Gravity flow of some steelmaking raw materials with particular reference to the effects of vibration

Alan S. Kaaden

University of Wollongong
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Signed
GRAVITY FLOW OF SOME STEELMAKING RAW MATERIALS WITH PARTICULAR REFERENCE TO THE EFFECTS OF VIBRATION

BY

A. S. KAADEN, B.E. (MECH) HONS (MELB), GRAD. I. E. AUST.

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The University of New South Wales

Department of Mechanical Engineering
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(formerly Wollongong University College)

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SUMMARY

The thesis comprises two main parts:

(i) A compilation of flow properties of the most difficult bulk solid materials handled by the Raw Materials area of Australian Iron & Steel, and a check on the various bin designs used in this area of the plant to predict flow behaviour with these materials.

(ii) A study of the effect of vibration on the strength of bulk solid materials and on the frictional forces between a surface and a material such as exists in vibrating baffles or vibrating panels.

The first part of the work was done using the experimental methods of Jenike, a pioneer in this field. The vibration section involved the construction of a suitable vibrating machine and shaking samples of ore in the Jenike shear cell either before shearing or during shearing. To get more accurate results when shearing was taking place during vibration, a special tilting device was used in preference to the Jenike machine.

It may be concluded from the first part of the work that many bins are not designed properly for the material they are expected to handle, and only the use of mechanical flow promoting devices and frequent cleaning particularly in wet weather keep the materials flowing.
The vibration work showed two contrasting points:

(a) that a confined sample of material gains considerable strength due to over-consolidation during vibration, and

(b) the same confined sample has a considerably lower wall friction when the wall is being vibrated.
ACKNOWLEDGEMENTS

The author is extremely grateful for the encouragement, guidance and advice given by Dr. Peter Arnold in the supervision of this thesis, particularly with helping to overcome the numerous problems that were encountered in the various stages of the work. Thanks are also due for the early assistance given by the then Associate Professor Alan Roberts.

Mr. Kevin Brady in the Materials Handling Laboratory gave invaluable help with the equipment, and also the work done by the Engineering Workshop in manufacturing the mechanical shaker was appreciated. The Raw Materials Electrical Section at Australian Iron and Steel gave the author much assistance in the production of the electrical shaker, for which gratitude must be expressed. The vibration aspect of this research has received support by the Australian Research Grants Committee. This support is gratefully acknowledged.

Finally I would like to thank Associate Professor S. E. Bonamy, Chairman, Department of Mechanical Engineering, for the opportunity to undertake the research work in the Department.
NOMENCLATURE

A  area of a Jenike shear cell, (l/13 ft²).

a  amplitude of vibration.

B  width of a rectangular outlet, side of a square outlet, diameter of a circular outlet, ft.

D  diameter of a pipe, major dimension of an outlet, diameter of a circular outlet, ft.

Dₜ  major dimension of a hopper outlet, ft.

E.Y.L.  effective yield locus.

e  voids ratio.

eₚ  voids ratio in the plane of failure at the moment of failure.

F = A x fc  unconfined yield force, lb.

Fₜ  unconfined yield force after time consolidation, lb.

FF  flow function of a solid.

f  frequency, Hz.

fc  unconfined yield pressure of a solid, lb/ft².

ff  flow factor of a hopper.

G(₀ₜ)  factor for no-piping in core flow.

g  gravity constant, (lbm-ft)/(lbf-sec²).

H  moisture content, % w.b.

H(₀')  factor for no-doming in mass flow.

L  length of a rectangular outlet, ft.

N  total normal force exerted on plane of material in contact with vibrating plate during vibration wall yield loci tests, lbf.
\( S = A \times \tau \) shearing force applied to a shear cell during consolidation, lb.

\( \bar{S} \) shearing force applied to a shear cell during shear, lb.

\( S_1 \) peak shearing force due to overconsolidation, lb.

\( S_2 \) steady shearing force after passing peak, lb.

\( S_3 \) shear force at lower value of normal force, lb.

\( S_{avg} \) average of \( S \) values from a number of tests.

\( \bar{S} \) corrected shear force during shearing of a cell, corrected for variations in \( S \).

\( T \) Temperature.

\( T.Y.L. \) Time yield locus.

\( t \) time.

\( V = A \times \sigma \) normal force applied to a shear cell during consolidation, lb.

\( \bar{V} \) normal force applied to a shear cell during shear, lb.

\( V_1 = A \times \sigma_1 \) major consolidating force, lb.

\( V_2 = A \times \sigma_2 \) minor consolidating force, lb.

\( \bar{V}_1 = A \times \sigma_1 \) major force in a dome or pipe, lb.

\( V_t \) vertical force applied to a shear cell during twisting, lb.

\( W.Y.L. \) wall yield locus.

\( x \) nominal distance, ft.

\( \ddot{Y} \) acceleration of vibrating plate, ft/sec\(^2\).

\( Y.L. \) yield locus

\( \alpha \) angle of inclination of vibrating plate from horizontal.
angle of inclination of ore face from horizontal.

effective angle of friction.

bulk density of a solid, lbs/ft\(^3\).

angle of inclination of a wall from horizontal

coefficient of friction.

conical hopper slope measured from vertical.

plane flow or pyramid hopper slope measured from vertical.

Kinematic angle of internal friction of a solid.

static angle of internal friction of a solid.

pressure, lbs/ft\(^2\).

major consolidating pressure, lbs/ft\(^2\).

minor consolidating pressure, lbs/ft\(^2\).

designations for differing \(\sigma\) values

major pressure in a dome or pipe, lbs/ft\(^2\).

minor pressure in a dome or pipe, lbs/ft\(^2\).

effective normal stress on the plane of failure, lbs/ft\(^2\).

shearing stress, lbs/ft\(^2\).

peak shear stress at failure, lbs/ft\(^2\).

frequency of vibration, radians/sec.

natural frequency, radians/sec.

Kinematic angle of friction between a solid and a wall.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Summary</th>
<th>Page No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acknowledgements</td>
<td>(iii)</td>
</tr>
<tr>
<td>Nomenclature</td>
<td>(iv)</td>
</tr>
<tr>
<td>Table of Contents</td>
<td>(vii)</td>
</tr>
<tr>
<td>List of Figures</td>
<td>(ix)</td>
</tr>
<tr>
<td>List of Graphs</td>
<td>(xii)</td>
</tr>
</tbody>
</table>

Section 1  INTRODUCTION                         1

Section 2  LITERATURE SURVEY                   4

Section 3  THEORETICAL CONSIDERATIONS          28

  3.1 Flowability Concepts                     28
  3.2 Theory Related to Density Changes and Vibration.  41

Section 4  ANALYSIS OF STEELMAKING RAW MATERIALS 52

  4.1 Preamble                               52
  4.2 Flow Properties of the Raw Materials    54
  4.3 Check on Use of Mt. Newman Fines in some A.I.S. Bins  63
  4.4 Design of a Bin for Mt. Newman Fines    79
  4.5 Discussion                             87

Section 5  ANALYSIS OF VIBRATION EFFECTS       89

  5.1 Preamble                               89
  5.2 Development of a Shaking Device         90
  5.3 Effect of Vibration on Shear Strength   96
  5.4 Effect of Vibration on Density and Density on Shear Strength  103
<table>
<thead>
<tr>
<th>Section</th>
<th>Page No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.5</td>
<td>105</td>
</tr>
<tr>
<td>5.6</td>
<td>108</td>
</tr>
<tr>
<td>6</td>
<td>123</td>
</tr>
<tr>
<td>6.1</td>
<td>123</td>
</tr>
<tr>
<td>6.2</td>
<td>129</td>
</tr>
<tr>
<td>7</td>
<td>137</td>
</tr>
<tr>
<td>7.1</td>
<td>137</td>
</tr>
<tr>
<td>7.2</td>
<td>140</td>
</tr>
<tr>
<td>8</td>
<td>142</td>
</tr>
<tr>
<td>9</td>
<td>146</td>
</tr>
<tr>
<td>9.1</td>
<td>146</td>
</tr>
</tbody>
</table>

Section 5.5: Effect of Vibration in Overconsolidation of Samples prior to testing for Yield Loci

5.6: Effect of a Vibrating Plate on Wall Yield Locus.

Section 6: SUMMARY OF CONCLUSIONS

6.1: Analysis of Steelmaking Raw Materials

6.2: Analysis of Vibration Effects

Section 7: SUGGESTIONS FOR FURTHER WORK

7.1: Further work from Steelmaking Raw Materials Section

7.2: Further work from Vibration Section.

Section 8: LIST OF REFERENCES

Section 9: APPENDIX

9.1: Graphical Results for Steelmaking Raw Materials.
<table>
<thead>
<tr>
<th>Fig. No.</th>
<th>Description</th>
<th>Page No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>Hvorslev failure surface.</td>
<td>6</td>
</tr>
<tr>
<td>2.2</td>
<td>Failure &amp; Yield surface.</td>
<td>6</td>
</tr>
<tr>
<td>2.3</td>
<td>Consolidation &amp; Yield surfaces in $(\gamma, \sigma, \tau)$ space.</td>
<td>8</td>
</tr>
<tr>
<td>2.4</td>
<td>Critical compaction overload graph.</td>
<td>9</td>
</tr>
<tr>
<td>2.5</td>
<td>Reduction of a family of yield loci to a single line.</td>
<td>9</td>
</tr>
<tr>
<td>2.6</td>
<td>Included angle of vertical wall hoppers.</td>
<td>11</td>
</tr>
<tr>
<td>2.7</td>
<td>Suggested theory using sections of Janssen, Walker &amp; Walters.</td>
<td>11</td>
</tr>
<tr>
<td>2.8</td>
<td>Conical insert expands flow.</td>
<td>16</td>
</tr>
<tr>
<td>2.9</td>
<td>Vertical vibration of hopper &amp; cohesionless contents.</td>
<td>16</td>
</tr>
<tr>
<td>2.10</td>
<td>Experimental apparatus of Scarlett &amp; Eastham.</td>
<td>19</td>
</tr>
<tr>
<td>2.11</td>
<td>Bin activator with vibrating baffle.</td>
<td>19</td>
</tr>
<tr>
<td>2.12</td>
<td>&quot;Bridge Breaker&quot; reduces wall friction.</td>
<td>24</td>
</tr>
<tr>
<td>2.13</td>
<td>Vibrating feeder.</td>
<td>24</td>
</tr>
<tr>
<td>2.14</td>
<td>Open air float conveyor.</td>
<td>26</td>
</tr>
<tr>
<td>2.15</td>
<td>Aeration cartridge.</td>
<td>26</td>
</tr>
<tr>
<td>3.1</td>
<td>Pressure acting on a flowing element.</td>
<td>29</td>
</tr>
<tr>
<td>3.2</td>
<td>Effective Yield Locus.</td>
<td>29</td>
</tr>
<tr>
<td>3.3</td>
<td>Yield Locus of a cohesionless solid.</td>
<td>32</td>
</tr>
<tr>
<td>3.4</td>
<td>Family of Yield Loci of a cohesive solid.</td>
<td>32</td>
</tr>
<tr>
<td>3.5</td>
<td>Time Yield Locus.</td>
<td>34</td>
</tr>
<tr>
<td>3.6</td>
<td>Wall Yield Locus.</td>
<td>34</td>
</tr>
<tr>
<td>3.7</td>
<td>Flow Functions of a solid.</td>
<td>36</td>
</tr>
<tr>
<td>Fig. No.</td>
<td>Description</td>
<td>Page No.</td>
</tr>
<tr>
<td>---------</td>
<td>------------------------------------------------------------------------------</td>
<td>----------</td>
</tr>
<tr>
<td>3.8</td>
<td>Family of Yield Loci required to draw Flow Function.</td>
<td>36</td>
</tr>
<tr>
<td>3.9</td>
<td>Intersection of solid Flow Function with hopper Flow Factor.</td>
<td>40</td>
</tr>
<tr>
<td>3.10</td>
<td>The Hvorslev failure surface.</td>
<td>42</td>
</tr>
<tr>
<td>3.11</td>
<td>Types of stress-strain curves.</td>
<td>42</td>
</tr>
<tr>
<td>3.12</td>
<td>Change of voidage during consolidation.</td>
<td>44</td>
</tr>
<tr>
<td>3.13</td>
<td>Force balance of Jenike Cell on vibrating plate.</td>
<td>44</td>
</tr>
<tr>
<td>3.14</td>
<td>Oscillation of total shear force.</td>
<td>48</td>
</tr>
<tr>
<td>3.15</td>
<td>Force balance of Jenike Cell on tilted vibrating plate.</td>
<td>48</td>
</tr>
<tr>
<td>3.16</td>
<td>Occurrence of slippage from combined shear and inertia forces.</td>
<td>50</td>
</tr>
<tr>
<td>4.1</td>
<td>Fine Ore Bins. Hopper Geometry.</td>
<td>69</td>
</tr>
<tr>
<td>4.2</td>
<td>No.2 Ore Screening Station Fines Bins. Hopper Geometry.</td>
<td>69</td>
</tr>
<tr>
<td>4.3</td>
<td>Approximate calculation of valley angle $\theta$.</td>
<td>76</td>
</tr>
<tr>
<td>4.4</td>
<td>Comparison of Mass Flow Hoppers for 72hr. Mt. Newman 8.1% w.b.</td>
<td>81</td>
</tr>
<tr>
<td>4.5</td>
<td>Geometry of a wedge hopper bin.</td>
<td>84</td>
</tr>
<tr>
<td>5.1a</td>
<td>General arrangement of mechanical shaker and drive.</td>
<td>91</td>
</tr>
<tr>
<td>5.1b</td>
<td>Mechanical shaker general arrangement.</td>
<td>92</td>
</tr>
<tr>
<td>5.1c</td>
<td>Arrangement of out-of-balance device with counter rotating masses.</td>
<td>92</td>
</tr>
<tr>
<td>5.2a</td>
<td>Arrangement of Electro-mechanical vibrator.</td>
<td>95</td>
</tr>
<tr>
<td>5.2b</td>
<td>Vibrator connected to Jenike Shear Cell.</td>
<td>95</td>
</tr>
<tr>
<td>5.3a</td>
<td>Arrangement of gravity device with vibrator.</td>
<td>97</td>
</tr>
<tr>
<td>5.3b</td>
<td>Gravity device and vibrator set up for wall yield tests.</td>
<td>97</td>
</tr>
<tr>
<td>Fig. No.</td>
<td>Description</td>
<td>Page No.</td>
</tr>
<tr>
<td>---------</td>
<td>--------------------------------------------------</td>
<td>----------</td>
</tr>
<tr>
<td>5.4</td>
<td>Schematic of overconsolidation peak.</td>
<td>107</td>
</tr>
<tr>
<td>5.5</td>
<td>Conical mass flow hopper.</td>
<td>107</td>
</tr>
<tr>
<td>7.1</td>
<td>Conveyor head chute.</td>
<td>139</td>
</tr>
<tr>
<td>7.2</td>
<td>Tripper dogleg chute.</td>
<td>139</td>
</tr>
<tr>
<td>No.</td>
<td>Description</td>
<td>Page No.</td>
</tr>
<tr>
<td>-----</td>
<td>------------------------------------------------------------------------------</td>
<td>----------</td>
</tr>
<tr>
<td>1.</td>
<td>Instantaneous Yield Loci. Mt. Newman Fines 8.1% w.b.</td>
<td>57</td>
</tr>
<tr>
<td>2.</td>
<td>Wall Yield Loci. Mt. Newman Fines 8.1% w.b.</td>
<td>58</td>
</tr>
<tr>
<td>3.</td>
<td>Flow Functions. Mt. Newman Fines 8.1% w.b.</td>
<td>60</td>
</tr>
<tr>
<td>4.</td>
<td>Time Yield Locus. Mt. Newman Fines 8.1% w.b.</td>
<td>64</td>
</tr>
<tr>
<td>6.</td>
<td>Flow Functions. Cockatoo Fines, varying moisture contents.</td>
<td>66</td>
</tr>
<tr>
<td>7.</td>
<td>Flow Functions. Flue dust.</td>
<td>67</td>
</tr>
<tr>
<td>8.</td>
<td>Effect of Vibration on Shear Strength.</td>
<td>99</td>
</tr>
<tr>
<td>9.</td>
<td>Effect of Vibration on Yield Locus.</td>
<td>102</td>
</tr>
<tr>
<td>10.</td>
<td>Effect of Density on Shear Strength.</td>
<td>104</td>
</tr>
<tr>
<td>11.</td>
<td>Prolonged Shear of Overconsolidated Cells gives S3 Yield Loci.</td>
<td>109</td>
</tr>
<tr>
<td>12.</td>
<td>Vibrated Wall Yield Locus. Mt. Newman Fines 0.9% w.b. using Jenike Machine.</td>
<td>112</td>
</tr>
<tr>
<td>13.</td>
<td>Vibrated Wall Yield Locus. Mt. Newman Fines 9.3% w.b. using Jenike Machine.</td>
<td>113</td>
</tr>
<tr>
<td>14.</td>
<td>Vibrated Wall Yield Locus. Cockatoo Fines 7.8% w.b. using Jenike Machine.</td>
<td>114</td>
</tr>
<tr>
<td>15.</td>
<td>Vibrated Wall Yield Locus. Flue Dust 9.9% w.b. using Jenike Machine.</td>
<td>115</td>
</tr>
<tr>
<td>16.</td>
<td>Vibrated Wall Yield Locus. Mt. Newman Fines 0.9% w.b. using Tilting Device.</td>
<td>116</td>
</tr>
<tr>
<td>18.</td>
<td>Vibrated Wall Yield Locus. Cockatoo Fines 8.9% w.b. using Tilting Device.</td>
<td>118</td>
</tr>
<tr>
<td>19.</td>
<td>Vibrated Wall Yield Locus. Flue Dust 17.4% w.b. using Tilting Device.</td>
<td>119</td>
</tr>
<tr>
<td>No.</td>
<td>Description</td>
<td>Page No.</td>
</tr>
<tr>
<td>-----</td>
<td>------------------------------------------------------------------------------</td>
<td>----------</td>
</tr>
<tr>
<td>20.</td>
<td>Instantaneous Yield Locus. Mt. Newman Fines 0.9% w.b.</td>
<td>147</td>
</tr>
<tr>
<td>21.</td>
<td>Wall Yield Locus. Mt. Newman Fines 0.9% w.b.</td>
<td>148</td>
</tr>
<tr>
<td>22.</td>
<td>Instantaneous Yield Locus. Mt. Newman Fines 5.6% w.b.</td>
<td>149</td>
</tr>
<tr>
<td>23.</td>
<td>Wall Yield Locus. Mt. Newman Fines 5.6% w.b.</td>
<td>150</td>
</tr>
<tr>
<td>24.</td>
<td>Instantaneous Yield Locus. Cockatoo Fines 0.3% w.b.</td>
<td>151</td>
</tr>
<tr>
<td>25.</td>
<td>Wall Yield Locus. Cockatoo Fines 0.3% w.b.</td>
<td>152</td>
</tr>
<tr>
<td>26.</td>
<td>Instantaneous Yield Locus. Cockatoo Fines 4.0% w.b.</td>
<td>153</td>
</tr>
<tr>
<td>27.</td>
<td>Wall Yield Locus. Cockatoo Fines 4.0% w.b.</td>
<td>154</td>
</tr>
<tr>
<td>28.</td>
<td>Instantaneous Yield Locus. Cockatoo Fines 7.8% w.b.</td>
<td>155</td>
</tr>
<tr>
<td>29.</td>
<td>Wall Yield Locus. Cockatoo Fines 7.8% w.b.</td>
<td>156</td>
</tr>
<tr>
<td>30.</td>
<td>Instantaneous Yield Locus. Cockatoo Fines 8.9% w.b.</td>
<td>157</td>
</tr>
<tr>
<td>31.</td>
<td>Wall Yield Locus. Cockatoo Fines 8.9% w.b.</td>
<td>158</td>
</tr>
<tr>
<td>32.</td>
<td>Instantaneous Yield Locus. Flue Dust 1.1% w.b.</td>
<td>159</td>
</tr>
<tr>
<td>33.</td>
<td>Wall Yield Locus. Flue Dust 1.1% w.b.</td>
<td>160</td>
</tr>
<tr>
<td>34.</td>
<td>Instantaneous Yield Locus. Flue Dust 5.3% w.b.</td>
<td>161</td>
</tr>
<tr>
<td>35.</td>
<td>Wall Yield Locus. Flue Dust 5.3% w.b.</td>
<td>162</td>
</tr>
<tr>
<td>36.</td>
<td>Instantaneous Yield Locus. Flue Dust 9.9% w.b.</td>
<td>163</td>
</tr>
<tr>
<td>37.</td>
<td>Wall Yield Locus. Flue Dust 9.9% w.b.</td>
<td>164</td>
</tr>
<tr>
<td>38.</td>
<td>Instantaneous Yield Locus. Flue Dust 13.4% w.b.</td>
<td>165</td>
</tr>
<tr>
<td>No.</td>
<td>Description</td>
<td>Page No.</td>
</tr>
<tr>
<td>-----</td>
<td>--------------------------------------------------</td>
<td>----------</td>
</tr>
<tr>
<td>39.</td>
<td>Wall Yield Locus. Flue Dust 13.4% w.b.</td>
<td>166</td>
</tr>
<tr>
<td>40.</td>
<td>Instantaneous Yield Locus. Flue Dust 17.4% w.b.</td>
<td>167</td>
</tr>
<tr>
<td>41.</td>
<td>Wall Yield Locus. Flue Dust 17.4% w.b.</td>
<td>168</td>
</tr>
</tbody>
</table>
SECTION 1

