OKLAHOMA INSTITUTE OF TECHNOLOGY
OF THE
OKLAHOMA AGRICULTURAL AND MECHANICAL COLLEGE
SCHOOL OF CHEMICAL ENGINEERING
STILLWATER

March 31, 1953

Mr. Keith Buell, Manager
Chemical Engineering Division
Phillips Petroleum Company
Bartlesville, Oklahoma

Dear Mr. Buell:

In accordance with the agreement between Mr. Richardson and Dr. Bartlett, Head of the department at that time and now deceased, it is agreed that the thesis by Mr. Allan Richardson, "Gas-Lift for Commercial Moving-Bed Heat Exchanger", and its abstract will be kept in the library under the restrictions of not being allowed for public use by anyone until such time as the proper authorities from Phillips Petroleum Company shall release it for public use. A copy of this letter will be attached to each thesis and abstract and will remain so attached until proper release is given.

Very truly yours,

Charles L. Nickolls
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Acting Head

CLN:ek

Date of Degree: May 25, 1953

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Title of Study: GAS LIFT FOR INDUSTRIAL MOVING-BED HEAT EXCHANGER

Number of Pages in Study: 47 Candidate for What Degree: Master of Science

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Statement of Problem: One of the essential parts of an industrial, moving-bed heat exchanger is a mechanism to elevate ceramic pellets from the bottom of the unit and return them to the system, thereby completing the cycle. A gas lift for transporting 3/8-inch diameter pellets in a large scale unit was proposed as an improvement over a bucket type elevator formerly used. Investment, maintenance and operating costs should be less for the gas lift than for the bucket hoist.

Method of Procedure: A 3-inch air lift was operated for a series of test runs in order to obtain design data for an industrial installation. A correlation was obtained which permitted extrapolation to larger pipes and greater rates of pellet flow.

Findings and Conclusions: The data obtained at lift-air temperatures up to 1000°F were in satisfactory agreement with published correlations of the terminal velocities of spheres in air. Approximately four-fifths of the total pressure drop was accounted for by known "heads". The remaining 20 per cent of the observed loss was attributed to pellet acceleration and pellet friction losses. A correlation of the experimental data indicated that 2245 SCFM of air delivered at 3.1 psig pressure and 1000°F temperature to the base of a 10-inch lift pipe would satisfactorily lift the 3/8-inch diameter pellets at rates of up to 90,000 pounds per hour. Since start-up conditions require up to 50 per cent increase in the air volume, the use of steam is suggested to supplement the air flow at lower temperatures. Additional pressure at the blower discharge would be necessary to allow for pressure drop through the burner, control valves and auxillary piping. A reduction in blower investment may be realized if superheated steam is used as all or a part of the lifting medium at operating temperatures. A choice between the steam and the air lift will depend on the comparison between the blower investment and the cost of steam at the plant installation and on alleviation of operating difficulties associated with water from condensed steam in the lift pipe. Stable operation can be achieved if the pellets are transported from four to five feet above the upper end of the lift pipe before they are allowed to fall into the return duct which carries them to the heat exchange unit. The pellet flow rate can be ascertained by measurements of the differential pressure across the top section of the lift pipe.

ADVISER'S APPROVAL [signature]

GAS LIFT FOR INDUSTRIAL MOVING-BED

HEAT EXCHANGER

By

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Bachelor of Science in Chemical Engineering

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1941

Submitted to the faculty of the Graduate School of
the Oklahoma Agricultural and Mechanical College
in partial fulfillment of the requirements
for the degree of
MASTER OF SCIENCE
May, 1953

GAS LIFT FOR INDUSTRIAL MOVING-BED

HEAT EXCHANGER

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PREFACE

In connection with his professional employment, the writer was asked to investigate the development of a correlation which would allow extrapolation of semi-works scale gas-lift data to industrial size. As a result of this investigation a gas-lift was designed to elevate ceramic pellets in a large scale installation to complete the cycle of a moving-bed heat exchanger. This thesis reports the development of the aforementioned correlation.

The writer wishes to express his appreciation to the members of the Chemical Engineering Division of the Research and Development Department of Phillips Petroleum Company in Bartlesville, Oklahoma for their helpful advice and assistance and for their willingness to allow the data contained herein to be used as the basis for this thesis. The writer also wishes to express his appreciation to the staff of the Chemical Engineering Department of Oklahoma Agricultural and Mechanical College for their excellent advice in the critical reading of this thesis.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF ILLUSTRATIONS

I. INTRODUCTION

One of the essential parts of an industrial moving-bed heat exchanger is a mechanism used to elevate ceramic pellets from the bottom of the unit and return them to the system, thereby completing the cycle. A gas lift for transporting 3/8-inch diameter pellets in a large scale unit was proposed as an improvement over a bucket type elevator formerly used. Investment, maintenance and operating costs for the gas lift should be less than for the bucket hoist.

A 3-inch diameter air lift was operated for a series of test runs in order to obtain design data for an industrial installation. These data were examined with respect to published data on the transport of spheres in various fluids. A summation of all the calculable energy heads was compared with the total observed pressure drop at various pellet flow rates. From the resulting correlation, the design specifications for an air lift for a large scale installation were estimated.

II. EXPERIMENTAL PROCEDURE

A series of test runs was made to obtain data on elevating 3/8-inch ceramic pellets using air in a 3-inch diameter by 41.5-foot high insulated carrier pipe. Figure 1 is a diagramatic sketch of the apparatus used. Lift air was supplied to the bottom of the pipe at controlled temperatures from 133°F to 1010°F. No measurement of the air temperature at the top of the lift pipe was obtained. Pellets were supplied to the vertical carrier at rates ranging from 100 to 8000 pounds per hour. The higher pellet rate was not limited by the lift line capacity but rather by the capacity of the pellet feeding mechanism. Lift air was prevented from flowing into the pellet supply system by a sealing mechanism. During normal operation of a gas lift, the air temperature would be controlled to approximate the pellet temperature. The experimental data presented herein were obtained with the pellets entering the lift at atmospheric temperature. The comparatively cool pellets removed some of the heat from the air and, consequently, estimated values of fluid requirements for an industrial installation, based on these data would be somewhat high.

A pellet disengaging section was located at the top of the air lift pipe. This vessel was 20 inches in diameter and 8 feet high. As the air issued from the carrier pipe it expanded into this larger diameter vessel, thereby losing much of its velocity. The pellets, no longer being carried by the lifting velocity of the air, fell to the bottom of the disengaging section from which they flowed back to the heat exchange unit.

At each of the temperature conditions, data were obtained at "maximum", "normal" and "minimum" air flow. The "maximum" air flow was defined as the quantity of air at lift temperature which would

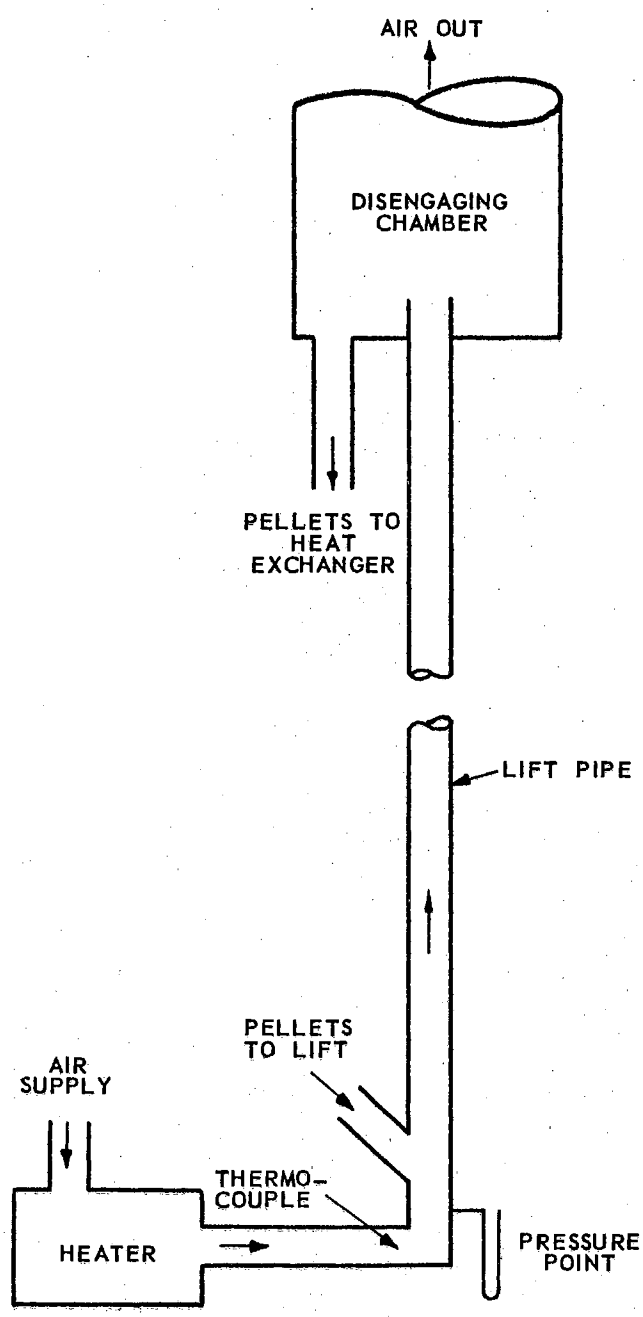

FIGURE 1 DIAGRAMATIC SKETCH OF AIR LIFT APPARATUS

transport the pellets up the lift pipe and to within a short distance of the top of the disengaging vessel. This quantity of air was determined by transporting pellets with an oversupply of air and then slowly reducing the hot air flow until the majority of the pellets no longer hit the top of the disengaging section. It was estimated that the average distance travelled by the pellets past the top of the carrier pipe, under these conditions, was about seven feet. The "minimum" air flow was defined as the quantity of air at lift temperature which just barely transported all of the pellets over the top of the lift pipe. To determine the "minimum" air flow, more than sufficient air to lift all of the pellets was supplied. This air flow was gradually reduced until plugging of the lift pipe indicated insufficient air was available.

