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ABSTRACT

Exergy analysis can quantify defects in compressor rational efficiency and attribute them to specific regions and causes. Improvements in compressor design can be based on this information. An exergy analysis methodology is presented for a reciprocating compressor. The required information about internal processes is obtained from a simulation program. The methodology applies equally where data on internal processes are gathered by measurement.

NOMENCLATURE

Roman Letters

b specific flow exergy function ($b = h - T_o s$)
C velocity
m mass
$\dot{m}$ mass flow rate
g acceleration due to gravity
h specific enthalpy
$\dot{I}$ exergy destruction rate
$\dot{Q}$ heat transfer rate
p pressure
T absolute temperature
T_o absolute temperature of the environment
t temperature relative to the freezing point of water
s specific entropy
V volume
W work
$\dot{W}_c$ shaft power input to the compressor
y valve lift
z elevation

Greek Letters

δ_η total rational efficiency defect
η rational efficiency
$\dot{\Xi}_c$ exergy transfer rate to the fluid being compressed
$\dot{\Xi}_Q$ exergy transfer rate associated with heat

General Subscripts

d discharge state
e exit
h high
i inlet
l low
s suction state

INTRODUCTION

Exergy is the potential for useful work from the interaction of a system with a specified environment. It is conserved only in thermodynamically ideal plant. Inevitably some exergy is destroyed in compressors. In exergy analysis, the overall performance parameter is the rational efficiency, which is defined below. It is common for compressors to have rational efficiencies of about fifty percent (1, 2). There is a possibility, according to the laws of thermodynamics, of reducing the required power input for a given duty very considerably. Any improvements made would have important economic benefits.

OVERALL EXERGY ANALYSIS

For a compressor that operates within a particular environment the rate of exergy transfer to the compressed fluid can be evaluated from measurements of the mass flow rate, thermodynamic properties at suction and discharge, and the temperature of the environment. It is given by

$$\dot{\Xi}_c = \dot{m}(b_d - b_s). \tag{1}$$

The total exergy destruction rate is the difference between the exergy input rate and the rate of exergy transfer to the fluid stream. The exergy input rate is equal to the shaft power, which can be measured. Hence

$$\dot{I} = \dot{W}_c - \dot{\Xi}_c. \tag{2}$$

The rational efficiency is the rate of exergy transfer to the fluid divided by the rate of exergy input as shaft power.

$$\eta = \dot{\Xi}_c / \dot{W}_c \tag{3}$$

The difference between unity and the measured rational efficiency is the total rational efficiency defect. It is a decimal fraction representing the "rational inefficiency" of the compressor.

$$\delta_\eta = 1 - \eta \tag{4}$$

DETAILED EXERGY ANALYSIS

Detailed information about internal processes is required to localise and quantify exergy destruction or rational efficiency defects. This type of information can be obtained from measurements. However, it is impossible to describe fully the internal processes in terms of measurements alone: a combination of measurements and hypotheses about the nature of the processes must be used. This involves the description of a somewhat simplified model of the compressor and its processes. As the level of detail of this model is increased, so the detail of the exergy analysis increases. The best results can be achieved by combining internal measurements with detailed computer simulation of a compressor. With this approach it is possible to develop a clear and quantitative description of the causes and locations of exergy destruction. The total rational efficiency defect can be apportioned to these causes and locations. References 3 and 4 describe exergy analysis and include a "toolbox" of expressions—only essential details are given here.

Some Relevant Exergy Expressions

Heat transfer with finite temperature difference is one mechanism of exergy destruction in compressors. This occurs within the fluid being compressed, within the material of the compressor itself, and between bodies of fluid and the material of the compressor. The exergy transfer associated with heat at a boundary over which temperature is uniform is given by the following expression:

$$\dot{\Xi}_Q = \frac{T - T_o}{T}\dot{Q} \tag{5}$$

In the above expression, T is the absolute temperature at the boundary. When steady heat transfer occurs through a finite temperature difference the exergy output from the region of heat transfer is always less than the exergy input. For example, if steady heat transfer occurs through a region where the high temperature boundary is at T_h and the low temperature boundary is at T_l the exergy destruction rate within the region is given by

$$\dot{I} = \left(\frac{T_h - T_o}{T_h} - \frac{T_l - T_o}{T_l}\right)\dot{Q}$$

$$= \left(\frac{1}{T_l} - \frac{1}{T_h}\right)T_o\dot{Q}. \tag{6}$$

If the temperature at which heat transfer occurs varies over the boundary, an integral form of equation 5 must be used as follows:

$$\dot{\Xi}_Q = \int \frac{T - T_o}{T} d\dot{Q}. \tag{7}$$

As well as being transferred in association with work or heat, exergy can be transported with material that crosses the boundary of a system. For a given system an exergy balance takes the following form:

