These three equations establish a relationship between the wet bulb temperature and the absolute humidity of the air.

(b) Heat and mass transfer

The rate of heat transfer across a boundary layer of air between the rock and the main airstream is given by

$$R_1 = H(\theta_s - \theta_D) \qquad (1.4)$$

where H is the surface heat transfer coefficient (see Chapter 2), θ_s is the rock surface temperature and θ_D is the dry bulb temperature of the main airstream.

The rate of mass transfer across this boundary layer is given by

$$R_2 = E(p_s(\theta_s) - p) \qquad (1.5)$$

where E is the mass transfer coefficient (see Chapter 2), $p_s(\theta_s)$ is the saturated vapour pressure of the boundary layer of air at the rock surface temperature θ_s and p is the vapour pressure of the main airstream.

If a fraction f of the rock surface is wet then the total rate of heat transfer across the boundary layer is

$$R = R_1 + fLR_2 = H(\theta_s - \theta_D) + fLE(p_s(\theta_s) - p) \qquad (1.6)$$

where L is the latent heat of water vapour.

Consider a volume of air flowing along a duct of cross-sectional perimeter P, at a rate Q. Let the direction of flow be taken as the positive x direction. Then from (1.4) the rate of increase of the dry bulb temperature along x is

$$\frac{d\theta_D}{dx} = \frac{P}{\rho C_p Q} H(\theta_s - \theta_D) + \frac{q}{\rho C_p Q} \text{ ---------- } (1.7)$$

where q is a heat source per unit length per unit time which may be used to simulate autocompression, lighting, broken rock etc., ρ is the density of the air and C_p is the specific heat of air at constant pressure.

Similarly, from equation (1.4) the rate of increase in the absolute humidity along x is

$$\frac{dw}{dx} = \frac{f\,P\,E}{Q}(p_s(\theta_s) - p)$$

Substituting from (1.1) and (1.3) this becomes

$$\frac{dw}{dx} = \frac{f\,P\,E}{Q}\{a\theta_s^2 + b\theta_s + c - w(\mu + \nu\theta_D)\} \text{ ---- } (1.8)$$

Equations (1.7) and (1.8) represent expressions for the increase in the dry bulb temperature and absolute humidity along the duct. In order to obtain a solution to these equations, a knowledge of the rock surface temperature θ_s is required.

(c) Heat conduction in the rock

The rate of heat flow across a rock-air interface is usually described by a forced convection boundary condition viz.

$$R = H(\theta_s - \theta_a) \text{ ---------------------------- } (1.9)$$

where θ_s is the rock surface temperature and θ_a is a constant ventilating air temperature.

If the rock surface temperature is taken as zero /....

zero when equal to the VRT and unity when equal to the air temperature θ_a then the rock surface temperature $\psi(t)$ may be written in a dimensionless form, viz.

$$\psi(t) = \frac{\theta_s - \theta_v}{\theta_a - \theta_v} \quad \text{------------------------------} \quad (1.10)$$

where θ_v = VRT. $\psi(t)$ is calculated for all time by using a one dimensional quasi-analytic approach[8]. In the ducts concerned the ventilating air temperature θ_a is not constant. By comparing equations (1.6) and (1.9) which both give the flux through the rock surface it may be seen that $\psi(t)$ is only valid if a fictitious air temperature

$$\theta_a(t) = \theta_D - \frac{f\ L\ E}{H}(p_s(\theta_s) - p) \quad \text{------------} \quad (1.11)$$

is used.

The temperature history of any part of the duct may be taken into account by using Duhamel's theorem[38], which states that the variable surface temperature $\theta_s(t)$ obtained when using a variable air temperature $\theta_a(t)$ is

$$\theta_s(t) = \theta_v\{1 - \psi(t)\} + \int_0^t \theta_a(u)\,\frac{\partial}{\partial t}\,\psi(t-u)\,du \quad \text{--} \quad (1.12)$$

where u is a dummy variable.

Equations (1.12), (1.11), (1.7) and (1.8) provide sufficient information to completely solve the problem, and they may be solved numerically to give the dry bulb temperature θ_D, the absolute humidity w, and hence the wet bulb θ_W at any point along the duct.

It is assumed that θ_D and w_n are known at x = 0 for all

/.

n where n is the number of time increments DT after the commencement of the calculation, i.e. the inlet conditions are known for all time.