"Normal" operation carried the pellets into the disengaging chamber at velocities between the "maximum" and "minimum" extremes. At "normal" and "minimum" air rates, the vertical travel of the pellets beyond the top of the air lift pipe was approximately four feet and two feet respectively, before the pellets fell from the air stream. Accurate measurements of pellet travel into the disengaging chamber could not be made because of the construction of the vessel.

The total pressure drop across the 41.5-foot air-lift pipe was read from a water manometer which was connected just below the duct which carried the pellets to the carrier pipe. Since the disengaging chamber at the top of the air lift was open to the atmosphere, the pressure drop through the pipe was essentially equal to the static pressureat the bottom of the pipe.

III. FACTORS INFLUENCING GAS-LIFT PERFORMANCE

The basic data needed for the design of a gas lift are the gas flow rate required for transport and the anticipated pressure drop. These two parameters are defined when the pipe size, the velocities of the gas and of the solid, and the physical properties of the gas and of the solid are determined.

When the fluid velocity through the pipe exceeds the terminal velocity of the particles, the particles take on a motion relative to the pipe in the direction of the flowing fluid. The terminal velocity is sometimes defined as the gas velocity necessary to suspend a particle at zero net velocity. The difference between the fluid velocity and the net pellet velocity is equal to the terminal velocity of the air. In equation form:

$$V_t = V_a - V_p \qquad (1)$$

where:

V_t is the terminal velocity of air necessary to freely suspend the particles

V_a is the true air velocity at flowing temperature

V_p is the pellet velocity under steady state conditions

The pellet velocities, V_p, in the lift will vary at different levels in the pipe. At the bottom of the lift, the pellets will be undergoing acceleration and the bulk density at this level will be high. Near the upper end of the carrier pipe a steady state will exist where the ratio of pellet velocity to fluid velocity will be essentially constant. An estimate of the pellet velocity in this steady-state section of the lift may be obtained from the average distance the pellets travel after leaving the lift pipe, since at that point their energy will be spent only in overcoming the force of gravity. Mathematically this may be expressed:

$$V_p = \sqrt{2gS}$$

where:

g is the acceleration due to gravity

S is the vertical distance travelled by the pellets after leaving the effect of the lift air

The bulk density of the pellets in the steady-state section is a function of the pellet velocity, pellet flow rate and the cross-sectional area of the pipe.

$$\rho_b = \frac{W_p}{A\ V_p} \qquad (3)$$

where:

ρ_b is the bulk density of the pellets in the pipe

W_p is the weight of pellets being elevated per unit time

A is the cross-sectional area of the pipe.

This bulk density term is required to determine the true gas velocity at the flowing conditions since it is a measure of the reduction in the cross-sectional area of the pipe due to the pellets. The expression, $(1 - \rho_b/\rho_s)$, indicates the proportion of the pipe area, A, which is free space. The true gas velocity is, therefore:

$$V_a = \frac{Q_a}{A\ (1 - \rho_b/\rho_s)} \qquad (4)$$

where:

Q_a is the volume of fluid per unit time at flowing temperature

ρ_s is the density of the individual pellets based on the weight of a unit volume of the material of composition

Examination of Equation (3) indicates the possibility of estimating the pellet velocity (i.e., kinetic energy) at <u>any</u> section of the lift by measuring the bulk density in the various sections.

The total energy loss in transporting the pellets through the lift pipe may be measured in terms of pressure drop as the sum of the contributing "heads" against which the work is done.

$$\Delta P_T = \Delta P_a + \Delta P_s + \Delta P_{V_p} + \Delta P_{Pel} + \Delta P_{Acc} \tag{5}$$

where:

ΔP_T is the total pressure drop in the gas lift

ΔP_a is the pressure drop due to air friction on the wall of the carrier pipe

ΔP_s is the pressure drop due to the head of pellets in the gas lift

ΔP_{V_p} is the pressure drop due to the velocity (i.e., kinetic energy) of the pellets at the top of the lift pipe

ΔP_{Pel} is the pressure drop due to pellet friction on pellets and on the wall of the lift pipe

ΔP_{Acc} is the pressure drop lost to acceleration of the pellets

The pressure drop due to the friction of the air on the walls of the carrier pipe may be evaluated from the Fanning equation:

$$\Delta P_a = \frac{\rho_a f V_a^2 H}{2gcD} \tag{6}$$

where:

ρ_a is the density of the fluid

f is the friction factor

H is the length of the air lift pipe

gc is a constant for dimensional conversion = 32.17 ft lb mass/lb force sec^2

D is the diameter of the carrier pipe

The loss required to support the column of pellets in the pipe is:

$$\Delta P_s = \rho_b H \tag{7}$$

The additional energy required to give the pellets their exit velocity is a function of the kinetic energy of the pellets at the top of the lift. The pressure drop necessary to give the pellets this amount of energy can be evaluated by the relation:

$$\Delta P_{V_p} = \rho_b(V_p^2/2g) \tag{8}$$

The pellet friction loss, ΔP_{Pel}, and the acceleration loss, ΔP_{Acc}, are complex terms, the total of which is best determined by difference. In a detailed set of data the two terms may be subdivided by obtaining pressure readings and pellet densitities in both the steady-state section and the acceleration section of the lift pipe. In the analysis herein reported the two terms were grouped together.

IV. CORRELATION OF DATA

In the correlation of the experimental data obtained for this work the following quantities remain constant:

D - diameter of the lift pipe - 0.256 ft

A - cross-sectional area of the lift pipe - 0.0513 ft^2

H - height of the lift pipe - 41.5 ft

D_s- average pellet diameter - 0.0313 ft

ρ_a- density of air at 60°F and 1 Atm - 0.0763 lb/ft^3

ρ_s- density of the individual pellets based on the weight of a unit volume of the material of composition - 249.0 lb/ft^3

gc- constant for dimensional conversion - 32.17 ft lb mass/lb force sec^2

The experimental data obtained during the course of this study appear in the appendix as Table I, Experimental Data on 3-Inch Air Lift For 3/8-Inch Diameter Pellets. These data include the lift air temperature, the air flow at lift temperature, the pellet flow and the total pressure drop as well as the estimated vertical distance the pellets travelled into the disengaging section before their upward motion was overcome by the force of gravity.

Table II, Estimation of Pressure Heads For 3-Inch Air Lift, is a tabulation of the estimated pressure "heads" for the 3-inch air lift calculated from the experimental data. The discussion which follows describes the evaluation of the various terms which appear in the table. Data point (1) will be used for example purposes.

For the three conditions of pellet flow, namely, "maximum", "normal" and "minimum", it was estimated that the pellets continued their vertical rise seven, four and two feet respectively after leaving the carrier pipe. Substituting in Equation (2) for data point (1) the average pellet velocity was:

$$V_p = \sqrt{2 \times 32.2 \times 7} = 21.3 \text{ ft/sec}$$

The bulk density of the pellets, ρ_b, was obtained by substitution in Equation (3). For data point (1):

$$\rho_b = \frac{1250}{0.0513 \times 21.3 \times 3600} = 0.318 \text{ lb/ft}^3$$

To calculate the velocity of the air, V_a, substitution in Equation (4) for data point (1) yielded:

$$V_a = \frac{555}{0.0513\ (1 - 0.318/249)\ 60} = 180.5 \text{ ft/sec}$$

Data were obtained on the flow of air through the three-inch pipe at different temperature levels without pellet flow in order to determine ΔP_a, the pressure drop due to the friction of the air on the walls of the carrier pipe. Table III, Calculation of ΔP_a With No Pellet Flow, presents the data obtained and a listing of the calculated values. In Equation (6)

$$\Delta P_a = \frac{\rho_a f V_a^2 H}{2gcD}$$

The value of ρ_a is

$$\frac{0.0763 \times 520}{T^oR} = \frac{39.7}{T^oR}$$

From data point (46)

$$\rho_a = \frac{39.7}{1410} = 0.0282 \text{ lb/ft}^3$$

Similarly:

$$V_a = \frac{Q_a}{0.0513 \times 60} = \frac{Q_a}{3.08}$$

For data point (46)

$$V_a = \frac{550}{3.08} = 169.4 \text{ ft/sec}$$

The friction factor is here defined as:

$$f = \frac{2gcD\Delta P}{V_a^2 H \rho_a} \tag{9}$$

where:

f is the friction factor or drag coefficient

D is the pipe diameter in feet

ΔP is the pressure drop in lb/ft^2

In order to simplify the friction factor with known quantities let:

$Q_a = ft^3/min$

Δ_p = pressure drop in inches of water

T_a = temperature of the air feed in degrees Rankine

then:

$$\Delta P = \Delta_p \frac{144}{27.7} = 5.1984 \Delta_p$$

$$V_a = \frac{Q_a 4}{60 \pi D^2}$$

$$V_a^2 = \frac{Q_a^2 16}{3600 \pi^2 D^4}$$

Substituting in Equation (9):

$$f = (2)(32.17)(0.256)(5.1984)(\Delta_p) \frac{(3600)(\pi^2)(0.256)^4}{(Q_a^2)(16)} \frac{1}{41.5} \frac{T_a}{39.7}$$

or:

$$f = 0.498 \frac{\Delta_p T_a}{Q_a^2}$$

For data point (46):

$$f = 0.498 \frac{(6.3)(1410)}{(550)^2} = 0.0142$$

Substitution of the now known values for data point (46) in Equation (6):

$$\Delta P_a = \frac{(0.282)(0.0142)(169.4)^2(41.5)}{(2)(32.17)(0.256)} = 5.7 \text{ in. } H_2O$$

The above calculation was carried out for data points (46) through

(83) as shown in Table III. As indicated by the dividing lines in the table, sets of data were obtained at three temperature levels to correspond roughly with the temperature levels used for obtaining data with pellet flow.