Net rate of exergy transfer and transport into a system	=	Rate of increase of exergy of the system	+	Rate of exergy destruction within the system.	(8)

By combining an energy balance and an exergy balance it can be shown that the exergy destruction rate within a system is given by the following general expression:

$$\dot{I} = \sum_i \dot{m}_i\left(-T_o s_i + \frac{C_i^2}{2} + gz_i\right) - \sum_e \dot{m}_e\left(-T_o s_e + \frac{C_e^2}{2} + gz_e\right) - \int \frac{T_o}{T}\dot{Q} + T_o\left(\sum_i \dot{m}_i - \sum_e \dot{m}_e\right)s + T_o m\dot{s}. \tag{9}$$

In equation 9 the usual sign convention for heat is assumed; i.e., heat transfer is positive when to the system. This equation gives the total exergy destruction rate due to all causes within a specified boundary. In order to evaluate it, all mass flow rates entering or leaving the system must be identified and the intensive thermodynamic states must be known where the streams cross the boundary. The specific kinetic and potential energy of each of the streams must be known at the boundary. The heat transfer rate over the boundary must be known in such a way that equation 7 can be evaluated. The mass and the intensive thermodynamic state throughout the system must be known—the latter, particularly, requires assumptions about the state of the system even when some measured data are available. The temperature of the environment must also be known.

Evaluation of exergy destruction within a compressor by means of equations 7 and 9 may seem a daunting task. However, the purpose of this paper is to show that this can realistically be done for a compressor by making appropriate assumptions. The results that can be obtained justify the effort.

COMPRESSOR MODEL DESCRIPTION

Figure 1 is a schematic representation of a compressor for which an exergy analysis is presented. In figure 3 five fluid regions within the compressor are identified for analysis: the first four are the suction plenum, discharge plenum, cylinder, and crankcase. The fifth is a volume surrounding the discharge pipe flow restriction and extending downstream of it within the discharge pipe. The main flow path is from the suction pipe through the suction plenum to the cylinder and from there through the discharge plenum and the discharge flow restriction region into the discharge pipe. In this path there are four flow restrictions. Those at entry to the suction plenum and at exit from the discharge plenum are modelled as simple orifices. The suction and discharge valves are modelled as orifices of variable area; in conjunction with valve dynamics modelling. There is also some mass flow between the cylinder and the crankcase due to leakage in both directions past the piston. This is modelled as flow through an equivalent orifice. For analysis purposes it is assumed that all exergy destruction associated with an orifice occurs downstream of it. If flow reverses, as in the case of leakage past the piston, the exergy destruction will occur on different sides of the orifice, depending on the flow direction.

The compressor body is assumed to consist of four metal elements. The temperature of each is represented by a single value. It is assumed there are simple thermal resistances where these elements connect. The heat transfer paths are represented schematically in figure 4. The effects of friction between moving parts and windage are not taken into account.

ANALYSIS RESULTS

Results were obtained by adapting a conventional simulation program to the requirements for exergy analysis of the hypothetical compressor that has been described. The fluid was R-12, treated as an ideal gas. Simple and somewhat arbitrary expressions for variations in internal heat transfer coefficients were used. An energy balance approach was adopted in determining the successive states of the gas volumes. Experience with a particular compressor was drawn upon, as a rough guide, in specifying the input data of the program. As the purpose was only to demonstrate the principle of undertaking exergy analysis within a compressor, the model was crude. The results are subject to the assumptions made.

Figure 5 shows the pressures within the suction and discharge plenums, the cylinder, and the crankcase, plotted against crank angle. Figure 6 illustrates the temperatures within the same regions. Valve lift is plotted in figure 7 for the suction and discharge valves.

Figure 8 shows the rates of exergy destruction within the seven constituent analysis regions of the compressor. These are the suction plenum, the discharge plenum, the cylinder, the crankcase, the discharge pipe restriction region, the metal mass, and the region just outside the outer surface of the compressor, where exergy destruction occurs due to heat transfer to the surroundings. The total exergy destruction rate is also shown in this figure. The causes of exergy destruction that are taken into account are fluid throttling, mixing of throttled fluid with a downstream body of fluid, heat transfer within the metal body, and heat transfer between metal surfaces and fluid. It turns out that the exergy destruction due to heat transfer is small: the more significant of the instantaneous exergy destruction rates due to heat transfer are shown in figure 9—the values are plotted within the range 0 to 60 watts, whereas the values in figure 8 (which include all the data in figure 9) are plotted within the range 0 to 1500 watts (except in two regions where exergy destruction is due to heat transfer only).

