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ABSTRACT

This study first reviews, discusses, and compares existing correlations for heat transfer in reciprocating compressors and engines. Then after a discussion of the significance of obtaining accurate instantaneous heat transfer data, the authors describe an experimental investigation of the instantaneous heat transfer rates to the cylinder wall of a reciprocating refrigerating compressor. Finally, this report presents an expression which correlates the time averaged data within 20%.

INTRODUCTION

Previous attempts at describing and predicting the heat transfer occurring in reciprocating engines and compressors have in some cases been moderately successful and in others completely unsuccessfuly. Most of the reported investigations have attempted to predict the instantaneous heat transfer from measured average heat transfer rates [1 through 5]. More recent investigations, however, have been concentrating on the problems of measurement and correlation of instantaneous heat transfer rates [6 through 10]. All these investigations were carried out on internal combustion engines and, to the authors' knowledge, there has been no previous measurement of instantaneous heat transfer rates in reciprocating compressors. Thus, this investigation was undertaken with the hope of providing additional insight into the fundamentals of heat transfer processes in reciprocating machinery.

The present investigation, which is part of a more extensive research program [11], begins with a review and discussion of the presently available heat transfer correlations. A comparison of these correlations exposes their basic characteristics and differences and is followed by a prediction of the maximum errors introduced when the estimation of compressor efficiencies is based upon a poor knowledge of the instantaneous heat transfer rates. Finally, experimental methods and results are presented together with a correlation of the data obtained.

REVIEW OF AVAILABLE HEAT TRANSFER CORRELATIONS

The available correlations on cylinder heat transfer have all been developed for internal combustion engines and they can be divided into two main groups according to the choice of variables. Representative for the first group are the correlations by Nusselt [1], Eichelberg [2], and Pflaum [3]. In this group the heat transfer coefficient chosen is of the following form:

$$h(t) = f[V_p, P_g(t), T_g(t)] \qquad (1)$$

where t denotes the instantaneous value of a quantity. Representative of the second group are Woschni [6], Annand [4], Sitkei [5] and LeFeuvre [7]. This group utilizes correlations of the type

$$Nu(t) = f[Re(t), \text{gas properties}]. \qquad (2)$$

All the correlations in these two groups include constants that have to be determined experimentally and are, therefore, empirical in the first group and semi-empirical in the second.

The correlations are given below.

First group:

Nusselt:

$$h(t)=.0278(1+.38\ V_p)[P_g(t)^2 T_g(t)]^{1/3} \qquad (3)$$

Eichelberg:

$$h(t)=.0565\ V_p^{1/3}[P_g(t)T_g(t)]^{1/2} \qquad (4)$$

Pflaum:

$$h(t)=.0399[P_g(t)T_g(t)]^{1/2}$$

$$[6.2-5.2(5.7)^{-(.0305\ V_p)^2}$$

$$.00762\ V_p][1.175\ (P_s)^{1/4}] \tag{5}$$

The dimensions of the parameters appearing in these equations are $h(Btu/(hr\ ft^{2\circ}R)$, $V_p(ft/sec)$, $P(psia)$, $T_g(^\circ R)$ and $P_s(psia)$.

The constants and exponents have been determined from average heat transfer measurements on internal combustion engines.

Second group:

Woschni:

$$h(t)=\text{Constant X }[\frac{k(t)}{D}][Re(t)]^{0.7} \tag{6}$$

where

$$Re(t)\equiv\frac{\rho(t)(2.28\ V_p)D}{\mu(t)} \tag{7}$$

Annand:

$$h(t)=\text{Constant X }[\frac{k(t)}{D}][Re(t)]^{0.8} \tag{8}$$

where

$$Re(t)\equiv\frac{\rho(t)V_pD}{\mu(t)} \tag{9}$$

Sitkei:

$$h(t)=\text{Constant X }[\frac{k(t)}{D_e(t)}][Re(t)]^{0.7} \tag{10}$$

where

$$Re(t)\equiv\frac{\rho(t)V_pD_e(t)}{\mu(t)}$$

and

$$D_e(t)\equiv\frac{2DS(t)}{2S(t)+D} \tag{12}$$

LeFeuvre:

$$h(t)=\text{Constant X }[\frac{k(t)}{D}][Re(t)]^{0.8}$$

$$[Pr(t)]^{0.333} \tag{13}$$

where

$$Re(t)\equiv\frac{\rho(t)(D/2)^2\omega_g}{\mu(t)} \tag{14}$$

Although these correlations too contain empirical constants, there are two notable advantages. Sitkei's correlation contains a time-varying equivalent diameter in the Reynolds number, and LeFeuvre's correlation contains a characteristic swirl velocity rather than the average piston speed. Sitkei's modification will predict the notably higher heat transfer when the piston moves towards the top-dead-center position. LeFeuvre's swirl velocity permits the incorporation of artificially empirically introduced squish and swirl into the correlation.

Next the above correlations can be compared for possible use in predicting the heat flow in a refrigerating compressor by using a simple mathematical model. The cylinder is treated as an open system with uniform properties throughout. The manifold pressures are held constant. The valves open instantaneously to fully open when the pressure drop across them exceeds zero and close instantaneously when the pressure drop returns to zero. The empirical constants in the heat transfer correlations are all normalized to give the same total heat transfer over a cycle. Comparing the different correlations sheds light on differences in their basic natures. Figure 1 presents one such set of curves where the heat transfer in the compressor has been calculated according to the seven correlations for one set of operating conditions. The empirical constants were normalized at this operating point and the average heat transfer is, therefore, the same for all correlations. The average-over-a-cycle of the instantaneous heat transfer coefficients are not equal for the seven correlations because of variations in the wall-to-fluid temperature differences which are functions of both crank angle and the particular correlation.

This method was used to predict the heat transfer at other operating conditions, and Figure 2 gives the resulting curves when the compressor speed is doubled.

These two sets of curves show that as RPM increases there is a pronounced increase in the peaking of the curves near top dead center, due to the decreased time for discharge and hence an increase in pressure.

INSTANTANEOUS HEAT TRANSFER RATES

General:

All the previously described correlations will predict heat transfer into the cylinder walls when the average gas temperature exceeds the wall temperature. In reality because of unsteady conditions and capacitive effects of the boundary layer, the temperature of the gas in the boundary layer close to the cylinder wall may differ significantly from the steady state distribution and the heat transfer may actually be in the reverse direction of what is predicted by the existing correlations. In order to get an idea of how important it is to know the time variations of the heat fluxes, a computer code was used to calculate the compressor efficiency for four assumed thermodynamic cyclic processes, all with the same average heat transfer. The four arbitrary processes used were:

1) Heat flow during expansion and suction only,
2) Heat flow during compression only,
3) Heat flow during exhaust only,
4) Heat flow over the whole cycle, and
5) Heat flow calculated by LeFeuvre's correlation.

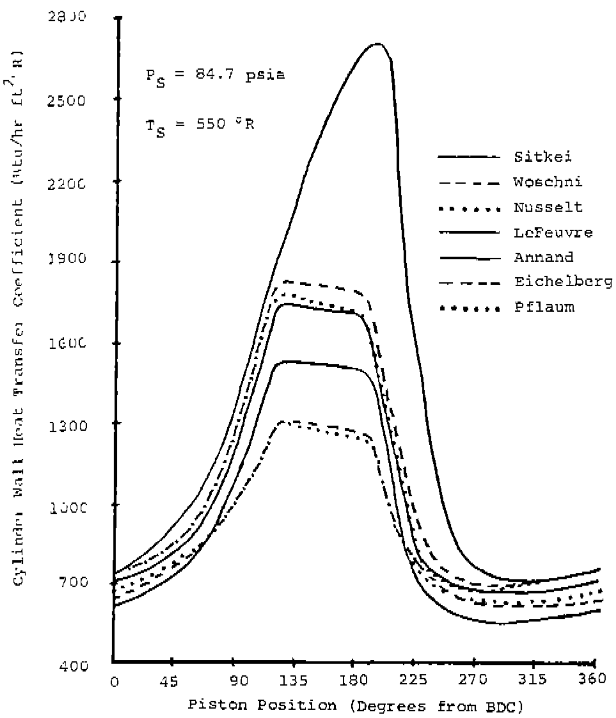


Figure 1 Instantaneous Heat Transfer Coefficients Using Seven Correlations at the RPM of the Experimental Data