Figure 2 is a plot of the data in Table III and shows the pressure drop due to friction of air on the walls of the carrier pipe at various air flow rates for the three temperature levels.

The values of ΔP_a for the data obtained with flowing pellets, as indicated in Table II were obtained from Figure 2. For data point (1) where $Q_a = 555$ at $T_a = 935$:

$$\Delta P_a = 5.70 \text{ in. } H_2O$$

The values of ΔP_s were obtained by substitution in Equation (7). for data point (1):

$$\Delta P_s = \frac{(0.318)(41.5)(27.7)}{(144)} = 2.50 \text{ in. } H_2O$$

The pressure drop due to the velocity of the pellets, ΔP_{V_p}, was obtained by substitution in Equation (8). On data point (1):

$$\Delta P_{V_p} = \frac{(0.318)(21.3)^2\ (27.7)}{(2)(32.2)(144)} = 0.43 \text{ in. } H_2O$$

A summation of the calculated pressure "heads" for data point (1) is then found to be:

$$\Delta P_a + \Delta P_s + \Delta P_{V_p} = 5.70 + 2.50 + 0.43 = 8.63 \text{ in. } H_2O$$

As previously discussed, the pellet friction loss, ΔP_{Pel}, and the acceleration loss, ΔP_{Acc}, were obtained by difference. From Equation (5) we have for data point (1):

$$12.0 = 8.63 + \Delta P_{Pel} + \Delta P_{Acc}$$

or:

$$\Delta P_{Pel} + \Delta P_{Acc} = 3.37 \text{ in. } H_2O$$

The foregoing term also includes any errors in calculation, in original data or in estimation of the distance the pellets travelled after leaving

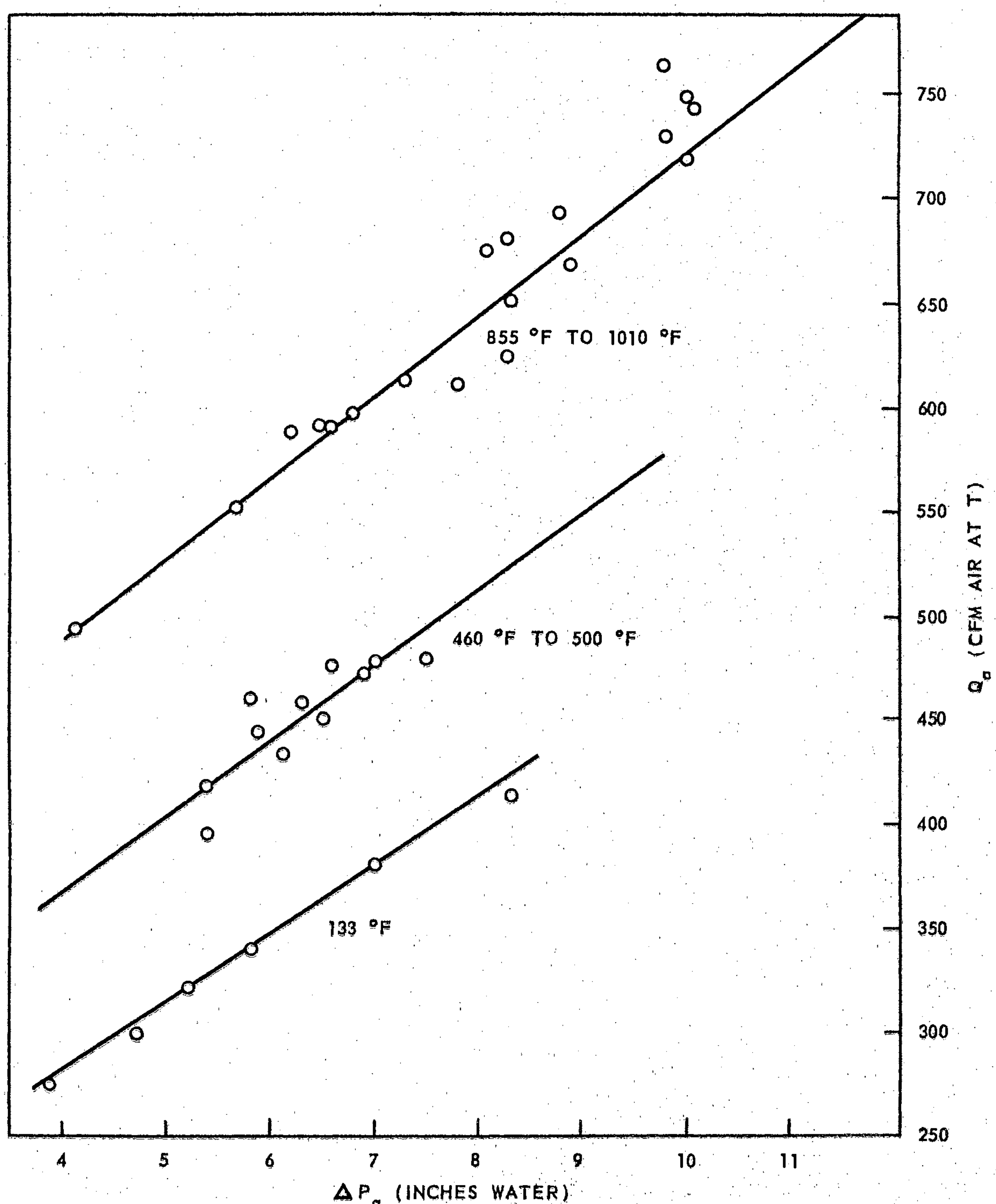

FIGURE 2 PRESSURE DROP DUE TO FRICTION OF AIR ON WALLS OF THE CARRIER PIPE

the lift pipe.

Because the pellet velocities were not determined exactly, little correlation was obtained between the per cent pressure drop unaccounted for and the pellet flow rate. An average of about 80 per cent of the energy requirements was attributed to known "heads". For design purposes, therefore, a factor of 1.25 will account for the losses to pellet acceleration and pellet friction. The percentage of the energy lost to these two factors should be little different in the prototype of the experimental air lift if the pellet density in the designed carrier does not exceed that reported in these data.

The estimated terminal velocities of the pellets in air are presented in Table IV, Terminal Velocity of 3/8-Inch Pellets in Air, and were calculated from Equations (2), (3), (4) and (1). For data point (1):

$$V_p = \sqrt{(2)(32.2)(7)} = 21.3 \text{ ft/sec}$$

$$\rho_b = \frac{(1250)}{(0.0513)(21.3)(3600)} = 0.318 \text{ lb/ft}^3$$

$$V_a = \frac{555}{(0.0513)(1 - 0.318/249)(60)} = 180.5 \text{ ft/sec}$$

$$V_t = 180.5 - 21.3 = 159.2 \text{ ft/sec}$$

Figure 3 is a plot of the calculated terminal velocities of the pellets in air versus the flowing air temperature. The data of Waddel[1] as calculated for similar pellets in air are superimposed upon the same figure. No attempt was made to draw a curve through the experimental data points since several points were obtained at only three temperature levels. Although the experimental data points do not duplicate the curve calculated from published data, the agreement is considered satisfactory since

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[1] H. Waddel. Journal of the Franklin Institute, 217 (1934) p. 459