The average rate of exergy transfer to the stream of fluid is 447.7 watts and the average total exergy destruction rate 236.54 watts. The average power input is the sum of these two quantities: 684.24 watts. (Calculation of the average power by integration of the computed $p-V$ diagram yields 684.9 watts.) The rational efficiency is thus 0.654 and the total rational efficiency defect 0.346. Figure 2 illustrates the composition of the total rational efficiency defect.

CONCLUSIONS

A methodology has been presented and applied that allows the thermodynamic shortcomings of a compressor to be quantified and assigned to particular regions. This is a new approach in the field of compressor analysis. It has considerable potential as a means of compressor design optimisation and, thus, for economic benefits.

ACKNOWLEDGEMENT

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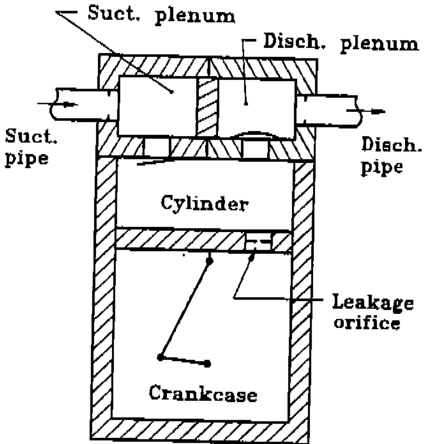

Fig. 1 Schematic representation of the compressor model.

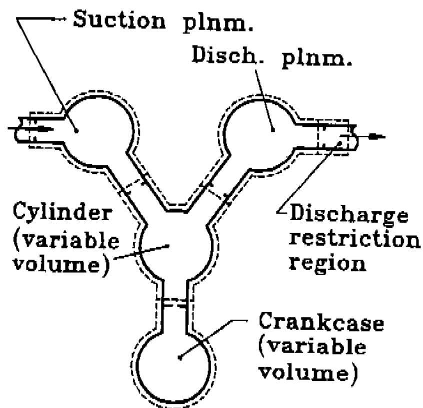

Fig. 3 Fluid analysis regions within the compressor.

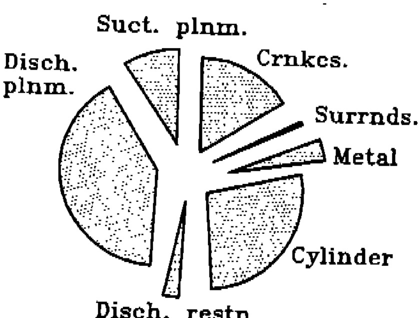

Fig. 2 Composition of the total rational efficiency defect.

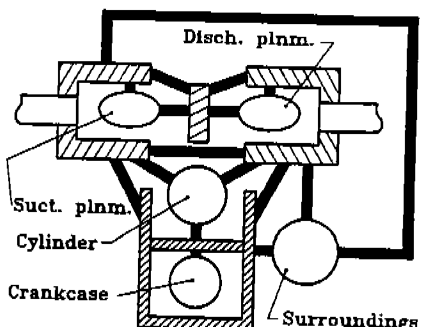

Fig. 4 Heat transfer interactions between compressor elements, fluid volumes, and the surroundings.

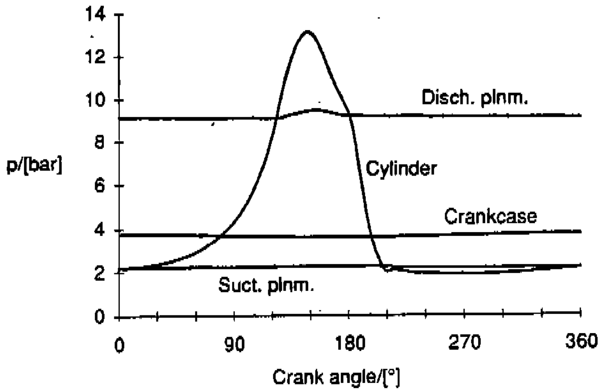

Fig. 5 Pressure versus crank angle diagram.

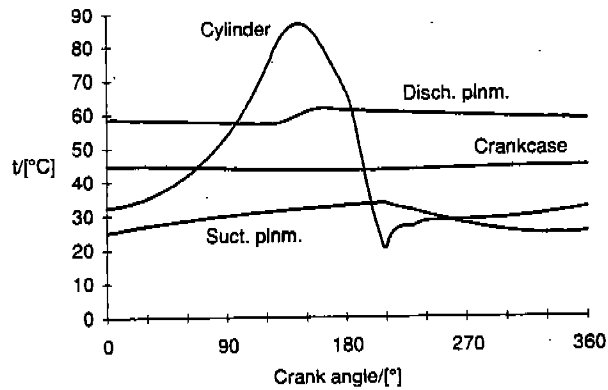

Fig. 6 Temperature versus crank angle diagram.

