On Refrigerant Compressors

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DECLARATIONS

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__________________________________________
James A. McGovern
The purpose of the work was to critically re-examine and investigate the evaluation of refrigerant compressors and to determine and discriminate between the factors which influence their characteristics. The objectives also included the investigation of techniques by which the characteristics can be established and the suggestion of ways in which compressor performance can be described and quantified.

The particular compressor which was tested was of the reciprocating open type.

Existing theories relating to the efficiency of displacement utilisation are extended to yield an overall framework in which the various influences can be described and quantified.

Conventional parameters for the quantification of shaft power utilisation are reviewed critically and alternatives are presented and justified. These alternatives are rational efficiencies based on the Second Law of Thermodynamics.

Load stands and specialised measurement techniques for testing refrigerant compressors are reviewed as background to the experimental work which was carried out. A design rationale for the load stand which was built and used for compressor testing is presented.

The load stand was a bypass type which involved full condensation of part of the flow and a mixing process. The operating conditions of the compressor were controlled by means of throttle valves only.

Data illustrating the main characteristics of the compressor which was tested, i.e. mass flow rate, shaft power, discharge temperature and oil concentration in the discharge vapour, are presented over a range of speeds and over a range of suction superheat values.

In addition, many parameters of the compressor which underlie its performance characteristics are presented graphically versus speed and versus suction temperature. Dynamic measurements of cylinder pressure and valve displacements are presented, analysed and discussed.

The displacement utilisation efficiency remained almost
constant at about 66% over the speed range. However, the proportions of the losses in displacement utilisation, which were quantified discretely under five headings, varied significantly. One of these headings comprised heat transfer, leakage and solubility effects within the cylinder. The test data did not allow full discrimination between these and the measurement difficulties are discussed in detail. The concept of a ‘phantom mass’ of refrigerant which dissolves and re-emerges from solution in the lubricating oil is conjectured and discussed.

The indicated rational efficiency of the compressor is presented graphically. Data describing indicated work, the suction pumping work and the discharge pumping work are presented, as are data describing the overall heat transfer rate during compression and the time averaged plenum heat transfer rates. The evaluation of the irreversibilities associated with throttling and with heat transfer would require precise data on the mass of charge within the cylinder and on the temperature distribution within the charge throughout the compression process. There is clearly a need for further work on the experimental problems of temperature and mass measurement within the cylinder. It is concluded that thermodynamic availability analysis is essential to the understanding of the factors underlying shaft power utilisation.

The outputs of a compressor simulation model are compared with test results. The model was found to require the support of test data as it would not necessarily give accurate predictions otherwise.

While there was scope for improvements in design to the compressor, its characteristics were found to be favourable for the purposes of capacity control by speed variation.

In conclusion, new ways of looking at the utilisation of shaft power and of volume displacement are presented along with many other new ideas relating to the evaluation of refrigerant compressors.
DEDICATION

If I were a great literary author I would dedicate my greatest works to the one I love. If I were a poet I would write her a poem. If I were a composer I would write my finest symphony for her. To Stephanie my wife I dedicate this PhD thesis about refrigerant compressors. Masterpiece or not I know this dedication will mean as much to her as a beautiful bunch of flowers, given with love.
ACKNOWLEDGEMENTS

I wish to acknowledge the constructive support and thoughtful observations which I have had over the years while I have been working on the thesis from my supervisor Professor W.G. Scaife. I was assisted in the practical aspects of the work by the technicians of the Department of Mechanical and Manufacturing Engineering and particularly by Mr. Tom Haveron. Mr. Alan Reid, the Senior Experimental Officer of the Department, assisted with some of the design aspects. Some of my final year students over a number of years carried out projects in areas such as the development of computer programs for calculating refrigerant properties or simulating compressor performance, measurement of valve characteristics and the setting up of instrumentation. I am grateful to them for sharing in my learning experience. At the start of the work I was fortunate to receive a grant from the Trinity College Research Fund. This enabled me to purchase equipment to begin work in the area of refrigerant compressor performance testing, which was a new research area in the Department. I am grateful to my Head of Department, Professor David Taplin, for encouraging me to visit Waterloo University in Canada and Purdue University in the U.S. at the stage when I was finalising my research objectives for the thesis. The Dean of the Faculty at the time, Dr. R. Cox, also assisted me in making that trip. I am grateful to a number of people abroad who took time to show me their research facilities or spent time talking about our mutual research interests. These include Professors James Hamilton and Raymond Cohen of Purdue University, Professor Harry Sullivan of Waterloo University, Professor Horst Kruse of the University of Hannover, Professor Fritz Steimle and Dipl.-Ing. E.L. Schmidt of the University of Essen, Dr. Tony Tramschek of the University of Strathclyde and Mr. Adalbert Stenzel of Bitzer AG, Sindelfingen. I acknowledge also a grant which I received from the National Board for Science and Technology for a contract entitled "Microcomputer based automation of an existing refrigerant compressor load stand" and a travel grant from the British Council.
LIST OF SYMBOLS

A = effective flow area at position identified by subscript [m²]
    or
    refers to discharge line throttle valve
    or
    = sum of terms in Bambach's solubility equation

a = constant

A = rate of useful availability transfer provided directly by the compressor [W]

a, b, c,... refer to measurement locations on load stand
    or
    = constants
    or
    represent thermodynamic state points

aa, bb, .. refer to measurement locations on load stand

B refers to bypass throttle valve

b = specific flow availability function [J/kg]
    or
    = constant

bara = bar absolute (units of pressure)

BDC = bottom dead centre

b_{hosat} = specific flow availability function of saturated refrigerant, evaluated at T₀ [J/kg]

C refers to liquid line metering valve
    or
    = coefficient of discharge

c = velocity of sound [m/s]

C_c refers to coarse metering valve

CCW = counterclockwise

C_f refers to fine metering valve

c.o.p. = coefficient of performance

c_p = specific heat at constant pressure [J/kgK]

CW = clockwise

## List of Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>D</td>
<td>refers to suction line throttle valve or pipe diameter [m]</td>
</tr>
<tr>
<td>d</td>
<td>orifice diameter [m]</td>
</tr>
<tr>
<td>DEP</td>
<td>discharge mean effective pressure [N/m²] or [bar], as indicated</td>
</tr>
<tr>
<td>E</td>
<td>velocity of approach factor</td>
</tr>
<tr>
<td>$E_{es}$</td>
<td>electrical isentropic efficiency</td>
</tr>
<tr>
<td>$E_{ir}$</td>
<td>indicated compressor rational efficiency</td>
</tr>
<tr>
<td>$E_{ircr}$</td>
<td>indicated rational efficiency for compression and heat rejection</td>
</tr>
<tr>
<td>$E_{is}$</td>
<td>indicated isentropic efficiency</td>
</tr>
<tr>
<td>$E_{it}$</td>
<td>indicated isothermal efficiency</td>
</tr>
<tr>
<td>$E_{mech}$</td>
<td>mechanical efficiency</td>
</tr>
<tr>
<td>$E_{sr}$</td>
<td>shaft compressor rational efficiency</td>
</tr>
<tr>
<td>$E_{vind}$</td>
<td>volumetric induction efficiency</td>
</tr>
<tr>
<td>f</td>
<td>frequency [Hz]</td>
</tr>
<tr>
<td>f.o.</td>
<td>fully open</td>
</tr>
<tr>
<td>$F_p$</td>
<td>pressure force [N]</td>
</tr>
<tr>
<td>$F_v$</td>
<td>viscous shear force [N]</td>
</tr>
<tr>
<td>$f_o$</td>
<td>natural frequency [Hz]</td>
</tr>
<tr>
<td>h</td>
<td>specific enthalpy at position identified by subscript [J/kg]</td>
</tr>
<tr>
<td>$h_{ro}$</td>
<td>specific enthalpy of saturated liquid refrigerant at $T_o$ [J/kg]</td>
</tr>
<tr>
<td>$h_{go}$</td>
<td>specific enthalpy of dry saturated vapour refrigerant at $T_o$ [J/kg]</td>
</tr>
<tr>
<td>$H_o$</td>
<td>Hodgson number</td>
</tr>
<tr>
<td>$h_{2s}$</td>
<td>specific enthalpy at state 2 which is a hypothetical state after an isentropic compression process [J/kg]</td>
</tr>
</tbody>
</table>
List of Symbols

\( h_{2t} = \) specific enthalpy at state 2 which is a hypothetical state after an isothermal compression process [J/kg]

\( I = \) time interval counter (fig. 7-19)

\( j = \) constant

\( K = \) overall heat loss coefficient [W/K]

\( k = \) constant

\( = \) isentropic exponent

\( or \)

\( = \) a fraction such that \( 0 < k < 1 \)

\( L_p = \) length of passage [m]

\( L_p^* = \) effective length of passage [m]

\( L_s = \) length of space in front of transducer [m]

\( L_1, L_2 = \) constants

\( m = \) mass [kg]

\( \dot{m} = \) mass flow rate [kg/s]

\( m_{bd} = \) mass which flows back through the discharge valve per cycle (due to late closure) [kg]

\( m_{bs} = \) the mass which flows back through the suction valve per cycle (due to late closure) [kg]

\( m_d = \) mass discharged into the discharge plenum per cycle [kg]

\( \dot{m}_d = \) mass flow rate out through discharge valve [kg/s]

\( MEP = \) mean effective pressure \([N/m^2]\) or [bar], as indicated

\( m_E = \) mass transferred externally per cycle [kg]

\( \dot{m}_{hx} = \) mass flow rate through heat exchanger [kg/s]

\( m_I = \) mass induced into the cylinder [kg]

\( m_{d} = \) mass leakage through the discharge valve per cycle (while valve is seated) [kg]
List of Symbols

\( m_p \) = mass which leaks past the piston per cycle [kg]

\( m_{leak} \) = mass leakage through the suction valve per cycle [kg]

\( m_p \) = phantom mass of refrigerant which enters into solution with lubricating oil and subsequently re-emerges [kg]

\( m_s \) = mass within volume \( V_s \) [kg]

\( m_f \) = mass flow rate in through suction valve [kg/s]

\( m_{tot} \) = total mass flow rate [kg/s]

\( m_{1v} \) = mass of refrigerant in the vapour phase at point 1 [kg]

\( m_{4v} \) = mass of refrigerant in the vapour phase at point 4 [kg]

\( N \) = rotational speed [revs./s]

\( n \) = polytropic index

\( N_{cy} \) = number of cycles per unit time [s\(^{-1}\)]

\( p \) = pressure [N/m\(^2\)] or [bara], as indicated

\( \bar{p} \) = mean pressure [N/m\(^2\)]

\( P_e \) = electric power [W]

\( P_{factor} \) = pressure equalisation factor

\( P_{hx} \) = heat exchanger pressure [N/m\(^2\)]

\( P_i \) = indicated power [W]

\( P_m \) = pressure being measured [N/m\(^2\)]

\( P_{me} \) = mean effective pressure [N/m\(^2\)]

\( P_{mix} \) = mixing pressure [N/m\(^2\)]

\( P_{tr} \) = pressure at transducer [N/m\(^2\)]

\( Q \) = heat transfer to the refrigerant [J]
List of Symbols

\( \dot{Q} \) = rate of heat transfer [W]

\( q \) = heat transfer per unit mass [J/kg]

\( q_c \) = net heat transfer from the vapour during compression [J/kg]

\( Q_{cy} \) = heat transfer to the refrigerant within the cylinder between the points of suction pressure equalisation [J]

\( Q_{sp} \) = heat transfer to the refrigerant in the suction plenum per cycle [J]

\( \overline{q_v} \) = time mean volume flow rate [m³/s]

\( (q_v)_{rms} \) = root mean square volume flow rate [m³/s]

\( q_t \) = heat transfer per unit mass from the gas during an assumed constant temperature compression process [J/kg]

\( R \) = specific gas constant [J/kgK]

\( R_e \) = volume ratio

\( R_e^* \) = effective volume ratio

\( r_c \) = clearance ratio

\( R_e D \) = Reynolds number based on diameter

\( R_{u q} \) = volumetric efficiency loss due to suction heat transfer

\( R_{u st} \) = volumetric efficiency loss due to suction throttling

\( r_p \) = radius of passage [m]

\( r_s \) = radius of space in front of transducer [m]

\( R_{up} \) = underpressure loss ratio
List of Symbols

\( s \) = specific entropy at state identified by subscript \([J/kgK]\) or maximum allowable percentage error in indicated flow rate due to pulsations [%]

\( \text{SEP} \) = suction mean effective pressure \([N/m^2]\) or \([\text{bar}]\), as indicated

\( s_{f0} \) = specific entropy of saturated liquid at \( T_0 \) \([J/kgK]\)

\( s_{g0} \) = specific entropy of dry saturated vapour at \( T_0 \) \([J/kgK]\)

\( s_{2t} \) = specific entropy at state 2 which is a hypothetical state after isothermal compression \([J/kgK]\)

\( T \) = temperature \([K]\) or number of turns open

\( \text{TDC} \) = top dead centre

\( T_{dv} \) = mean temperature of refrigerant leaving the discharge valve \([K]\)

\( T_{hx} \) = measured surface temperature on the heat exchanger \([K]\)

\( T_s \) = refrigerant saturation temperature \([K]\)

\( T_{sv} \) = mean temperature of refrigerant entering the suction valve \([K]\)

\( T_{wi} \) = water inlet temperature \([K]\)

\( T_0 \) = temperature of the surroundings \([K]\)

\( T_{1a} \) = adiabatic suction temperature after induction \([K]\)

\( u \) = specific internal energy \([J/kg]\)

\( V \) = volume \([m^3]\)

\( \dot{V} \) = volume flow rate of water \([m^3/s]\)

\( v \) = specific volume \([m^3/kg]\)

\( V_{cl} \) = clearance volume per cylinder \([m^3]\)

\( V_p \) = volume of passage \([m^3]\)
List of Symbols

\( V_p \) = effective volume of passage \([m^3]\)

\( V_s \) = volume of space \([m^3]\)

\( V_{sw} \) = swept volume per cylinder \([m^3]\)

\( w \) = work per unit mass \([J/kg]\)

\( w \) = mass fraction of refrigerant in a refrigerant/oil mixture

\( W_c \) = work of compression (done on vapour) \([J/kg]\)

\( W_{di} \) = indicated discharge work \([J]\)

\( W_{de} \) = ideal work of compression \([J/kg]\)

\( W_{si} \) = indicated suction work \([J]\)

\( X \) = digital value of transducer output

\( x \) = displacement \([m]\)

\( \dot{x} \) = velocity \([m/s]\)

\( \ddot{x} \) = acceleration \([m/s^2]\)

\( X_0 \) = digital value of transducer output corresponding to a zero measurement input

\( Y \) = effective digital output of transducer

1,2,3,4,... \( 1',2',... \)\( 1''',2''',... \) points on an ideal or actual compressor indicator diagram

Greek Letters

\( \beta \) = diameter ratio

\( \gamma \) = ratio of specific heats

\( \Delta h \) = specific enthalpy difference \([J/kgK]\)

\( \Delta p \) = pressure difference \([N/m^2]\)

\( \delta p \) = small increase in pressure \([N/m^2]\)
List of Symbols

$\Delta p^\text{m}$ = time mean pressure difference between a chamber and a source/sink of constant pressure [N/m$^2$]

$\Delta T$ = temperature difference [K]

$\delta v$ = small increase in specific volume [m$^3$/kg]

$\varepsilon$ = expansibility factor

$\eta_{ds}$ = discharge side sealing efficiency

$\eta_{ss}$ = suction side sealing efficiency

$\eta_v$ = displacement utilisation efficiency, also known as real volumetric efficiency

$\eta_{vi}$ = indicated volumetric efficiency

$\eta_{vst}$ = volumetric efficiency taking account of suction throttling

$\eta_{vstq}$ = volumetric efficiency taking account of suction throttling and heat transfer

$\eta_{vstqs}$ = volumetric efficiency, taking account of suction throttling, heat transfer and solubility effects

$\eta_{vstqsl}$ = volumetric efficiency taking account of suction vapour throttling, heat transfer, solubility effects and leakage

$\lambda$ = relative magnitude of flow fluctuation

$\lambda_{clp}$ = clearance volumetric efficiency expressed in terms of suction and discharge pressures

$\lambda_{clv}$ = clearance volumetric efficiency expressed in terms of specific volumes

$\mu$ = absolute viscosity [Ns/m$^2$]

$\nu_{bd}$ = discharge valve backflow ratio

$\nu_{bs}$ = suction valve backflow ratio

$\nu_{ld}$ = discharge valve leakage ratio

$\nu_{lp}$ = leakage past piston ratio

$\nu_{ls}$ = suction valve leakage ratio

$\nu_{pm}$ = phantom mass ratio

xiii
List of Symbols

\[ \mathcal{f} = \text{damping coefficient} \]
\[ \rho = \text{fluid density} \quad [\text{kg/m}^3] \]
\[ \omega_n = \text{angular natural frequency} \quad [\text{rad./s}] \]

General Subscripts

\begin{align*}
\text{a,b,c,...} & \quad \text{refer to measurement locations on load stand} \\
\text{cd} & \quad \text{at closing of discharge valve} \\
\text{cs} & \quad \text{at closing of suction valve} \\
\text{d} & \quad \text{refers to the discharge condition} \\
& \quad \text{or} \\
& \quad \text{refers to downstream condition} \\
\text{hx} & \quad \text{refers to a property or parameter within the heat exchanger on the load stand} \\
\text{od} & \quad \text{at opening of discharge valve} \\
\text{os} & \quad \text{at opening of suction valve} \\
\text{s} & \quad \text{refers to the suction condition (mean value)} \\
& \quad \text{or} \\
& \quad \text{refers to an isentropic process} \\
& \quad \text{or} \\
& \quad \text{refers to a saturation temperature or pressure} \\
\text{sv} & \quad \text{indicates a property mean value at entry to the suction valve} \\
\text{u} & \quad \text{refers to upstream condition} \\
1,2,3,4,... & \quad \text{refer to points on an ideal or actual compressor indicator diagram} \\
& \quad \text{or} \\
& \quad \text{refer to thermodynamic state points at positions around a load stand circuit}
\end{align*}
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Chapter</th>
<th>Section</th>
<th>Figure Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ch. 1</td>
<td>1-1</td>
<td>General diagram of a compressor</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>1-2</td>
<td>Heat pump heating capacity versus speed</td>
<td>12</td>
</tr>
<tr>
<td>Ch. 2</td>
<td>2-1</td>
<td>Indicator diagram</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>2-2</td>
<td>A hypothetical cycle</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>2-3</td>
<td>Fluid transfer paths</td>
<td>39</td>
</tr>
<tr>
<td>Ch. 3</td>
<td>3-1</td>
<td>Comparison between compression processes:</td>
<td>48</td>
</tr>
<tr>
<td></td>
<td></td>
<td>a) isentropic and b) isentropic and isothermal stages</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3-2</td>
<td>An ideal compression process with heat rejection at T_o</td>
<td>49</td>
</tr>
<tr>
<td></td>
<td>3-2</td>
<td>An ideal compression process between specified suction and discharge states</td>
<td>51</td>
</tr>
<tr>
<td>Ch. 4</td>
<td>4-1</td>
<td>Schematic diagram of a calorimeter load stand</td>
<td>58</td>
</tr>
<tr>
<td></td>
<td>4-2</td>
<td>Thermodynamic processes of a calorimeter load stand</td>
<td>58</td>
</tr>
<tr>
<td></td>
<td>4-3</td>
<td>Schematic diagram showing heat and work transfer paths of a calorimeter load stand</td>
<td>60</td>
</tr>
<tr>
<td></td>
<td></td>
<td>with internal heat transfer</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4-4</td>
<td>A gas loop load stand (two versions)</td>
<td>61</td>
</tr>
<tr>
<td></td>
<td>4-5</td>
<td>Cycles of gas loop load stands</td>
<td>62</td>
</tr>
<tr>
<td></td>
<td>4-6</td>
<td>Diagram of a gas loop load stand with a reservoir</td>
<td>63</td>
</tr>
<tr>
<td></td>
<td>4-7</td>
<td>Marriott load stand schematic</td>
<td>64</td>
</tr>
<tr>
<td></td>
<td>4-8</td>
<td>Marriott load stand processes</td>
<td>65</td>
</tr>
<tr>
<td></td>
<td>4-9</td>
<td>DIN calorimeter, version J (schematic and processes)</td>
<td>67</td>
</tr>
<tr>
<td></td>
<td>4-10</td>
<td>Diagram of a pressure induction system</td>
<td>69</td>
</tr>
</tbody>
</table>
List of Figures

Ch. 5 Design Rationale of the Load Stand Used
5-1 Load stand basic circuit ................................. 75
5-2 Idealised thermodynamic processes of the load stand ................................. 76
5-3 Load stand circuit including bypass and liquid line throttle valves ................................. 78
5-4 Thermodynamic processes including liquid line and bypass throttling ................................. 79
5-5 Heat transfer rate characteristic of heat exchanger ................................. 86
5-6 Sensitivity of load stand control valves to suction saturation temperature ................................. 89
5-7 Sensitivity of load stand control valves to discharge saturation temperature ................................. 90
5-8 Sensitivity of load stand control valves to suction superheat ................................. 91
5-9 Sensitivity of load stand control valves to compressor speed ................................. 92
5-10 Sensitivity of load stand control valves to heat exchanger water flow rate ................................. 93

Ch. 6 The Load Stand and Its Instrumentation
6-1 Schematic diagram of the compressor load stand ................................. 97
6-2 The compressor load stand (photo.) ................................. 98
6-3 The compressor and its driving motor ................................. 99
6-4 View of the compressor cylinder head ................................. 101
6-5 Diagram of equipment setup for mass flow rate measurement by the orifice plate method ................................. 103
6-6 Measured differential pressure waveform ................................. 106
6-7 View of the liquid line and sampling apparatus ................................. 113
6-8 Diagram of the liquid refrigerant sample vessel ................................. 114
6-9 View of the valve plate (cylinder side) ................................. 116
6-10 View of the inside of the cylinder head ................................. 116
6-11 View of the valve plate (plenum side) ................................. 117
List of Figures

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>6-12</td>
<td>Discharge valve displacement transducer mounting arrangement</td>
<td>117</td>
</tr>
<tr>
<td>6-13</td>
<td>Pressure inducting system of the cylinder pressure transducer</td>
<td>119</td>
</tr>
<tr>
<td>6-14</td>
<td>Load stand and compressor diagram (p-h)</td>
<td>122</td>
</tr>
<tr>
<td>6-15</td>
<td>Section through the mixing venturi</td>
<td>123</td>
</tr>
<tr>
<td>7-1</td>
<td>Ambient temperature versus speed</td>
<td>128</td>
</tr>
<tr>
<td>7-2</td>
<td>Ambient temperature versus suction temp.</td>
<td>128</td>
</tr>
<tr>
<td>7-3</td>
<td>Mass flow rate measured by the orifice plate method versus speed</td>
<td>129</td>
</tr>
<tr>
<td>7-4</td>
<td>Mass flow rate measured by the orifice plate method versus suction temperature</td>
<td>129</td>
</tr>
<tr>
<td>7-5</td>
<td>Mass flow rate determined by the heat balance method versus compressor speed</td>
<td>130</td>
</tr>
<tr>
<td>7-6</td>
<td>Mass flow rate determined by the heat balance method versus suction temperature</td>
<td>130</td>
</tr>
<tr>
<td>7-7</td>
<td>Shaft power versus compressor speed</td>
<td>133</td>
</tr>
<tr>
<td>7-8</td>
<td>Shaft power versus suction temperature</td>
<td>133</td>
</tr>
<tr>
<td>7-9</td>
<td>Discharge temperature versus compressor speed</td>
<td>134</td>
</tr>
<tr>
<td>7-10</td>
<td>Discharge temperature versus suction temperature</td>
<td>134</td>
</tr>
<tr>
<td>7-11</td>
<td>Oil concentration in the discharge vapour versus speed</td>
<td>135</td>
</tr>
<tr>
<td>7-12</td>
<td>Oil concentration versus suction temperature</td>
<td>135</td>
</tr>
<tr>
<td>7-13</td>
<td>View of the Bitzer IV compressor with the cylinder head removed</td>
<td>136</td>
</tr>
<tr>
<td>7-14</td>
<td>Compressor temperatures versus speed</td>
<td>138</td>
</tr>
<tr>
<td>7-15</td>
<td>Compressor temperatures versus suction temp.</td>
<td>138</td>
</tr>
<tr>
<td>7-16</td>
<td>Cylinder pressure and valve lifts vs. crank angle, 300 r.p.m.</td>
<td>139</td>
</tr>
<tr>
<td>7-17</td>
<td>Cylinder pressure and valve lifts vs. crank angle, 590 r.p.m.</td>
<td>140</td>
</tr>
<tr>
<td>7-18</td>
<td>Cylinder pressure and valve lifts vs. crank angle, 902 r.p.m.</td>
<td>141</td>
</tr>
<tr>
<td>7-19</td>
<td>Detail diagram of valve lift vs. time</td>
<td>143</td>
</tr>
</tbody>
</table>
List of Figures

7-20 Cylinder pressure and valve lifts vs. volume ........................................... 146
7-21 Effective pressures versus speed ......................................................... 148
7-22 Effective pressures vs. suction temp. .................................................. 148
7-23 Pressure equalisation factor vs. speed .................................................. 149
7-24 Press. equalisation factor vs. suct. temp. ..... 149
7-25 Volumetric efficiencies vs. speed ......................................................... 151
7-26 Volumetric efficiencies vs. suct. temp. ................................................ 151
7-27 Plenum heat transfer rates vs. speed ..................................................... 154
7-28 Plenum heat transfer rates vs. suct. temp. ........................................... 154
7-29 Suction leakage ratio vs. speed ............................................................. 158
7-30 Suction leakage ratio vs. suct. temp. ................................................... 158
7-31 Discharge leakage ratio vs. speed .......................................................... 159
7-32 Discharge leakage ratio vs. suct. temp. ................................................ 159
7-33 log(p) vs. log(V) ...................................................................................... 163
7-34 Polytropic indices for compression vs. speed ........................................... 165
7-35 Polytropic indices for compr. vs. suct. temp. ........................................... 165
7-36 Polytropic index for re-expansion vs. speed ............................................. 167
7-37 Polytropic index for re-exp. vs. suct. temp. ............................................ 167
7-38 Displacement utilisation efficiencies at 300 r.p.m. .................................. 173
7-39 Displacement utilisation efficiencies at 900 r.p.m. .................................. 173
7-40 Displacement utilisation efficiencies at 0°C .......................................... 174
7-41 Displacement utilisation efficiencies at 30°C ........................................... 174
7-42 Underpressure loss ratio vs. speed ......................................................... 176
7-43 Underpressure loss ratio vs. suct. temp. ................................................ 176
7-44 Losses in displacement utilisation at 300 and at 900 r.p.m. .................... 179
7-45 Losses in displacement utilisation at 0°C and at 30°C ............................... 179
7-46 Mechanical efficiency vs. speed ............................................................ 180
7-47 Mechanical efficiency vs. suct. temp. .................................................... 180
7-48 Indicated and pumping power vs. speed .................................................. 182
7-49 Indicated and pumping power vs. suct. temp. ......................................... 182
7-50 Indicated efficiencies vs. speed ............................................................. 183
7-51 Indicated efficiencies vs. suct. temp. ...................................................... 184
List of Figures

7-52 Net heat rejection rate vs. speed ............... 185
7-53 Net heat rejection rate vs. suct. temp. ...... 185

Ch. 8 Compressor Simulation

8-1 Effective force area versus valve lift ......... 191
8-2 Flow chart of the simulation program .......... 195
8-3 Simulation results: cylinder pressure and valve lifts versus volume ....................... 196
8-4 Comparison between simulated and measured mass flow rate .................................. 198
8-5 Comparison between simulated and measured indicated power ................................. 198
8-6 Comparison between simulated and measured discharge temperature ....................... 199
8-7 Comparison between simulated and measured indicated volumetric efficiency ............ 199
8-8 Comparison between simulated and measured mean effective pressure and suction effective pressure .................................................. 200
8-9 Comparison between simulated and measured discharge effective pressure .............. 200
<table>
<thead>
<tr>
<th>Chapter</th>
<th>Section</th>
<th>Table Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ch. 1</td>
<td>1-1</td>
<td>Volumetric efficiency versus speed</td>
<td>11</td>
</tr>
<tr>
<td>Ch. 4</td>
<td>4-1</td>
<td>Results of Mainardi et al.</td>
<td>73</td>
</tr>
<tr>
<td>Ch. 5</td>
<td>5-1</td>
<td>Heat exchanger mass flow rate as a proportion of the total compressor mass flow rate</td>
<td>77</td>
</tr>
<tr>
<td></td>
<td>5-2</td>
<td>Heat exchanger mass flow rate for an ideal compressor</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td>5-3</td>
<td>Heat exchanger heat transfer rate for an ideal compressor</td>
<td>81</td>
</tr>
<tr>
<td></td>
<td>5-4</td>
<td>Effective flow areas of valves on the test stand for an ideal compressor</td>
<td>84</td>
</tr>
<tr>
<td></td>
<td>5-5</td>
<td>Comparison of observed load stand valve positions with predicted values</td>
<td>85</td>
</tr>
<tr>
<td>Ch. 6</td>
<td>6-1</td>
<td>Calculated errors of component frequencies within cylinder pressure versus time waveforms due to the characteristics of the pressure inducting system</td>
<td>120</td>
</tr>
<tr>
<td></td>
<td>6-2</td>
<td>Estimated maximum errors associated with the data transformations within the pressure measurement system</td>
<td>121</td>
</tr>
<tr>
<td></td>
<td>6-3</td>
<td>Readings for test no. 64</td>
<td>124</td>
</tr>
<tr>
<td>Ch. 7</td>
<td>7-1</td>
<td>Test conditions</td>
<td>126</td>
</tr>
<tr>
<td></td>
<td>7-2</td>
<td>Valve natural frequencies</td>
<td>145</td>
</tr>
<tr>
<td></td>
<td>7-3</td>
<td>Results of heat transfer and leakage analyses</td>
<td>155</td>
</tr>
<tr>
<td></td>
<td>7-4</td>
<td>Sensitivity of polytropic indices to errors</td>
<td>168</td>
</tr>
<tr>
<td></td>
<td>7-5</td>
<td>Refrigerant phantom mass analysis</td>
<td>171</td>
</tr>
</tbody>
</table>
CONTENTS

DECLARATIONS i

SUMMARY ii

DEDICATION iv

ACKNOWLEDGEMENTS v

LIST OF SYMBOLS vi

LIST OF FIGURES xv

LIST OF TABLES xx

INTRODUCTION 1

Chapter 1 BACKGROUND 3

1.1 THE COMPRESSOR WHICH WAS TESTED AND ANALYSED 3

1.2 CHARACTERISATION OF COMPRESSOR PERFORMANCE 5

1.3 CAPACITY CONTROL BY SPEED VARIATION 6

1.4 INFORMATION FROM THE LITERATURE ON VARIABLE SPEED OPERATION 8

1.4.1 THE WORK OF ITAMI ET AL. 8

1.4.1.1 Modifications to Standard Compressors for Variable Speed Use 9

1.4.1.2 Capacity 9

1.4.1.3 Volumetric Efficiency 9

1.4.1.4 Efficiency of the Motor/Inverter 9

1.4.1.5 Overall Efficiency 10

1.4.1.6 Seasonal Energy Efficiency Ratio 10

1.4.2 THE WORK OF PAUL ET AL. 10

1.4.2.1 Volumetric Efficiency 11

1.4.2.2 Heating Capacity 11

1.4.3 THE WORK OF BREDESEN, PAUL, ET AL. ON VALVE BEHAVIOUR OVER A RANGE OF SPEEDS 13

1.5 COMPUTER CALCULATION OF REFRIGERANT THERMODYNAMIC PROPERTIES 15

1.5.1 PROGRAMS AND DATA FOR PURE REFRIGERANTS 16

1.5.2 REFRIGERANT/LUBRICANT MIXTURES 16

1.5.3 PROPERTY PROGRAMS USED 18

Chapter 2 UTILISATION OF VOLUME DISPLACEMENT 19

2.1 THE PROCESSES OF THE RECIPROCATING COMPRESSOR CYCLE 19
# Contents

Chapter 4  **COMPRESSOR LOAD STANDS**  
4.1  **THE CALORIMETER LOAD STAND**  
4.2  **ZLATKOV'S LOAD STAND**  
4.3  **GAS LOOP OR VAPOUR BYPASS LOAD STANDS**  
4.4  **THE MARRIOTT FAST RESPONSE LOAD STAND**  
4.5  **STANDARDS FOR THE PERFORMANCE TESTING OF REFRIGERANT COMPRESSORS**  
   4.5.1  **ASHRAE STANDARD**  
   4.5.2  **GERMAN STANDARD**  
4.6  **SOME SPECIALISED MEASUREMENT TECHNIQUES**  
   4.6.1  **DYNAMIC PRESSURE MEASUREMENT**  
   4.6.2  **MEASUREMENT OF PULSATING FLOW WITH AN ORIFICE PLATE**  
   4.6.2.1  **Dynamic Differential Pressure Measurement**  
   4.6.2.2  **Measurement of Mean Differential Pressure with Flow Damping**  
4.7  **CONCLUSIONS**

Chapter 5  **DESIGN RATIONALE OF THE LOAD STAND USED**  
5.1  **THERMODYNAMIC PROCESSES**  
5.2  **THE IDEAL LOAD STAND AND COMPRESSOR**  
5.3  **EFFECTIVE VALVE FLOW AREAS**  
5.4  **THROTTLE VALVES: SELECTION AND CHARACTERISTICS**  
5.5  **HEAT EXCHANGER SELECTION, CHARACTERISTICS AND ANALYSIS**  
   5.5.1  **HEAT TRANSFER RATE**  
   5.5.2  **HEAT EXCHANGER WATER SIDE PRESSURE DROP**  
5.6  **COMPUTER MODELLING OF THE COMPLETE LOAD STAND**  
   5.6.1  **SENSITIVITY OF THE REQUIRED VALVE OPENINGS TO THE OPERATING CONDITIONS**  
   5.6.2  **COMPARISON OF THE LOAD STAND MODEL WITH TEST OBSERVATIONS**  
5.7  **CONCLUSIONS**
## Contents

### Chapter 6  THE LOAD STAND AND ITS INSTRUMENTATION  96

6.1 CONTROL OF OPERATING CONDITIONS ......................... 96
6.2 MEASUREMENT OF TORQUE ....................................... 98
6.3 MEASUREMENT OF SPEED ......................................... 99
6.4 MEASUREMENT OF TEMPERATURES ................................ 100
6.5 MEASUREMENT OF PRESSURE ..................................... 101
6.6 MASS FLOW RATE OF REFRIGERANT .............................. 102
6.6.1 ORIFICE PLATE METHOD ...................................... 102
6.6.1.1 Calibration .................................................. 104
6.6.1.2 Development of the Orifice Plate Flow Measurement System .................................................. 105
6.6.2 HEAT BALANCE METHOD ....................................... 110
6.7 OIL CONCENTRATION MEASUREMENT ............................ 111
6.7.1 RATIONALE OF THE METHOD AS APPLIED TO THE LOAD STAND .................................................. 112
6.7.2 EQUIPMENT DETAILS .......................................... 112
6.7.3 PROCEDURE ...................................................... 113
6.8 VALVE LIFT MEASUREMENT ...................................... 115
6.8.1 CALIBRATION .................................................... 118
6.8.2 INTERFACING ................................................... 118
6.9 CYLINDER PRESSURE MEASUREMENT ............................ 118
6.9.1 FREQUENCY RESPONSE CHARACTERISTICS .................. 119
6.9.2 CALIBRATION .................................................... 120
6.10 DATA LOGGING ..................................................... 121
6.11 THE SUCTION LINE MIXING PROCESS ............................ 121
6.12 TEST PROGRAMME AND SAMPLE READINGS ................... 123
6.13 CONCLUSIONS ...................................................... 124

### Chapter 7  COMPRESSOR PERFORMANCE CHARACTERISTICS  126

7.1 TEST CONDITIONS .................................................. 126
7.2 MAIN OPERATING CHARACTERISTICS ............................ 127
7.2.1 REFRIGERANT MASS FLOW RATE .............................. 127
7.2.1.1 Influence of Entrained Oil on Flow Rate Precision .................................................. 131
7.2.2 SHAFT POWER INPUT .......................................... 132
7.2.3 DISCHARGE GAS TEMPERATURE .............................. 132
7.3 OIL CONCENTRATION IN THE DISCHARGE VAPOUR ............ 132
7.3.1 OIL CONCENTRATION CHARACTERISTICS .......... 132
7.3.2 OIL TRANSFER WITHIN THE COMPRESSOR .......... 137
7.4 VALVE LIFT AND PRESSURE DIAGRAMS ............... 137
7.4.1 DETERMINATION OF POINTS OF VALVE OPENING AND CLOSING ........................................... 137
7.4.2 VALVE LIFT AND CYLINDER PRESSURE VERSUS CRANK ANGLE ............................................... 144
7.4.3 LIFT AND PRESSURE DIAGRAMS VERSUS VOLUME .. 145
7.4.4 MEAN EFFECTIVE Pressures ........................ 147
7.4.5 UNDER OR OVERPRESSURE AT VALVE OPENING OR CLOSING .................................................. 147
7.5 UTILISATION OF VOLUME DISPLACEMENT ............... 150
7.5.1 DISPLACEMENT UTILISATION EFFICIENCY ............ 150
7.5.2 INDICATED VOLUMETRIC EFFICIENCY ................ 150
7.5.3 HEAT TRANSFER TO THE REFRIGERANT WITHIN THE SUCTION AND DISCHARGE PLENUMS ............... 152
7.5.4 HEAT TRANSFER AND/OR LEAKAGE/BACKFLOW DURING INDUCTION AND DISCHARGE ....................... 155
7.5.4.1 The 'No Leakage' Assumption .................... 155
7.5.4.2 The 'Adiabatic Vapour Transfer' Assumption ............................................................... 156
7.5.4.3 Possible Solubility Effects ..................... 160
7.5.4.4 The Balance of Uncertainty ..................... 161
7.5.5 COMPRESSION AND RE-EXPANSION PROCESSES ....... 161
7.5.5.1 Basis of the Analyses ........................... 161
7.5.5.2 Indices Based on p-V Data for Compression 164
7.5.5.3 Indices Based on p-V Data for Re-expansion ................................................................. 164
7.5.5.4 Indices Based on p-T Data ........................ 165
7.5.5.5 Exclusion of Low Polytropic Index Values . 166
7.5.5.6 Sensitivity to Zero Errors in Pressure and Volume ......................................................... 168
7.5.5.7 Leakage and Plenum Pulsation Effects ......... 169
7.5.5.8 Solubility Effects ............................... 169
7.5.5.9 Conclusions Regarding Polytropic Indices .. 171
7.5.6 ATTRIBUTION OF LOSSES IN DISPLACEMENT UTILISATION EFFICIENCY ............................... 172
Contents

7.5.6.1 Loss Due to Clearance Volume ................. 172
7.5.6.2 Loss Due to Mass Addition During
Re-expansion and Underpressure at the
Bottom Dead Centre Position ................. 175
7.5.6.3 Loss Due to Suction Throttling and Heat
Transfer Within the Suction Plenum,
According to the 'Adiabatic Vapour
Transfer' Assumption ....................... 175
7.5.6.3.1 Loss Due to Heat Transfer Within the
Suction Plenum ........................... 177
7.5.6.3.2 Loss Due to Suction Throttling and
Re-compression .......................... 177
7.5.6.3.3 Reapportionment of Suction Plenum
Heating and Suction Throttling Losses .... 177
7.5.6.4 Loss Due to Heat Transfer During the
Induction and Discharge Processes or Due
to Backflow, Leakage, and Oil Solubility
Effects Not Already Included ............... 178
7.5.6.5 Summary of Losses ....................... 178
7.6 UTILISATION OF SHAFT POWER ............... 178
7.6.1 MECHANICAL EFFICIENCY .................... 178
7.6.2 INDICATED POWER ........................ 181
7.6.3 INDICATED POWER UTILISATION EFFICIENCIES .. 181
7.6.4 NET HEAT TRANSFER FROM THE REFRIGERANT ..... 184
7.7 SUMMARY .................................... 186

Chapter 8 COMPRESSOR SIMULATION ..................... 187
8.1 BASIS OF THE PURDUE SIMULATION MODEL ........ 188
8.2 INPUTS TO THE SIMULATION PROGRAM ............ 188
8.2.1 GEOMETRIC PARAMETERS ..................... 188
8.2.2 WORKING FLUID PARAMETERS .................. 188
8.2.3 OPERATING CONDITIONS ...................... 188
8.2.4 INITIAL CONDITIONS ........................ 189
8.2.5 CALCULATION STEP PARAMETERS ............... 189
8.2.7 VALVE EFFECTIVE FLOW AREAS ................ 189
8.2.8 VALVE EFFECTIVE FORCE AREAS ................ 190
8.2.9 AREAS OF VALVE REED AND PORT ELEMENTS ...... 191
Contents

8.2.10 NATURAL FREQUENCIES AND MODE SHAPES OF VALVE REEDS ........................................ 192
8.2.11 DAMPING RATIOS ................................................................. 193
8.3 NUMERICAL SOLUTION TECHNIQUE ........................................... 193
8.4 OUTPUTS OF THE SIMULATION PROGRAM ........................................ 195
8.4.1 PRESSURE AND VALVE LIFT VERSUS CRANK ANGLE DIAGRAMS .................................. 195
8.4.2 PERFORMANCE CHARACTERISTICS OVER THE SPEED RANGE ....................................... 197
8.4.2.1 Mass Flow Rate ............................................................... 197
8.4.2.2 Indicated Power ............................................................. 197
8.4.2.3 Discharge Temperature .................................................. 197
8.4.2.4 Indicated Volumetric Efficiency .................. 197
8.4.2.5 Effective Pressures from the Indicator Diagrams ...................................................... 201
8.5 COMMENTS .......................................................... 201

Chapter 9 CONCLUSIONS ......................................................... 202
9.1 THE LOAD STAND ................................................................. 202
9.2 TESTING METHODS ............................................................... 204
9.3 COMPRESSOR CHARACTERISTICS ............................................. 206
9.4 IMPLICATIONS FOR OVERALL PLANT PERFORMANCE ... 207
9.5 FACTORS WHICH INFLUENCE THE UTILISATION OF VOLUME DISPLACEMENT ................. 207
9.6 FACTORS WHICH INFLUENCE THE UTILISATION OF SHAFT POWER ..................................... 208
9.7 COMPRESSOR SIMULATION .................................................... 210
9.8 CONCLUSION ............................................................... 211

REFERENCES ................................................................. 212

APPENDICES
A COMPRESSOR SPECIFICATIONS ................................................. A-1
B BAMBACH'S SOLUBILITY RELATIONSHIPS ................................ B-1
C DERIVATION OF GOSNEY'S VOLUMETRIC EFFICIENCY EQUATION ......................... C-1
Contents

D DERIVATION OF THE DAMPING COEFFICIENT FOR A HELMHOLTZ RESONATOR .................. D-1
E VALVE FLOW CHARACTERISTICS ..................... E-1
F LIST OF MAIN ITEMS OF EQUIPMENT USED ............ F-1
G MEASUREMENT ORIFICE FLOW EQUATION ............... G-1
H A DIFFERENTIAL PRESSURE TRANSDUCER FAILURE MECHANISM ......................... H-1
I DETAILS OF THE TESTING PROGRAMME .......... I-1
J VOLUMETRIC INDUCTION EFFICIENCY ANALYSIS PROCEEDURE ........................... J-1
K INPUT DATA FOR THE PURDUE SIMULATION PROGRAM .. K-1
INTRODUCTION

The purpose of the work was to critically re-examine the performance evaluation process as applied to refrigerant compressors, to investigate the methods used, the parameters which quantify performance and the ways in which the data can be presented, and to determine the factors which influence compressor characteristics. The work was carried out at the Department of Mechanical and Manufacturing Engineering, Trinity College Dublin.

