizing the cause for the previous disagreement in the turbulence models used in his initial work.

Obviously, this makes a strong case for direct heat flux measurement in any heat exchange investigation involving oscillating flow.

Going now over to the IC engine heat transfer research, Annand and Pinfold [5] proposed a model that accounts for the above discussed phase difference by making the local convection coefficient also a function of the rate of change of gas temperature:

$$Nu = 0.3 \cdot Re^{0.7} \cdot \left[1 + 0.27 \cdot \frac{D}{w_g \cdot (T_w - T_g)} \cdot \frac{dT_g}{dt}\right] \tag{11}$$

In this formula w_g is the local instantaneous velocity, used also to calculate Re, and the characteristic length is the cylinder diameter D. Kornhauser [19] demonstrated that the above equation can be generalized in terms of the complex Nusselt number. , Annand and Pinfold mention difficulties in finding a correspondence between the phase shift observed and other quantities characterising the gas and the operating conditions.

The above model of Annand and Pinfold does not seem to have attracted much interest in the IC engine community. As already mentioned, it was compared in the versions with and without the temperature differential term to the model of Liu and Zhou [30] by simulating the same labyrinth-piston compressor working with helium [33]. The global results of the machine were consistently worse than those obtained with the model of [30], but it must be mentioned that this particular test is very difficult, for the performance of this compressor type is extremely sensitive to the size of the cylinder-piston gap, which in turn is dependent upon the piston heat balance, i.e. the heat transfer model used.

In 1987, Lawton [24] reported on an experimental study of cylinder heat transfer carried out with a motored Diesel engine, deriving the heat flux data from the measurements of instantaneous gas temperature. He also performed a theoretical analysis that included the wall boundary layer, the result of which was a heat transfer model expressed as a differential equation. Solving the latter by means of a finite-difference procedure, Lawton obtained very good agreement between the data and the predictions. Arguing that his heat transfer model in this form may not be attractive for implementation in performance prediction software of the time, he developed a closed-form model equation by extending the quasi-steady formula of Annand [4]. The proposed model reads

$$Nu = 0.28 \cdot Re^{0.7} - 2.75 \cdot \frac{L \cdot T_w}{T_g - T_w} \tag{12}$$

wherein the quantity L is referred to as compressibility factor, to be calculated as:

$$L = (\kappa - 1)\frac{\dot{V}}{V}\sqrt{\frac{D^3}{\alpha_0 \cdot w_p}} \tag{13}$$

with κ denoting the isentropic exponent, w_p the piston speed, and $\alpha_0 = k/(\rho \cdot c_p)$ thermal diffusivity of the inlet air. The above model agrees slightly worse with the measured data than the one based on solving the differential equation of heat transfer, but the differences are not serious. It was compared to the models of Adair et al [1], Annand [4], and Kornhauser and Smith [21] in a study by Fagotti and Prata [12] aimed at finding a replacement for the models of [1] and [4] in their software for performance prediction of small refrigeration compressors. The model of Lawton was first calibrated against measurement data, yielding:

$$Nu = 0.28 \cdot Re^{0.65} + 0.25 \cdot \frac{L \cdot T_w}{T_g - T_w} \tag{14}$$

In the above mentioned study as well as in a subsequent one by Catto and Prata [8], where the model of Lawton in the form of Eq. (14) was compared to numerical simulations, the latter achieved the best agreement with the respective references. However, seeing that the second term factor in the model changes by an order of magnitude between a motored engine and a refrigerating compressor it is obvious that the model coefficients must always be determined for the machine in question, as already remarked earlier in the present paper.

Numerical Methods

Strictly speaking, this is not a class of cylinder heat transfer models, for the modelling proceeds by discretizing the compression space by means of any of the standard methods, i.e. finite volumes, differences, or elements, and numerically integrating the equation system thus obtained in space and time. Heat exchange between the gas and compression chamber walls is then treated as a part of the boundary conditions that must be specified in order to be able to perform the numerical integration of the governing equations. The collection of these methods is customarily referred to as Computational Fluid Dynamics (CFD).

