SUPERSATURATED STEAM, AND ITS EFFECT ON THE FLOW OF STEAM THROUGH TURBINE NOZZLES.

When dry saturated steam is discharged through a nozzle the theoretical quantity of steam passing per second through the smallest section of the nozzle can readily be calculated when the pressure difference (p_1–p_2) existing between the two ends of the nozzle is known. Up to a certain point, for a given initial pressure (p_1), the lower the final pressure (p_2) and the greater the value of (p_1–p_2) the greater will be the mass of steam discharged through a given sized nozzle. This quantity does not go on increasing indefinitely, however, and when the final pressure is less than a certain proportion of the given initial pressure, it is found that the mass discharged remains constant however much the final pressure may be further reduced and that this mass is the maximum that can flow through the nozzle. The limiting final pressure at which maximum flow first occurs is given by the ratio,

$$p_2 = \cdot 577\ p_1$$

for steam initially dry and saturated, and for any terminal pressure less than this value the discharge does not further increase, though the *velocity* of discharge will go on increasing as the terminal pressure is reduced, provided the nozzle is fitted with a properly designed divergent end.

The pressure (p_2) at which maximum mass flow of the steam first occurs is called the "critical pressure," and the theoretical maximum value of the steam flow under these conditions is given by the expression,

$$W_{max} = 3 \cdot 604 A \sqrt{\frac{p_1}{v_1}}$$

where W is in lbs. per second, A is the smallest cross-sectional area of the nozzle in sq. feet, and p_1 and v_1 are the initial pressure in lbs. per sq. foot and initial volume in cu. feet, respectively.

This expression should give the correct value of the discharge from an orifice with a properly designed entrance if there were no friction or eddy resistance.

Actually the conditions are never perfect and the discharge might be expected to be slightly less than the value so calculated.

It has been noted, however, by many experimenters that the discharge is *always greater* than that obtained from the above expression, the excess varying under different circumstances but having a maximum value of about 5 per cent.

There has been considerable discussion as to the reason for this discrepancy, but the explanation now generally accepted is that between the entrance and the throat of the nozzle the steam is in a peculiar condition which is termed "supersaturated."

According to the usual theory, when the steam expands in a nozzle the velocity energy produced is obtained at the expense of the heat energy in the steam and hence if it starts dry it loses heat during expansion and becomes partially wet at exit from the throat.

It is generally recognised that a certain time is required for this condensation to take place and it is inferred that in an actual nozzle the interval between the steam passing the entrance and reaching the throat (which may be of the order of $\frac{1}{6000}$ of a second in a turbine), is too small for condensation to occur and supersaturation results.

In this condition the steam is in an unstable thermal state, since no condensation has taken place and at successive pressures the temperature of the steam falls below the corresponding dry saturation values, and its temperature is no longer a criterion of its pressure.

According to this hypothesis the steam continues to expand like a gas, and the law of expansion is the same as for superheated steam, namely,

$$p.u^{1 \cdot 3} = \text{constant.}$$

The volume of the steam per lb. (u) at the throat can thus be calculated and it will be found appreciably less than that obtained under the usual assumptions, while the mass of steam discharged will correspond to that for the flow of superheated steam. Moreover, the ratio of the throat pressure to the initial pressure for maximum discharge will no longer have the previous value but will have the same as for superheated steam, viz. :—

$$p_2 = \cdot 546\ p_1$$

On the whole the different method of calculation leads to an increase in the maximum mass flow of from 4 per cent. to 5 per cent. which may introduce an appreciable correction when designing the areas of turbine nozzles.

For any given drop in pressure the heat drop, on the assumption that supersaturation takes place, is less than that due to ordinary adiabatic expansion, so that a loss of efficiency occurs. The degree of supersaturation existing will depend on the pressure fall for which the nozzle is arranged, but the difference between observed stage pressures in actual turbines and the pressures for which they were designed on the usual theory leaves little doubt that something of this nature does take place.

It is extremely difficult to measure the temperature of supersaturated steam since the steam in its unstable state at once condenses on the thermometer used and the heat given up by the condensation raises the temperature reading above the true supersaturation value.

Since the steam behaves as a gas the temperature of supersaturation (T_2) at any pressure (p_2) can however be obtained from the formula,

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\cdot 3}{1 \cdot 3}}$$

where T_2 and T_1 are the absolute temperatures of the steam at the pressures p_2 and p_1 respectively.

It has been shown experimentally that this unstable condition of supersaturation cannot persist beyond a certain point, which corresponds to a pressure of about one-third of the initial pressure before expansion, the actual limit being when the pressure after expansion is eight times the saturation pressure, corresponding to the actual supersaturated temperature of the steam.

What happens when supersaturation ceases is doubtful, but it is probable that condensation takes place with great rapidity, the steam returning to its wet state condition while heat is lost in the process.

This certainly occurs at the supersaturation limit just referred to, but in ordinary impulse turbine design where the nozzle pressure ratio at each stage does not approach the limiting value of 3 to 1 it is possible that supersaturation is relatively slight and the steam returns to its stable state by the time it has passed through the moving blades.

At present these are suppositions which can only be confirmed by very careful comparisons between the trial results of actual turbines and the anticipated results based on the usual methods of design, and though from data available there are distinct indications that supersaturation does occur in modern turbines, until more information of this nature is available the exact extent of any correction necessitated by the theory of " supersaturation " remains somewhat uncertain.