It was felt at the outset that there was considerable potential for matching the output of a heat pump to the heating load, or, the cooling effect of a refrigeration plant to the cooling load, by varying the speed of the compressor. The possibility of using an electronic inverter speed controller for this purpose had been described in the literature, e.g. by McMullan and Morgan in their book [1, p.73].

The term 'inverter', as used here, means a solid state electronic device in which an alternating mains electricity supply of constant frequency is 'converted', or rectified, to d.c., which is then 'inverted' to a.c. at any desired frequency within the continuous range of the device. The output frequency range may extend above, as well as below, the supply value.

When the work commenced, in 1982, inverters which operated with good electrical efficiency were commercially available. These were viable for some fan and pump driving applications. Although the cost per kilowatt capacity would have been rather high for most heat pump or refrigeration applications, it was felt that inverter costs would reduce in real terms, as had been the case in solid state electronics generally. In fact, as was later discovered, Japanese manufacturers were already making inverter driven heat pumps for their home market [2].

The scenario highlighted the need for data on the performance of compressors over a range of speeds. It was felt, however, that the time was ripe for the re-examination of the whole process of performance evaluation and quantification. A growing number of refrigerant compressor types, e.g. reciprocating, twin screw, and vane type, were in common use. Newer types such as the rolling piston and scroll compressor [3]
Introduction

seemed to have excellent potential for variable speed operation. It was decided, however, to study only the reciprocating type for the purposes of this thesis. A mechanically driven compressor was selected, in order that the results would not be obscured by drive system characteristics.

The approach adopted involved a review of the literature on variable speed reciprocating compressor performance, design and construction of a load stand, and the carrying out of performance tests. Analyses of the load stand and of the test results were undertaken. The compressor was also simulated with a computer model.

An aspect of the work which may appear transparent is the breadth of the computer programming which was involved. No program listings are included, but, the essential theory and calculation structures are described in the text. The power of computer based analysis of thermodynamic processes of real fluids was harnessed to good effect. In particular, the analyses of processes within the compressor cylinder, where experimental data was insufficiently precise or incomplete and where many different influences were superimposed, yielded useful insights. Conjecture was built into analysis programs where the solutions would otherwise have been indeterminate.

In this thesis a new framework for the quantification of losses in displacement utilisation is proposed and applied. Shaft power utilisation is assessed in terms of thermodynamic availability. Many other new ideas relating to compressor performance evaluation are presented, e.g. control of the load stand by throttle valves only, an improved method of measuring lubricating oil concentration, and the use of logarithmic diagrams of cylinder pressure versus volume.
Chapter 1
BACKGROUND

In this chapter background information relating to the research is presented. The compressor which was tested is described, and the parameters of compressor performance are introduced. Germane published results of other researchers are referred to as background to the work. Details are given of the computer programs which were used for the calculation of refrigerant thermodynamic properties.

1.1 THE COMPRESSOR WHICH WAS TESTED AND ANALYSED

An open type air-cooled belt driven compressor of relatively straightforward design was chosen for the work, fig. 1-1. This was the Bitzer model IV, which had two cylinders. In most commercial or industrial applications it would have been belt driven by an induction motor operating at constant speed. According to the manufacturers' data the speed would have been in the range 380 r.p.m. to 750 r.p.m, depending on the pulley ratios used.

The compressor was chosen for convenience in testing and analysis. By selecting an open compressor the measurement of shaft power was facilitated and the complications of motor characteristics and heat transfer from the motor to the refrigerant were avoided. The physical size of the machine was convenient for the installation of measurement transducers and access was relatively good. The dimensions of the compressor are given in appendix A.

A further advantage of the compressor which was chosen was that this type had been in widespread use for a considerable time and was still in production, so parts were readily available. The compressor was suitable for use with refrigerants R-12, R-22 or R-502. The first of these was selected for the work.

The refrigeration oil used in the compressor was SUNISO 3GS, a naphthenic mineral oil widely used in the industry.
Fig. 1-1 General diagram of a compressor from the Bitzer series, of which the model IV is a member (source: product catalogue: "Open Compressors 0 to VII", Bitzer Kühlmaschinenbau GmbH & Co. KG). Two discharge ports and one suction port can be seen on the diagram, above the left hand cylinder. The compressor which was tested, model IV, had only one discharge and one suction port for each cylinder. See Appendix A for further details.
1.2 CHARACTERISATION OF COMPRESSOR PERFORMANCE

The operating conditions of a reciprocating refrigerant compressor, such as the Bitzer model IV, are defined, with reasonable completeness, by six parameters. These are:

1. the refrigerant type
2. the suction pressure, or the corresponding saturation temperature
3. the discharge pressure, or the corresponding saturation temperature
4. the suction gas temperature, or the amount of suction superheat
5. the compressor speed
6. the temperature of the surroundings to or from which heat transfer can occur.

For greater rigour in defining the operating conditions it would be necessary to specify, for example,

7. the nature of the flow of air around the compressor
8. the characteristics of the lubricating oil used
9. the nature of the suction and discharge pipework attached to the compressor.

In the refrigeration industry, the suction and discharge pressures are usually described in terms of the corresponding saturation temperatures of the refrigerant being used. In an actual plant these temperatures would roughly correspond with the evaporating and condensing temperatures. The terms 'suction saturation temperature' and 'discharge saturation temperature' will be used in this thesis.

Under a given set of steady operating conditions the compressor will have unique values of the following main characteristics:

A. refrigerant mass flow rate (or, volume flow rate at the suction condition)
B. shaft power input
C. discharge gas temperature

Other operating characteristics of lesser significance, or, with design or reliability implications, could also be determined, e.g.

D. mass concentration of oil in the discharge vapour
E. sump oil temperature.

Compressor characterisation is the process whereby operating characteristics A to C are determined for a range of operating conditions. Compressor characteristics can be presented in
tabular or graphical form and are essential information to a system designer who wishes to consider adopting the compressor.

It is common practice amongst manufacturers of compressors and experimentalists to characterise compressors by presenting data for the heating or cooling effect and the heat pump or refrigeration cycle coefficient of performance (C.O.P.) over a range of operating conditions. While many end users appreciate data in this form, it was considered preferable to present data for mass flow rate, shaft power input, and discharge temperature as already described. The vapour compression cycle performance parameters of the compressor are readily calculated from these.

In order to understand and explain the operating characteristics of a compressor, characterisation at a deeper level is necessary. This may involve analysis of indicator diagrams, valve lift diagrams and pressure pulsation waveforms, as well as internal and external heat transfer processes. Various 'figures of merit' may be used to describe how closely the actual compressor performance approaches theoretical ideals. In chapter 2 the utilisation of volume displacement is considered in detail and in chapter 3 the utilisation of shaft power is examined. Appropriate definitions of efficiency are presented in both cases.

1.3 CAPACITY CONTROL BY SPEED VARIATION

Capacity control by speed variation offers the advantage of precision in matching the heating or cooling load, or, in maintaining an exact temperature. Energy consumption and the number of start-stop cycles are also reduced, as discussed in the following paragraphs.

A conventional system, operating in the start-stop mode in response to a thermostat, allows the controlled temperature to vary within a band in the region of the desired value. Similar temperature fluctuations are produced by a step controlled system in which, perhaps, one compressor of a number is switched on and off, or, one or more cylinders of a multi-cylinder compressor are intermittently loaded and unloaded. In the storage of frozen foods, for example, such temperature
fluctuations are undesirable, as they encourage the growth of large ice crystals which can damage the produce.

The argument in support of lower energy consumption when capacity is modulated by speed variation is as follows. If, for example, the required heating or cooling demand over a period is half the system capacity, then, the thermostat in an on/off controlled system will ensure that the plant runs for approximately half of the time and is idle for the remainder. A system which can operate over a continuous range of speeds, however, will run at considerably less than maximum speed over the entire period. The instantaneous rates of heat transfer in the evaporator and condenser, and the power input to the compressor, will be approximately half those of the on/off system. In consequence, the temperature differences across the heat exchangers will be smaller in the variable speed system and it will operate at a higher coefficient of performance, whether it is a heat pump or a refrigerator. This is due to the fact that when the saturation temperatures in the evaporator and condenser are closer together the pressure difference (or the pressure ratio) across the compressor is reduced and less work is required per unit mass of refrigerant. The continuously running variable speed system thus requires less energy input, as work, per unit of heating or cooling provided.

The comparison given between an intermittent and a continuously operating system is an interesting one for the thermodynamicist. The rates of the heat transfer processes determine the degree to which reversibility is approached. The system which operates continuously scores more highly in terms of a rational system efficiency based on the Second Law of Thermodynamics and, of course, in terms of running costs.

The lower number of start/stop cycles is also an advantage of a system which runs continuously. Electrical and mechanical reliability are both likely to be increased, as system start up imposes increased stresses in these areas. There are also energy, or more precisely, availability losses, associated with on/off operation. When the system starts, temperatures within the plant begin to move towards steady state (or, quasi-steady-
Ch. 1 Background

state) values. Some parts of the system move towards temperatures above ambient and other parts, e.g. the evaporator and suction piping, are cooled below ambient. The plant components and the charge of refrigerant possess heat capacity, and a work (or electrical energy) input is required to bring them into a state of thermal disequilibrium with the environment. The system, thus, possesses thermodynamic availability at the moment it is switched off. This availability is lost irreversibly as the various parts of the system heat up and cool down towards ambient temperature.

It was decided to test the steady state performance of the compressor only. The justification for this, and for the neglect of transient effects, was that a speed modulated compressor would be expected to operate in a quasi-steady-state manner. There would, ideally, be relatively few sudden changes in operating conditions.

In these paragraphs the positive performance benefits of capacity control by speed variation have been stressed. A possible negative effect would be any disproportionate increase in losses, e.g. mechanical or volumetric, over a range of speeds, compared with losses at the design speed. Such effects were not expected to outweigh the benefits of variable speed operation, and the test results confirmed this (see ch. 7).

1.4 INFORMATION FROM THE LITERATURE ON VARIABLE SPEED OPERATION

1.4.1 THE WORK OF ITAMI ET AL.

Itami, Okoma and Misawa [2] of the Toshiba Corporation presented a paper at the 1982 Purdue Compressor Technology Conference entitled "An experimental Study of Frequency Controlled Compressors". They described test results for a rolling piston type and for a reciprocating piston type compressor, driven by an inverter in each case. Both compressors were fully hermetic. The motors, which were of the two pole induction type, had been newly developed for inverter drive. The authors reported that the frequency controlled compressors developed by their company had been mass produced since 1980 in
Ch. 1 Background

the case of the reciprocating compressor and since 1981 in the case of the rotary unit.

Some of the findings of Itami et al., which were relevant to the work for this thesis, are stated below.

1.4.1.1 Modifications to Standard Compressors for Variable Speed Use

In the Toshiba reciprocating compressor tested by Itami et al., a ring discharge valve was adopted instead of a leaf valve to improve volumetric efficiency over a wide operating frequency range. A two stage oil pump was used to improve oil circulation at the low end of the frequency range, and a damper was incorporated to prevent discharge piping resonance.

1.4.1.2 Capacity

For the reciprocating and rotary compressors, capacity (presumably cooling) was found to vary in proportion to the operating frequency. The reciprocating compressor was operated over the range 25 Hz - 75 Hz and the rotary compressor over the range 30 Hz - 90 Hz. Furthermore, it was claimed, the capacity changes were obtained at high efficiency in relation to mains power input.

1.4.1.3 Volumetric Efficiency

Relative, rather than absolute values were quoted. The volumetric efficiency of the reciprocating compressor varied from about 1.04 times the 60 Hz reference value, at 30 Hz, to about 0.87 times, at 90 Hz.

1.4.1.4 Efficiency of the Motor/Inverter

Itami et al. did not quantify the absolute efficiency of the motor/inverter combination. Rather, the change relative to a reference output frequency of 60 Hz was given. On their graphs the efficiency of the inverter-driven motor was about 1.04 times the reference value at 90 Hz and about 0.89 times the reference value at 30 Hz.

The fact that higher motor/inverter efficiencies were
Ch. 1 Background

achieved at frequencies above the mains synchronous value suggested that advantage could be taken of this in specifying the design operating speed ranges of compressors.

1.4.1.5 Overall Efficiency

The overall efficiency characteristic of the inverter driven reciprocating compressor was found by Itami et al. to be approximately flat, varying by no more than about 4 percent from the reference value over the range 25 to 75 Hz. This overall efficiency was not precisely defined in the paper. It was based on the power input to the inverter and included the effects of valve throttling. It was, perhaps, an electrical isentropic efficiency of the form

$$E_{es} = \frac{\dot{m} (h_{2s} - h_1)}{P_e}$$

where

- $E_{es}$ = electrical isentropic efficiency
- $\dot{m}$ = mass flow rate of refrigerant [kg/s]
- $h_1$ = specific enthalpy of the refrigerant at the suction state [J/kg]
- $h_{2s}$ = specific enthalpy after assumed isentropic compression to the discharge pressure [J/kg]
- $P_e$ = electric power input to the inverter [kW]

1.4.1.6 Seasonal Energy Efficiency Ratio

Itami et al. also stated in their paper that a 20 percent to 40 percent improvement in seasonal energy efficiency ratio (SEER), otherwise known as seasonal coefficient of performance (SCOP), could, on the basis of their calculations, be realised in air conditioners equipped with frequency controlled compressors, in comparison with equipment operated under on/off control.

1.4.2 THE WORK OF PAUL ET AL.

Joachim Paul and co-workers at the University of Essen, West Germany, also carried out tests on compressors operated at different speeds. An extensive final report entitled "Basic
Ch. 1 Background


A calorimeter load stand with a mechanical variable speed unit was used to test two open type reciprocating compressors at different speeds and for various operating conditions. The discharge saturation temperatures used were 50°C, 65°C and 80°C.

One of the compressors was a two cylinder type and was tested over the speed range 500 to 1,000 r.p.m. Although the make or model of the compressor was not given in the report, it was established from a scaled photograph of the valve plate, in a paper presented by Paul [5], that its dimensions were the same as those of the valve plate of a Bitzer IV compressor. Thus, the results were particularly interesting and relevant, as the compressor was apparently of the same type as used in the work for this PhD thesis. It is referred to hereafter as 'the low speed open compressor'.

The second open compressor tested by Paul et al. was a four cylinder model. This was a more modern type and was tested over the speed range 500 to 2,000 r.p.m. It is referred to hereafter as 'the high speed open compressor'.

1.4.2.1 Volumetric Efficiency

For the low speed open compressor, using refrigerant R-22 with 10 K of superheat at suction, results were presented for three speeds as a function of pressure ratio. Volumetric efficiency was found to decrease with speed, e.g. for a pressure ratio of 4.4 (approximately the same value as applied for test results to be presented in chapter 7), the following results were read from the graphs presented [4, p.146].

<table>
<thead>
<tr>
<th>Speed/r.p.m.</th>
<th>500</th>
<th>750</th>
<th>1,000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vol. Eff.</td>
<td>0.70</td>
<td>0.68</td>
<td>0.64</td>
</tr>
</tbody>
</table>

Table 1-1

1.4.2.2 Heating Capacity

For the high speed open compressor a comprehensive set of
graphs of heating capacity versus speed was presented for a wide range of operating conditions, Fig. 1-2. It can be concluded from the graphs that heat pump heating capacity was very nearly directly proportional to speed for most operating conditions.

Fig. 1-2 Variation in the 'heat pump heating capacity' with speed of a high speed open compressor for various evaporating and condensing temperatures, t_e/t_c, in °C.
(from [4, p.162])
1.4.3 THE WORK OF BREDESEN, PAUL, ET AL. ON VALVE BEHAVIOUR OVER A RANGE OF SPEEDS

The performance characteristics of a variable speed reciprocating compressor may depend, to a considerable extent, on the dynamic characteristics of the suction and discharge valves. These characteristics are determined by factors such as the masses and stiffnesses of the moving valve elements, the valve areas, and the degree of damping. In a single speed compressor, the valves can be designed to open and close at about the correct moments within the cycle, and without excessive velocities, which could lead to fatigue failure due to impact or bending stresses. The design problem is more difficult, however, if the valves are required to function satisfactorily over a range of speeds.

Bredesen and Paul contributed a pair of complementary papers on valve behaviour, energy losses, and efficiency of reciprocating compressors, under the influence of speed control, to the "2. Essener Wärmepumpentagung" held in Essen, West Germany in Sept. 1978. The two papers were included under a joint heading in a congress report volume [5].

The results presented by Bredesen were based on a computer simulation, as far as variable speed operation was concerned. Comparative experimental results were, however, presented at fixed speed, for a range of suction saturation temperatures. The ideal valve stiffness was defined as that which caused the valve to close without oscillation when pressure equalisation occurred between the gas in the cylinder and the suction or discharge plenum chamber. Bredesen employed a set of design equations, which had been previously derived and validated [6], to specify the ideal valve stiffnesses at the mid point and extremes of the speed range 500 to 1500 r.p.m. The computer model was then used to predict the valve lift behaviour, the volumetric efficiency, and the energy losses due to the valve pressure differences.

From the simulation results Bredesen concluded that the valves designed for 1,500 r.p.m., which were therefore too stiff for low speed operation and tended to close early, oscillated...
excessively at low speeds. The high impact frequencies which resulted were considered undesirable. Volumetric efficiency was good over the full speed range and energy losses increased only at very low speed.

The valves designed for 500 r.p.m. which were too weak for operation at higher speeds and tended to close late, were deemed unsatisfactory in that volumetric efficiency was reduced, particularly towards the upper end of the speed range.

The valves designed for 1,000 r.p.m., the midpoint of the range, were found to give acceptable volumetric efficiency values and energy losses over the full speed range, when compared with the weaker and stiffer valves.

The main conclusion from Bredesen's paper was, thus, that the valves should be designed for the midpoint of the operating speed range.

The paper presented by Paul at Essen described performance tests which were carried out on two different compressors: the low speed open type and the high speed open compressor already referred to in the last section. Piezoelectric pressure transducers and thermocouples were installed in the valve ports of the low speed compressor. Further details, not included in the Essen paper, were contained in [4].

In the low speed open compressor test setup described by Paul, one piezoelectric pressure transducer measured the pressure on the suction plenum side of the inlet valve while the second measured the pressure on the cylinder side of the discharge valve. Results were presented for the amplitude of both waveforms versus speed. For example, with R-12, a suction saturation temperature of -5°C, a discharge saturation temperature of 50°C, and 10 K suction superheat, the amplitude of the suction port pressure waveform varied from about 0.4 bar at 500 r.p.m. to about 0.7 bar at 1,000 r.p.m. These figures represented the amplitude of pressure pulsations within the suction port. A photograph of oscilloscope traces for both piezoelectric pressure transducers was also presented in the paper by Paul. The suction port waveform was jagged, showing that there were considerable pulsations, within the suction port. These
Ch. 1 Background

'pressure' curves resembled suction valve lift curves measured in the course of the work for this PhD thesis, e.g. fig. 7-17, ch. 7. It was therefore concluded that the pulsations related to the instantaneous flow rates through the port and did not necessarily represent the situation in the suction plenum, as Paul seemed to assume.

The discharge port waveform was smooth. It was similar in form to diagrams of the cylinder pressure obtained in the course of work for this PhD thesis.

It was considered that it was possible to quantify the valve losses over a range of speeds more precisely than was done by Paul et al. This was one of the objectives of the work described in this thesis. Whereas in the final report presented by Paul et al., diagrams were presented of excess pressure within the cylinder due to valve characteristics, it was considered that it would be more useful to quantify the valve losses in terms of excess indicated power (see figs. 7-48 and 7-49, ch. 7, for 'suction pumping power' and 'discharge pumping power'). Also, Paul et al. did not have experimental data for the valve lifts. Such data is presented in this thesis (see figs. 7-16 to 7-18, ch. 7).

1.5 COMPUTER CALCULATION OF REFRIGERANT THERMODYNAMIC PROPERTIES

A key aspect of the detailed evaluation of refrigerant compressor performance is the evaluation of thermodynamic properties, such as specific enthalpy, specific volume or specific entropy, from experimental measurements. This function can be carried out within computer programs either using 'table lookup' in conjunction with a data set of refrigerant property values, or, by using empirical equations of state and calculating property values as required. If a refrigerant/oil mixture is to be considered, then, further data or empirical relationships on solubility and the properties of the oil are required. In the work for this thesis extensive use was made of computer programs, based on real fluid equations of state, for refrigerant and refrigerant/oil property evaluation. Subroutines, data,
and empirical relationships from the literature were incorporated into simulation and analysis programs which were written.

1.5.1 PROGRAMS AND DATA FOR PURE REFRIGERANTS

Kartsounes and Erth [7 (1971)] published a complete set of FORTRAN subroutines for the calculation of the thermodynamic properties of refrigerants 12, 22 and 502. These were based on earlier publications of equations and constants by Martin, Downing, and the "Freon" products division of E.I. du Pont de Nemours & Co. The latter company also published a computer program for the calculation of properties under the authorship of Downing and Knight [8 (1971)]. The author [9 (1977)] wrote a small subroutine, for use with those of Kartsounes and Erth, to calculate refrigerant state properties after isentropic compression, given the specific entropy and the final pressure. Downing [10 (1974)] presented equations for the calculation of basic refrigerant properties and tabulated the constants required for twelve refrigerants including 12, 22 and 502. McMullan and Morgan, as an appendix to their book [1 (1981)], published property subroutines and a calling program in FORTRAN, together with the constants required for refrigerants 11, 12, 21, 22, 114 and 502. These subroutines were based on those of Kartsounes and Erth. The same authors, with Hughes [11 (1985)], later published a revised suite of computer programs with data for twelve refrigerants. Their paper also listed the equations in algebraic form and outlined the calculation methods used.

1.5.2 REFRIGERANT/LUBRICANT MIXTURES

Refrigerant R-12 in the liquid state is fully miscible in any concentration with common mineral refrigeration oils. If such an oil is brought into contact with R-12 in the vapour state some of the vapour will go into solution with the oil and the proportion of liquid refrigerant in the mixture will depend on the temperature and pressure. This comes about as the oil has the effect of elevating the boiling point or lowering the vapour pressure of the liquid refrigerant.
Ch. 1 Background

At conditions which would represent a superheated state of pure refrigerant, some small part of the refrigerant in a mixture with oil will remain in the liquid state. No single saturation temperature corresponds with the pressure of the mixture, as would be the case for pure refrigerant. Ideal solutions can be characterised quantitatively by Raoult's law [12], and a solubility relationship can, in principle, be derived on this basis. Spauschus [13 & 14 (1963)] examined the thermodynamic properties of refrigerant/oil solutions and found that R-12/oil solutions exhibited deviations from Raoult's law over the temperature range -30 to 90°C. Spauschus and other authors have quoted solubility relationships for R-12 with mineral oil by Bambach [15] (see Appendix B). These relationships were given in [14] (a copy of the original publication was not located). Cooper and Mount [16 (1972)] used Bambach's solubility relationships, together with empirical relationships for the specific heat capacity and density of a common refrigeration oil, to derive expressions for the specific enthalpy and density of refrigerant/oil mixtures in two phases. Hughes, McMullan et al. [17 (1980)] used these expressions to study the effects of oil circulation on c.o.p. and later [18, 19 (1982)] to produce pressure - specific enthalpy diagrams for refrigerant/oil mixtures. McMullan, Hughes and Morgan [20 (1985)] published a suite of FORTRAN subprograms for use in conjunction with those given in [11] to calculate the approximate thermodynamic properties of refrigerant/oil mixtures. Bambach's solubility relationships were assumed to apply for R-12/oil mixtures, and the solution was assumed to behave as an ideal mixture when calculating the reduction in vapour pressure. Solubility relationships for mixtures of refrigerants other than R-12 with oil were not included.

When the flow of refrigerant contains oil it is not strictly correct to refer to the amount of superheat, since some of the refrigerant remains in the liquid state and there is no fixed saturation temperature. The amount by which the temperature exceeds the saturation temperature of pure refrigerant is termed the 'apparent superheat' hereafter.
1.5.3 PROPERTY PROGRAMS USED

In simulating the steady state performance of the load stand, the subprograms of Kartsounes and Erth [7] were used to calculate the properties of R-12. A further subroutine was written to calculate refrigerant properties after a throttling process, given the pressure and specific enthalpy, and use was made of the author's previously written subroutine to find the properties after an isentropic compression process. These programs were in FORTRAN and were run on a mainframe DEC 2060 computer. It was found that thermodynamic properties calculated using these programs agreed closely with values published in tables, e.g. in [21].

The subroutines of McMullan and Morgan, from their book [1], which were originally in FORTRAN, were rewritten in Hewlett Packard BASIC and were run on an HP9845B desktop computer which was used for the analysis of test results. Here too, the calculated thermodynamic properties were found to agree closely with published tables. A subroutine was written, based on theory outlined in references already mentioned, to calculate the equilibrium properties of an R-12/oil mixture, given the temperature, pressure and oil concentration. This was based on Bambach's solubility equations and Raoult's law. Specific enthalpy and dryness fraction values calculated by the subroutine agreed with those presented graphically by Hughes et al. in [18]. Iteration techniques were used wherever it was required to find equilibrium properties, given two properties other than temperature and pressure.
Chapter 2

UTILISATION OF VOLUME DISPLACEMENT

In this chapter the factors affecting utilisation of volume displacement are analysed. An attempt is made to formulate a coherent system of parameters to quantify these. Reference is made to the work of other researchers and to parameters they have defined. Some new parameters are introduced and inter-relationships are described. The overall purpose of the theory is to apportion shortcomings in the volumetric efficiency of reciprocating compressors to specific causes, on the basis of test measurements.

The measurement problems which would arise in applying the theory fully are not addressed. Indeed, in the course of the experimental work for the thesis the available data did not allow all of the effects which are described to be quantified, nor all of the parameters to be evaluated.

Inevitably, generalisations and approximations must be made in practice and in theory, due to the limitations of test measurements, and in order to separate out and roughly quantify effects which, in reality, overlap, interact and are not totally distinct.

The theory within the following sections is put forward as a set of tools for the detailed analysis of displacement utilisation in reciprocating compressors.

2.1 THE PROCESSES OF THE RECIPROCATING COMPRESSOR CYCLE

Fig. 2-1 illustrates the cycle of a reciprocating compressor as a diagram of cylinder pressure versus volume. In this case, it is assumed that the suction and discharge pressures outside the cylinder remain constant. These are represented by the horizontal lines through points 4 and 1 and through points 3 and 2 respectively.

The cycle can be regarded as four processes. From point 1 to point 2 an enclosed mass of vapour is compressed. This is referred to as the compression process. From point 2 to point 3, vapour passes out of the cylinder through the discharge valve: the discharge process. From 3 to 4 the vapour remain-
Ch. 2 Utilisation of Volume Displacement

![Diagram](image)

Fig. 2-1 An indicator diagram such as might be measured experimentally for a reciprocating compressor, showing overpressure and underpressure caused by valve throttling. In the clearance volume is re-expanded: the re-expansion process. From 4 to 1 vapour enters the cylinder through the suction valve: the induction process.

2.2 DISPLACEMENT UTILISATION EFFICIENCY

The displacement utilisation efficiency, which is also known as the volumetric efficiency, quantifies the utilisation of the volumetric displacement of the piston and is defined as follows:

\[
\eta_v = \frac{\dot{m} v_s}{N_c y V_{sw}}
\]  

(2.1)

where

- \( \eta_v \) = displacement utilisation efficiency, or, volumetric efficiency
- \( \dot{m} \) = actual external mass flow rate produced by the compressor [kg/s]
- \( N_c y \) = number of cycles per unit time [s⁻¹]
- \( V_{sw} \) = swept volume per cylinder [m³]
- \( v_s \) = specific volume of the refrigerant at the suction condition [m³/kg]

Displacement utilisation efficiency can be determined
Ch. 2 Utilisation of Volume Displacement

experimentally from measurements of the mass flow rate and the compressor speed. It is always less than unity due to the factors listed in the following sections.

2.3 VOLUMETRIC INDUCTION EFFICIENCY

The term 'volumetric induction efficiency' is used in this thesis to quantify the mass of refrigerant actually induced into the cylinder, as a fraction of the mass which would occupy the swept volume at the suction pressure and temperature.

The points of suction pressure equalisation, defined as the points on the indicator diagram where the cylinder pressure equals the mean suction pressure (e.g. 4 and 1, fig. 2-1), are used as the basis for evaluating the induced mass, whether or not the valves open or close at these points, and even in the presence of pressure pulsations within the suction and discharge plenums.

\[ E_{\text{ind}} = \frac{m_l V_l}{V_{\text{sw}}}, \]  

(2.2)

where

\[ E_{\text{ind}} = \text{the volumetric induction efficiency} \]
\[ m_l = \text{mass induced [kg]} \]

The mass induced is defined as follows:

\[ m_l = m_1 - m_4 \]  

(2.3)

where

\[ m_1 = \text{mass of refrigerant present within the cylinder at the point of suction pressure equalisation nearest bottom dead centre (point 1, fig. 2-1) [kg]} \]
\[ m_4 = \text{mass of refrigerant within the cylinder at the point of suction pressure equalisation nearest top dead centre (point 4, fig. 2-1) [kg]} \]

The volumetric induction efficiency is not the same as the volumetric efficiency, e.g. some of the induced mass may leak, or flow backwards, through the suction valve to the suction side of the compressor.

2.4 FACTORS WHICH AFFECT DISPLACEMENT UTILISATION

Kleinert and Najork [22] presented an expression for the volumetric efficiency as the product of four efficiency terms which
they described. These terms quantified different influences on the overall utilisation of displacement. Pandeya and Soedel [23] had previously described an approach wherein the losses in mass flow rate were additive.

The interrelationships between the various displacement losses were re-examined, taking account of earlier work, and it was found that the effects were not simply multiplicative or additive. In the following sections the various factors which affect displacement utilisation are discussed.

2.4.1 RE-EXPANSION OF THE CLEARANCE MASS

The refrigerant in the clearance volume when the piston is at the top dead centre position is re-expanded in the initial part of the induction stroke. The volume occupied by this 'clearance mass' of refrigerant at the suction pressure reduces the volume of fresh charge taken into the cylinder correspondingly. The clearance volumetric efficiency of a compressor with ideal valves is given by the well known expression

\[
\lambda_{cl,p} = 1 - r_c \left[ \left( \frac{p_d}{p_s} \right)^{1/n} - 1 \right]
\]

where

- \( \lambda_{cl,p} \) = clearance volumetric efficiency, expressed in terms of the suction and discharge pressures
- \( r_c \) = clearance ratio
- \( p_s \) = suction pressure [bara]
- \( p_d \) = discharge pressure [bara]
- \( n \) = polytropic index for the re-expansion process

It should be noted that \( \lambda_{cl,p} \) depends on the value of the polytropic index for the re-expansion process, \( n \), and decreases with decreasing \( n \), e.g.

For \( p_s = 2.19 \) bara, \( p_d = 9.61 \) bara, \( r_c = 0.027 \)
- if \( n = 1.18 \) then \( \lambda_{cl,p} = 93.2\% \)
- if \( n = 1.00 \) then \( \lambda_{cl,p} = 90.8\% \).

In the case of R-12, an index of 1.18 is usually assumed for a reversible adiabatic process. For an ideal gas, a value
of unity represents an isothermal process. This would involve heat transfer to the gas during re-expansion. The clearance volumetric efficiency, $A_{clp}$, thus includes the effect of heat transfer to the refrigerant during re-expansion. If the polytropic index is determined from an indicator diagram for the re-expansion process, using pressure and cylinder volume (rather than pressure and specific volume) measurements, then other effects such as leakage may also be included.

It is also possible to use a more fundamental expression for the clearance volumetric efficiency, which is written in terms of specific volumes, thus avoiding the use of the polytropic index.

$$A_{c1v} = 1 - r_c \left[ \frac{v_s}{v_d} - 1 \right]$$

(2.5)

where

$A_{c1v}$ = clearance volumetric efficiency, expressed in terms of specific volumes

$v_d$ = specific volume of the refrigerant at the discharge condition [m$^3$/kg]

Using the above expression, the clearance volumetric efficiency is precisely defined using the specific volume values corresponding to the measured suction and discharge pressures and temperatures. Other definitions, having the same form as equation 2.5, in which the specific volumes are precisely specified, could also be used, e.g. the specific volumes could be the mean values within the suction and discharge plenums, or, ideally, the mean values within the cylinder at points 4 and 3 in the cycle (fig. 2-1).

The parameter $A_{c1v}$, as defined in equation 2.5 and based on the suction and discharge conditions, can be regarded as a rough, but precisely defined and measurable estimate, of $A_{c1v}$ or $A_{clp}$ based on points 4 and 3 in the cycle. A better estimate would be one based on measurements within the plenums.
2.4.2 THE INDUCTION PROCESS

2.4.2.1 The Indicated Volumetric Efficiency

This parameter is calculated from measurements on an indicator diagram such as that shown in fig. 2-1.

\[
\eta_{vi} = \frac{V_1 - V_4}{V_{sw}}
\]  \hspace{1cm} (2.6)

where

\[
\eta_{vi} = \text{indicated volumetric efficiency}
\]

\[
V_1, V_4 = \text{volumes at the points of suction pressure equalisation, from the indicator diagram, e.g. fig. 2-1.}
\]

The indicated volumetric efficiency is less than the clearance volumetric efficiency for two main reasons:

1. The volume at the end of the re-expansion stroke (point 4, fig. 2-1) may be greater than that in the case of an ideal compressor with clearance, due to factors such as leakage through the discharge valve during re-expansion.

2. Due to underpressure when the cylinder reaches the bottom dead centre position the induction process continues while the piston begins to move back up the cylinder, and so the volume within the cylinder is less than the sum of the clearance and swept volumes when pressure equalisation occurs.

To quantify the former effect the 're-expansion loss ratio' is defined:

\[
R_{rl} = \frac{V_{sw} + V_{cl} - V_4}{V_{sw}}
\]  \hspace{1cm} (2.7)

where

\[
V_{cl} = \text{clearance volume per cylinder [m}^3\text{]}
\]

This loss ratio quantifies the difference between the volume increase of the actual re-expansion process and that of an ideal compressor having the same value of \(A_{c1v}\), as a proportion of the swept volume.

To quantify the second effect described above, the 'under-pressure loss ratio' is defined:

\[
R_{upl} = \frac{V_{sw} + V_{cl} - V_1}{V_{sw}}
\]  \hspace{1cm} (2.8)

The indicated volumetric efficiency is related to the clearance volumetric efficiency by the loss terms defined in equations 2.7 and 2.8.
Ch. 2 Utilisation of Volume Displacement

\[ \eta_{vi} = \lambda_{ci} V - R_{r1} - R_{d1} \]  \hspace{1cm} (2.9)

Various other effects, which are described subsequently, cause the actual displacement utilisation efficiency to be less than the indicated volumetric efficiency.

2.4.2.2 Throttling at the Suction Valve

As well as causing underpressure at the bottom dead centre position, as already mentioned, this reduces the suction vapour density. Costagliola analysed these effects [24 (1950)] and presented an equation (2.10) for volumetric efficiency, based on an indicator diagram such as that in fig. 2-1.

\[ \eta_{vst} = \eta_{vi} - \frac{Y - 1}{Y} \int \left( 1 - \frac{p}{p_{si}} \right) \frac{dV}{V_{sw}} \]  \hspace{1cm} (2.10)

where

- \( \eta_{vst} \) = volumetric efficiency taking account of suction throttling.
- \( Y \) = ratio of specific heats

A derivation of a more comprehensive form of this equation (due to Gosney) is included in appendix C.

The second term in Costagliola's equation (2.10) takes account of the irreversible suction throttling process which has the overall effect of increasing the temperature and reducing the density at point 1 in the diagram. The equation can also be written in the following form:

\[ \eta_{vst} = \eta_{vi} - \frac{Y - 1}{Y} \frac{W_{si}}{V_{sw} p_{si}} \]  \hspace{1cm} (2.11)

where

- \( W_{si} \) = indicated suction work [J]
  = lower crosshatched area in fig. 2-1.

The right hand term in equation 2.11 represents the loss in volumetric efficiency due to suction throttling.

\[ R_{s1} = \frac{Y - 1}{Y} \frac{W_{si}}{V_{sw} p_{si}} \]  \hspace{1cm} (2.12)

where

- \( R_{s1} \) = volumetric efficiency loss due to suction throttling.
In order to illustrate the likely significance of the right hand term in equation 2.11, typical values from compressor tests are given below.

for
compressor speed = 600 r.p.m.
suction pressure = 2.19 bara
ratio of specific heats = 1.18

typical test results:
\[ \eta_{vi} = 0.865 \]
\[ W_{si} = 6.30 \text{ J} \]
\[ V_{sw \; ps} = 36.33 \text{ J} \]
\[ R_{st} = 0.026 \]
\[ \eta_{vst} = 0.865 - 0.026 = 0.839 \]

Throttling at the suction valve depends not only on the geometry and flow area versus lift characteristics of the valve port and reed, but, also on the stiffness and dynamic characteristics of the reed. Furthermore, if there is a preload on the valve reed, holding it in position, or if there is any tendency for it to adhere to the seat, there may be a delay from the time when pressure equalisation occurs across the valve to the moment when the reed begins to lift. This effect too influences the efficiency loss due to suction throttling.

2.4.2.3 Heat Transfer to the Suction Vapour

Heat transfer which occurs to the refrigerant after it leaves the suction pipe and before the suction valve closes on the induction stroke further reduces the refrigerant density at point 1 in fig. 2-1. Gosney [25 (1951), 26 (1953)] extended the type of analysis carried out by Costagliola to include the effect of heat transfer to the suction vapour: equation 2.13.

A derivation of the equation, based on the approach he used, is given in Appendix C.

\[ \eta_{vstq} = \eta_{vi} - \frac{Q + W_{si}}{V_{swps}} \left( \frac{Y - 1}{Y} \right) \]  
\[ (2.13) \]

where
\[ \eta_{vstq} = \text{volumetric efficiency taking account of suction vapour throttling and heat pick up} \]
\[ Q = \text{heat transfer to the refrigerant per cycle from the point where it leaves the suction pipe to closure of the suction valve [J]} \]

Equation 2.13 is the same as equation 2.11 in the special case
Ch. 2 Utilisation of Volume Displacement

where the amount of heat transfer is zero.

The term representing the loss in displacement utilisation efficiency due to suction heat transfer can be separated out as follows:

$$ R_{q} = \frac{Q}{V_{sw}} \frac{\gamma - 1}{P_{s}} \frac{Y}{\gamma} $$

(2.14)

where

$ R_{q} = $ volumetric efficiency loss due to suction heat transfer

In order to estimate $\eta_{vstq}$, using equation 2.13, the amount of heat transfer to the vapour must be determined. This could be done, if, in addition to the indicator diagram, the mean vapour temperature entering the suction valve and the mean vapour temperatures within the cylinder at points 1 and 4 in fig. 2-1 were available.

Heat transfer occurs to the suction vapour in two stages. The first stage occurs within the suction plenum before the fluid enters the suction valve. It can be evaluated if the mean temperature of the fluid entering the suction valve is measured.

$$ Q_{sp} = m_{E} (h_{sv} - h_{s}) $$

(2.15)

For an ideal gas,

$$ Q_{sp} = m_{E} c_{p} (T_{sv} - T_{s}) $$

(2.16)

where

$ Q_{sp} = $ heat transfer to refrigerant in the suction plenum per cycle [J]

$ m_{E} = $ the mass transferred externally per cycle [kg]

$ h = $ specific enthalpy [J/kg]

$ c_{p} = $ specific heat capacity of refrigerant at constant pressure [J/kgK]

$ T = $ temperature [K]

subscripts:

$ s $ indicates a property mean value in the suction pipe

$ sv $ indicates a property mean value at entry to the suction valve

The second stage of heat transfer to the suction vapour occurs within the cylinder, from point 4 to point 1 of the cycle (fig. 2-1).

$$ Q = Q_{sp} + Q_{cyl} $$

(2.17)

where

$ Q_{cyl} = $ heat transfer to the refrigerant within the
Assuming $T_1$, $T_4$ and $T_{sv}$ are known, the first step in the evaluation of $Q_{cyl}$ is to determine the temperature, $T_{1a}$, which would exist at point 1 if there were no heat transfer during the induction process and if the same indicator diagram applied. The term 'adiabatic suction temperature after induction' is used to denote this quantity. Whereas the throttling process itself produces no temperature change, if ideal gas behaviour is assumed, a temperature increase comes about due to the compression which occurs as the pressure of the induced charge increases adiabatically to the suction pressure at point 1. The derivation of equation 2.18 for the adiabatic suction temperature after induction is included in Appendix C.

\[
T_{1a} = \frac{1}{\frac{V_4}{V_1} \left( \frac{T_1}{T_4} - 1 \right) - \frac{\gamma - 1}{\gamma} \frac{W_{si}}{p_b V_1} + \frac{T_{sv}}{T_1}}
\]  
(2.18)

where

$T_{1a} = \text{adiabatic suction temperature after induction [K]}$

The quantity of heat transfer to the suction vapour, from the start of the induction process to the start of the compression process, is given by equation 2.19 (a derivation is given in Appendix C).

\[
Q_{cy1} = \frac{\gamma}{\gamma - 1} p_b V_1 T_s \left( \frac{1}{T_{1a}} - \frac{1}{T_1} \right)
\]  
(2.19)

Thus, a procedure has been presented whereby $T_{1a}$, $Q$ and thence $\eta_{vstq}$ can be determined from an indicator diagram with the corresponding mean gas temperatures at the points of suction pressure equalisation (4 and 1 in fig. 2-1) and the mean temperature of the refrigerant entering the suction valve.

2.4.3 THROTTLING AT THE DISCHARGE VALVE

Throttling at the discharge valve may cause overpressure at the TDC position. However, the amount of piston movement which occurs before pressure equalisation with the discharge plenum is slight. Any effect due to overpressure is already taken into
Ch. 2 Utilisation of Volume Displacement

account in the indicated volumetric efficiency, equation 2.6, and thus in equations 2.11 for \( \eta_{\text{vst}} \) and 2.13 for \( \eta_{\text{vstq}} \).

A further term, similar to Gosney's correction term to the indicated volumetric efficiency in equation 2.13, could be included in equations 2.11 and 2.13 to take account of the density change due to discharge throttling and heat transfer. This effect can generally be neglected, however, on the basis that the clearance ratio is small, typically about 3%, and so density changes when the piston is near the top dead centre position are much less significant than density changes when it is near bottom dead centre.

2.4.4 PLENUM PRESSURE PULSATIONS

Pulsations occur in the suction and discharge plenum chambers (see fig. 2-2) and also in the intake and discharge pipes. In principle these pressure pulsations could enhance volumetric efficiency if the suction or discharge pulsations were favourably 'tuned'. However, they may adversely affect volumetric efficiency, particularly if the compressor is operated over a range of speeds. These effects are taken into account in backflow ratios described subsequently. As noted in section 1.4.1.1, ch. 1, a damper was fitted to a variable speed compressor described by Itami et al. [2] to prevent discharge piping resonance.

2.4.5 EVAPORATION, CONDENSATION AND ENTRY OF REFRIGERANT INTO SOLUTION

Gosney [26 (1953)] reported on and discussed an effect whereby the volumetric efficiency of a compressor increased from about 65% to 75% when the suction temperature was increased by about 24 K. The refrigerant used was R-12 and the original suction temperature was 2.2°C, at an evaporating temperature of -17.6°C. The condensing temperature was 30°C.