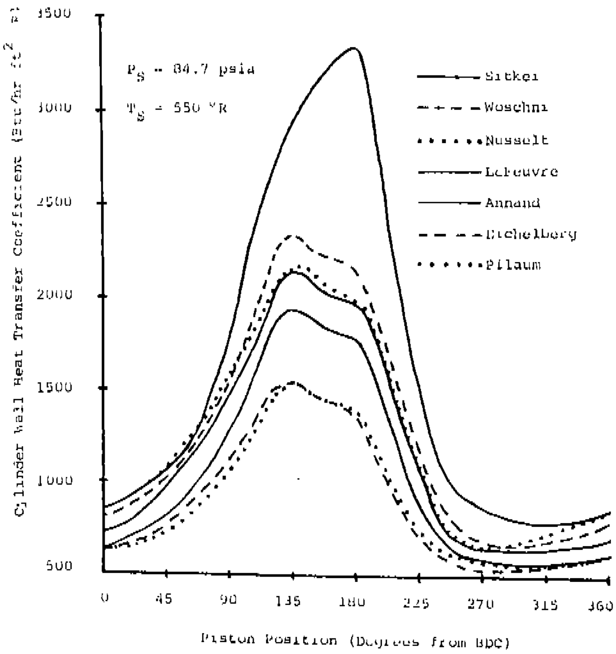


Figure 2 Instantaneous Heat Transfer Coefficients Using Seven Correlations at Twice the RPM of the Experimental Data.

The results given in Table 1 indicate the importance of knowing the heat transfer rates accurately. Since this knowledge is even more important for the calculation of the local, instantaneous heat flux, one must apply existing information to different operating conditions and to new or different compressors carefully.

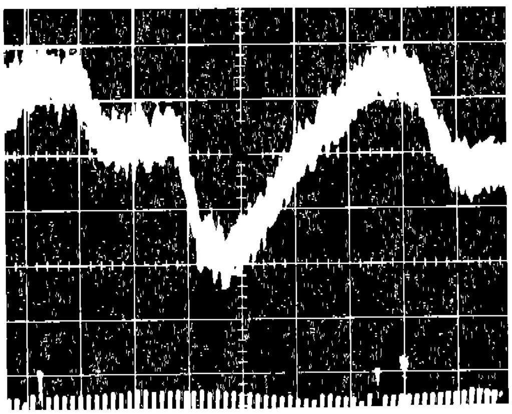
Experimental Investigation:

The present investigation was carried out on a three-cylinder open compressor operating at approximately constant RPM. Time averaged temperatures of the cylinder wall and piston surfaces were recorded at a number of locations. The instantaneous heat flux was calculated from recordings of the output from a fast response surface thermocouple mounted in the cylinder head (Medtherm Corp., Huntsville, Alabama). The amplified signal from the thermocouple was displayed on an oscilloscope, and Figure 3 shows a typical trace. It is important to note that the cylinder wall temperature varies less than $\pm$ 1 ^{o}R from its time averaged value. Variations in the cylinder gas temperature over a cycle are of the order of $\pm$ 100 ^{o}R. Therefore, when considering the instantaneous value of (T_g-T_w) in the calculation of h(t), it is safe to assume T_w = constant.

TABLE 1 THE EFFECT OF FIVE HYPOTHETICAL CYCLIC HEAT TRANSFER RATE VARIATIONS ON COMPRESSOR PERFORMANCE

	η_A	$T_{g_{BDC}}$ (°R)	$P_{g_{BDC}}$ (psia)
Heat Transfer During:			
Expansion & Suction	96.76%	577.7	84.6
Compression	97.89%	566.5	84.5
Exhaust	98.23%	575.1	84.4
Full Cycle	97.53%	571.9	84.4
LeFeuvre Correlation	95.53%	577.6	84.5

Approximately 0.5 °R per cm



Approximately 5 msec per cm

Figure 3 Typical Oscilloscope Trace of Dynamic Response of Cylinder Wall Surface Temperature Probe.