INTRODUCTION

The flow of bulk solids is of vital importance to many parts of industry today, and with greater demands on productivity and reliable flow the technical side is being thoroughly researched. The author has seen many of the plant problems inherent in bad design of materials handling plant and believes that many of these problems could be alleviated by closer attention to the theory. The use of mechanical aids, air lances, jack hammers, vibrating aids and even explosives is common in an effort to get materials flowing from bins and hoppers.

As no information on flow characteristics of some of the more difficult materials used in Steelmaking - Australian fine iron ores - was available for design work, a study of the flow characteristics of the more difficult materials was undertaken - these being Mt. Newman fine Iron ore, Cockatoo fine Iron ore and Flue Dust (the dust extracted from the Blast Furnace Dustcatcher). Using this information calculations were then done on hoppers in the Raw Materials and Sinter Plant area to ascertain whether, for the worst materials, they were mass flow, core flow or no-flow bins. Particular interest was shown in the feed bins for the new No. 3 Sinter Machine because any reduction in capacity due to material build up has critical effects on the blend mix as well as the emptying time for these bins.

Little work of practical application has been
done on the effect of vibration on the flow of raw materials. When materials will not flow from a bin, the use of various vibrators is tried with little knowledge of the consequences. Research was undertaken into how vibration of a surface reduces the friction between itself and a bulk material in contact with it.

Vibration of a confined sample was also studied to determine the effect on the strength of the sample. Further tests were carried out to determine what effect frequency, extra weight and longer vibration times had on strength. This work was undertaken to try to improve the practical knowledge available on how vibration affects material flow.

The objectives of the work can be summarised as follows:

1. To obtain flow functions for the strongest bulk solid materials handled by Australian Iron & Steel in their Raw Materials Section and to use this information to check whether the bins used in that section are mass-flow, core-flow or no-flow for those materials. These design checks can then be confirmed from actual plant experience.

2. To determine what effect vibration has on the strength of a confined sample and to examine how changing frequency, normal force and the length of vibration affects the strength.
3. To determine what effect vibration of a surface has on the frictional forces between itself and a material. This has particular relevance to trying to predict the performance of vibrating baffles and vibro feeders which are commonly used in industry. These devices rely on either a decrease in wall friction or on cyclic acceleration to throw the particles in small jumps towards the outlet.
The theory of the gravity flow of bulk solids is largely a derivative of soil mechanics theory, because of the many similarities in the two subjects. Jenike, Ref.(1), has pioneered the recent work in this field and has developed a direct shear tester which can be used for testing both cohesive and non-cohesive powders. He developed a theory for flow assuming a rigid plastic material and based his predictions on the flow - no flow postulate; "flow will occur if the material cannot develop sufficient strength to support obstructions to flow". The theory was developed to predict the maximum consolidating stresses acting in a material and the maximum stresses acting in an obstruction from experimental measurements for the material. If the maximum consolidating stresses are larger than the peak stresses which would just cause failure in an obstruction, then flow will occur. From this theory, mass flow and core flow can be predicted for hoppers of various shapes if the flow properties of the material are known. The theory is explained in detail in Section 3.1, as it is used extensively in Section 4.

Jenike's work intuitively related the density of a given material at a certain moisture content and temperature to the material's behaviour under normal and shearing forces - by obtaining a steady shear value during the consolidation process the critical density can be obtained for those normal
and shear forces. Other authors have concentrated on this aspect to try to relate the density to the normal and shear forces being applied.

Hvorslev, Ref.(2), showed in 1937 that for saturated cohesive soils, the peak shear stress at failure is a function of the effective normal stress on, and the voids ratio (or density) in, the plane of failure at the moment of failure and is independent of the stress history of the sample. Roscoe et al, Ref.(3), extended this to show that Hvorslev's equation for failure defines a failure surface in a space of three variables, shear stress, normal stress and voidage (Fig. 2.1). Under loading a sample reaches this surface and then as failure occurs, the sample follows a path across the surface until it reaches a unique critical voids ratio line. At this critical voids ratio state, unlimited deformation occurs with no change in the three variables. Roscoe and co-workers verified the existence of the failure surface and the critical voids ratio line by the analysis of triaxial tests on a clay.

In addition, they showed from testing that the C.V.R. line is not vertically above the normal consolidation line, but that a three dimensional yield surface exists which emanates from the normal consolidation line and intersects the Hvorslev failure surface at the C.V.R. line, Fig. (2.2).

To obtain readings of the voids ratio in the failure plane at the moment of failure, a special apparatus was designed which applied uniform strain to the sample. The
**Fig. (2.1)**

Hvorslev failure surface

**Fig. (2.2)**

Failure & yield surface
average voids ratio of the whole sample could then be equated to the failure plane voids ratio. The tests of Roscoe and co-workers showed that the concepts of the three dimensional failure surface and the critical voids ratio are applicable to cohesionless as well as cohesive media. Ashton et al, Ref.(4), proposed a similar three dimensional surface with a yield and a consolidation surface intersecting at the critical voids ratio line, with the exception that the surface extended into the tensile region, Fig.(2.3). Their results showed a functional relationship between bulk density and strength for a given powder at constant humidity.

Williams and Birks, Ref.(5), continued in the vein of Ref.(3) and proposed that the Jenike consolidation procedure was actually attaining the critical density for a given normal load, because, for correctly consolidated samples, there was virtually no density change during shearing. The details of their predictions are contained in Section 3.2. They also developed an alternate method for correctly consolidating samples in the Jenike shear cell, in which samples which would not fail under normal consolidation had heavier normal loads applied to them during twisting. The stress intercept of each line plotted in the plane of maximum shear stress versus the ratio of twisting weight/shearing weight gave the critical compaction load for that shearing weight, Fig.(2.4). That load, if used in compaction, would cause failure in shear to occur with negligible change of density. This method is only used if a sample cannot be made to fail by the normal Jenike consolidation method.
Consolidation and yield surfaces in $(\gamma, \sigma, \tau)$ space.
Critical compaction overload graph.

Fig. (2.4)

Reduction of a family of yield loci to a single line, using reduced shear stress and reduced compound stress on logarithmic scales.

Fig. (2.5)
A second paper by Williams and Birks, Ref.(6), showed that some powders have appreciable tensile strength as shown by Ashton et al, Ref.(4). They proposed substituting the sum of the applied normal stress and the tensile stress - the "compound normal stress" in place of the applied normal stress for shear failure tests. This allowed the angle of internal friction of the powder to be directly read from the yield loci graphs, where this angle is a measure of internal friction of the powder during flow with no volume changes.

The authors propose several means of simplifying the yield loci graphs and for a dry powder illustrated that a family of yield loci could be reduced to a single straight line by the use of the compound stress concept, dimensionless stresses and logarithmic scales, Fig.(2.5). The Hvorslev failure surface of a dry powder can be reduced similarly to a single straight line by the use of compound stress, dimensionless stress and logarithmic scales. Experimental results verified the reduction of a family of yield loci to a single straight line.

Lee, Ref.(7), reviews some bin design in a practical paper and recommends that straight sided bins should not be used with troublesome materials due to arching. This effect is confirmed by Jenike who suggests that the included angle of a vertical sided hopper is not to exceed 75% of the allowed conical hopper included angle, Fig.(2.6).
Included angle of vertical wall hoppers should not exceed $1.5 \theta'c$.

Suggested theory using sections of Janssen, Walker and Walters, from Clague's experimental results.
Wright, Ref.(8), tested the Jenike hopper design method with both a variable geometry wedge hopper and a conical hopper. He concluded that the method was accurate for mass flow bins operating under dynamic flow conditions and even showed 5 - 10° overdesign for the wedge hopper. There were two deficiencies which Wright found in the theory:

(a) It does not allow for the impact loading which may occur during loading and can cause arching.

(b) It does not give a design which is certain to eliminate the possibility of arching at the transition in the hopper.

Walker, Ref.(9), gave a theory for the approximate stresses occurring throughout a bin both on filling and during flow for wedge hoppers and conical hoppers. This theory predicts the pressure in the vertical and converging sections.

Clague and Wright, Ref.(10), tested the theory of Walker for both wedge hoppers and conical hoppers and found the wall pressures to be in reasonable agreement with the theory for a variety of flow conditions and loading conditions, including impact loading. The point brought out by Ref.(8), that Jenike does not allow for impact loading, was again confirmed by the results. The pressure measurements showed that bins having a high vertical section above the hopper experience up to five times the predicted Janssen pressure, Ref.(11), at the transition zone when material is flowing.
Since the Janssen theory is a popular one, this fact may account for some of the structural failures at the transition zone. The pressures due to impact loading could be reduced by:-

(a) Ensuring that when the lower part of the hopper is being filled there is some material flow out of the hopper outlet to keep the material 'live'.

(b) Using an impact breaker in the bin.

Walters, Ref.(12) and (13), extended the work of Walker, Ref.(9), in developing a theoretical analysis of the stresses acting at the walls of mass flow bins, for silos with vertical walls and axially symmetric hoppers.

Arnold and Roberts, Ref.(14), showed some deficiencies in the Walters theory, and proposed a simpler theory using sections of Walker, Ref.(9), Janssen, Ref.(11), Clague and Wright, Ref.(10), and Walters, Ref.(12) and (13). Fig. (2.7).

Jenike, Johanson and Carson, Ref.(15) and (16), applied the principle of minimum recoverable strain energy to determine the stresses acting at vertical bin walls. Their procedure is difficult to apply and relies on data in chart form to satisfy the range of bin configurations.
Perry and Jangda, Ref.(17), extended the work of Clague and Wright, Ref.(10), by testing the pressures in flowing and static sand, both at the walls and in the material interior by using pressure sensitive radio pills which transmit signals to a recorder outside the bin. They confirmed the findings of Clague and Wright in that mass flow bins with conical hoppers experience high stresses at the transition point during flow. The wedge hopper does not lead to such high stresses at the transition zone, so is to be preferred. They tested core flow bins and found that they have lower dynamic pressures, but do not give reliable flow. Early in this work it was found that the filling rate affects the static pressures in the bin, but once flow commenced the fill rate had no effect on the dynamic pressure. The work confirmed that the order of pressures developed can be calculated by the theory of Jenike, Johanson and Carson, Ref.(15) and (16) or Walker, Ref.(9).

A paper by Lakshman Rao and Venkateswarlu, Ref.(18), did, in a similar fashion to Clague and Wright, Ref.(10), a series of tests on static and dynamic wall pressures for mass flow bins. With tests on conical and wedge hopper bins the authors reinforced the findings of Clague and Wright, Ref.(10), and Perry and Jandga, Ref.(17). They showed that Janssen's wall pressures, Ref.(11), were in good agreement with the static and dynamic pressures in the vertical section, and that Jenike et al, Ref.(15) and (16) and Walker's, Ref.(9), pressures gave fair agreement also, particularly in the transition region.
These papers - Ref.(8), (9) and (17) have pointed out two major deficiencies in the Jenike theory enunciated in Ref.(1) :

a. It does not predict the effect of impact loading.

b. It does not ensure that arching will not occur at the hopper transition due to the high localized pressures.

Johanson, Ref.(19), reinforces the use of an impact breaker as recommended by Ref.(10). He proposes a solution to the problem of changing existing core flow bins to mass flow without extensive modifications by the use of flow-corrective impact breaking inserts, Fig.(2.8). The inserts are shown to expand the flow out to the walls to produce mass flow underneath and also act as pressure breakers for material being impact loaded.

Another solution to the problem of converting core flow bins to mass flow bins lies in the use of vibration. Takahashi et al., Ref.(20), found that for cohesionless materials, vertical sinusoidal vibration applied to a hopper caused the material within to cycle and mix rapidly. The movement was upwards along the wall of the vessel and down towards the bottom through a central core, Fig.(2.9). Quantitative measurements were taken which showed the circulating velocity to increase uniformly with increasing frequency. The zone of descending particles was thought to be fluidized, while the ascending particles acted as a solid
Fig. (2.8)
Conical insert expands flow out to walls to produce mass flow.

Fig. (2.9)
Vertical vibration of hopper and cohesionless contents causes cyclic mixing in definite zones.
block. A theory based on these points showed reasonable agreement with practical measurements. The effect of the vibration is to make the entire hopper 'live' thus eliminating core flow.

A further paper by the same authors, Ref.(21), showed that a certain vertical vibration level attains the greatest consolidation of particles, while less violent or more violent vibration causes less compaction. When vertical vibration was applied to a hopper with an orifice using cohesionless materials, vibration was shown to be effective for an outlet smaller than that required for gravity discharge, particularly at higher frequencies. Vibration had little major effect for an outlet size larger than that required for gravity discharge, and in fact, at low frequencies the discharge was less than the gravity rate. In general, discharge rate increases with frequency but peaks at a certain acceleration rate. The authors developed a theory from the motion of a particle on a vibrating plate, which confirmed the experimental results reasonably well.

Suzuki and Tanaka, Ref.(22), studied the discharge of cohesive solids from a vertical vibrating hopper. Assuming core flow occurred, they developed a theory for the critical vibration intensity to commence flow, and also derived a flow rate equation. The flow rate tests showed that low frequency or high intensity of vibration gave the best flow rates, with good theoretical agreement. This is in contrast to the results for cohesionless materials,
Ref. (21), which showed that high frequency gave the best
discharge rates. Further findings from Suzuki and Tanaka
were that increasing intensity of vertical vibration gave a
higher density in the particle bed and that increasing
density gave a linear increase in shear strength and wall
shear strength.

Scarlett and Eastham, Ref. (23), carried out some
excellent work on the way shock waves from vibration were
transmitted in a bed of granular material, in this case
coarse and fine sand. Vertical vibration was applied at the
bottom of a particle bed, and an output transducer used to
record the vibration strength at the top of the bed, Fig.
(2.10). It was found that the vibration reaching the top
surface was strongest vertically above the energy source, and
fell off in strength until there was no vibration received
outside a 45° cone emanating from the energy source. The
velocity through the bed was independent of the height of
the bed and the size fraction of the sand used, but was
greatest for the vertical direction while decreasing for
increasing angles of inclination. This effect was thought
to be due to the shock wave being passed between particles
in direct contact vertically, but more through frictional
forces for angles of inclination from the vertical. It was
further shown that high frequency vibration gave better
penetration than low frequency vibration, and the vibration
strength was found to decay exponentially with bed height.
A theory based on statistical energy transfer between
particles in 3 dimensions gave reasonable agreement between
Fig. (2.10)
Experimental apparatus of Scarlett and Eastham, Ref. (23).

Fig. (2.11)
Bin activator with vibrating baffle.
experimental and theoretical results for the intensity of vibration at the top surface.

Gray and Rhodes, Ref.(24), examined the mechanism of vibratory energy transfer both experimentally and theoretically for vertical vibration. They modelled the beds as (i) plastic bodies, and (ii) visco-elastic bodies. The authors found that at frequencies less than 150 Hz, the powder acts as a coherent mass and is projected from the container base, subsequently colliding with it. This had the effect of making the motion of the electromagnetic vibrator non sinusoidal. Other non-linearities with the electromagnetic vibrator caused the energy transferred to the powder to be frequency dependent. The authors felt that much of the literature reporting the frequency dependence of powder behaviour is due to the non linearity of electromagnetic vibrators used. Both models predicted a decrease in the energy transferred to the powder as acceleration is raised at constant frequency, but the models fail when the powder ceases to behave as a coherent mass. This occurs at frequencies over 150 Hz, accelerations over 10 g or when the bed is fluidized by vibration. Within the model's limits the visco-elastic model was superior, because it allowed some parameters of the vibrator to be included and did not assume instantaneous reversal of motion. Both models incorporated on air drag term for when the bed is in free flight, and thus are superior to the single plastic body subjected to vibratory forces only, as assumed by Ref.(20), (21) and (22).
Three practically oriented papers by Myers, Ref.(25), Wahl, Ref.(26), and Carroll and Colijn, Ref.(27), produced some further recommendations to improve material flow. Myers recommends cycling the vibration from a vibrating hopper as being preferable to continual vibration, as it does not produce excessive consolidation and the momentary tensile stresses occurring tend to collapse arches and pipes. This cycling would produce considerable fatigue particularly in the transition zone of the hopper as switch stresses are produced each 2 - 5 seconds. Myers found that high amplitude, low frequency vibration was most effective for discharge, concurring with the findings of Suzuki and Tanaka, Ref.(22) for materials which are cohesive. Materials not recommended for use in vibrating hoppers were sludges, materials with high consolidation strengths and those with low densities.

