THE DESIGN OF A PORTABLE AIR-CIRCULATING UNIT

Thesis by

Harry L. Masser, Jr.

In Partial Fulfillment of the Requirements
For the Degree of
Industrial Designer

California Institute of Technology
Pasadena, California

1950


SPREAD FLOW STRAI
HI OFF HI
PRODUCTS CORP

ACKNOWLEDGEMENTS

The author wishes to express his appreciation and gratitude to Mr. J. Paul Youtz and Mr. David F. Welch, without whose assistance and encouragement this thesis could not have been completed. The aid and advice of other members of the Engineering and Industrial Design faculties of the California Institute of Technology have been most helpful.

Thanks must also go to the many manufacturers, distributors, and retailers of fans, motors, and other appliances and components who supplied material required in the development of this design.

A B S T R A C T

A portable electric air-circulating and heating unit has been developed for home or office use. The problem was approached through evaluation of the atmospheric factors relating to human comfort, and included a basic reconsideration of the means by which control of these factors might be achieved. Study of competitive products on the home appliance market, and of the requirements and desires of both the trade and the consumer, indicated the practical scope of such a device.

The design combines, in a single attractive unit, the functions of both straight- and diffused-flow fans with a forced-convection heater. A propeller of special design, driven by an adjustable-speed reversible motor, operates at optimum efficiency in either direction of rotation. Its housing provides, in one direction of flow, an injector inlet with axial discharge; with flow reversed, radially diffused air at low velocity gives draft-free direct circulation. The maximum capacity is equivalent to that of the best conventional twelve-inch fans. Any required directional setting may be obtained with the unit on the floor, table or wall.

Increased flexibility of operation, with modification of the seasonal aspects of manufacture, distribution, and use, are thus offered, in a unit which is fully effective in each of its functions.

CONTENTS

Contents

FIGURES

INTRODUCTION

Purpose and Scope

The purpose of this thesis is to develop the complete design of a portable air-circulating unit for use in home and office. The function of the unit is to provide a high degree of control or regulation of atmospheric conditions for human health and comfort, in a simple, compact device suitable for wide application. The unit is intended to replace the conventional electric fan, combining new and standard features for improved performance and greater flexibility of use. It is assumed that it is to be manufactured and marketed by a representative established manufacturer of home electric appliances.

The scope of the problem includes a determination of the requirements of such a unit through consideration of the variable atmospheric factors which are controllable, and their relative effect on comfort. Means of accomplishing the required control within a relatively simple unit are studied, with particular reference to technological advances which have not been embodied in products now available. In spite of the highly competitive nature of the home appliance field, the electric fan of today is so similar to its prototype as to suggest that no basic reconsideration of design criteria has recently been attempted.

In order to increase the practical value of the results, the conditions which would be present in an actual designer-client relationship are maintained. The final design is synthesized by reconciliation of the theoretical considerations with the practical limitations imposed by: the production facilities available to the manufacturer; the characteristics of the trade including distribution, promotion, and sales; and the desires of the consumer.

Historical Notes

The value of air movement as a factor in human comfort has been recognized since early man first noted the cooling effect of a breeze in warm weather. The "palm-leaf" fan was the first means used for providing air movement artificially, and was followed by more elaborate reciprocating fans such as the Indian punkah, a hanging curtain of matting or cloth pulled back and forth by a servant, and the highly ornamental feather fans manipulated by the attendants of Egyptian and African rulers, which grew to have a purpose as much decorative and ceremonial as functional. The use of reciprocating fans, operating inefficiently as they do by continuously stopping and reversing the direction of air flow, persists to some degree today, but in industrialized countries has been almost entirely replaced by mechanical units. With modern provisions for air circulation and ventilation, the hand-held fan, embellishment of which was once practically a fine art in Europe and Asia, is almost obsolete even as a feminine adjunct to the social graces.

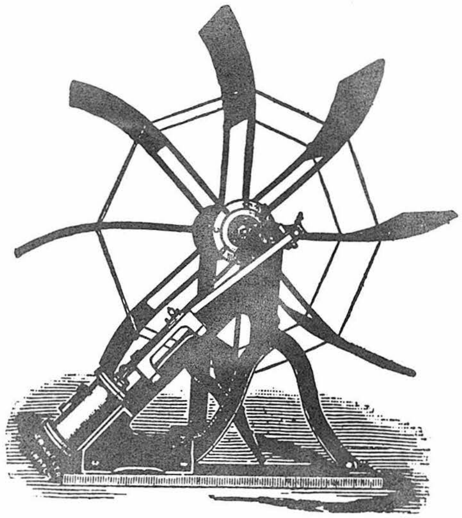
The Egyptians in their pyramids, and the Assyrians in elaborate cave temples, were among the first to realize the need for artificial ventilation and provide specifically for it, by employing passages in the stone structures to utilize natural draft for freshening the air supply. Development of mechanical air-moving units followed closely the first application of reciprocating engines. The earliest such devices consisted merely of a series of paddles arranged radially around a shaft (Fig. 1-A) and were employed for mine ventilation, then a critical problem in English collieries.[1*] The addition of a housing was an early contribution toward increased efficiency, and led to the perfection of the centrifugal blower, as well as to the development of controlled ventilation by means of ducting.