The air temperatures θ_{a_n} may be calculated at $x = 0$ from equation (1.11) by substituting (1.1) and (1.3) as follows:

$$\theta_{a_n} = \theta_{D_n} - \frac{f\,L\,E}{H}\{a\theta_{s_n}^2 + b\theta_{s_n} + c - w_n(\mu + \nu\theta_{D_n})\} \quad \text{------ (1.13)}$$

where θ_{s_n} is given by the numerical expression of Duhamel's theorem, viz.

$$\theta_{s_n} = \theta_v(1 - \psi_n) + \tfrac{1}{2}\sum_{i=0}^{n-1}(\theta_{a_i} + \theta_{s_{i+1}})(\psi_{n-i} - \psi_{n-i-1}) \quad (1.14)$$

where ψ_n is known for all n. By solving (1.13) and (1.14) simultaneously, the values of θ_{a_n} and θ_{s_n} may be calculated at $x = 0$ for all n. It is assumed that the initial conditions, i.e. θ_{a_0} and θ_{s_0} are known.

Having the complete solution at $x = 0$, the situation at a distance DX along the duct may now be considered. Let θ'_{D_n}, w'_n, θ'_{a_n} and θ'_{s_n} represent the values of the variables at $x = DX$. Then

$$\theta'_{D_n} = \theta_{D_n} + \tfrac{1}{2}\,DX\left(\frac{d\theta_{D_n}}{dx} + \frac{d\theta'_{D_n}}{dx}\right) \quad \text{---------------- (1.15)}$$

and

$$w'_n = w_n + \tfrac{1}{2}\,DX\left(\frac{dw_n}{dx} + \frac{dw'_n}{dx}\right) \quad \text{---------------- (1.16)}$$

By substituting from equations (1.7) and (1.8) and then solving equations (1.12), (1.14), (1.15) and (1.16) in time steps DT, the solution at $x = DX$ may be built up. Using this solution as the starting point, the solution at $x = 2DX$ may now be found and so on as far as desired.

Using Starfield's models, particular attention was focussed on wetting down and it was shown that significant improvements could be obtained by carefully controlling the time and duration of wetting.

Starfield's one dimensional stope model has several inherent assumptions(8)(28) and the publication of a recently discovered finite difference technique for the solution of the two dimensional diffusion equation together with the availability of an extremely large computer made possible the refinement of his method.

This refinement is considered in Part II where the solution of a two dimensional mesh by Saul'ev's(33) method and the subsequent application of this result to the problem of heat flow into an advancing stope is examined.

Finally a complete two dimensional stope model is formulated.

PART I

CHAPTER 2

The sensitivity of heat pick-up to changes in H and E

Two of the parameters governing the flow of heat from rock to air in a stope are the surface heat transfer coefficient H, and the mass transfer coefficient E. Much literature is available on the determination of these quantities in idealised conditions[21] but their significance in underground conditions is not clearly defined. In an effort to establish the sensitivity of the computed heat flow and of the increase in temperature of the ventilating air to these parameters, several computer experiments were run using Starfield's one dimensional stope model[18].

2.1 An outline of Starfield's stope model

As a prerequisite to the determination of wet bulb and dry bulb temperature increases in an advancing stope, the dimensionless surface temperature of the stope for all time is required. Starfield obtained this by a one dimensional quasi-analytic method[8], with the basic assumption that the rock ahead of the stope face immediately after blast (for any rate of advance), was at virgin rock temperature (VRT). The implications of this assumption will be discussed later. By then simultaneously solving equations comprised of psychrometric relations, a heat transfer balance at the air-rock interface, and a numerically tractable approximation to Duhamel's theorem, an iterative solution which progressed along the length of the stope in previously defined space intervals was developed.

Factors which may be taken into account in this model include -

(i) the VRT;

(ii) the temperature and moisture content of the ventilating air;

(iii) the dimensions of the stope;

(iv) the rate of advance of the face;

(v) the fraction of the surface area of the stope which is wet (wetness factor f), and

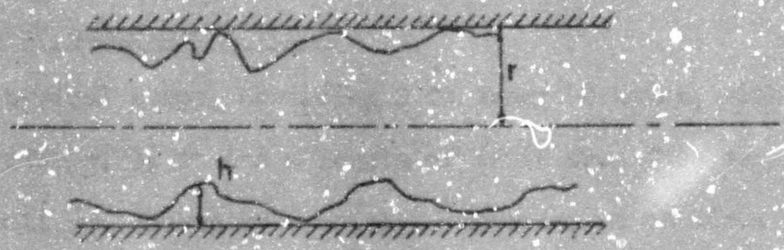
(vi) a heat source to simulate the presence of broken rock, or autocompression in a non-horizontal stope.

