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EFFICIENCY OF ACCELERATING TUBES OF JET GRINDING MILLS

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An analysis is made of the energy efficiency of accelerating tubes in counterflow jet grinding mills. The dependence of the efficiency of these tubes on the parameters of the two-phase flow is established.

The grinding of solid materials is one of the most energy-intensive processes in industry. This fact makes it particularly important to select the proper method of grinding for a given case. Thus, analysis of the energy efficiency of grinding mills is of definite interest with regard to improving grinding technology and mill design. One promising trend in grinding is the use of jet mills [1, 2], in which the material is ground by high-speed impact. Gas or steam is usually used as the working substance, the energy of the gas or steam accelerating the starting material to velocities at which it breaks up upon impact against an obstacle (the wall of the mill or another portion of the material being ground). Here, the energy of the working substance is spent on the completion of useful work in accelerating particles of the material being ground, as well as on irreversible losses connected with the evolution of heat in interphase friction. Both types of energy expenditures depend on the phase slip

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velocity and increase with an increase in the latter. The gas, doing work on accelerating particles in the straight tube, itself accelerates along the tube. However, its velocity cannot exceed the sonic velocity at the outlet. The critical tube length for which the gas reaches the speed of sound depends on many factors, particularly on the pressure at the inlet, the rate of flow of the solid phase, and its terminal velocity. The same factors determine the relationship between the useful work done by the gas and the irreversible losses. Below we study the energy efficiency of straight accelerating tubes. The complete system of equations is written for both phases and is solved numerically. An approximate analytical solution to the problem is also obtained. The good agreement the analytical solution and the exact numerical solution makes it possible to obtain fairly simple formulas which can be used to design accelerating tubes that are energetically optimum.

We will examine a unidimensional steady two-phase flow consisting of a gaseous carrier medium and monodisperse solid particles of volumetric concentration τ. It is assumed that τ is small and that the particles interact only with the carrier medium. The equations of motion of the phases [3] in this approximation will be written in the form

$$\varepsilon\rho v v' = -p' - F,$$
$$\tau\rho_1 w w' = F,\ \varepsilon = 1 - \tau. \tag{1}$$

Here

$$F = \frac{3}{4}\frac{\zeta\tau}{d}\rho(v-w)^2 - \tau p' \tag{2}$$

is the force acting on the particles from the flow side. The first term in (2) is associated with the resistance force of the particles contained in a unit volume during separated flow of the gas about them. The second term determines the force due to the presence of a pressure gradient in the continuous medium. The prime denotes a derivative with respect to the longitudinal coordinate. Assuming that the concentration of the solid phase is small, in the first term we have omitted factors connected with the change in the velocity of flow about a particle due to the presence of other particles (see [3, 4]). In the equation of motion of the gas phase we omitted the resistance force of the accelerating tube F_1, which is much less than the resistance force of the solid phase:

$$F_1/F \sim (\lambda d)/(\tau\zeta D) \ll 1. \tag{2a}$$

For the case examined, Eq. (2a) is valid despite the condition $\tau \ll 1$, since the size of the particles being accelerated is much less than the diameter of the accelerating tube, and the resistance coefficient of the circular tube is considerably less than the resistance coefficient of a sphere. Equations of motion (1) must be augmented by the energy and continuity equations for both phases:

$$\varepsilon\rho v = G,\ \tau\rho_1 w = G_1. \tag{3}$$

As already noted above, the energy of the gas is expended on doing mechanical work in accelerating particles and on work in overcoming interphase friction. The latter is connected with an increase in the entropy of the gas, i.e., with irreversible losses. The rate of heat release per unit mass of gas as a result of interphase friction is equal to

$$\frac{dQ}{dt} = (fu)/(\varepsilon\rho),$$

where $f = (3/4)(\zeta\tau/d)\rho u^2$, $u = (v - w)$ is the phase slip velocity. Using the first law of thermodynamics in the form

$$dQ = di - dp/\rho,$$

where $i = c_pT$ is the enthalpy of the gas, and also using Eqs. (1)-(3) and the relationship between the longitudinal coordinate and time for a chosen element of the gas $dx = vdt$, we obtain the sought energy equation in the form

$$di_0 = -G_*[d(w^2/2) + dp/\rho_1],\ G_* = G_1/G. \tag{4}$$

Here, $i_0 = c_pT + v^2/2$ is the total enthalpy of a unit mass of gas. The second term in the right side of Eq. (4) represents the elementary work $dA = -(\tau p')/(\varepsilon\rho)wdt$ due to the presence of the pressure gradient and completed by a unit mass of gas over particles in the gas at a given moment of time.