Gosney discussed the effects of suction vapour condition on volumetric efficiency and referred to a number of researchers who had found the volumetric efficiency of ammonia compressors to be higher with dry suction than with wet suction (i.e.
Fig. 2-2 A hypothetical cycle (description overleaf).
Fig. 2-2 (on previous page) A hypothetical cylinder pressure versus time diagram and corresponding valve lift diagrams, such as might be measured experimentally, are shown. Plenum pulsations about the mean suction and discharge pressures, $p_s$ and $p_d$, are also shown. The diagram on the bottom of the page illustrates how the mass within the cylinder might vary with time. Due to leakage effects the mass within the cylinder does not remain constant while the valves are closed for this hypothetical cycle.

saturated vapour. It had also been found by others (references were given by Gosney) that the volumetric efficiency of a compressor pumping R-12 increased significantly with increased superheat. Gosney referred to the following possible explanations for the variation of volumetric efficiency with suction superheat.

A. Liquid droplets, which could be present even in superheated vapour, could impinge on the cylinder walls and evaporate, reducing the volume of vapour taken in. Evaporation of these particles could continue during re-expansion, further reducing the volumetric efficiency.

B. In the later part of the compression stroke condensation of the high pressure vapour could take place on the cylinder walls. The mass discharged could thus be reduced and the liquid could evaporate again during the re-expansion stroke.

C. Refrigerant, particularly R-12 which is highly soluble in mineral oil, could be stored in the clearance space, by dissolving in the oil present within the cylinder, to come out of solution during either re-expansion or induction.

It appeared to Gosney that cyclic evaporation and condensation, perhaps involving interaction with the lubricating oil, could occur within refrigerant compressors under at least some operating conditions.

Cooper and Mount [16] described the effects of oil circulation on the evaporator cooling capacity of compressors. They pointed out that, at low superheat values, refrigerant R-12 was highly soluble in the lubricating oil and that this had implications for both the density and the specific enthalpy of the mixture. Hughes, McMullan et al. pursued these ideas in relation to heat pump performance [17, 27]. Their results showed that lubricating oil concentration could have a significant effect on the evaporator performance and heat pump c.o.p.
of refrigeration plant and heat pumps. They did not consider volumetric efficiency separately.

2.4.5.1 Entrained Liquid Droplets

Entrained liquid droplets in wet suction vapour would certainly reduce the volumetric efficiency by greatly increasing the sensitivity of the specific volume of the induced charge to any heat transfer within the suction plenum or cylinder and to suction throttling. The ideal gas model used in deriving equation 2.13 for $\eta_{vstq}$ would not apply.

Entrained liquid droplets in superheated suction vapour would not be in a state of equilibrium with the vapour, but, they would reduce the volumetric efficiency in the same way as liquid droplets in wet suction vapour. The practical difficulty in establishing the true volumetric efficiency, as with wet suction, would be to establish the true specific volume of the suction vapour: measurements of pressure and temperature alone would not suffice.

Care is needed, therefore, both in compressor installations and in performance testing, to ensure that liquid droplets of refrigerant are not present in the suction vapour.

2.4.5.2 Condensation Within the Cylinder

Consideration of the thermodynamics of a two phase substance leads to the conclusion that condensation of a pure refrigerant within the cylinder would only occur if the cylinder wall temperature were lower than the refrigerant saturation temperature corresponding to the instantaneous pressure at some point during the cycle.

2.4.5.3 Entrained Oil Droplets

Whenever there are entrained droplets of oil present within the refrigerant passing through a compressor, some of the refrigerant remains in the liquid state, dissolved in the oil. The proportion of the refrigerant which is in the liquid state changes with the pressure and temperature conditions. If the concentration of oil in the total mixture is known then the
Ch. 2 Utilisation of Volume Displacement

thermodynamic properties at any state can be calculated e.g. using the computer programs of Hughes, McMullan et al. [11]. Analysis techniques are thus available. Derivations such as that given in appendix C for $\eta_{\text{vs}}$, equation 2.13, which assume ideal gas behaviour, are not applicable, but, equivalent parameters can be calculated using computer methods. However, if the oil concentration is sufficiently low (less than 1% perhaps) and the apparent superheat is greater than about 5 K, then reasonable accuracy can be achieved by treating the refrigerant as pure, or even treating it as an ideal gas. In the analyses leading to the test results presented in chapter 7, measured values of the oil concentration were used in evaluating the thermodynamic properties of the refrigerant/oil mixture.

2.4.5.4 Cyclic Solubility and the Phantom Mass

As pointed out by Gosney, refrigerant vapour could enter into solution with liquid oil, present as droplets or on surfaces, during the compression stroke, and release of this vapour could occur from the oil during re-expansion. Kleinert and Najork [22] also referred to this type of effect. Although the concentration of oil in the refrigerant vapour when it leaves the compressor cylinder may be very small, the ratio of oil mass to refrigerant mass is likely to be high within the cylinder due to presence of oil coating the cylinder walls.

The questions discussed here are addressed in chapter 7, section 7.5.5.8, with the support of test observations. It is concluded on the basis of the work done for this thesis that a 'phantom mass' of refrigerant may dissolve cyclically in the lubricating oil, reducing the induced mass correspondingly.

As a thesis, it is assumed that oil on the cylinder surfaces, which is at a temperature near the cylinder wall temperature, contains a mass of refrigerant, and that the refrigerant and oil would form a saturated solution at this temperature and at a pressure which lies somewhere between the suction and discharge values. At higher pressures the oil can accommodate more refrigerant, while at lower pressures it can accommodate less and refrigerant is released from solution. Refrigerant
Ch. 2 Utilisation of Volume Displacement

thus evolves from the oil during the later part of the re-expansion process, the induction process and the early part of the compression process. Conversely, refrigerant goes into solution with the oil during the later part of the compression process, the discharge process and the early part of the re-expansion process. Insofar as the quantity of the oil/refrigerant mixture on the surfaces does not change from one cycle to the next, the oil and liquid refrigerant in the mixture can be regarded as 'resident' in the cylinder.

That part of the resident refrigerant which evolves from the resident oil/refrigerant mixture during the cycle is not discharged, but, occupies part of the cylinder volume during the induction process, reducing the intake of fresh charge, and subsequently dissolves again. This mass of refrigerant which dissolves cyclically in the oil is mainly in the liquid state at the end of the discharge process and is mainly in the vapour state at the end of the induction process. The term 'phantom mass' is used for that mass which dissolves in the oil on the cylinder surfaces from point 1 to point 3 of the cycle (fig. 2-1) and evolves from the oil from point 3 to point 1. The resident mass of refrigerant would be higher than the phantom mass, but, refrigerant which remains in the liquid state does not significantly affect the volumetric efficiency.

Evolution of part of the phantom mass influences the re-expansion process and is one of the effects included in the re-expansion loss ratio. That part of the phantom mass which evolves from point 4 to point 1 of the cycle reduces the induced mass from the value which would be implied from an analysis of the indicator diagrams, assuming only the vapour phase is present.

A phantom mass ratio is used to express the phantom mass as a fraction of the mass of refrigerant which would occupy the swept volume at the suction condition.

$$v_{pm} = \frac{m_p}{V_{sw}}$$  \hspace{1cm} (2.20)

where

$v_{pm}$ = the phantom mass ratio
An underlying assumption in the analyses leading to equation 2.13 for \( \eta_{\text{vstq}} \) is that the increase in volume which occurs between points 4 and 1 of the cycle is due to flow of refrigerant vapour into the cylinder. The equation must be modified to take account of the cyclic solubility effect. It is assumed that specific volume of refrigerant in the liquid state is negligible in comparison with the specific volume of refrigerant vapour.

\[
m_l = m_{lv} - m_{4v} - k \, m_p
\]  

where

\[
m_l = \text{the mass induced per cycle [kg]}
\]
\[
m_{lv} = \text{mass of refrigerant in the vapour phase at point 1 (fig. 2-1 or fig. 2-2) [kg]}
\]
\[
m_{4v} = \text{mass of refrigerant in the vapour phase at point 4 (fig. 2-1 or fig. 2-2) [kg]}
\]
\[
k = \text{a fraction such that } 0 < k < 1
\]

\( k \) = that part of the phantom mass which evolves during the induction process

The induced mass, thus, excludes refrigerant which comes out of solution from within the oil on the cylinder surfaces between points 1 and 4. However, it should be noted that equation 2.3 for the induced mass is also valid, even when the cyclic solubility effect is present, i.e.

\[
m_l = m_1 - m_4
\]  

where

\( m_1 \) and \( m_4 \) refer to the total charge of refrigerant in the cylinder, including resident refrigerant dissolved in the lubricating oil.

From equation 2.21,

\[
m_l = \eta_{\text{vstq}} \left( \frac{V_{sw}}{V_s} \right) - k \, m_p
\]

\[
\eta_{\text{vstq}} = \frac{m_l}{\left( \frac{V_{sw}}{V_s} \right)} - k \, \nu_{pm}
\]

Hence,

\[
\eta_{\text{vstqs}} = \eta_{\text{vstq}} - k \, \nu_{pm}
\]  

where

\( \eta_{\text{vstqs}} \) = volumetric efficiency, taking account of suction throttling, heat transfer and solubility effects.
Ch. 2 Utilisation of Volume Displacement

In equation 2.22 a fraction, \( k \), of the phantom mass ratio, \( V_{pm} \), appears as a loss in volumetric efficiency. However, the balance of the phantom mass ratio, i.e. \( (1-k)V_{pm} \), is already included as a loss within \( \eta_{vstq} \). That part of the phantom mass which evolves during the re-expansion process contributes to the re-expansion loss ratio \( R_{ri} \) (equation 2.7), which is included as a loss in the indicated volumetric efficiency \( \eta_{vi} \) and thus in \( \eta_{vstq} \). The entire phantom mass ratio, therefore, represents a loss in volumetric efficiency.

2.4.6 DESCRIPTION OF A HYPOTHETICAL REAL CYCLE

Fig. 2-2 illustrates many of the aspects of actual reciprocating compressor cycles, including leakage, backflow and non-ideal valve behaviour. A description of the processes follows:

TDC: Cylinder volume is at a minimum. The discharge valve is still open. Due to pulsations, the pressure in the discharge plenum is higher than the mean pressure in the discharge pipe, \( p_d \). Refrigerant is continuing to flow out of the cylinder through the discharge valve since the pressure within the cylinder is still higher than in the discharge plenum.

TDC to 3': Cylinder pressure falls and the discharge valve continues to close. The mass within the cylinder is still reducing. The rate of increase of cylinder volume is low. At 3' the pressure difference across the discharge valve is zero and hence the instantaneous flow rate is zero.

3' to 3: The cylinder pressure is just below the discharge plenum pressure and backflow through the discharge valve commences. At 3 the cylinder pressure is equal to the mean pressure in the discharge plenum, \( p_d \).

3 to 3'': The discharge valve is still open. Backflow begins and continues as the piston moves away from top dead centre and the cylinder pressure falls. The mass of refrigerant within the cylinder increases. At 3'' the discharge valve closes.

3'' to 4': Both valves are closed and re-expansion of the clearance mass continues. Leakage into the cylinder may occur through the discharge valve and leakage out of the cylinder may occur through the suction valve and past the piston. As the pressure within the cylinder falls some refrigerant, dissolved in oil coating the cylinder surfaces, may come out of solution. This effect would increase the mass of vapour within the cylinder. At 4' the pressure difference across the suction valve is zero. However, the valve remains closed.

4' to 4: Due to pulsations the pressure within the suction plenum is higher than the mean suction pressure, although it is subject to a lifting force due to the pressure difference across
it. Re-expansion continues. The suction valve is still closed. At 4 the pressure within the cylinder is equal to the mean pressure in the suction plenum. Point 4 is the first point of suction pressure equalisation.

4 to 4'': Re-expansion continues and at 4'' the suction valve opens.

4'' to 1: As the piston moves down the cylinder the cylinder pressure is less than the suction plenum pressure and vapour is induced into the cylinder. Maximum volume is reached at bottom dead centre, but, the mass of vapour within the cylinder continues to increase since the valve remains open. At point 1, the second point of suction pressure equalisation, cylinder pressure equals \( p_a \).

1 to 1': As the suction valve is still open the flow of refrigerant into the cylinder continues until the pressure difference across the valve becomes zero at 1'.

1' to 1'': Due to the reversed pressure difference, backflow occurs through the suction valve until it closes at 1''.

1'' to 2 & 2': Both valves are closed and the charge is compressed. As the pressure increases, some refrigerant may dissolve in lubricating oil on the cylinder surfaces, reducing the mass of vapour present. Leakage also proceeds from the discharge plenum and to the suction plenum. At 2, the cylinder pressure equals the mean discharge pressure, \( p_d \), and, by chance, the plenum pressure has the same value.

2 & 2' to 2'': Cylinder pressure rises above the plenum value, but, no flow occurs as the valve has not opened. The discharge valve only begins to lift at 2''.

2'' to TDC: Discharge of refrigerant from the cylinder proceeds. The next cycle continues from the top dead centre position.

2.4.7 BACKFLOW AND LEAKAGE

2.4.7.1 Backflow Through the Suction and Discharge Valves

Due to late closure of the suction valve, as illustrated in fig. 2-2, refrigerant may flow back into the suction plenum. Similarly, backflow may occur from the discharge plenum into the cylinder through the discharge valve if it closes late. This backflow could be quantified by analysis of measured pressure-time diagrams in conjunction with valve lift diagrams. The pressure and valve lift diagrams in fig. 2-2 could be used for this type of analysis. Data on the flow characteristics of the valves as a function of lift would also be required in order to calculate the instantaneous backflow rates. An alternative
Ch. 2 Utilisation of Volume Displacement

approach would be to use compressor simulation. Careful validation of the model would be necessary to ensure reliability of the results. Parameters to quantify backflow rates are defined subsequently.

2.4.7.2 Leakage Past the Piston and Through the Valves

Figure 2-3 shows, schematically, the types of backflow and leakage paths which can occur within a compressor. The symbols used are explained in section 2.4.7.3. Leakage past the piston and past the suction valve when it is seated causes a loss in mass from the cylinder during compression, during the discharge process, and during re-expansion. A further consequence of this would be an apparent reduction in the polytropic index of the compression process determined from an indicator diagram [28] and an apparent increase in the index for the re-expansion process. In a similar way, leakage through the discharge valve when it is closed would tend to increase the mass in the cylinder during re-expansion, during the induction process and during compression. It would cause an apparent reduction in the polytropic index for the re-expansion process and an increase in the index for the compression process.

For well designed valves, leakage rates would be expected to be negligible, provided the contact surfaces were clean and undamaged.

Ferreira and Lilie [29 (1984)] described a theoretical model for the calculation of the leakage past a piston without rings, which they found to give good agreement with experimental results. The method included the use of Bambach's solubility equations to take account of refrigerant dissolved in the oil which leaked past the piston. Lui and Yu [30 (1986)] described a theoretical model for the leakage past a piston with rings. This included the use of an empirical 'oil sealing coefficient'.

2.4.7.3 Leakage and Backflow Ratio Definitions

With reference to fig. 2-3,

\[
\begin{align*}
\dot{m}_E &= \dot{m}_L - \dot{m}_p - \dot{m}_{bs} - \dot{m}_{ls} = \dot{m}_D - \dot{m}_{bd} - \dot{m}_{ld} \\
\end{align*}
\] (2.23)
Fig. 2-3 Fluid transfer paths of a reciprocating compressor.
Ch. 2 Utilisation of Volume Displacement

where

\[ m_{bs} = \text{the mass which flows back through the suction} \]
\[ \text{valve per cycle (due to late closure and due to plenum} \]
\[ \text{pulsations) - this can have a negative value, i.e. the volumetric} \]
\[ \text{efficiency can be enhanced [kg]} \]

\[ m_s = \text{mass leakage through the suction valve per cycle} \]
\[ \text{(while the valve is seated) [kg]} \]

\[ m_p = \text{mass which leaks past the piston per cycle [kg]} \]

\[ m_d = \text{mass discharged into the discharge plenum per} \]
\[ \text{cycle [kg]} \]
\[ = m_2 - m_3 \text{ (refer to indicator diagram fig. 2-1, or, to fig. 2-2)} \]

\[ m_{bd} = \text{mass which flows back through the discharge} \]
\[ \text{valve per cycle (due to late closure and due to plenum} \]
\[ \text{pulsations) [kg]} \]

\[ m_a = \text{mass leakage through the discharge valve per} \]
\[ \text{cycle (while valve is seated) [kg]} \]

Mass transfers due to leakage take place at different parts of the cycle as follows:

\[ m_g: \text{occurs throughout the cycle, but, mainly from 4 to 1} \]
\[ \text{on the suction side and from 2 to 3 on the discharge side.} \]

\[ m_l: \text{occurs, by definition, from 4 to 1.} \]

\[ m_{bs}: \text{occurs from 1 to 1''. From 1 to 1' it is negative,} \]
\[ \text{while from 1' to 1'' it is positive.} \]

\[ m_s: \text{occurs from 1'' to 4''. It has a positive value} \]
\[ \text{from 1'' to 4'', which is likely to be significant only if} \]
\[ \text{the sealing faces are damaged. From 4' to 4'' it has a} \]
\[ \text{negative value, which is probably negligible due to the} \]
\[ \text{small pressure differences involved.} \]

\[ m_{ld}: \text{occurs from 1' to 4'.} \]

\[ m_p: \text{occurs, by definition, from 2 to 3} \]

\[ m_{bd}: \text{occurs from 3' to 3''} \]

\[ m_d: \text{occurs from 3'' to 2''. It has a positive value} \]
\[ \text{from 3'' to 2'', which is likely to be significant only if} \]
\[ \text{the sealing faces are damaged. From 2' to 2'' it is} \]
\[ \text{negative and probably negligible due to the small pressure} \]
\[ \text{differences involved.} \]

Leakage ratios can be defined as follows:

\[ \psi_{bs} = \frac{m_{bs}}{m_l} \quad \text{suction valve backflow ratio} \quad (2.24) \]

\[ \psi_i = \frac{m_s}{m_l} \quad \text{suction valve leakage ratio} \quad (2.25) \]
Ch. 2 Utilisation of Volume Displacement

\[ \nu_{lp} = \frac{m_{lp}}{m_l} \quad \text{ratio for leakage past piston} \quad (2.26) \]

\[ \nu_{bd} = \frac{m_{bd}}{m_b} \quad \text{discharge valve backflow ratio} \quad (2.27) \]

\[ \nu_{ld} = \frac{m_{ld}}{m_b} \quad \text{discharge valve leakage ratio} \quad (2.28) \]

The overall 'suction side sealing efficiency' can be defined as

\[ \eta_{ss} = 1 - \nu_{bs} - \nu_{ls} - \nu_{lp} \quad (2.29) \]

Similarly a 'discharge side sealing efficiency' can be defined as

\[ \eta_{ds} = 1 - \nu_{bd} - \nu_{ld} \quad (2.30) \]

The influence of the discharge side sealing efficiency is already included in the expression for \( \eta_{vstqs} \) (eqn. 2.22), as the mass of vapour at point 4 on an actual indicator diagram has been increased due to leakage and backflow from the discharge plenum. The volumetric efficiency allowing for valve throttling, heat transfer to the suction vapour and leakage is given by

\[ \eta_{vstqs1} = \eta_{vstqs} \eta_{ss} \quad (2.31) \]

where

\[ \eta_{vstqs1} = \text{volumetric efficiency taking account of suction vapour throttling, heat transfer, solubility and leakage.} \]

2.4.8 OTHER FACTORS

Throttling of the vapour entering the suction plenum chamber, from the inlet pipe where the suction pressure is measured, will reduce volumetric efficiency. This effect is included in the equations already listed.

There may also be a loss of refrigerant from the cylinder with lubricating oil which bypasses the piston, as the solubility of refrigerant in the oil is high at the conditions encountered within the cylinder. This is, in fact, another form of leakage and would be included in the ratio for leakage past the piston.
2.5 CONCLUSIONS

The processes which occur within a reciprocating compressor and particularly the factors which affect its displacement utilisation have been described. These factors have been discussed, with reference to the literature, and parameters have been defined to assist in their quantification.

The expressions given in this chapter, which follow on from the work of Costagliola, are based on the assumption that the refrigerant can be regarded as an ideal gas. These expressions do not strictly apply to real refrigerants which have rather complicated empirical equations of state, or, to refrigerant/oil mixtures. Equivalent parameters to those described in this chapter can be calculated for real refrigerant/oil mixtures, but, cannot be written as relatively simple expressions involving a small number of parameters. The ideal gas assumption is therefore a useful tool in identifying and quantifying, if a little crudely, the main factors which influence the utilisation of volume displacement.

As already mentioned, a more accurate analysis, making use of computer subroutines for the evaluation of the thermodynamic properties, was carried out to yield the compressor performance characteristics which are presented in chapter 7.
Chapter 3

UTILISATION OF SHAFT POWER

In the field of refrigerant compressor engineering an isentropic (i.e. reversible and adiabatic) compressor has been regarded as the ideal model for the purposes of quantifying shaft power utilisation. The particular discharge state produced by each compressor type has, generally, been ignored. The isentropic model is readily applied and provides a consistent basis for the comparison of different compressors. Its drawback, however, is that it may not always lead to an optimum overall design for a cyclic vapour compression plant and may lead to difficulties in the attribution of power utilisation deficiencies to the true causes.

It is sometimes necessary to make a distinction between refrigerant compressors used for refrigeration, i.e. to provide a cooling effect, and those used for heat pump applications, i.e. to provide a heating effect. It will be seen that this distinction, on the basis of function, is necessary when the effectiveness of shaft power utilisation is being considered.

In this chapter compressor efficiencies are described and discussed and a case is made for quantifying shaft power utilisation in a way which recognises the thermodynamic worth of the discharge refrigerant state produced by a compressor (and of intermediate states in more detailed analyses), using the principles of availability analysis.

In the previous chapter, losses in displacement utilisation due to many different effects were described. It would be logical to attempt the same type of breakdown of shaft power utilisation, using the concept of availability analysis. Such a detailed application of availability analysis is outside the scope of this thesis. However, the technique which is described, and applied to the analysis of the overall shaft power utilisation, has scope for the detailed attribution of losses to discrete causes, such as heat transfer with finite temperature differences, throttling, and perhaps even cyclic solubility.
3.1 MECHANICAL EFFICIENCY

The mechanical efficiency of a compressor is the ratio of the work done on the refrigerant vapour to the shaft work input. It can also be expressed as the ratio of indicated power to shaft power, equation 3.1. The indicated power is normally calculated according to equation 3.2.

\[ E_{\text{mech}} = \frac{\text{indicated power}}{\text{shaft power}} \]  

\[ P_i = P_{\text{me}} V_{\text{sw}} N_{\text{cy}} \]  

where

- \( E_{\text{mech}} \) = mechanical efficiency
- \( P_i \) = indicated power [W]
- \( P_{\text{me}} \) = mean effective pressure calculated from the indicator diagram [N/m²]
- \( V_{\text{sw}} \) = swept volume [m³]
- \( N_{\text{cy}} \) = number of cycles per unit time [s⁻¹]

The difference between the shaft power and the indicated power is due to:

a) friction between the moving parts of the compressor i.e. between the piston and cylinder, at the connecting rod bearings, and at the crankshaft bearings

b) the power used in distributing oil within the compressor (The Bitzer IV compressor did not have an oil pump, but, used splash lubrication.) and

c) windage losses within the crankcase and around the external flywheel.

These are termed mechanical losses. The total mechanical power loss can be determined experimentally by subtracting the indicated power from the shaft power input.

3.2 TWO DEFINITIONS OF INDICATED EFFICIENCY

The terms 'indicated isentropic efficiency' and 'indicated isothermal efficiency' are commonly used to express 'ideal theoretical work' as a fraction of the indicated work. The former parameter is used for refrigerant compressors and the latter is normally used for gas compressors. The reason for the two different efficiency definitions is that a refrigerant compressor is usually employed as part of a thermodynamic cycle which produces a heating or cooling effect, whereas a gas compressor, generally, has the sole purpose of increasing the...
pressure of the gas, without necessarily changing its temperature.

\[ E_i_s = \frac{\text{ideal power for reversible adiabatic compression}}{\text{indicated power}} \quad (3.3) \]

\[ E_i_t = \frac{\text{ideal power for rev. isothermal compression}}{\text{indicated power}} \quad (3.4) \]

where

\[ E_{i_s} = \text{indicated isentropic efficiency} \]
\[ E_{i_t} = \text{indicated isothermal efficiency} \]

### 3.2.1 ISENTROPIC EFFICIENCY

Isentropic efficiency (sometimes called the adiabatic efficiency) is defined by the following equation:

\[ E_{i_s} = \frac{\dot{m} (h_{2s} - h_1)}{P_1} \quad (3.5) \]

where

\[ h_1 = \text{specific enthalpy evaluated at the compressor suction condition [J/kg]} \]
\[ h_{2s} = \text{specific enthalpy evaluated after an assumed isentropic compression process to the discharge pressure [J/kg]} \]
\[ \dot{m} = \text{the actual externally measured mean mass flow rate of the compressor [kg/s]} \]

Note: the kinetic energy of the fluid at compressor suction and discharge is assumed to be negligible.

The isentropic efficiency which, as defined in equation 3.5, includes the externally measured compressor mass flow rate, incorporates three fundamentally different types of effect. The numerator in the fraction, the ideal power for isentropic compression, is not equal to the denominator, the indicated power, because:

1. The actual compression process is not reversible. In addition to a non flow compression process which is somewhat irreversible, mainly due to temperature gradients, the fluid undergoes two flow throttling processes, which are highly irreversible, in passing through the suction and discharge valves. The irreversibilities of the valve flow processes cause an increase in the area of the indicator diagram and thus in the indicated power. A small proportion of the mass which passes through the compressor
also undergoes the irreversible processes of backflow or leakage through the valves or leakage past the piston. These effects always tend to reduce the indicated isentropic efficiency.

2. There is heat transfer to and from the refrigerant as it passes through the compressor. A steady flow energy equation can be written as follows:

\[ W_c = (h_2 - h_1) + q_c \]  

(3.6)

where

\[ W_c \] = work of compression (done on vapour) per unit mass [J/kg]
\[ h_2 \] = specific enthalpy evaluated at the actual compressor discharge condition [J/kg]
\[ q_c \] = net heat transfer from the vapour during compression per unit mass [J/kg]

Any heat transfer from the vapour during compression increases the indicated isentropic efficiency by reducing the required work of compression and hence the indicated work. Similarly, any heat transfer to the vapour during compression reduces the isentropic efficiency. If indicated isentropic efficiency were to be used as the basis for compressor design optimisation the compression process would need to approach the isothermal model in order to maximise the calculated 'isentropic' efficiency. Since an ideal isothermal compressor would have a lower work input than an ideal isentropic compressor operating between the same pressures, this could, in theory, result in the isentropic efficiency being greater than unity. However, it is pointed out in the next section that an isothermal compressor is unsuitable for refrigeration or heat pump vapour compression cycles.

3. Some of the vapour, on which work has been done by the piston, leaks back into the suction plenum or back into the cylinder from the discharge plenum. This causes a decrease in the indicated isentropic efficiency due to a reduction in the external mass flow term in equation 3.5 which is proportionally greater than any reduction in the indicated power.
Indicated isentropic efficiency is discredited as an efficiency measure of refrigerant compressor performance since its value can conceivably be greater than unity, as explained in point 2. Furthermore, in the case of heat pump compressors, this parameter does not attribute any additional value to high temperature discharge vapour or provide any distinction between different discharge temperatures.

### 3.2.2 ISOThermal EFFICIENCY

Isothermal efficiency is defined in equation 3.7.

\[
E_{i_{it}} = \frac{\dot{m} [h_{2t} - h_{1} + q_{1}]}{P_{1}}
\]

\[
E_{i_{it}} = \frac{\dot{m} [h_{2t} - h_{1} + T_{1}(s_{2t} - s_{1})]}{P_{1}}
\]

where

- \( h_{2t} \) = specific enthalpy evaluated at the discharge pressure and the suction temperature [J/kg]
- \( q_{1} \) = heat transfer per unit mass from the gas during an assumed constant temperature compression process [J/kg]
- \( T_{1} \) = suction temperature [K]
- \( s_{1} \) = specific entropy at the suction condition [J/kgK]
- \( s_{2t} \) = specific entropy at the discharge pressure and the suction temperature [J/kgK]

Note: \( h_{2t} \) and \( s_{2t} \) are only valid if the fluid is in a gaseous or vapour state at the discharge pressure and the suction temperature.

For an ideal gas, specific enthalpy is a function of temperature only. The isothermal compression work is therefore equal to the isothermal heat rejection. The expression for indicated isothermal efficiency can be written as follows:

\[
E_{i_{it}} = \frac{\dot{m} p_{1} v_{1} \ln(p_{2}/p_{1})}{P_{1}}
\]

\[
E_{i_{it}} = \frac{\dot{m} R T_{1} \ln(p_{2}/p_{1})}{P_{1}}
\]

where

- \( p_{1} \) = suction pressure [Nm⁻²]
- \( p_{2} \) = discharge pressure [Nm⁻²]
- \( v_{1} \) = suction specific volume [m³/kg]
- \( R \) = specific gas constant [J/kgK]
While isothermal efficiency is a very important and useful figure of merit for gas compressors it has little direct relevance to refrigerant compressors since

1. in a refrigeration system the compressor must raise the temperature of the refrigerant to at least the temperature of the surroundings so that heat rejection can occur.
2. in a heat pump system the compressor must raise the temperature of the refrigerant to at least the temperature of the medium to be heated.

3.3 An isentropic/isothermal compression process

The concept of isothermal efficiency could be applied to the second or higher stages of a refrigerant compressor or, conceptually, to the part of the compression process in which the temperature of the refrigerant is greater than or equal to the temperature of the reservoir to which heat rejection occurs.

The ideal refrigerant compressor would compress the vapour isentropically until the heat rejection temperature (condensing temperature) was reached and then isothermally until the dry saturated condition was reached as in Fig. 3-1b.

![Fig. 3-1 Comparison between a) an ideal isentropic compression process 1-2 and b) an ideal compression process 1-a-2 consisting of an isentropic and an isothermal stage.](image)

However, the process shown in fig. 3-1b, for which the ideal work of compression can be readily evaluated, is still unsatisfactory as an ideal with which to compare the actual
indicated work of a compressor. The following are some considerations.

a) Two heat pump compressors which operated on the basis of process 1-a-2 would not be of equal worth if one transferred even part of the heat rejected during compression to the heated medium, while the second transferred the heat unadvantageously (and irreversibly) to the surroundings.

b) In a refrigeration plant heat rejection in the condenser always occurs at a temperature higher than that of the surroundings. The relationship between the condensing temperature and the temperature of the cooling medium ('the surroundings') depends on the characteristics of the condenser being used. It is conceivable that a compressor could operate on the basis of a process such as 1-a-b-c-2 as shown in fig. 3-2. The work of compression would be less than for process 1-a-2 shown in fig. 3-1b.

c) Actual reciprocating compressors, even those with intercooling, do not involve processes which approximate to the isothermal compression process a-2 of fig. 3-1b.

Fig. 3-2 An ideal compression process in which part of the heat rejection during compression occurs at the temperature of the surroundings and part occurs at the condensing temperature, i.e. the saturation temperature corresponding to the discharge pressure.

3.4 SECOND LAW RATIONAL EFFICIENCY
The concept of availability analysis has been described in the literature, e.g. by Moran [31 (1982)] and by Kotas [32 (1985)].
For specified entry and exit states of a fluid passing through a steady flow system and for a specified temperature of the surroundings the maximum work which can be done by the fluid or the minimum work which must be done on the fluid is given by the change in the flow availability function between the two states. The value of this function depends on the temperature of the surroundings as well as on the thermodynamic state properties. Furthermore, the maximum or minimum work quantity can be achieved only if the flow process between the states is reversible (internally and externally). The approach is readily applied to compressors.

The ideal compression work and a rational efficiency based on availability analysis can be defined for the compression process as follows:

\[
\begin{align*}
W_{\text{ideal}} &= b_2 - b_1 \\
&= h_2 - T_0 s_2 - (h_1 - T_0 s_1) \\
E_{ir} &= \frac{\dot{m} W_{\text{ideal}}}{P_1}
\end{align*}
\]  

(3.11)

(3.12)

where

- \(W_{\text{ideal}}\) = the ideal work of compression (done on the fluid) between specified suction and discharge states [J/kg]
- \(b\) = specific flow availability function [J/kg]
- \(h_2 - T_0 s_2 - (h_1 - T_0 s_1)\) = specific entropy [J/kg K]
- \(T_0\) = temperature of the surroundings [K]
- \(E_{ir}\) = indicated compressor rational efficiency

Also,

\[
E_{sr} = E_{ir} E_{mech}
\]  

(3.13)

where

- \(E_{sr}\) = shaft compressor rational efficiency

It is proposed that the rational efficiency be used to quantify the indicated power utilisation (equation 3.12) and the shaft power utilisation (equation 3.13) of both heat pump and refrigeration compressors. In determining the rational efficiency of an actual compressor the measured suction and discharge states and the temperature of the surroundings would be used in equation 3.11 to calculate the ideal work of compression and then equation 3.12 would be used to calculate \(E_{ir}\). On this
basis an ideal reversible compressor which involved two isentropic compression processes and an isothermal process at the temperature of the surroundings, as shown in Fig. 3-3, would have a rational efficiency of unity.

It should be noted that the rational efficiency, as defined by equations 3.11 and 3.12, depends on the temperature of the surroundings. It is important, therefore, that this temperature is stated whenever a rational efficiency is quoted.

In cases where useful heating (or, indeed, useful cooling) is provided directly by the compressor, e.g. by a cooling water jacket, this can be incorporated in the rational efficiency definition as follows:

\[
E_r = \frac{\dot{m} \cdot w_{\text{deal}} + \dot{A}}{P_i}
\]

(3.14)

where

\[\dot{A}\] = rate of useful availability transfer provided directly by the compressor [W]

Fig. 3-3 An ideal compression path between specified or experimentally measured suction and discharge states.

3.4.1 RATIONAL EFFICIENCY APPLIED TO REFRIGERATION COMPRESSORS

The rational efficiency defined in equation 3.13 quantifies the effectiveness of shaft power utilisation in transforming the thermodynamic state of the refrigerant from the suction to the discharge condition. The parameter does not incorporate any information on the appropriateness of either the suction or discharge state to the overall purpose of the plant, i.e. the
production of a cooling effect in the evaporator. The characteristics of a compressor in producing particular discharge states when operating with specified suction states and specified discharge pressures must be described separately.

In a refrigeration plant the discharge pressure must be higher than the saturation pressure corresponding to the temperature of the surroundings and depends on the interaction of the characteristics of the compressor and the condenser. For a given condenser a lower condensing temperature and pressure will result if a significant part of the heat rejection occurs in the compressor. For this reason, and because the work of compression is less when there is heat rejection, compressors which involve a high degree of heat rejection are to be preferred for refrigeration applications.

In an extreme case an ideal reversible compressor could involve isentropic compression to the temperature of the surroundings followed by isothermal compression and condensation at that same temperature. In fact, whether condensation takes place within the compressor, or within the condenser, or partly within each, does not affect the ideal compression work, once the discharge vapour is saturated and at the temperature of the surroundings. This is because the flow availability function has a single value for saturated refrigerant at the temperature of the surroundings, no matter what the dryness fraction.

If the compressor discharge vapour is at a temperature above that of the surroundings, then, unnecessary work has been expended in the compression process. The extent of the unnecessary work depends on the characteristics of the compressor and on the characteristics of the condenser. In a condenser of finite heat transfer area, the condensing pressure and the discharge pressure of the compressor will be higher than the saturation pressure corresponding to the temperature of the surroundings. However, even with an ideal condenser of infinite heat transfer area and zero pressure drop, the discharge temperature of the compressor at the saturation pressure corresponding to the temperature of the surroundings, $T_o$, might be higher than $T_o$. This would invariably be the case for current technology.
Ch. 3 Utilisation of Shaft Power

reciprocating refrigeration compressors.

On the basis of the above discussion a new type of rational efficiency, which depends on the characteristics of a refrigeration compressor combined with a condenser, can be defined.

\[ E_{ircr} = \frac{\dot{m}(b_{tosat} - b_i)}{P_i} \]  

(3.15)

where

- \( E_{ircr} \) = indicated rational efficiency for compression and heat rejection
- \( b_{tosat} \) = specific flow availability function of saturated refrigerant, evaluated at the temperature of the surroundings, \( T_o \) [J/kg]
  \[ = h_{f_0} - T_o s_{f_0} \]
  and also
  \[ = h_{g_0} - T_o s_{g_0} \]
- \( h_{f_0} \) = specific enthalpy of saturated liquid refrigerant at the temperature of the surroundings, \( T_o \) [J/kg]
- \( h_{g_0} \) = specific enthalpy of dry saturated vapour refrigerant at \( T_o \) [J/kg]
- \( s_{f_0} \) = specific entropy of saturated liquid at \( T_o \) [J/kg K]
- \( s_{g_0} \) = specific entropy of dry saturated vapour at \( T_o \) [J/kg K]

Given the compressor suction condition, the indicated power and the temperature of the surroundings, the rational efficiency for compression and heat rejection \( E_{ircr} \) can be evaluated for a specified suction pressure, suction temperature and discharge pressure. This parameter represents the performance of the compressor, combined with a condenser which operates at the specified pressure and de-superheats, condenses and subcools the refrigerant to the temperature of the surroundings.

Where the compressor is to be used to provide cooling only, the value of \( E_{ircr} \) can be quoted to quantify its performance in conjunction with a condenser which will cause it to operate at the specified discharge pressure. This parameter incorporates information on the suitability of the discharge state to the purpose of the plant.

\( E_{ircr} \) could also be evaluated experimentally for the special case where the discharge pressure is the saturation value corresponding to the temperature of the surroundings. This would represent the rational efficiency of the compressor combined with an ideal condenser - this figure could be of
interest where the highest possible rational efficiency for the entire plant was to be achieved by using a highly oversized condenser.

Situations sometimes arise where, in addition to the cooling effect, some useful heating is provided. In these cases the indicated rational efficiency, \( E_r \), and the discharge temperature should be quoted in order to describe the merit of the refrigeration compressor in utilising its shaft power input, when operating between a specified suction state and discharge pressure with a specified ambient temperature. An availability analysis of the entire plant would be necessary to evaluate the suitability of the compressor for its purpose.

### 3.4.2 RATIONAL EFFICIENCY APPLIED TO HEAT PUMP COMPRESSORS

The indicated rational efficiency defined in equation 3.14 is an appropriate measure of the effectiveness of shaft power utilisation for 'heat pump compressors'. It is appropriate also for compressors which provide simultaneous heating and cooling. A further point is that the expression for rational efficiency takes account of any useful direct heating (or cooling) provided from the compressor.

As in the case of refrigeration compressors, the parameter does not include information on the suitability of either the suction or discharge states of the refrigerant to the purposes of the plant.

Where the rational efficiency of a compressor is quoted the discharge condition must be specified by means of a second thermodynamic state property, normally temperature, as well as the discharge pressure.

### 3.5 SUCTION AND DISCHARGE PUMPING WORK

With reference to the indicator diagram shown in fig. 2-1, chapter 2, the crosshatched area below the horizontal line representing the mean suction pressure can be referred to as the 'indicated suction work', \( W_{si} \). In the same way the area of that part of the diagram which is above the mean discharge pressure can be described as the 'indicated discharge work',
Ch. 3 Utilisation of Shaft Power

$W_{di}$. These are the quantities of 'pumping work' which are necessary per cycle in order to induce refrigerant into the cylinder and to discharge it. The necessity for pumping work arises from the pressure losses across the suction and discharge valves.

The 'suction mean effective pressure' and the 'discharge mean effective pressure' are defined in equations 3.16 and 3.17. Both are analogous to the familiar 'mean effective pressure'.

\[
\text{suction mean effective pressure} = \frac{W_{si}}{N_{cy}} \quad (3.16)
\]

\[
\text{discharge mean effective pressure} = \frac{W_{di}}{N_{cy}} \quad (3.17)
\]

where

\[
W_{si} = \text{indicated suction work} \quad [J]
\]

\[
W_{di} = \text{indicated discharge work} \quad [J]
\]

By multiplying the indicated suction work or the indicated discharge work by the number of cycles per second, the suction pumping power or the discharge pumping power can be calculated.

\[
\text{suction pumping power} = W_{si}N_{cy} \quad (3.18)
\]

\[
\text{discharge pumping power} = W_{di}N_{cy} \quad (3.19)
\]

In the reciprocating compressor part of the pumping work quantities add to the availability of the vapour being compressed, but, not all, due to the high degree of irreversibility involved in the throttling processes at the valves. Availability is destroyed in any irreversible process.

By carrying out availability analyses of the suction and discharge processes the availability destruction corresponding to the pumping work quantities could be determined, on the basis of test measurements (including the thermodynamic state of the charge within the cylinder). This approach would be entirely consistent with the use of the rational efficiency as a compressor performance parameter.

3.6 CONCLUSIONS

Parameters have been described and discussed for use in quantifying the shaft or indicated power utilisation of refrigerant compressors. A strong case has been made for the use of a rational efficiency founded on the Second Law of Thermodynamics. It is not felt that the rational efficiency must necessarily
Ch. 3 Utilisation of Shaft Power

replace the more conventional isentropic efficiency. Rather, it should be used to gain a deeper understanding of the factors which tend to reduce power utilisation performance. It provides a basis for more detailed examination of these factors in which correct thermodynamic values can be apportioned to various losses.
Chapter 4

COMPRESSOR LOAD STANDS

A refrigerant compressor load stand is a system which is capable of subjecting a compressor to predetermined conditions such as it would encounter in normal operation. In some cases it may also be required to subject the compressor to off-design conditions. The power to drive the compressor, either electrical or mechanical, must be provided and this may also be regarded a function of the load stand. The four main operating conditions to be controlled in the case of a variable speed compressor are

A. the suction pressure
B. the discharge pressure
C. the suction vapor temperature
D. the shaft speed

A load stand normally takes the form of a closed loop so that the refrigerant can be circulated continuously through the compressor. Since the compressor transfers energy to the refrigerant, means must be provided for rejecting this energy from some part of the circuit.

Instrumentation must be provided on the load stand to confirm the operating conditions and for the measurement of compressor performance parameters, the main ones being the shaft power input and the refrigerant mass flow rate.

4.1 THE CALORIMETER LOAD STAND

The most obvious type of refrigerant compressor load stand would be a refrigeration system consisting of an evaporator, a condenser, and an expansion valve. Such a system, in which the heat transfer quantities of the cycle can be directly measured, is known as a calorimeter load stand. It is represented schematically in fig. 4-1 and the corresponding cycle is shown on a pressure-enthalpy diagram in fig. 4-2. This is the vapour compression refrigeration cycle. It is referred to as the reversed Rankine cycle since it resembles the Rankine cycle of a power plant, in reverse. (The expander of the power plant is replaced by the compressor and the feed pump is replaced by the expansion valve, which can be regarded as a crude substitute for a work producing expander.)
Calorimeter load stands are commonly used and have the particular advantage that the heating and cooling effects, of the cycle which includes the compression process, can be measured directly. The refrigerant mass flow rate can be calculated by means of a heat balance on the evaporator or condenser, or on both.

A common version of a calorimeter type load stand employs an electrically heated evaporator. The electric heating may be direct or indirect. In the direct case, for example, a heating tape could be wrapped around the tubes of a dry expansion evaporator. In the case of a flooded evaporator the heater could be immersed directly in the liquid refrigerant. More
commonly, a secondary heat transfer fluid, either a liquid or a
two phase refrigerant is used. In one such variation the
evaporator is enclosed in a pressure vessel containing a sec­
dary refrigerant in the liquid and vapour phases. The secon­
dary refrigerant liquid is heated by an electric immersion
heater, producing saturated vapour which transfers latent heat
to the primary evaporator. The evaporator cooling effect of
the refrigeration cycle is equal to the power input to the
heater and is readily measured by a wattmeter. Paul et al.,
for instance, used and described load stands of the latter type
[4].

The main disadvantage of this type of load stand is that
the rates of heat transfer in the evaporator and condenser are
high in comparison to the compressor power: perhaps three or
more times its value in the evaporator, and four or more times
in the condenser.