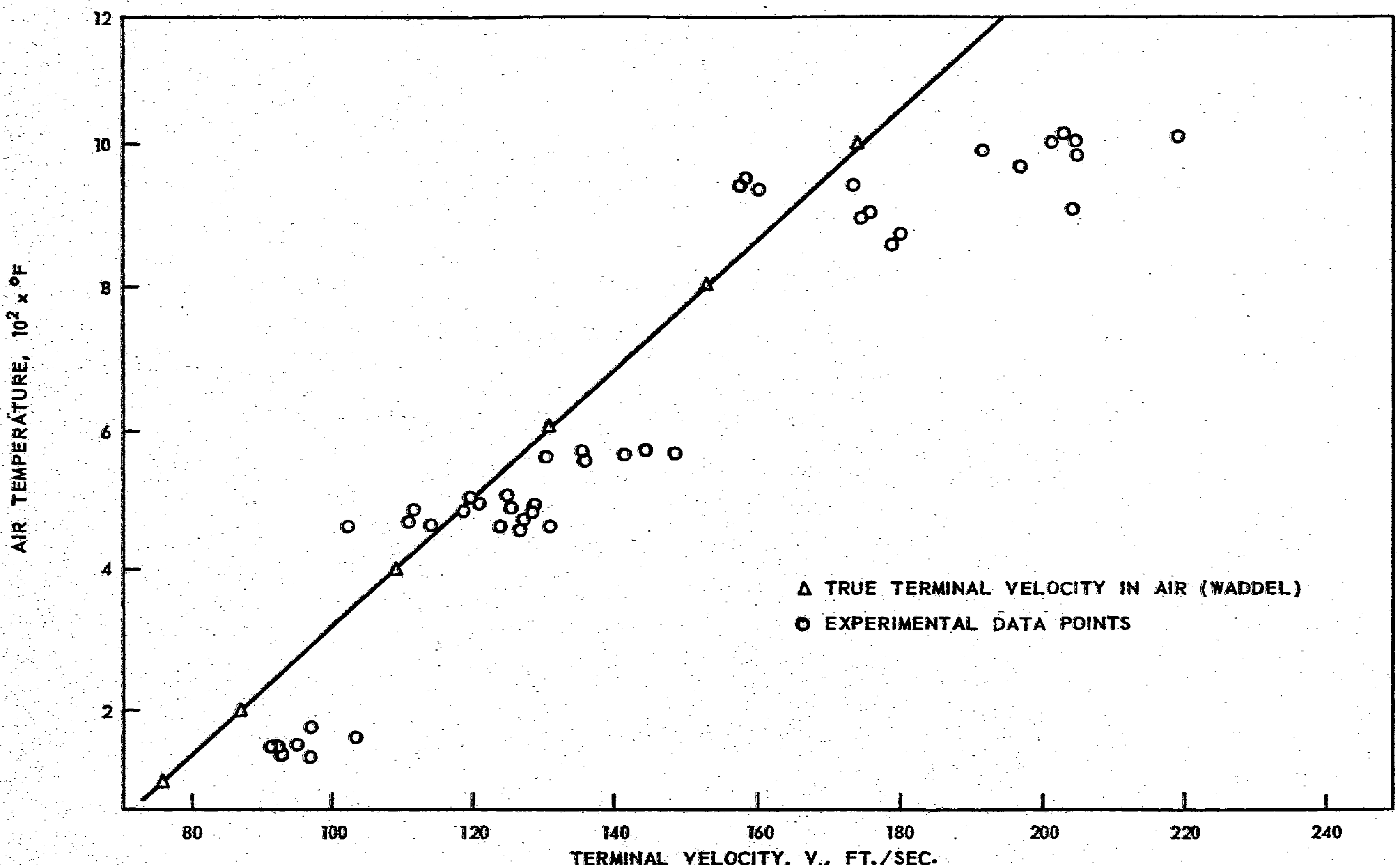

FIGURE 3 TERMINAL VELOCITY OF 3/8–INCH PELLETS IN AIR

the terminal velocities calculated from the data obtained were based on the estimated average distances travelled by the pellets above the carrier pipe. The agreement is sufficient to confirm the more exact data of Waddel.

The data of Waddel appeared as a plot of the Reynolds number versus the friction factor for spheres. Figure 4 is a reproduction of the portion of the curve which is applicable to the conditions encountered in this study.

The Reynolds number is defined as

$$Re = \frac{D_s V_t \rho_a}{\mu_a} \tag{10}$$

where:

Re is the Reynolds number

D_s is the average diameter of the pellets

μa is the viscosity of the fluid

The form of the friction factor used here is:

$$f_D = \frac{4(\rho_s - \rho_a) g D_s}{3 V_t^2 \rho_a} \tag{11}$$

where:

f_D is the friction factor

Since the terminal velocity term, V_t, appears in both parameters it was necessary to combine the two parameters to form a new dimensionless factor, R, in such a way as to eliminate the velocity term. By definition:

$$R = Re^2 f_D \tag{12}$$

where:

R is a new dimensionless number

Substitution in Equation (12) gives:

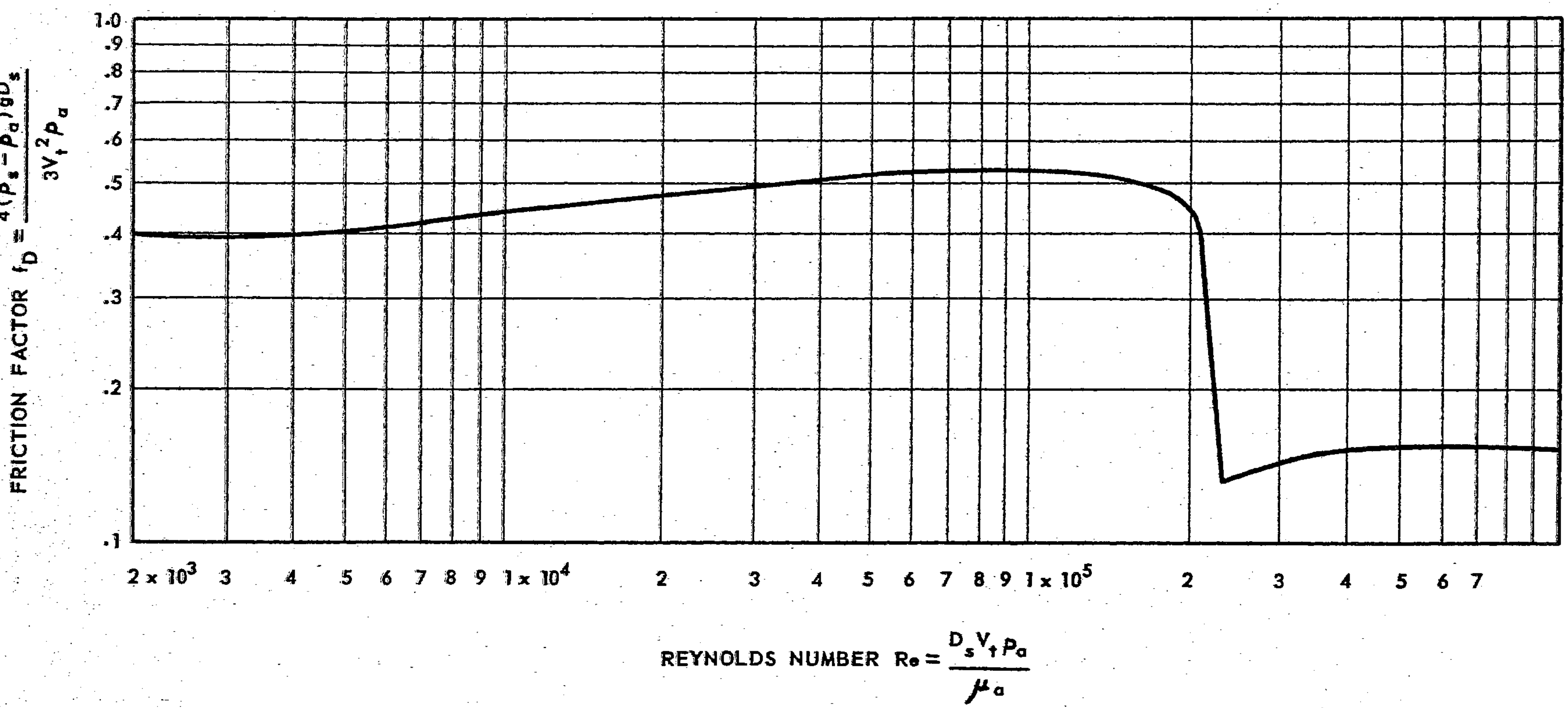

FIGURE 4 TRANSPORT OF "FREE-FALLING" SPHERES
(WADDEL, J. FRANKLIN INST. 217 (1934)

$$R = \frac{(D_s^{\,2} V_t^{\,2} \rho_a^{\,2})}{\mu_a^{\,2}} \left[\frac{4(\rho_s - \rho_a) g D_s}{3 V_t^{\,2} \rho_a} \right] \qquad (13)$$

$$R = \frac{4}{3} \frac{D_s^{\,3} \rho_a g(\rho_s - \rho_a)}{\mu_a^{\,2}}$$

Since the density of the fluid is very small compared with the density of the solid it was assumed in this study that:

$$\rho_s - \rho_a = \rho_s \qquad (14)$$

For the pellets used in this study:

$D_s = 0.0313$ ft

$\rho_s = 249.0$ lb/ft^3

then substituting in Equation (13) and (14):

$$R = \frac{4}{3} \frac{(0.0313)^3(32.2)(249)}{(0.672 \times 10^{-3})^2} \quad \frac{\rho_a}{\mu_a^{\,2}}$$

or:

$$R = 2.18 \times 10^5 \frac{\rho_a}{\mu_a^{\,2}} \qquad (15)$$

where:

μ_a is the viscosity in centipoises

A plot of the Reynolds number versus the new dimensionless number, R, is presented as Figure 5 and was obtained from the data listed in Table V, Calculation of Dimensionless Number R From Figure 4. For example, using Equation (12), for the first point listed:

$$R = (1 \times 10^3)^2(0.44) = 4.4 \times 10^5$$

Since R is a function of ρ_a and μ_a, it is also a function of temperature. A temperature scale was therefore superimposed on Figure 5. To locate the temperature scale it was necessary to calculate values of R at various temperature levels. The viscosity was obtained from an alignment chart in Perry[1]. Calculation of the foregoing relationship