A similar paper by Wahl, Ref.(26), described a bin activator comprised of a curved baffle just above the outlet which is vibrated horizontally by a mechanical gyrator, Fig.(2.11). The curved baffle is said to eliminate overhead bridging by sending vibrations vertically up the hopper and causes flow to occur in at least the area vertically above the bin activator. An effect not mentioned is the reduction in wall friction on the surface of the baffle which will enhance material flow off the edges of the baffle.

Carroll and Colijn, Ref.(27), wrote an excellent review on the vibratory equipment available for improved flow and for flow metering from hoppers. Bin vibrators
were discussed initially, being devices used to dislodge bulk solids from the bin walls and promote flow. Mechanical out-of-balance devices, electromagnetic vibrators, pneumatic vibrators and even a sonic vibrator were mentioned. The location should be in the area where arching occurs. For breaking arches, low frequency high amplitude vibration was recommended, while for fine compacting materials or to induce mass flow in a core flow bin, medium frequency (60-120 Hz) and low amplitude was preferred. For complete hopper cleanout, high frequency and low amplitude vibration is suggested.

Bin bottom activators discussed, such as that described by Wahl, Ref.(26), consisted of a suspended and isolated hopper bottom with a baffle or cone inside either vibrating by itself or together with the whole hopper bottom. The feed rate increased with stroke and frequency. The baffle inside was said to act as an impact breaker, (which both Ref.(10 and (19) recommend), and furthermore produces a large flow channel in the upper bin. Low frequency and high amplitude vibration gave the best results with the bin bottom activators.

Carroll and Colijn proposed four areas of useful application for vibration:-

(a) To reduce wall friction to promote mass flow for shallow walls. For this case, high frequency (60-100 Hz) and low amplitude is recommended.
(b) To break up time consolidated material or to encourage flow through openings which would arch under gravity flow. To break up the material the vibration must penetrate well and medium frequency (30 - 60 Hz) and amplitude is recommended.

(c) To deaerate and densify aerated powders to prevent flooding, deep penetration at low energy levels is required.

(d) To discharge very sticky or fibrous materials, deep penetration at high energy levels is required.

One particular vibrator was a "Bridge Breaker", a vibrating panel inside the bin which vibrates along the wall, Fig.(2.12). It will be shown in Section 5 later, how this type of device reduces the wall friction considerably.

The second part of the paper described vibrating feeders, which are used to meter flow from a hopper, Fig.(2.13). Since they are isolated from the hopper the only vibration passed to the hopper is through the mounting connections or via the material itself. Because of this a vibrating feeder fitted to a core flow bin is unlikely to change the bin's flow characteristics.

Some of the properties of feeders are relevant to this work; for example the curved baffle of the bin activator in Ref.(26) is virtually a feeder operating in a bin. Feeders have a trough under the bin outlet supporting the material
Fig. (2.12)
"Bridge Breaker" reduces wall friction by vibrating along the hopper wall.

Fig. (2.13)
Vibrating feeder - used to meter flow from a hopper.
and acceleration is used to throw the material down the trough in a series of jumps. The feed rate is generally linear and almost directly proportional to the product of frequency and stroke. Fine particle beds, particularly those which aerate, feed slower and require less deep beds than granular material. This would be explained by the lower penetration of vibration through fines than roughs, as found by Scarlett and Eastham in Ref. (23). It was also interesting to note that while the free-flowing material feed rate increased directly with increasing stroke, the feed rate for aerating material had a non-linear curve with an optimum stroke.

Leitzel and Morrisey, Ref. (28), in a general outline of air float conveyors showed that aeration could be used to clear a large core flow bin. The material in use should contain 100% minus 850 micron particles and 10-15% minus 74 micron particles, with the moisture content below 1%. Higher moisture contents can be conveyed, as well as coarser materials as long as there is a greater percentage of fine material to fill the voids. Open type air float conveyors are used in several large storage silos to transport material to the opening, in effect making a mass flow bin out of a core flow bin, Fig. (2.14).

Emery, Ref. (29), showed that what was a core flow bin, could be converted to a mass flow bin by the insertion of an aerating cone at the outlet, Fig. (2.15). The aerating cone acts as a system of open air conveyors moving material to the outlet. Because of this a hopper angle of $\theta^{\circ}c = 40^\circ$
OPEN AIR FLOAT CONVEYORS SET IN FLOOR OF STORAGE SILO.

Fig. (2.14)
Open air float conveyors transform core flow bin into mass flow bin.

Fig. (2.15)
Aeration cartridge changes pressurised core flow bin into mass flow bin.
with resulting low head room is obtainable with mass flow. The hoppers used were pressurized.

Sutton and Richmond, Ref.(30), studied the de-aeration of powders in a hopper. By preventing de-aeration of aerated powders smooth flow from small outlets can be obtained because the powder does not consolidate. The experimental program showed that once de-aeration has occurred it is impossible to re-aerate the powder effectively. The authors proposed the use of aerated bins of squat design which could produce mass flow from outlets smaller than allowed by gravity flow. Again the material which could be used in this type of bin is very limited.

Ref.(31), an anonymous article in the journal "Material Handling", describes the use of a sonic actuator to induce flow in non-sticky materials. The device fluidizes the bed and also has a self cleaning effect on the hopper walls because the sound reverberates in the empty hopper.
SECTION 3
THEORETICAL CONSIDERATIONS

3.1 Flowability Concepts

The tests of flowability of local iron ores were performed using the theory enunciated by Jenike (Ref. 1). His theory is based on the flow-no flow postulate. This states, if the material has insufficient strength to support an obstruction to flow, then flow will occur. If the material develops sufficient strength under consolidating forces to form a dome or pipe, then no flow will occur. The theory details how the variation of shear strength of a material varies with the consolidating forces applied to it, the time of consolidation and the moisture content of the material. By graphing the consolidation forces versus shear forces for a given material, and obtaining from Jenike the flowability conditions for a particular hopper of opening size $D_H$, slope $\theta'$ and wall friction $\theta'$, it can be predicted whether flow will occur or not.

Following an element during flow as it passes down a bin, Fig. (3.1), it can be seen that the major and minor stresses $(\sigma_1, \sigma_2)$ acting on the element first increase, pass a maximum then decrease. During this time, the shape of the element changes, causing a change in density of the element.

Experiment has shown that the ratio of major and minor stresses $\sigma_1/\sigma_2$ varies little under flow, as
Fig. (3.1)
Pressure acting on a flowing element.

Fig. (3.2)
Effective Yield Locus.
long as moisture content and temperature remain constant.

That is,

\[
\frac{\sigma_1}{\sigma_2} = \frac{1 + \sin \phi}{1 - \sin \phi} \quad (H = \text{constant}, \ T = \text{constant}) \quad \ldots \ldots (3.1)
\]

Angle \( \phi \) is the effective angle of friction and for a given solid varies only a few degrees with consolidating pressure. Because \( \frac{\sigma_1}{\sigma_2} \propto \) constant, this relation gives a straight line passing through the origin. This envelope is called the "effective yield locus" or E.Y.L. for short. (Fig. 3.2). In testing it is simpler to use force instead of pressures.

So transforming,

\[
V_1 = A \sigma_1 \quad \ldots \ldots (3.2)
\]

\[
V_2 = A \sigma_2 \quad \ldots \ldots (3.3)
\]

\[
\frac{V_1}{V_2} = \frac{1 + \sin \phi}{1 - \sin \phi} \quad \ldots \ldots (3.4)
\]

where \( A = \) area of shear cell tester

Under pressure some solids attain strength, due to consolidation.

The higher the consolidation forces, the greater strength the material attains. Some materials do not gain strength with pressure such as hard lumpy materials e.g. Iron ore roughs, or dry sand. To shear such materials only their angle of internal friction \( \phi \) has to be overcome. This leads to a yield locus given by:

\[
S = V \tan \phi \quad \ldots \ldots (3.5)
\]

that is slip occurs whenever the shear force \( S \) attains the value of \( V \tan \phi \). For such solids the angle of internal friction \( \phi \) equals the effective angle of
friction $\phi$, Fig.(3.3). For cohesive solids, which will be used in this thesis, each consolidation at a given normal load gives one family of yield loci. Thus if a solid is flowing under a certain pressure and if a section of that solid is sheared under a vertical load $\overline{V}$ by a force $\overline{S}$, then point $(\overline{V}, \overline{S})_1$ defines one point on the yield locus , Fig.(3.4). If a larger normal force is then used for that same element, point $(\overline{V}, \overline{S})_2$ might result. The yield locus must also be tangential to the Mohr force circle for those consolidating conditions, because during flow the section is continuously at yield.

Similarly for a lower consolidating pressure, the lower yield locus defined by the point $(\overline{V}, \overline{S})_3$ and $(\overline{V}, \overline{S})_4$ is drawn. Each yield locus terminates at the point of tangency with the Mohr circle for the consolidating conditions, because that circle defines the maximum forces in the section. The effective yield locus E.Y.L. forms an envelope of these Mohr circles describing the consolidating conditions.

The kinematic angle of internal friction $\phi$ is the angle between the yield locus and the axis $V$. Since the yield loci are generally convex curves, this angle varies with the consolidating pressure and the normal force $V$.

**Time Yield Locus (T.Y.L.)** Once a solid is left for a period of time $t$ under consolidating forces, the solid tends to gain strength. This means the yield locus after a period of time (time yield locus) is above the yield locus for no time delay , Fig. (3.5).
Fig. (3.3)
Yield Locus of a cohesionless solid.

Fig. (3.4)
Family of Yield Loci of a cohesive solid.
Unconfined Yield Forces When an obstruction to flow such as a dome is about to collapse, it is most likely that failure starts at the exposed surface. The stresses are simple to analyse because the normal force is zero and the major principal force is tangential to the surface. When the force causes yield, it is referred to as the unconfined yield force $F$. The corresponding Mohr force circle is tangential to the yield locus and passes through the origin.

Static Angle of Internal Friction $\phi_t$ For core flow, it is necessary to know the angle of internal friction within the solid at an exposed surface. This angle is measured where the T.Y.L. is tangential to the Mohr circle, for the unconfined yield force $F_t$.

Wall Yield Locus (W.Y.L.) When a solid slides along the wall of a channel the stress conditions at the wall become significant. The normal and shearing forces at the wall during flow are given by $(V', S')$ respectively, (Fig. 3.6).

Kinematic Angle Of Friction Between A Solid And A Wall, Angle $\phi'$ Since the W.Y.L. is usually curved, the angle $\phi'$ varies with the normal force, $V$. Generally the force $V$ at which the angle $\phi'$ is needed is $V_1$, the point of intersection of the material Flow Function $FF$ and the hopper flow factor $ff$. The angle is found by drawing a Mohr force circle on the W.Y.L. with the principal force $V_1$ and tangent to the Effective Yield Locus. The angle between the origin and the larger intercept of the Mohr circle with the W.Y.L.
Fig. (3.5)

Time Yield Locus.

Fig. (3.6)

Wall Yield Locus.
gives the kinematic angle of friction $\phi'$ corresponding to $V_1$.

**Flow Function**  Thus for a solid with a given moisture content $H$, temperature $T$ and with zero consolidation time, its yield locus and unconfined yield force $F$ become a function of the consolidation of the solid only. If the effect of the third principal pressure is neglected, then:

$$F = f(V_1) \ (t = 0, \ H = \text{constant}, T = \text{constant}) \ .... \ (3.6)$$

That is the unconfined yield force $F$ and the yield locus are a function of the major consolidating force $V_1$ only. This relation is called the flow-function of a solid, $(FF)$, (Fig.3.7). The higher the flow function, the stronger the material and the easier it is to support obstructions to flow. Thus $FF(c)$ represents the strongest material, while $FF(a)$ represents the weakest. Any solid which is cohesionless has a flow function coinciding with the $V_1$ axis, as it has no unconfined yield strength. The three points $F$ and $V_1$ are required to draw one flow function curve for a given temperature $T$ and moisture content $H$. (Fig. 3.8) If the flow of the solid is interrupted for some time $t_1$ the flow-function $FF$ changes because the material gains strength. This function is called a time flow-function and it has the relation:

$$F_t = f(V_1, t) \ (H, T = \text{constant}) \ .... \ (3.7)$$

Again if moisture or temperature change, the relation becomes general in that:
Fig. (3.7)
Flow Functions of a solid.

Fig. (3.8)
Family of Yield Loci required to draw Flow Function.
\[ F_t = f(V_1, t, H(t), T(t)) \quad \ldots \quad (3.8) \]

That is \( F_t \) is a function of normal force, time, temperature and moisture content.

The changes of a material's strength with time could be caused by one or more of the following:-

1) Escape of entrained air with an increase in density.
2) Migration of water.
3) External vibrations causing a rearrangement of particles with corresponding density increase.
4) Evaporation of free water causing precipitation of dissolved salts which then cement the particles.
5) Break up of particles under pressure causing an increase in the surface of contact and cohesion.
6) Changes in the surface of particles due to crystallization, fermentation or whatever.

In general, solids without particles under 850 microns are free flowing, although there are some exceptions. Flow conditions of a solid are generally governed by the fines in the solid, because shearing takes place across the fines, while the roughs are a passive agent.

**Flowability of Channels - Flow Factor \( ff \)**

In general a solid will flow if a dome does not develop across the channel. Also for a core flow channel, the solid should be unable to sustain an empty vertical pipe of excessive height. For an obstruction to be stable, the solid must be consolidated sufficiently to develop enough strength to support the weight of the obstruction. Thus the lower the pressures \( \sigma \) acting in
an obstruction, and the higher the consolidating pressures \( \tau \) in the channel, the lower the flowability of that channel.

Defining flow factor by:

\[
ff = \frac{\sigma_i}{\sigma_t} \quad \ldots \quad (3.9)
\]
or in force co-ordinates

\[
ff = \frac{V_1}{V_t} \quad \ldots \quad (3.10)
\]
then the lower \( ff \) values give better flowability. Jenike (Ref. 1) has computed \( ff \) values for conical channels and plane flow channels, both symmetric and asymmetric.

For mass flow, \( ff \) is a function of the slope angle \( \theta ' \) of the channel, the kinematic angle of friction between the flowing solid and the wall \( \phi ' \), and the effective angle of friction \( \phi \).

I.e. \( ff = f(\theta ', \phi ', \phi) \) for mass flow \ldots (3.11)

For core flow, the flow factor depends on the slope angle \( \theta ' \) of the channel and the effective angle of friction \( \phi \). Similarly the no-piping flow factor is a function of the static angle of internal friction \( \phi_t \) and the effective angle of friction \( \phi \).

I.e. for doming in core flow,

\[
ff = f(\theta ' , \phi) \quad \ldots \quad (3.12)
\]
for no piping

\[
ff = f(\phi_t , \phi) \quad \ldots \quad (3.13)
\]
For no-doming in core flow, Ref. 1 recommends the use of \( ff = 1.7 \) for design.

**Flow - No Flow Postulate** Jenike postulated that:

"Gravity flow of a solid in a channel will take place provided the yield strength which the solid develops
as a result of the action of the consolidating pressures is insufficient to support an obstruction to flow." For flow to commence, the stresses in an obstruction must be higher than the yield strength of the material, so that failure occurs. The principal stress must occur at the exposed surface of the obstruction, and it is denoted \( \bar{\sigma}_1 \). The principal force is then

\[
\bar{V}_1 = A \cdot \bar{\sigma}_1 \quad \ldots \ldots \ldots (3.14).
\]

Since the yield strength is given by the unconfined yield pressure \( f_c \), the flow postulate is given by

\[
\bar{\sigma}_1 \geq f_c \quad \ldots \ldots \ldots (3.15).
\]

But since the flow factor \( f = \sigma_1/\bar{\sigma}_1 \) from Eqn. (3.7), then eliminating \( \bar{\sigma}_1 \) gives

\[
\sigma_1/f_c \geq f \quad \ldots \ldots \ldots (3.16)
\]

or

\[
V_1/F \geq f \quad \ldots \ldots \ldots (3.17)
\]

\( f \) plots as a straight line on \( V_1, \bar{V}_1 \) co-ordinates, (Fig. 3.9). \( V_1/F \) is the flow function FF, and for the eqn. (3.17) to hold \( f \) must be less than \( V_1/F \). This means that flow occurs always for FF(a), never for FF(c) and will occur for FF(b) when FF(b) is less than \( f \) at the corresponding \( V_1 \) value,

\[
i.e. \quad F \geq \bar{V}_1 \quad \ldots \ldots \ldots (3.18).
\]

The minimum dimension of the channel opening which allows flow is then given by:

\[
B = H (\theta') \cdot \bar{V}_1/A \gamma \quad \ldots \ldots \ldots (3.19).
\]

Where \( B \) is the minimum dimension

\[
A = \text{area of a shear cell} \left( \frac{1}{13} \text{ ft}^2 \right)
\]

\( \gamma = \text{bulk density of material} \).
Fig. (3.9)

Intersection of solid Flow Function with hopper Flow Factor.
and \( H(\theta') \) is a function of the geometry of the opening and the hopper slope \( \theta' \) - Ref.(1) pp. 83 Fig.43. Similarly for no-piping in core flow, a flow factor is drawn, and a point of intersection may be obtained. For failure to occur, \( F < \bar{V}_1 \) is required. The formula for the minimum pipe diameter is then given by:

\[
D = G(\theta) \frac{\bar{V}_1}{A \gamma} \quad \ldots \quad (3.20)
\]

where \( G(\theta) \) is given by Jenike Ref.(1) pp. 67 Fig. 36. \( D \) is the largest dimension of the opening.

3.2 - Theory Related to Density Changes and Vibration

Hvorslev, Ref.(2), in 1937 gave an equation for the failure of a remoulded saturated clay,

\[
\tau_f = m_0 \sigma_f + \gamma \exp(-B \epsilon_f) \quad \ldots \quad (3.21)
\]

where

- \( \tau_f \) = peak shear stress at failure.
- \( \sigma_f \) = effective normal stress on the plane of failure at the moment of failure.
- \( \epsilon_f \) = voids ratio in the plane of failure at the moment of failure,

and \( m_0, \gamma \) and \( B \) are constants of the particular clay.