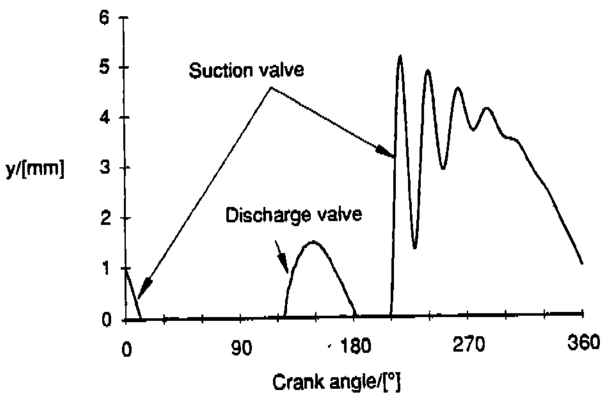

Fig. 7 Valve lift versus crank angle diagram.

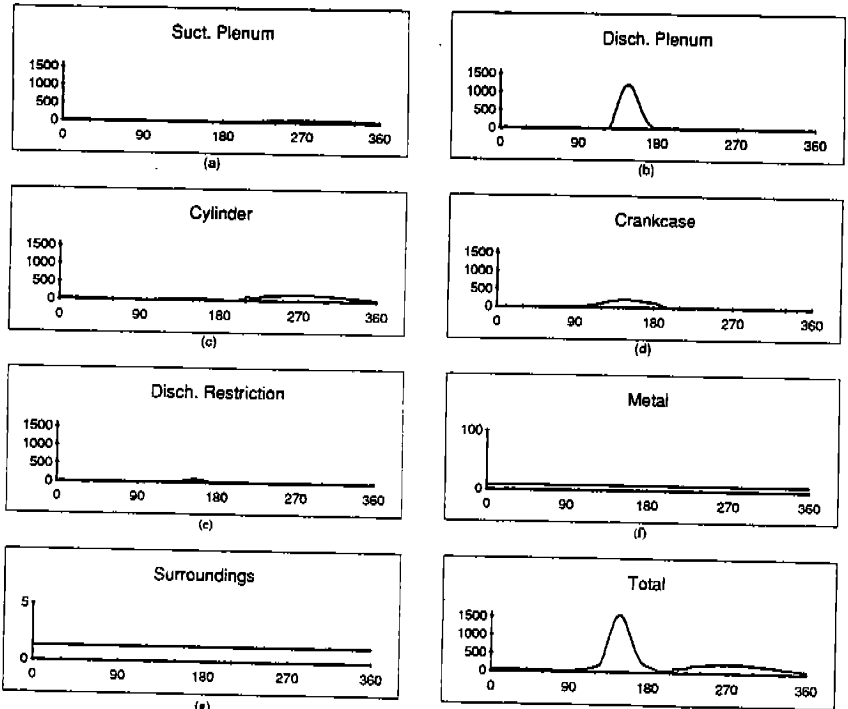

Fig. 8 Graphs (a) to (g) represent the exergy destruction rates (watts) due to all causes in all regions versus crank angle (°). Graph (h) shows the total exergy destruction rate of the compressor due to all causes—this is the sum of the values in the previous graphs.

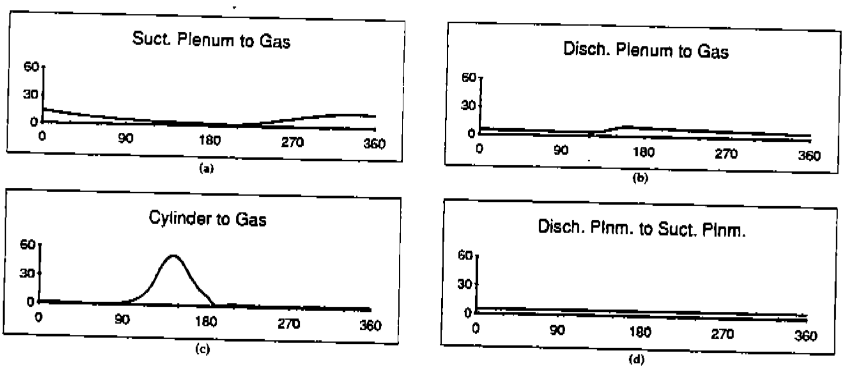

Fig. 9 Some of the exergy destruction rates (watts) due to heat transfer only.