There was considerable cycle-to-cycle scatter, and therefore, 20 traces were averaged at each operating condition. The resulting average trace was then analyzed by a standard Fourier Analysis to arrive at the instantaneous values of the heat flux.

At this point an acceptable correlation to accomodate the recorded data must be developed. Attempts to adapt the existing correlations of Woschni, Annand, and LeFeuvre to the results were without much success. Following Sitkei's technique of introducing the variable equivalent diameter improved the agreement; however, it became clear that the crucial quantity in a successful correlation would be the characteristic velocity. Shipinski [9] suggests that the swirl velocity is approximately twice the angular velocity of the crankshaft. Knowing this information together with the fact that the velocity decays quite rapidly after closing the suction valve, we can approximate the swirl velocity by

$$\omega_g(t)=\begin{cases} 2\omega[1.04+\cos(2\theta)] & \text{for } \frac{3}{2}\pi<\theta<\frac{1}{2}\pi \\ 2\omega[\frac{1}{2}][1.04+\cos(2\theta)] & \text{for } \frac{1}{2}\pi<\theta<\frac{3}{2}\pi \end{cases} \tag{15}$$

If we define the Reynolds number by

$$Re(t)\equiv\frac{\rho(t)D_e(t)[\frac{D_e(t)}{2}\omega_g(t)]}{\mu(t)} \tag{16}$$

where

$$D_e(t)=\frac{6\ \text{Volume}}{\text{Area}}=\frac{6\pi(\frac{D}{2})^2 S(t)}{\pi D S(t)+2\pi(\frac{D}{2})^2} \tag{17}$$

and choose a heat transfer correlation of the form

$$Nu(t)\equiv\frac{h(t)D_e(t)}{k(t)}=A[Re(t)]^{0.8}[Pr(t)]^{B} \tag{18}$$

we arrive at values of A=0.053 and B=0.6 as the best fit for the data. Recasting this equation in terms of the instantaneous cylinder wall heat flux one obtains

$$\dot{q}''_w=0.053[\frac{k(t)}{D_e(t)[T_g(t)-T_w]}][Re(t)]^{0.8}$$

$$[Pr(t)]^{0.6} \tag{19}$$

Unfortunately, this correlation cannot predict the previously described instantaneous reversed heat flux, but a study of Figures 4,5,6 and 7 and a comparison of the experimental and the predicted heat flux using the above correlation indicates that the agreement is quite good. None the less, instantaneous measured and predicted values of heat flux at top dead center are quite far apart.

Other more complex correlations were considered: for example,

$$h(t)=\text{Constant X }[\frac{\rho(t)}{\rho_c}]^a\left|\frac{T_g(t)-T_w}{T_g(t)}\right|^b$$

$$[Re(t)]^c[Pr(t)]^d$$

However, none produced a better agreement for the overall or average heat flux than the simpler one presented above.

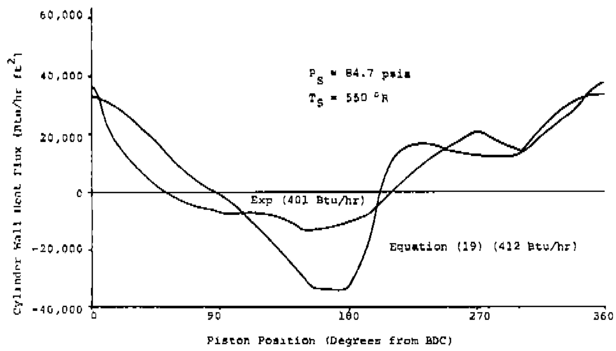


Figure 4 Comparison of Experimental Instantaneous Heat Flux and the Correlation Equation.

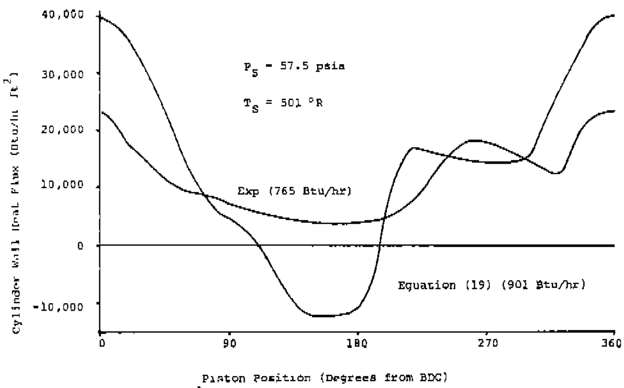


Figure 5 Comparison of Experimental Instantaneous Heat Flux and the Correlation Equation.