This equation states that the peak shear stress at failure of the clay is a function of the effective normal stress on, and of the voids ratio in, the plane of failure at the moment of failure, and this function is independent of the stress history of the sample. Eqn.(3.21) defines a 3-dimensional failure surface as shown in Fig.(3.10) in \( \tau, \sigma, \epsilon \) space. Roscoe and co-workers, Ref.(3), set the bounds of the surface as \( \tau \geq 0, \sigma \geq 0 \) and \( \epsilon \geq \epsilon_{\text{min}} \). This meant that the soil has no tensile strength, and the voidage could not be less than a certain minimum. A further boundary exists because under any normal load there is a
The Hvorslev failure surface.

Types of stress-strain curves.
limit to how loose the packing can be. Roscoe et al showed that this boundary rises from the normal consolidation line and is formed by the loading paths of normally consolidated samples. The surface is curved and intersects the Hvorslev failure surface at the critical voids ratio line. This line is the end point for any path on the failure surface - when it is reached further strain ensues with no change in shear force or voidage under a given normal load.

Williams and Birks, Ref.(5), showed how the density changes during a shear consolidation, when the sample is underconsolidated, normally consolidated and overconsolidated, see Fig.(3.11). In the preparation of a sample the normal consolidation level is the desired state. How this state is attained can be shown on the failure surface.

Referring to Fig.(3.10), a loosely packed powder might be represented by the point A, with a small normal stress due to the weight of the powder and Jenike cell above it. When weights are added the normal stress increases and the cell is compacted to point B along the normal consolidation line. As shown on Fig.(3.12), the twisting shear applied will be in a constant normal stress plane passing through the points BN. CD is the intersection of the plane with the Hvorslev failure surface and C is a point on the C.V.R. line. If the powder is initially at a point B, each twist produces a small change in voidage BJ. If the twisting is stopped at point G and the specimen then subjected to shear strain, then the voidage will fall until it reaches the point C, where failure takes place. Since point C is on the C.V.R. line, strain continues with no
**Fig. (3.12)**

Change of voidage during consolidation.

**Fig. (3.13)**

Force balance of Jenike Cell on vibrating plate.
further changes in shear stress or voidage. If the strain stops before point C is reached no failure occurs, giving a stress-strain curve of the form shown in Fig.(3.11) line A. If the specimen is originally densely compacted (point K, Fig.(3.10)) as might occur through vibration, each twist will slightly increase the voidage. If twisting is stopped at L and the specimen sheared to failure, the voidage increases and the shear stress rises to point M in the failure surface. Then voidage increases further and the shear stress falls along MC until at point C there is no further change in voidage or shear stress. The stress-strain curve obtained is of the form shown in Fig.(3.11) line C. The aim of the consolidation procedure is to end the twisting process at N, directly under C on the C.V.R. line. When shear is applied failure occurs with no change in voidage, giving a stress-strain curve of the form shown in Fig.(3.11) line B.

After the twisting process, Jenike's method, Ref.(1), subjects the specimen to shear stress failure until a steady shear is obtained. If the specimen is under-consolidated, the point C may not be obtained and a stress-strain curve of the form of line A, Fig(3.11), may result. In this case either more normal force and/or more twists must be applied in the twisting stage. If the specimen reaches point C under shear, then when the shear stress is removed the voidage will remain close to point N. In the case of an overconsolidated specimen, stopping the twisting at point L to the right of N, and then shearing to steady failure will bring the state of the powder to C. When the shear stress
is removed the voidage will end up close to point N. It must be emphasized that in the overconsolidated case, the specimen must be sheared until a steady shear value is attained as shown by Fig. (3.11) line C. If this is not done the correct voidage will not be obtained and a bogus shear value will occur at the lower value of normal stress.

The determination of a yield locus can be also illustrated on the failure surface. If the specimen is correctly consolidated to point N, then when a lower normal stress is applied ($\sigma''$), the specimen will move to point P, assuming no elasticity. Since P is not under the C.V.R. line, the specimen is now overconsolidated for its new load. Applying shear force will produce the overconsolidation form of stress - strain curve shown as line PRS, with R the peak shear stress and S the steady shear point on the C.V.R. line. A test at a higher normal load $\sigma''$ would give the line QTU. The line CTR drawn on the failure surface is the yield locus and the locus obtained from shear cell tests is the projection of CTR in the $(\tau,\sigma)$ plane.

One point about this failure surface is that with the Jenike cell, the partial restriction on the material within the cell (instead of being perfectly restricted) means the point C is not on the failure surface but just below it. This is illustrated by the drawing of the failure curve, Fig. (3.12), as not passing through the consolidation conditions for the yield locus curve.

In Section 5, research is reported describing the effect on wall friction of vibrating the plate in contact with the Jenike shear cell. In the initial stage the
Jenike machine was used to provide the shear force, and in the latter stages a tilting plate was used so shearing took place under gravity forces.

The force balance for the first case is shown in Fig.(3.13). Using the English Engineering system of units, the total vertical force exerted on the plane of material in contact with the plate is \( N(\text{lbf}) \). The Jenike machine is applying a shear force \( S(\text{lbf}) \) through the moving stem. Since the plate is being accelerated with a physical motion

\[
\dot{Y} = a\omega^2 \sin\omega t (\text{ft/sec}^2) \quad \ldots \quad (3.22)
\]

the shear cell feels an inertia force \( (NY/g) \text{lbf} \) as long as no slippage occurs. Prior to slippage then, the force balance is:

\[
S + Na\omega^2 (\sin\omega t)/g = MN \quad \ldots \quad (3.23)
\]

The term \( (Na\omega^2 (\sin\omega t)/g) \) causes the term \( (S + Na\omega^2 (\sin\omega t)/g) \) to oscillate as shown in Fig.(3.14), and if at any point the L.H.S. of eqn.(3.23) exceeds the maximum restraining force \( MN \) then slippage occurs. Thus, if the acceleration \( (a\omega^2 \sin\omega t) \) is high enough, only a small shear force \( S \) will be required to move the cell. For the case with constant amplitude, constant frequency acceleration, the shear force measured \( (S) \) should be a constant distance \( (Na\omega^2 (\sin\omega t)/g) \) below the static shear force measured for the case when slippage is just occurring. If slippage occurs for most of the cycle of the \( (a\omega^2 \sin\omega t) \) term then the cell is not experiencing the full acceleration from the plate and the inertia force felt by the cell will be less than \( (Na\omega^2 (\sin\omega t)/g) \text{lbf} \).
Fig. (3.14)
Oscillation of total shear force.

Fig. (3.15)
Force balance of Jenike Cell on tilted vibrating plate.
For the measurement of wall friction using the tilting plate device, the force balance for the cell is shown in Fig. (3.15). The total normal force felt by the plane of material in contact with the plate is \( N \cos \alpha \), where \( N \) is the total weight of cell, ore and weights expressed as \( N(\text{lbf}) \).

The plate acceleration is given by Eqn. (3.22), which causes the cell to experience an inertia force given by

\[
\text{Inertia force} = N\omega^2 \sin(\omega t)/g (\text{lbf}) \quad ... \quad (3.24)
\]

The inertia force is only of this magnitude if no slippage occurs. Prior to slippage, the sum of forces along the plate is

\[
N \sin \alpha + \frac{N\omega^2 \sin(\omega t)}{g} = M \cos \alpha \quad ... \quad (3.25)
\]

The sinusoidal term causes the driving forces for slippage to oscillate in an identical fashion to Fig. (3.14).

As \( \alpha \) increases, the driving forces for slippage also increase until the coefficient of friction \( M \) reaches its limit. If \( \alpha \) is increased more, the driving forces exceed the friction limit and slippage occurs; Fig. (3.16), i.e. whenever

\[
N \sin \alpha + \frac{N\omega^2 \sin(\omega t)}{g} > M \cos \alpha \quad ... \quad (3.26)
\]

When slippage does occur, the full acceleration of the plate will not be felt by the cell, and the inertia force on the cell will be less than \( (N\omega^2 \sin(\omega t)/g) \text{lbf} \).

This theory predicts several factors:

(a) The wall friction force can be decreased by an amount not exceeding \((N\omega^2 \sin(\omega t)/g)\), if the plate in contact with the cell is vibrated with a sinusoidal motion \((a \sin \omega t)\). If this term is
Fig. (3.16)

Occurrence of slippage from combined shear and inertia forces.
large then the shearing force required for slippage will be close to zero.

(b) The shear force - normal force graph obtained will show a low angle of internal friction $\phi'$; $\phi'$ being dependent on the magnitude of the inertial acceleration $(a\omega^2\sin\omega t)$. The higher this inertial acceleration, the lower the angle $\phi'$ will be.

(c) Slippage is predicted when the sum of the cell's inertia force and the external shear force applied exceed the frictional limit of the cell-plate system.
SECTION 4

ANALYSIS OF STEELMAKING RAW MATERIALS

4.1 Preamble

After discussions with production personnel at the Raw Materials Section of A.I. & S., it was confirmed that the worst materials handled in that section were Mt. Newman fine ore, Cockatoo fine ore and flue dust (the dust extracted by the blast furnace dust catcher from blast furnace gas). These three materials were the most prominent offenders in chute blockages and build up in hoppers. Samples of each material, obtained from the primary storage yards of Raw Materials, were sieved through a 20 mesh sieve to give a maximum particle size of 850 microns for testing. The chemical analyses of the 3 materials is given in Table (4.1).

For the 3 materials, samples with moisture contents of nominally 0%, 5%, 10%, 12% and 15% w.b. were made up, where

\[ \% \text{ w.b.} = \frac{\text{weight of water}}{\text{weight of water} + \text{weight of dry ore}} \] .... (4.1)

Each sample was then left sealed to the atmosphere for at least 24 hours prior to testing, to allow even distribution of the moisture. After testing, some of the material was placed in an air oven at 105 - 110°C for at least 4 hours and the moisture content calculated from the weight decrease. The final moisture content used was the average of the figures for a sample taken
CHEMICAL ANALYSES OF RAW MATERIALS

<table>
<thead>
<tr>
<th>MATERIAL</th>
<th>FLUE DUST</th>
<th></th>
<th>MT. NEWMAN</th>
<th></th>
<th>COCKATOO FINES</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>% BY WEIGHT</td>
<td>TYP.</td>
<td>RANGE</td>
<td>TYP.</td>
<td>RANGE</td>
<td>TYP.</td>
<td>RANGE</td>
</tr>
<tr>
<td>% Fe</td>
<td>35</td>
<td>30-40</td>
<td>62</td>
<td>61-63</td>
<td>63</td>
<td>61-65</td>
</tr>
<tr>
<td>% Mn</td>
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<td>.2-1.0</td>
<td>0.14</td>
<td>.08-.18</td>
<td>.15</td>
<td>.08-.2</td>
</tr>
<tr>
<td>% CaO</td>
<td>9</td>
<td>5-15</td>
<td>0.20</td>
<td>.05-.50</td>
<td>.16</td>
<td>.06-.25</td>
</tr>
<tr>
<td>% SiO₂</td>
<td>7</td>
<td>5-10</td>
<td>6</td>
<td>5-7</td>
<td>4.5</td>
<td>3.5-6</td>
</tr>
<tr>
<td>% Al₂O₃</td>
<td>3</td>
<td>2-4</td>
<td>2.5</td>
<td>2-3</td>
<td>2.5</td>
<td>2-3</td>
</tr>
<tr>
<td>% Free Carbon</td>
<td>22</td>
<td>10-30</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

**TABLE (4.1)**

*Note:* The additional percentage amount to come up to the 100% total for flue dust is comprised mainly of oxygen in the form FeO and Fe₂O₃, plus some CO₂ in the form CaCO₃. For the two iron ores it is also mainly in the form of oxygen, but purely as Fe₂O₃.
before testing and a sample taken after testing.

4.2 Flow Properties of the Raw Materials

This section gives the flow properties found for the 3 materials.

The Mt. Newman fines was obtained from the yards in hot dry weather, and the other two obtained after a very wet 2 week period. The percentage moisture contents of the three samples were:

Mt. Newman fines 0.9% w.b.
Cockatoo fines ore 8.3% w.b.
Flue dust 13.5% w.b.

This indicates the variation in moisture content between wet and dry periods and shows how design must cater for the worst possible (i.e. strongest) materials that the system is expected to handle. The yield and wall loci tests were done on the following samples:

Mt. Newman fine iron ore 0.9%, 5.6%, 8.1% w.b. (The saturation level was approximately 10.0%).
Cockatoo fine iron ore 0.3%, 4.0%, 7.8% and 8.9% w.b. (The saturation level was approximately 10%).
Flue dust 1.1%, 5.3%, 9.9%, 13.4% and 17.4% w.b. (The saturation level was approximately 26%).

To illustrate the processing of results, the detailed results for 8.1% w.b. Mt. Newman fine ore are given. From testing, it was found that a normal force range of 4, 8 and 12 lbs gave a satisfactory range of yield loci curves. The weight of the top part of the Jenike shear cell plus the top half of the ore was constant at
1.11lbs for all tests. Over the range of tests carried out with 4, 8 and 12lbs normal force, the shear values were all corrected to take account of variations in the consolidation shear level as set out below in Table (4.2). For example, for the 12lb case, the consolidation shear values $S$ varied, and when averaged out came to 14.01lbs. The values of shear force $\bar{S}$ were then corrected by the ratio $(\frac{S_{av}}{S})$

i.e. $\bar{S}$ corrected = $\bar{S} \times (\frac{S_{av}}{S}) \quad \ldots \quad (4.2)$

The results are plotted on Graph 1 as the instantaneous yield loci.

Wall yield loci tests were then performed for black steel and polished stainless steel. For 8.1% w.b. Mt. Newman fines, wall yield loci were also carried out for two common hopper lining materials, Linatex and Trelleborg. Linatex is a very flexible red rubber, generally used in fines chutes for its impact absorption and ability to shed wet fines. Trelleborg is a very hard black rubber generally used for direct impact of roughs. It was decided to test these two materials for their frictional properties due to their common use in plant applications. The consolidation procedure for all wall yield tests followed Jenike's suggestion of light consolidation. (Ref. 1 page 52)

The results are tabulated as Table (4.3), and the wall yield loci are plotted on Graph 2. From the yield loci graph (Graph 1) the flow function graph FF is obtained, Graph 3. The yield loci graph also gives the angle of effective friction ($\phi$), the angle between the
CORRECTION OF SHEAR FORCE VALUES

MT. NEWMAN FINES 8.1% W.B.

INSTANTANEOUS YIELD LOCUS

<table>
<thead>
<tr>
<th>V</th>
<th>S</th>
<th>V̅</th>
<th>S̅</th>
<th>Sav.</th>
<th>Scorr.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal Force during Consolidn.</td>
<td>Shear Force during Consolidn.</td>
<td>Normal Force in test</td>
<td>Shear Force in test</td>
<td>Average of values of S</td>
<td>S x (Sav/S)</td>
</tr>
<tr>
<td>12.1</td>
<td>13.4</td>
<td>10.1</td>
<td>12.2</td>
<td>14.0</td>
<td>12.75</td>
</tr>
<tr>
<td>12.1</td>
<td>14.2</td>
<td>6.1</td>
<td>9.7</td>
<td>&quot;</td>
<td>9.55</td>
</tr>
<tr>
<td>12.1</td>
<td>14.3</td>
<td>4.1</td>
<td>7.8</td>
<td>&quot;</td>
<td>7.65</td>
</tr>
<tr>
<td>12.1</td>
<td>14.0</td>
<td>2.1</td>
<td>5.5</td>
<td>&quot;</td>
<td>5.5</td>
</tr>
<tr>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8.1</td>
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<td>6.1</td>
<td>9.25</td>
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<td>2.1</td>
<td>5.2</td>
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<td>5.05</td>
</tr>
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<td>1.1</td>
<td>3.45</td>
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<td>V = 8.1 lbs</td>
<td></td>
<td></td>
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<td></td>
<td></td>
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<td>V = 4.1 lbs</td>
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</tbody>
</table>

TABLE (4.2)

See Graph 1 for plotting of these points.
GRAPH 2 WALL YIELD LOCUS. MT. NEWMAN FINES 8.1% W.B.
## WALL YIELD LOCI RESULTS

**MT. NEWMAN FINES 8.1% W.B.**

<table>
<thead>
<tr>
<th>POLISHED STAINLESS</th>
<th>BLACK STEEL</th>
<th>LINATEX</th>
<th>TRELLEBORG</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>V</strong></td>
<td><strong>S</strong></td>
<td><strong>V</strong></td>
<td><strong>S</strong></td>
</tr>
<tr>
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<td>5.1</td>
<td>12.1</td>
<td>7.6</td>
</tr>
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<td>4.4</td>
<td>10.1</td>
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<td>2.95</td>
<td>6.1</td>
<td>3.8</td>
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<tr>
<td>1.1</td>
<td>0.8</td>
<td>1.1</td>
<td>0.6</td>
</tr>
</tbody>
</table>

**TABLE (4.3)**
V-axis and the envelope of the major consolidating force circles. This angle is used for flow calculations and as it is not constant it is shown graphed at the top of Graph 3.

From the Wall Yield Locus (Graph 2), the kinematic angle of friction (\(\theta'\)) is obtained, being the angle between the V-axis and the point on the W.Y.L. for the particular normal force \(V_{1}\). The angle \(\theta'\) varies with pressure, so an iterative procedure is required to find it knowing the hopper angle \(\theta_{c}\), the angle of effective friction \(\phi\) and the flow function \(FF\) for that material. An example will be used to illustrate the method of iteration.

**EXAMPLE 1 - Stainless Steel**

From Graph 3, \(\phi\) varies from 54.5 \(\rightarrow\) 63.5°. Take \(\phi = 60°\). From Graph 2, take an assumed angle of \(\theta' = 23°\). For example, using a hopper angle \(\theta_{c} = 20°\) - conical hopper. From Fig 48 of Jenike's Bulletin 123 (Ref. 1), an \(FF\) value of 1.21 is obtained. This \(FF\) line is then drawn on to the Flow Function graph, Graph 3. The point of intersection of \(FF = 1.21\) with the Flow Function for 8.1% w.b Mt. Newman is given by \(V_{1} = 8.2\)lbs. A Mohr force circle is then drawn on the wall yield locus, Graph 2, passing through the point \((V_{1} = 8.2, S = 0)\) and tangent to the effective yield locus.

The intercept of the Mohr circle with the stainless wall locus gives the angle \(\theta'\) as 25°. On Fig. 48 of Bulletin 123 (Ref. 1) this angle is on the limit for mass flow. The calculated angle \(\theta' = 25°\) must be iterated until
the calculated angle $\varnothing'$ is repetitive.