Generally speaking, along with heat liberation in the gas due to interphase friction, heat exchange occurs between the phases because the gas cools as it accelerates and does mechanical work. However, it is not hard to see that the relaxation time for relatively short accelerating tubes and fairly high particle velocities is much greater than the particle transit time (the diameter of the particles being accelerated is not very small), i.e.,

$$c\rho_1 d/\alpha \gg l/w.$$

Here, c is the specific heat of the solid material, $\alpha = \alpha(\mathrm{Re}, \mathrm{Pr})$ is the heat-transfer coefficient of a particle, which depends on the dimensionless Reynolds and Prandtl criteria [5]. In many cases of practical interest, heat exchange between the phases can be ignored.

Equations (1)-(4), together with the equation of state of the gas, give us a closed system to determine all of the necessary parameters of both phases at any point of the accelerating tube. If we solve this system relative to the derivatives of the phase velocities, and if we introduce the Mach number $M = v/a$, where $a = (kRT)^{1/2}$, then the system can be written as follows:

$$\begin{gathered}
w' = \frac{f_*}{\varepsilon G}\,\frac{(1-M^2) + k(\tau - \alpha_*\varepsilon)M^2}{(1-M^2) + (\tau/\varepsilon)^2 (v/w)/G_*},\\
v' = \frac{f_*}{\varepsilon}\,\frac{[k\varepsilon(1+\alpha_*) - (k-1)w/v]M^2 - \{(\tau/\varepsilon)(v/w)/G_*\}}{(1-M^2) + (\tau/\varepsilon)^2 (v/w)/G_*},\\
(M^2)' = [2v'/v - \alpha_*(\varepsilon v)'/(\varepsilon v)]M^2 + (k-1)[v'/v + G_*(w/v)^2(w'/w)]M^4,\\
\tau' = -\tau(w'/w),\\
\alpha_* = G_*(\rho/\rho_1)(k-1)/k,\\
f_* = \frac{3}{4}\,\frac{\tau\zeta}{\varepsilon v d}(v-w)^2.
\end{gathered} \tag{5}$$

The presence of the density of the gas ρ in the expression for α_* gives a closed system of equations represented in the form (5). However, since $\rho/\rho_1 \ll 1$, then for relatively low values of G_* we find that $\alpha_* \ll 1$. This allows us to close (5) and ignore terms on the order of α_*. Assuming $\tau \ll 1$, we can omit terms containing τ in the first two equations of system (5) and set ε = 1. This corresponds to a linear approximation with respect to τ. Such an assumption essentially changes little. For example, the term in the braces in the second equation of (5) represents the change in gas velocity due to its expansion in the volume occupied previously by solid particles due to a reduction in particle concentration downflow. At low particle concentrations, this factor can be ignored.

System (5), augmented by the equations of continuity and state for the gas phase, can be solved numerically without the above-noted limitations. However, such an approach is justified when it is necessary to obtain an exact solution to a specific engineering problem. For general analysis, when the problem contains many parameters, it is better to attempt to find the most acceptable approximate analytic solution – even a solution that is simplified somewhat but that still reflects the basic content of the problem.

The ratio of the phase velocities for the chosen acceleration scheme $z = w/v < 1$, while the Mach number cannot be greater than unity because the speed of sound cannot be exceeded for the gas flow in the straight tube. We will make use of both of these facts: the first, for an expansion in z; the second, to replace the argument. The equation for z, with allowance for the above-noted approximations, has the form

$$\frac{dz}{dy} = \frac{(1-y) - G_* yz[k-(k-1)z]}{G_* y^2\{(k-1)(1-y)z + [2+(k-1)y][k-(k-1)z]\}}, \tag{6}$$

here we have introduced the notation $y = M^2$.