2.2 Theoretical limits on the value of H

In the case of the smooth pipe, values of H may be calculated from one of several empirically established formulas[21]. Roughness on the inner surface of a pipe introduces an increased friction effect. Roughness is defined as the ratio of the height of protuberances on the surface to the radius of the pipe as shown in Fig. (2.1). The relation between the value of H and different magnitudes of roughness may be clearly seen from results obtained by Nunner[22], shown in Fig. (2.2). Curves are presented for Reynolds numbers ranging from 5,000 to 80,000.

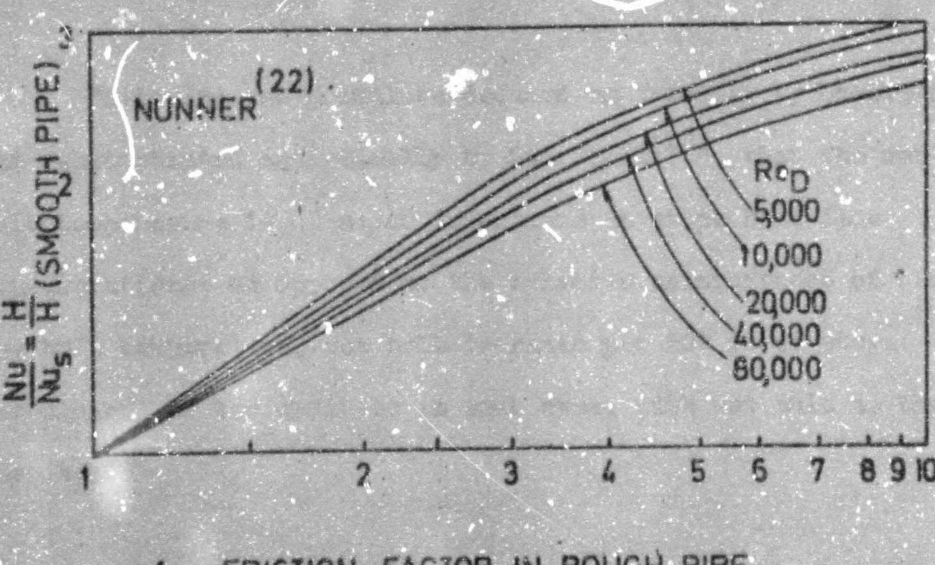
From Fig. (2.2) it may be seen that the lower bound for H is the "smooth pipe value" while an upper bound may be taken as three times that value. This range probably includes most values of roughness likely to be encountered underground.

FIG 2·1 ROUGHNESS IN A PIPE



ROUGHNESS $= \frac{h}{r}$

FIG 2·2 EFFECT OF ROUGHNESS ON HEAT TRANSFER IN TURBULENT FLOW (NUNNER)[22]



$\frac{f}{f_s} = \frac{\text{FRICTION FACTOR IN ROUGH PIPE}}{\text{FRICTION FACTOR IN SMOOTH PIPE}}$

In applying this theory to rectangular stopes, the pipe diameter must be replaced by the equivalent diameter D_E[23] where

$$D_E = \frac{4 \times \text{area of stope}}{\text{Perimeter of stope}}$$

2.3 <u>The computer experiments for H</u>

A typical stope, advancing 2 ft every 48 hours, with a stoping width of 3½ ft and a length of 12 ft from face to scatter wall was chosen as a model. In order to obtain a comprehensive set of results, the VRT, the air velocity and the wetness of the stope were each varied independently of the other parameters. Inlet air temperatures of 70/75° F, 75/75° F, 80/85° F and 90/105° F were used with H values of once, twice and three times the smooth pipe value respectively. The value of the mass transfer coefficient was calculated from the Lewis relation[24] (which was assumed valid) in the form

$$E = \text{const} \times H$$

Following the postulate deduced empirically [illegible]hts[20] and substantiated analytically by Starfield[18], that [illegible] wet bulb temperature is of major importance in stope heat flow considerations, as opposed to the relative unimportance of the dry bulb temperature, wet bulb increase per 100 ft of stope was plotted as the ordinate in each case, with wet bulb as the abscissa.

In all cases the conductivity of rock k was taken as 3.0 BTU/ft^2 hr (°F/ft) and the diffusivity D as 0.10 ft^2/hr.

<u>Case (i)</u> The VRT was varied keeping the air velocity constant

/at

at 220 ft/min with ... ...ness factor f = 0.8. VRT values of 100°F, 120°F and 140°F were used, giving a range wider than that encountered at present.