From Fig. 48 of Ref.(1), with $\varnothing' = 20^\circ$ and $\varnothing' = 25^\circ$, $ff = 1.23$ is obtained. When $ff = 1.23$ is drawn on the Flow Function graph, the intercept with FF 8.1% w.b. Mt. Newman produces $V_1 = 8.6$. Still the angle $\varnothing'$ remains at $25^\circ$ for stainless steel. As a second check the angle $\varnothing'$ should be $63.5^\circ$ as $V_1 = 8.6$. However this change does not affect the angle $\varnothing' = 25^\circ$.

EXAMPLE 2 - Black Steel - Taking $\varnothing' = 60^\circ$ approximately, and knowing that $\varnothing'$ steel is constant at $32^\circ$, for a conical bin with $\varnothing' = 20^\circ$ the point is clearly outside the mass flow region on Fig. 48, Ref. 1. Hence if any flow does develop it will be core flow and the effective angle of friction $\varnothing'$ is not applicable to this case as there is no flow down the walls.

These yield loci graphs, wall yield loci graphs and flow function graphs were drawn for all the materials of different moisture content. The flow function graphs are shown as Graphs 5, 6 and 7. From a comparison of the instantaneous flow function graphs it can be seen that Cockatoo fines 7.8% w.b. and Mt. Newman fines 8.1% w.b. have the highest ff graphs and hence the greatest strength of all the samples tested. The two flow functions intersect at $V_1 = 20.21$bs, with the Mt. Newman stronger in the higher $V_1$ range and the Cockatoo stronger at the lower end of the range. (Graph 5)

Time yield loci were carried out for both of these materials, but the preliminary results showed
virtually no shear strength increase for the Cockatoo fines above the instantaneous level. Since the Mt. Newman showed a large increase in shear strength in time tests, the Cockatoo time yield locus was discontinued because the purpose of this section was to find the strongest material possible.

The time yield locus carried out for the 8.1% w.b. Mt. Newman fines, is shown on Graph 4. The static angle of internal friction \( \phi_t \) is obtained from this graph and is shown tabulated on Graph 4. This angle is the inclination angle at the point of tangency of the T.Y.L. with the Mohr circle which determines the unconfined yield pressure, \( F_t \). The angle \( \phi_t \) is required for core flow design. All the remaining graphs are in Appendix 1.

4.3 Check On Use Of Mt. Newman Fines In Some A.I.S. Bins

To determine whether various A.I. & S. bins are capable of feeding properly with 8.1% w.b. Mt. Newman fines after 72 hours consolidation, the following bins were examined.

A) Fine Ore Bins, No. 2 and 3 Sinter Machines, A.I. & S.

1) Mass Flow These steel bins originally took screened iron ore, each bin holding a different type of ore. The mixing for the sinter feed was done by the rotary table feeders at the bottom of the bins. With the recent installation of new plant, blending is now carried out earlier in the sequence and these bins carry the mixed sinter feed, a combination of iron ore, limestone, coke and other materials. Problems have been encountered
GRAPH 4  72 HOUR TIME YIELD LOCUS. MT. NEWMAN FINES 8.1% W.B.
Graph 5
Flow functions, Mt. Newman fines - varying moisture contents
Unconfined Yield Force F lbs

MAJOR CONSOLIDATING FORCE - V_i LBS

FLOW FUNCTIONS. COCKATOO FINE ORE - VARYING MOISTURE CONTENTS
with bridging of the material in the bins, and all the bins have "dongers" - 3 foot diameter by 4 inch thick slabs of steel suspended on chain, to beat the side of the bins.

The geometry of the conical section of the bin is shown in Fig. (4.1) and the angle $\theta'$ is $20^\circ$. From Graph 3, for Mt. Newman fines 8.1% w.b., the effective angle of friction varies between $54.5^\circ$ and $63.5^\circ$. A value of $60^\circ$ will be assumed initially and corrected later if the interaction requires it. The kinematic angle of friction $\phi'$ is required from the W.Y.L. Graph 2.

**Black Steel**

$\phi'$ is constant at $32^\circ$. $\theta'c = 20^\circ$, $\phi = 60^\circ$.

The point is outside the limit of $\phi$ contours given on Fig. 48 Bulletin 123; that is with these conditions mass flow will not occur. (Even with $\phi = 50^\circ$ mass flow will not occur.). It is interesting to note that if $\theta'c$ was reduced to $12^\circ$, mass flow would commence.

**If Stainless Steel was used:**

Assume $\phi' = 24^\circ$. With $\theta'c = 20^\circ$, and $\phi = 60^\circ$, the point is just inside the limit of $\phi$ contours, that is, mass flow may occur.

A value of $\phi = 1.21$ is obtained from these points. Plotting $\phi = 1.21$ on the Flow Function graph, Graph 3, the point of intersection of $\phi = 1.21$ with the Flow Function for 8.1% w.b. Mt. Newman 72 hrs. is given by $V_1 = 12.1$ lbs.

When a Mohr force circle passing through the point $V_1 = 12.1$ lbs. and tangent to the Effective Yield Locus
**Fig. (4.1)**
Fine Ore Bins. Hopper geometry.

**Fig. (4.2)**
No. 2 Ore Screening Station Fines Bins. Hopper geometry.
is drawn on Graph 2, the point of intersection of the circle with the Stainless steel wall yield locus gives a kinematic angle of friction $\phi' = 23^\circ$.

**Repeat of Calculation**

With $\phi' = 23^\circ$, and $\phi = 60^\circ$, a new value of $ff = 1.22$ is obtained. With this value of $ff = 1.22$, the value of $V_1$ changes from 12.1 to 12.4 lbs. This gives no significant change to the kinematic angle of friction $\phi'$. Secondly the assumed angle of $\phi = 60^\circ$ was not exact, and this must be checked. With $V_1 = 12.4$ lbs., the effective angle of friction is actually given by $63.5^\circ$.

Repeating with $\theta_c = 20^\circ$, $\phi = 63.5^\circ$ and $\phi' = 23^\circ$, by linear interpolation between Fig. 48 and 49 Ref. (1)

$$ff = 1.22 - \frac{3.5}{10} (1.22 - 1.15)$$

$$ff = 1.22 - .35 (.07) = 1.2200 - .0245$$

$$ff = 1.1955 \approx 1.20$$

When this value of $ff$ is drawn on the Flow Function graph, a new point of intersection of $V_1 = 11.8$ lbs. is obtained. The new Mohr circle on the wall yield locus through $V_1 = 11.8$ and tangent to $\phi = 63.5^\circ$ gives a new angle of $\phi' = 23.4^\circ$. A final repeat shows that changing from $\phi'$ of $23.0^\circ$ to $\phi' = 23.4^\circ$ makes no significant change to the $ff$ value, just allowing mass flow.

Summarising: A mild steel lined conical hopper of angle $\theta_c = 20^\circ$ containing ore with an effective angle of friction $\phi = 63.5^\circ$ and having a kinematic angle of friction of $32^\circ$ with the mild steel wall, will not allow mass flow.
The same hopper, if lined with stainless steel, has a kinematic angle of friction of $23.4^\circ$ with the walls and is just capable of allowing mass flow, using the same ore. The angle $23.4^\circ$ is $1\frac{1}{2}^\circ$ inside the mass flow limit.

A) Fine Ore Bins, No. 2 and 3 Sinter Machines  
2) Core Flow  From the preceding section A - 1 it was shown that mass flow will not occur for steel lined bins. A core flow analysis will now be done for these steel lined bins to determine the critical opening dimensions to prevent piping and doming in core flow.

Using the Time Yield Locus for Mt. Newman fines 8.1% w.b. (Graph 4), the worst value of the static angle of friction is $54^\circ$. From the no-piping graph, Fig. 35, Ref. (1), a no-piping flow factor is obtained using $\theta = 60^\circ$ and $\theta_t = 54^\circ$, of $ff = 3.3$. This $ff$ line does not intersect with the Time Yield Locus for Mt. Newman fines, indicating that the material is stable enough to form a pipe because it can support itself vertically.

If the figure of $ff = 1.7$ is used, (recommended for design purposes) the corresponding value of $\bar{V}_1$ from the intersection of $ff$ and the time yield flow function is 24.5 lbs. From Eqn. (3.20), computing the diameter $D$ required for no-piping,

$$D = \frac{13\bar{V}_1 \cdot G (\theta_t)}{\sqrt{161}} = \frac{(13) \cdot (24.5) \cdot (6.0)}{161} = 11.9 \text{ ft.}$$

This illustrates clearly that this material will easily support piping, up to a width of 11.9 ft. which is over half the maximum diameter of the bin. Now if the lowest angle
of effective friction \((\phi^f)\) of 54.5° had been used instead of 60°, and the lowest static angle of friction, 42.5° had been also used instead of 54°, the no-piping flow factor is 2.2. This still has no intersection with the T.Y.L. and thus shows piping is still stable. So no matter what size the opening is at the bottom, if core flow commences, a vertical pipe of any diameter can form and it will be stable.

To see if doming occurs also, the minimum minor dimension \(B\) of the outlet will be computed. Using the intercept of the T.Y.L. and \(ff = 1.7, \overline{V_1} = 24.5\) lbs., from eqn. (3.19),

\[
B = 15 \overline{V_1}/\gamma = \frac{(15)(24.5)}{161} = 2.28 \text{ ft.}
\]

Since the opening is to the side via a rotary table feeder and a plough, and the opening is of the order of 18 inches by 18 inches square then doming also appears to be possible. However the rotary table feeder would cause dislocation of the dome and collapse it as soon as it formed. Stable piping will be the principal problem.

B) **Fines Bins, No. 2 Ore Screening Station A.I. & S.**

There are 4 fines bins in No. 2 Ore Screening Station, and above each bin is mounted a double deck iron ore screen which screens ore - 6mm through the screens into the fines bins. The ore + 6mm feeds off the end of the screen onto a conveyor belt and hence to the Blast Furnace, while the fine ore passes through the fines bins to the Blending Yards and to the Sinter Machine for sintering. Initially, the bins were to be lined with bonded Linatex rubber 20mm thick
to prevent wear and to stop substantial build up. But since there was a possibility of bypassing rough ore through the fines bins by removing screen cloths, Trelleborg impact rubber was fitted to the bin walls instead. This has an inverted washboard shape to collect ore so that under impact the ore is wearing on itself rather than the rubber. The calculations will show results for both clean Trelleborg liners and then liners with a complete ore coverage. Under wet weather conditions, ore has built up on the back wall of the bins out to approximately 4 feet thick, and at one stage this built up the full height of the back wall until it contacted the vibrating screen and caused breakage of a rear beam of the screen. The geometry of the bins is in Fig. (4.2), and by A.I.S. standards is a good, steep bin.

The angle of slope $\theta_{p1} = \arctan \frac{455}{1000} = 24.5^\circ$

The angle of slope $\theta_{p2} = \arctan \frac{376}{1000} = 20.6^\circ$

The angle of slope $\theta_{p3} = \arctan \frac{167}{1000} = 9.5^\circ$

1) **Mass Flow Check (i) Assuming Clean Trelleborg Liners**

The top part of the bin is effectively a plane flow section, and using symmetric plane flow design criteria the bin will be checked for mass flow, using Mt. Newman fines 8.1% w.b. after 72 hours time delay.

From Graph 3, choose the effective angle of friction $\phi^* = 63.5^\circ$.

From Graph 2, the actual kinematic angle of friction $\phi'$ for steel is given by $\phi' = 32^\circ$ and the angle for clean Trelleborg is very close.

So assume $\phi' = 32^\circ$.
The $\theta'$ which just touches the dashed line limit recommended by Jenike, Ref. (1), Figs 53 and 54 for good mass flow is by linear interpolation,

$$\theta' = 18^\circ \times 0.35 \times (21 - 18) = 19^\circ$$

Using $\theta' = 19$ gives by linear interpolation,

$$ff = 1.1 - 0.35 \times (1.1 - 1.05) = 1.0825 \approx 1.08$$

Drawing this on the flow function graph, (graph 3) gives an intersection of $V_1 = 9.21$ lbs with the Time Flow Function. When the corresponding Mohr circle is drawn on the W.Y.L. Graph, (Graph 2) to intersect with the Trelleborg W.Y.L., the recalculated angle $\vartheta' = 32^\circ$, and the recalculated angle $\varphi' = 63.5^\circ$. Thus the actual bin angles are very close to that required for mass flow, and since plane flow bins have a very wide margin before core flow develops, this section of bin would give mass flow. The small difference in slope angle of the two walls is not considered significant for plane flow. The lower part of the bin is an inverted pyramid with one wall vertical. This is difficult to analyse directly, but Jenike (Ref. 1, P76, 77) suggests that the flow factor of a pyramidal hopper is above 4 for mass flow, and that the valley angles should be within the mass flow region for conical hoppers. In addition he states that a conical hopper with one side vertical will lead to mass flow but the included angle should be only $1.5\theta'$, where $\theta'$ is the slope angle of a symmetric core for mass flow. If the included angle is too large core flow will develop along the vertical wall.
Using these points it will be possible to estimate the required angles for mass flow. If the bin lower section is thought of as symmetric in both directions, i.e. with its centre-line vertical, the angle $\theta^p$ will become

$$\frac{24.5}{2} = 12.2^\circ \quad \text{See Fig. (4.3)}$$

The distance $x$ is given approximately by:

$$x^2 = \sqrt{1.200^2 + .850^2}$$

$$x = \sqrt{1.44 + .72}$$

$$x = \sqrt{2.16} = 1.47 \text{m}$$

Then the valley angle

$$\theta = \arctan \frac{1470}{5500}$$

$$\theta = \arctan .267$$

$$\theta = 14.9^\circ$$

For the Mt. Newman fines 8.1% w.b., the properties needed are:

Estimate $\phi = 63.5^\circ$.

Estimate $\phi' = 32^\circ$, which for clean Trelleborg is fairly constant.

From Ref. (1) Figs. (48) and (49), the max. $\theta^c$ for a symmetric conical channel giving mass flow $= 9^\circ$ (taking $3^\circ$ inside mass flow limit). The corresponding flow factor $ff \approx 1.13$, gives an intersection of $V_1 = 10.1 \text{ lbs.}$ and gives no significant change in $\phi$ or $\phi'$. (See Graphs 1, 2 and 3).

Thus for the Mt. Newman fines 8.1% w.b. after 72 hours time delay, a hopper angle $\theta^c$ of $9^\circ$ or less would be needed to give mass flow in a Trelleborg lined conical bin mounted with the centre-line vertical. But
Approximate calculation of valley angle $\theta$. 
with a vertical pyramidal hopper, it is recommended that the valley angles not exceed this 9° angle if mass flow is to be ensured. So with the pyramidal hopper mounted with its centreline vertical, mass flow could not be expected because the valley angle of 14.9° is outside the mass flow angle for a conical bin ($\theta_c = 9°$). When the pyramidal bin is set as it is in the plant with one wall vertical, the situation is worsened because one wall is made even more shallow. It was said in Ref. (1) that to ensure mass flow with a tilted conical bin the included angle should not exceed $1.5\theta_c$, where $\theta_c$ is the angle recommended for mass flow in a symmetric conical bin. If this is applied to the valley angle of the pyramidal bin, then the included angle of the valleys should not exceed $1.5 \times (9°) = 13.5°$ to ensure mass flow. In fact the included angle of the valleys $= 2\theta$
\[ = 2(14.9) = 29.8°, \]
so the tilting of the back wall has ensured there will be no mass flow. Core flow will develop at the vertical wall and the material on the back wall will build up.

1) Mass Flow (ii) With Ore Buildup On Trelleborg Liners

The wall lining of Trelleborg washboard rubber during bad conditions builds up a surface of almost complete ore coverage, which will give a worse kinematic angle of friction $\phi'$ than the Trelleborg rubber itself. An estimate of the angle $\phi'$ when ore is the actual surface can be obtained from the angle of inclination of the yield loci. The 4lb curve on Graph 1 should be used as that is the standard consolidation for Wall Yield Locus
tests, and the material in question is Mt. Newman 8.1% w.b. The angle $\theta'$ varies between 40° and 50° for the 4 lb. curve so take $\theta' = 45°$.

From Fig. (53), Ref. (1), the effect on the upper part of the bin is to give a flow function $ff \approx 1.60$ for the hopper angle, $\theta'_p = 24.5°$, using $\theta = 63.5°$. From Graph 3, this is unlikely to give mass flow except at very high pressures; core flow would be expected in general.

For the lower part of the bin, a kinematic angle of friction of $\theta' = 45°$, for $\theta = 63.5°$, would not give mass flow for the pyramidal section even if the walls were all almost vertical - so core flow is assured. Thus if the situation develops where the bin wall surfaces are filled with ore, the flow tends to worsen. Mass flow in the upper plane flow section of the bin would be transformed to core flow, while for the lower section there would be even less chance of having mass flow develop.

2) **Core Flow Check**

As it has been shown in part 1 that mass flow cannot be expected in the bin lower sections, because the valley angles are too large and the included valley angle is much too large for mass flow, the design will now be checked for core flow.

From the previous work for the conical bins, it was shown that the no-piping flow factor $= 3.3$. Since this is always below the time flow factor, this shows that
piping is always stable. Also if the design ff of 1.7 was used, this predicted a stable pipe up to 11.9 ft. diameter or 3.63 metres diameter, wider than the whole bin.

To check for doming, the minimum minor dimension B was found previously to be 2.28 feet or 0.7 metres if doming was not to occur. The opening dimensions of the bins are 0.70 x 1.0 metres, so the theory predicts doming may occur across the bin opening.

In conclusion, although A.I.S. Design considers this bin was a good design, its flowability is very poor and doming and piping are predicted. It illustrates that conical bins are to be much preferred over inverted pyramid bins, and also A.I.S. Design needs some re-education.

4.4 Design Of A Bin For Mt. Newman Fines

A conical bin and a plane flow bin will be designed for mass flow and then core flow using 72-hr. Mt. Newman 8.1% w.b., to illustrate the method of design.

A) Conical Bin 1) Mass Flow a) Steel Walls

From Graphs 1, 2 and 3

Estimate $\phi = 63.5^\circ$

Actual $\phi' = 32^\circ$

Try $\theta'c = 20^\circ$

The point on Fig. 48 and 49 Ref. (1) is outside the range of mass flow.