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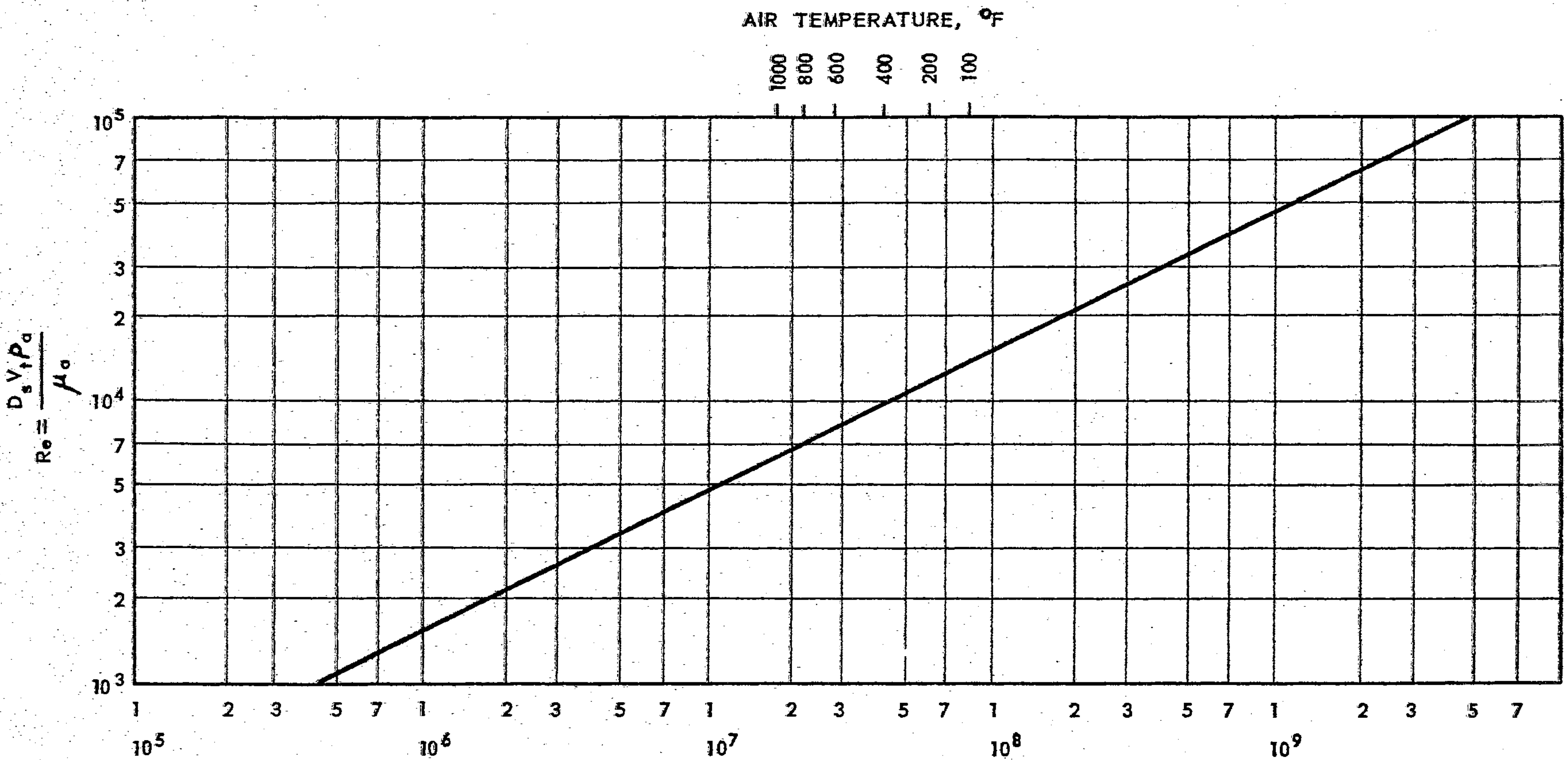
[1]J. Perry. <u>Chemical Engineers Handbook</u>, 2nd Ed., (1941) p. 791



$$R = Re^2 f_D = \frac{4}{3}\,\frac{D_s{}^3 \rho_s g(\rho_s - \rho_a)}{\mu_a{}^2}$$

FIGURE 5 REYNOLDS NUMBER IN TERMS OF R AND T appears in Table VI, Calculation of Dimensionless Number R From Equation (15). For example:

$$\text{at } 100^{\circ}\text{F} \quad \rho_a = 7.08 \times 10^{-2} \text{ lb/ft}^3$$

$$\mu_a = 1.87 \times 10^{-2} \text{ centipoises}$$

then:

$$R = (2.18 \times 10^5) \frac{(7.08 \times 10^{-2})}{(1.87 \times 10^{-2})^2} = 8.26 \times 10^7$$

The exact data of Waddel[1] could then be applied to calculate the terminal velocity, V_t, at various temperatures. The straight line shown on Figure 3 represents this plot. Values of V_t were calculated from the Reynolds numbers obtained at various temperatures from Figure 5. For example at 100°F:

$$Re = 1.35 \times 10^4$$

then:

$$1.35 \times 10^4 = \frac{(0.0313)V_t(7.08 \times 10^{-2})}{(0.672 \times 10^{-3})(1.87 \times 10^{-2})}$$

or:

$$V_t = 75.2 \text{ ft/sec}$$

Values obtained at the several other temperature plotted appear in Table VII, Terminal Velocity of 3/8-Inch Pellets in Air.

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[1] Waddel, p. 459.

V. PROPOSED DESIGN FOR INDUSTRIAL INSTALLATION

The calculation of a design of an air lift for an industrial installation follows. This design was based on a 10-inch, Schedule 40, steel pipe, 90-feet long. An 8-foot high disengaging chamber at the top of the carrier pipe was proposed, and velocities sufficient to transport the pellets four and three-fourths feet into the disengaging chamber would be used. The maximum lift-air temperature was assumed to be 1000°F. The maximum pellet flow rate was assumed to be 90,000 pounds per hour.

The pellet velocity in the steady-state section of the pipe by Equation (2) is:

$$V_p = \sqrt{(2)(32.2)(4.75)} = 17.5 \text{ ft/sec}$$

The bulk density of the pellets in the carrier pipe is within the range of densities covered by the experimental data; from Equation (3):

$$\rho_b = \frac{90,000}{(0.548)(17.5)(3600)} = 2.60 \text{ lb/ft}^3$$

The required air rate may be calculated from Equations (1) and (4) by using the terminal velocity correlation shown as the straight line on Figure 3:

$$V_a = 174.0 + 17.5 = 191.5 \text{ ft/sec}$$

$$Q_a = 191.5(0.548)(1 - 2.60/249)(60) = 6300 \text{ CFM}$$

The air-volume specification at standard conditions is:

$$6300(520/1460) = 2245 \text{ SCFM}$$

The pressure at which the blower must deliver 2245 SCFM of air to the bottom of the carrier pipe is calculated in a stepwise fashion from Equation (5). A standard friction-factor chart is used to determine the friction of the air on the walls of the pipe. Under these conditions, by Equation (6):

$$\Delta P_a = \frac{(0.0272)(0.0165)(191.5)^2(90)(12)}{(2)(32.17)(10.02)} \quad \frac{(27.7)}{(144)} = 5.3 \text{ in. } H_2O$$

From Equation (7):

$$\Delta P_s = (2.60)(90)\frac{(27.7)}{(144)} = 45.2 \text{ in. } H_2O$$

From Equation (8):

$$\Delta P_{V_p} = \frac{(2.60)(17.5)^2}{(2)(32.17)} \quad \frac{(27.7)}{(144)} = 2.4 \text{ in. } H_2O$$

Accounting for the acceleration and pellet friction loss by a factor of 1.25, the total calculated pressure drop from Equation (5) is:

$$\Delta P_T = (5.3 + 45.2 + 2.4)(1.25) = 66.2 \text{ in. } H_2O$$

or:

$$\Delta P_T = \frac{66.2}{27.7} = 2.39 \text{ psig}$$

If a factor of safety of 1.3 is used, the operating pressure at the bottom of the lift pipe should be about 3.1 psig.

Similar calculations were made for several different diameter lift pipes at different pellet flow rates and air temperatures. The requirements at the different conditions are summarized in Table VIII, Requirements For Industrial Air Lift Installations.

Since the upper limit of pellet flow was 90,000 pounds per hour, a 12-inch carrier pipe was found to be unnecessary. A 6-inch lift pipe would require higher flowing pellet densities than found practical in these data. The 10-inch carrier pipe is the most satisfactory for pellet flow rates up to 90,000 pounds per hour. If start-up temperatures were about 100°F, approximately 3400 SCFM of air would be required. At operating temperatures, the flow requirements would be decreased to about 2200 SCFM. Because of the wide difference between the air requirements at 100°F and at 1000°F, it would be preferable to use steam as a supplementary lift gas at the lower temperatures.

A gas lift for the same industrial installation discussed in the foregoing paragraphs can be operated with steam as the lift fluid and with a smaller amount of air and fuel used for superheating the steam. The design of a steam lift can be calculated from the analytical expressions previously discussed with the use of a terminal velocity curve for steam which could be calculated in a manner similar to that used to prepare Figure 3. An exact solution would be obtained only by trial and error since the air and fuel necessary to superheat the steam would alter the physical properties of the lift medium. The indicated advantage of the steam lift over the air lift is the substantial reduction in the cost of the blower which would be achieved with the steam lift. A choice between the steam and the air lift would depend on the comparison between the blower investment and the cost of steam at the plant installation and on alleviation of operating difficulties associated with water from condensed steam in the carrier pipe. A complete evaluation of the use of superheated steam as the lifting medium is beyond the scope of this thesis.

VI. OPERATION AND CALIBRATION OF INDUSTRIAL AIR LIFT

Stable operation of the experimental 3-inch air lift was obtained when slightly higher than "minimum" air and pellet exit velocities were maintained. The following tabulation indicates the significance of slightly higher pellet velocities. The data presented were taken from Tables I and II.

Air Flow Description	(CFM)	Temp. (°F)	Pellet Flow (lb/hr)	Flowing Pellet Density ρ_b (lb/ft^3)	Pressure Drop (in. H_2O)
"Maximum"	739	1010	7000	1.78	27.8
"Normal"	677	1000	7000	2.38	28.6
"Minimum"	654	1010	7000	3.87	31.5

At the "maximum" air flow rate the value of the flowing pellet density decreased to less than one-half the density calculated at the "minimum" air flow rate and a concurrent decrease in the pressure drop was experimentally noted. It was found that a pellet density of 3.87 pounds per cubic foot would not plug the carrier pipe, however, operation with this high density was unsteady and should not be practiced.

For the reasons stated, the recommended minimum height of the pellet disengaging section for the proposed lift was eight feet. The diameter should be sufficient to reduce the air velocity below the terminal velocity at the operating temperature. A two to four-foot diameter chamber would satisfy this requirement.