From the results (Fig. (2.3)) it may be seen that over the range considered, the wet bulb increase per 100 ft is virtually insensitive to changes in H for a wide variety of VRT's.

(VRT - wet bulb) has been widely used as the driving force for heat flux in empirical studies but this graph shows that wet bulb increase is not well correlated on a basis of (VRT - wet bulb), since for the same (VRT - wet bulb) the wet bulb increase at a VRT of 140°F is half as much as that at a VRT of 120°F. Fig. (2.4) illustrates this point over a wide range of temperatures.

<u>Case (ii)</u> The air velocity was varied for a constant VRT of 120°F, again with f = 0.8. Values of 110 ft/min, 220 ft/min, 440 ft/min and 880 ft/min were used, covering the practical conditions found underground.

The results are presented in Fig. (2.5) which shows that the smaller the air speed, the greater the discrepancy between extreme cases of H = 1 x (smooth pipe value) and H = 3 x (smooth pipe value). The reason for this is that at 110 ft/min, E is extremely small and doubling it (since E = constant

/xH)

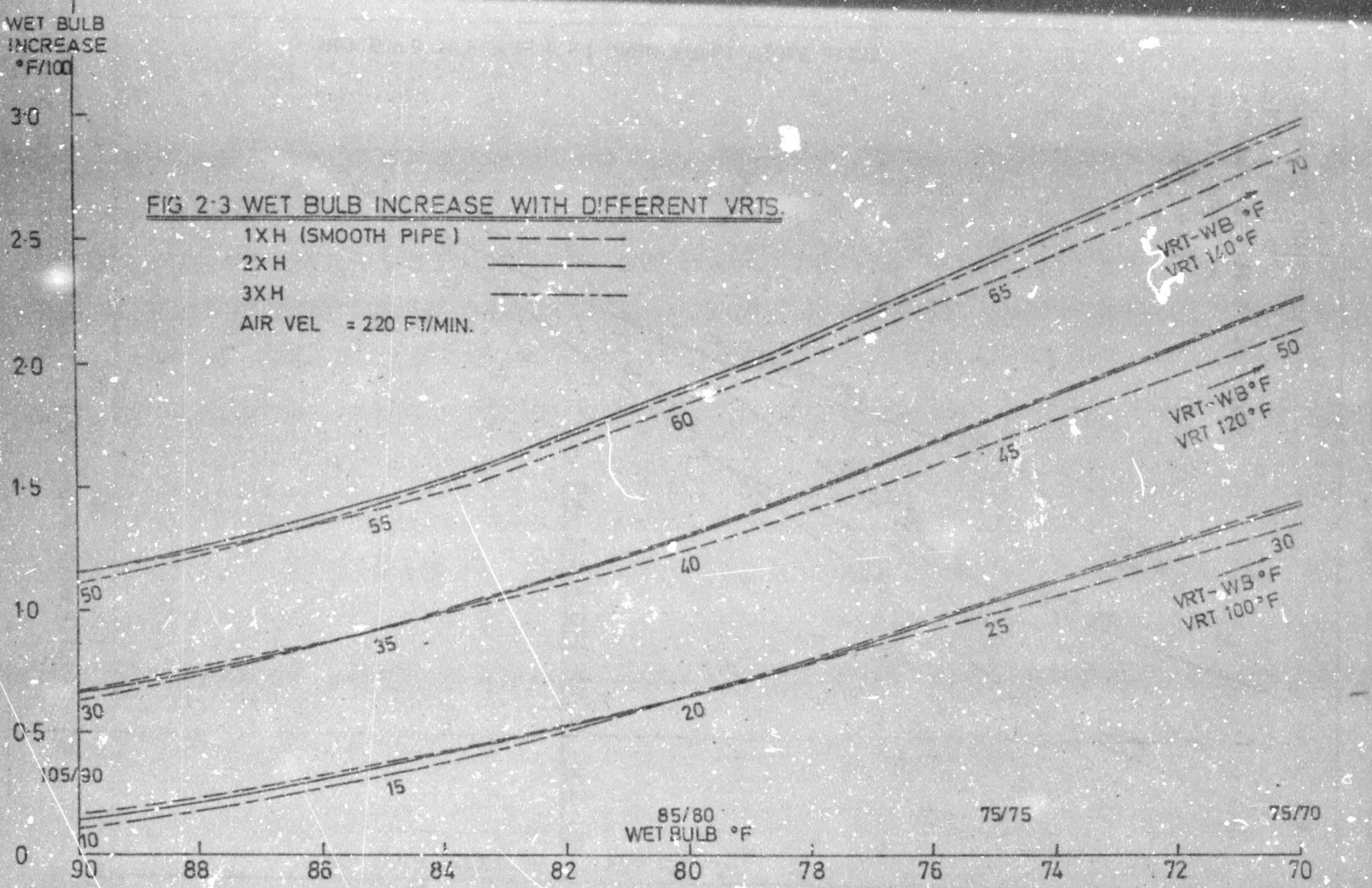


FIG 2·3 WET BULB INCREASE WITH DIFFERENT VRTS.