The problem of providing a high rate of heat transfer to
the evaporator and rejecting energy at a high rate in the
condenser can be lessened if the energy rejected from the
condenser is transferred to the evaporator. This can be done
by circulating a secondary fluid between the heat exchangers.
As the heat rejection rate in the condenser is greater than the
evaporator heat transfer rate, by an amount approximately equal
to the indicated power of the compressor, this excess must be
rejected externally, see fig. 4-3.

In a system of this type, means must be provided for the
control of the operating conditions, e.g. by varying the temper­
ature differences between the refrigerant and the secondary
fluid in the heat exchangers. Such control is complicated by
the fact that the same secondary fluid passes through both the
evaporator and condenser. Although the external rate of heat
transfer is reduced, the rates of heat transfer within the
system are high and require correspondingly large heat transfer
areas. It would appear that calorimeters of this type are not
in common use.
4.2 ZLATKOV’S LOAD STAND

A further progression in the development of the calorimeter test stand was described in a paper by Zlatkov [33 (1977)]. A three way heat exchanger was developed and patented which incorporated the evaporator and condenser in one unit and provided for external heat rejection to cooling water. In this system the entire refrigerant flow passed through the full reversed Rankine cycle. Zlatkov reported that, compared to an electrically heated calorimeter, the energy consumption was reduced by 100% i.e. no electric heating was required. He seems to have ignored the compressor power input! Water consumption was reduced by 80%. The dimensions, materials and labour requirements for construction were all substantially reduced. Also, the time required to reach a desired set of operating conditions was reduced by 30%, compared to electrically heated calorimetric load stands, due to reduced thermal inertia of the test stand. As pointed out by Zlatkov these advantages are particularly significant when the power rating of the compressor is in the medium to high range. The calorimeter he described was for testing compressors with a rating of 20 kW.

The controls on the calorimeter described by Zlatkov
Ch. 4 Compressor Load Stands

consisted of two water circuit flow adjustment valves, one to regulate the condensing temperature and one to regulate the amount of suction superheat, and an expansion valve in the refrigerant circuit.

4.3 GAS LOOP OR VAPOUR BYPASS LOAD STANDS

An alternative approach to the testing of refrigerant compressors is to assume that the thermodynamic properties of refrigerants are well understood and that heating and cooling capacity can therefore be calculated if the refrigerant mass flow rate and the temperature and pressure at suction and discharge are measured. There is no necessity for the working fluid actually to pass through the reversed Rankine cycle.

Means must be provided within the circuit to bring the refrigerant from a state at high temperature, pressure and enthalpy in the superheat region to one at a low pressure, with a low degree of superheat and lower enthalpy. The most convenient processes to achieve this are constant pressure cooling and adiabatic expansion. A work producing expansion device could be used in principle, but, a throttle valve has the advantages of being simple and readily controlled. Fig. 4-4 illustrates two possible arrangements.

![Diagram of gas loop load stands](image)

Fig. 4-4 A gas loop type load stand. In version (a) heat rejection occurs at the suction pressure, while in version (b) heat rejection occurs at the discharge pressure.

The gas loop load stand cycles corresponding to the diagrams in fig. 4-4 are shown in fig. 4-5. The refrigerant
remains in the superheated vapour state throughout cycle (a), but, passes through the saturated region in cycle (b). The flow rate could be measured by a gas flow meter or by a heat balance on the heat exchanger. It would also be possible to heat the refrigerant electrically at some point in the bypass loop and to calculate the flow rate from the electric power consumed and the refrigerant state properties determined from measurements before and after heating.

![Graph](a)

![Graph](b)

Fig. 4-5 Two possible cycles of a gas loop type compressor load stand involving constant pressure and throttling processes. These correspond with the plant diagrams in fig. 4-4.

Control of the operating conditions would be by

1. adjusting the rate of cooling, normally by varying the flow rate of coolant. (A balance must be reached between the heat exchanger and compressor characteristics so that the rate of heat rejection is equal to the indicated power. The operating point also fixes the pressure of the refrigerant in the heat exchanger.)

2. by adjusting the throttle valve (Together with the compressor characteristics, this determines the pressure difference across the compressor.) and

3. by varying the charge of refrigerant in the system.

The need for the last adjustment requires some elaboration. In a system of fixed volume any cycle during which the working fluid is a vapour, such as that shown in fig. 4-5 (a), will require a unique mass of fluid to be present, and the desired operating conditions can only be met when this quantity is provided. This is, perhaps, the main disadvantage of the gas
loop load stand. On the other hand, it excels in terms of compactness, low operating costs and low thermal inertia. A second disadvantage of the gas loop load stand is that the mass concentration of oil circulating with the refrigerant cannot be determined by the usual method of taking a liquid phase sample.

In a gas loop type load stand refrigerant is normally added or removed from the system through valves to a reservoir containing liquid and vapour refrigerant. This reservoir should be maintained at about ambient temperature - the saturation pressure will then normally be below the compressor discharge pressure and above its suction pressure. The arrangement is shown schematically in fig. 4-6.

![Diagram of a gas loop load stand with a reservoir which allows refrigerant vapour to be added to or removed from the circuit.](image)

**Fig. 4-6** Diagram of a gas loop load stand with a reservoir which allows refrigerant vapour to be added to or removed from the circuit.

### 4.4 THE MARRIOTT FAST RESPONSE LOAD STAND

Most load stands require a considerable time period before equilibrium at a desired operating condition can be established. This is typically from one to four hours and is due mainly to the need to establish full thermal equilibrium both within the compressor and within the load stand.

Marriott [34 (1973), 35 (1974)] described a load stand which had been built and evaluated and which was capable of measuring steady state performance parameters of a hermetic compressor within seventeen minutes of start-up.
Ch. 4 Compressor Load Stands

This was a gas loop load stand in which the greater part of the flow passed through a water cooled heat exchanger (fig 4-7). The cooled refrigerant was then mixed with hotter refrigerant which bypassed the heat exchanger through a throttle valve. The refrigerant remained superheated throughout the cycle. By oversizing the heat exchanger, by employing a very high water circulation rate and by controlling the temperature of the water to a predetermined value, the refrigerant exit temperature from the heat exchanger was maintained approximately constant. Thus the temperature after mixing, and thence the suction temperature, could be controlled by adjusting the bypass throttle valve. The pressure in the heat exchanger was allowed to float.
between the suction and discharge pressures, depending on the operating conditions and the system charge. According to Marriott, the heat exchanger acted as a vapour accumulator in the system. However, he did not state its volume or indicate whether it was necessary to add or remove refrigerant when the operating conditions of the system were changed significantly. It would seem that his system allowed changes in operating conditions, over a limited range, without alteration of the system charge. The processes of Marriott's load stand are shown in fig. 4-8.

Fig. 4-8 The thermodynamic processes of the Marriott load stand [34, p.51]. "The numbers on this diagram correspond to those of [fig. 4-7] and represent the approximate steady state conditions of the refrigerant circulating within the calorimeter system. The path from 1 to 2 depicts the compression process across the compressor. Path 2 to 3 represents the nearly isenthalpic expansion of the refrigerant across the discharge pressure control valve. Path 3 to 4 represents the portion of the refrigerant which flows through the exchanger, while path 3 to 5 depicts the expansion of the remaining refrigerant across the temperature control valve. The mixing process as the refrigerant streams recombine is depicted by paths 4 to 6 and 5 to 6. Isenthalpic expansion occurs from points 6 to 1 across the suction pressure control valve."
4.5 STANDARDS FOR THE PERFORMANCE TESTING OF REFRIGERANT COMPRESSORS

4.5.1 ASHRAE STANDARD

An ASHRAE standard "Methods of testing for rating positive displacement compressors" outlined standard methods for the determination of compressor capacity (flow rate), power input, performance factor (c.o.p.) and oil circulation [36 (1978)]. According to the standard a primary method for capacity measurement and a confirming method should be used and these should agree to within 3%. The standard listed the following as acceptable primary capacity measurement methods:

1. Secondary refrigerant calorimeter
2. Secondary fluid calorimeter
3. Primary refrigerant calorimeter
4. Gaseous refrigerant flow meter

Suggested confirming methods were:

5. Gaseous refrigerant flow meter
6. Condenser calorimeter
7. Liquid refrigerant flow meter

4.5.2 GERMAN STANDARD

A German standard entitled "Testing of refrigerant compressors" [37 (1973)] outlined ten methods of compressor performance testing and provided schematic diagrams for each. The methods listed were as follows:

A. Secondary fluid calorimeter: heat transfer to the evaporator via a secondary refrigerant.
B. Flooded evaporator calorimeter: direct heat transfer to the liquid refrigerant in the flooded evaporator.
C. Dry expansion calorimeter: direct heat transfer to refrigerant in the dry expansion type evaporator.
D. Refrigerant vapour flow meter calorimeter: measurement of refrigerant vapour flow rate.
E. Refrigerant liquid collection calorimeter: measurement of refrigerant liquid quantity by collection over a period of time.
F. Refrigerant liquid flow meter calorimeter: measurement of instantaneous liquid flow rate.
G. Water cooled condenser calorimeter: measurement of refrigerant mass flow rate by a heat balance on the condenser.

In methods A to G the entire refrigerant mass flow undergoes the reversed Rankine cycle. Full condensation takes place.
in a condenser and evaporation takes place in a separate evaporator.

**Fig. 4-9** Refrigerant vapour cooling calorimeter: version J, according to DIN 8977 [37] (alternative to version H).

**H.** Refrigerant vapour cooling calorimeter: part of the refrigerant flow is expanded directly to the low pressure side of the system while part goes through a condenser where it is condensed to a liquid and is then expanded to a wet mixture. The low pressure hot gas and the low pressure wet mixture pass through a heat exchanger in which the gas is cooled and liquid present in the mixture is evaporated.
Ch. 4 Compressor Load Stands

On leaving the heat exchanger the two gas streams are mixed to give the required suction condition.

J. Refrigerant vapour cooling calorimeter: an alternative to version H. In this version mixing of the two fluid streams takes place within the gas cooling heat exchanger (Fig. 4-9).

K. Calorimeter in the compressor discharge pipe: heat is transferred to the compressor discharge vapour and the mass flow rate is determined from a heat balance. This version can also be combined with version J.

4.6 SOME SPECIALISED MEASUREMENT TECHNIQUES

4.6.1 DYNAMIC PRESSURE MEASUREMENT

In order to measure a fluctuating pressure accurately it is necessary that the natural frequency of the pressure transducer is very much higher than the highest frequency component of interest within the pressure waveform. Fortunately, transducers having very high natural frequencies are available e.g. quartz piezoelectric types having natural frequencies greater than 150 kHz.

In measuring the pulsating pressure within the cylinder or plenums of a compressor, it may not be possible to flush mount the face of the pressure transducer in the region where the pressure is to be measured. As a compromise, a pressure tapping at the measurement point may be linked to the transducer by a pressure inducting system. The dynamic response of the measurement system is often limited by the pressure inducting system, rather than by the transducer. It is thus necessary to be able to calculate the natural frequency of the inducting system, and its damping coefficient. The design objective is to maximise the natural frequency and minimise the damping coefficient so as to ensure the best possible amplitude and phase response.

The most common type of pressure inducting system consists of a connecting passage and a volume at the face of the transducer, as shown in fig. 4-10.

The following parameters are defined, based on fig. 4-10:

\[
\begin{align*}
\text{volume of passage, } V_p &= \pi r_p^2 L_p \text{ [m}^3\text{]} \\
\text{volume of space, } V_s &= \pi r_s^2 L_s \text{ [m}^3\text{]} \\
\text{volume ratio, } R &= \frac{V_s}{V_p} \\
&= \frac{(r_s^2 L_s)}{(r_p^2 L_p)}
\end{align*}
\]
Various models have been described in the literature for this type of pressure inducting system and variations on it. On the basis of these models analytical expressions for the natural frequency and the damping coefficient have been proposed.

If the volume ratio is zero, i.e. a passage of constant area connects the pressure source to the transducer, without a volume $V_s$ at the end, the "organ pipe" natural frequencies apply [38]:

$$ f_o = \frac{(2n + 1) c}{4 L_p} \quad (n = 0, 1, 2...) \quad (4.1) $$

where

- $f_o$ = natural frequency of the pressure inducting system [Hz]
- $c$ = velocity of sound within the fluid [m/s]

Elson and Soedel [38] pointed out that equation 4.1 should not be used except for very small values of the volume ratio $R$ (e.g. for $R > 0.1$, error > 10%).

The pressure inducting system can be idealised as a "Helmholtz resonator" if the length of the connecting channel $L_p$ is less than the quarter wavelength of the fluctuating pressure and the volume ratio $R$ is large [38, 39].

$$ f_o = \frac{c}{2\pi L_p^{0.5}} \left( \frac{1}{R^2 + 0.5} \right)^{0.5} \quad (4.2) $$
where

\[ L_p^* = L_p + \frac{\pi r_p}{4} \]  
\[ = L_p + 0.785 r_p \]  
\[ R^* = \frac{(r_s^2 L_s)}{(r_p^2 L_p^*)} \]  

It can be noted that an effective length of the connecting passage \( L_p^* \), to take account of gas inertia at entry and exit, is used in equation 4.2.

Doeblin [40] gave an expression similar to equation 4.2 for the natural frequency, using \( R \) in place of \( R^* \) and without the term \( 0.5 \), but, defined the effective length as

\[ L_p^* = L_p + \frac{16 r_p}{3\pi} \]
\[ = L_p + 1.70 r_p \]

Doeblin's expressions apply where the volume ratio is large and the gas in the connecting passage moves as a unit, as opposed to exhibiting wave motion within the tube, i.e. the same assumptions as already stated for equation 4.2.

Elson and Soedel [38] proposed a formula which they claimed was generally more accurate than equation 4.2 and could be applied over the volume ratio range \( 0 \leq R^* \leq 100 \), equation 4.6.

\[ f_0 = \frac{1}{2\pi L_p^* (R^* + 0.3905)^{0.5}} \]  

An expression for the damping coefficient \( f \) was given by Hanjalic and Stosic [39 (1978)] as

\[ f = \frac{4\pi \mu L_p}{f C V_p} (R + 1/c)^{0.5} \]  

where

\( f \) = damping coefficient

\( \mu \) = absolute viscosity [Ns/m²]

\( f \) = fluid density [kg/m³]

The above expression, as given in [39], is dimensionally incorrect due, most probably, to printing errors. A derivation is given in appendix D, based on a lumped parameter Helmholtz resonator model, assuming laminar flow in the connecting passage, and it is concluded that the correct form of the equation is

\[ f = \frac{4\pi \mu L_p^*}{f C V_p^*} (R^* + 0.3905)^{0.5} \]  

(Doeblin [41] also gave an equation similar to 4.7.)

Hanjalic and Stosic gave a resonant frequency equation
Ch. 4 Compressor Load Stands

equivalent to 4.2. Their equations, 4.2 and 4.7, were based on a lumped parameter model with the assumption that $V_p$ and $V_s$ were of the same order of magnitude. The latter assumption is not made in arriving at equation 4.7 in appendix D. This assumption would, perhaps, allow the replacement of $V_p^*$ by $V_s$ in equation 4.7.

Xiangxing [42 (1986)] derived a slightly different form of the resonant frequency equation to equations 4.2 and 4.6, and also derived an equation for the damping coefficient as follows:

$$f_o = \frac{c}{2.31 \pi L_p} \frac{1}{\sqrt{R}} \quad (4.8)$$

$$f = 2 \sqrt{(3/\pi)} \frac{\mu \sqrt{(L_p V_s)}}{c \sqrt[3]{r_p^3}} \quad (4.9)$$

These equations were based on the assumption that $(L_p/r_p) > 8$. Expressions for the amplitude and phase angle response of a pressure inducting system (i.e. a single degree of freedom system with linear damping) were also presented and graphed. These could be used to determine the response of any pressure inducting system for which the natural frequency and damping coefficient could be determined.

An important aspect of the paper by Xiangxing was its demonstration that an accurate p-V diagram could be produced from data obtained with an otherwise poor pressure inducting system, of known transfer function. This was done by digitally reprocessing the data to correct the amplitude and phase errors.

Buchholz [43 (1978)] presented design guidelines for pressure inducting systems, based on analytical expressions from the literature and on a special test programme carried out at his company. In summary:

1. The connecting passage length $L_p$ should be as short as possible.
2. The connecting passage radius $r_p$ should be as large as possible.
3. The volume $V_s$ should be as small as possible.
4. Orifices or restrictions within the pressure inducting system should not be used as they were found to distort waveforms and cause phase shifts.

4.6.2 MEASUREMENT OF PULSATING FLOW WITH AN ORIFICE PLATE

Orifice plates have not normally been regarded as suitable
for the direct measurement of pulsating flow rates such as occur in the suction or discharge pipes of a reciprocating compressor. The traditional approach was to dampen the pulsations by means of a large volume between the source of the pulsations and the metering orifice. A study of the literature showed that, as a result of developments in differential pressure measuring devices, this was no longer the only approach possible.

4.6.2.1 Dynamic Differential Pressure Measurement

Lee and Smith [44 (1980)] designed their differential pressure measuring system to have natural frequencies well above the component frequencies of the pulsating flow associated with a compressor and treated the flow through the orifice as quasi-steady. The flow rate waveform was determined from the differential pressure waveform by using the instantaneous differential pressures in the steady state flow equation. As some resonance did occur it was found necessary to smoothen the differential pressure waveform numerically before transforming it to a mass flow rate waveform. Lee and Smith calculated average mass flow rates from results obtained in this way and obtained values which were 101%, at suction, and 102%, at discharge, of the calorimetrically determined mass flow rates.

4.6.2.2 Measurement of Mean Differential Pressure with Flow Damping

The more traditional approach, which had been used since the early nineteen hundreds, where only the average mass flow rate was required, was to install a pulsation damping chamber between the source of pulsations and the orifice plate. Hodgson described a method of calculating the size of chamber required, depending on the form of the flow pulsations. Oppenheim and Chilton, in a literature survey [45 (1955)], described Hodgson's results and related results from other workers.

A criterion for sizing the pulsation damping chamber for a specified maximum error was included in British Standard 1042 [46 (1984)]. This was as follows:
Damping is adequate if

\[
\frac{H_0}{k} > 0.563 \frac{1}{\sqrt{s}} \left( \frac{(q_v)_{\text{rms}}}{q_v} \right) \tag{4.10}
\]

where

\[
\begin{align*}
H_0 &= \text{Hodgson number} \\
&= \frac{V \Delta p^s}{q_v f \bar{p}} \tag{4.11} \\
k &= \text{isentropic exponent} \\
s &= \text{maximum allowable percentage error in indicated flow rate due to pulsations} \\
(q_v)_{\text{rms}} &= \text{root mean square volume flow rate [m}^3/\text{s}] \\
\bar{q}_v &= \text{time mean volume flow rate [m}^3/\text{s}] \\
V &= \text{volume of chamber [m}^3] \\
f &= \text{pulsation cycle frequency [Hz]} \\
\Delta p^s &= \text{time mean pressure loss between the chamber and the source/sink of constant pressure [N/m}^2] \\
\bar{p} &= \text{mean pressure within the receiver [N/m}^2]
\end{align*}
\]

Mainardi et al. [47 (1977)] investigated the variations in the coefficient of discharge and the coefficient of pressure drop for turbulent pulsating flow through an orifice plate. In their experimental work the flow rate waveform was approximately sinusoidal. It was found that the coefficient of pressure drop continued to have the same value as in steady flow, but, the coefficient of discharge was lower than the value measured in steady flow. The discharge coefficient results are summarised as follows in table 4-1.

<table>
<thead>
<tr>
<th>relative amplitude (\lambda) of flow fluctuation</th>
<th>(effective discharge coeff.)/(steady state coeff.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 %</td>
<td>100.0 %</td>
</tr>
<tr>
<td>6 %</td>
<td>101.5 %</td>
</tr>
<tr>
<td>15 %</td>
<td>104.0 %</td>
</tr>
</tbody>
</table>

Table 4-1 Results of Mainardi et al [47].

where

\[
\lambda = \frac{\text{maximum flow} - \text{minimum flow}}{\text{maximum flow} + \text{minimum flow}} \tag{4.12}
\]
4.7 CONCLUSIONS

In this chapter compressor load stands have been described, based on information available in the literature. Many other variations are possible and it is reasonable to design a load stand to suit the needs or constraints of a particular application. Information from the literature relating to specialised techniques for the measurement of dynamic pressures and pulsating flow rates has also been included.
Chapter 5

DESIGN RATIONALE OF THE LOAD STAND USED

It was decided to build a load stand similar in its principle of operation to version J in the DIN standard [37] (fig. 4-9, ch. 4). In this type of load stand part of the refrigerant flow is fully condensed and then mixed with the remainder which undergoes throttling only.

The intention was to provide control of the suction and discharge pressures and the suction superheat by means of throttle valves in the refrigerant circuit. A bypass type load stand was selected in order to minimise the requirements for cooling water and electric power. A further consideration in deciding upon a type which involved condensation of part of the refrigerant flow was that it should be possible to take a liquid refrigerant sample in order to determine the concentration of oil in the compressor discharge vapour. The circuit arrangement which evolved in the course of the work differed from version J in the DIN standard in the locations of some of the throttle valves and in that it required only one heat exchanger instead of two.

5.1 THERMODYNAMIC PROCESSES

A circuit diagram to implement the basic processes of the load stand is shown in fig. 5-1. It is assumed that condensation to the saturated liquid state occurs in the heat exchanger and that the saturation pressure can be held constant. The suction and

![Diagram](image-url)

Fig. 5-1 Load stand basic circuit.
Ch. 5 Design Rationale of the Load Stand Used

discharge pressures of the compressor are established with respect to the heat exchanger pressure by the throttle valves D and A. By some, as yet unspecified means, part of the total refrigerant flow leaving valve A is caused to pass through the heat exchanger, while the balance is bypassed to valve D. By varying the proportion of refrigerant which is bypassed the suction superheat can be controlled. The idealised thermodynamic processes are shown on a p-h diagram in fig. 5-2.

![Fig. 5-2 Idealised thermodynamic processes of the load stand.](image)

By applying the steady flow energy equation the flow rate of refrigerant through the heat exchanger can be expressed as a proportion of the total as follows:

\[
\frac{\dot{m}_{hx}}{\dot{m}_{tot}} = 1 - \frac{h_1 - h_4}{h_2 - h_4}
\]

(5.1)

where

- \(\dot{m}_{hx}\) = mass flow rate of refrigerant through the heat exchanger [kg/s]
- \(\dot{m}_{tot}\) = total mass flow rate produced by the compressor [kg/s]
- \(h\) = specific enthalpy [J/kg]

Subscript numbers refer to refrigerant state points identified in figures 5-1, 5-2 and subsequent figures.

Table 5-1 is a set of results of this calculation over a wide range of operating conditions for R-12 with a saturation temperature in the heat exchanger of 20°C. Similar tables were produced for other values of the heat exchanger operating temperature. For example, at a saturation temperature of 30°C the proportion of the total flow rate which passes through the
### HEAT EXCHANGER MASS FLOW RATE AS A PROPORTION OF THE TOTAL COMRESSOR MASS FLOW RATE

**REFRIGERANT NO.: 12**  
**HEAT EXCHANGER OPERATING TEMP., DEG. C: 20.0**

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<tr>
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</table>

Table 5-1
Ch. 5 Design Rationale of the Load Stand Used

heat exchanger is higher than in table 5-1 by from 5% to 7%, depending on the pressure and superheat conditions.

The main conclusion from table 5-1 is that it is possible, in principle, to produce any operating condition of interest by varying the relative amounts of refrigerant which are condensed and bypassed.

There would be practical difficulties in implementing the cycle as described thus far. There would be a pressure fall, however small, in any real heat exchanger and so a corresponding pressure fall would be required in the bypass line. A means of proportioning the flow would also be necessary. Both difficulties can be overcome by introducing throttle valves B and C as shown in fig. 5-3.

![Diagram](image)

Fig. 5-3 Load stand circuit including bypass and liquid line throttle valves.

The effect these valves have on the thermodynamic cycle is shown on a p-h diagram in fig. 5-4. The inclusion of these extra throttle valves does not alter the values presented in table 5-1 for the proportion of the refrigerant flow which passes through the heat exchanger. The pressure at which adiabatic mixing of the condensed liquid and bypassed vapour occurs can lie at any level between that of the heat exchanger and that at suction, depending on the settings of valves B, C and D.

5.2 THE IDEAL LOAD STAND AND COMPRESSOR

For the purpose of designing or sizing a load stand, an ideal
compressor which can be described in terms of its displacement rate and clearance ratio is considered a suitable model. The volumetric efficiency of such a compressor is given by equation 2.5, ch. 2.

By using this equation in conjunction with the displacement rate and the thermodynamic relationships for the cycle, the ideal mass flow rate, the proportions of refrigerant condensed and bypassed and the rate of heat rejection in the heat exchanger can be calculated for any combination of operating conditions. The results of such calculations, assuming a displacement rate of 1 litre/sec., a clearance ratio of 3% and a heat exchanger saturation temperature of 20°C are given in tables 5-2 and 5-3. The values of heat rejection rate given in table 5-3 apply regardless of the heat exchanger operating temperature. These figures can be used generally in selecting suitable heat exchangers for load stands of this type.

5.3 EFFECTIVE VALVE FLOW AREAS
The value chosen for the mixing pressure is a matter of control strategy and, when this has been decided upon, the required flow areas for the four throttle valves can be calculated. The valves are modelled as orifices having a coefficient of discharge of 1 and can thus be fully described by their effective flow areas.

For valve C, which the refrigerant enters as a liquid,
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Table 5-2
### HEAT EXCHANGER HEAT TRANSFER RATE IN KW FOR AN IDEAL
### COMPRESSOR OF GIVEN DISPLACEMENT RATE AND CLEARANCE RATIO

**REFRIGERANT NO.: 12**

**COMPRESSOR DISPLACEMENT RATE, LITRES PER SECOND: 1.00**

**CLEARANCE RATIO: 0.030**

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</tr>
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</table>

**Table 5-3**
incompressible flow is assumed and the effective flow area is given by:

\[ A = \frac{\dot{m}}{\sqrt{(\nu_u/2Ap)}} \]  

(5.2)

where:

- \( \nu_u \) = specific volume of refrigerant upstream of valve \([\text{m}^3/\text{kg}]\)
- \( Ap \) = pressure difference across valve \([\text{N/m}^2]\)

For valves A, B and D the flow is compressible and, depending on the pressure ratio in each case, may be choked. In the calculations either the actual or the critical pressure ratio, whichever is the greater, is used. For the purposes of calculating the required orifice area isentropic flow is assumed, although, for the throttling process as a whole, enthalpy is the property which remains unchanged. The isentropic process is modelled as being polytropic, i.e. one for which

\[ pv^n = \text{constant} \]  

(5.3)

where:

- \( p \) = refrigerant pressure \([\text{N/m}^2]\)
- \( v \) = refrigerant specific volume \([\text{m}^3/\text{kg}]\)
- \( n \) = polytropic index

A value of 1.18 is used for the polytropic index.

While the assumption of a polytropic process is considered reasonably valid for the refrigerant vapour, even within the saturation region, the ideal gas equation cannot be assumed to apply, especially if the refrigerant is saturated for part of the process. It is concluded therefore that the equation for the required flow area should be in terms of the properties pressure and volume, rather than, for example, pressure and temperature. The following equation is used:

\[ A = \frac{\dot{m}}{\sqrt{\left(\frac{p_u}{\nu_u} \cdot \frac{2n}{n-1} \left(\frac{r_p^{2/n}}{n-1} - \frac{r_p^{(n+1)/n}}{n+1}\right)\right)^{1/5}}} \]  

(5.4)

where:

- \( p_u \) = refrigerant pressure upstream of valve \([\text{N/m}^2]\)
- \( r_p \) = actual or critical pressure ratio, whichever is the greater

\[ = \frac{p_u}{p_u} \quad \text{or} \quad = \left[\frac{2}{n+1}\right]^{n/(n-1)} \]
Ch. 5 Design Rationale of the Load Stand Used

\[ p_d = \text{downstream pressure, [N/m}^2 \text{]} \]

It should be noted that for some suction conditions point 7 in fig. 5-4 lies within the saturation region on the p-h diagram. The specific volume at this point can be found since the specific enthalpy is the same as at point 1.

Values of the required effective flow areas for the valves are presented over a range of operating conditions in table 5-4. It is assumed that one fifth of the pressure fall from the heat exchanger saturation pressure to the suction pressure occurs across the bypass and liquid line throttle valves B and C. It can be noted that any change in the operating conditions requires a change in the areas of all four valves. The required flow areas for valve C, the liquid line throttle valve, are very much smaller than for the other valves which throttle vapour.

5.4 THROTTLE VALVES: SELECTION AND CHARACTERISTICS

The data presented in table 5-4 were used as the basis for the selection of the valves for the load stand.

A type of regulating valve having an orifice of 9.5 mm diameter was selected for valves A, B and D. This had a maximum effective flow area of 30.56 mm² at about 4.5 turns open. Its flow characteristics were represented graphically by the manufacturer. A polynomial was fitted to the data for analysis purposes. The equation and the constants which were used are included in appendix E.

In order to control the load stand over a wide range of conditions and to achieve the necessary metering accuracy, two valves were installed in parallel to fulfil the function of valve C for liquid refrigerant metering. These are referred to as \( C_c \) and \( C_f \). The coarser of the two, \( C_c \), had an orifice of 3.18 mm diameter with a maximum effective flow area of 2.547 mm² at 11 turns open. The finer metering valve, \( C_f \), had an orifice diameter of 0.79 mm and a maximum effective flow area of 0.067 mm² at 10 turns open. Curves were fitted to the graphical data presented by the manufacturers for the effective flow areas of the valves. The resulting equations are given in appendix E. Both liquid line metering valves were fitted with micrometer...
handles which facilitated accurate setting to less than one twenty-fifth of a revolution.

5.5 HEAT EXCHANGER SELECTION, CHARACTERISTICS AND ANALYSIS

The displacement rate of the compressor to be tested was about 3.7 litres/second and, using data from table 5-3, the required

<table>
<thead>
<tr>
<th>EFFECTIVE FLOW AREAS OF VALVES ON THE TEST STAND FOR AN IDEAL COMPRESSOR OF GIVEN DISPLACEMENT RATE AND CLEARANCE RATIO</th>
</tr>
</thead>
<tbody>
<tr>
<td>SYMBOLS: C - CHOKED FLOW, I - IMPOSSIBLE OPERATING CONDITION,</td>
</tr>
<tr>
<td>0 - VALVE FULLY OPEN, Z - ZERO COMPRESSOR FLOW</td>
</tr>
<tr>
<td>It is assumed that the adiabatic mixing pressure ((P7=P6=P5)) lies between the heat exchanger pressure ((P3=P4)) and the suction pressure ((P1)) and that (P7 = P3 - (P3-P1)*0.20)</td>
</tr>
<tr>
<td>REFRIGERANT NO.: 12 ADIABATIC INDEX (BASED ON P AND V): 1.180</td>
</tr>
<tr>
<td>HEAT EXCHANGER OPERATING TEMP., DEG. C: 20.0</td>
</tr>
<tr>
<td>COMPRESSOR DISPLACEMENT RATE, LITRES PER SECOND: 1.00</td>
</tr>
<tr>
<td>CLEARANCE RATIO: 0.030</td>
</tr>
</tbody>
</table>

### VALVE A ###

<table>
<thead>
<tr>
<th>SUCTION SAT. TEMP., DEG. C</th>
<th>DISCHARGE SAT. TEMP., DEG. C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20.0</td>
</tr>
<tr>
<td>-30.0</td>
<td>99.999</td>
</tr>
<tr>
<td>-10.0</td>
<td>99.999</td>
</tr>
<tr>
<td>10.0</td>
<td>99.999</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>SUCTION SAT. TEMP., DEG. C</th>
<th>DISCHARGE SAT. TEMP., DEG. C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20.0</td>
</tr>
<tr>
<td>-30.0</td>
<td>99.999</td>
</tr>
<tr>
<td>-10.0</td>
<td>99.999</td>
</tr>
<tr>
<td>10.0</td>
<td>99.999</td>
</tr>
</tbody>
</table>

### VALVE B ###

<table>
<thead>
<tr>
<th>SUCTION SAT. TEMP., DEG. C</th>
<th>DISCHARGE SAT. TEMP., DEG. C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20.0</td>
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<tr>
<td>-30.0</td>
<td>2.082</td>
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<tr>
<td>-10.0</td>
<td>5.741</td>
</tr>
<tr>
<td>10.0</td>
<td>17.705</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>SUCTION SAT. TEMP., DEG. C</th>
<th>DISCHARGE SAT. TEMP., DEG. C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20.0</td>
</tr>
<tr>
<td>-30.0</td>
<td>1.958</td>
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<td>-10.0</td>
<td>5.391</td>
</tr>
<tr>
<td>10.0</td>
<td>16.539</td>
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</table>

Table 5-4 (continued on next page)
### VALVE C

<table>
<thead>
<tr>
<th>SUCTION SUPERHEAT, K: 0.0</th>
<th>SUCTION SAT. TEMP. DEG. C</th>
<th>DISCHARGE SAT. TEMP., DEG. C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20.0</td>
<td>30.0</td>
</tr>
<tr>
<td>-30.0</td>
<td>0.071</td>
<td>0.076</td>
</tr>
<tr>
<td>-10.0</td>
<td>0.105</td>
<td>0.129</td>
</tr>
<tr>
<td>10.0</td>
<td>0.100</td>
<td>0.185</td>
</tr>
</tbody>
</table>

### VALVE D

<table>
<thead>
<tr>
<th>SUCTION SUPERHEAT, K: 30.0</th>
<th>SUCTION SAT. TEMP. DEG. C</th>
<th>DISCHARGE SAT. TEMP., DEG. C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20.0</td>
<td>30.0</td>
</tr>
<tr>
<td>-30.0</td>
<td>0.062</td>
<td>0.067</td>
</tr>
<tr>
<td>-10.0</td>
<td>0.092</td>
<td>0.113</td>
</tr>
<tr>
<td>10.0</td>
<td>0.087</td>
<td>0.162</td>
</tr>
</tbody>
</table>

### VALVE E

<table>
<thead>
<tr>
<th>SUCTION SUPERHEAT, K: 0.0</th>
<th>SUCTION SAT. TEMP. DEG. C</th>
<th>DISCHARGE SAT. TEMP., DEG. C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20.0</td>
<td>30.0</td>
</tr>
<tr>
<td>-30.0</td>
<td>2.234 C</td>
<td>2.125 C</td>
</tr>
<tr>
<td>-10.0</td>
<td>4.945 C</td>
<td>4.834 C</td>
</tr>
<tr>
<td>10.0</td>
<td>10.467</td>
<td>10.342</td>
</tr>
</tbody>
</table>

### Table 5-4 (continued)
Refer to fig. 5-3 for the circuit and fig. 5-4 for the thermodynamic processes.

Heat rejection rate was found to be about 2.2 kW, which also coincided with the motor rating.

A water cooled condenser heat exchanger with some capacity to hold excess liquid was required and a shell and tube type was selected.

#### 5.5.1 Heat Transfer Rate

The manufacturers presented the heat transfer rate as a function of the water flow rate and the temperature difference defined in terms of the saturation temperature and the water inlet temperature.
\[ \Delta T = T_{hx} - T_{wi} \]  

(5.5)

where

\[ \Delta T = \text{temperature difference [K]} \]
\[ T_{hx} = \text{refrigerant saturation temperature within the heat exchanger [K]} \]
\[ T_{wi} = \text{water inlet temperature [K]} \]

Heat transfer correlations suggest that the water side heat transfer coefficient depends on the flow rate, to the power of 0.8 approximately. Hence, the following expression was derived for the rate of heat transfer

\[ \frac{\dot{Q}}{1,000} = \left( \frac{a (1,000 \dot{V})^{0.8}}{1 + b (1,000 \dot{V})^{0.8}} \right) \Delta T \]  

(5.6)

where

\[ \dot{Q} = \text{rate of heat transfer [W]} \]
\[ \dot{V} = \text{volume flow rate of water [m}^3/\text{s]} \]

By regression analysis of points taken from the manufacturers' graphs the following coefficients were obtained:

\[ a = 2.009 \]
\[ b = 1.933 \]

A graph of this expression is given in fig. 5-5.

![Graph](attachment:Fig. 5-5 Heat transfer rate per degree temperature difference as a function of water flow rate.)

5.5.2 HEAT EXCHANGER WATER SIDE PRESSURE DROP

It was found that the manufacturers' data on the water side pressure drop could be represented reasonably closely by the following expression:

\[ \Delta p = k \dot{V} \]  

(5.7)
Ch. 5 Design Rationale of the Load Stand Used

where

\[
\Delta p = \text{pressure drop [N/m}^2]\]
\[k = 1.151 \times 10^5\]
\[j = 1.833\]

The maximum available water supply rate was about 4 litres per minute. The head tank was installed at a height of about 5.2 metres above the drain to provide for the pressure losses through the heat exchanger and associated piping, while maintaining a constant flow rate.

5.6 COMPUTER MODELLING OF THE COMPLETE LOAD STAND

A number of computer programs were written to assist in the analysis of the load stand. Some of these produced the tables already included in this chapter. The programs called subroutines to find the refrigerant properties at the various thermodynamic states. For various operating conditions, and assuming an ideal compressor of known clearance volume, the flow rates within the load stand were calculated. The flow equations for valves A, B, C, C, and D were solved to yield the required effective flow areas and hence the 'turns open' of each. The program logic took full account of compressible flow conditions and where choking of any of the valves occurred this was indicated in the output by a flag (see table 5-4). Flags were also used to indicate impossible operating conditions and situations where valves were fully open.

In the most elaborate analysis program, suction line flow restrictions, i.e. a flow straightener, a measurement orifice and a baffle plate, were included. Flags indicated choked flow at these locations also if it occurred. A venturi which was incorporated in the load stand at the point of mixing of the liquid and vapour streams was also included in the calculation process. The heat transfer characteristics of the heat exchanger were taken into account. The program calculated results for specified ranges of the four operating conditions of the compressor (speed, suction saturation temperature, discharge saturation temperature and suction superheat) and an additional operating condition of the load stand (the cooling water flow rate).
Two main control strategies were examined:

1. Valve B in the vapour bypass line was left at a fixed setting and the settings of Cc and Cf were computed, together with the settings of A and D.

2. The drop in pressure from the mixing point to the suction line was expressed as a fixed ratio of the pressure drop from the heat exchanger to the suction line. The settings of all five throttle valves were then calculated.

The results were printed out in tabular form, with flags indicating choking, and were also stored for plotting later (see figs. 5-6 to 5-10).

5.6.1 SENSITIVITY OF THE REQUIRED VALVE OPENINGS TO THE OPERATING CONDITIONS

The first control strategy mentioned above was preferred as it required adjustment of four throttle valves rather than five and did not require monitoring of the mixing pressure. In figures 5-6 to 5-10, graphs are presented, based on the output data of the modelling program, of the required valve openings over ranges of the five load stand operating conditions. For each figure only one operating condition is varied.

For the control of the suction and discharge saturation temperatures, valves A, B and C all require adjustment. Control of the suction superheat requires very fine metering of the liquid by means of valve Cf and slight adjustment of the other valves, mainly of D, the suction line throttling valve.

Changes in operating speed require changes mainly in valves A and D, but some fine adjustment of valve Cf is also required. Valves A, Cc and D show some sensitivity also to the flow rate of cooling water through the heat exchanger.

It was concluded from these graphs that, based on a steady state analysis, the operating conditions could be controlled by adjustment of the throttle valves.

5.6.2 COMPARISON OF THE LOAD STAND MODEL WITH TEST OBSERVATIONS

The predictions of the load stand computer model are compared with test observations in table 5-5. In making the comparison the refrigerant flow rate within the model (which
Ch. 5 Design Rationale of the Load Stand Used

COMPRESSOR LOAD STAND ANALYSIS

Discharge sat. temp.: 35 deg. C
Suction superheat: 10 K
Compressor speed: 600 r.p.m.
Cooling water flow rate: 3 litres/min.
Cooling water temp.: 15 deg. C
Valve B fully open
Computer run ref.: TSTND8 30-May-87-12:01

Fig. 5-6 Sensitivity of load stand control valves to suction saturation temperature.
Ch. 5 Design Rationale of the Load Stand Used

COMPRESSOR LOAD STAND ANALYSIS

Suction sat. temp.: -10 deg. C
Suction superheat: 10 K
Compressor speed: 600 r.p.m.
Cooling water flow rate: 3 litres/min.
Cooling water temp.: 15 deg. C
Valve B fully open
Computer run ref.: TSTND8 30-May-87-12:01

![Graphs of valves A, C (coarse), C (fine), and D showing sensitivity to discharge saturation temperature.]

Fig. 5-7 Sensitivity of load stand control valves to discharge saturation temperature.

90
Ch. 5 Design Rationale of the Load Stand Used

COMPRESSOR LOAD STAND ANALYSIS
Suction saturation temp.: -10 deg. C
Discharge sat. temp.: 35 deg. C
Compressor speed: 600 r.p.m.
Cooling water flow rate: 3 litres/min.
Cooling water temp.: 15 deg. C
Valve B fully open
Computer run ref.: TSTND8 30–May–87–12:01

Fig. 5-8 Sensitivity of load stand control valves to suction superheat.
Ch. 5 Design Rationale of the Load Stand Used

COMPRESSOR LOAD STAND ANALYSIS

Suction saturation temp.: -10 deg. C
Discharge sat. temp.: 35 deg. C
Suction superheat: 10 K
Cooling water flow rate: 3 litres/min.
Cooling water temp.: 15 deg. C
Valve B fully open
Computer run ref: TSTND8 30-May-87-12:01

Fig. 5-9 Sensitivity of load stand control valves to compressor speed.
Ch. 5 Design Rationale of the Load Stand Used

COMPRESSOR LOAD STAND ANALYSIS

Suction saturation temp.: -10 deg. C
Discharge sat. temp.: 35 deg. C
Suction superheat: 10 K
Compressor speed: 600 r.p.m.
Cooling water temp.: 15 deg. C
Valve B fully open
Computer run ref.: TSTND8 16-Jun-87-17:03

VALVE A

VALVE C: COARSE

VALVE C: FINE

FULLY CLOSED

VALVE D

Cooling water flow rate, litres/min.

Fig. 5-10 Sensitivity of load stand control valves to the cooling water flow rate of the heat exchanger.
normally assumed an ideal compressor) was corrected to allow for the actual volumetric efficiency of the compressor. Also, the actual turns open of valves Cc and Cr were expressed in terms of the equivalent turns open of valve Cc alone.

There is not close agreement between the predictions and observations and a number of factors account for this, but, the level of agreement does, nonetheless, justify the model as a design tool. The factors which account for the differences are:

1. The equations which were used to model the valves were based on flow characteristics supplied in the product catalogue by the manufacturers. These did not represent precise calibration data for the specific valves used.

2. The actual numbers of turns open of valves A, B and D were determined rather crudely, to an accuracy of perhaps plus or minus 10°. In addition, there was considerable uncertainty as to the exact 'fully closed' position of each of the valves, including Cc and Cr.

3. Pressure losses within the pipe runs were neglected within the model and perfect insulation was assumed. In practice these effects would have had some effect on the required valve openings.

4. The model assumed a homogeneous equilibrium state after mixing of the throttled liquid and bypass vapour streams. This may not have been the case in practice, particularly when low suction superheat values were being maintained.

It was found that the sensitivity of the required valve settings to the operating conditions did correspond qualitatively in practice with the types of characteristics shown in figures 5-6 to 5-10. It was also found that the operating conditions could, in general, be controlled satisfactorily by means of the throttle valves which had been selected on the basis of the computer model. An aspect which the model did not predict, however, was that it proved virtually impossible to maintain stable operating conditions when the suction superheat was less than about 10 K. This problem was probably due to a combination of effects: heat capacity of the pipework and of the suction accumulator, separation of liquid droplets from the wet mixture and heat transfer to the suction line. There may also
Ch. 5 Design Rationale of the Load Stand Used

have been an element of instability due to oil solubility effects.