Using $\theta'c = 9^\circ$, gives by linear interpolation
ff = 1.14 - .35(.04) = 1.14 - .014 = 1.126 ≈ 1.13
From plotting ff = 1.13, the intercept with T.Y.L. is
V₁ = 10.1 lbs. This low value of V₁ confirms that
φ = 63.5°. Thus for Mt. Newman fines 8.1% w.b. left
over 72 hours in the bin, a steel lined bin with angle
θ'c = 9° would just give mass flow (taking 3° inside the
mass flow limit). The minimum dimension B of the outlet
necessary to avoid doming, is given by eqn (3.19)
B = \frac{13V₁ H(θ')}{γ}
V₁ = Point of intersection = 9.0 lbs
γ = Bulk density
H(θ') given by Ref. (1) in Fig. 43
∴ B = (13)(9.0)(2.18)/161 = 1.59 ft.
Thus a conical steel bin with θ'c = 9° and an opening
larger than 1.59 feet will give mass flow for this
material. See Fig. (4.4)
1) b) Stainless Steel Walls
From Graphs 1, 2 and 3
Use φ = 63.5°
Estimate θ' = 23°
Try θ'c = 19° as this is 3° inside the mass flow limit.
By linear interpolation, ff
ff = 1.21 - .35 (1.21 - 1.15) = 1.21 - (.35)(.06)
= 1.189 ≈ 1.19
The intersection of ff = 1.19 with the Time FF gives
V₁ = 11.6. From the Mohr circle intersection the
corrected θ' = 23.4°, which does not require further
iteration. Thus for a stainless lined bin with this
a) Conical hopper, steel liners.

b) Conical hopper, stainless steel liners.

c) Wedge hopper, steel liners.

d) Wedge hopper, stainless steel liners.

*Fig. (4.4)*

Comparison of Mass Flow Hoppers for 72hr. Mt. Newman 8.1% w.b.
material, a hopper angle of 18.5° (giving 3° inside the mass flow limit) will ensure mass flow, compared to 9° for black steel. The minimum opening is given by

\[
B = \frac{13VH(\theta')}{\phi} = \frac{(13)(9.5)(2.29)}{161} = 1.76 \text{ ft.}
\]

See Fig. (4.4) for mass flow bin design.

A) Conical Bin 2) Core Flow  It was shown in Section 4.3 Part A.2 that with the Mt. Newman fines 8.1% wet after 72 hours consolidation, its effective angle of friction (\(\phi^c\)) and angle of static friction \(\phi_t\) are such that a pipe 11.9 ft. diameter is stable. Thus any bin which is not a mass flow bin for this material, will produce stable piping up to 11.9 feet in diameter. The only criterion to ensure flow does occur through the pipe is that the opening is too wide to allow doming. The figure calculated for the minimum dimension of the opening is 2.28 feet, where this is the diameter of a circle, the side of a square or the small side of a rectangle.

Thus if a bin was to be designed for core flow for this material, it would need an opening larger than 2.28 feet to prevent doming. Apart from that requirement, the slope of the walls and the wall material have no effect on the flowability as long as mass flow is not possible. However it would make little sense to design a core flow bin for this material because piping is so stable that once a pipe formed in the bin, it would need external vibration or some physical force apart from gravity to dislodge the material around the walls. If this pipe remained for some time undisturbed the pipe
would consolidate and leave the bin with much reduced live capacity.

B) **Plane Flow - Wedge Hopper Bin** The design of a wedge hopper as shown in Fig.(4.5) will now proceed. Plane flow hoppers such as a wedge hoppers are much more capable of producing mass flow than conical or pyramidal hoppers, because the material is only converging in one direction instead of two directions. The theory also recommends a smooth transition from the vertical to the inclined wall with a radius \( R > D/3 \). If this is done, the final calculated slope angle \( \Theta'p \) can be increased by 5°, giving a less steep bin wall.

Another point with plane flow bins is that there is an optimum value of slope angle \( \Theta'p \) for any given material. If a less steep or a more steep angle than that calculated is used, the flowability will be decreased.

B) 1) **Mass Flow** a) **Steel Walls** Using Mt. Newman fines 8.1% w.b. after 72 hours delay. (Graphs 1, 2 and 3)

Estimate \( \phi' = 63.5° \)

Actual \( \Theta' = 32° \)

Try \( \Theta'p = 18° \)

From Fig. 53 and 54, Ref. (1), the point is just on the dashed line for

\[ \phi' = 60°. \]

By linear interpolation,

\[ \phi = 1.1 - (0.35)(0.05) = 1.100 - 0.0175 = 1.0825 \approx 1.08 \]

Plotting this flow factor on graph 3, the point of intersection with the flow function is given by
Fig. (4.5)

Geometry of a wedge hopper bin with radiused transition.
\[ V_1 = 9.2 \text{ lbs}, \bar{V}_1 = 8.4 \text{ lbs} \]

The angle of \( \phi' \) for steel does not change, so there is no need for iteration, and \( \phi' = 63.5^\circ \) is accurate. However this hopper will easily give mass flow and if it has the radiused transition, a \( \Theta'p \) of 23° could be used with confidence.

The minimum opening size is given by Eqn. (3.19)
\[ B_{\text{min}} = 13V_1 \frac{H(\Theta')}{\gamma} = (13)(8.4)(1.12)/161 \]
\[ = 0.76 \text{ ft}, \]
as long as the length of outlet \( L \) is greater than 3B where \( B = \text{width of outlet} \). This geometry of hopper would give an extremely reliable mass flow bin as flow will commence at low pressures. However larger angles of \( \Theta'p \) could be used if Jenike's recommendation Page 72, Ref. (1) to stay to the left of the dashed line is not followed. The effect would be to develop some non flowing regions at the top and the sides of the bin.

\( \Theta'p \) angles up to 35° (giving \( \frac{\rho}{\gamma} \approx 1.4 \)) could be used to give mostly mass flow in the bin although there would be some non flowing regions present. The minimum opening \( B \) would increase to
\[ B'_{\text{min}} = (13)(13.0)(1.18)/161 = 1.24 \text{ feet}, \]
and the minimum pressure in the material to commence flow virtually doubles. Still this shows how flexible the design is with plane flow bins rather than conical bins - some amount of bad design can be tolerated and still get mass flow. See Fig.(4.4)
1) Mass Flow  b) Stainless Steel Walls  The exercise will be repeated with stainless steel to show the improvement due to the lower wall friction.

Estimate $\phi = 63.5^\circ$

Estimate $\phi' = 25^\circ$

Try $\phi' = 28^\circ$

(on dashed line). By linear interpolation,

$$ff = 1.1 - .35(.05) = 1.0825 \approx 1.08$$

This gives $V_1 = 9.2$lbs, and the reiterated $\phi' = 24.5^\circ$.

This new angle of $\phi'$ makes no significant difference to the intersection point $V_1 = 9.2$lbs, and secondly the angle $\phi = 63.5^\circ$ is accurate. Again by using a properly radiused transition, the angle $\phi' = 28^\circ$ can be safely increased 5° and still obtain steady mass flow, i.e. up to 33°. The minimum opening B is given by:

$$B_{\text{min}} = 13V_1(H(\phi'))/\gamma = (13)(8.4)(1.18)/161 = 0.80 	ext{ feet wide}$$

Again assuming $L > 3B$. See Fig. (4.4)

Thus it can be seen that for the Jenike recommended design, P72 Ref. (1) an angle of $\phi' = 23^\circ$ is recommended for the steel walls and $\phi' = 33^\circ$ for the stainless steel, for the material in question.

If one was prepared to tolerate some areas of non flow, at the ends of the bin, the angle $\phi'$ could be increased up to $35^\circ$ for the steel and up to $45^\circ$ for the stainless steel, with increased opening sizes $B_{\text{min}}$ also.

B) 2) Core Flow  The core flow design is identical to that done in both Section 4.3 Part A.2 and Section 4.4
Part A.2, so will not be repeated.

4.5 - DISCUSSION

The work on the design of a bin to suit the material Mt. Newman fines 8.1% w.b. after 72 hours consolidation highlights several points.

(1) Any bin which produces core flow with this material is going to need continual cleaning out to maintain its full live capacity. The piping phenomenon has been shown to be extremely stable and needing an opening larger than 2.28 ft. across to prevent doming.

(2) To ensure mass flow in steel lined conical hoppers, the hopper angle required is 9° or less. This is not feasible for large capacity.

(3) The use of stainless steel liner plates improves the hopper angle θ'c by the order of 9° - 10° because of their lower frictional properties, compared to black steel. In the conical bins they improve the angle from 9° to 18.5°, which would make the construction of such a bin feasible.

(4) The results show that the plane flow bins can have shallower angles than the conical bins and still have mass flow. Again the stainless steel improved the hopper θ'p from 23° for black steel up to 33° for stainless steel. These plane flow bins could have angles up to 35° for black steel and 45° hopper angles for stainless and still retain mainly mass flow, if desired.

(5) Thus to secure mass flow the plane flow bins are much superior to the conical bins, and these are superior
again to the inverted pyramidal bins.

The main problem with plane flow bins is ensuring that the feeder arrangement used maintains flow over the complete length of the slotted outlet. If it does not there will be dead regions which will cause blockages.
SECTION 5

ANALYSIS OF VIBRATION EFFECTS

5.1 Preamble

This section on vibration follows several avenues. The first need was to produce a reliable vibrator. Once the vibrator was operative it was used to see how effective vibration was in over-consolidation of shear cells. The level of over-consolidation was graphed for varying frequency, time of vibration and normal force. Following this the effect of vibration in producing samples of increased density was observed, because over-consolidation was believed to be related to increased density. The next part of the work was to see if over-consolidated samples would return to their instantaneous yield locus curve if the samples were sheared to steady values. These tests were performed to evaluate a theory described by Williams and Birks Ref. (5), which said that for a given normal load there was a critical density which a sample would tend towards. Any over-consolidation would show a high peak of shear force which would then revert to the steady normal shear values.

Finally, research into the reduction of wall friction due to vibration was carried out. The use of vibrating baffles in the bottom of bins and vibrating panels in the sides of bins was thought to be mainly due to a reduction in wall friction which encourages mass flow. To test this effect wall yield loci were performed on plates which were actually being vibrated.
5.2 Development of Shaking Device

The first requirement for this section of the work was to construct a reliable vibrator. Development commenced with a mechanical vibrator driven by an electric universal motor and subsequently proceeded on the actual vibrator used, an electro-mechanical device operated by a coil.

(a) Mechanical Vibrator

This vibrator comprised two out-of-balance masses driven counter relationally by a small variable speed motor, Fig.(5.1.) The motor was geared down by a factor of ten through vee pulleys and was connected by a flexible rubber tube to the vibrating out-of-balance device. The out-of-balance device consisted of two shafts, one driven by the motor and the shafts held in mesh through two identical spur gears. Set onto each shaft was an off centre piece of steel with a 2½ inch long bolt screwed into the steel so that the assembly of steel and bolt was statically balanced. A known weight of steel (.044 lbs) with a tapped hole through the centre, was set on both bolts. The weights were positioned with respect to each other so that all side forces were eliminated and only a sinusoidal back-and-forwards force was produced. The motor and vee pulleys were mounted on a stand and the out-of-balance device was hung from this stand with chains. The out-of-balance device was then connected with a clamping ring to the top ring of the shear cell.

The mechanical vibrator was to be used to shake the top half of the Jenike shear cell to test the effect
91

TO POTentiOMETER & POWER SUPPLY.

10:1 V-PULLEY REDUCTION.

VARIABLE SPEED SEWING MACHINE MOTOR

FLEXIBLE TUBE

SUPPORT CHAIN

OUT-OF-BALANCE DEVICE

THRUst COLLARS

CLAMPING RING FOR JENIKE SHEAR CELL

STATICALLY BALANCED COLLAR & BOLT ASSEMBLY

OUT-OF-BALANCE MASSES

SPUR GEARS

Fig. (5.1a)
General arrangement of mechanical shaker and drive

SECTION 'AA'
GENERAL ARRANGEMENT

SECTION 'BB'
Fig. (5.1b)
Mechanical shaker general arrangement.

Fig. (5.1c)
Arrangement of out-of-balance device with counter rotating masses.
of vibration on material strength. It had variable frequency due to the variable speed motor, and controlled force due to the known out-of-balance weights; adjustment in or out varied the force.

Unfortunately the mechanical vibrator developed secondary vibrations because the torque from the flexible driving hose as well as its resistance to lateral movement set up torsional oscillations in the system. With minimal damping these became a steady vibration while the motor kept driving. Also the amplitude of the vibrator was too large and with the large size of the unit the inertial acceleration delivered to the top ring of the shear cell may have been enough to shear the material in the cell. Further development was planned but at this stage it was decided to try to build a simple electrical vibrator, because the problems involved with the mechanical device were fairly severe.

(b) Electro-Mechanical Vibrator

A commonly used large scale vibrator consists of a large coil energizing a ferrous core, set a small distance away from a fixed ferrous core. The energized core is restrained by very large springs and AC current is fed to this core to cause attraction. To generate vibration a certain amount of this current, controlled by a potentiometer is fed through a diode. The diode removes one half of the wave form, allowing the springs to retract the movable core while the next half waveform attracts the movable core, closing the gap. This cyclic action creates the vibration. To increase the strength of the vibration the potentiometer
is adjusted to feed more current through the diode.

The Electrical Maintenance section of Raw Materials at A.I.S. kindly donated a Cutler Hammer coil-operated contactor, a device which closes a hinged ferrous loop when the coil is activated. See Fig. (5.2) for General Arrangement. This was fitted with suitable strength springs and a diode, and connected to a Phillips Power Amplifier and a Variable frequency supply unit. The vibrator operated from 3Hz (the lower limit of the variable frequency supply) up to 60-70 Hz, where its operation became unstable. The maximum amplitude could be controlled by adjusting the clamping bolts which varied the contactor gap. The amplitude varied with frequency also being the maximum at low frequency and smaller at high frequency.

The unit was strengthened and fitted with mounting brackets for ease of assembly. To connect the vibrator to the shear cell, a perspex arm was constructed which bolted to the moving part of the vibrator. At the other end was a perspex flange with two holes drilled in it. These attached via two bolts to a flange glued to the top half of the shear cell.

To compare the effect of vibrating the whole cell rather than the top half only, a sieve shaker with variable frequency and amplitude was used. This shaker shook the cells vertically and compacted the complete shear cell whereas the electro-mechanical device was used to compact the top half of the cell only.

(c) **Gravity Device**

Most of the initial vibration work was done by
SPRING DEVICE TO LOCATE HINGED ARM POSITIVELY.

JENIKE SHEAR CELL WITH PERSPEX CONNECTING FLANGE GLUED TO TOP RING.

Arrangement of Electro-mechanical Vibrator.

Fig. (5.2a)

SECTION 'AA' TO VARIABLE FREQUENCY SUPPLY VIA DIODE & POWER AMPLIFIER.

3-3° ANGLE MOUNTING BRACKET.

HINGED FERROUS LOOP AIR GAP EXCITING COIL

Vibrator connected to Jenike Shear Cell.

Fig. (5.2b)
vibrating a sample and then shearing it in the Jenike machine to check how much vibration affected the sample's strength.

In a later part of the work (Section 5.6) it was desired to determine a Wall Yield locus using a plate which was actually being vibrated horizontally. This was attempted on the Jenike machine, but the vibration made accurate pen recording difficult at low shear values.

To overcome this a tilting plate was devised where the shear cell was placed on the vibrating plate. The complete plate, cell and vibrating plate could be tilted and the lowest angle at which the cell commenced to move down the plate was measured with a spirit level. By geometry the normal forces and shearing forces could be then be calculated. The Gravity Device is shown in Fig. (5.3).

5.3 Effect of vibration on shear strength using electrical shaker

It was decided to test the effect of vibration in consolidation of the Jenike Shear cell. The electrical vibrator was used to vibrate the top ring of the shear cell only. The samples were consolidated normally, first by twisting with a known normal force, then sheared to a steady value generally with the same normal force. Then the samples were vibrated for a set period of time and sheared again. This procedure allowed the change in shear strength after vibration to be determined. Furthermore how this change in shear strength is affected by increased vibration time, changes in frequency and changes in normal force on the shear cell was also examined. Note that all shear tests were carried out using a set normal load of
Fig. (5.3a)
Arrangement of gravity device with vibrator.

Fig. (5.3b)
Gravity device and vibrator set up for Wall Yield tests.
V = 8.1 lbs.

It was thought that with the electrical vibrator vibrating the top half of the cell only, a change in density would occur in the top half of the cell. This could lead to a plane of weakness at the join of the top and the bottom due to differing densities.

If this did occur then a lower shear value could be expected. For the following tests, the material used was 0.9% w.b. Mt. Newman Fine Ore.

The tests were done using the electrical shaker to shake the top ring of the Jenike shear cell after the cell had been consolidated at V = 8.1 lbs. Graph 8 shows the increase in shear strength above the unvibrated level for various conditions plotted against frequency:

(a) Shear strength after 15 seconds vibration with a normal force of 1.1 lbs. on the shear plane.

(b) Shear strength after 15 seconds vibration with a normal force of 8.1 lbs. on the shear plane.

(c) Shear strength after 60 seconds vibration with a normal force of 1.1 lbs. on the shear plane.

(d) Shear strength after 3 minutes vibration with a normal force of 1.1 lbs. on the shear plane.

As mentioned previously in the development of the electrical shaker, the amplitude was dependent on the frequency.

For the range of values of frequency, the following amplitudes were attained:
Graph 8  
Effect of Vibration on Shear Strength

Shear Force (lbs)

3 mins. vibration, V=1-lbs.
1 min vibration, V=1-lbs.
15 secs. vibration, V=1-lbs.

Average shear force during consolidation, S=6.5, V=8-l.

Note: All shear tests done with V=8-lbs, on Mt. Newman 0.5% WB.
<table>
<thead>
<tr>
<th>FREQUENCY (cycle/sec.)</th>
<th>AMPLITUDE (inches)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.010</td>
</tr>
<tr>
<td>15</td>
<td>0.009</td>
</tr>
<tr>
<td>30</td>
<td>0.009</td>
</tr>
<tr>
<td>50</td>
<td>0.007</td>
</tr>
<tr>
<td>60</td>
<td>0.005</td>
</tr>
</tbody>
</table>

As can be seen from Graph 8, the increase in shear strength due to vibration is most affected by increase in time of vibration. Presumably there is some limiting time where the material approaches a critical density and any further vibration will have little further effect. The increase in normal force had only a marginal effect while the change of frequency did make some difference although no conclusions should be drawn from this because amplitude was also being changed. To test if shear strength depends on frequency the amplitude would have to be held constant while frequency was varied. This was not feasible with the electrical shaker. To see if the similar effects to those discussed above happen for yield loci, extra tests were undertaken to establish the effect of extra time of vibration, extra normal force on the sample during vibration and a higher frequency of vibration compared to a standard vibration yield locus. The standard was taken to be a test done using normal twisting, vibrating at 10 Hz with a normal force of 3.2 lbs. for 30 seconds then shearing.