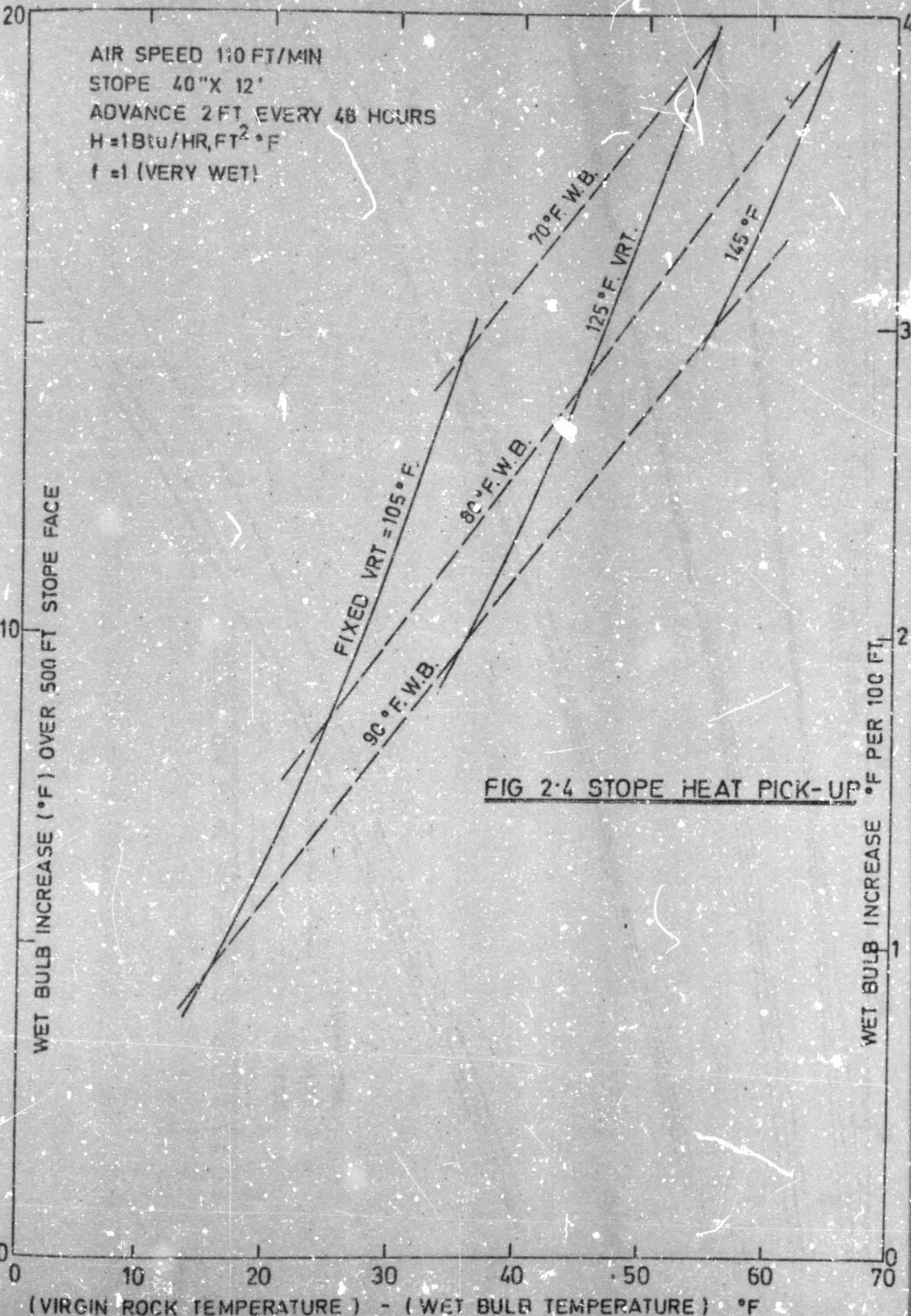


FIG 2·4 STOPE HEAT PICK-UP

FIG 2·5 WET BULB INCREASE WITH AIR VELOCITY