We expand the right side of Eq. (6) into a series in $z < 1$:

$$\begin{gathered}
\frac{dz}{dy} = \frac{(1-y)}{kG_* y^2[2+(k-1)y]} - \frac{z}{y[2+(k-1)y]} \times\\
\times\left\{\left[1 - \frac{(k-1)(1-ky)}{k[2+(k-1)y]}z\right] - \frac{(k-1)(1-y)(1+ky)}{k^2 G_* y[2+(k-1)y]}\left[1 + \frac{(k-1)(1+ky)z}{k[2+(k-1)y]}\right]\right\} + O(z^3).
\end{gathered} \tag{7}$$

It is easy to find an approximate solution to Eq. (6) if we make several quite reasonable assumptions. First, the expression in the braces is on the order of unity if the following condition is satisfied:

$$y_0 \gg \frac{(k-1)}{2k^2 G_*}. \tag{8}$$

Henceforth the index (0) will denote a value of a quantity at the tube inlet, i.e., $y_0 = y|_{x=0}$. Since we are interested in the maximum value of G_* for a given ratio of total pressure at the tube inlet to the static pressure at the outlet P, it will be shown below that there is a range of P for which Eq. (8) is satisfied. Second, k = 1.4 for diatomic gases and k = 1.3 and $(k - 1)y \ll 2$ for steam, since $y \leq 1$. Thus, the following expression is an approximate solution of Eq. (6):

$$z = Cy^{-1/2} - \frac{1}{kG_*}(1 + 1/y),$$
$$C = y_0^{1/2}[z_0 + (1/kG_*)(1 + 1/y_0)]. \tag{9}$$

The equation for gas velocity has the form

$$\frac{dv}{vdy} = \frac{k-(k-1)z}{z\{(k-1)(1-y)z + [2+(k-1)y][k-(k-1)z]\}}. \tag{10}$$

Using the procedure described above and the form of solution (9), we can write an approximate solution of Eq. (10) as follows:

$$v/v_0 = \left[\frac{(k-1)/2 + 1/y_0}{(k-1)/2 + 1/y}\right]^{1/2} \exp[\Phi(y, y_0)], \tag{11}$$
$$\Phi(y, y_0) = \left\{\frac{C(k-1)(1+y)}{2ky^{1/2}} - \frac{(k-1)(1+y^2)}{4k^2G_* y}\right\}\Bigg|_{y_0}^{y}.$$

The velocity of the particles is found from the relation w = vz.

We will introduce characteristic scales of the physical quantities. For the scales of temperature and density we take their values at the inlet. For the scale of velocity we take the speed of sound in the gas phase at the tube inlet, while for the characteristic dimension we take the length of the accelerating tube l. We can use the equations of continuity and state for the gas phase to find the pressure

$$p/p_0 = (\varepsilon_0 v y_0^{1/2})/(\varepsilon y). \tag{12}$$

Here we used the expression for temperature. By definition, $T = v^2/y$, $p_0 = RT_0\rho$, $v_0 = y_0^{1/2}$. The ratio of the total pressure at the inlet to the static pressure at the outlet can be written by using Eq. (12) and determining the total pressure of an adiabatically slowed gas:

$$P = \frac{\varepsilon}{\varepsilon_0 y_0^{1/2} v_1}\left(1 + \frac{k-1}{2}y_0\right)^{k/(k-1)}. \tag{13}$$

Here the index (1) denotes a value of a quantity at y = 1. After all of the necessary relations have been written, the parameter ε can be omitted in accordance with the above-made approximations. Also, the expression in brackets in (13) differs very little from unity. Thus it, too, can henceforth be ignored. If we use the relation $w_1 = v_1 z_1$, we can write Eq. (13) in the form

$$kG_*(Pw_1y_0^{1/2} - w_0) = \frac{(1-y_0^{1/2})^2}{y_0^{1/2}}. \tag{14}$$

The exponent of the exponential function in (11) at y = 1 is

$$|\Phi(1, y_0)| \simeq \left|\frac{(k-1)}{2k}\left[(z_1 - z_0) + y_0^{1/2}\left(z_0 + \frac{1+y_0}{kG_* y_0}\right) - \frac{1}{2kG_* y_0}\right]\right| \ll 1,$$

since $1/(kG_* y_0) \ll 2k/(k-1)$, in accordance with condition (8). With allowance for this and Eq. (14), we write Eq. (13) in the form