<table>
<thead>
<tr>
<th>Operating Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>test no.</td>
</tr>
<tr>
<td>suction saturation temp. [°C]</td>
</tr>
<tr>
<td>discharge saturation temp. [°C]</td>
</tr>
<tr>
<td>suction temperature [°C]</td>
</tr>
<tr>
<td>compressor speed [r.p.m.]</td>
</tr>
<tr>
<td>cooling water inlet temp. [°C]</td>
</tr>
<tr>
<td>cooling water flow rate [l/s]</td>
</tr>
<tr>
<td>control strategy:</td>
</tr>
</tbody>
</table>

Valve Position in Turns Open
(predicted values in brackets)

<table>
<thead>
<tr>
<th></th>
<th>max. turns</th>
<th>test no. 39</th>
<th>test no. 48</th>
</tr>
</thead>
<tbody>
<tr>
<td>valve A</td>
<td>4.51</td>
<td>0.5 (0.29)</td>
<td>0.32 (0.20)</td>
</tr>
<tr>
<td>valve B</td>
<td>4.51 f.o.</td>
<td>f.o. (f.o.)</td>
<td>0.43 (0.43)</td>
</tr>
<tr>
<td>valve C</td>
<td>11.0</td>
<td>0.65 (0.37)</td>
<td>0.23 (0.14)</td>
</tr>
<tr>
<td>valve D</td>
<td>4.51</td>
<td>0.42 (0.47)</td>
<td>0.44 (0.50)</td>
</tr>
</tbody>
</table>

Table 5-5 Comparison of observed load stand valve positions with predicted values.

5.7 CONCLUSIONS

A design rationale for the load stand which was built and used for compressor testing has been presented. It is not claimed that the design which was adopted was in any sense the optimum for compressor testing. However, it is felt that the techniques which have been described should prove useful to other workers in designing and analysing compressor load stands in which the operating conditions are controlled by means of throttle valves. Indeed, the analysis methods could be applied to refrigeration systems generally and to other flow networks involving throttling and mixing of condensable fluids.
Chapter 6
THE LOAD STAND AND ITS INSTRUMENTATION

The load stand which was constructed is shown schematically in fig. 6-1. It was a hot gas bypass type, employing a single heat exchanger, in which part of the total refrigerant flow coming from the compressor was fully condensed to the liquid state and then mixed with the remainder to produce a homogeneous vapour, which was returned to the compressor suction side. A particular feature was that the suction and discharge pressures and the suction temperature were controlled by means of throttle valves. Fig. 6-2 is a photograph of the plant. The main items of equipment which were used are listed in appendix F.

6.1 CONTROL OF OPERATING CONDITIONS
The compressor was belt driven by a standard three phase induction motor and the speed was set by means of a control knob on an electronic frequency inverter.

The saturation temperature in the heat exchanger was determined by the temperature and flow rate of the cooling water and by the rate of heat rejection from the refrigerant. This rate was approximately equal to the indicated power of the compressor. The cooling water for the heat exchanger was taken from the normal cold supply of the building, via a constant head tank. With the maximum water flow rate of about three litres per minute, the sensitivity of the heat exchanger saturation temperature to compressor indicated power was, typically, 5 K/kW. Thus, the saturation temperature changed little with the operating conditions of the compressor.

The control valves are identified in fig. 6-1 by the letters A to E. Discharge pressure was controlled by valve A while valve D was used to control the compressor suction pressure. The settings of valve B in the vapour bypass line and valve C in the liquid line determined the suction superheat temperature. Valve B also provided the necessary pressure difference across valve C to ensure adequate liquid feed to that valve, which was located above the heat exchanger level. For the most part valve B could be left fully open, but, had to be
Fig. 6-1. Schematic diagram of the compressor load stand. Lower case letters indicate measurement locations: refer to table 6-3, section 6.12, for descriptions and values.
closed partially at very low compressor speeds. Valve C on the schematic diagram was realised in the form of two valves, \( C_c \) and \( C_f \), in parallel, one for coarse and one for fine adjustment. These metered the flow rate of liquid from the heat exchanger. It was found necessary to install two valves to obtain adequate sensitivity over a wide range of operating conditions. They were fitted with micrometer handles so that their settings could be noted and repeated.

The refrigerant circuit control valves A, B, C and D were interactive to some extent, as described in the previous chapter.

Valve E controlled the flow rate of the heat exchanger cooling water - it was normally left in the fully open position. Any reduction of the cooling water flow rate had the effect of raising the saturation temperature within the heat exchanger.

6.2 **MEASUREMENT OF TORQUE**

The electric motor was mounted on a cradle which was pivoted about the axis of rotation. Its reaction torque was measured
Fig. 6-2  The compressor load stand.
by a spring balance attached to a radius arm on the cradle. Care was taken to ensure a right angle between the radius arm and the line of action of the spring balance. A pointer which was attached to the cradle for this purpose could be realigned, by means of a hand wheel on the spring balance support, after any change in torque. The motor cradle and the spring balance can be seen in fig. 6-3.

Shaft power was computed from the motor torque and speed, and thus included any power loss in the belt drive system.

Fig. 6-3 The compressor and its driving motor.

6.3 MEASUREMENT OF SPEED

Motor speed and compressor speed were measured by the use of thin serrated rims attached to the belt pulleys (see fig. 6-3). There were 180 teeth on the rim attached to the compressor pulley and 98 teeth on the rim attached to the motor pulley. The tooth spacing was approximately the same on both rims. Slotted opto switches, which straddled these rims, provided square wave pulse trains. The opto switches were connected to
Fig. 6-3  The compressor and its driving motor.
counting modules in the data acquisition system and the rotational speeds were computed from the pulse count of each switch over a period of one second.

A third opto switch, together with a tag attached to the compressor pulley, provided a top dead centre marker pulse. This switch was connected to a digital storage oscilloscope, where a trace containing the marker was recorded on one of its four channels for transfer to the data logging computer.

6.4 MEASUREMENT OF TEMPERATURES

Temperatures on the refrigerant and water circuits were measured by means of type K thermocouple junctions attached to the pipework. Each junction was electrically insulated from the pipework by a thin patch of plastic insulating tape and was held in position by fabric reinforced adhesive tape. The thermal insulation around the pipework ensured that the temperatures measured by the thermocouples were close to those of the fluid within.

Towards the end of the testing programme thermocouples were inserted within the suction and discharge plenums. These were type K and were situated to measure the mean temperatures of the fluid entering the suction valve and leaving the discharge valve. The thermocouple leads can be seen in fig. 6-4. A hexagonal headed setscrew was used to support each thermocouple junction. The two insulated wires of each junction were passed through two small holes at the bottom of a deep hole drilled from the setscrew head. The emerging ends were stripped and twisted to form a junction and a bead of epoxy resin covering both small holes provided gas sealing. The setscrew heads were sealed against the cylinder head by means of copper washers. The suction plenum thermocouple junction was located directly above the suction port. The discharge plenum thermocouple was located just to the side of the discharge valve, so as to be in the path of the jet emerging from between the valve reed and the valve plate.

Thermocouples were also fitted on the outer surface of the compressor to measure the cylinder temperature (at a
Fig. 6-4 View of the compressor cylinder head. The twisted pair thermocouple leads and the cream coloured leads of the valve displacement transducers can be seen where they enter the cylinder head. The cylinder pressure transducer adapter can be seen where it enters the valve plate - it has a green lead attached.

representative location between two of the fins) and the crankcase temperature. These were attached by adhesive tape and were covered with thermal insulation pads.

The thermocouples were connected to a dedicated module within the data acquisition system. This module incorporated a platinum resistance thermometer to measure the temperature of a copper block to which the reference junctions were attached. Plugs were fitted to the thermocouples so that they could be easily connected to the data acquisition system, or, to a portable meter for spot checks.

6.5 MEASUREMENT OF PRESSURE

There were four main pressure levels within the refrigerant circuit: suction pressure, discharge pressure, heat exchanger pressure and mixing pressure. The last mentioned was the pressure in the suction line before valve D (refer to fig. 6-1).
Fig. 6-4  View of the compressor cylinder head.
Pressure gauges were installed to give visual indications of these and, for the suction and discharge pressures, transducers were also installed for data logging purposes. A further gauge and transducer were installed to measure the refrigerant pressure just upstream of the flow measurement orifice. This pressure, which was needed for the orifice flow rate calculations, was higher than the suction pressure, due to pipe friction and the presence of the suction line baffle and measurement orifice (see fig. 6-5). The transducers were energised by a regulated 10 V supply and their outputs were connected to a programmable gain amplifier and multiplexer module in the data acquisition system. A manifold with shut off valves allowed the pressure transducers and their associated gauges to be isolated from the system and connected separately, or severally, to a master gauge for calibration purposes.

6.6 MASS FLOW RATE OF REFRIGERANT

Two independent methods were used to determine the mass flow rate through the compressor. In the primary method an orifice plate in the suction line was used, while the confirming method consisted of a heat balance between the refrigerant and cooling water circuits.

6.6.1 ORIFICE PLATE METHOD

The equipment setup which was used to measure the mass flow rate by means of an orifice plate is shown diagrammatically in fig. 6-5. The flow system consisted of an initial straight pipe run, a tube bundle flow straightener, a second straight run, the measurement orifice, a third straight run, a perforated baffle plate, a short length of pipe with a reduced diameter, and a pulsation damping chamber.

In order to measure the total refrigerant flow rate through the compressor the orifice could have been located in either the suction line or the discharge line. The suction line was chosen as pressure pulsations were known to be less severe in it. Also, the specific volume of the refrigerant was higher in the suction line than in the discharge line. Thus, for a given
Fig. 6-5 Diagram of the equipment setup which was used to measure the refrigerant mass flow rate by the orifice plate method.
mass flow rate and pressure difference, a larger orifice could be used in the suction line. This was an important consideration at the mass flow rates involved, since the orifice diameter would have been small in either case. It was also felt that the orifice pressure drop could be accommodated more readily on the suction side of the heat exchanger and that the flow straightener and pipe run to the orifice would ensure that the mixing process to produce the suction vapour condition was complete.

The temperature and pressure of the refrigerant upstream of the orifice were measured and logged by the data acquisition system in order to calculate the specific volume for use in the orifice flow equation.

For the early tests a differential pressure transducer was used to record the pressure difference across the orifice. This was connected to the programmable amplifier module of the data acquisition system. The data acquisition was controlled by a high speed clock module. A rotary four way changeover valve was installed which allowed the pressure across the transducer to be equalised, and the transducer to be re-zeroed, while testing was in progress. Two further valves were provided which allowed a mercury manometer to be connected across the transducer to calibrate it, or, to indicate the mean pressure difference across the orifice visually. A mercury manometer was specially constructed to safely withstand the operating pressures of the system. For most of the tests described in this thesis, this was the instrument used for the measurement of the differential pressure across the orifice.

6.6.1.1 Calibration

The design of the orifice flow meter was based on British Standard 1042 [48, 49, 50, 46], but, all of the constraints imposed by the standard were not met. Judicious extrapolation of the calibration data was necessary due to the fact that the standard included within its scope only orifices in pipes of 25 mm inside diameter or greater. The pipe actually used had an inside diameter of 20.2 mm. It was considered to be unwise
to increase the diameter further, as sufficient velocity was required to transport entrained oil droplets. The flow equation which was used for the orifice flow meter is described in Appendix G. Rather than interpolating for two of the parameters in the equation which were tabulated in the British Standard, analytical expressions, as found in reference [51], were used instead.

6.6.1.2 Development of the Orifice Plate Flow Measurement System

Originally an orifice plate of 12.8 mm diameter was installed, without the pulsation damping chamber or baffle plate shown in fig. 6-5. The connections from the pressure tappings were of capillary tubing and the calculated natural frequency of the pressure inducting system at either side of the transducer was about 49 Hz, with a damping coefficient of about 0.04 (refer to section 4.6.1, DYNAMIC PRESSURE MEASUREMENT, ch. 4).

The differential pressure transducer output was read 400 times at intervals of 1,100 μs (equivalent to 91 readings per crank revolution at 600 r.p.m.). The zero reading was subtracted from each value and the effective transducer output was calculated. The formula used, equation 6.1, was the equivalent of calculating the refrigerant flow rate corresponding to each instantaneous differential pressure measurement at closely spaced intervals over a number of revolutions (e.g. 4.4 revolutions, or 8.8 pulsation cycles, at 600 r.p.m.) and then calculating the mean value. The use of absolute algebraic values in formula 6.1 took account of situations where the pressure transducer indicated a reversed pressure differential for part of the measurement period. This could occur due to reversed flow through the orifice during part of the pulsation cycle. The formula also avoided problems to do with the square root of a negative number, which could arise due to noise on the transducer output. Such noise could result in small negative (and positive) apparent pressure differentials when no actual differential existed.
where

\[ Y = \left( \frac{\sum_{i=1}^{n} |X_i - X_0| \cdot 5 (X_i - X_0) / |X_i - X_0|}{400} \right)^2 \]

(6.1)

\( Y \) = effective digital output of pressure transducer relative to zero differential pressure (in range 0 - 2048 approx.)

\( X \) = digital value of transducer output (in range 0 - 4095)

\( X_0 \) = digital value of transducer output with zero differential pressure applied (2048 corresponded to zero millivolts)

Fig. 6-6 is an example of a measured differential pressure waveform. In this case the peak-to-peak amplitude of pressure pulsations across the transducer was roughly 150% of the mean value.

While the differential transducer was in use there were significant apparent differences (28% at one stage) between the mass flow rates measured by the orifice plate and those
calculated from the heat balance. It was found retrospectively, after a number of problems had been investigated and resolved, that the flow measurement method outlined thus far did actually yield accurate results. The following paragraphs outline the factors which led to a change of approach to the flow measurement problem.

Difficulties were experienced with the differential pressure transducer and complete failure occurred eventually. The output corresponding to zero differential pressure shifted a number of times and, as a result, fell well outside the manufacturers' specifications. Erratic waveforms were also produced, sometimes alternating with seemingly normal smooth waveforms as in fig. 6-6.

After discussions with the manufacturers and further testing with a new transducer it was thought that the first transducer could have been damaged by the application of a vacuum. In this transducer, which was a wet/wet type, specified to tolerate liquid oil in the pressure lines, silicone oil filled the space between a stainless steel isolating diaphragm on the positive pressure side and the silicon measurement diaphragm. It was thought that the application of a vacuum, before the load stand was charged with refrigerant, had caused a bubble to appear within the silicone oil. The hypothesis was that, when pressure was applied, the bubble did not collapse immediately and the increased volume caused distortion of the measurement diaphragm, changing the zero offset of the strain gauge bridge. It was thought that the erratic waveforms were caused by the continued presence or reappearance of this bubble within the silicone oil. In order to continue using this type of transducer it was decided to modify the plant and the refrigerant charging procedure so that the transducer would not be subjected to a vacuum at any time. It was also considered expedient to dampen the flow pulsations in order to reduce any likelihood of vapour bubble formation within the silicone oil in the transducer.

As the next phase in the development of the flow measurement system, a pulsation damping chamber of about 1.5 litres
volume was installed and a perforated baffle plate was designed for an allowable flow rate error of less than 0.5\%, based on measurement of the mean differential pressure and the criterion given in British Standard 1042 [46], see equations 4.10 and 4.11. The four way changeover valve shown in fig. 6-5 was also installed, in order to isolate the differential pressure transducer during charging. The pressure connections to and from this were of capillary tubing. With these components the peak to peak amplitude of pressure pulsations at a compressor speed of 680 r.p.m. was found to be about 40\% of the mean differential pressure. Equation 6.1 continued to be used to calculate the equivalent steady differential pressure.

There were still significant differences between the refrigerant mass flow rate calculated from the orifice plate measurements and that calculated from the heat balance. It was decided to reduce flow pulsations still further and so the damping chamber was replaced with a larger one of 5.4 litres volume. This reduced the calculated maximum error due to pulsations to 0.014\%. The pressure line capillary tubes to and from the four way valve were replaced by tubes having an inside diameter of 4.5 mm. The latter changes would have increased the natural frequency and reduced the damping coefficient of the pressure measurement system considerably. The estimated minimum natural frequency for each side of the transducer was about 170 Hz, with a damping coefficient of about 0.005. Thus, as a result of the changes, the level of flow pulsations was reduced further while the response of the measurement system to dynamic pressure variations was improved considerably. These modifications reduced the observed peak-to-peak amplitude of pressure fluctuations to less than 25\% of the mean value (at 600 r.p.m.). However, there were still significant differences between the mass flow rates measured by the orifice plate and heat balance methods and the modifications had not reduced these appreciably.

Further changes were made which did give rise to substantial improvement in the agreement between the two flow
measurement methods. These were the following:

1. The refrigerant/water heat exchanger was carefully insulated to minimise heat losses or gains. Any external heat transfer was taken into account by means of a calculated heat transfer coefficient.

2. The water flow meter was calibrated and a calibration equation was established so that the indicated flow rate could be corrected within the data acquisition software.

3. The discharge coefficient for the orifice plate was calculated on the basis of the measured temperature and pressure of the refrigerant at entry, and the flow rate. The latter aspect required an iteration loop in the evaluation, starting with an approximation to the flow rate. The expansibility factor, in particular, within the flow equation, depended strongly on the measured pressure drop across the orifice and so a constant value would have caused errors if it were used over a range of flow rates. The discharge coefficient calculations, which also involved lookup of refrigerant state properties, and were too tedious for manual repetition, were incorporated in the computer program for the calculation of the test results.

4. The presence of entrained lubricating oil was taken into account in the heat balance calculations.

The next modification to the orifice plate measurement system was to replace the orifice plate with one having a reduced orifice diameter of 6.02 mm. The purpose of this was to increase the pressure difference across the pressure transducer to make better use of its full range of 350 mbar. Also, as a result of this change, the calculated error due to pressure pulsations was further reduced to a negligible 0.006%. Equation 6.1 continued to be used as a means of averaging the instantaneous differential pressure values. A simple arithmetic mean value would, of course, have been equally suitable as the pulsation level was negligible.

The new differential pressure transducer failed due, it was concluded, to a transient overload associated with choking at either the orifice plate, or, more probably, at valve D. The computer model of the load stand showed that, with a 6 mm orifice, it was impossible to achieve steady state operation with low suction saturation temperatures, due to choking at the orifice plate. In the actual plant, attempts to control the suction pressure resulted in sudden drops to low vacuum levels downstream of the orifice. It may be that an external overload, sufficient to cause rupture of the transducer diaphragm,
was applied, due to choking at the orifice. From discussions with the manufacturers, a differential pressure of up to about 6 bar would have been necessary to cause rupture. Hence, it was considered more likely that failure was due to a sudden lowering of the line pressure associated with choking at valve D. In efforts to control the operating conditions this valve would have been closed sufficiently to cause choked flow. A failure mechanism was postulated in which the transducer diaphragm was overloaded internally due to the external lowering of the line pressure. This is described in Appendix H. On the basis of this failure mechanism it was considered that the particular transducer was unsuitable for use on the load stand. However, the manufacturers did not accept responsibility for the unsuitability.

As funds were not available to replace the transducer again, it was decided to use a mercury manometer in conjunction with an 11.06 mm diameter measurement orifice. The final version of the orifice plate flow measurement system was as shown in fig. 6-1, without the differential pressure transducer. With this version, the calculated flow measurement error due to pulsations was estimated at about 0.017% and was considered negligible.

Improvements to the calculation procedures were applied retrospectively to measurements which had been made with the differential pressure transducer. Good agreement was obtained between the flow rates measured using the transducer and also using the manometer with those measured by the heat balance method.

6.6.2 HEAT BALANCE METHOD

All parts of the system were insulated to minimise heat gains or losses. The mass flow rate of refrigerant through the compressor was determined by a heat balance on the condenser, combined with a heat balance for the liquid and vapour mixing process. The rate of heat transfer to the cooling water was obtained from its flow rate and temperature rise. Any heat exchange with the surroundings was taken into account by means
Ch. 6 The Load Stand and its Instrumentation

of an overall heat loss coefficient. This had a calculated value of 0.712 Watts per Kelvin temperature difference between the surface of the condenser, beneath the insulation, and the temperature of the surrounding air. The mass flow rate of refrigerant through the compressor was given by

\[
m = \frac{Q + k(T_{hx} - T_o)}{h_f - h_h} \left[ 1 + \frac{h_j - h_h}{h_f - h_j} \right]
\]  

(6.2)

where,

- \( m \) = refrigerant mass flow rate through compressor [kg/s]
- \( Q \) = rate of heat transfer to cooling water [W]
- \( k \) = overall heat loss coefficient from the surface of the condenser to the surrounding air [W/K]
  (Its value was based on the thermal conductivity of the insulating material and on an assumed surface convection resistance of 0.123 m²K/W.)
- \( T_{hx} \) = measured surface temperature on the heat exchanger [K]
- \( T_o \) = measured temperature of the surroundings [K]
- \( h \) = specific enthalpy at measurement position indicated by subscript [J/kg]

It was found that at low apparent superheat values the specific enthalpy \( h_j \) was sensitive to the amount of lubricating oil present within the refrigerant (refer to ch. 7, sections 7.2 and 7.3). All refrigerant enthalpy values used within the calculations were actually those of the refrigerant/oil mixture.

Water flow rate was read visually from a variable area flow meter. A calibration equation was applied to the values when the data was processed.

6.7 OIL CONCENTRATION MEASUREMENT

An ASHRAE standard [52] described a method, based on weighing, to determine the mass concentration of lubricating oil in liquid refrigerant. This method was considered unsuitable for use in conjunction with the compressor test stand as it required that a sample of about 1 lb (0.454 kg) of refrigerant be taken from the system. The entire system charge was in fact only about 2.4 kg and the ASHRAE sample size would have been too large a proportion of this. A weight over volume method was developed.
instead, which required a sample of less than 16 grammes. The method also had the advantage that only a minute quantity of refrigerant was lost from the system for each sample which was taken. This was achieved by venting the refrigerant sample to the suction side of the compressor after the volume had been noted.

6.7.1 RATIONALE OF THE METHOD AS APPLIED TO THE LOAD STAND

Ideally, a sample should have been taken of the full flow of refrigerant. However, as the method used required the refrigerant to be in the liquid state, the liquid line coming from the heat exchanger was chosen as the location for the measurement. A major assumption underlying the technique was, therefore, that the oil concentration in the liquid refrigerant coming from the heat exchanger was the same as that in the refrigerant vapour coming from the compressor. In the long term no build up of oil could occur in the heat exchanger due to the high solubility of the oil in liquid R-12. Furthermore, if oil were to separate out within the heat exchanger it would lie below the liquid refrigerant and, due to the position of the outlet tube, would leave first. It was necessary to ensure that, as far as possible, the refrigerant which went to the heat exchanger had the same oil concentration as the greater part of the flow, which bypassed it. The angle which the branch to the heat exchanger formed with the main pipe was made as small as possible (about 30°) to minimise any effect of the tendency of oil droplets to continue in a straight line. The diameter of the branch pipe to the heat exchanger was chosen so that the vapour velocity within it would be about the same as in the main stream. The size of the sample bleed valve to the compressor suction side was chosen to ensure a low vapour velocity within the sample vessel, to avoid entrainment of liquid particles.

6.7.2 EQUIPMENT DETAILS

The locations of the sample vessel and its associated valves on the load stand are shown in fig. 6-1. The three port valve which was used to take the sample was of a type which
Ch. 6 The Load Stand and its Instrumentation

contained no dead space, where an unrepresentative liquid sample could lodge. This characteristic was essential for the satisfactory operation of the measurement system. A transparent hinged housing made of Perspex was fitted around the sample vessel as a safety device in case the glass should rupture under pressure. The vessel and its transparent safety cover can be seen in fig. 6-7. Fig. 6-8 is a diagram of the sample vessel.

Fig. 6-7 View showing the liquid line, in which two sight glasses are installed, and the refrigerant sample vessel with its safety cover. The three port sampling valve is directly above the sample vessel and has a red knob. The cruciform pipe connector above the sample vessel links the sampling valve, the vacuum pump connection, the sample vessel and the bleed connection. The isolating valve for the vacuum pump (which has no knob) and the bleed valve to the suction line can be seen. Also in this photograph are the two micrometer handles of the liquid metering valves. The two valves with the upright spindles and the horizontal black handles are: on the left, valve B in the vapour bypass line and, on the right, valve D in the suction line. On the lower left hand side of the photograph is the four way changeover valve which is connected to the orifice plate pressure tappings.

6.7.3 PROCEDURE

A sample of up to about 12 ml of refrigerant was drawn into
Fig. 6-7  View showing the liquid line, in which two sight glasses...
the evacuated glass vessel. The volume of refrigerant was read from a scale and it was then allowed to evaporate slowly by the opening of a bleed valve to the suction line. When there appeared to be only oil left in the vessel the bleed valve was closed and the sample vessel was vented to atmosphere. It was then connected to a vacuum pump and evacuated to ensure the
evaporation of any remaining refrigerant. The vessel was then opened to atmosphere, removed, and weighed to determine the weight including the sample. After cleaning, the vessel was re-weighed and, by subtracting the second weight from the first, the mass of oil was calculated.

During evaporation of the liquid refrigerant the temperature within the sample vessel dropped and this had the effect of slowing down the evaporation process. A visible manifestation of this effect was condensation on the outside surface of the glass vessel, indicating that its temperature was below the dew point of the surrounding air. It was found that the evaporation process could be speeded up significantly by gently heating the vessel using a hair drier. The disappearance of the condensation was taken as the signal to discontinue the heating.

6.8 VALVE LIFT MEASUREMENT

Eddy current type proximity transducers were used to measure valve lift (see figs. 6-9 to 6-12).

The suction valve transducer was mounted almost flush with the inner valve plate surface. As it was desired not to alter the compressor characteristics, the choice of location was very restricted and compromises had to be accepted. The transducer was installed in a position a little to the side of the valve port and directly under the web separating the suction and discharge plenums in the cylinder head. A clearance hole was drilled through the valve plate and the threaded transducer was mounted, using thread locking fluid, in a tapped hole through the web (fig. 6-10). At this location the web was modified slightly to increase its thickness by deposition of cast iron weld. It was not possible to provide a recess of one diameter all around the tip of the transducer, as recommended in the manufacturer's data sheet. The maximum recess which did not interfere with the valve seat was used instead. A PTFE insert was press-fitted into the recess and was milled so that the groove around the valve seat continued into the insert (fig. 6-9), maintaining the original geometry. The clearance volume of the cylinder was thus unaffected.
Fig. 6-9 View showing the tip of the suction valve displacement transducer surrounded by the white PTFE insert. The pressure port for the piezoelectric pressure transducer is also visible (to the right of the suction valve port).

Fig. 6-10 View of the inside of the cylinder head showing the suction valve lift transducer (protruding from the web which separates the suction and discharge plenums) and the discharge valve lift transducer (with its two lock nuts).
Fig. 6-9  View showing the tip of the suction valve displacement transducer...

Fig. 6-10  View of the inside of the cylinder head...
Fig. 6-11 View of the plenum side of the valve plate: the right hand valve back plate and tie bar were modified. The hole through the valve plate for the suction valve transducer can also be seen.

Fig. 6-12 Diagram illustrating the mounting of the discharge valve displacement transducer (not to scale).
Fig. 6-11  View of the plenum side of the valve plate.
The discharge valve lift transducer was mounted over the midpoint of the valve. Fig. 6-12 is a diagram of the arrangement. The transducer was supported in a tapped hole in the cylinder head and two lock nuts were used (figs. 6-10 and 6-12). Radial clearance for the tip was provided by drilling a large hole through the spring-loaded valve back plate. Reinforcement of the back plate was necessary as a consequence. A hole had also to be drilled through the tie bar between the two valve mounting bolts.

6.8.1 CALIBRATION
The manufacturers' calibration data did not apply to either of the transducers due to insufficient radial clearances at the tips and due to the fact that the target area on the suction valve was less than ideal. For this reason the transducers were calibrated after they had been mounted on the cylinder head and with the valve plate bolted to the head. The gasket between the two was in place when this was done so that its thickness was taken into account in the calibration. All ten bolts were tightened to full torque. Access to both valves for displacement measurement during the calibration was from the cylinder side of the assembly. Polynomial curves were fitted to the measured lift versus output data and were used in processing the measurements in the course of compressor testing.

6.8.2 INTERFACING
Each transducer was connected to its own proximitator (the unit which contained the electronic circuitry). The proximitators were powered by a -24 V d.c. power supply and gave an output in the range 0 to -22 V. These outputs were connected to two of the input channels of a digital storage oscilloscope, from which the waveform data was transferred to the data logging computer on an IEEE bus.

6.9 CYLINDER PRESSURE MEASUREMENT
A miniature piezoelectric pressure transducer (6.3 mm dia.) was installed in a hole drilled into the edge of the valve plate
Ch. 6 The Load Stand and its Instrumentation

(the end of the transducer adaptor and the connecting lead can be seen in fig. 6-4). A 4 mm pilot drill, and a step drill supplied by the transducer manufacturers, were used. A 3.3 mm hole was drilled from the cylinder side of the valve plate as the pressure port (see photograph, fig. 6-9). The geometry of the pressure inducting system is shown in fig. 6-13.

Fig. 6-13 Section through the pressure inducting system of the piezoelectric pressure transducer used to measure the cylinder pressure.

6.9.1 FREQUENCY RESPONSE CHARACTERISTICS

The calculated organ pipe frequency was 4929 Hz (eqn. 4.1, ch. 4, taking the velocity of sound as 138 m/s and the total passage length as 7 mm). If the small space between the transducer and the 3.3 mm pressure port passage were regarded as a 'spring' element of a Helmholtz resonator, the calculated natural frequency would have been 3063 Hz (eqn. 4.6). The damping coefficient was calculated as 0.088 (eqn. 4.7). The transducer itself had a natural frequency of 150 kHz, while the upper cut off frequency of the charge amplifier was 10 kHz.

The maximum rotational frequency of the compressor was 15 Hz (900 r.p.m.). Using the worst case estimate of the pressure inducting system's natural frequency and the calculated damping coefficient, table 6-1 was produced to illustrate possible distortions to the higher frequency components of the pressure.
versus time waveform.

<table>
<thead>
<tr>
<th>Pressure Frequency Component / Hz</th>
<th>Amplitude Error / %</th>
<th>Phase Angle Error / °</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>negligible</td>
<td>-0.05</td>
</tr>
<tr>
<td>150</td>
<td>0.2</td>
<td>-0.49</td>
</tr>
<tr>
<td>1500</td>
<td>30.7</td>
<td>-6.45</td>
</tr>
</tbody>
</table>

Table 6-1 Calculated amplitude and phase angle errors of component frequencies within pressure/time waveforms due to the characteristics of the pressure inducting system.

6.9.2 CALIBRATION

The transducer had a sensitivity of -16.2 pC/bar and this value was used in setting the calibrated gain of the charge amplifier to provide a signal at an appropriate level as an input to the digital storage oscilloscope. The captured waveform was transferred to the data logging computer over an IEEE bus, together with the valve lift waveforms and the trace containing the top dead centre marker.

The transducer did not measure the absolute level of the pressure within the cylinder, but rather, the changes which occurred during each cycle. The absolute level was deduced by assuming pressure equalisation existed across the suction valve at the point where it began to lift and that this pressure was equal to the mean suction pressure. Thus, the determination of the absolute pressure within the cylinder depended on the measurements of suction pressure and valve lift as well as on the dynamic measurement made by the the piezoelectric pressure transducer. Furthermore, the underlying assumption, that the cylinder pressure was equal to the mean suction pressure at the moment of valve lift, was questionable. This is discussed in section 7.4.1 of chapter 7. One of the main purposes of the cylinder pressure transducer was to determine the mean effective pressure, and hence, the indicated power. This determination was not affected by any imprecision in the absolute pressure level, but, depended solely on pressure changes within the cycle.
The maximum errors associated with the data transformations within the measurement system are summarised in Table 6-2.

<table>
<thead>
<tr>
<th>Component</th>
<th>Error Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure transducer maximum error</td>
<td>≤ 0.3% of range 0 - 25 bar</td>
<td>≤ 0.075 bar</td>
</tr>
<tr>
<td>Charge amplifier error</td>
<td>&lt; 1% of 16.2 bar</td>
<td>&lt; 0.162 bar</td>
</tr>
<tr>
<td>Storage scope A/D linearity error</td>
<td>&lt; 3% of range (typically 16 bar)</td>
<td>&lt; 0.480 bar</td>
</tr>
</tbody>
</table>

Table 6-2 Estimated maximum errors associated with the data transformations within the cylinder pressure measurement system.

6.10 DATA LOGGING

A computer interfacing system with secondary addressing, multiplexing and A to D facilities communicated with a host microcomputer on an IEEE bus. The thermocouples, opto switches and pressure transducers were connected to this unit.

A four channel digital storage oscilloscope with an IEEE bus output was used to capture valve lift and cylinder pressure data. This was triggered by a pulse from the top dead centre marker on the compressor pulley. One channel was used to mark the length of the cycle using the same TDC pulse. The stored data was transferred to the host microcomputer on the IEEE bus.

6.11 THE SUCTION LINE MIXING PROCESS

The thermodynamic processes which took place within the load stand and compressor are shown in Fig. 6-14, which is based on test measurements (from an early set of tests reported on by the author in [53]). Liquid coming from the heat exchanger was mixed with vapour from the bypass line and the mixture was throttled through valve D (refer to Fig. 6-1).

Thorough mixing of the liquid and vapour streams was essential to the operation of the load stand. Originally the liquid line coming from valve C was connected as the vertical limb of an inverted T-junction to the horizontal vapour line between valves B and D. Detailed monitoring of temperatures at close intervals along the pipework led to the conclusion that
liquid could build up in the bottom of the vapour line and, after a period, could flood through valve D. This effect introduced a considerable time lag between any adjustment of valve C and the corresponding change in the suction temperature at the compressor. Valve D had been mounted with its stem in the horizontal plane and this contributed to the problem also. This valve had an eccentric entry port which could form a liquid dam in the bottom of the pipe if not installed correctly.

Three steps were taken to overcome the accumulation of liquid:

(i) A venturi was installed in the vapour pipe (fig. 6-15). The line from valve C (in fact two lines, one from the coarse and one from the fine metering valve) was connected to the throat of the venturi.

(ii) The liquid metering valves, Cc and Cf, were moved as
close as possible to the venturi to minimise erratic lodgement of liquid in the short connecting pipes.

(iii) The stem of valve D was moved into the vertical upwards position so that the eccentric entry port lay on the bottom of the pipe.

These measures had the desired effect. Test readings confirmed that the refrigerant in the pipe, after valve D, was superheated and that further temperature changes in the long suction line were small and could be attributed to a small amount of heat exchange with the surroundings.

![Section through the venturi](image)

Fig. 6-15 Section through the venturi whose purpose was to improve mixing between the liquid and vapour refrigerant streams.

6.12 TEST PROGRAMME AND SAMPLE READINGS

The load stand and the testing procedures were developed over a considerable period of time and there were many modifications to the plant and instrumentation. Details of the testing programme and of the developments in equipment and methodology are included in appendix I.

The data in table 6-3 are the readings which were taken in test no. 64. Refer to the lower case letters on fig. 6-1 for the measurement locations.
Ch. 6 The Load Stand and its Instrumentation

| a | suction pressure [bara] | 2.19 |
| b | discharge pressure [bara] | 9.62 |
| c | suction temperature ['C] | 16.15 |
| d | compressor speed [r.p.m.] | 599 |
| e | discharge temperature ['C] | 90.54 |
| f | refrigerant temperature at branch to heat exchanger ['C] | 84.95 |
| g | heat exchanger surface temperature ['C] | 30.49 |
| h | refrigerant liquid line temperature ['C] | 19.93 |
| i | temperature of refrigerant after mixing ['C] | 19.63 |
| j | refr. temperature, suction line, 1st ['C] | 16.39 |
| k | refr. temperature, suction line, 2nd ['C] | 16.03 |
| l | refr. temperature, suction line, 3rd ['C] | 16.03 |
| m | refr. temperature, suction line, 4th (before orifice) ['C] | 16.33 |
| n | temp. of water into heat exchanger ['C] | 14.2 |
| o | temp. of water out of heat exchanger ['C] | 19.44 |
| p | ambient temperature ['C] | 18.89 |
| q | motor speed [r.p.m.] | 1248.9 |
| r | motor torque [Nm] | 11.33 |
| s | heat exchanger pressure [bara] | 7 |
| t | refrigerant pressure after mixing [bara] | 5.6 |
| u | water flow rate [l/min] | 3.00 |
| v | refrigerant sample volume [ml] | 7.81 |
| w | mass of oil [g] | 0.0289 |
| x | refrigerant pressure before orifice [bara] | 2.69 |
| y | differential pressure across orifice [mbar] | 70.04 |
| aa | mean temperature entering suction valve ['C] | 34.3 |
| bb | mean temperature leaving discharge valve ['C] | 96.6 |
| cc | temperature on external surface of cylinder ['C] | 55.37 |
| dd | crankcase temperature ['C] | 46.0 |

Table 6-3 Readings for test no. 64. In addition to these data, 251 values were stored for each of the three parameters: suction valve lift, discharge valve lift and cylinder pressure.

6.13 CONCLUSIONS

The load stand and the instrumentation have been described in some detail. These were set up in order to measure the performance of the compressor over a wide range of operating conditions. Control of the these conditions was by means of throttle valves and an electronic frequency inverter. It was intended that at a later stage control could be automated by using computer controlled motor actuators on the valves and by supplying a computer generated analogue input to the inverter.
Emphasis was placed on the computer based acquisition of all essential measurements to speed up the testing process and to facilitate computation and presentation of performance data.

Extensive changes were made to the orifice plate flow measurement system in the course of the work. It evolved from a system without pulsation damping based on dynamic pressure measurements to one with smoothened flow and static measurement of the differential pressure. Both systems yielded results which agreed well with measurements made by the heat balance method.
Chapter 7
COMPRESSOR PERFORMANCE CHARACTERISTICS

From the test measurements it was possible to derive, and graph or tabulate, many performance characteristics. Both the shaft power utilisation and the volumetric displacement utilisation were examined in some detail. As many different influences on the performance characteristics were superimposed in the operation of the compressor it was not possible to quantify all of these discretely on the basis of the measured data. Nonetheless, by making assumptions and carrying out sensitivity analyses, useful insights were obtained.

In all calculations involving the thermodynamic properties of specific volume, specific enthalpy and specific entropy the oil entrained in the refrigerant, as measured in the tests, was taken into account.

7.1 TEST CONDITIONS
The Bitzer IV compressor was tested over the speed range 300 to 900 r.p.m. and also over the range of suction temperatures from 0°C to 28°C. The nominal operating conditions are shown in table 7-1. In the tests versus suction temperature the minimum suction temperature, 0°C, corresponded to 10 K of suction superheat. At lower superheat values it was extremely difficult to achieve stable operating conditions of the compressor with the load stand. Thus, the minimum suction superheat for which results are presented is 10 K.

<table>
<thead>
<tr>
<th>Nominal Operating Conditions</th>
<th>Tests Versus Speed</th>
<th>Tests Versus Suction Temp.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suction saturation temperature [°C]</td>
<td>-10</td>
<td>-10</td>
</tr>
<tr>
<td>Discharge saturation temperature [°C]</td>
<td>40</td>
<td>40</td>
</tr>
<tr>
<td>Suction temperature at intake pipe [°C]</td>
<td>5</td>
<td>0 to 28</td>
</tr>
<tr>
<td>Speed [r.p.m.]</td>
<td>300 to 900</td>
<td>600</td>
</tr>
<tr>
<td>Mean ambient temperature [°C]</td>
<td>22.3</td>
<td>22.8</td>
</tr>
</tbody>
</table>

Table 7-1 Test conditions.
Ch. 7 Compressor Performance Characteristics

Ambient temperature was not controlled. Fig. 7-1 and fig. 7-2 show the ambient temperature measurements corresponding to all test results which are presented subsequently. All ambient temperatures fell within a band of approximately 6 K. It can be seen in fig. 7-2 that, by coincidence, the ambient temperature showed a decreasing trend with suction temperature.

7.2 MAIN PERFORMANCE CHARACTERISTICS

In this section the main performance characteristics of the compressor are presented. Refrigerant mass flow rate, shaft power input and discharge gas temperature are plotted over the speed range and over the suction temperature range.

7.2.1 REFRIGERANT MASS FLOW RATE

The mass flow rate of refrigerant (or, more correctly, of the refrigerant/oil mixture) was found to vary almost in direct proportion to speed over the range tested, fig 7-3. Good agreement was achieved between the primary and secondary (fig. 7-5) measurement methods, i.e. the orifice plate and the heat balance respectively, over the speed range.

Over the range of suction temperatures from 0°C to 28°C the mass flow rate, as measured by the orifice plate, remained roughly constant, fig. 7-4. The heat balance mass flow rate measurement method resulted in slightly lower calculated flow rates at high suction temperatures, fig. 7-6. As ambient temperature also decreased a little with increasing suction temperature, this may have had an influence on the mass flow rate as measured by the heat balance method. It is likely, however, that the main cause of the flow rates at high suction temperatures appearing to be lower according to the heat balance method was heat exchange between the suction line and the surroundings. At high suction temperatures this heat exchange was more strongly influenced by the temperature within the suction line than by the temperature of the surroundings. The maximum error of the secondary method with respect to the primary method, about 4.5%, occurred at 28°C suction temperature.
Ch. 7 Compressor Performance Characteristics

Fig. 7-1 Ambient temperature measurements for tests versus speed.

Fig. 7-2 Ambient temperature measurements for tests versus suction temperature.
Ch. 7 Compressor Performance Characteristics

TEST RESULTS vs. SPEED

![Graph showing mass flow rate vs. compressor speed](image)

Fig. 7-3 Mass flow rate measured by the orifice plate method versus speed.

TEST RESULTS vs. SUCT. TEMP.

![Graph showing mass flow rate vs. suction temperature](image)

Fig. 7-4 Mass flow rate measured by the orifice plate method versus suction temperature.

129
Fig. 7-5 Mass flow rate determined by the heat balance method versus compressor speed.

Fig. 7-6 Mass flow rate determined by the heat balance method versus suction temperature.
Bars are used on figs. 7-3 to 7-6 to illustrate the calculated precision of the mass flow rate data, based on the known or estimated accuracies of the measurement devices involved. Any inaccuracies in the heat balance data due to shortcomings of the mixing process or lack of thermal equilibrium are not included. The primary measurements were therefore considered to be significantly more accurate and reliable.

From the primary data, mass flow rate was found to vary in direct proportion to compressor speed and was insensitive to the suction temperature over a 28 K range.

7.2.1.1 Influence of Entrained Oil on Flow Rate Precision

Particular attention is drawn to the precision bar of the data point on fig. 7-6 corresponding to a suction temperature of 0.81°C. The low precision of the flow rate, calculated by the heat balance method, for this particular data point was due to a lack of precision in the specific enthalpy value $h_j$ (refer to equation 6.2, ch. 6) of the refrigerant after valve D in the suction line. The temperature at measurement location $j$ (refer to fig. 6-1, ch. 6) was -3.03°C and the corresponding pressure, which was used in evaluating the specific enthalpy, was 2.65 bara (measured at location $x$, fig. 6-1). At these conditions the 'apparent' superheat, assuming pure refrigerant, was 1.53 K. However, the enthalpy, $h_j$, was evaluated taking account of the entrained oil, which represented 0.33% of the mass flow. The oil caused part of the refrigerant to remain in the liquid state, giving a dryness fraction of 0.986. To illustrate the difference this made to the specific enthalpy the values are given below:

for pure refrigerant, $h_j = 186.54$ kJ/kg
for refrigerant/oil mixture, $h_j = 184.01$ kJ/kg

With the low apparent superheat at location $j$ for this data point the specific enthalpy value was particularly sensitive to the pressure and this caused the disimprovement in precision. At high apparent superheat values the specific enthalpy was relatively insensitive to pressure.
7.2.2 SHAFT POWER INPUT

Shaft power increased linearly with speed over the range 300 to 900 r.p.m. (fig. 7-7) and remained, in effect, constant over the range of suction temperatures tested (fig. 7-8).