The pellet flow rate can be ascertained by means of a differential-pressure recorder with leads attached across the top section of the lift pipe. Since the pellet flow in the upper section of the carrier pipe should be in a steady-state condition, losses due to pellet acceleration would be negligible. By assuming, for example, six per cent of the total pressure-drop as loss to pellet frictions, the equations outlined in Chapter III of this thesis could be used to calculate the pellet flow rate at any pressure drop. These following equations:

$$\Delta P_T = \Delta P_a + \Delta P_s + \Delta P_{V_p} + \Delta P_{Pel} + \Delta P_{Acc} \quad (5)$$

$$\Delta P_{Pel} + \Delta P_{Acc} = 0.06\, \Delta P_T \text{ (assumed)}$$

$$\Delta P_s = \rho_b H \quad (7)$$

$$\Delta P_{V_p} = \rho_b (V_p^2/2g) \quad (8)$$

$$\rho_b = W_p / AV_p \quad (3)$$

can be combined as follows:

$$\Delta P_T = \Delta P_a + \rho_b H + \rho_b (V_p^2/2g) + 0.06\, \Delta P_T$$

$$0.94\, \Delta P_T - \Delta P_a = \rho_b (H + V_p^2/2g)$$

$$0.94\, \Delta P_T - \Delta P_a = (W_p/AV_p)(H + V_p^2/2g)$$

rearranging:

$$W_p = AV_p \; \frac{(0.94\, \Delta P_T - \Delta P_a)}{(H + V_p^2/2g)} \quad (16)$$

Figure 6 presents one calibration curve estimated from equation (16). This curve was calculated for the anticipated pressure drop across the top 20-feet of the 10-inch air lift pipe. The exit pellet velocity of 17.5 feet per second and the air flow at 1000°F of 2245 SCFM were determined by the design conditions previously stated. Table IX, Estimated Calibration For 10-Inch Industrial Air Lift, contains the calculated pellet flow rates at several pressure-drop readings across the top 20 feet of carrier pipe. These data are plotted on Figure 6.

For the 6-inch pressure drop:

$$\Delta P_a = 5.3 \text{ in. } H_2O$$

Substituting in Equation (16) and using the proper conversion factors:

$$W_p = \frac{(0.548)(17.5)(3600)[0.94 \times 6 - 5.3 \times (20/90)]\,(144)}{[20 + (17.5^2/64.4)]\; 27.7} = 25{,}000 \text{ lb/hr}$$

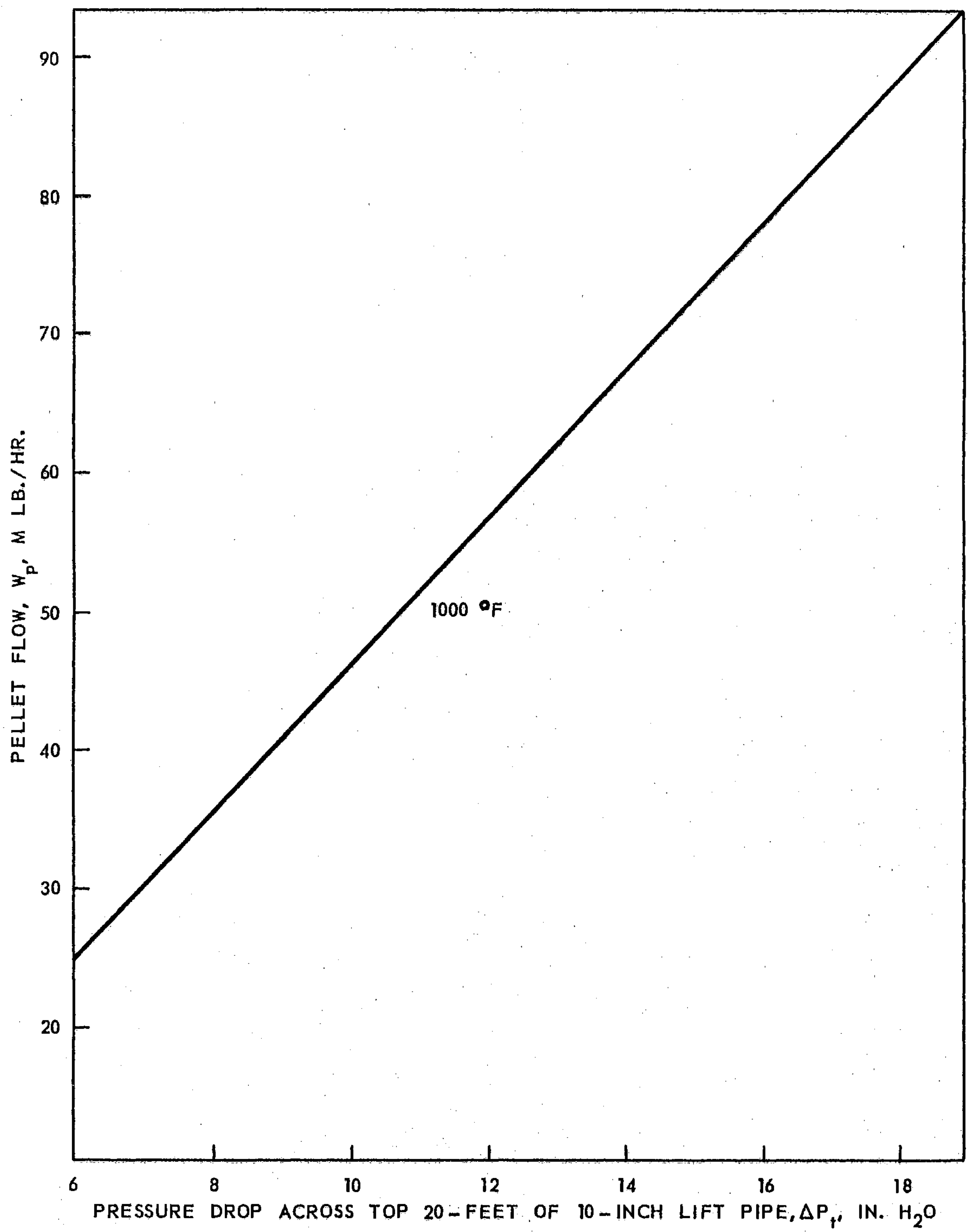

FIGURE 6 ESTIMATED CALIBRATION CURVE FOR 10-INCH INDUSTRIAL AIR LIFT

Since the air flow rate will vary in operation only with variations in temperature, a set of similar curves at 100-degree increments of temperature should complete the calibration. The sensitivity of the calibration depicted in Figure 6 is approximately 5250 pounds of pellets per inch-of-water differential pressure. This sensitivity can be increased to about 2400 pounds per inch-of-water, for example, by increasing the distance between the pressure taps from 20 feet to 40 feet.

Equation (9) and Figure 6 indicate the controlling factors in the use of a differential-pressure recorder to indicate the pellet flow rate. An actual calibration by a sample-weighing procedure should be conducted after installation of the proposed lift, and the results should be compared with the predicted behavior.

VII. SUMMARY AND CONCLUSIONS

One of the essential parts of an industrial, moving bed, heat exchanger is a mechanism to elevate ceramic pellets from the bottom of the unit and return them to the system, thereby completing the cycle. A gas lift for transporting 3/8-inch diameter pellets in a large scale unit was proposed as an improvement over a bucket type elevator formerly used. Investment, maintenance and operating costs for the gas lift should be less than for the bucket hoist.

A 3-inch air lift was operated for a series of test runs in order to obtain design data for a large scale installation. A correlation was obtained which permitted extrapolation to larger pipes and greater rates of pellet flow.

The data obtained at lift-air temperatures up to 1000°F were in satisfactory agreement with published correlations of the terminal velocities of spheres in air. Approximately four-fifths of the total pressure drop was accounted for by known "heads". The remaining 20 per cent of the observed loss was attributed to pellet acceleration and pellet friction losses.

A correlation of the experimental data indicated that 2245 SCFM of air delivered at 3.1 psig pressure and 1000°F temperature to the base of a 10-inch lift pipe would satisfactorily lift the 3/8-inch diameter pellets at rates of up to 90,000 pounds per hour. Since start-up conditions require up to 50 per cent increase in the air volume, the use of steam is suggested to supplement the air flow at lower temperatures. Additional pressure at the blower discharge would be necessary to allow for pressure drop through the burner, control valves and auxiliary piping.

A reduction in blower investment may be realized if superheated steam is used as all or part of the lifting medium at operating temperatures. A choice between the steam and the air lift will depend on the comparison between the blower investment and the cost of steam at the plant installation and on alleviation of operating difficulties associated with water from condensed steam in the lift pipe.

Stable operation can be achieved if the pellets are transported four to five feet above the upper end of the lift pipe before they are allowed to fall into the return duct which carries them to the heat exchange unit. The pellet flow rate can be ascertained by measurements of the differential pressure across the top section of the lift pipe.