According to Recktenwald et al. [40] this approach was first applied to the modelling of piston-cylinder configurations by Gosman and Watkins [15] in 1977. However, the work of Chong and Watson [9] precedes the latter. Recktenwald developed further the model of Gosman and Watkins and applied it to a reciprocating compressor. His results were already mentioned in the discussion of the model of Adair et al. [1] of this paper.

The potential of the numerical methods for predicting heat transfer in oscillating flow was quickly recognized [18], and they have enjoyed an intensive development ever since. Judging by the numerous papers on the subject of numerical cylinder modelling presented every year at the Purdue Compressor Conference this approach seems to be employed rather in the research than in the industrial practice. This is understandable, for it is impracticable and indeed not necessary to model every detail of the cylinder in order to achieve the prediction accuracy that is satisfactory for most of the projects being routinely dealt with in industry. Using CFD still requires qualified engineers and considerable computer resources, in terms of both the software and the hardware; and the time needed to obtain the results and interpret them is also an important factor. As in some other industry branches, the calculation and sizing engineers need to be able to compare quickly several alternatives in a project, which means that the run time is an important property of the performance prediction software, see e.g. [35]. Generally, there is a strong correlation between the model's spatial dimensionality and the software run time.

On the other hand, it is also true that the accuracy attainable with zero-dimensional cylinder models is in some cases no longer sufficient for meeting the requirements stipulated by the customers. The approach chosen in the joint EFRC project [2] whereby the spatial dimensionality of the model is increased in accordance with the cylinder complexity is a good example of finding a compromise between the modelling effort and the accuracy required.

SIMULATION RESULTS

As the present survey shows, cylinder heat transfer has been treated in a large number of theoretical and experimental studies carried out both in the industry and academia. However, with the exception of gas springs and Stirling engines, heat transfer measurements are seldom reported. Present authors are thus fortunate to have been provided with an internal report from Burckhardt Compression Ltd., documenting the results of cylinder heat transfer measurements carried out in 2008 with a double-acting, labyrinth-piston reciprocating compressor 2K90-1A. Main machine data are summarized in Table 1 below.

Table 1 *Principal data of the 2K90-A compressor*

Cylinder diameter	0.22 m
Stroke	0.09 m
Connecting rod ratio	0.18
Upper piston side clearance	1.6 mm
Valves	110K12, metal plate
Nominal motor speed	995 rpm
Suction pressure	Barometric

Heat exchange measurements were made with a commercial heat flux sensor Vatell HFM-7E/L at three locations: at the cylinder wall (15 mm from the cover), and at the cover, at a distance of 45 mm from the suction and delivery valve slits, respectively. The sensor measures also the temperature of its housing, i.e. of the metal surface surrounding the tip. Heat flux is measured with a first-order response time of 6 µsec.

The report contains the heat flux data acquired at a pressure ratio of 4:1, i.e. at a delivery pressure of approx. 4 bar. Large differences were observed between the heat fluxes at the two cover positions in the high-temperature parts of the process, i.e. in the discharge and back-expansion phases, the peak values reaching 85 kW/m^2 and 140 kW/m^2 at the suction and discharge valves, respectively. The heat fluxes recorded in the low-temperature phases of the compressor cycle at all three sensor positions were approximately the same.

The heat transfer model that produced the best agreement in terms of global machine results between predictions and measurements in the paper of Fagotti and Prata [12], namely the one proposed by Lawton [24], was chosen for comparison with the measurements; it is believed that this is the first comparison of this model with unsteady heat transfer data acquired in a piston compressor. Since the measurement report does not contain all information needed to formulate the simulation input data, the missing values were estimated. For example, the valve data were estimated on the basis of the information quoted in [3], since the same machine was also used in the latter project.