7.2.3 DISCHARGE GAS TEMPERATURE

The discharge temperature showed a small linear increase with increasing speed over the range tested. This characteristic is shown in fig. 7-9.

Discharge temperature also increased linearly with suction temperature within the range 0°C to 28°C, the increase being about 23.8 K for the change of 28 K in the suction temperature, fig. 7-10.

7.3 OIL CONCENTRATION IN THE DISCHARGE VAPOUR

7.3.1 OIL CONCENTRATION CHARACTERISTICS

It was found that a considerable time interval at a given operating condition was necessary before reasonably consistent measurements of the oil concentration could be obtained. For instance, much higher oil concentration values were observed shortly after start up of the plant. Even with care the data showed a lot of scatter, figs. 7-11 and 7-12.

The steady state oil concentration was clearly low at low speeds. It reached a maximum of perhaps 0.4% in the mid speed range and appeared to decline at higher speeds. The value of nearly 0.6% shown on fig. 7-11 at about 800 r.p.m. was considered to be spurious. Considerable further testing would have been necessary to clarify the exact nature of the oil concentration versus speed characteristics.

Allowing for the scatter in the test results, the oil concentration showed, perhaps, just a small decline with increasing suction temperature. This would lend support to the assumption that the oil concentration in the discharge vapour depended mainly on the mechanics of oil throwing and entrainment by high velocity refrigerant, rather than on solubility effects. The possibility that the decreasing trend with suction temperature may have been related to the decrease in ambient
Fig. 7-7 Shaft power versus compressor speed.

Fig. 7-8 Shaft power versus suction temperature.
Ch. 7 Compressor Performance Characteristics

**TEST RESULTS vs. SPEED**

![Graph showing discharge temperature vs. compressor speed.]

Fig. 7-9 Discharge temperature versus compressor speed.

**TEST RESULTS vs. SUCT. TEMP.**

![Graph showing compressor discharge temperature vs. suction temperature.]

Fig. 7-10 Compressor discharge temperature versus suction temperature.
Ch. 7 Compressor Performance Characteristics

**TEST RESULTS vs. SPEED**

![Diagram showing oil concentration in the discharge vapour versus compressor speed.](image)

Fig. 7-11 Oil concentration in the discharge vapour versus compressor speed.

**TEST RESULTS vs. SUCT. TEMP.**

![Diagram showing oil concentration in the discharge vapour versus compressor suction temperature.](image)

Fig. 7-12 Oil concentration in the discharge vapour versus compressor suction temperature.
temperature could not be excluded.

In the concentrations measured in the tests over the speed range and over the range of suction temperatures, entrained oil did not greatly affect the thermodynamic properties of the refrigerant within the compressor, as it was well superheated. However, the effect on the properties would have been significant if the 'apparent superheat' was less than about 5 K. In the tests over the range of speeds the superheat at entry to the compressor was 15 K, while the minimum superheat over the range of suction temperatures was about 10 K.

Fig. 7-13 View of a Bitzer IV compressor with the cylinder head and valve plate removed. The first suction plenum can be seen as the box shape on the left hand side of the compressor. This connected, through the port which can be seen on the top face, with the second plenum within the cylinder head.
Fig. 7-13  View of a Bitzer IV compressor with the...
7.3.2 OIL TRANSFER WITHIN THE COMPRESSOR

In the Bitzer compressor, oil was transferred to the cylinder walls by splashes from the crankcase. The connecting rod big ends dipped into the sump oil near the bottom of the stroke and there was also an oil thrower on the crankshaft (see fig. 1-1, ch. 1). The vapour returning to the compressor in the suction line entered a plenum (the first of two) which had a drain to the crankcase, where some of the entrained oil droplets could fall out of suspension, due to the low velocity. This plenum can be seen in fig. 7-13.

It seemed likely that at low speeds the amount of oil thrown onto the cylinder walls, of which part was subsequently entrained in the refrigerant vapour within the cylinder, was less than at speeds in the mid range.

The fall off in oil concentration at the high end of the speed range may have been due to a number of factors such as the following:

1. a decrease in the viscosity of the oil, reducing the film thickness on the cylinder surfaces and therefore reducing the amount of oil available to be entrained. Cylinder temperature and oil sump temperature rose a little with increased speed, fig. 7-14.
2. a reduction in the oil throwing effectiveness due to the high rotational speed of the crankshaft - less oil may have remained on the crankshaft or thrower long enough to leave in an upwards direction.

The low oil concentration in the discharge vapour at low compressor speeds may have been an indication that cylinder lubrication was inadequate. Modifications to the lubrication system might be required for continuous running at low speeds. It was noted in chapter 1, section 1.4.1.1, that Itami et al. used a two stage pump to improve oil circulation at low speeds.

7.4 VALVE LIFT AND PRESSURE DIAGRAMS

These diagrams, figs. 7-16 to 7-18, were plotted from test readings captured on the digital storage oscilloscope.

7.4.1 DETERMINATION OF POINTS OF VALVE OPENING AND CLOSING

The crank angle positions at which the suction and
Fig. 7-14 Compressor temperatures versus speed.

Fig. 7-15 Compressor temperatures versus suction temperature.
Ch. 7 Compressor Performance Characteristics

TEST RESULTS

BITZER IV COMPRESSOR

Ps = 2.19 bara
Ts = 5.46 deg. C

Refrigerant 12

Pd = 9.6 bara
RPM = 300.3

Test 53

Fig. 7-16
TEST RESULTS

BITZER IV COMPRESSOR

Test 49

Ps = 2.19 bara

Ts = 4.73 deg. C

Refrigerant 12

Pd = 9.6 bara

RPM = 592.3

---

Fig. 7-17
TEST RESULTS

BITZER IV COMPRESSOR

Ps = 2.19 bar a  Pd = 9.63 bar a
Ts = 4.97 deg. C  RPM = 902.3
Refrigerant 12

---

**Fig. 7-18**

Crank angle
discharge valves opened and closed are represented by vertical lines in figs. 7-16 to 7-18. While these lines could be roughly drawn by hand on the valve lift diagrams, the identification, in a consistent way, of the precise points of opening and closing proved somewhat more difficult within a computer program.

Valve displacement data were available for, typically, 250 points (on an equal time interval basis) throughout the cycle. Due to a small degree of deflection while the valve remained seated and due to noise on the signal, an increase in displacement did not necessarily indicate opening. The criterion which was finally adopted was that the valve was considered to have opened when the lift rate exceeded 10 mm/s for three successive intervals. This was based on observations that the valves normally rose after opening at a rate higher than this for more than three sample intervals. In this way, the data point at or before valve lift was identified (position 1-3 in fig. 7-19). Using the slope of the second interval with a slope greater than 10 mm/s, the fractional increment, as shown in fig. 7-19, was calculated by trigonometry. Thus, the 'calculated position of valve opening' was determined to a resolution smaller than the sample interval. The displacement of the valve at opening was taken to be the displacement of the data point at or just before the calculated position of valve opening. This technique was applied to calculate the opening positions of both the suction and discharge valves.

As can be seen in figs. 7-16 to 7-18, the suction valve closed very gradually. Consequently, it was not possible to accurately identify the position of closing. The technique adopted was to search for the last data point, starting at a position within the compression stroke and working backwards, where the displacement was less than or equal to the opening displacement plus a small tolerance (to allow for the resolution of the analogue to digital converter). A similar approach was used to calculate the position of closing of the discharge valve. As the discharge valve was observed to close relatively quickly, a fractional interval was also calculated by projecting
NOTE: If $x < 0$, let $x = 0$
If $x > 1$, let $x = 1$

slope > 10 mm/s

Fig. 7-19 Diagram of valve displacement data versus time to illustrate the calculation of the position of valve opening.

the closing slope.

The accuracy of identification of the points of suction and discharge valve opening was, in principle, within one sample interval (typically about 1.5'). The point of discharge valve closure was identified with slightly less accuracy. The point of suction valve closure was not accurately identified, and should be regarded as a nominal position at which the valve opening was very small, perhaps zero.

In deciding on the absolute level of the cylinder pressure it was assumed that zero pressure difference existed between the cylinder and the suction pipe at the moment when the suction valve began to lift. This assumption ignored pressure losses from the suction pipe to the plenum side of the suction valve and also neglected pulsations within the plenum, but, was considered to be the best approximation possible in the absence of data for the fluctuating pressures within the plenums. The option of referring the absolute level of the cylinder pressure to the discharge pressure (assuming the two to be equal at
discharge valve opening) was also examined. This was found to be very much less satisfactory as it caused excessive variation in the apparent level of the minimum cylinder pressure with respect to the suction pipe pressure level, from one test to another, at the same operating conditions. It was concluded from a detailed examination of the test readings (see section 7.4.5, 'UNDER OR OVERPRESSURE AT VALVE OPENING OR CLOSING') that suction and/or discharge pressure pulsations were present and, thus, the absolute levels of cylinder pressure should be regarded as approximate only. The error in the absolute pressure level, which could not be determined from the measurements, would have been roughly equal to the amplitude of the suction plenum pressure waveform at the point of suction valve opening, for any particular test.

7.4.2 VALVE LIFT AND CYLINDER PRESSURE VERSUS CRANK ANGLE

Figs. 7-16 to 7-18 illustrate typical valve lift and cylinder pressure curves plotted against crank angle. The mean suction and the mean discharge pressure (measured at locations a and b, fig. 6-1, ch. 6, respectively) are shown by horizontal lines. The calculated angles of valve opening and closing are shown by vertical lines.

It can be noted that the discharge valve, which was a relatively stiff deflecting plate supported at its ends, rose and fell without significant oscillation. The suction valve, which was of the cantilever type, without a stop, showed considerable oscillation at its first natural frequency.

During the discharge process the cylinder pressure varied in a very smooth manner. The top of the pressure curve was rounded. Considerable care was taken to ensure that this was not an effect caused by a poor dynamic response characteristic of the pressure transducer, or, more particularly, of the pressure inducting system. It was concluded that it was indeed a characteristic of the slow speed compressor, and was due to the stiffness of the discharge valve. This conclusion was supported by simulation work and by the fact that at the very high speed, for this compressor, of 900 r.p.m., oscillation of
the discharge valve occurred and caused a rough top on the cylinder pressure curve, fig. 7-18.

The first natural frequency of the suction valve reed was estimated from the diagrams of lift versus crank angle. It was about 200 Hz and remained approximately constant over the full range of operating speeds. Thus, at lower speeds there were more oscillations than at higher speeds: compare fig. 7-16 at 300 r.p.m. with fig. 7-18 at 900 r.p.m. The lowest natural frequency of the discharge valve was estimated, very roughly, on the basis of the distance between the peaks on the discharge valve lift curve in fig. 7-18, as about 490 Hz. Tests had also been carried out to measure the natural frequencies of the valve reeds directly and the results are shown in table 7-2 [54]. In these tests the mounting arrangements, particularly for the discharge valve reed, did not correspond exactly with the way in which the reeds were normally supported on the valve plate.

<table>
<thead>
<tr>
<th>Valve Type</th>
<th>1st nat. freq.</th>
<th>2nd nat. freq.</th>
</tr>
</thead>
<tbody>
<tr>
<td>suction valve reed</td>
<td>136</td>
<td>698</td>
</tr>
<tr>
<td>discharge valve reed</td>
<td>768</td>
<td>2939</td>
</tr>
</tbody>
</table>

Table 7-2 Natural frequencies of the suction and discharge valve reeds determined with the reeds in a test fixture and excited by an electromagnet [54].

Over the range of suction temperatures from 0°C to 27°C there was little perceptible change in the pressure and valve lift diagrams. The main feature noted was that the suction valve began to open a little earlier at higher suction temperatures.

7.4.3 LIFT AND PRESSURE DIAGRAMS VERSUS VOLUME

The same data as shown in fig. 7-17 are shown in fig. 7-20, plotted against cylinder volume. The closed pressure versus volume curve is the traditional indicator diagram.
TEST RESULTS

BITZER IV COMPRESSOR

Ps = 2.19 bara  \hspace{1cm} Pd = 9.6 bara
Ts = 4.73 deg. C  \hspace{1cm} RPM = 592.3

Ref: Refrigerant 12

![Graphs showing discharge and suction valve lifts and cylinder pressure variation.]

Fig. 7-20
7.4.4 MEAN EFFECTIVE PRESSURES

The mean effective pressure (MEP), the suction effective pressure (SEP) and the discharge effective pressure (DEP) were calculated by numerical integration from the indicator diagrams. Values for these quantities are shown on fig. 7-20 and are graphed in figs. 7-21 and 7-22. Both the suction and discharge effective pressures increased with speed, as would be expected due to the higher flow rates through the valves. The mean effective pressure increased just a little with increasing speed. Allowing for the scatter of the data, all three parameters appeared to be insensitive to the suction temperature.

7.4.5 UNDER OR OVERPRESSURE AT VALVE OPENING OR CLOSING

If there were no pulsations or pressure losses within the plenums, the suction and discharge valves should have begun to open when the cylinder pressure was equal to the suction or discharge pressure respectively. To analyse such effects over a range of speeds a 'pressure equalisation factor' was defined as follows:

\[
P_{\text{factor}} = \frac{\text{disch. valve opening press.} - \text{suct. valve opening press.}}{\text{(discharge press.} - \text{suction press.)}}
\]  

(7.1)

The pressures in the numerator of equation 7.1 were measured by the cylinder pressure transducer, while the values in the denominator were measured by the suction and discharge pressure transducers.

Figs. 7-23 and 7-24 show calculated values of the pressure equalisation factor versus compressor speed and versus suction temperature. There was considerable scatter of the results, but, overall, the factor decreased with increasing speed. The fact that it was not always unity over the speed range suggested that pulsations within the plenums existed and caused over-pressure at the point of suction valve opening, or underpressure at the point of discharge valve opening. These effects increased with increasing speed.

Due to scatter and fewer data points, it was unclear from the data whether the pressure equalisation factor changed
Ch. 7 Compressor Performance Characteristics

TEST RESULTS vs. SPEED

Fig. 7-21 Mean effective pressure, suction effective pressure and discharge effective pressure, as calculated from indicator diagrams, versus compressor speed.

TEST RESULTS vs. SUCT. TEMP.

Fig. 7-22 Mean effective pressure, suction effective pressure and discharge effective pressure, as calculated from indicator diagrams, versus suction temperature.
Ch. 7 Compressor Performance Characteristics

TEST RESULTS vs. SPEED

Fig. 7-23 Pressure equalisation factor versus compressor speed.

TEST RESULTS vs. SUCT. TEMP.

Fig. 7-24 Pressure equalisation factor versus suction temperature.
significantly with suction temperature. It appeared to decrease a little with increasing suction temperature.

7.5 UTILISATION OF VOLUME DISPLACEMENT

7.5.1 DISPLACEMENT UTILISATION EFFICIENCY

The displacement utilisation efficiency (\( \eta_v \), eqn. 2.1, ch. 2) was found to remain almost constant over the range of compressor speeds, at a value of about 66%, fig. 7-25. This fact would be particularly significant if the compressor were to be operated over this speed range for the purposes of capacity control, as there would be no further net displacement utilisation losses due to such operation.

Paul et al. measured values from 70% to 64% (table 1-1, ch. 1) over a range of speeds for a similar compressor with a similar compression ratio and 10 K suction superheat. They found a decrease in displacement utilisation efficiency with increasing speed. However, the refrigerant used was R-22, which is not fully miscible with refrigeration oil. This comparison of results would suggest that solubility effects could be significant when the refrigerant is R-12.

From the line drawn through the plotted results in fig. 7-25, the displacement utilisation efficiency showed an increase from about 66% to about 74% when the suction temperature increased from 0°C (10 K superheat) to 30°C (40 K superheat). The mass flow rate, however, remained roughly constant over the same range, fig. 7-4. This effect, of increased volumetric efficiency with increased superheat, has been noted by other workers too. In fact, the figures given above are remarkably similar to those quoted by Gosney (see ch. 2, 2.4.5, 'EVAPORATION, CONDENSATION AND ENTRY OF REFRIGERANT INTO SOLUTION').

7.5.2 INDICATED VOLUMETRIC EFFICIENCY

In analysing the indicator diagrams it was found that, for many tests, the second point of suction pressure equalisation apparently occurred before the piston reached bottom dead centre on the induction stroke (fig. 7-16 is an example). This effect was possibly due to lack of precision in positioning the
Ch. 7 Compressor Performance Characteristics

Fig. 7-25 Volumetric efficiencies versus compressor speed.

Fig. 7-26 Volumetric efficiencies versus suction temperature.
Ch. 7 Compressor Performance Characteristics

indicator diagram on the absolute pressure scale, or, to suction plenum pulsations. It was not therefore possible, on the basis of the test data, to say where pressure equalisation actually occurred. In these cases the bottom dead centre position was taken as the second point of suction pressure equalisation for the purposes of evaluating the indicated volumetric efficiency and the underpressure loss ratio.

The indicated volumetric efficiency ($\eta_{vi}$, eqn. 2.6, ch. 2) showed a significant decrease from about 89% to about 81% over the speed range tested, fig. 7-25. Furthermore, these values were considerably higher than the actual displacement utilisation efficiency values. The differences were due to effects which may have included heat transfer, backflow, leakage, and refrigerant solubility in the lubricating oil. It was not possible to discriminate fully between these effects from the data available. Nonetheless, reasoned deductions were made and, where discrimination became impossible, due to the limitations of the test data, two extreme cases were examined in order to estimate the maximum extents of either heat transfer or leakage type effects.

The limited number of data points and the considerable scatter meant that it was difficult to make accurate deductions about the variation of indicated volumetric efficiency with suction temperature, fig. 7-26. However, it appeared that the value did not change significantly over the range 0°C to 28°C. This may have been due to the fact that the suction valve opened a little earlier at higher suction temperatures. The longer induction stroke may have compensated for the reduced vapour density at higher temperatures.

7.5.3 HEAT TRANSFER TO THE REFRIGERANT WITHIN THE SUCTION AND DISCHARGE PLENUMS

Towards the end of the compressor testing programme some thermocouple measurements were made of the mean refrigerant temperature within the suction plenum at entry to the suction valve port and within the discharge plenum adjacent to the discharge valve.
The thermocouples did not respond to any cyclic variations in temperature. The measured discharge temperature may therefore have been significantly lower than the instantaneous values during flow through the valve, due to the relatively short cyclic flow duration and cooling during the intervening periods. Flow occurred over the suction valve thermocouple over about half the cycle duration and this fact would probably have ensured that the measured temperature was closer to the instantaneous values during flow through the valve. It is likely that the measured suction valve inlet temperature was a little higher than the instantaneous values during flow. The calculated rates of heat transfer to the suction vapour within the plenum were thus, most likely, a little high, while the calculated rates of heat transfer from the vapour within the discharge plenum were probably somewhat low.

In spite of the above reservations, it was felt that the results were roughly correct, in representing the plenum heat transfer effects. In fig. 7-27 the plenum heat transfer rates are plotted against compressor speed. Heat transfer to the suction vapour within the plenum would have reduced the displacement utilisation efficiency by increasing the specific volume before the refrigerant passed through the suction port into the cylinder. Heat transfer which occurred within the discharge plenum would not have affected the displacement utilisation efficiency. Both rates of heat transfer increased linearly with speed.

It should be noted that although the rates of heat transfer within the plenums increased with speed, they did not increase in the same proportion as the mass flow rate of refrigerant. Therefore, the quantities of heat transfer to and from the refrigerant, within the suction and discharge plenums respectively, per unit mass, decreased with speed.

The rate of heat transfer to the suction vapour within the plenum showed a small decrease with increased suction temperature. The rate of heat transfer within the discharge plenum was insensitive to the suction temperature, fig. 7-28.
Fig. 7-27 Suction and discharge plenum heat transfer rates versus compressor speed.

Fig. 7-28 Suction and discharge plenum heat transfer rates versus suction temperature.
7.5.4 HEAT TRANSFER AND/OR LEAKAGE/BACKFLOW DURING INDUCTION AND DISCHARGE

The calculation procedure which was used for this aspect of the analysis of the test readings is shown as a flow diagram in appendix J and the results for one test are presented in table 7-3.

<table>
<thead>
<tr>
<th>Test no. 61</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Suction saturation temperature:</strong></td>
</tr>
<tr>
<td><strong>Discharge saturation temperature:</strong></td>
</tr>
<tr>
<td><strong>Suction temperature (location c, fig. 6-1, ch. 6):</strong></td>
</tr>
<tr>
<td><strong>Compressor speed:</strong></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Thermocouple measurements:</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Discharge temperature:</strong></td>
</tr>
<tr>
<td><strong>Temperature entering suction valve:</strong></td>
</tr>
<tr>
<td><strong>Temperature leaving discharge valve:</strong></td>
</tr>
<tr>
<td><strong>Cylinder temperature:</strong></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Plenum heat transfer rates:</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>To refrigerant within suction plenum:</strong></td>
</tr>
<tr>
<td><strong>From refrigerant within discharge plenum:</strong></td>
</tr>
</tbody>
</table>

**Assuming no valve or piston leakage and no backflow:**

| Volumetric induction efficiency: | 0.634 |
| Refrigerant temperature after induction: | 70.2 °C |
| Refrigerant temperature before discharge: | 137.5 °C |
| Heat transfer rate to vapour during induction | 0.957 kW |
| Heat transfer rate from vapour during discharge | 0.992 kW |

**Assuming adiabatic vapour transfer:**

| Volumetric induction efficiency: | 0.736 |
| Refrigerant temperature after induction: | 27.8 °C |
| Refrigerant temperature before discharge: | 99.6 °C |
| **Suction leakage ratio:** | 0.138 |
| **Discharge leakage ratio:** | 0.119 |

Note: the rates of heat transfer shown above were averaged over the full cycle duration for the full compressor mass flow rate.

Table 7-3 Results of heat transfer and leakage analyses for one test.

7.5.4.1 The 'No Leakage' Assumption

In the first case it was assumed that there was no leakage or backflow and that the actual displacement utilisation efficiency could be explained on the basis of the indicator diagram and the increased temperature of the vapour at the points where
Ch. 7 Compressor Performance Characteristics

the pressure within the cylinder was equal to the suction pressure, i.e. the points of suction pressure equalisation. The volumetric induction efficiency was thus assumed equal to the displacement utilisation efficiency. It was further assumed that the specific volume was the same at both points of suction pressure equalisation.

The results of the calculations showed that the assumption of no leakage or backflow was not valid. As shown in table 7-3 for test no. 61, at a speed near the top of the range, the temperature of the refrigerant at the end of the induction process would have to have been 70.2°C. This was not possible since the cylinder temperature was only 52°C in that particular case and no source of the necessary heat transfer at a higher temperature was available. Furthermore, it was noted that the rates of heat transfer during the suction and discharge processes were very large in themselves, and in comparison with the rates of heat transfer within the suction and discharge plenums: this was considered to be unlikely. The same conclusions were drawn for tests over the full speed range.

7.5.4.2 The 'Adiabatic Vapour Transfer' Assumption

In this case it was assumed that no heat transfer occurred to the refrigerant within the cylinder during the suction process, i.e. while flow occurred through the suction valve, between the points of suction pressure equalisation. Similarly, it was assumed that no heat transfer occurred from the vapour within the cylinder while there was flow through the discharge valve i.e. between the points of discharge pressure equalisation.

It can be noted from table 7-3 that, notwithstanding the adiabatic transfer of the suction vapour into the cylinder, the temperature of the refrigerant at suction pressure equalisation (point 1, fig. 2-1, ch. 2) was higher than the temperature at entry to the suction valve by 6 K. This temperature rise was related to the indicated suction pumping work (c.f. eqn. 2.18, ch. 2, for the 'adiabatic suction temperature after induction'). In a similar way, the temperature of the refrigerant at
Ch. 7 Compressor Performance Characteristics
discharge pressure equalisation (point 2, fig. 2-1, ch. 2) was
higher by 8.8 K than the discharge temperature. (These effects
were also taken into account in the analysis assuming no leak-
age.)

Under the assumptions of this analysis the volumetric
induction efficiency was higher than the displacement utilisa-
tion efficiency, the difference between the two being accounted
for by backflow and leakage past the suction and discharge
valves, by leakage past the piston, and by refrigerant/oil
solubility effects. The volumetric induction efficiency,
assuming adiabatic suction and discharge processes, is shown
plotted against speed in fig. 7-25 and against suction tempera-
ture in fig. 7-26. It was found to remain approximately
constant with increasing speed and, allowing for the scatter of
the data points, to increase slightly with increased suction
temperature. The heat transfer in the suction plenum, and the
temperature increase due to suction throttling and re-com-
pression, were taken into account in the calculated values.

Leakage ratios were evaluated and the first of these, the
suction leakage ratio (0.138 for test 61), included suction
valve and piston leakage, suction valve backflow and any evolu-
tion of refrigerant from the lubricating oil within the cylinder
during the re-expansion stroke. The data did not allow sepa-
rate quantification of these effects. The discharge leakage
ratio (0.119 for test 61) included backflow and leakage past the
discharge valve.

Values of the suction and discharge leakage ratios were
plotted against speed and showed a high degree of scatter, see
figs. 7-29 and 7-31. Due to the scatter of the data there was
doubt about the nature of the suction leakage ratio variation
with speed. The discharge leakage ratio, however, clearly
decreased with speed. The calculated values of both leakage
ratios were very sensitive to the volumes determined from the
indicator diagram at the points of suction and discharge pres-
sure equalisation. These volumes, in turn, were highly sensi-
tive to the accuracy in the positioning of the indicator diagram
on the pressure scale with respect to the suction and discharge
Fig. 7-29 Suction leakage ratio versus compressor speed.

Fig. 7-30 Suction leakage ratio versus suction temperature.
Fig. 7-31 Discharge leakage ratio versus compressor speed.

Fig. 7-32 Discharge leakage ratio versus suction temperature.
pressures. The scatter was thus attributed to limitations of the measurement methods and equipment, particularly the lack of a precise absolute reference for the internal cylinder pressure. Leakage through the discharge valve probably decreased with increased speed due to the reduced time available per unit of mass flow. In the case of the suction valve, a reduction in leakage at higher speeds may have been counterbalanced by increased backflow.

It can be seen from figs. 7-30 and 7-32 that both the suction and discharge leakage ratios decreased with increasing suction temperature. The decreases in the calculated 'leakage ratios' may, in fact, have represented decreases in the amount of heat transfer which occurred within the cylinder as the suction temperature increased.

7.5.4.3 Possible Solubility Effects

Any solution and evolution of refrigerant into and out of oil within the compressor could affect the induction and discharge processes. Refrigerant could evolve from the oil during the induction process, lessening the mass induced from the suction line. Similarly, refrigerant could go into solution in the oil during the discharge process, reducing the mass discharged to a value below that which might otherwise be implied from the indicator diagram. These effects, insofar as they occurred in the tests, could not be discriminated from leakage or backflow, given the test data. The suction and discharge leakage ratios presented in table 7-3 and figs. 7-29 to 7-32, therefore, included these effects also.

A further consideration is that the extent of the solubility effects would be determined by the rates of heat transfer between the cylinder surfaces, the oil film, and the refrigerant vapour. Due to the latent heat involved in solution or evolution, the induction and discharge processes could be close to isothermal, while being highly non-adiabatic, if the system boundary were chosen on the metal surface rather than on the surface of the oil film.
7.5.4.4 The Balance of Uncertainty

Undoubtedly heat transfer did occur to and from the vapour during the induction and discharge processes respectively, as temperature differences to promote such heat transfer existed. It was found that the test data could not support the extreme possibility that heat transfer effects, to the exclusion of leakage type effects, accounted for the actual displacement utilisation efficiency.

The leakage ratio values, calculated on the assumption of adiabatic suction and discharge processes, represented the maximum possible extents of the range of leakage type effects comprising: suction and discharge valve backflow and leakage, piston leakage, and solubility effects.

Further experimental work would clearly be required to fully resolve the question as to which effects, heat transfer type or leakage type, were more significant and whether the balance shifted over the range of speeds tested. The test results and the analyses helped to clarify the questions, at least.

7.5.5 COMPRESSION AND RE-EXPANSION PROCESSES

7.5.5.1 Basis of the Analyses

A polytropic compression or expansion process is described by equation 7.2.

\[ p V^n = a \]  

where

- \( p \) = cylinder pressure [Nm\(^2\)]
- \( V \) = cylinder volume [m\(^3\)]
- \( n \) = polytropic index
- \( a \) = constant

Taking the logarithm of both sides,

\[ \log(p) + n \log(V) = \log a = b \]  

where

- \( b \) = constant

When a polytropic process is plotted with logarithmic \( p \) and \( V \) axes, a straight line results and, from equation 7.3, its negative slope is the polytropic index.

In order to study the compression and re-expansion
processes of the compressor, therefore, diagrams were prepared of the log of the cylinder pressure plotted against the log of the cylinder volume, e.g. fig. 7-33. For the purpose of evaluating the overall polytropic index the compression process was considered to begin at the calculated point of suction closing and to end at the calculated point of discharge valve opening. The re-expansion process was considered to begin at the point of discharge closing and to end at the point of suction valve opening (the latter point was also considered the point of suction pressure equalisation, where the cylinder pressure equalled the mean suction line pressure: refer to section 7.4.1). The positions of valve opening and closing are marked by vertical lines on fig. 7-33. The following formulae were used to evaluate the polytropic indices:

\[
T_{lc} = \log \left( \frac{P_{od}}{P_{os}} \right) - \log \left( V_{cs} \right) \log \left( V_{cd} \right)
\]

\[
n_c = \log \left( \frac{P_{od}}{P_{os}} \right) - \log \left( V_{cd} \right)
\]

where

- \( n_c, n_e \) = polytropic indices for compression and re-expansion respectively
- \( p \) = cylinder pressure [Nm\(^{-2}\)]
- \( V \) = cylinder volume [m\(^3\)]

subscripts:
- os = at opening of suction valve
- od = at opening of discharge valve
- cs = at closing of suction valve
- cd = at closing of discharge valve

For an ideal gas the polytropic relationship (7.2) can be combined with the ideal gas equation to yield an expression for the polytropic index in terms of the initial and final pressures and temperatures as follows:

\[
n = \frac{\log \left( \frac{p_a}{p_s} \right)}{\log \left( \frac{p_a}{p_s} \right) - \log \left( \frac{T_{dv}}{T_s} \right)}
\]

where

- \( p_a \) = mean discharge pressure [Nm\(^{-2}\)]
- \( p_s \) = mean suction pressure [Nm\(^{-2}\)]
- \( T_{dv} \) = mean temperature measured close to the valve within the discharge plenum [K]
Ch. 7 Compressor Performance Characteristics

TEST RESULTS

BITZER IV COMPRESSOR

Ps = 2.19 bara
Ts = 4.73 deg. C
Refrigerant 12

Pd = 9.6 bara
RPM = 592.3

Disch. press. equalisation
Suction press. equalisation

Fig. 7-33 Diagram of log(p) versus log(V) for test number 49.
Values of the polytropic indices, based on the p-T data (treating the refrigerant as an ideal gas) were calculated (eqn. 7.6) using the temperatures entering and leaving the cylinder, as measured within the suction and discharge plenums (locations aa and bb respectively, fig. 6-1, ch. 6). As mentioned in section 6.4, ch. 6, these temperatures were only measured towards the end of the testing programme. However, straight lines drawn through the data points were used to extrapolate or interpolate (see figs. 7-14 and 7-15) and so polytropic indices based on p-T data were calculated for earlier tests also.

The values of the polytropic indices calculated from p-V and p-T measurements differed considerably. In the following sections the calculated values are first presented and then discussed.

7.5.5.2 Indices Based on p-V Data for Compression

It was found that the line representing the compression process on the log(p) vs. log(V) diagrams approximated to a straight line in all cases, indicating that the process was approximately polytropic. The value of the polytropic index varied from about 0.94 at 300 r.p.m. to about 1.04 at 600 r.p.m. and to about 1.02 at 900 r.p.m., as shown by the curve fitted to the scattered data points in fig. 7-34.

The polytropic indices, calculated from the p-V data, for the compression process are shown plotted against suction temperature in fig. 7-35. The values were close to unity at 0°C and at 28°C suction temperature, but rose to about 1.08 in the middle of this range.

7.5.5.3 Indices Based on p-V Data for Re-expansion

On the log/log diagrams, e.g. fig. 7-33, the line representing the re-expansion process showed a small amount of curvature, apparently indicating a changing polytropic index during the process. In the case of the diagram shown in fig. 7-33, the apparent polytropic index was about 0.77 at the start of the re-expansion process and about 1.08 at the end, with an
TEST RESULTS vs. SPEED

- \(x\) = Index Derived from \(p \& T\)
- \(+\) = Index Derived from \(p \& V\) (Compression)

Fig. 7-34 Polytropic indices based on \(p-T\) data and based on \(p-V\) data for the compression process versus compressor speed.

TEST RESULTS vs. SUCT. TEMP.

- \(x\) = Index Derived from \(p \& T\)
- \(+\) = Index Derived from \(p \& V\) (Compression)

Fig. 7-35 Polytropic indices based on \(p-T\) data and based on \(p-V\) data for the compression process versus suction temperature.
overall value of about 0.93. The calculated overall polytropic indices for the re-expansion processes are shown in figs. 7-36 and 7-37, versus speed and suction temperature respectively. The indices increased a little with increasing speed over the speed range and more noticeably with increasing suction temperature over its range. Over both ranges the values were, for the most part, below the isothermal index of unity.

7.5.5.4 Indices Based on p-T Data

These indices increased linearly with speed from about 1.15 at 300 r.p.m. to 1.17 at 900 r.p.m., fig. 7-34. The values were close to the adiabatic index of 1.18 and this suggested that the actual compression process approached an adiabatic process a little more closely as the speed increased.

Indices calculated from the p-T data showed a decrease from about 1.16 to about 1.14 with increasing suction temperature over the range 0°C to 28°C, fig. 7-35. This suggested that the compression process approached an adiabatic process less closely as the suction temperature rose.

7.5.5.5 Exclusion of Low Polytropic Index Values

Values of the polytropic index less than unity would, in the absence of leakage or vapour solubility effects, suggest an increase in gas temperature during the re-expansion process, or, a decrease during the compression process. This was ruled out due to the lack of suitable thermal reservoirs to provide or accept the necessary heat transfer. Even isothermal re-expansion and compression processes could be excluded on this basis. For example, using data from table 7-3 and assuming a closed system during re-expansion of the clearance mass: the refrigerant temperature before re-expansion could have been 137.5°C (assuming no leakage or backflow) or 99.6°C (assuming adiabatic vapour transfer from the cylinder). In either case no temperature rise could have occurred during re-expansion since the measured temperature of the cylinder was 52.3°C. Also, the re-expansion could not have been isothermal since this would have required a reservoir to accept heat transfer at the initial
Fig. 7-36 Polytropic index based on p-V data for the re-expansion process versus compressor speed.

Fig. 7-37 Polytropic index based on p-V data for the re-expansion process versus suction temperature.
7.5.5.6 Sensitivity to Zero Errors in Pressure and Volume

Sensitivity analyses were carried out on hypothetical isothermal compression and re-expansion processes between the test levels of suction and discharge pressure. The initial cylinder volume for each hypothetical process was approximately the same as the value in the actual compressor. Based on assumed pressure and volume tolerances the overall tolerances on the polytropic indices were evaluated and the effects on the curvature of the lines on the log/log diagrams were examined, see table 7-4.

For the re-expansion process:

- pressure tolerance: -0.5 bar to +0.5 bar
- resulting polytropic index tolerance: +0.14 to -0.10
- curvature: CW to CCW
- volume tolerance: -3 ml to +5 ml
- resulting polytropic index tolerance: -0.369 to + 0.52
- curvature: CW to CCW

For the compression process:

- pressure tolerance: -0.5 bar to +0.5 bar
- resulting polytropic index tolerance: +0.14 to -0.10
- curvature: CW to CCW
- volume tolerance: -5 ml to +5 ml
- resulting polytropic index tolerance: -0.06 to +0.07
- curvature: CW to CCW

where

- CW indicates clockwise curvature in the direction of increasing volume
- CCW indicates counterclockwise curvature in the direction of increasing volume

Table 7-4 Sensitivity of calculated polytropic indices to pressure and volume zero errors.

From table 7-4 it can be seen that the polytropic indices and the curvature of the lines representing compression and re-expansion were quite sensitive, in slope and curvature, to inaccuracies in the absolute values of the cylinder pressure and volume.
Ch. 7 Compressor Performance Characteristics

The curve representing the re-expansion process was particularly sensitive to the clearance volume. Special care was taken with this input, e.g. allowance was made for the groove around the suction valve port and the volume of the port itself. When the clearance volume had been calculated carefully from the geometry of the space an additional amount of about 11% was added to allow for chamfers on corners and the small space between the piston and cylinder above the first piston ring. This increased estimate of the clearance volume gave rise to an increase in the polytropic index for re-expansion, e.g. from 0.92 to 0.97, but did not significantly affect the curvature of the line on the log/log diagram nor did it significantly affect other results. It was more likely, therefore, that any significant errors in the volume when the piston was near the top of the cylinder were related to inaccuracies in the location of the top dead centre position.

7.5.5.7 Leakage and Plenum Pulsation Effects

The low values of the polytropic index calculated from the p-V data for the compression process may have been due in part to leakage past the piston and suction valve. Plenum pulsations may also have contributed to the low values. This latter effect would have meant that at least one of the pressures used in calculating the polytropic index was incorrect.

Leakage through the discharge valve could have accounted for a low overall polytropic index for the re-expansion process, as indeed could backflow caused by late closure of the discharge valve, if there were underpressure in the discharge plenum, due to pulsations, at the point of discharge valve closing.

7.5.5.8 Solubility Effects

The possibility, as suggested by Gosney [26] and already described in chapter 2, was considered: that a small part of the superheated vapour condensed and went into solution with lubricating oil on the cylinder walls during the compression stroke and re-evaporated, coming out of solution, during the re-expansion stroke.
Condensation of the refrigerant could not occur if it were pure since at all times the cylinder wall temperature was well above the saturation temperature of pure refrigerant. This was confirmed by measurements of the cylinder temperature (refer to figs. 7-14 and 7-15). The following thesis was proposed:

Lubricating oil in contact with the internal surfaces of the cylinder of a compressor is saturated with refrigerant at the mean temperature of the cylinder walls. The amount of refrigerant which can be dissolved in the oil, and which can be determined from Bambach’s solubility equations (Appendix B), depends on both temperature and pressure. At a given mean temperature the solubility is greater at the discharge pressure than at the suction pressure. Thus, vapour can evolve from the oil resident on the internal surfaces during re-expansion and can dissolve again during compression. This process of solution and evolution would involve heat transfer to and from the oil film. It may well be that the time available during expansion or compression does not permit the concentration of liquid refrigerant in the oil to vary between the theoretical equilibrium extremes, but, an involvement of a very small quantity of oil could explain the peculiarity in the apparent polytropic indices calculated from the p-V data wherein the indices for compression were generally higher than those for re-expansion.

Various values for the 'phantom mass' of refrigerant were tried in a computer program for analysis of the results (see table 7-5), on the basis of an assumption that the mass which evolved during re-expansion went back into solution during compression. An increase in the assumed phantom mass increased the calculated polytropic index for re-expansion and also the index for compression, but, the effect on the re-expansion index was much more pronounced. As an extension of this sensitivity analysis a phantom mass was calculated for each set of test results which made the overall polytropic indices equal for the compression and re-expansion processes. In the calculation of the value of the phantom mass the assumption of adiabatic vapour transfer was used and it was also assumed that the same thermodynamic state existed at the two points of suction pressure equalisation.

As it was realised that there were many uncertainties, e.g. in absolute pressure levels, leakage rates, plenum pulsations and solubility effects, which could have influenced the
calculated values of the polytropic indices for the compression and re-expansion processes and as there was no basis for assuming equal polytropic indices for both compression and re-expansion, it was decided not to present values of the calculated 'phantom mass' as results over the speed and temperature ranges. The results for test no. 49 (refer also to fig. 7-33), table 7-5, are given for illustrative purposes only. These results are not entirely consistent with the data for the polytropic indices presented in figs. 7-34 to 7-37 since the algorithm used in analysing the test results, of which the phantom mass analysis formed part, was not the final version.

<table>
<thead>
<tr>
<th>Phantom mass of refrigerant [mg]</th>
<th>polytropic index for compression</th>
<th>polytropic index for re-expansion</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.000</td>
<td>1.087</td>
<td>0.940</td>
</tr>
<tr>
<td>0.025</td>
<td>1.098</td>
<td>1.027</td>
</tr>
<tr>
<td>0.050</td>
<td>1.109</td>
<td>1.150</td>
</tr>
<tr>
<td>0.075</td>
<td>1.120</td>
<td>1.338</td>
</tr>
<tr>
<td>0.100</td>
<td>1.132</td>
<td>1.672</td>
</tr>
</tbody>
</table>

Table 7-5 Refrigerant phantom mass analysis.

It was concluded that solubility effects could significantly affect the apparent polytropic indices, but, due to measurement uncertainties could not be discriminated from other effects on the basis of the data which were available.

7.5.5.9 Conclusions Regarding Polytropic Indices

In view of the uncertainties mentioned it seemed unlikely that the compression and re-expansion processes were close to isothermal, as suggested by the p-V data. The p-T data, on the other hand, yielded values close to and a little below the
Ch. 7 Compressor Performance Characteristics

adiabatic index of 1.18. Given that the temperatures were measured outside the valves and that they represented averages of fluctuating quantities, the derived polytropic indices cannot be regarded as very accurate. Nonetheless, it was concluded that the compression process approached the adiabatic model, particularly at high speeds.

7.5.6 ATTRIBUTION OF LOSSES IN DISPLACEMENT UTILISATION EFFICIENCY

Based on the analyses which have been described, bar charts were constructed of decreasing displacement utilisation efficiency values. These are shown in figs. 7-38 and 7-39 for the extreme ends of the speed range tested and in figs. 7-40 and 7-41 for the extreme ends of the suction temperature test range. Each bar, identified by a letter, represents a displacement utilisation efficiency defined in a particular way, as detailed below:

A. Ideal compressor, no clearance volume.
B. Ideal compressor, with clearance volume, polytropic compression process with \( n = 1.18 \) (= ratio of specific heat values approx.).
C. Ideal compressor, with clearance volume, polytropic compression process, \( n = \) polytropic index calculated from pressure and temperature measurements.
D. Indicated volumetric efficiency.
E. Volumetric induction efficiency assuming adiabatic heat transfer (evaluated as described in section 7.5.4.2).
F. Actual displacement utilisation efficiency.

On the basis of the values shown in figs. 7-38 to 7-41 the causes of the losses in volumetric displacement utilisation were ascribed as follows:

7.5.6.1 Loss Due to Clearance Volume

Loss = A - C
= 7.1% at 300 r.p.m., 5°C suct. temp.
= 6.9% at 900 r.p.m., 5°C suct. temp.
= 7.0% at 600 r.p.m., 0°C suct. temp.
= 7.2% at 600 r.p.m., 30°C suct. temp.

These values would have been very slightly less if the compression and re-expansion processes had been closer to an adiabatic process, with a polytropic index of 1.18, in which case the loss would have been:
Fig. 7-38 Displacement utilisation efficiencies, as described in section 7.5.6, at 300 r.p.m.

Fig. 7-39 Displacement utilisation efficiencies, as described in section 7.5.6, at 900 r.p.m.
Ch. 7 Compressor Performance Characteristics

Fig. 7-40 Displacement utilisation efficiencies, as described in section 7.5.6, at 0°C suction temperature.

Fig. 7-41 Displacement utilisation efficiencies, as described in section 7.5.6, at 30°C suction temperature.
Ch. 7 Compressor Performance Characteristics

Loss = A - B = 6.8%.

