Fan manufacturers' performance data are usually obtained either through accurately conducted tests of scale models, by theoretical calculation, or through the use of limited full-scale field testing. Well-made models under laboratory conditions provide accurate data which are finding increased use among fan producers, although the basis of published ratings is often vague.

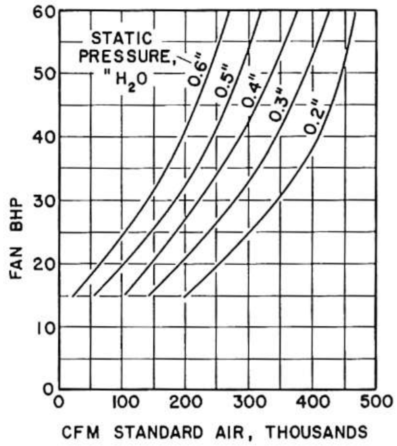

Fig. 10 Performance characteristics of 19-ft diameter 6-bladed fan with 63-in. diameter hub based upon 3-ft diameter scale model tested in standard flared wood cooling tower fan cylinder

Test data (or other basis of rating) may be presented in many ways, one of which is shown in Fig. 10. Note that this plot is for a specific fan as tested in a fan throat of a particular configuration, and for a tip speed of 10,000 feet per minute. The static pressure drop due to air flow through the fan cylinder is here charged against the fan, and as such the fan and its cylinder are rated as a unit. Fig. 10 was obtained from a 36-in. diameter model of the 19-ft fan discussed in the example and indicates that 58 horsepower is required to handle 372,000 cfm at 0.425 in. water standard static pressure. Thus the power required at an actual air density of 0.0705 lb/cu ft,

$$HP = 58 \times 0.0705/0.075 = 54.5$$

and if the power transmission efficiency is 90 per cent, the motor horsepower required is $54.5/0.90 = 60.5$. Agreement with the full scale field test value of 50.8 is not entirely satisfactory. As in this example, the efficiency of models is usually somewhat lower than full scale fans, due to the difficulty of maintaining tip clearances that are accurately to scale, and the fact that the model operates at a lower Reynolds number.

Fan Cylinders and Stacks

As previously mentioned, the theoretical method of performance estimation set forth in this paper is based on the static pressure difference across the fan disk. In using the method of calculation, it is necessary to estimate the static pressure loss in the fan cylinder in order to obtain best accuracy. The fan cylinder static pressure loss may be taken as some percentage of the velocity pressure at the fan disk. The factor may be as low as 2 per cent, if the cylinder is of well designed proportions with a flared inlet and may be as high as 200 per cent for a sharp edge orifice mounting. The usual commercial design of cooling tower fan cylinder will have a static pressure loss in the order of one quarter of a velocity head at the cylinder.

Ordinarily, extended stacks as are sometimes used to assist in discharging the humid air at a distance of 20 or 30 feet above the fan will have a negligible effect on the fan performance. If the stack is flared outward some kinetic energy recovery will result, which may be of practical significance under conditions of very low static pressure (say in the order of $^1/_4$ in. of water or less). The method of fan performance estimation presented here may be used with reasonable accuracy, and the static pressure reduced by the amount of velocity head recovery. It is doubtful that more than about one half of the kinetic head can be recovered economically.

Summary

A simple method of predicting the performance of cooling tower fans has been presented and compared by example with results of both model and full scale tests. The method may be used to advantage by purchasers of cooling tower fans to determine the validity of performance data presented by prospective suppliers of cooling tower fans. Although the calculated horsepower may be in error by as much as 20 per cent, the predicted air flow will be within approximately six per cent (the cube root of 1.2), which is within field test accuracy. Manufacturers of cooling towers and cooling tower fans have, over the years, indicated accuracies of fan performance and horsepower prediction that are inconsistent within the state of knowledge of the art.

DISCUSSION

J. D. Holmberg[3]

This paper is an excellent presentation of a method of calculating the maximum capability of a cooling tower fan. It clearly shows the development of the simple blade element theory which has generally been used in cooling tower fan design. This is the first time the theory has been presented in this form, which greatly simplifies calculations through judicious choice and use of curves. The authors are to be commended for the effectiveness of their approach.

It should be kept in mind, however, that the calculated performance represents a performance "ceiling." The actual performance is always something less, partly because of losses which occur in the application of the fan and partially because the theory is based on several assumptions which are not precisely true. These basic assumptions are discussed, not as a criticism of the authors but to emphasize the fact that the calculations do establish a maximum performance which the fan can, at best, only approach.

This theory assumes that the discharge air flow is truly axial; actually it always has a rotational component. Airfoil drag coefficients are higher and maximum lift coefficients are lower at the lower Reynolds numbers corresponding to actual fan operation than at the Reynolds numbers for which the NACA has published airfoil characteristics. The method of calculation also assumes that the performance of the fan as a whole is represented by that of the blade elements at 75 per cent radius. This is most nearly true at design blade pitch and at design static pressure, and it becomes less accurate as an approximation as either of these is changed.

Calculations from the theory are based upon operation of the fan under ideal conditions. In the actual application, losses will occur because of back flow at the hub and blade tips, nonuniform velocity distribution of air approaching the fan, and/or angularity of inlet air flow. These losses are influenced by the design and arrangement of other cooling tower components, particularly the fan cylinder. In applying a cooling tower fan, the ultimate goal is to obtain fan performance very nearly equal to that calculated by the method of the paper. How closely that goal is approached can be determined only by test.

The rating of a cooling tower is no more accurate than the rating of the fan. The fan designer, guided by limitations learned

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from experience, can successfully use the theory in the fan design. One fan design can be compared with other fan designs by thorough tests of an accurate scale model. Performance ratings, however, are acceptable only after having been confirmed through field tests of the full-scale fan as applied on the cooling tower.

The paper points out the difficulty of conducting accurate field tests. It must be agreed that it is a difficult, painstaking task. It is only after numerous tests and gradual refinement of testing methods that we have reached a point of being able to obtain accurate field test results. Volume measurements consistently agree with the fan rating within three per cent. Differences greater than that would call for a thorough investigation as to the cause. It has been through investigations of past differences that correct procedures for applying, rating, and testing cooling tower fans have been developed.