Referring to the LHS plot of Fig 2, there is a small phase difference between the calculated heat flux at the cylinder wall and the measured data. Assuming the same phase between the two signals, the magnitudes would agree very well in the expansion part of the process, although the sensor is fully or partly masked by the piston during the first 45 degrees on each side of the UDP. The agreement is worse in the compression and discharge phases.

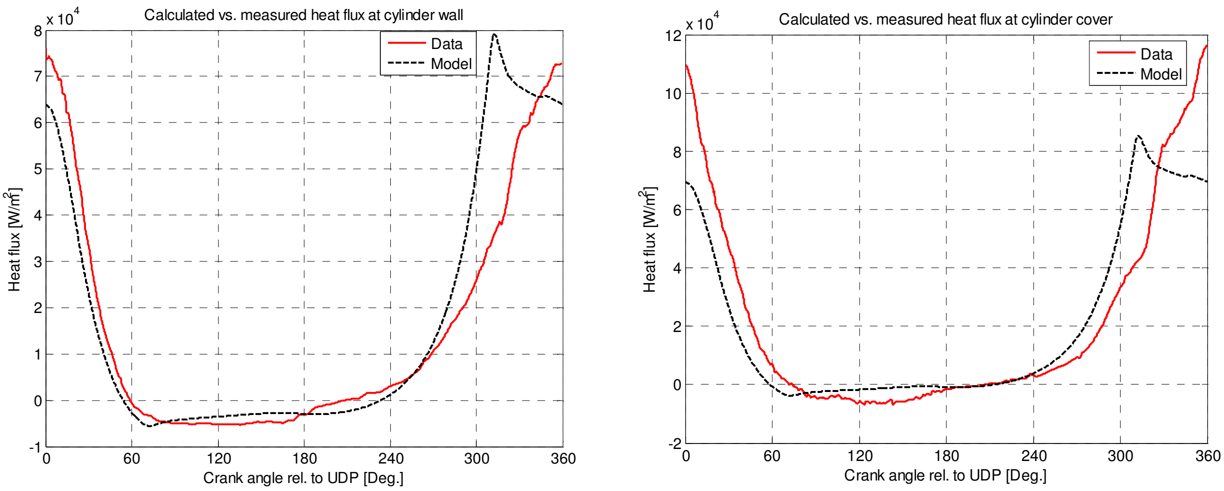

***Figure 2** Comparison of the model of Lawton [24] with measured heat flux data*

In the RHS plot of Fig. 2, the predicted wall heat flux is compared to the arithmetic mean of the heat fluxes measured at the suction and discharge valve locations. The phase difference is somewhat larger than in the case of the wall heat flux, but the main disagreement in magnitude is observed when the piston is very close to the UDP. Apparently, the mean piston velocity used to calculate the Reynolds number in the Lawton's formula is not representative of the gas velocity in this part of the process. The latter is mainly determined by the valve action, whose simulation is dependent on the respective models, but also on the knowledge of the valve data. In any case, since the measurement report does not contain the indicator diagram it is difficult to speculate on the reasons for the disagreement between the prediction and the measurement results at the UDP. One should also recall that a labyrinth-piston compressor was used as the test machine, the performance of which (especially the gas temperature) is strongly dependent upon the piston blow-by [33]. Unfortunately, the piston leakage could only be very crudely approximated in the present simulation model.

CONCLUSIONS

The survey shows that there is still the need for improvement of the cylinder heat transfer modelling for the performance prediction of reciprocating compressors. For reasons of run time and flexibility, the approach must still be based on relatively simple cylinder models, integrated with neighboring attachments and piping into compressor stage models. This has a potential for achieving a better accuracy in the prediction of valve action, which in its turn improves not only the prediction of heat transfer, but also the overall performance prediction.

Based on the comparison with limited data available to the authors, the simple unsteady model of Lawton [24] is apparently capable of predicting the cylinder heat transfer with an acceptable accuracy. However, more measurement data are required before a fully qualified statement as to its general utility can be made. The authors hope to obtain these on the test rig for small air compressors in the Engine Laboratory of the Faculty of Engineering of Kragujevac, which is currently being brought into operation.

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