7.5.6.2 Loss Due to Mass Addition During Re-expansion and Underpressure at the Bottom Dead Centre Position

Loss = C - D = 3.1% at 300 r.p.m., 5°C suet. temp.
= 9.9% at 900 r.p.m., 5°C suet. temp.
= 6.2% at 600 r.p.m., 0°C suet. temp.
= 2.4% at 600 r.p.m., 30°C suet. temp.

The above loss figures included contributions due to discharge valve leakage and backflow and any evolution of refrigerant from the resident oil. These factors determined the position of point 4 (fig. 2-1, ch. 2) on the indicator diagram. The position of point 1 on the diagram was a result of underpressure at the bottom dead centre position. Any effect of overpressure at the top dead centre position was negligible in comparison with the effects mentioned, as seen on the indicator diagrams.

The effects of underpressure could not be accurately quantified due to the fact that suction pressure equalisation apparently occurred before the bottom dead centre position in many of the tests. In these cases, under the assumptions made, the calculated underpressure loss ratio was zero. The values of underpressure loss ratio are shown in figs. 7-42 and 7-43. These losses were significant at higher compressor speeds and possibly also at mid range suction temperatures.

The overall loss under this heading increased with speed, possibly due to underpressure at the bottom dead centre position. It decreased with increasing suction temperature, possibly due to a decrease in evolution of vapour from liquid during the re-expansion stroke.

7.5.6.3 Loss Due to Suction Throttling and Heat Transfer Within the Suction Plenum, According to the 'Adiabatic Vapour Transfer' Assumption

Loss = D - E = 12.0% at 300 r.p.m., 5°C suet. temp.
= 5.3% at 900 r.p.m., 5°C suet. temp.
= 8.2% at 600 r.p.m., 0°C suet. temp.
= 7.2% at 600 r.p.m., 30°C suet. temp.

The values for the volumetric induction efficiency (E
Fig. 7-42 Underpressure loss ratio versus compressor speed.

Fig. 7-43 Underpressure loss ratio versus suction temperature.
above) resulted from the iterative analysis calculations described in appendix J. A rough breakdown of the losses under this heading was possible as outlined below. These were only rough estimates as leakage effects were ignored and ideal gas behaviour was assumed, whereas in the above figures the assumed leakage effects were taken into account in deriving the volumetric induction efficiency and real fluid properties of the refrigerant/oil mixture were used.

7.5.6.3.1 Loss Due to Heat Transfer Within the Suction Plenum

This was estimated using equation 2.14, ch. 2.

\[
\text{Loss} = \begin{array}{ll}
8.5\% & \text{at 300 r.p.m., 5°C suet. temp.} \\
6.0\% & \text{at 900 r.p.m., 5°C suet. temp.} \\
6.9\% & \text{at 600 r.p.m., 0°C suet. temp.} \\
6.3\% & \text{at 600 r.p.m., 30°C suet. temp.}
\end{array}
\]

The contribution to the loss in displacement utilisation efficiency due to heating within the suction plenum decreased with increasing speed, due to the shorter transit time through the plenum chambers. It decreased slightly with increasing suction temperature, due to the reducing temperature differential between the suction vapour and the internal compressor surfaces.

7.5.6.3.2 Loss Due to Suction Throttling and Re-compression

This loss was estimated using equation 2.12, ch. 2.

\[
\text{Loss} = \begin{array}{ll}
1.2\% & \text{at 300 r.p.m., 5°C suet. temp.} \\
3.1\% & \text{at 900 r.p.m., 5°C suet. temp.} \\
2.7\% & \text{at 600 r.p.m., 0°C suet. temp.} \\
2.7\% & \text{at 600 r.p.m., 30°C suet. temp.}
\end{array}
\]

This loss was due to the further increase in temperature, while assuming adiabatic vapour transfer, due to suction throttling and re-compression. It was related to the suction pumping work and so increased with compressor speed. It did not change significantly with suction temperature.

7.5.6.3.3 Reapportionment of Suction Plenum Heating and Suction Throttling Losses

It can be noted that the estimated losses presented in the previous sections do not add up to the overall values presented
in section 7.5.6.3. This is a good example of the difficulty in attempting discrete quantification of effects which are superimposed. As the overall loss figures were considered to be more accurate the component losses were reapportioned as follows:

Loss due to heat transfer in the suction plenum:
- = 10.5% at 300 r.p.m., 5°C suet. temp.
- = 3.5% at 900 r.p.m., 5°C suet. temp.
- = 5.9% at 600 r.p.m., 0°C suet. temp.
- = 5.0% at 600 r.p.m., 30°C suet. temp.

Loss due to suction throttling:
- = 1.5% at 300 r.p.m., 5°C suet. temp.
- = 1.8% at 900 r.p.m., 5°C suet. temp.
- = 2.3% at 600 r.p.m., 0°C suet. temp.
- = 2.2% at 600 r.p.m., 30°C suet. temp.

7.5.6.4 Loss Due to Heat Transfer During the Induction and Discharge Processes or Due to Backflow, Leakage, and Oil Solubility Effects Not Already Included

\[ \text{Loss} = E - F = 11.5\% \text{ at } 300 \text{ r.p.m., 5°C suet. temp.} \]
\[ = 12.9\% \text{ at } 900 \text{ r.p.m., 5°C suet. temp.} \]
\[ = 12.8\% \text{ at } 600 \text{ r.p.m., 0°C suet. temp.} \]
\[ = 9.0\% \text{ at } 600 \text{ r.p.m., 30°C suet. temp.} \]

These losses were less significant at higher suction temperatures. The available data did not allow a further breakdown of the losses, but, it was shown that heat transfer effects alone could not account for the values given.

7.5.6.5 Summary of Losses

The losses in displacement utilisation are summarised on the bar chart in fig. 7-44 for the two extremes of the speed range and in fig. 7-45 for the extremes of the suction temperature range.

7.6 UTILISATION OF SHAFT POWER

7.6.1 MECHANICAL EFFICIENCY

Mechanical efficiency was found to vary little with speed. It showed a small increase from about 92% to about 94% over the speed range (fig. 7-46) and did not vary with suction temperature (fig. 7-47). The friction power increased from about 56 W
Ch. 7 Compressor Performance Characteristics

AT 300 & 900 R.P.M.

- a. clearance volume
- b. re-exp. & mass addn. & underpress.
- c. suct. plenum heat transf.
- d. suct. throttling & re-comp.
- e. heat transf. & leakage & solub.

Fig. 7-44 Losses in displacement utilisation at the ends of the speed range. The left hand bar of each pair is at 300 r.p.m. and the right hand bar is at 900 r.p.m.

AT 0 & 30 deg. C
Suction Temperature

See above for descriptions of losses

Fig. 7-45 Losses in displacement utilisation at 0°C and 30°C suction temperature (left hand and right hand bar of each pair respectively).
Ch. 7 Compressor Performance Characteristics

Fig. 7-46 Mechanical efficiency versus compressor speed.

Fig. 7-47 Mechanical efficiency versus suction temperature.
at 300 r.p.m. to about 126 W at 900 r.p.m.

7.6.2 INDICATED POWER

The mean effective pressure, the suction effective pressure and the discharge effective pressure from the indicator diagrams were multiplied by the swept volume and the compressor speed to give the indicated power, the suction pumping power and the discharge pumping power respectively. All three quantities increased linearly with speed, as shown in fig. 7-48, and did not vary significantly with suction temperature, fig. 7-49.

7.6.3 INDICATED POWER UTILISATION EFFICIENCIES

The indicated isentropic efficiency $\eta_{is}$, the indicated rational efficiency, $\eta_{ir}$, and the indicated rational efficiency for compression and heat rejection, $\eta_{ircr}$ (eqn. 3-15, ch. 3), all showed small linear decreases with increased speed (fig. 7-50). The indicated rational efficiency was a little higher than the isentropic efficiency, while $\eta_{ircr}$ was considerably lower than both. All three characteristics versus speed had roughly the same slope.

The indicated rational efficiency for compression and heat rejection was not sensitive to the suction temperature, fig. 7-51. Both the indicated rational efficiency and the indicated isentropic efficiency increased with suction temperature, reflecting the increased worth in terms of the Second Law of Thermodynamics of the higher temperature discharge vapour.

The indicated rational efficiency, which had a value of about 60% in the middle of the speed range, is of particular significance in the context of heat pump applications of the compressor. 40% of the availability of the indicated power was lost, due mainly to the irreversible throttling processes at the suction valves and the consequent suction and discharge pumping work requirements. Part of the loss in availability would also have been due to the heat loss from the compressor to the surroundings and due to the fact that the heat transfer processes within the compressor were irreversible.

The approach adopted in presenting the indicated rational
Ch. 7 Compressor Performance Characteristics

**TEST RESULTS vs. SPEED**

- $\times$ = Indicated Power
- $+$ = Discharge Pumping Power
- $\circ$ = Suction Pumping Power

**Fig. 7-48** Indicated power, discharge pumping power and suction pumping power versus compressor speed.

**TEST RESULTS vs. SUCT. TEMP.**

- $\times$ = Indicated Power
- $+$ = Discharge Pumping Power
- $\circ$ = Suction Pumping Power

**Fig. 7-49** Indicated power, discharge pumping power and suction pumping power versus suction temperature.
TEST RESULTS vs. SPEED

Fig. 7-50 Three indicated efficiencies versus compressor speed.

TEST RESULTS vs. SUCT. TEMP.

Fig. 7-51 Three indicated efficiencies versus suction temperature.
efficiency as a performance parameter of the compressor had scope to trace the availability of the refrigerant throughout the compressor cycle, and thus attribute the overall loss in availability to specific quantified causes. This was outside the objectives of the present work and would have required better test data on the thermodynamic states of the refrigerant within the cylinder. The more traditional parameter, the isentropic efficiency, does not have scope for this type of analysis, even if the data were available.

In the context of refrigeration applications the values of \( E_{rr} \) showed that even if the compressor were combined with an ideal condenser, which de-superheated, condensed and subcooled the discharge vapour to the liquid state at the temperature of the surroundings, about 50% of the availability of the indicated power would be lost. This loss was due mainly to the factors mentioned above, but part was also due to the inappropriateness of the discharge pressure, as it was higher than the saturation pressure corresponding to the temperature of the surroundings.

7.6.4 NET HEAT TRANSFER FROM THE REFRIGERANT

Based on indicated power, the measured flow rate and the enthalpy values at entry and exit, the net rate of heat transfer from the refrigerant to the surroundings increased with speed from 0.19 kW to 0.43 kW over the range 300 r.p.m. to 900 r.p.m., fig. 7-52. As the flow rate increased almost in direct proportion to the speed over the same range, the net loss from the refrigerant per unit mass decreased with increasing speed.

The net heat loss rate from the refrigerant increased from 0.30 kW to 0.36 kW over the suction temperature range 0°C to 30°C, fig. 7-53. This also represented a increase in the net heat loss per unit mass of refrigerant passing through the compressor. The net heat transfer from the refrigerant plus the friction power would give the net heat transfer from the compressor to the surroundings. Based on the figures already quoted this varied from about 0.25 kW at 300 r.p.m. to about 0.56 kW at 900 r.p.m.
Fig. 7-52 Net heat rejection rate from the refrigerant passing through the compressor versus compressor speed.

Fig. 7-53 Net heat rejection rate from the refrigerant passing through the compressor versus suction temperature.
Ch. 7 Compressor Performance Characteristics

7.7 SUMMARY

The main compressor performance characteristics were determined with reasonable accuracy over a range of speeds and over a range of suction temperatures. These are presented in section 7.2. Data are presented on the oil concentration in the discharge vapour in section 7.3. Further testing would be required to confirm the exact nature of the oil concentration variation with compressor speed. Valve lift and pressure diagrams are presented and described in section 7.4. These were studied in considerable detail. The need for a precise absolute pressure datum for the cylinder pressure measurements is highlighted. Utilisation of volume displacement is analysed, quantified and discussed in section 7.5. There were difficulties in discriminating between leakage, heat transfer and solubility effects. Experimental data for temperatures within the cylinder would reduce these difficulties. Shaft power utilisation is quantified in section 7.6 and rational efficiency values are presented. It is pointed out that the rational efficiency approach has scope for more detailed analysis of the losses in shaft power utilisation.
Simulation techniques are widely used by researchers in the area of refrigerant compressors and by manufacturers of these machines in developing their products. A simulation model allows variations in compressor design parameters to be assessed without the need to carry out physical modifications. This can greatly reduce development time by reducing the number of prototypes which must be built. Simulation models can also predict how a compressor will perform over a wide range of operating conditions. A model used in this way can be validated by carrying out a relatively small number of test measurements. In seeking a better theoretical understanding of compressor characteristics researchers compare simulation results with test measurements. If a high degree of agreement is found this can be taken as evidence that the theoretical model is correct. Where there is lack of agreement, a detailed comparison of results may help to identify the deficiencies in the model and thus may lead to improved understanding of aspects of compressor performance.

A number of short courses were run, the first in 1972, at the Ray W. Herrick Laboratories of Purdue University, West Lafayette, Indiana, on compressor simulation. The course notes which were published provide a comprehensive guide to the techniques and include complete listings of simulation programs written in the FORTRAN language [55 to 60].

It was decided to apply the basic Purdue simulation program [56] to the Bitzer IV compressor over a range of operating speeds as a tool in establishing and explaining its performance characteristics. Some relatively minor changes were made to the original program listing, e.g. metric units were used rather than British units, step increments were described in degrees rather than radians and new input and output file structures were created. The FORTRAN language had been considerably enhanced since the program listing was published in 1974 (at that time data input was from punched cards) and some updating of the coding was done to take advantage of FORTRAN 77 features.
Ch. 8 Compressor Simulation

Additions were made to the program in order to evaluate parameters such as the mean effective pressure, the indicated power, the mass flow rate and the volumetric efficiency from the simulation results. A graphics program was also written to plot the results.

8.1 BASIS OF THE PURDUE SIMULATION MODEL
The Purdue model simulates the processes which occur within the compressor cylinder over a small number of revolutions for specified operating conditions and for a specified initial thermodynamic state of the charge. Calculations are carried out at small increments of crank rotation. After each interval the mass and thermodynamic state of the charge, the incremental work done on the piston, the suction and discharge valve displacements and the instantaneous mass flow rates through the valves are evaluated.

8.2 INPUTS TO THE SIMULATION PROGRAM
The inputs to the program are described below and details are given of the ways in which these were established. A full set of input data is included in appendix K.

8.2.1 GEOMETRIC PARAMETERS
These consisted of the crank and connecting rod radii, the piston diameter and the clearance volume.

8.2.2 WORKING FLUID PARAMETERS
The refrigerant was modelled as an ideal gas and was specified by its specific gas constant and the ratio of its specific heat capacities.

8.2.3 OPERATING CONDITIONS
1. Crank speed.
2. Suction plenum pressure specified as a function of crank angle. This was taken to be constant and was assumed to be the same as the suction pressure.
3. Discharge plenum pressure as a function of crank angle. This was taken to be constant and was assumed to be the same as the discharge pressure.
4. Suction plenum temperature. This was assumed to be the same as the suction temperature.

5. Discharge plenum temperature. This parameter was not strictly an operating condition, but rather, was a characteristic of the compressor. It was used within the model only when the rate of backflow through the discharge valve was being calculated. It was assumed to be the same as the discharge temperature and an estimate was used.

8.2.4 INITIAL CONDITIONS

The initial crank position and the initial temperature and pressure of the refrigerant were specified.

8.2.5 CALCULATION STEP PARAMETERS

The calculation step in degrees of crank angle and the total change in crank angle were specified.

8.2.6 POLYTROPIC INDEX

For an ideal gas and a reversible adiabatic compression process this would be equal to the ratio of the specific heats. It would be equal to unity for an isothermal compression process involving heat transfer from the refrigerant. Actual compression and re-expansion processes are not reversible and involve heat transfer at varying temperatures. Also, factors such as leakage or solubility of refrigerant in the lubricating oil within the cylinder may affect the apparent polytropic indices. In selecting a single polytropic index as an input to the simulation program a value should be chosen which would best represent the actual processes.

For the simulation results presented in this chapter a polytropic index of 1.1 was specified to take account of some heat transfer from the refrigerant during compression.

8.2.7 VALVE EFFECTIVE FLOW AREAS

The suction and discharge valves were modelled as simple orifices and were described by effective flow areas which depended on valve lift. Data were input for the normal and backflow directions of each valve as a function of lift.
Ch. 8 Compressor Simulation

For the simulation results presented in this chapter these data were determined experimentally [61] in steady flow tests carried out on a valve plate test rig based on one described by Soedel [55, pp.50-53].

Schwerzler and Hamilton [62 (1972)] described an analytical method of estimating the effective flow areas based on valve geometry and dimensions. It involved the representation of the actual valve geometry as a flow network of simple orifices and the use of assumed flow coefficients. This method was applied to the Bitzer compressor at an earlier stage of the work and yielded usable input data for the simulation program. The main drawback, however, was the subjective nature of the model and the assumed flow coefficients. Different effective flow area characteristics were obtained with different sets of reasonable assumptions.

8.2.8 VALVE EFFECTIVE FORCE AREAS

In the simulation model the force acting on a valve reed at any lift position was assumed to be equal to the product of the pressure difference across the valve and the effective force area.

The effective force area input data used for the results presented in this chapter were determined experimentally in steady flow tests on a valve plate test rig [61]. This, like the test rig for flow areas, was based on one described by Soedel [55, pp.59-62]. Ferreira and Driessen [63 (1986)] presented experimental data in dimensionless form for a wide range of flow conditions and for different valve geometries based on a circular port and a circular valve disk. These data did not apply directly to the valves of the Bitzer IV compressor, but, did corroborate minima which occurred on the effective force area versus lift characteristics of both the suction and discharge valves at a lift of about 1.7 mm, see fig. 8-1. This effect was due to the occurrence of low pressure regions underneath the valve reeds, due to the high velocity fluid stream passing between the reed and the seat.

Schwerzler and Hamilton [62] also described how the
effective force areas could be estimated. This method was used at an earlier stage of the work and, as was the case for the effective flow areas, gave usable data. However, the results depended on the particular set of reasonable assumptions made and did not predict the minima in the force area characteristics, such as the one shown in fig. 8-1.

![Effective force area versus lift](image)

**Fig. 8-1** Effective force area versus lift for the discharge valve of the Bitzer IV compressor, as determined on a valve plate test rig [61].

### 8.2.9 AREAS OF VALVE REED AND PORT ELEMENTS

In solving the equations for flow rates, forces and valve dynamics, the reeds and ports were each divided into a number of finite elements and values were input for the areas of these elements. The program structure required a corresponding port element for each reed element, even though some of the port elements had zero area.

The effective flow and force areas specified for the complete valves as inputs to the model were based on uniform lift, i.e. it was assumed that the valve reeds remained flat and parallel to the valve plate. The effective flow and force areas of each port element at any given lift value were assigned
within the model in the same proportion as the port element area over the total port area. This allowed the curvature and the nonparallel lift of the reed to be taken into account within the iteration scheme by calculating the effective areas of the port elements using the lift values of the reed elements.

8.2.10 **Natural Frequencies and Mode Shapes of Valve Reeds**

The program could handle up to two mode shapes for each valve reed. The density of the reed material and the reed thicknesses were input and from these values, together with the reed element areas, the masses of the reed elements were calculated within the program. The natural frequencies for each mode shape of each valve were also required as inputs. These values were determined experimentally [54] in a test fixture by causing the reed to vibrate at its natural frequencies. The test fixture was based on one described by Soedel [55, pp. 63-68].

The boundary conditions for the valve reed dynamic equations were taken into account in the simulation program by obtaining the solutions in terms of the natural mode shapes. The mode shapes were specified by inputting the relative lift of each valve reed element for each mode. These were determined experimentally [54] (in the same test fixture as was used to determine the natural frequencies) by measuring the displacement amplitude at points along the lengths of the valve reeds while they were subjected to excitation at the natural frequencies by an electromagnet.

For the simulation results presented in this chapter two mode shapes of the suction valve reed and one of the discharge valve reed were taken into account. The values used for the first mode natural frequencies were not those determined in the test fixture, but, were chosen to agree more closely with the frequencies estimated from experimental indicator diagrams (refer to table 7-2, ch. 7). The fact that the natural frequency values determined in the test fixture differed from those observed in the indicator diagrams was attributed to the fact that the actual mounting conditions were not exactly reproduced
8.2.11 DAMPING RATIOS

In the simulation program damping ratios were incorporated in the valve reed dynamic equations. No theoretical methods were found in the literature for evaluating these. The recommended procedure was that the values used as inputs to the simulation model should be adjusted to obtain the best possible agreement between the simulated valve lift diagrams and those measured on the compressor. For the results presented in this chapter a value of 0.12 was used for the two modes of the suction valve and a value of 2.0 was used for the single mode of the discharge valve.

8.3 NUMERICAL SOLUTION TECHNIQUE

The algorithm used in the simulation program involved the solution of a number of differential equations for each increment of crank rotation. These were:

A. Discharge valve dynamics equations, one for each vibration mode.

B. Suction valve dynamics equations, one for each vibration mode.

C. The cylinder mass equation:

\[
\frac{dm}{dt} = \dot{m}_s - \dot{m}_d \tag{8.1}
\]

where

- \(\frac{dm}{dt}\) = rate of increase in cylinder mass [kg/s]
- \(\dot{m}_s\) = mass flow rate in through suction valve [kg/s]
- \(\dot{m}_d\) = mass flow rate out through discharge valve [kg/s]

D. One of the valve mass flow rate equations.

The second order differential equations were each replaced by two first order equations and the Runge-Kutta technique was used to solve the full set of first order differential equations.

An outline flow chart of the Purdue simulation program is presented in fig. 8-2 and full details are available in references [55] and [56].
Ch. 8 Compressor Simulation

Start

Read input data.

Evaluate constants to be used in calculations.

Set initial conditions.

Runge-Kutta solution procedure for next time step begins.

Compute valve deflections from modal participation factors of previous solution point.

Call the "valve logic" subroutine to determine whether the valves are open or closed, whether flow is in the normal or backwards direction and if flow is choked.

If any one of the valves are open, look up applicable effective flow and force areas by calling an interpolation subroutine.

Call the "valve mass flow" subroutine if any one of the valves is open.

Compute new instantaneous values of crank angle, cylinder volume, mass in cylinder, cylinder pressure, temperature of charge.

Have the intermediate steps of the Runge Kutta solution procedure been completed?

Fig. 8-2 Continued on next page.

194
8.4 Outputs of the Simulation Program

8.4.1 Pressure and Valve Lift versus Crank Angle Diagrams

Figure 8-3 is a sample of the output data from the simulation program. It shows the valve displacements and the cylinder pressure plotted against cylinder volume. The mean effective pressure (MEP), the discharge effective pressure (DEP) and the suction effective pressure (SEP) are also shown on the diagram. This diagram can be compared with data obtained experimentally for similar operating condition which are presented in the same format in fig. 7-20, ch. 7.

The suction and discharge effective pressures from the simulation results (fig. 8-3) are in very good agreement with the experimental values (fig. 7-20, ch. 7). There is also very good agreement between the peak cylinder pressure in both cases. The mean effective pressure is lower, however, in the simulation results. The discharge valve lift is about twice as great in the simulation results as in the test data and the suction valve lift is also considerably greater. The shape of the peak on
SIMULATION RESULTS

BITZER IV COMPRESSOR

Ps = 2.19 bara  
Pd = 9.61 bara

Ts = 5.00 deg. C  
RPM = 600.

R = 68.76 J/kgK

Discharge Valve Lift

Cylinder Pressure /bara

MEP = 3.50

DEP = 0.59

SEP = 0.41

Suction Valve Lift

Fig. 8-3
the pressure/volume curve also differs between the two diagrams.

In the simulated results the second point of suction pressure equalisation did not occur before the bottom dead centre position was reached for any speed over the range 300 to 900 r.p.m., as was noted in the test results (section 7.5.2, ch. 7). The underpressure loss ratio varied from about 3.7% at 300 r.p.m. to about 9% at 900 r.p.m. (c.f. fig. 7-42, ch. 7)

8.4.2 PERFORMANCE CHARACTERISTICS OVER THE SPEED RANGE

Figures 8-4 to 8-9 illustrate the comparisons between the simulated and experimental results over the speed range.

8.4.2.1 Mass Flow Rate

The simulated mass flow rates, fig. 8-4, were higher than the test values. Effects such as leakage, heat transfer and oil solubility were not taken into account in the simulation model. The higher mass flow rates in the simulation results would account to some extent for the fact that the simulated valve lifts were also higher.

8.4.2.2 Indicated Power

The simulated values of indicated power, fig. 8-5, were a little lower than the measured values.

8.4.2.3 Discharge Temperature

The simulation program predicted discharge temperatures which were in the order of 40 K lower than the measured values, fig. 8-6. This was largely due to the low polytropic index which was used in the simulation model. Yet, when higher values were used the agreement between the indicator diagrams was not as good.

8.4.2.4 Indicated Volumetric Efficiency

Remarkably good agreement was achieved between the simulated data for volumetric efficiency and the measured indicated volumetric efficiency, fig. 8-7. The actual displacement utilisation efficiency values were considerably lower due to
Ch. 8 Compressor Simulation

**Fig. 8-4** Comparison between simulated and measured values of refrigerant mass flow rate.

**Fig. 8-5** Comparison between simulated and measured values of indicated power.
Ch. 8 Compressor Simulation

**Fig. 8-6** Comparison between simulated and measured values of discharge temperature.

**Fig. 8-7** Comparison between simulated and measured values of indicated volumetric efficiency.
Fig. 8-8 Comparison between simulated and measured values of mean effective pressure and suction effective pressure.

Fig. 8-9 Comparison between simulated and measured values of the discharge effective pressure.
effects discussed in chapter 7, section 7.5. With the exception of backflow, the simulation model did not take account of these effects.

8.4.2.5 Effective Pressures from the Indicator Diagrams

The mean effective pressures predicted by the simulation program were a little lower than those measured experimentally, but, agreement between the simulated and measured suction and discharge effective pressures was quite good, figs. 8-8 and 8-9.

8.5 COMMENTS

The simulation program was a success in modelling the general nature of performance variations over the speed range. Heat transfer effects were not taken into account within the model, except in the input of a single number to represent the polytropic index for both the compression and re-expansion processes. It was not surprising, therefore, that the simulation results did not accurately represent the measured discharge temperatures. In the results presented in this chapter the magnitudes of the simulated valve displacements were greater than the measured values. This may have been due in part to experimental limitations in obtaining the input data and was also due to the fact that the exact mounting arrangements of the valves were not modelled. It was found that the model could be made to agree well with measured indicator diagrams and to predict the shaft power. The level of agreement did, however, depend on the careful adjustment of the input parameters for the polytropic index and the valve damping ratios.
Chapter 9

CONCLUSIONS

The objectives of the work for this PhD thesis were to critically re-examine the ways in which the performance of a refrigerant compressor is evaluated and described and to find out which factors influence its characteristics. These objectives were met.

A load stand for testing compressors was developed and analysed. It made use of throttle valves, with condensation of part of the flow and a mixing process, for control of the operating conditions. The details presented in chapters 4, 5 and 6 can be regarded as an information resource and a set of guidelines for the design of similar load stands.

A fresh look was taken at the utilisation of volume displacement and of shaft power. Many aspects of both topics were considered, but, it was necessary to place limits on the scope of the work and, inevitably, the depth of examination varied somewhat.

9.1 THE LOAD STAND

The load stand described in this thesis is not presented as the ideal load stand for testing refrigerant compressors. It was an evolving product of a learning process. It fulfilled its purpose, in that it facilitated the testing of a compressor over a range of operating conditions.

Control of the operating conditions was achieved by means of throttle valves only and this was considered an important step, as it demonstrated the feasibility of load stand control using purely mechanical actuation. No input of energy was required other than the shaft power to drive the compressor. Full automation of the compressor testing process would be possible by motorising the valves and setting up a suitable microcomputer based control system.

A detailed steady state computer analysis of the complete load stand over a full range of operating conditions was undertaken, using real fluid refrigerant properties. This served a number of purposes as follows:
Ch. 9 Conclusions

1. It confirmed that the circuit arrangement could produce any desired set of operating conditions and quantified the proportions of the total flow which would be condensed and bypassed in each case.

2. By assuming an ideal compressor of known clearance ratio and displacement rate it provided design data for use in selecting the heat exchanger.

3. It also provided design data which enabled suitable liquid and vapour throttle valves to be selected. In particular, it allowed a fine liquid metering valve to be selected which would permit the desired precision to be attained in the control of the suction superheat.

4. It predicted the required settings of the throttle valves to achieve any set of operating conditions over the full range.

5. It allowed different control strategies to be assessed and provided an indication of where any choking at valves or restrictions within the circuit occurred. It also indicated impossible combinations of control valve settings and operating conditions.

The suction and discharge pressures of the compressor were readily controlled by means of the throttle valves (A and D), fig. 6-1, chapter 6. The suction temperature responded to very fine adjustments of valves Cc and Ct for a fixed setting of valve B, but, there was a noticeable time delay and compensatory adjustments of valves A and D were necessary.

The final version of the load stand, as described in this thesis, incorporated a large suction side pulsation damping chamber and a baffle plate. This was required to smoothen the flow through the measurement orifice in order that the flow rate could be calculated from the differential pressure measured by a mercury manometer. The volume and mass of the damping chamber added considerably to the 'thermal inertia' of the plant. This increased the time taken to reach steady state operating conditions and caused a thermal time lag in the suction line whenever adjustments were made to the control valves. An earlier version of the load stand, which did not have the pulsation damping devices, had the advantage of a more rapid response to control adjustments.

The load stand did not allow a low degree of suction superheat (below 10 K) to be maintained. This shortcoming may have been related to the high heat capacity within the suction line, downstream of the flow mixing point. Also, at low
suction superheat values, there appeared to be either incomplete mixing, or, separation of liquid from the vapour after mixing, and response to adjustments in valves \( C_e \) and \( C_f \) was very slow. Further work and plant modifications (perhaps involving a reduction in the suction line diameter as well as the elimination of the pulsation damping chamber) would have been necessary to solve the problems and allow operation with low suction superheat values.

The electronic frequency inverter performed very satisfactorily in allowing the speed to be set at any desired value over the operating range of the compressor.

9.2 TESTING METHODS

The mass flow rate was determined by means of an orifice plate in the suction line and, independently with lower accuracy, by a heat balance involving the heat exchanger and the mixing process. Reasonable agreement was obtained between the methods, the maximum variation between the two being 4.5%. The use of a transducer to measure the dynamic differential pressure across the orifice was found to be a feasible method of measuring the average flow rate when pulsations were present (see section 6.6.1.2, ch. 6). The final version of the test stand relied on pulsation damping and the static measurement of the differential pressure.

The lack of a fixed datum for the pressure measurements made within the cylinder by means of the piezoelectric transducer led to difficulties and uncertainties in analysing the processes which occurred within the cylinder. Dynamic measurement of the pressure pulsations within the suction and discharge plenums would also have been useful.

It had been hoped to derive the mean temperature of the charge within the cylinder throughout the cycle on the basis of the measured cylinder pressure, the volume derived from the crank position (assuming constant rotational speed), and the externally measured mass flow per cycle. This was not feasible due to the following factors:

1. there was uncertainty as to the absolute level of the
205

Ch. 9 Conclusions

cylinder pressure

2. leakage and backflow rates were not known and so the mass of fluid induced into the cylinder could not be directly related to the externally measured mass flow rate

3. the extent to which the refrigerant dissolved and re-emerged from solution with the lubricating oil was not known.

Other workers [44, 64] have measured dynamic temperatures within the cylinders of reciprocating machines and, clearly, such measurements would be desirable.

There was particular difficulty in distinguishing between leakage type effects and heat transfer effects. In chapter 7, section 7.5.4, the uncertainties in the quantification of displacement utilisation losses due to these effects, resulting from measurement limitations, are highlighted. Similar difficulties would be encountered in attempting a detailed breakdown of shaft power availability losses. This emphasises the need for accurate experimental data on the temperature and pressure of the fluid within the cylinder throughout the cycle.

The method described in chapter 6, section 6.7, for the measurement of oil concentration in the refrigerant liquid is an improvement on the technique described in the ASHRAE standard [52]. It requires only a very small sample to be taken and, since the sample is vented to the low pressure side of the system, only a minute quantity of material is permanently lost for each measurement. This technique could be applied in refrigeration equipment in general, the sample being taken from the liquid line before the expansion valve and vented to the compressor suction line.

It was concluded that the particular type of differential pressure transducer which was used across the orifice plate was not suitable for use in refrigeration systems where a sudden drop in line pressure could occur.

It was found useful to install a master gauge and a manifolding system so that the calibrations of the pressure transducers could be checked frequently and without difficulty.
The mass flow rate and the indicated power varied approximately in direct proportion to compressor speed over the range 300 to 900 r.p.m. Shaft power (which was only a little higher than the indicated power) and the discharge gas temperature varied linearly with speed. The measured oil concentration in the discharge vapour was lowest at the low end of the speed range and reached a maximum near the middle of the range.

The mass flow rate and the shaft power were almost insensitive to suction temperature over a range of suction superheat values from 10 K to 38 K at a constant speed of 600 r.p.m. Over the same suction superheat range the discharge temperature increased by 23.8 K and the oil concentration in the discharge vapour showed a moderate decrease (from about 0.3% to 0.25%).

The displacement utilisation efficiency was found to remain almost constant at about 66% over the speed range. It showed an increase of about 8% (from about 66% to about 74%) over the range of suction superheat values from 10 K to 40 K.

Mechanical efficiency, at about 93%, varied little with speed or suction superheat. The indicated rational efficiency, at about 61%, showed only a slight decrease with increased speed. It increased moderately with increased suction superheat (from about 60% at 10 K to about 65% at 40 K). The indicated rational efficiency for compression and heat rejection, at about 47%, showed only a slight decrease with increased speed and was insensitive to suction superheat.

On the basis of these results the Bitzer IV compressor was considered suitable for operation over the speed range 300 to 900 r.p.m., with the proviso that some modification of the lubrication system might be required to ensure adequate oil distribution at low speeds.

The performance figures indicated that for both heat pump and refrigeration applications there was scope for improvement in the utilisation of volume displacement and of shaft power, by improved design.
9.4 IMPLICATIONS FOR OVERALL PLANT PERFORMANCE

The characteristics of the compressor are good for operation over a range of speeds since indicated power and mass flow rate vary in proportion to speed. In a refrigeration or heat pump plant, operated at variable speed and with fixed evaporating and condensing temperatures, the heating and cooling effects would vary approximately in proportion to speed (the characteristics would be similar to those shown in fig. 1-2, ch. 1). Also, for fixed evaporating and condensing temperatures, the rational efficiency or the c.o.p. of the entire plant would be almost insensitive to compressor speed over the range 300 to 900 r.p.m., based on the compressor's characteristics. However, as the temperature differences in the evaporator and condenser of an actual plant would be expected to vary in proportion to their heat transfer rates, these differences would be smaller at the low end of the speed range than at the high end. In consequence, the evaporating and condensing temperatures of an actual plant would be closer together at low speeds than at high speeds. The rational efficiency or the c.o.p. of the plant would therefore be highest at 300 r.p.m. and would decrease linearly with increasing speed. Speed modulation to match the load would ensure that the instantaneous rational efficiency of the overall plant was always as high as possible.

9.5 FACTORS WHICH INFLUENCE THE UTILISATION OF VOLUME DISPLACEMENT

In chapter 2, existing theories for the analysis of volumetric displacement utilisation are extended and a number of new parameters are defined. A comprehensive framework is provided for the discrete quantification of losses in displacement utilisation due to

(1) re-expansion of the clearance mass
(2) throttling and re-compression of the vapour during induction
(3) underpressure at the bottom dead centre position
(4) heat transfer to the vapour within the suction plenum and within the cylinder before the end of the induction process
(5) cyclic solubility of the refrigerant in the lubricating oil within the cylinder.
Ch. 9 Conclusions

(6) backflow through the valves due to late closure
(7) leakage through the valves or past the piston

It is shown that the parameters which quantify all of these
effects can be combined to give a displacement utilisation
efficiency ($\eta_{\text{dis}}$), which should approximate to the experimentally measured value. In addition, the relationship between
the indicated volumetric efficiency and the value for an ideal
compressor of known clearance volume is clarified.

These theories were applied, within the limitations of the
measurements which were made, in evaluating the compressor which
was tested. Results of the analyses are presented in chapter 7
and are summarised in figs. 7-44 and 7-45. Due to measurement
limitations it was not possible to distinguish fully between
heat transfer, leakage and solubility effects within the cylin-
der. Together these accounted for losses amounting to 11.5% of
the displacement rate at 300 r.p.m. and 12.9% at 900 r.p.m. It
was concluded, however, that heat transfer effects alone could
not account for these figures.

The question as to whether oil solubility effects within
the cylinder were significant or not was not fully resolved.
It was shown, however, that a 'phantom mass' of refrigerant
which dissolved in the lubricating oil within the cylinder over
part of the cycle and re-emerged over the remainder could affect
the displacement utilisation adversely, as any vapour release
from solution would occur mainly during the re-expansion and
induction processes. This pointed to the need for specific
experimental and theoretical work in this area.

9.6 FACTORS WHICH INFLUENCE THE UTILISATION OF
SHAFT POWER

The use of rational efficiencies based on the Second Law of
Thermodynamics is advocated to quantify the effectiveness of
shaft power utilisation. The 'rational efficiency' and the
'rational efficiency for compression and heat rejection' are
described and justified in chapter 3 and experimental values are
given in chapter 7. This concept, if taken to its conclusion,
would involve the attribution of losses in shaft power
availability to specific causes, such as heat transfer within the cylinder, suction valve throttling, etc. The detailed application of availability theory to the internal processes of refrigerant compressors was not addressed as part of the work for this thesis, however. Moreover, the experimental data did not permit such detailed analysis, with accuracy, due mainly to the lack of definitive temperature data for the fluid within the cylinder throughout the cycle. The consideration of rational efficiencies in section 3.4, chapter 3, is put forward as a starting point for further work which would trace the flows of thermodynamic availability within the compressor.

Suction and discharge pumping power quantities and power loss due to friction (as quantified in the mechanical efficiency) are presented in chapter 7, but, it should be emphasised, these do not correspond exactly with availability losses since some of the 'lost work' goes to increase the availability of the refrigerant. The suction and the discharge pumping power requirements both increased as a proportion of total indicated power as the speed increased and these effects would tend to cause a decrease in indicated rational efficiency with increased speed. However, the negative effects were cancelled out somewhat by the fact that leakage type effects (including backflow and solubility effects) and heat transfer effects were less significant at higher speeds.

Heat loss occurred from the refrigerant as it passed through the compressor at a rate which increased from about 190 W at 300 r.p.m. to about 425 W at 900 r.p.m. (fig. 7-52). Heat transfer occurred to the refrigerant within the suction plenum and from the refrigerant, at a considerably higher rate, in the discharge plenum (fig. 7-27). Combining the latter two effects, the net heat rejection rate which occurred outside the cylinder varied from about 150 W at 300 r.p.m. to about 275 W at 900 r.p.m. Consideration of these results leads to the conclusion that the greater part of the net heat rejection from the refrigerant occurred outside the cylinder. The rate of heat rejection which occurred within the cylinder can be estimated from the results presented. It varies from about 40 W at 300
Ch. 9 Conclusions

r.p.m. to 150 W at 900 r.p.m. These figures should not be regarded as accurate (due to the limitations of the temperature measurements near the valves), but, they are estimates of the magnitudes of the effects involved.

It was not found possible, on the basis of the test data, to evaluate heat transfer quantities between different points within the cycle.

The compression process within the cylinder was found to be polytropic, but, the polytropic index evaluated on the basis of pressures and volumes was less than that evaluated on the basis of pressures and temperatures (using thermocouples within the suction and discharge plenums near the valves). The latter method of evaluating the polytropic indices was considered more reliable (although not precise, due to the limitations of the temperature measurements near the valves) and on this basis it was concluded that the compression process within the cylinder approached the adiabatic model, particularly at high speeds.

Solubility effects, if present within the cylinder, could not be distinguished from other effects on the basis of the test measurements. It is pointed out in section 7.5.5.8 that a 'phantom mass' of refrigerant could change the apparent polytropic indices for the compression and re-expansion processes. This would influence the indicated power. Also, any influence of solubility effects on the displacement utilisation efficiency would affect the indicated power as well since, for example, the discharge pumping work per cycle depends on the quantity of fluid discharged.

9.7 COMPRESSOR SIMULATION

The Purdue simulation program correctly modelled the nature of compressor performance variations over the range of speeds. It was also possible to obtain good agreement between the simulated and measured indicator and valve lift diagrams by adjustments to the valve effective flow and force areas, the damping coefficients and the polytropic index. Simplified analytical methods of estimating the effective flow and force areas did not produce definitive data. Steady flow tests on the valve reeds
Ch. 9 Conclusions

to measure these parameters and the natural frequencies also failed to give values which produced good agreement between simulated and measured valve lift and pressure diagrams, although more accurate tests might improve the comparison considerably. Effects such as heat transfer, solubility and leakage were not incorporated in the model and, therefore, it provided no information in these areas. These aspects could be incorporated into the model, but, this would prove useful only if accompanied by appropriate experimental work for validation of the results.

9.8 CONCLUSION

In this thesis the evaluation of refrigerant compressors has been critically re-examined. This has involved a review of relevant literature, extension of the theoretical base for the quantification and description of characteristics and performance, the design, implementation and development of a compressor test rig and methodology, compressor testing and detailed analysis of the results, and the evaluation of a simulation program. Computer based techniques have been applied extensively to: the thermodynamic property evaluation of a refrigerant and of refrigerant oil mixtures; load stand simulation and design; calibration, data acquisition and software led testing; analysis of results including conjectural analysis; and compressor simulation.

New ways have been proposed for quantifying and describing the two main aspects of compressor performance, i.e. the utilisation of volume displacement and the utilisation of shaft power. These are based on a new and deeper understanding of the underlying influences. Their use will lead to improved compressor design, since depth of understanding is the basis of ingenuity.
REFERENCES


References


References


References


36. ASHRAE standard 23-78. 1978. "Methods of testing for rating positive displacement refrigerant compressors".


References


49. British Standard 1042: Section 1.2: 1984: Specification for square edged orifice plates and nozzles (with drain holes, in pipes below 50mm diameter, as inlet and outlet devices) and other orifice plates with Borda inlets.