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APPENDIX

DEFINITIONS OF TERMS

A	Cross-sectional area of pipe
D	Diameter of the carrier pipe
D_s	Average pellet diameter
f	Friction factor
f_D	Friction factor
g	Acceleration due to gravity
gc	Constant for dimensional conversion = 32.17 ft lb mass/lb force sec^2
H	Length of the air lift pipe
Q_a	Volume of fluid per unit time at flowing temperature
R	A new dimensionless number
Re	Reynolds number
S	Vertical distance travelled by pellets after leaving the effect of the lift air
Ta	Temperature of air feed in degree Rankine
T^oF	Temperature of the air feed in degrees Fahrenheit
Va	True air velocity at flowing temperature
V_p	Pellet velocity under steady state conditions
V_t	Terminal velocity of air necessary to freely suspend the particles
W_p	Weight of pellets being elevated per unit time
Δp	Pressure drop in inches of water
ΔP	Pressure drop in lb/ft^2
ΔPa	Pressure drop due to air friction on the wall of the carrier pipe
ΔP_{Acc}	Pressure drop lost to acceleration of the pellets
ΔP_{Pel}	Pressure drop due to pellet friction on pellets and on the wall of the lift pipe
ΔP_s	Pressure drop due to the head of pellets in the gas lift
ΔP_T	Total pressure drop in the gas lift
ΔP_{V_p}	Pressure drop due to the velocity of the pellets at the top of the lift pipe
μ_a	Viscosity of the fluid
ρ_a	Density of the fluid
ρ_b	Bulk density of the pellets in the pipe
ρ_s	Density of the individual pellets based on the weight of a unit volume of the material of composition

TABLE Ia

EXPERIMENTAL DATA ON 3-INCH AIR LIFT FOR 3/8-INCH DIAMETER PELLETS

("Maximum" Air Flow)

Data Point	S (1)	T Lift Air Temp (°F)	Q_a Air Flow (CFM at Lift Temp)	W_p Pellet Flow (lb/hr)	ΔP_T Experimental Pressure Drop (in. H_2O)
1	7	935	555	1250	12.0
2	7	900	605	2000	16.4
3	7	855	615	3000	18.2
4	7	1000	683	4200	23.4
5	7	900	694	5500	25.4
6	7	1010	739	7000	27.8
7	7	470	470	120	7.2
8	7	465	429	600	9.9
9	7	500	446	1440	13.0
10	7	460	460	2805	16.4
11	7	460	469	4200	20.8
12	7	560	481	6000	31.7
13	7	565	526	8280	37.5
14	7	133	358	3240	21.0
15	7	142	357	4500	25.5
16	7	150	357	6060	34.0
17	7	175	374	7260	38.4

(1) The distance, S, is the estimated vertical distance the pellets travelled into the disengaging section before their upward motion was overcome by the force of gravity.

TABLE Ib

EXPERIMENTAL DATA ON 3-INCH AIR LIFT FOR 3/8-INCH DIAMETER PELLETS

("Normal" Air Flow)

Data Point	S (1)	T Lift Air Temp (°F)	Q_a Air Flow (CFM at Lift Temp)	W_p Pellet Flow (lb/hr)	ΔP_T Experimental Pressure Drop (in. H_2O)
18	4	940	533	1250	12.1
19	4	940	584	2000	16.7
20	4	870	602	3000	17.7
21	4	990	644	4200	24.7
22	4	980	676	5500	26.3
23	4	1000	677	7000	28.6
24	4	470	417	120	6.9
25	4	480	418	600	10.9
26	4	490	448	2800	20.9
27	4	495	432	1440	13.5
28	4	475	450	4200	21.0
29	4	560	481	6000	31.7
30	4	560	522	8280	40.2

(1) The distance, S, is the estimated vertical distance the pellets travelled into the disengaging section before their upward motion was overcome by the force of gravity.

TABLE Ic

EXPERIMENTAL DATA ON 3-INCH AIR LIFT FOR 3/8-INCH DIAMETER PELLETS

("Minimum" Air Flow)

Data Point	S (1)	T Lift Air Temp (°F)	Q_a Air Flow (CFM at Lift Temp)	W_p Pellet Flow (lb/hr)	ΔP_T Experimental Pressure Drop (in. H_2O)
31	2	950	515	1250	13.2
32	2	895	566	3000	19.6
33	2	965	633	5500	28.0
34	2	1010	654	7000	31.5
35	2	460	388	120	8.9
36	2	480	407	600	11.0
37	2	500	427	1440	15.6
38	2	490	439	2805	21.0
39	2	460	430	4200	23.7
40	2	555	468	6000	34.3
41	2	560	481	8280	42.6
42	2	133	338	3240	23.0
43	2	135	320	4500	38.0
44	2	144	332	6060	41.3
45	2	160	360	7260	42.6

(1) The distance, S, is the estimated vertical distance the pellets travelled into the disengaging section before their upward motion was overcome by the force of gravity.

TABLE IIa

ESTIMATION OF PRESSURE HEADS FOR 3-INCH AIR LIFT

(855°F to 1010°F Lift Air Temperature)
(Experimental Data on Table I)

Data Point	ρ_b (lb/ft^3) (Eq. 3)	V_a (ft/sec) (Eq. 4)	ΔP_a (in.H_2O) (Fig. 2)	ΔP_s (in.H_2O) (Eq. 7)	ΔP_{V_p} (in.H_2O) (Eq. 8)	$\Delta P_a + \Delta P_s + \Delta P_{V_p}$ (in.H_2O)	ΔP_T (in.H_2O) (exp)	$\Delta P_{Pel} + \Delta P_{Acc}$ (in.H_2O) (by diff)
	"Maximum" Air Flow S = 7 feet V_p = 21.3 ft/sec (Eq. 2)							
1	0.318	180.5	5.70	2.50	0.43	8.63	12.0	3.37
2	0.509	196.5	7.05	4.06	0.69	11.80	16.4	4.60
3	0.763	200.0	7.13	6.10	1.04	14.27	18.2	3.93
4	1.068	222.0	9.30	8.53	1.45	19.28	23.4	4.12
5	1.400	225.0	9.36	11.20	1.90	22.46	25.4	2.94
6	1.780	240.0	10.52	14.20	2.42	27.14	27.8	0.66
	"Normal" Air Flow S = 4 feet V_p = 15.9 ft/sec (Eq. 2)							
18	0.425	173.0	5.20	3.39	0.32	8.91	12.1	3.19
19	0.680	189.5	6.77	5.43	0.51	12.71	16.7	3.99
20	1.020	195.5	6.97	8.14	0.77	15.88	17.7	1.82
21	1.430	209.0	7.42	11.41	1.08	19.91	24.7	4.79
22	1.870	220.0	8.90	14.90	1.42	25.22	26.3	1.08
23	2.380	220.0	8.91	19.00	1.80	29.71	28.6	(1)
	"Minimum" Air Flow S = 2 feet V_p = 9.8 ft/sec (Eq. 2)							
31	0.691	167.3	4.52	5.51	0.19	10.22	13.2	2.98
32	1.660	184.0	6.05	13.24	0.47	19.76	19.6	(1)
33	3.040	206.0	7.80	24.20	0.86	32.86	28.0	(1)
34	3.870	212.1	8.32	30.90	1.09	40.31	31.5	(1)

(1) In these instances the calculated pressure drop exceeded the experimental pressure drop. Such a result indicates that the pellets actually travelled farther than the distance, S, which was estimated.

TABLE IIb

ESTIMATION OF PRESSURE HEADS FOR 3-INCH AIR LIFT

(460°F to 500°F Lift Air Temperature)
(Experimental Data on Table I)

Data Point	ρ_b (lb/ft^3) (Eq. 3)	V_a (ft/sec) (Eq. 4)	ΔP_a (in.H_2O) (Fig. 2)	ΔP_s (in.H_2O) (Eq. 7)	ΔP_V (in.H_2O) (Eq. 8)	$\Delta P_a + \Delta P_s + \Delta P_{V_p}$ (in.H_2O)	ΔP_T (in. H_2O) (exp)	$\Delta P_{Pel} + \Delta P_{Acc}$ (in.H_2O) (by diff)
	"Maximum" Air Flow S = 7 feet V_p = 21.3 ft/sec (Eq. 2)							
7	0.031	148.0	6.94	0.24	0.04	7.22	7.2	(1)
8	0.153	135.0	5.67	1.22	0.21	7.10	9.9	2.80
9	0.366	140.3	6.20	2.92	0.50	9.62	13.0	3.38
0	0.714	144.6	6.60	5.70	0.95	13.25	16.4	3.15
1	1.070	147.5	6.90	8.53	1.45	16.88	20.8	3.92
2	1.530	151.3	7.10	12.20	2.07	21.37	31.7	10.33
3	2.110	165.5	8.07	16.82	2.86	27.75	37.5	9.75
	"Normal" Air Flow S = 4 feet V_p = 15.9 ft/sec (Eq. 2)							
4	0.041	131.2	5.33	0.33	0.03	5.69	6.9	1.21
5	0.204	131.5	5.35	1.63	0.15	7.13	10.9	3.77
6	0.952	141.0	6.28	7.61	0.72	15.61	20.9	5.29
7	0.490	136.0	5.82	3.91	0.37	10.10	13.5	3.40
8	1.430	141.6	6.32	11.41	1.08	18.81	21.0	2.19
9	2.820	164.1	7.18	22.50	2.13	31.81	40.2	8.39
0	2.040	151.3	8.32	16.28	1.54	26.14	31.7	5.56
	"Minimum" Air Flow S = 2 feet V_p = 9.8 ft/sec (Eq. 2)							
5	0.066	122.0	4.57	0.53	0.02	5.12	8.9	3.78
6	0.331	128.0	5.10	2.64	0.09	7.83	11.0	3.17
7	0.795	134.4	5.58	6.35	0.23	12.16	15.6	3.44
8	1.550	138.0	6.04	12.38	0.44	18.86	21.0	2.14
9	2.320	135.2	5.75	18.50	0.66	24.91	23.7	(1)
0	3.310	147.1	6.84	26.40	0.95	34.19	34.3	0.11
1	4.570	151.2	7.17	36.50	1.31	44.98	42.6	(1)

(1) In these instances the calculated pressure drop exceeded the experimental pressure drop. Such a result indicates that the pellets actually travelled farther than the distance, S, which was estimated.