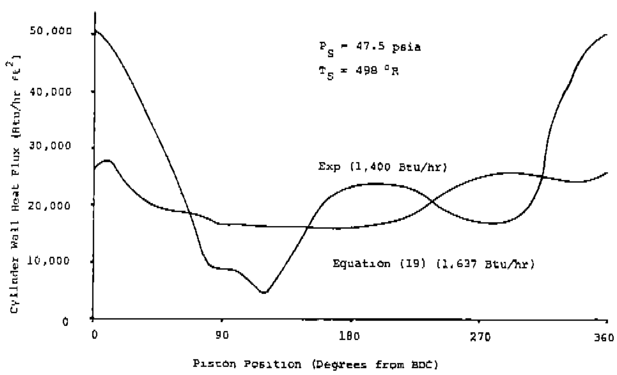


Figure 6 Comparison of Experimental Instantaneous Heat Flux and the Correlation Equation.

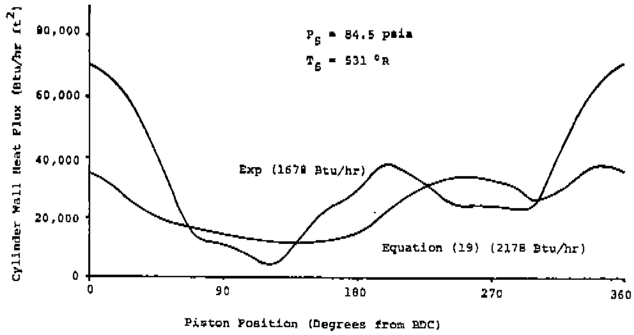


Figure 7 Comparison of Experimental Instantaneous Heat Flux and the Correlation Equation.

CONCLUSIONS

This investigation reviewed, discussed and compared the existing correlations for heat transfer in reciprocating compressors and engines. These correlations can be divided according to their basic forms into two groups:

$$h(t)=f[V_p, P(t), T_g(t)] \quad (1)$$

and

$$Nu(t)=f[Re(t), \text{gas properties}] \quad (2)$$

The primary importance of an accurate knowledge of the instantaneous heat transfer rates as opposed to average rates lies not in the prediction of the compressor efficiency which can error approximately 3 percent, but rather in the understanding of the basic mechanisms of heat transfer which can account for as much as 10 to 20 percent decrease in compressor volumetric and thermodynamic efficiencies.

The experimental data on the time averaged heat transfer is correlated within $\pm$ 20 percent by the expression

$$Nu(t)=0.053[Re(t)]^{0.8}[Pr(t)]^{0.6} \quad (21)$$

ACKNOWLEDGEMENT

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NOMENCLATURE

D = Diameter of cylinder

D_e = Equivalent or hydraulic diameter

f = Denotes functional relationship

$h(t)$ = Instantaneous heat transfer coefficient $h(t) \equiv \dot{q}''_w(t)/(T_g(t)-T_w)$

k = Thermal conductivity of gas in cylinder

Nu = Nusselt number

P_g = Cylinder gas pressure

P_D = Discharge plenum pressure

P_s = Suction plenum pressure

Pr = Prandtl number

$\dot{q}''_w$ = Cylinder wall heat flux

RPM = Revolutions per minute

Re = Reynolds number

S = Piston to cylinder lead distance

t = Time, denotes instantaneous value. Quantities without instantaneous notation have values averaged over a cycle.

T_g = Cylinder gas temperature

T_s = Temperature of gas suction plenum

T_w = Temperature of cylinder wall

V_p = Mean piston speed

η_A = Adiabatic efficiency

θ = Crank angle from bottom dead center

ρ = Density of gas in cylinder

ρ_c = Reference density of gas for cycle

μ = Viscosity of gas in cylinder

ω = Angular crankshaft speed

ω_g = Swirl velocity of gas in cylinder

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