The extra time test was done for 2 minutes instead of 30 seconds with no other changes. The extra normal force test was done with 8.1 lbs. instead of 3.2 lbs. with no other
changes. The higher frequency test was done at 30 Hz instead of 10 Hz with no other changes. In this last case although frequency was changed the amplitude remained constant at 0.009 inches. Graph 9 shows the yield loci with the instantaneous yield locus and the 48 hour time yield locus also plotted. Again the increased time of vibration gave the greatest increase above the standard vibration yield locus as was illustrated for the constant normal force tests. Both extra normal force during vibration and higher frequency of vibration gave an increase above the standard but not as great as the effect of increased time. The increase in strength due to extra normal force is due to the higher forces on the particles forcing them to fill gaps, forcing air out, and allowing surface tension to increase between particles. The increase in strength due to higher frequency at a set amplitude could be explained by the fact that higher inertial accelerations and decelerations are happening at the higher frequency and these larger forces cause more gap filling.

The reason why increased time of vibration gave a higher increase in shear strength than increasing normal force or frequency is probably due to the larger number of impacts. To increase the shear strength of a Jenike cell the density in the region of shear must be increased. It is known that vibration is much more effective in increasing density instead of adding extra normal force or twisting of the shear cell lid more times. The point to consider is that the number of impacts are much higher with the extra time of vibration. With the increased normal force
GRAPH 2  EFFECT OF VIBRATION ON YIELD LOCUS
there are less impacts and the extra normal force is not acting in the same direction as the cyclic force is. Thus the extra normal force has little effect.

Similarly the reason why the increased time of vibration case gave a larger shear strength increase than the higher frequency case could be that at 30 Hz for 30 secs the total number of reversals is less than 10 Hz for 120 secs. The number of reversals would be related to the number of impacts between particles and hence the compacting of the shear region should be higher for the 10 Hz for 2 minutes case. This is in fact what did occur; the situation with the greatest number of cycle reversals gave a larger strength increase.

5.4 Effect of Vibration on Density and Density on Shear Strength

It is a well known fact that there is some form of correlation between shear strength and density, because the more consolidated a sample becomes with a higher density the higher strength it has. To examine this correlation more closely tests were done with both the sieve shaker and the electrical vibrator, vibrating over-full unconsolidated samples for a certain time, then scraping them level to give a set volume. By weighing the cell and material and knowing the cell's weight and volume, the density of the material could be found. The density was then plotted against the shear strength to give a curve for both vibrators. Graph 10 shows the results obtained using 0.9% w.b. Mt. Newman fines. Generally increasing time of vibration gave a corresponding increase in density
Graph 10 - Effect of Density on Shear Strength

Shear Force (lbs)

Weight of Ore in full shear Cell, gms.

Cells vibrated with sieve shaker.

Cells vibrated with electrical shaker.

Cells not vibrated.

All tests with Mt. Newman fines 0.9% WB.
as would be expected.

Similarly higher density gave to reasonable agreement a directly proportional increase in shear strength, on both the sieve shaker vibrating the whole cell and the electrical shaker vibrating the top half only. The vibration of the whole cell vertically gave more strength than the vibration of the top half horizontally, as shown clearly in the graph. The probable reason for this effect is that vibrating the whole cell vertically gives uniform strength and density throughout the material in the cell. Vibrating the top half of the cell horizontally will certainly increase the density in the top half of the cell but will have a lesser effect in the region of the shear plane and particularly the bottom half of the cell. This change in density from fairly dense and uniform in the top half through to medium density in the middle graduating to the original low density in the bottom will produce an area of varying density in the middle and hence a plane of weakness. Thus the shear force will be lower than for the complete cell vibrated vertically.

The main illustration from this section is the directly proportional relation between density of the material and its shear strength, for this particular material similar to Ref. (22).

5.5 The effect of vibration in overconsolidation of samples prior to testing for yield loci

Williams and Birks, Ref. (5) predict that there is a certain critical density which a given material will tend towards under a certain compacting load. If the
material is underconsolidated, the first part of the shearing process is taken up by consolidating the material to the critical density in the shear region, after which shear commences. Similarly if the material is over consolidated, the theory predicts that shearing will reduce the density to the critical value in the region of shearing, after which the material shears as if it were consolidated correctly. To test this theory a normal yield locus on 0.9% w.b. Mt. Newman fines was carried out with the normal force being 8.1 lbs. Then two tests were done similarly except that the samples were overconsolidated by vibration, one with twisting and one without. The samples were then sheared to a steady value with the original 8.1 lbs, then sheared at lower values to give a yield locus. If the theory was correct the values of sheady shear under 8.1 lbs. normal force would be identical, as well as the yield loci. The only difference would be that the curves for the vibrated sample would show a high peak before descending to the steady shear value, as shown in Fig. (5.4). The results confirmed in both cases that the overconsolidated yield loci returns closely to the normal instantaneous yield loci after shearing is taken to a steady value. It was also evident that the peak shearing loads \( S_1 \) were significantly higher than the final steady shearing loads \( S_2 \), indicating just how strong a material can become under vibration. To illustrate this point further tests were carried out on the same material to consolidate samples by twisting, overconsolidate them by vibration, and then to shear them over a range of normal values. This
Schematic of overconsolidation peak, steady shear after prolonged shearing and shear at reduced normal lead.

Conical mass flow hopper fitted with vibrating wall panels in lower cone.
was to show the direct increase in yield loci values due to overconsolidation by vibration.

A 48 hour time yield locus was also performed to compare the change in the yield loci due to vibration against that due to time.

The results are all shown on Graph 11. The tests where vibration was used to overconsolidate the samples for yield loci showed a 20-35% increase in strength where as the 48 hour time yield loci gave only a marginal increase in strength above the normal yield locus. Thus for this material vibration can cause much increased strength, while time consolidation has only a marginal effect on strength.

5.6 Effect of a Vibrating Plate on Wall Yield Loci

From observations of vibrating feeders used around the materials handling section, A.I.S., the main function of these feeders is to reduce the amount of friction between a vibrating plate and the material in contact with it. In some cases the vibrating feeders are designed to throw the material clear of the plate, while in others the function appears to be to reduce the friction to a small value so that a slight decline will move the material down the plate. In the former case the acceleration is inclined at some angle to the plate, while with the latter the acceleration is parallel to the plate.

It was decided to test this latter effect by comparing standard Jenike wall yield loci on mild steel plates, with the same tests done when the mild steel plate was actually being vibrated. This would show directly what effect the vibration had on friction. An idealised
GRAPH 11 - PROLONGED SHEAR OF OVERCONSOLIDATED CELLS GIVES $S_3$ YIELD LOCUS
theory was evolved in Section 3.2, by considering the system of forces on the cell during vibration prior to slippage occurring. The predictions from this theory will be compared with the experimental results.

Initial tests were undertaken using the Jenike machine to record the shear force as the plate was being vibrated. At low frequencies the recording pens oscillated severely while following the varying external shear force, so the frequency was increased to 25Hz with the amplitude approximately .009 inches. This gave a steadier pen movement, but readings could still not be obtained satisfactorily at low values of shear force.

To overcome this the tilting device described in Section 5.2 (c) was developed, in which gravity forces were used to shear the cell. This allowed readings to be obtained over a full range of shear force values.

Tests were carried out using the Jenike machine on the following materials with the corresponding graphs:-

<table>
<thead>
<tr>
<th>Material</th>
<th>Moisture Content</th>
<th>Graph</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mt. Newman Fines</td>
<td>0.9% w.b.</td>
<td>Graph 12</td>
</tr>
<tr>
<td>Mt. Newman Fines</td>
<td>9.3% w.b.</td>
<td>Graph 13</td>
</tr>
<tr>
<td>Cockatoo Fines</td>
<td>7.8% w.b.</td>
<td>Graph 14</td>
</tr>
<tr>
<td>Flue Dust</td>
<td>9.9% w.b.</td>
<td>Graph 15</td>
</tr>
</tbody>
</table>

Further tests were done with the tilting device as follows:-

<table>
<thead>
<tr>
<th>Material</th>
<th>Moisture Content</th>
<th>Graph</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mt. Newman Fines</td>
<td>0.9% w.b.</td>
<td>Graph 16</td>
</tr>
<tr>
<td>Mt. Newman Fines</td>
<td>9.3% w.b.</td>
<td>Graph 17</td>
</tr>
<tr>
<td>Cockatoo Fines</td>
<td>8.9% w.b.</td>
<td>Graph 18</td>
</tr>
<tr>
<td>Flue Dust</td>
<td>17.4% w.b.</td>
<td>Graph 19</td>
</tr>
</tbody>
</table>
The vibration used was all 25Hz with approximately .009 inches amplitude, corresponding to a peak acceleration of 0.575 g, assuming a sinusoidal motion. All samples were consolidated exactly the same as the static wall yield tests - 30 twists with 4 lbs normal load.

Graphs 12, 13, 14 and 15 with the Jenike Machine show the shear force during vibration decreasing to 5 - 30% of the static wall yield values. The shear values during vibration were extremely low at low values of normal force. Graphs 16, 17, 18 and 19, using the tilting device, showed very low shear values down to 3% of the static wall yield at low normal force values. However, the shear force during vibration climbed up to 60% of the static values at the high end of the normal force values.

The predicted results from the theory of Section 3.2 confirmed the trend of the experimental results, in general predicting a larger reduction in shear than actually occurred. In Graphs 12, 15, and 16 the predicted results were marginally below the normal force axes, so were not plotted. Graphs 13 and 14 with the Jenike machine showed good agreement for the range of values, while Graphs 17, 18 and 19 with the tilting device gave good agreement at the lower normal force values only.

Some possible reasons for the discrepancies between the idealized model analysed in Section 3.2 and the experimental results are:

(a) The effect of the weight carrier with the majority of weights some distance below the cell causes pendulum behaviour. When this occurs, the cell will not
GRAPH 12 WALL YIELD LOCUS USING JENIKE MACHINE. MT. NEWMAN FINES 0.9% W.B. (ON BLACK STEEL)
GRAPH 13: WALL YIELD LOCUS USING JENIKE MACHINE. MT. NEWMAN FINES 9.3% W.B. (ON BLACK STEEL)
Graph 14  WALL YIELD LOCUS USING JENIKE MACHINE. COCKATOO FINES 7.8% W.B. (ON BLACK STEEL)
GRAPH 17 WALL YIELD LOCUS USING TILTING DEVICE. MT. NEWMAN FINES 9.3% W.B. (ON BLACK STEEL)
GRAPH 18 WALL YIELD LOCUS USING TILTING DEVICE. COCKATOO FINES 8.9% W.B. (ON BLACK STEEL)
GRAPH 19  WALL YIELD LOCUS USING TILTING DEVICE. FLUE DUST 17.4% W.B. (ON BLACK STEEL)
feel the full inertia force because all the mass of the cell-weight system is not rigidly connected to the cell. This will have the greatest effect in the high normal force range because the majority of the cell-weight system is then in the hanging weights.

(b) The amplitude is most likely decreasing at higher loads, which lowers the inertial acceleration and hence increases the friction forces.

Point b) may also explain why the tilting device gave uniformly higher shear force values than the Jenike machine. The tilting device tilts the vibrator, connecting arm and vibrating plate down the slope as well as the cell. Because the vibrator must lift the connecting arm and vibrating plate up the slope each stroke, this puts extra load on the vibrator which reduces its amplitude, particularly at high normal force values where larger slopes were used.

The applications of this vibration effect in promotion of flow are widespread. The "Bridge-Breaker" described by Carroll and Colijn, Ref. (27) is one example. The device reduces the Kinematic angle of friction $\phi'$ appreciably (down to 2-5°), without over-consolidating the material in contact with the plate which would occur if the plate was vibrated towards the material. By reducing $\phi'$, mass flow can be obtained from a core flow bin.

To illustrate this effect the bin design exercise will be repeated in part, to show the improvement that vibrating panels could make for the Mt. Newman fines 8.1% w.b. after 72 hours. Using the results from Graph 17 on
Mt. Newman fines 9.3% w.b. (the closest to 8.1% w.b. Mt. Newman) being vibrated at 25 Hz, the design of a conical mass flow hopper will proceed.

The effective angle of friction (φ) and a flow function curve must be assumed for this exercise; in effect predicting what happens to the shear strength of an element in the bin interior while a vibration of 25 Hz frequency and 0.009 inches amplitude is applied to the material in contact with the wall panels. Since the element of interest is where failure is most likely to occur at an exposed surface of an obstruction, it is by no means certain what will be transmitted to this element from the vibrating panels. Nor could it be assumed that the element in question would gain shear strength from overconsolidation as occurred in the Jenike shear cell, because the element in the bin is not constrained as the shear cell sample is. Since it is quite possible that a reduction in shear strength may occur, it is proposed to use the Flow Function and effective angles of friction for the 8.1% w.b. Mt. Newman fines after 72 hours time delay. This is the closest moisture content to 9.3% w.b. for which yield loci were drawn, and is also most probably conservative.

From Graphs 3 and 17

Estimate $\phi = 63.5^\circ$

Estimate $\phi'$ vibrated = $5^\circ$

For a conical bin an angle $\theta'c$ of $38^\circ$ would be $3^\frac{1}{2}\circ$ inside the mass flow limit as recommended by Jenike, Ref. (1), and this gives a bin flow factor $ff \approx 1.25$. 
The intersection of \( ff = 1.25 \) with the 72 hour Flow Function for 8.1\% w.b. Mt. Newman fines gives \( V_1 = 13.0 \) lbs, \( \bar{V}_1 = 10.4 \) lbs. When the corresponding Mohr circle is drawn on Graph 17, the iterated angle \( \phi' = 2.5^\circ \).

With \( \phi' = 2.5^\circ \), a hopper angle of \( 41^\circ \) would still be \( 3^\circ \) inside the mass flow limit, so mass flow could be expected in the region of the panels. The minimum opening dimension \( B \) to prevent doming is from eqn (3.19),

\[
B = \frac{13\bar{V}_1 \cdot H(\phi')}{y} = \frac{(13)(10.4)(2.62)}{161} = 2.20 \text{ feet}
\]

Thus a conical bin could be designed with \( \theta'c = 41^\circ \) with a minimum opening of 2.20 feet diameter, and where vibrating panels were fitted mass flow would occur. However, without vibration a hopper angle of \( 9^\circ \) or less is needed to give mass flow for steel walls and this material. In the regions where the panels had no effect, core flow would occur if \( \theta'c = 41^\circ \). The bin would need to have vibrating panels fitted to the complete section where \( \theta'c = 41^\circ \) to ensure mass flow.

A good compromise bin would be a conical bin in two sections; a steep section with \( \theta'c \) from \( 9^\circ - 0^\circ \) which would give mass flow without any vibration, and a lower section of \( \theta'c = 41^\circ \) fitted completely with vibrating panels.

It is interesting to note that if the vibrating panels were fitted to the steep \( 9^\circ \) walls a flow factor of 3.0 would result and flow would be retarded. These panels are only useful for \( \theta'c \) angles of \( 40 - 50^\circ \) to ensure that low values of Flow Factor are obtained. See Fig. (5.5) for Mass Flow Bin.
SECTION 6

SUMMARY OF CONCLUSIONS

6.1 Analysis of Steelmaking Raw Materials

The aim of this section of the work was to compile flow measurements on the most difficult materials handled in the Raw Materials Section of A.I.S., and with this information check some bin designs used in the plant to predict flow behaviour in the bins. The materials chosen for testing were Mt. Newman fine iron ore, Cockatoo fine iron ore and Flue Dust (the dust extracted from blast furnace gas).

The analysis of the moisture contents of samples taken from the primary storage yards showed the large variation possible, from 0.9% w.b. Mt. Newman in dry weather up to 13.5% w.b. Flue Dust in very wet weather. Plant experience has shown that actual slurries are being conveyed during bad weather conditions, in such a condition that the material will not convey up a 15° slope. Clearly the worst possible material at its worst moisture content must be catered for in hopper and chute design. Flow tests were done on Mt. Newman fines 0.9%, 5.6% and 8.1% w.b.; Cockatoo fines 0.3% 7.8% and 8.9% w.b.; and Flue dust 1.1%, 5.3%, 9.9%, 13.4% and 17.4% w.b., giving instantaneous yield loci, instantaneous flow functions and wall yield loci for both black steel and stainless steel.

From a comparison of the instantaneous yield loci, the strongest materials were Mt. Newman fines 8.1% w.b.
and Cockatoo fines 7.8% w.b. Time yield loci carried out for both materials showed virtually no shear strength increase for Cockatoo but a large increase for the Mt. Newman fines 8.1% w.b. Thus the strongest material found from these tests was Mt. Newman fines 8.1% w.b. after 72 hours time consolidation.

As well as determining wall yield loci for black steel and stainless steel for Mt. Newman fines 8.1% w.b., loci were also determined for two common lining materials used extensively in the Raw Materials area at A.I.S. - Trelleborg rubber, an impact resisting bin lining for roughs, and Linatex rubber, commonly used for fines to reduce wear and stop material build up. As can be seen on Graph 2, their frictional properties were very similar to that of black steel. Thus the effect of Linatex in minimizing fines build up must be due to its "springiness" rather than its low frictional properties.

Two designs of bins which hold fine ore in the Raw Materials section were then checked using the results from the 8.1% w.b. Mt. Newman fine ore after 72 hours delay. One design was the Fine Ore Bins for No. 2 and 3 Sinter Machines, and the other was the Fines Bins at No. 2 Ore Screening Station. The Fine Ore Bins are cylindrical steel-lined bins with a conical lower section of angle θ°c = 20°, and they discharge onto collecting conveyors from rotary table feeders with stationary ploughs. Each bin has a "donger" fitted - a 3 ft. diameter 4" thick slab of steel suspended by chain to beat the side of the bin. The bins have a history of doming and/or
piping.

The calculations predicted that mass flow would not occur with the 72 hour - 8.1% s.b. Mt. Newman fines for the steel-lined bin, but if stainless steel liners were fitted an angle $\theta$ of 23.4° would result which will probably give mass flow being $1\frac{1}{2}$° inside the mass flow limit. Calculations on the Fine Ore Bins for core flow showed that piping was stable up to 11.9 ft. diameter - over half the width. To prevent doming, the minimum minor dimension of the outlet was found to be 2.28 feet. Since the actual opening is on the side of the bin and is less than 18 inches by 18 inches square, then doming appears to be likely. However with the rotary table feeder underneath the lower sections of a dome, dislocation would occur and the dome should collapse. The problem with these bins would be if a pipe of a greater diameter than the table feeder formed, then the only way to get this material to flow would be by external vibration.

The prediction that the Fine Ore Bins will display core flow for the strongest Mt. Newman fines, and that piping will occur up to very large diameters, concurs with what has been observed to happen in practice. The collapse of pipes under vibration and the widespread Production practice to 'dong' every bin as it is emptied to get the remaining material down onto the rotary feeder illustrates these points.