TABLE IIc

ESTIMATION OF PRESSURE HEADS FOR 3-INCH AIR LIFT

($133^{o}F$ to $175^{o}F$ Lift Air Temperature)
(Experimental Data on Table I)

ata oint	ρ_b (lb/ft^3) (Eq. 3)	V_a (ft/sec) (Eq. 4)	ΔP_a (in. H_2O) (Fig. 2)	ΔP_s (in. H_2O) (Eq. 7)	ΔP_V (in. H_2O) (Eq. 8)	$\Delta P_a + \Delta P_s + \Delta P_V$ (in. H_2O)	ΔP_T (in. H_2O) (exp)	$\Delta P_{Pel} + \Delta P_{Acc}$ (in. H_2O) (by diff)
	"Maximum" Air Flow $S = 7$ feet $V_p = 21.3$ ft/sec (Eq. 2)							
14	0.824	112.5	6.37	6.59	1.12	14.08	21.0	6.92
15	1.140	112.3	6.35	9.09	1.54	16.98	25.5	8.52
16	1.540	112.3	6.35	12.30	2.09	20.74	34.0	13.26
17	1.850	117.6	6.85	14.76	2.51	24.12	38.4	14.28
	"Minimum" Air Flow $S = 2$ feet $V_p = 9.8$ ft/sec (Eq. 2)							
42	1.790	106.2	5.77	14.30	0.51	20.58	23.0	2.42
43	2.480	100.6	5.20	19.80	0.71	25.71	38.0	12.29
44	3.350	104.2	5.55	26.70	0.96	33.21	41.3	8.09
45	4.010	113.1	6.05	32.10	1.15	39.30	42.6	3.30

TABLE III

CALCULATION OF ΔP_a WITH NO PELLET FLOW

Data Point	T °F	T °R	Q_a CFM at T	Δp in.H_2O	V_a ft/sec	ρ_a lb/ft^3 at T	f	ΔP_a
46	950	1410	550	6.3	169.4	0.0282	0.0142	5.7
47	950	1410	572	6.8	176.2	0.0282	0.0142	6.2
48	940	1400	578	7.1	178.0	0.0284	0.0144	6.5
49	935	1395	576	7.2	177.5	0.0285	0.0146	6.6
50	895	1355	590	7.5	181.5	0.0293	0.0145	6.8
51	950	1410	621	8.0	191.5	0.0282	0.0145	7.3
52	870	1330	617	8.5	190.2	0.0298	0.0148	7.8
53	1000	1460	672	8.9	206.5	0.0272	0.0143	8.1
54	940	1400	648	9.1	199.5	0.0284	0.0151	8.3
55	855	1315	635	9.2	195.5	0.0302	0.0149	8.3
56	1010	1470	688	9.6	212.0	0.0270	0.0148	8.8
57	980	1440	679	9.7	209.0	0.0276	0.0151	8.3
58	980	1440	665	9.8	204.5	0.0276	0.0159	8.9
59	1000	1460	728	10.8	224.5	0.0272	0.0148	9.8
60	1000	1460	747	10.9	230.0	0.0272	0.0142	10.0
61	1000	1460	764	10.9	235.0	0.0272	0.0136	9.8
62	900	1360	741	11.0	228.2	0.0292	0.0136	10.1
63	1000	1460	717	11.0	221.0	0.0272	0.0155	10.0
64	460	920	395	5.9	121.7	0.0432	0.0173	5.4
65	480	940	418	5.9	128.8	0.0422	0.0158	5.4
66	470	930	417	6.1	127.5	0.0427	0.0162	5.4
67	500	960	458	6.4	141.2	0.0413	0.0146	5.8
68	465	925	433	6.5	133.3	0.0429	0.0165	6.1
69	495	955	443	6.5	136.2	0.0416	0.0157	5.9
70	490	950	458	6.9	141.2	0.0417	0.0155	6.3
71	500	960	457	7.0	141.0	0.0413	0.0160	6.3
72	460	920	448	7.2	137.0	0.0432	0.0164	6.5
73	470	930	473	7.2	145.5	0.0427	0.0149	6.6
74	460	920	472	7.6	145.2	0.0432	0.0156	6.9
75	475	935	476	7.6	146.5	0.0425	0.0156	7.0
76	460	920	488	8.2	150.2	0.0432	0.0157	7.5
77	133	593	247	3.7	76.0	0.0670	0.0179	3.4
78	133	593	275	4.3	84.7	0.0670	0.0168	3.9
79	133	593	298	5.2	91.7	0.0670	0.0173	4.7
80	133	593	322	5.7	99.2	0.0670	0.0162	5.2
81	133	593	342	6.4	105.2	0.0670	0.0161	5.8
82	133	593	379	7.8	116.8	0.0670	0.0160	7.0
83	133	593	417	9.1	128.5	0.0670	0.0154	8.3

TABLE IV

TERMINAL VELOCITY OF 3/8-INCH PELLETS IN AIR
(Calculated From Experimental Data)

Data Point	Lift Air Temp (°F)	V_a (ft/sec) (Table II)	V_p (ft/sec) (Eq. 2)	V_t (ft/sec) (Eq. 1)
(855°F to 1010°F Lift Air Temperature)				
1	935	180.5	21.3	159.2
2	900	196.5	21.3	175.2
3	855	200.0	21.3	178.7
4	1000	222.0	21.3	200.7
5	900	225.0	21.3	203.7
6	1010	240.0	21.3	218.7
18	940	173.0	15.9	157.1
19	940	189.5	15.9	173.6
20	870	195.5	15.9	179.6
21	990	209.0	15.9	193.1
22	980	220.0	15.9	204.1
23	1000	220.0	15.9	204.1
31	950	167.3	9.8	157.5
32	895	184.0	9.8	174.2
33	965	206.0	9.8	196.2
34	1010	212.1	9.8	202.3
(460°F to 565°F Lift Air Temperature)				
7	470	148.0	21.3	126.7
8	465	135.0	21.3	113.7
9	500	140.3	21.3	119.0
10	460	144.6	21.3	123.3
11	460	147.5	21.3	126.2
12	560	151.3	21.3	130.0
13	565	165.5	21.3	144.2
24	470	131.2	15.9	115.3
25	480	131.5	15.9	115.6
26	490	141.0	15.9	125.1
27	495	136.0	15.9	120.1
28	475	141.6	15.9	125.7
29	560	164.1	15.9	148.2
30	560	151.3	15.9	135.4
35	460	122.0	9.8	112.2
36	480	128.0	9.8	118.2
37	500	134.4	9.8	124.6
38	490	138.0	9.8	128.2
39	460	135.2	9.8	125.4
40	555	147.1	9.8	137.3
41	560	151.2	9.8	141.4
(133°F to 175°F Lift Air Temperature)				
14	133	112.5	21.3	91.2
15	142	112.3	21.3	91.0
16	150	112.3	21.3	91.0
17	175	117.6	21.3	96.3
42	133	106.2	9.8	96.4
43	135	100.6	9.8	90.8
44	144	104.2	9.8	94.4
45	160	113.1	9.8	103.3

TABLE V

CALCULATION OF DIMENSIONLESS

NUMBER R FROM FIGURE 4

Re (Fig. 4)	f_D (Fig. 4)	R ($Re^2 f_D$)
1×10^3	0.44	4.4×10^5
2×10^3	0.40	1.6×10^6
5×10^3	0.40	1.0×10^7
1×10^4	0.42	4.2×10^7
2×10^4	0.45	1.8×10^8
5×10^4	0.48	1.2×10^9
1×10^5	0.50	5.0×10^9

TABLE VI

CALCULATION OF DIMENSIONLESS

NUMBER R FROM EQUATION (15)

T (°F)	ρ_a (lb/ft³)	μ_a (centipoises)	R
100	7.08×10^{-2}	1.87×10^{-2}	8.26×10^{7}
200	6.02×10^{-2}	2.10×10^{-2}	5.82×10^{7}
400	4.61×10^{-2}	2.47×10^{-2}	4.12×10^{7}
600	3.74×10^{-2}	2.88×10^{-2}	2.83×10^{7}
800	3.15×10^{-2}	3.25×10^{-2}	2.11×10^{7}
1000	2.72×10^{-2}	3.60×10^{-2}	1.65×10^{7}

TABLE VII

TERMINAL VELOCITY OF 3/8-INCH PELLETS IN AIR

(Calculated From Data of Waddel)

T (°F)	Re (Fig. 5)	V_t (ft/sec)
100	1.35×10^4	75.2
200	1.15×10^4	86.5
400	9.60×10^3	108.7
600	7.90×10^3	130.8
800	6.90×10^3	152.5
1000	6.10×10^3	174.0

TABLE VIII

REQUIREMENTS FOR INDUSTRIAL AIR LIFT INSTALLATION

Pipe Size (in)	Pellet Rate (lb/hr)	Lift Temp (°F)	Flowing Pellet Density (lb/ft^3)	Calc Pressure Drop (psig)	Design Specifications Air Flow (SCFM)	Delivery Pressure (psig)
10	90,000	1000	2.60	2.39	2245	3.11
10	60,000	1000	1.73	1.66	2245	2.16
10	30,000	1000	0.87	0.96	2245	1.25
10	60,000	500	1.73	1.63	2740	2.12
10	30,000	100	0.87	0.91	3410	1.19
8	90,000	1000	4.11	3.70	1390	4.81
8	60,000	1000	2.74	2.58	1390	3.36
8	30,000	1000	1.37	1.45	1390	1.88
8	60,000	500	2.74	2.55	1740	3.32

TABLE IX

ESTIMATED CALIBRATION FOR 10-INCH INDUSTRIAL AIR LIFTS

$T = 1000^{\circ}F$
$D = 0.835$ ft
$A = 0.548$ ft^2
$H = 20.0$ ft
$\Delta P_a = 5.3$ in. H_2O
$V_p = 11.35$ ft/sec (Eq. 2)

Pressure Drop Over Top 20 Feet $\Delta P_T'$ (in. H_2O)	W_p (lb/hr)
6	25,000
8	35,500
10	46,100
12	56,500
14	67,000
16	77,600
18	88,300
20	98,800

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