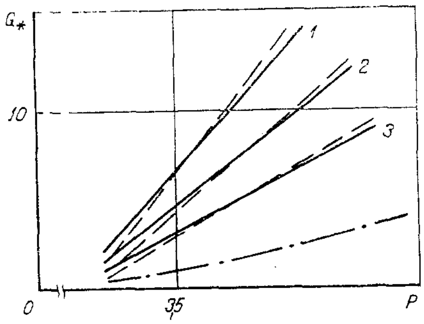

Fig. 1

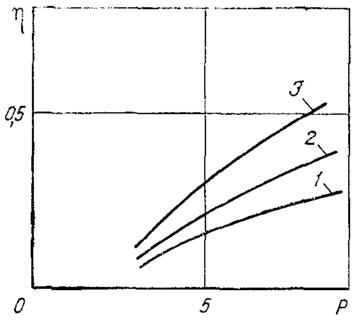

Fig. 2

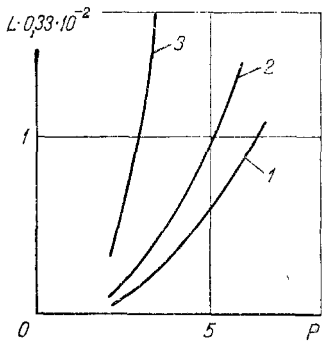

Fig. 3

Fig. 1. Dependence of the ratio of the mass flow rates of the solid phase to the gaseous phase on the ratio of the total pressure at the tube inlet to the static pressure at the outlet for air: 1-3) w_1/w_0 = 5, 7, 10 at w_0 = 1/34.

Fig. 2. Efficiency of an accelerating tube in relation to the ratio of the total pressure at the tube inlet to the static pressure at the outlet. See Fig. 1 for significance of curves 1-3.

Fig. 3. Dependence of the dimensionless length of the accelerating tube L = $(3\zeta l)/(4d)$ on the ratio of the total pressure at the tube inlet to the static pressure at the outlet. See Fig. 1 for the significance of curves 1-3.

$$P=\left(\frac{k+1}{2y_0}\right)^{1/2}\left\{1+\frac{k-1}{4}\,G_*\,(P^2w_1^2y_0-w_0^2)\right\}. \tag{15}$$

Equations (14) and (15) give the dependence of G_* on P in parametric form. In accordance with the above assumptions, they must be augmented by inequality (8). Here, the condition $w_1 < 1$ is always satisfied because the velocity of the particles is referred to the speed of sound a_0. Analysis of the numerical examples shows that the second term in braces in (15) can be ignored with good accuracy in a broad range of parameters. Inserting $P = [(k+1)/(2y_0)]^{1/2}$ into (14) and (8), we finally have

$$G_*=\left(\frac{2}{k+1}\right)^{1/2}\frac{P-2\left(\frac{k+1}{2}\right)^{1/2}}{k\left[\left(\frac{k+1}{2}\right)^{1/2}w_1-w_0\right]}, \tag{16}$$

if

$$G_*>\frac{1}{k^2}\,\frac{(k-1)}{(k+1)}\,P^2.$$

To check approximate solution (16), we performed an exact numerical integration of the initial system of equations (5) and examined several examples. The solid lines in Fig. 1 show data from the numerical integration; the dashed lines correspond to approximate solution (16), and the dot-dash line represents the region of existence of the approximate solution [the inequality in (16)].

As the efficiency of the accelerating tube we will examine the ratio of the work done by the gas in increasing the kinetic energy of the particles to the work done by an ideal compressor in compressing a gas from a pressure equal to the static pressure at the tube outlet to a pressure equal to the total pressure at the inlet [6, 7]. Using the characteristic scales introduced earlier, we have the following expression for the efficiency:

$$\eta = \frac{(k-1)}{2} G_* \frac{(w_1^2 - w_0^2)}{1 - P^{(1-k)/k}}. \tag{17}$$

In combination with (16), Eq. (17) answers the question posed at the outset regarding the energy efficiency of the accelerating tube. The length of the tube does not enter into the final expression for the efficiency because, in the formulation given, it is a sought quantity under specified external conditions and corresponds to the critical length for these conditions. However, when necessary, any quantity – the relative flow rate of the solid phase, for example – can be chosen as the sought quantity for a fixed tube length.

Figure 2 shows the dependence of the efficiency of the accelerating tube on P for the case shown earlier in Fig. 1. It is apparent that η increases with an increase in both the total pressure at the tube inlet and the terminal velocity of the particles. It follows from the latter that, from an energy point of view, in the case of a fixed ratio of total pressure at the tube inlet to the static pressure at the tube outlet P, it is more expedient to accelerate the particles to greater velocities. However, there are fewer particles in this instance. As was noted in [8, 9], an increase in the number of particles is accompanied by an increase in the efficiency of the grinding operation. However, an increase in the terminal velocity of the particles is also accompanied by an increase in the length of the accelerating tube at which this velocity is reached. The corresponding results are shown in Fig. 3. The curves correspond to the numerical solution of system (5). Here we examined a straight tube of constant cross section. If the formulation of the problem is expanded to tubes of variable cross section, then with specified parameters of the two-phase flow the required tube length may be chosen on the basis of its shape. Here, the conclusions made above regarding the efficiency of the tube remain valid.