References


APPENDIX A

COMPRESSOR SPECIFICATIONS

Type: Bitzer IV
Motor pulley dia: 180 mm
Compressor speed (with motor speed of 1450 r.p.m.): 670 r.p.m.
Cubic capacity: 13.33 m³/h
Cylinders: number: 2
           bore/stroke: 65 mm/ 50 mm
Oil charge: 1.5 litres
Weight: 45.5 kg
V-belts: number x profile to DIN 2215 2 x 17

Connections:
SL  Suction line: screwed pipe connection with solder tailpiece 22 mm
DL  Discharge line: screw conn. (flare) 18 mm
1  Discharge side: 1/8"-27 NPTF
3  Oil filling: 1/8"-27 NPTF
4  Oil drain: 1/8"-27 NPTF
5  Oil return (suction side): 1/8"-27 NPTF
10 Pressure gauge, discharge valve: 7/16"-20 UNF
11 Pressure gauge, suction valve: 7/16"-20 UNF

Dimensions/mm:
A  Length: 310
B  Width: 275
C  Height: 410
D x E x dia. fastening holes: 220x165x11
F  Flywheel dia.: 380
G  250 L 92 O 55
H  200 M 395 P 290
I  100 N 255 R 115
K  5

Source: product catalogue: "Open Compressors 0 to VII", Bitzer Kühlmaschinenbau GmbH & Co. KG.
Bambach's equations for the solubility of R-12 in paraffinic oil are as follows:

for temperatures < 0°C:

\[
\log_{10} p = a - b w^{-0.5} - \frac{c - d w^{-0.5}}{T} = A \quad (B.1)
\]

for temperatures > 0°C:

\[
\log_{10} p = A - (T - 273.16) [e (w - 0.6)^2 - f] \quad (B.2)
\]

where

- \( w \) = mass fraction of refrigerant in the refrigerant/oil mixture
- \( p \) = pressure [bara]
- \( T \) = absolute temperature [K]
- \( a \) = 4.9972
- \( b \) = 0.558
- \( c \) = 1177.67
- \( d \) = 98.753
- \( e \) = 0.002338
- \( f \) = 0.000075

The above equations were taken from Hughes et al. [17] & [19], who referred to the original source, Bambach [15]. Hughes et al. also listed the equations in [18], but, in this case constant 'e' was incorrect as it contained an extra zero after the decimal point. The equations were also listed by Cooper and Mount [16] and by Spauschus [14], where the value of constant 'a' was slightly different to account for pressure units of [kg/cm²].
Applying the energy equation to the process which takes place between points 4 and 1 (fig. 2-1, ch. 2),

\[ Q = m_1 u_1 - m_4 u_4 - (m_1 - m_4) h_s + \int_{V_4}^{V_1} p \, dV \]  
(C.1)

where

\[ Q = \text{heat transfer to the refrigerant [J]} \]
\[ u = \text{specific internal energy [J/kg]} \]
\[ h = \text{specific enthalpy [J/kg]} \]
\[ m = \text{mass of refrigerant present in cylinder [kg]} \]
\[ V = \text{instantaneous cylinder volume [m}^3\text{]} \]
\[ p = \text{pressure [N/m}^2\text{]} \]
\[ v = \text{specific volume [m}^3/\text{kg]} \]
\[ p_s = \text{subscript s indicates a property of the refrigerant in the suction pipe.} \]

But \( p_1 = p_4 = p_s \), and therefore

\[ Q = m_1 h_1 - m_4 h_4 - (m_1 - m_4) h_s - p_s \int_{V_4}^{V_1} (1 - p/p_s) \, dV \]  
(C.2)

Also

\[ W_{si} = p_s \int_{V_4}^{V_1} (1 - p/p_s) \, dV \]  
(C.3)

where

\[ W_{si} = \text{indicated suction work [J]} \]
\[ = \text{lower crosshatched area in fig 2-1, ch.2.} \]

Therefore,

\[ Q = m_1 h_1 - m_4 h_4 - (m_1 - m_4) h_s - W_{si} \]  
(C.4)

Since, for an ideal gas, \( \Delta h = c_p \Delta T \)

\[ Q + W_{si} = m_1 c_p T_1 - m_4 c_p T_4 - (m_1 - m_4) c_p T_s \]  
(C.5)

where

\[ c_p = \text{specific heat at constant pressure [J/kgK]} \]
\[ T = \text{absolute temperature [K]} \]

Also,
\[ T = pv/R \]

where

\[ R = \text{specific gas constant [J/kgK]} \]
Therefore,

\[ Q + W_{s1} = \frac{m_1 p_1 V_1}{C_p / R} - m_4 p_4 V_4 \frac{C_p / R}{(m_1 - m_4) C_p T_s} \]

Since \( C_p / R = \frac{\gamma}{(\gamma - 1)} \) and \( V = mv \)

\[ Q + W_{s1} = \left(\frac{\gamma}{(\gamma - 1)}\right) (p_1 V_1 - p_4 V_4) - (m_1 - m_4) C_p T_s \]

where

\( \gamma \) = ratio of specific heat capacities

\[ Q + W_{s1} = \left(\frac{\gamma}{(\gamma - 1)}\right) \frac{p_s}{C_p T_s} \frac{\gamma}{(V_1 - V_4)} - \frac{Q + W_{s1}}{C_p T_s} \]  \hspace{1cm} (C.6)

\[ \eta_{vstq} = \frac{V_s (m_1 - m_4)}{V_{sw}} \]  \hspace{1cm} (by definition)  \hspace{1cm} (C.7)

\[ = p_s V_s \frac{\gamma}{C_p T_s} \frac{V_1 - V_4}{(V_1 - V_4)} - \frac{Q + W_{s1}}{C_p V_{sw} T_s / V_s} \]

\[ = V_{sw} \frac{V_1 - V_4}{V_{sw} p_s C_p / R} - \frac{Q + W_{s1}}{\gamma} \]  \hspace{1cm} (C.8)

where

\( \eta_{vstq} \) = volumetric efficiency taking account of suction vapour throttling and heat pick up

\( \eta_{vi} \) = indicated volumetric efficiency

DERIVATION OF THE ADIABATIC SUCTION TEMPERATURE AFTER INDUCTION

In the definition of the adiabatic temperature after induction it is assumed that the actual indicator diagram applies for a hypothetical adiabatic induction process.

Equation C.5 can be applied from a point where the refrigerant enters the suction valve to point 1 on the indicator diagram.

In this case \( T_s \), the temperature in the suction pipe, is replaced by \( T_{sv} \), the mean temperature at entry to the suction valve.

\[ Q_{cy1} + W_{s1} = m_1 C_p T_1 - m_4 C_p T_4 - (m_1 - m_4) C_p T_{sv} \]  \hspace{1cm} (C.9)

where

\( Q_{cy1} \) = heat transfer to the refrigerant within the cylinder during the induction stroke [J]

\( T_{sv} \) = the mean temperature of the refrigerant entering the suction valve [K]
APPENDIX C

From equation C.9, if there is no heat transfer, \( Q_{c y1} = 0 \) and

\[
m_{1a} c_p T_{1a} - m_4 c_p T_4 = W_{si} + (m_1 - m_{1a}) c_p T_{svv}
\]

(C.10)

where

- \( T_{1a} \) = the adiabatic suction temperature after induction [K]
- \( m_{1a} \) = the mass within the cylinder at point 1 on the indicator diagram after adiabatic induction [kg]

But \( m = pV/RT \) and so

\[
p_s V_1 \frac{c_p}{R} - p_s V_4 \frac{c_p}{R} = W_{si} + p_s \frac{c_p}{R} \left( \frac{V_1}{T_{1a}} - \frac{V_4}{T_4} \right) T_{svv}
\]

Also \( c_p/R = \gamma/(\gamma - 1) \) and therefore

\[
p_s \frac{\gamma}{\gamma - 1} (V_1 - V_4) = W_{si} + p_s \frac{\gamma}{\gamma - 1} \left( \frac{V_1}{T_{1a}} - \frac{V_4}{T_4} \right) T_{svv}
\]

\[
(V_1 - V_4) \frac{1}{T_{svv}} - \frac{W_{si}}{T_{svv} p_s} \frac{\gamma}{\gamma - 1} = \frac{V_1}{T_{1a}} - \frac{V_4}{T_4}
\]

\[
\frac{V_1}{T_{1a}} = \frac{V_4}{T_4} + \frac{V_1 - V_4}{T_{svv}} - \frac{W_{si}}{T_{svv} p_s} \frac{\gamma}{\gamma - 1}
\]

\[
T_{1a} = \frac{V_4}{T_4} \frac{V_1 - V_4}{T_{svv}} - \frac{W_{si}}{T_{svv} p_s} \frac{\gamma}{\gamma - 1} \frac{1}{T_{svv}}
\]

(Evaluation of the heat transfer to the suction vapour within the cylinder)

From equations C.9 and C.10

\[
Q_{c y1} = m_1 c_p T_1 - m_{1a} c_p T_{1a} - (m_1 - m_{1a}) c_p T_{svv}
\]

But \( m = pV/RT \) and so

\[
Q_{c y1} = p_s V_1 \frac{c_p}{R} - p_s V_4 \frac{c_p}{R} - \left( p_s V_1 \frac{c_p}{T_1} - p_s V_4 \frac{c_p}{T_{1a}} \right) \frac{1}{T_{svv}}
\]

\[
= \frac{\gamma}{\gamma - 1} p_s V_1 T_{svv} \left( \frac{1}{T_{1a}} - \frac{1}{T_4} \right)
\]

(C.12)
DERIVATION OF 'TIlE DAMPING COEFFICIENT AND NATURAL FREQUENCY FOR A HEILHO1TZ RESONATOR

Note 1: all symbols have the same meanings as in Ch. 4, section 4.6.1, under the heading 'DYNAMIC PRESSURE MEASUREMENT'.

Fig. D-1 Model of a Helmholtz resonator consisting of a volume $V_s$ and a passage of length $L_p$ and radius $r_p$.

Note 2: effective passage length = $L_p + \pi r_p/4$

Effective mass of element in passage,
$$m_p = \pi r_p^2(L_p + \pi r_p/4)/v$$  \hspace{1cm} (D.1)

Pressure force on element when it is displaced by distance $x$,
$$F_p = \pi r_p^2(-\delta p/\delta v)\delta v = \pi r_p^2(-\delta p/\delta v)\pi r_p^2 x/m_s$$

where
$$\delta p = \text{increase in pressure within the volume [N/m}^2]\text{]}$$
$$\delta v = \text{increase in specific volume within the space [m}^3/\text{kg}\text{]}$$
$$m_s = \text{mass within volume } V_s [\text{kg}] = V_s/v$$

But, sonic velocity, $c = \sqrt{(-v^2(\delta p/\delta v)_s)}$

where  
subscript $s$ indicates an isentropic process.

Therefore, if the pressure disturbance is propagated reversibly and adiabatically,
$$\delta p/\delta v = -c^2/v^2$$

Hence, pressure force, $F_p = \pi r_p^2(c^2/v^2)(\pi r_p^2/m_s)x$
$$= \pi^2 r_p^4 c^2 x/v V_s$$  \hspace{1cm} (D.2)

Viscous shear force, $F_v = 8\mu(1 + \pi r_p/4)x$  \hspace{1cm} (D.3)
(from the Hagan Poiseuille formula for viscous pipe flow [40])

Note 3: the effective passage length is used in the viscous flow equation also.
APPENDIX D

From Newton's second law:

\[ -F_p - F_v = m \ddot{x} \]

\[ \pi r_p^2 (L_p + \pi r_p / 4) / \sqrt{v} \dddot{x} + 8\pi \mu (L_p + \pi r_p / 4) \ddot{x} + \pi^2 r_p^4 c^2 / (\sqrt{V_s} v) x = 0 \]

\[ \ddot{x} + \frac{8\mu v}{r_p^2} \dot{x} + \frac{\pi c^2 r_p^2}{V_s (L_p + \pi r_p / 4)} x = 0 \quad (D.4) \]

This equation is of the same form as the second order differential equation for a single degree of freedom system with damping, such as a spring, mass and dashpot system. It is commonly written in terms of the natural frequency and the damping coefficient as:

\[ \ddot{x} + 2\omega_0 \dot{x} + \omega_0^2 x = 0 \quad (D.5) \]

Comparing equations D.4 and D.5,

\[ \omega_0 = \frac{cr_p \sqrt{\pi}}{(\pi / (V_s (L_p + \pi r_p / 4)))} \]

\[ f_0 = \frac{\omega_0}{2\pi} = \frac{cr_p \sqrt{\pi}}{(\pi / (V_s (L_p + \pi r_p / 4)))} \]

On simplifying, \( f_0 = (c/2\pi L_p^* \omega_0) / \sqrt{R^*} \quad (D.6) \]

\[ \mathcal{F} = 4\mu v / (r_p^2 \omega_0) \quad (D.7) \]

On substituting for \( \omega_0 \) and rearranging,

\[ \mathcal{F} = \frac{4\pi \mu L_p^* \omega_0^2}{\mathcal{F} c V_p^*} / \sqrt{R^*} \quad (D.8) \]

Note 4: density, \( \mathcal{F} = 1/v, \quad V_p^* = V_p L_p^*/L_p \)

Since equation 4.6, ch. 4, given by Elson and Soedel [38], was claimed to be more accurate than equations such as D.6 over a wide range of values of \( R \) it would seem reasonable to incorporate it in the expression for the damping coefficient. This yields the following expression:

\[ \mathcal{F} = \frac{4\pi \mu L_p^* \omega_0^2}{c V_p^*} \sqrt{(R^* + 0.3905)} \quad (D.9) \]
APPENDIX E

VALVE FLOW CHARACTERISTICS

Data was available in graphical form in the manufacturers' catalogues on the effective flow areas of the throttle valves which were used in the load stand. In these graphs the number of turns open was plotted against a flow coefficient. The flow coefficient was converted to an effective flow area by a simple multiplier and polynomial equations were fitted to the data. These equations are listed below.

Valves A, B and D (9.5 mm orifice)
maximum number of turns = 4.51
corresponding maximum effective flow area = 30.564 [mm²]

\[ T = 0.02535 (A \times 10^6) + 0.004 (A \times 10^6)^2 \]  (E.1)

where

\[ T = \text{turns open} \]
\[ A = \text{effective flow area [m}^2\text{]} \]

Valve C (3.18 mm orifice)
maximum number of turns = 11
corresponding maximum effective flow area = 2.547 [mm²]

\[ T = 2.23605 (A \times 10^6) - 12.03583 (A \times 10^6)^2 + 25.50511 (A \times 10^6)^3 - 18.75227 (A \times 10^6)^4 + 6.08525 (A \times 10^6)^5 - 0.73680 (A \times 10^6)^6 \]  (E.2)

Valve C (0.79 mm orifice)
maximum number of turns = 10
corresponding maximum effective flow area = 0.0679 [mm²]
for \( 0 \leq (A \times 10^6) \leq 0.03 \) [mm²]

\[ T = 520.44826 (A \times 10^6) - 26085.98 (A \times 10^6)^2 + 824185.07 (A \times 10^6)^3 - 12651946 (A \times 10^6)^4 + 74131705 (A \times 10^6)^5 \]  (E.3)

for \( (A \times 10^6) > 0.03 \) [mm²]

\[ T = 2.6525 + 109.66 (A \times 10^6) \]  (E.4)
APPENDIX F

LIST OF MAIN ITEMS OF EQUIPMENT USED

SLOTTED OPTO SWITCHES
R.S. Components Ltd. no. 304-560
Infrared slotted opto switch with built in Schmitt trigger. Used for rotational speed measurement and top dead centre marking.

DIFFERENTIAL PRESSURE TRANSDUCER
Druck Ltd. PDCR 120/WL
Integrated silicon strain gauge bridge differential pressure transducer in which the measurement diaphragm was separated from the positive pressure port by silicone oil behind a sealing diaphragm. Range: 350 mbar. Excitation: 10v d.c. Output: 50mV. Natural frequency: 10.5 kHz. Used to measure differential pressure across flow measurement orifice.

STATIC PRESSURE TRANSDUCERS
Druck Ltd. PDCR 10
Integrated silicon strain gauge bridge transducer. The back of the measurement diaphragm was exposed to a sealed volume of air and the instrument had thermal compensation to allow for expansion of this air and so provide a sealed gauge datum. Range: 35 bar sealed gauge. Excitation: 10v d.c. Output: 100 mV. Natural frequency: 360 kHz. Used to measure the mean discharge pressure of the compressor.

Druck Ltd. PDCR 110/W
Integrated silicon strain gauge bridge transducer in which the measurement diaphragm was separated from the pressure port by silicon oil behind a sealing diaphragm. Range: 0 - 35 bar absolute. Excitation: 10v d.c. Output: 100 mV. Natural frequency: about 66 kHz. Used to measure the mean suction pressure of the compressor.

Transamerica Instruments BHL-3040
This transducer incorporated piezoresistive strain gauges bonded to a stainless steel diaphragm. Range: 0 - 6 bar abs. Excitation 10v d.c. Output: 100 mV. Used to measure the mean refrigerant pressure on the upstream side of the flow measurement orifice.
APPENDIX F

DYNAMIC PRESSURE TRANSDUCER
Kistler
601-A
+ 6421A20

This transducer was of the piezoelectric type, incorporating a quartz crystal. Maximum measuring range 0 - 245 bar. Sensitivity: -16pC/bar approx. Natural frequency: > 150 kHz. Used to measure the pressure within the compressor cylinder.

CHARGE AMPLIFIER
Fylde
FE128CA

Minimum gain: 1v per 5,000 pC
Maximum gain: 1v per pC
Output: +/- 10v max.
Frequency response: 100 kHz (-3dB), normally limited to 10 kHz (-3dB).
Selectable time constants: 0.1 s, 1 s, 10 s, static (1 pC/s drift). Used to pre-amplify the output of the Kistler pressure transducer.

DISPLACEMENT TRANSDUCERS
Bentley Nevada
7200 system

This was an eddy current type proximity transducer system. It consisted of a threaded cylindrical probe having a tip diameter of 5mm, a connecting cable, and a proximiter unit containing the electronics to supply a radio frequency signal to the probe, demodulate the return signal, and provide a high level voltage output. Linear range: 0.25 - 2.5 mm. Excitation: -24 v.
Sensitivity: about 7.7 v/mm.
Frequency response: d.c. to 10 kHz flat. Used to measure suction and discharge valve lifts.

DATA ACQUISITION SYSTEM
Biodata Ltd.
Microlink system

This was a modular hardware interface which operated on the IEEE-488 bus and had secondary addressing of the installed modules. The mainframe contained the power supply, the control unit for bus interaction and secondary addressing, and module slots with a backplane. The following modules were installed.

A-12D: A twelve bit analogue to digital converter which accepted an analogue signal in the range 0 to 10V from one of the analogue input modules. Acquisition time < 12 µs.

HSC: A high speed clock which was used for fast data acquisition and analogue
APPENDIX F

to digital conversion.

PGA-16: A programmable gain multiplexing and amplifying module for up to sixteen differential analogue inputs. Amplifier gains x1, x10, x100, x1,000. Signal settling time (worst case, at max. gain, 670 µs). This was used to accept the analogue outputs of the pressure transducers.

FC: A frequency counter capable of counting pulses, with gate times of 0.1, 1.0 or 10 seconds. Max. frequency 10 MHz. Used to measure compressor and motor speed by counting pulses from opto switches.

TC-16: A multiplexing and amplifying module for fifteen thermocouple inputs and an input from an integral platinum resistance thermometer attached to a common copper block, for cold junction compensation. Used for temperature measurements.

HEAT EXCHANGER
Bitzer Kühlmaschinenbau GmbH & Co. KG K070H

A water cooled tube bundle refrigerant condenser. This served as the load stand heat exchanger.

ELECTRONIC FREQUENCY INVERTER
Reliance Electric Invertron VSI
Frequency Converter 11 A

This was a three phase electronic inverter capable of driving motors up to 4 kW. Output frequency: 1 - 50 or 1 - 100 Hz. Modulation: pulse width modulation. Motor protection: by overcurrent relay. Isolated reference input: 0 - 10V. This unit was used to control the compressor speed. It provided a constant V/Hz ratio to the induction motor.

INDUCTION MOTOR
ACEC BF4100L14A 2.2kW, 3-phase, 50 Hz. This was used to drive the compressor by a pair of V-belts.

THROTTLE VALVES
Whitey BS-18VS12

APPENDIX F

Nupro SS-SS4
Fine metering valve. Orifice: 0.79mm. Connections: 1/4 in. Swagelok. Used for valve C₁ on the load stand for fine adjustment.

Nupro SS-4L

SPRING BALANCE
Salter model 235
Range 0 - 25 kgf x 0.1 kgf. This was used to measure motor torque in conjunction with a radius arm attached to the motor cradle.

THERMOCOUPLE METER
Eirelec Ltd. Type K digital temperature meter with seven position selector switch. Resolution: 0.1°C. Used to check or display temperature measurements.

MASTER PRESSURE GAUGE
Budenberg 214F
250mm standard test gauge, graduated 0 - 16 barg x 0.1 bar. Used to calibrate and check gauges and transducers.

FOUR WAY CHANGEOVER VALVE
Whitey B-43YF2

WATER FLOW METER
GTV Gapmeter
Variable area flow meter with transparent glass scale. Range: to 4.5 l/min.

DIGITAL STORAGE OSCILLOSCOPE

THREE PORT SAMPLING VALVE
Castel RFT-2
Diaphragm sealed three way valve. Connections: 1/4 in. flare.

COMPUTERS
Hewlett-Packard
Desktop computer. This was used for data collection via the IEEE bus.
<table>
<thead>
<tr>
<th>Company</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hewlett-Packard 9845B</td>
<td>Desktop computer. This was used to read test data from cassette tapes and for analysis and presentation of test results. The test data and results were stored on floppy discs. A Hewlett-Packard flat bed plotter was used for graphical output of results.</td>
</tr>
<tr>
<td>Digital Equipment Co. DEC 2060</td>
<td>Mainframe computer running under TOPS 20. This was used to run FORTRAN programs for compressor simulation. PLOT79 software was used to write programs for graphical presentation of the results.</td>
</tr>
</tbody>
</table>
The mass flow rate of refrigerant was calculated according to the following equation:

\[
\dot{m} = CE \pi \left(\frac{d}{2}\right)^2 \sqrt{\frac{2(\Delta p)\rho_u}{f_u}} \quad (G.1)
\]

where

\[
\begin{align*}
\dot{m} & = \text{mass flow rate of refrigerant [kg/s]} \\
C & = \text{coefficient of discharge} \\
E & = \text{velocity of approach factor} \\
\beta & = \text{diameter ratio } = d/D \\
d & = \text{orifice diameter [m]} \\
D & = \text{pipe diameter [m]} \\
\epsilon & = \text{expansibility factor} \\
\Delta p & = \text{differential pressure across orifice [N/m}^2] \\
f_u & = \text{density of refrigerant upstream of orifice [kg/m}^3]
\end{align*}
\]

The coefficient of discharge was calculated from the Stolz equation (G.2) [51].

\[
C = 0.5959 + 0.0312\beta^{0.1} - 0.1840\beta^2 \\
+ 0.00029\beta^{1.5}(10^6/Re_D)^{0.75} \\
+ 0.0900L_1(0.0390) - 0.0337L_2\beta^3 \quad (G.2)
\]

where

\[
\begin{align*}
Re_D & = \text{Reynolds number based on pipe diameter} \\
L_1 & = 1 \text{ for D and D/2 taps} \\
L_2 & = 0.47 \text{ for D and D/2 taps}
\end{align*}
\]

The expansibility factor was calculated from an equation given in [51], (G.3).

\[
\epsilon = 1 - (0.41 + 0.35\beta^4)(\Delta p/\rho_u) \quad (G.3)
\]

where

\[
\begin{align*}
k & = \text{isentropic exponent} \\
\rho_u & = \text{density of refrigerant upstream of orifice [kg/m}^3]
\end{align*}
\]

Limitations to the Applicability of Equations G.1 to G.3

Equation G.1 is equivalent to the flow equation used in British Standard 1042 for square edged orifice plates with D and D/2 taps [48, 49, 46]. The standard imposed the following restrictions:

1. The orifice plate and pressure tappings should be manufactured according to the specifications given in the standard and specified minimum lengths of straight pipe run should be provided upstream and downstream.

2. \(50 \text{mm} < D < 760 \text{mm}\)

OR

3. \(25 \text{mm} \leq D \leq 50 \text{mm}\) if a further correction factor was applied to the discharge coefficient, C, and an additional uncertainty in the flow measurement was accepted.

4. \(0.2 \leq \beta \leq 0.75\)

5. \(Re_D \geq 1260\beta^2 D\)

6. \(0.2 \leq \beta \leq 0.75\)

7. \(0.032 \leq CE_D \leq 0.35\)

Equation G.3 for the expansibility factor was subject to the limitation that

8. \(\rho_d/\rho_u \geq 0.75\)

where

\[
\begin{align*}
\rho_d & = \text{downstream pressure [N/m}^2] \\
\rho_u & = \text{pressure of refrigerant upstream of orifice}
\end{align*}
\]
APPENDIX G

\[ p_u = \text{upstream pressure [Nm}^{-2} \text{]} \]

All of the above constraints except for (2) were met. The diameter of the pipe used was 20.2mm. As the Stolz equation G.2 was used to calculate the coefficient of discharge, rather than tables in the British Standard, additional corrections due to the small pipe diameter were not necessary.

Constants and Parameters Used in the Flow Equation

For tests to no. 33:
\[ d = 0.0128 \text{ m} \]
\[ D = 0.0202 \text{ m} \]
Hence, \[ \beta = 0.6337 \]

For tests 34 on:
\[ d = 0.01106 \text{ m} \]
\[ D = 0.0202 \text{ m} \]
Hence, \[ \beta = 0.5475 \]

In the computer program which was used to evaluate the refrigerant mass flow rate from the test measurements the upstream pressure and temperature were used to calculate the density, specific heat at constant pressure, and absolute viscosity of the refrigerant from property subroutines as described in chapter 1, section 1.5.3. The Reynolds number was evaluated initially from an approximate estimate of the flow rate. By a process of iteration, which involved re-evaluation of the measured flow rate, the correct value value of the Reynolds number was determined. Hence \( C \) was evaluated from equation G.2. The isentropic exponent was taken to be the ratio of the specific heats. The specific heat at constant volume was calculated from the specific gas constant and the specific heat at constant pressure. Hence \( \epsilon \) was evaluated according to equation G.3.
The transducer was exposed, unintentionally, to a vacuum (caused by flow choking upstream) simultaneously on both sides. This occurred a number of times for durations less than 30 seconds and failure occurred. However, the situation described did not appear to violate the conditions of use as defined by the manufacturers’ specification sheet.

It was considered likely that the failure of this transducer was due to its unsuitability for use in situations where it could be filled with liquid refrigerant on the low pressure side and relatively fast reductions in line pressure could occur. This would often be the case in a refrigeration circuit.

The transducer contained a very small bore (about 0.5mm) capillary tube on the low pressure side, fig. H-1. This led to a small chamber with a diameter of about 4mm and a length of about 6mm, which had the silicon measurement diaphragm at its opposite end. The high pressure side of the measurement diaphragm was in contact with silicone oil and this, in turn, was separated from the high pressure connection by a stainless steel diaphragm. Some rough calculations were carried out using the details mentioned and the properties of R-12 and, based on the results, a possible mechanism of failure was proposed as follows.

Suppose the transducer is at ambient temperature and contains liquid refrigerant on both sides, i.e. against the stainless steel diaphragm and within the small chamber against the silicon diaphragm, and a relatively fast decrease in line pressure occurs, say from the saturation value (e.g. 4.9 bara at 15°C) to moderate vacuum in a fraction of a second. Flash evaporation will occur within the high pressure side of the transducer and the pressure on the stainless steel diaphragm will decrease, remaining close to line pressure. On the low pressure side flash evaporation will occur within the capillary tube and choked flow will be established almost instantly. Evaporation within the chamber against the silicon diaphragm will proceed at a rate corresponding to the mass flow rate through the capillary. As long as liquid refrigerant remains within the chamber the pressure will remain close to its original saturation value (4.9 bara). Thus, a large differential pressure will be applied to the silicon diaphragm from the low pressure side causing it to rupture.

It should be noted that a relatively small decrease in line pressure on the low pressure side of the transducer would be sufficient to cause choked flow in the capillary. According to the rough calculations the time taken for the liquid within the chamber at the end of the capillary to evaporate would be at least 0.2 seconds. A differential pressure overload could thus occur internally within the transducer for a short period after
APPENDIX H

a drop in line pressure when no external overload whatever is applied.

Fig. H-1 Schematic diagram of the differential pressure transducer.
APPENDIX I

DETAILS OF THE TESTING PROGRAMME
HISTORY OF TESTS, METHODS AND PROCEDURES

Test Nos.

Notes: Differential pressure measurements for orifice plate (12.8 mm) were made by means of a transducer. No pulsation damping chamber.

1 TDC marker was in wrong place.

Changes: Differential pressure transducer failed. A four way changeover valve and two purging valves were installed to facilitate transducer isolation during system charging so as to prevent vacuum damage and also to facilitate re-zeroing with the compressor running. Capillary tubing was used to interconnect the valves and the orifice. A new transducer was installed.

2 - 3 Initial tests.

4 - 6 From test 5 a facility was included in the data logging program which simplified re-zeroing of the differential pressure transducer. This was used before logging the results for each test. The results were evaluated by calculator and there were large apparent disagreements (28% at one stage) between the flow rates according to the orifice plate and heat balance measurement methods.

4 - 12 Tests over the speed range 400 to 600 r.p.m. Suction saturation temp.: -10°C, discharge sat. temp.: 30°C, suction temp.: 5°C. The results were evaluated by computer program. These were presented at the 1986 Purdue conference [53]. Mass flow rates were based on the heat balance method as there were still some doubts about the orifice plate flow measurements.

Changes: Major modifications were made to the test rig. Capillary tubes to the differential pressure transducer from the orifice plate were shortened. A suction pressure pulsation damping chamber (about 1.5 litres) and a perforated damping plate were installed. These changes reduced the amplitude of the pressure pulsations across the orifice plate.

13 - 14 There were still large apparent differences between the flow rates measured by the orifice plate and heat balance methods.

Changes: A larger suction line pulsation damping chamber (about 5.4 litres) was installed. A pressure gauge was fitted just upstream of the orifice.
plate. The test rig was run before the damping chamber was insulated and was found to be highly unstable. There was a noticeable time lag between a change in pressure at the orifice and downstream at the suction side of the compressor. Stability improved considerably when the insulation was fitted.

The specific volume at the orifice plate was calculated using the measured pressure and temperature at the orifice, rather than the suction values which had been used previously. However, flow rate disagreements still persisted (at a level of about 12%).

Changes: The water flow meter was calibrated and it was found that it had been reading a little high. TDC marker on flywheel was realigned.

Apparent disagreements between flow measurement methods were considerably reduced. Three tests were carried out at each operating speed, with one oil sample, which was taken in conjunction with the first test of the triad. Some results for tests 19, 20 and 21 were lost due to an error in the data logging program.

Changes: Following careful theoretical analysis, the fine liquid metering valve was replaced with a finer one to improve control of suction superheat. The capillary tubes connecting the differential pressure transducer to the orifice were replaced by larger diameter tubes. A pressure transducer was installed upstream of the orifice to transfer the measurement directly to the data logging system. A smaller orifice (6.02mm) was installed to increase the differential pressure output.

Failure: The new differential pressure transducer failed. Several theories of failure were examined and eventually it was concluded that this was probably due to a transient overload associated with choking at valve D or at the orifice plate (the former was considered more likely).

Changes: As a suitable alternative transducer was not available within budgetary and time constraints, it was replaced by a mercury manometer which was specially designed to withstand the line pressure. The orifice was replaced with one of 11.06 mm diameter. A heat exchanger heat loss coefficient was introduced for use in the calculation of the flow rate by the heat balance method. The parameters used in the orifice flow
equation were calculated for each test using measured operating values. The expansibility factor was found to be particularly sensitive to the differential pressure. Properties of the refrigerant/oil mixture were used throughout in the calculations of results, rather than assuming the refrigerant was pure. The modified program for calculating results was applied retrospectively from test 22 onwards and gave good agreement between the flow rates measured by the orifice plate and heat balance methods. A large diameter master pressure gauge was installed and manifolded to all pressure transducers to facilitate calibration. Frequent pressure transducer calibration checks and corrections were carried out for all subsequent tests.

34 - 48

Good levels of agreement between the flow measurement methods were achieved, except at low superheat values. Also, it was not possible to maintain stable operating conditions at low superheat. From test 47 onwards, the number of points transferred from the storage scope to the data logging computer was doubled. Although still limited by the memory capacity of the computer this increase was achieved by declaring the arrays used to be 'short precision'.

Changes:

The requirements for steady state operation before readings were taken were defined much more strictly. The new criterion for stability was that the rate of change of the suction temperature should consistently be less than 0.1 K/min. (brief excesses were tolerated) and the rates of change of the suction and discharge pressures should consistently be less than 0.1 bar/min., while the nominal operating conditions were held for at least 30 minutes before each set of test readings.

49 - 54

Tests were considered good. Minor qualifications were noted.

55 - 64

For these tests additional thermocouples were installed to provide additional data relating to heat transfer effects.

Changes:

The computer program for the calculation of results was further developed and was applied retrospectively to tests from 22 onwards.
APPENDIX I

SELECTION OF RESULTS FOR FINAL PRESENTATION

Tests Versus Speed

**Operating Conditions**

| Suction saturation temperature | -10°C |
| Discharge saturation temperature | 40°C |
| Suction temperature | 5°C |

<table>
<thead>
<tr>
<th>Speed (r.p.m.)</th>
<th>Test No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>300.3</td>
<td>53</td>
</tr>
<tr>
<td>349.6</td>
<td>52</td>
</tr>
<tr>
<td>381.6</td>
<td>47</td>
</tr>
<tr>
<td>400.0</td>
<td>27</td>
</tr>
<tr>
<td>497.3</td>
<td>48</td>
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<tr>
<td>500.6</td>
<td>23</td>
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<td>557.6</td>
<td>62</td>
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<tr>
<td>592.3</td>
<td>49</td>
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<td>600.3</td>
<td>41</td>
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<td>650.3</td>
<td>55</td>
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<td>650.3</td>
<td>56</td>
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<td>651.3</td>
<td>57</td>
</tr>
<tr>
<td>698.0</td>
<td>51</td>
</tr>
<tr>
<td>699.0</td>
<td>29</td>
</tr>
<tr>
<td>699.3</td>
<td>28</td>
</tr>
<tr>
<td>768.0</td>
<td>59</td>
</tr>
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<td>769.0</td>
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<td>798.6</td>
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</tr>
<tr>
<td>799.3</td>
<td>33</td>
</tr>
<tr>
<td>835.0</td>
<td>50</td>
</tr>
<tr>
<td>879.0</td>
<td>61</td>
</tr>
<tr>
<td>902.3</td>
<td>54</td>
</tr>
</tbody>
</table>

Four successive tests at the same operating conditions. Only one oil sample taken, in conjunction with test 56.

Two successive tests at the same operating conditions. Separate oil samples taken.

Tests versus Suction Temperature

| Suction saturation temperature | -10°C |
| Discharge saturation temperature | 40°C |
| Compressor speed | 600 r.p.m. |

<table>
<thead>
<tr>
<th>Suct. Temp. (°C)</th>
<th>Test No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.81</td>
<td>39</td>
</tr>
<tr>
<td>4.71</td>
<td>41</td>
</tr>
<tr>
<td>4.73</td>
<td>49</td>
</tr>
<tr>
<td>9.65</td>
<td>42</td>
</tr>
<tr>
<td>16.15</td>
<td>64</td>
</tr>
<tr>
<td>19.58</td>
<td>44</td>
</tr>
<tr>
<td>24.68</td>
<td>43</td>
</tr>
<tr>
<td>27.16</td>
<td>63</td>
</tr>
</tbody>
</table>
APPENDIX J

VOLUMETRIC INDUCTION EFFICIENCY ANALYSIS PROCEDURE

Start

Set suction and discharge leakage ratios to zero.

Calculate the heat transfer rate to the suction vapour in the suction plenum and the heat transfer rate from the discharge vapour in the discharge plenum: from the suct. & disch. temperatures, the temperatures adjacent to the valves and the external mass flow rate.

Calculate the mass induced and the mass discharged from the external flow rate and the leakage ratios.

Calculate the mass of refrigerant present at each of the points of pressure equalisation, 1, 2, 3 & 4, assuming \( v_1 = v_4 \) and \( v_2 = v_3 \).

Calculate the specific volumes \( v_1 (= v_4) \) & \( v_2 (= v_3) \).

Evaluate \( T_1 (= T_4) \) and \( T_2 (= T_3) \) from the pressures and specific volumes.

Calculate the net heat transfer rates to the vapour during induction and from the vapour during the discharge process.

1st iter. ?

yes

Output the net heat transfer rates, the volumetric induction efficiency and the temperatures \( T_1 \) & \( T_4 \) assuming no leakage.

no

Fig. J-1 (contd. overleaf)

J-1
Estimate leakage ratios which would make the net heat transfer rates during induction and discharge zero.

Output the leakage ratios, the B temperatures and the volumetric induction efficiency assuming adiabatic vapour transfer.

Fig. J-1 Calculation procedure which was used to assess volumetric induction efficiency from test measurements based on two assumptions: A) heat transfer during the induction and discharge processes with no leakage or backflow and B) adiabatic vapour transfer with leakage and/or backflow.
APPENDIX K

INPUT DATA FOR THE PURDUE SIMULATION PROGRAM

INPUT DESCRIPTION: Bitzer IV compressor

<table>
<thead>
<tr>
<th>angle [°]</th>
<th>press. [bara]</th>
</tr>
</thead>
<tbody>
<tr>
<td>suction plenum:</td>
<td>0.000 2.1910 360.0 2.1910</td>
</tr>
<tr>
<td>discharge plenum:</td>
<td>0.000 9.6070 360.0 9.6070</td>
</tr>
</tbody>
</table>

Table K-1 Suction and discharge plenum pressures as a function of crank angle. These are taken to be constant.

<table>
<thead>
<tr>
<th>lift [mm]</th>
<th>Effective Flow Areas [cm²]</th>
<th>Suction Valve</th>
<th>Discharge Valve</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>normal backflow</td>
<td>normal backflow</td>
<td></td>
</tr>
<tr>
<td>0.00</td>
<td>0.000 0.000</td>
<td>0.000 0.000</td>
<td></td>
</tr>
<tr>
<td>0.50</td>
<td>0.140 0.020</td>
<td>0.100 0.005</td>
<td></td>
</tr>
<tr>
<td>1.00</td>
<td>0.270 0.060</td>
<td>0.190 0.020</td>
<td></td>
</tr>
<tr>
<td>1.50</td>
<td>0.360 0.150</td>
<td>0.370 0.090</td>
<td></td>
</tr>
<tr>
<td>2.00</td>
<td>0.440 0.300</td>
<td>0.440 0.200</td>
<td></td>
</tr>
<tr>
<td>2.50</td>
<td>0.520 0.460</td>
<td>0.480 0.400</td>
<td></td>
</tr>
<tr>
<td>3.00</td>
<td>0.580 0.580</td>
<td>0.510 0.560</td>
<td></td>
</tr>
<tr>
<td>3.50</td>
<td>0.660 0.680</td>
<td>0.550 0.670</td>
<td></td>
</tr>
<tr>
<td>4.00</td>
<td>0.730 0.770</td>
<td>0.600 0.760</td>
<td></td>
</tr>
<tr>
<td>4.50</td>
<td>0.730 0.820</td>
<td>0.630 0.830</td>
<td></td>
</tr>
</tbody>
</table>

Table K-2 Valve effective flow areas versus lift.

<table>
<thead>
<tr>
<th>lift [mm]</th>
<th>Effective Force Areas [cm²]</th>
<th>Suction Valve</th>
<th>Discharge Valve</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>normal backflow</td>
<td>normal backflow</td>
<td></td>
</tr>
<tr>
<td>0.00</td>
<td>1.501 1.501</td>
<td>1.522 1.522</td>
<td></td>
</tr>
<tr>
<td>0.50</td>
<td>1.371 1.439</td>
<td>1.398 1.426</td>
<td></td>
</tr>
<tr>
<td>1.00</td>
<td>1.119 1.335</td>
<td>1.107 1.287</td>
<td></td>
</tr>
<tr>
<td>1.50</td>
<td>0.590 1.175</td>
<td>0.609 1.107</td>
<td></td>
</tr>
<tr>
<td>2.00</td>
<td>0.554 1.015</td>
<td>0.623 0.913</td>
<td></td>
</tr>
<tr>
<td>2.50</td>
<td>0.818 0.873</td>
<td>0.934 0.734</td>
<td></td>
</tr>
<tr>
<td>3.00</td>
<td>0.916 0.720</td>
<td>1.024 0.574</td>
<td></td>
</tr>
<tr>
<td>3.50</td>
<td>0.923 0.572</td>
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<td>4.00</td>
<td>0.916 0.443</td>
<td>1.017 0.422</td>
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</tr>
<tr>
<td>4.50</td>
<td>0.910 0.369</td>
<td>1.010 0.408</td>
<td></td>
</tr>
</tbody>
</table>

Table K-3 Valve effective force areas versus lift.

K-1
### Areas of Valve Reed and Port Elements [cm²]

<table>
<thead>
<tr>
<th>element no.</th>
<th>Areas of Valve Reed and Port Elements</th>
<th>[cm²]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Suction Valve</td>
<td>Discharge Valve</td>
</tr>
<tr>
<td></td>
<td>reed</td>
<td>port</td>
</tr>
<tr>
<td>1</td>
<td>0.150</td>
<td>0.000</td>
</tr>
<tr>
<td>2</td>
<td>0.274</td>
<td>0.080</td>
</tr>
<tr>
<td>3</td>
<td>0.340</td>
<td>0.209</td>
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<tr>
<td>4</td>
<td>0.370</td>
<td>0.260</td>
</tr>
<tr>
<td>5</td>
<td>0.390</td>
<td>0.277</td>
</tr>
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<td>6</td>
<td>0.401</td>
<td>0.272</td>
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<tr>
<td>7</td>
<td>0.411</td>
<td>0.225</td>
</tr>
<tr>
<td>8</td>
<td>0.421</td>
<td>0.166</td>
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<tr>
<td>9</td>
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<td>10</td>
<td>0.441</td>
<td>0.000</td>
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<td>13</td>
<td>0.206</td>
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<td>14</td>
<td>0.209</td>
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<td>22</td>
<td>0.320</td>
<td>0.000</td>
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<tr>
<td>23</td>
<td>0.392</td>
<td>0.000</td>
</tr>
<tr>
<td>24</td>
<td>0.392</td>
<td>0.000</td>
</tr>
</tbody>
</table>

Table K-4 Areas of valve reed and port elements.

1. **GEOMETRIC PARAMETERS**
   - crank radius: 0.025 [m]
   - connecting rod length: 0.145 [m]
   - piston diameter: 0.065 [m]
   - clearance volume: 4.51 [ml]

2. **WORKING FLUID PARAMETERS**
   - specific gas constant (R-12): 68.76 [J/kgK]
   - ratio of specific heat values: 1.18

3. **OPERATING CONDITIONS**
   - crank speed: 600.0 [r.p.m.]
   - plenum pressures as a function as crank angle: table K-1
   - suction plenum temperature: 278.15 [K]
   - discharge plenum temperature: 333.15 [K]

4. **INITIAL CONDITIONS**
   - initial crank angle: 0.00 [']
   - initial cylinder pressure: 2.191 [bara]
   - initial temperature of mass in cylinder: 278.15 [K]
APPENDIX K

5. CALCULATION STEP PARAMETERS
step size: \( 0.05 \, [\text{\textdegree}] \)
total change in crank angle: \( 720.00 \, [\text{\textdegree}] \)

6. POLYTROPIC INDEX
index: \( 1.1 \)

7. VALVE EFFECTIVE FLOW AREAS
see table K-2

8. VALVE EFFECTIVE FORCE AREAS
see table K-3

9. AREAS OF VALVE REED AND PORT ELEMENTS
   total suction valve port area: \( 1.501 \, [\text{cm}^2] \)
   total discharge valve port area: \( 1.523 \, [\text{cm}^2] \)
   see table K-4

10. NATURAL FREQUENCIES AND MODE SHAPES OF VALVE REEDS
    mode shapes: table K-5
    density of valve reed material: \( 7900.0 \, [\text{kg/m}^3] \)

11. DAMPING RATIOS
    suction valve first mode: \( 0.12 \)
    suction valve second mode: \( 0.12 \)
    discharge valve first mode: \( 2.0 \)

Suction Valve:
no. of modes: \( 2 \)
1st natural frequency: \( 200 \, [\text{Hz}] \)
2nd natural frequency: \( 698 \, [\text{Hz}] \)
thickness of suction valve reed: \( 0.406 \, [\text{mm}] \)

Discharge Valve:
no. of modes: \( 1 \)
1st natural frequency: \( 493 \, [\text{Hz}] \)
thickness of discharge valve reed: \( 0.888 \, [\text{mm}] \)
### APPENDIX K

<table>
<thead>
<tr>
<th>element no.</th>
<th>Relative Displacements of Valve Reeds</th>
<th></th>
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<tr>
<td></td>
<td>Suction Valve</td>
<td>Disch. Valve</td>
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<tr>
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<td>2nd mode</td>
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Table K-5  Mode shapes of the suction and discharge valves.