The second bin design checked, the Fines Bins in No. 2 Ore Screening Station, showed similarly that core flow must be expected. These bins are in two sections,
the lower one an inverted pyramid with one wall vertical and the top section a rectangular wedge section. See Fig.(4.2). The bins each have a large 8'6" x 20' double deck vibrating screen above them and feed out at the bottom via electromagnetic Syntron vibro-feeders. Originally the bins were to be lined with bonded Linatex rubber, but a design change to Trelleborg inverted washboard plates occurred due to the possibility of passing rough ore through the bins.

From the calculations, it was shown that the top section being a plane flow section will give mass flow, if the Trelleborg liners are clean. The actual angle $\theta'$ of $20.6^\circ$, and an angle of $\theta'$ of $19^\circ$ was shown to give good mass flow. Since the plane flow bins have some latitude with this angle, mass flow can be expected.

The lower part of the bin was analysed by comparing the valley angles of the bin with its centreline vertical, against the angle of a conical bin giving mass flow. The valley angle required was $9^\circ$, and to ensure mass flow still occurs when the bin is tipped so that one wall was vertical, it is recommended that the included valley angle not exceed

$$1.5 \times 9^\circ = 13.5^\circ$$

Since the actual bin has a valley angle of $14.9^\circ$ and an included angle of $29.8^\circ$, mass flow is not anticipated.

A repeat calculation was performed to predict what would occur if the Trelleborg inverted washboard liner plates built up completely with ore, as tends to occur in
bad conditions. The kinematic angle of friction was derived from the inclination angle of the 41b yield locus curve for instantaneous 8.1% w.b. Mt. Newman fines, as 45° (compared to 32° for steel). In the top section of the bin this still predicted some degree of mass flow but much less reliable, and in the lower section even less chance of mass flow was predicted.

Core flow calculations are identical to the other bin design, and since the bin opening is 3.2 ft. x 2.3 ft, doming is predicted as well as piping. Although mass flow is predicted for the top section of the bin it is expected that the commencement of core flow in the bottom section will also induce core flow in the top. These predictions are what has actually occurred in practice. The shallow wall of the lower section has built up with ore in bad conditions to a thickness of 4 feet, and has continued building up from the bin opening until it contacted the vibrating screen, causing structural damage. The resultant downtime and cleaning out took approximately one week.

The final part of this section of the work was to design both a conical hopper and a wedge hopper which would have either mass flow or core flow, and to examine the effect of using different wall materials. For a conical bin to obtain mass flow with the 72 hr Mt. Newman 8.1% w.b. fines, a hopper angle less than 9° for steel and less than 18.5° for stainless steel is required, being 3° inside the mass flow limit. The corresponding minimum openings are 1.59 feet and 1.76 feet, being
larger for the stainless steel. The 9° angle is obviously not practical for sufficient storage capacity.

For the conical bin to have core flow, larger hopper angles than 9° and 18.5° are required respectively for steel and stainless steel. For this material, large diameter pipes have been shown to be stable, while the minimum minor dimension for no-doming is 2.28 feet. However it would be senseless to design a bin for core flow with this material because piping is so stable that the live bin capacity would be severely reduced.

The design of the wedge hopper bin with the same material as the conical bin showed how much better plane flow bins are than conical bins. With steel walls a hopper angle of 23° would concur with the recommendations for reliable mass flow, as long as the transition from vertical to angled wall was smoothly radiused. The minimum width B of the slot was 0.76 feet, provided the length of the bin L was greater than 3B. If some dead regions could be tolerated in the bin then the hopper angle could be increased to 35° for steel walls and still provide predominantly mass flow with the opening

\[ B_{\text{min}} = 1.24 \text{ feet}. \]

This produces a flow factor of 1.4 compared to 1.08 for the 23° case.

For stainless steel a hopper angle of 33° would have a flow factor of 1.08 and would give reliable mass flow together with an opening size

\[ B_{\text{min}} = 0.80 \text{ feet}. \]
If some dead regions could be tolerated then the hopper angle could be increased up to 45°. The comparison of these angles to those for the conical bin shows how much better the plane flow bins are from a practical standpoint.

Core flow is predicted for hopper angles in excess of 43° for the steel bin and 51° for the stainless bin, and since the characteristics of core flow are independent of hopper geometry the same observations as for the conical hopper apply here.

These results show the marked differences between inverted pyramid bins, conical bins and plane flow bins. If the advantages of the plane flow bins were widely known there would be many more used in industrial situations. The main problem with the plane flow bins is controlling the feed rate from the long slotted openings while at the same time ensuring that the mass flow characteristic of a fully active hopper outlet is maintained.

6.2 Analysis of Vibration Effects

The aim of this section of the work was to study the effect of vibration on the strength of materials, and on the frictional forces between a vibrating surface and a material such as exists in vibrating baffles or vibro feeders. The first need was to develop a reliable vibrator, and the search went through the range of mechanical out-of-balance vibrators, electrical speakers and electro-
mechanical devices before a suitable device was developed. The mechanical out-of-balance vibrator involved two out-of-balance masses driven counter rotationally by a small variable speed motor, but the inertia of the vibrating section as well as the extraneous vibrations encountered made its use impractical. A stripped down electrical speaker was considered but was not taken further, because the electro-mechanical coil-operated contactor with spring return was by then working satisfactorily. This electro-mechanical device was fed from a variable frequency supply, with a constant power supply. The vibration was induced by deleting one half of the waveform by passing the current through a diode. For all tests the power supply was set at its maximum and all the current passed through the diode, giving maximum shake. If desired both the power supply and the amount of current passing through the diode could have been varied. This would have made amplitude control possible as well as frequency control, but it was necessary at the time to get maximum amplitude. A further piece of equipment was developed which allowed shear and normal forces to be measured for Wall Yield testing, while the plate in contact with the material was being vibrated. This was the tilting device described in Section 5.2(c).

Tests were conducted initially, to see what effect vibrating the top half of a Jenike shear cell had on shear values with a set normal load, after normal consolidation of the cell. How this effect varied with
extra normal load on the cell, extra vibration and varying frequency was also examined. For a 15 second vibration there was a 30-45% increase in shear strength for all frequencies, and a marginally higher increase with an extra 71lbs weight on the cell. The greatest increase in shear strength was due to increasing the time of vibration, and for 3 minutes vibration instead of 15 seconds the increase was 60-80%. The effect of varying frequency could not be singled out for attention because amplitude was also changing at the same time, a trait of electromagnetic feeders – Ref. (24). Further to these vibration tests at set normal loads, vibration tests were carried out at varying normal loads to give yield loci curves. The increased normal force test and the increased frequency test gave substantial increases above both the standard yield locus, and the standard vibration yield locus (10 Hz vibration for 30 seconds with 4.31bs normal force). However the highest yield locus was again for the longest time of vibration. In general, the results indicated that considerable increases in strength can occur in confined samples of ore subject to vibration, particularly prolonged vibration. Particles are being rearranged to form a tighter, more dense packing. The increase in strength and density will reach a maximum when the majority of gaps are filled, so if this procedure were repeated for say 5, 10 and 15 minutes vibration, a level would be reached where shear strength would not increase any further.

To check how significant density changes were
after vibration, tests were carried out using the electro-
mechanical shaker and the sieve shaker to compact ore
samples in the Jenike shear cell. From the known volume
and weight the density was found; the shear force required
to shear the cell was also found. The sieve shaker which
vibrated the whole cell produced a denser packing than the
other vibrator, and both recorded weight increases of up
to 75 - 80 grams for 400 grams of material previously
unconsolidated, an increase of 20%. The sieve shaker also
produced a stronger material, because the whole cell is
being vibrated instead of the top half only. Both machines
showed, in a similar fashion to Ref. (22), a roughly
linear relation between density and strength, with 20%
density increases being correlated with 80 - 90% shear
strength increases. These results were for one material
only at a certain moisture content and it would be unwise
to predict comparable effects for other materials. The
size distribution of particles in the material for example,
would have a critical effect on strength because the
strongest material would occur when fines gave just the
correct volume to fill the voids between the roughs. The
main conclusions to be drawn are that increasing the
density of a material leads to increasing shear strength,
and that vibration is far more effective than normal
consolidation in obtaining a high density material.

In one reference, Ref. (5) on shearing a material
under a certain normal force, the statement was made that
there is a critical density in the shear region and a
critical load at which the material will shear steadily despite the consolidation conditions of that material. As a result of this statement tests were conducted to overconsolidate samples severely, and then to check if under shearing they returned to the steady shear values obtained for a correctly consolidated yield locus. The yield locus for normal consolidation and that for the severely overconsolidated specimen virtually coincided when the steady shear values were used, verifying the predictions of Williams and Birks, Ref. (5). As a comparison, the actual peak shear values were plotted as well as a 48 hour Time Yield Locus for this material (Mt. Newman 0.9% w.b.) The peak shear loads from vibration showed a 20-35% increase above the standard yield locus, while the 48 hour Time Yield Locus gave of the order of a 10% increase only. It appears that part of the work done in the initial shearing of an overconsolidated sample is used to return the density in the region of shear to the critical value. Once this lower density is obtained the shear value becomes steady, and the shear values obtained then are the same as for correctly consolidated samples. It can also be said for this material that far greater overconsolidation is produced by vibration than time delays, for the same consolidation conditions.

The final section of this work has examined what happened to the standard Wall Yield Loci when, during the tests, the plate in contact with the shear cell was actually being vibrated. The results from this could then be used to model vibrating panels in a bin, such as the
Bridge Breaker in Ref. (27).

Early tests were carried out with the Jenike machine, but the reading of very low shear force values was not accurate enough and some doubts were expressed as to the fatigue strength of the machine. The gravity tilting device described in Section 5.2(c) was developed to overcome these problems.

Graphs 12 - 15 produced from the Jenike machine, gave shear values during vibration of 5-30% of the static wall shear values, being lowest for low values of normal force. Graphs 16 - 19 from the tilting device gave shear values down to 3% at the low end but up to 60% of the static values at the high end of the normal force range.

The theory derived in Section 3.2 gave theoretical results which confirmed the trend of the experimental results, except for the large rise in shear values at high normal force values. The theoretical decrease in shear force predicted was generally larger than the decrease obtained. Graphs 13 and 14, done with the Jenike machine, showed excellent agreement between theoretical and experimental predictions. Graphs 17, 18 and 19 with the tilting device gave good agreement at low values of normal force only. It was also noticed that even for the same ore, the tilting device gave shear values consistently higher than the Jenike machine, especially at high values of normal force. The reasons for the discrepancies between theoretical predictions and experimental results possibly are:-

a) The pendulum effect of the weight carrier,
particularly at high normal force values, causes the full inertia force of the cell-weight system not to be felt. This reduction in inertial force causes an increase in shear values.

b) The amplitude is varying with load - a characteristic of electro-magnetic devices which have springs. Increasing load decreased the amplitude, and this lowers the inertial acceleration so increases the friction forces. This was also thought to be the reason for the tilting device having higher shear values than the Jenike machine, since the slope of the tilting device puts extra load on the vibrator which will decrease the amplitude.

From the results obtained it is obvious that at low - medium forces, wall friction can be very substantially decreased by the use of vibration. This could be particularly useful in bins where doming or piping was a problem, and when the vibration was started in the area of dead material, the kinematic angle of friction $\phi'$ would be reduced to such a low value that mass flow would commence and the obstruction would collapse. From a maintenance and design viewpoint though, such devices should only be used to make flow possible in existing bad bins - it would be unwise to deviate from the design of mass flow bins in new design unless absolutely restricted by existing constraints.

The results of the Mt. Newman fines 9.3% w.b. vibrated W.Y.L. were then applied to the bin design problem, assuming the Instantaneous Yield Locus and
Flow Function for the material as being applicable to the material in the bin while it is being vibrated by wall panels. Since the vibration is emanating from wall panels and the element which is being calculated is somewhere in the centre of the bin, it is a reasonable assumption that the only effect is a reduction in wall friction. For a conical bin, a hopper angle of 41° would give mass flow if vibrating panels were fitted to the full area of the 41° section. This compares to 9° without vibration. If vibrating panels are fitted to steep walls they can actually retard flow, so if a bin had to be designed with vibrating panels, the best design would be a steep conical section with θ'c of 9° or less. Then just before the outlet a shallow conical section of θ'c = 41° with vibrating panels should be fitted. The bin could thus have a large storage capacity but still have mass flow.

The work has shown two effects of vibration to be significant:

(1) That a confined sample of material will be overconsolidated by vibration, increasing the density and the shear strength of the material far more effectively than the twisting method and the time delay method of consolidation.

(2) That a confined sample of material will have its kinematic angle of friction θ' reduced to very low values (particularly at the low end of the normal force range) by vibrating the plate in contact with the material.
Section 7

SUGGESTIONS FOR FURTHER WORK

Section 7.1

Further Work from Steelmaking Raw Materials Section

As often happens with work of this nature, the results and conclusions determined point to further areas which need investigation. The following suggestions will attempt to outline some of these areas.

The work done on the Steelmaking raw materials was done on those materials which gave the most difficulty in the Raw Materials Section of A.I.& S., due to the author having been extensively associated with the area. The Sinter Plant Section mixes fine iron ore, coke, limestone, flue dust and other materials to produce the sinter feed, and the fine iron ore content produces most of the build-up problems in the Sinter Plant. Thus, the work done on the iron ores will cover the worst conditions of build-up also in the Sinter Plant.

However, in the Coal Washery Section of A.I. & S., it is known that some types of fine coal have a light oil added to them to improve flowability. This has been necessary to overcome many bin and chute blockage problems occurring with that coal.

Tests should be undertaken on some of these fine coals to establish their flow properties. The results could then be used to great benefit for the redesign of existing plant and as standards for new plant design, for the Coal Washery area.
There is a great need for research into conveyor head chute design and parallel chute design because for these two cases the Jenike theory, Ref.(1), is not sufficient. With conveyor head chutes, only when the chute is completely filled could the Jenike theory be applied to it. This is not normally the case because usually a free stream of material is being thrown from the conveyor head pulley, hitting the back wall of the chute and falling down onto the belt below to be conveyed away. The chute does not fill so no converging stresses are applied to the material.

The inclination angles required for lining materials not to have build-up on them would be very useful for design purposes, because it would determine most of the geometry of conveyor head chutes to prevent build-up of material. See Fig.(7.1). This angle would also be useful in the design of bypass chutes or tripper dogleg chutes, where the chutes are square or rectangular and not converging. See Fig.(7.2). In practice, steel liners with angles of inclination 𝜃 as steep as 70° are known to have fine ore build-up on them. This is much steeper than the worst Kinematic angle of friction of ore on steel - 32°.

When material does build up on walls, the measured angle of inclination of the ore surface 𝛽 has been as steep as 85° with wet iron ore. This is well in excess of the highest angle of effective friction for ore - 63°. The angle 𝛽 would also be useful for conveyor head chute design, because of the common use of impact boxes to prevent wear. See Fig.(7.1).
Fig. (7.1)

Conveyor head chute.

Fig. (7.2) - Tripper dogleg chute.
A theory relating these two angles, $\psi$ and $\beta$, to the physical properties of the bulk solid is needed. From observation, the velocity of the stream of solid impacting the surfaces has a definite effect and would constitute a possible starting point.

**Section 7.2**

**Further Work from Vibration Section**

To continue the vibration work, it would be desirable to construct another vibrator, so that amplitude and frequency were independent of each other. This is difficult with the springs in the present machine because when the forcing frequency is changed, the value of $(\omega/\omega_n)$ changes and this directly affects amplitude. There is some control with the present vibrator on amplitude, by limiting the movement of the moving arm with the lock nuts. However, this artificial limitation on amplitude by end stops ensures the motion will not be sinusoidal and the accelerations obtained would be hard to predict. As pointed out by Gray and Rhodes, Ref.(24), electromagnetic vibrators often lead to predictions of the frequency dependence of some aspect, where an identical investigation with mechanical vibrators reveals no such dependence. The solution here would be to build a mechanical vibrator such as the cam-type, with a higher G-range than the present machine and with amplitude recording.

The new machine could then test the frequency and amplitude dependence of:-
(i) The overconsolidation effect of vibration as discussed in Section 5.3;

(ii) The reduction in wall friction under vibration as discussed in Section 5.6.

The use of vibration in clearing material blockages deserves further research. Carroll and Colijn, Ref.(26), speculated that high frequency, low amplitude vibration towards the hopper centre is the best for reducing wall friction, while low frequency, high amplitude vibration is the best for breaking up arches. Their contention was that low frequency vibration had improved penetration; contradicting Ref.(23). These predictions would be worth testing using a variable angle, plane flow perspex hopper with vibrators attached to the walls. The wall friction tests could record the acceleration required at a particular frequency before slippage at the wall occurred, for a given hopper-material combination. This would indicate which frequency and which acceleration gave the best reduction in wall friction. For the arching tests, the hopper would need to be set up to obtain reproducible arching. Then the acceleration required at a particular frequency to collapse the arch would be obtained and a series of tests would indicate which frequency and which acceleration collapsed the arch most easily.

These tests are involved because many factors, including material type and size fraction, material wetness, uniform compaction of material in the hopper and hopper stiffness, affect the passage of vibration. However, there is so little experimental work available in this area and so much application for it, that the work should be undertaken.
SECTION 8

LIST OF REFERENCES


7) Lee, C.A., "Hopper Design Up to Date", Chemical Engineering, pp 75-78. April 1, 1963.


SECTION 9

APPENDIX

9.1 GRAPHICAL RESULTS FOR STEELMAKING RAW MATERIALS

GRAPHS 20 - 41
GRAPH 21 WALL YIELD LOCUS. MT. NEWMAN FINES 0.9% W.B.
GRAPH 22
INSTANTANEOUS YIELD LOCUS. MT. NEWMAN FINES 5.6% W.B.
GRAPH 23 WALL YIELD LOCUS. MT. NEWMAN FINES 5.6% W.B.
GRAPH 24
INSTANTANEOUS YIELD LOCUS. COCKATOO FINES 0.3% W.B.
INSTANTANEOUS YIELD LOCUS. COCKATOO FINES 4.0% W.B.
GRAPH 27 WALL YIELD. LOCUS. COCKATOO FINES 4.0% W.B.
GRAPH 29 WALL YIELD LOCUS. COCKATOO FINES 7.8% W.B.
GRAPH 30  INSTANTANEOUS YIELD LOCUS. COCKATOO FINES 8.9% W.B. (NEARLY SATURATED)
GRAPH 31  WALL YIELD LOCUS. COCKATOO FINES 8.9% W.B.
Graph 37: Wall yield locus. Flue dust 9.9% W.B.
GRAPH 39 WALL YIELD LOCUS. FLUE DUST 13.4% W.B.
GRAPH 40  INSTANTANEOUS YIELD LOCUS  FLUE DUST 17.4% W.B.
GRAPH 41  WALL YIELD LOCUS. FLUE DUST 17.4% W.B.