In conclusion, we would like to note that the approximate solution represented by Eqs. (16) and (17) makes it possible to limit oneself to a relatively small number of the most important parameters in analyzing the efficiency of accelerating tubes. This fact considerably simplifies the problem of optimizing accelerating tubes.

NOTATION

D, l, λ, diameter, length, and resistance coefficient of the accelerating tube; d, ρ_1, and ζ, diameter, density, and hydraulic resistance coefficient of a particle; ρ, v, density and velocity of gas; τ, w, volume concentration of solid particles and their velocity; ε, porosity; G, G_1, mass flow rates of the gaseous and solid phases through a unit area of the accelerating tube; M, Mach number; a, speed of sound in the gas; k, R, adiabatic exponent and gas constant of a unit mass of the gas; P, ratio of the total pressure at the tube inlet to the static pressure at the outlet; η, efficiency of the accelerating tube; L, dimensionless length of the tube; p, static pressure.

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DETERMINATION OF THE HEAT-TRANSFER CHARACTERISTICS IN A CHANNEL OF ANNULAR CROSS SECTION WITH SPIRAL FINS

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The characteristics of heat transfer in the developed turbulent flow of a viscous incompressible liquid in a slot channel of annular cross section with spiral fins are analyzed. Expressions are obtained for calculating the Nusselt numbers at the convex and concave walls of the channel.

In the literature there are experimental data on the asymmetry of the averaged velocity profiles and the distributions of pulsation components in curved channels [1-4]. One can presume the existence of the corresponding asymmetry of the conditions of heat transfer between the heat-transfer agent and the walls in a curved channel, which is also confirmed experimentally for plane curved channels [1].

Let us estimate the possible difference between the values of the heat-transfer coefficient $\alpha_{1,2} = q_{1,2}/(T_{1,2} - \bar{T})$ at the convex and concave walls of a channel of annular cross section with spiral fins as a function of the geometrical characteristics of the channel, the physical characteristics of the heat-transfer agent, and the hydrodynamic parameters of the flow. We carry out the analysis using the methods and assumptions adopted in the investigation of hydraulics and heat transfer in smooth annular channels without fins and plane curved channels [1-6]. We consider the turbulent flow of an incompressible viscous heat-transfer agent in an annular channel with spiral fins under steady-state conditions of moderate heat fluxes and velocities, outside the region of the disturbing action of the fins. We assume that the thermophysical properties of the liquid are constant and the heat flux through the walls of the annular channel is constant along the channel length and angularly.

We assume that secondary flows are absent, i.e.,

$$V_r = 0,\ V_z = V_\varphi \frac{S}{2\pi r},\ \ \tau_z = \tau_\varphi \frac{S}{2\pi r},$$

$$\frac{\partial P}{\partial z} = \frac{\partial P}{r\partial\varphi}\frac{S}{2\pi r},\ \ \frac{1}{r}\frac{\partial V}{\partial\varphi} = -\frac{S}{2\pi r}\frac{\partial V}{\partial z},\ \ \frac{\partial T}{\partial z} = \frac{\partial T}{r\partial\varphi}\frac{S}{2\pi r}. \tag{1}$$

At the inner and outer walls,

$$V_{1,2} = 0,\ q_{1,2} = \alpha_{1,2}(T_{1,2} - \bar{T}) = -\lambda \left.\frac{\partial T}{\partial r}\right|_{r=r_{1,2}}. \tag{2}$$

We determine the characteristics of heat transfer at the inner and outer walls of the channel in the form of the functions

$$\mathrm{Nu}_{1,2} = f(\mathrm{Re},\ \mathrm{Pr},\ S/2\pi r_2,\ \delta/r_2,\ q_1/q_2),$$

$$\mathrm{Nu}_{1,2} = \alpha_{1,2} d_h/\lambda = 2q_{1,2}\delta/\lambda(T_{1,2} - \bar{T}),\ \delta = r_2 - r_1.$$

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