The axial-flow or propeller type of unit, typified by the household electric fan of today, had its inception in two inventions which might better be classed as evolutionary developments. The first of these was the screw propeller, utilizing a principle familiar to the ancients as the Archimedean screw, which was first proposed for ship propulsion in the late 18th century.[2] Credit for its invention, although usually given to the Englishman Francis Pettit Smith and Captain John Ericsson, a Swede, for their operable patents of 1836, should probably be extended to the Austrian inventor Ressel, who patented

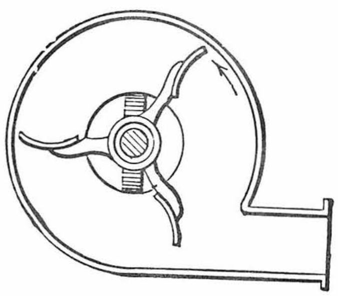
- -

* Superior numbers indicate specific references listed in Appendix I.

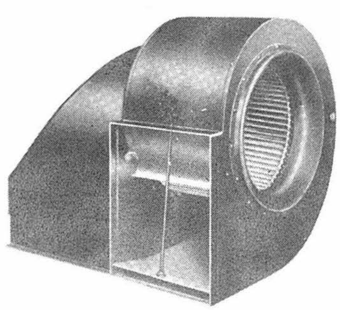
Centrifugal Fans



A. Steam Driven Mine Ventilator

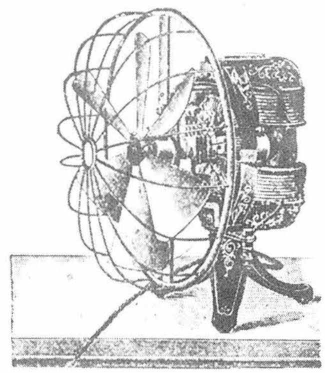


B. Early Blower

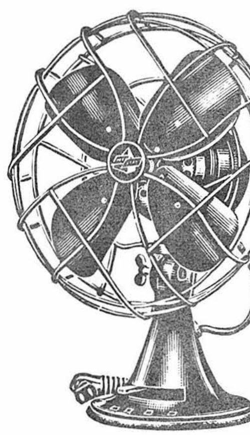


C. Modern Centrifugal Fan

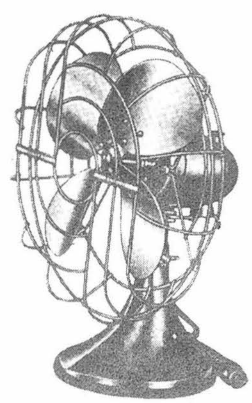
Propeller Fans



D. Battery-Driven "Edison Model" (1890)



E. Typical Oscillating Fan (1920)



F. Modern Electric Fan (1950)

Figure 1.

The Development of Fan Types

such a propeller in 1812 and used it practically in 1829, ten years before the others saw use.

The second prerequisite invention was that of the electric motor, which furnished a compact source of power. Henry's electromagnetic oscillating machine of 1829 was the forerunner of all electric motors,[3] which by the 1880's had become a common and dependable source of motive power in Europe and America.[4] It is difficult to tell exactly when these two inventions were first combined into a propeller fan, but such units were commercially available as early as 1890, according to the General Electric Company. By the beginning of the twentieth century, electric fans were in fairly common use in American homes and offices. In its usual form, today's "modern" fan differs little from its ancestors of sixty years ago (Fig. 1-D,E,F). Specialized, highly efficient types have, however, been developed for industrial applications.

Atmospheric Factors in Human Comfort

The physical comfort of man is primarily controlled by the conditions of the atmosphere in and around his immediate environment. The two general factors which most affect an individual or group within the atmospheric surroundings of an enclosed space (throughout the wide range of living and working conditions that might be termed "normal") are the weather and the rate of metabolism. The enclosure may be designed as a shelter for protection from wind, rain, and the direct radiation of the sun, as well as insulated against excessive gain or loss of heat by convection and conduction. Such a shelter cannot, however, be perfectly insulated, nor can it in itself control the conditions of the air needed to sustain life. As long as doors, windows, and other openings allow the flow of any substantial volume of outside air into the space, the problems of the temperature, humidity, and purity of the supply will remain.

If it is decided to limit the amount of air transferred between a given enclosure and its surroundings in order to control the exchange of heat, moisture, and impurities, the problem of conducted heat remains, and in addition the effects of the life processes of the occupants increase in importance. These processes of catabolism increase the temperature of the air and vitiate it by decreasing the oxygen content, raising the amount of carbon dioxide, and creating objectionable odors by the addition of organic matter given off in expiration and perspiration. The humidity also is increased by moisture evaporated from the skin and lungs.[5]

Metabolic loss of heat from the body normally takes place through radiation, convection, and evaporation, to the degree necessary to balance the heat production of the body and maintain its normal temperature of approximately 98.6° F.[6] The relative amounts of heat lost in each of these ways depend on the correlation of air dry-bulb temperature, relative humidity, exertion, and weight of clothing. As these four factors rise, maintenance of bodily temperature equilibrium is increasingly dependent on cooling by the evaporation of perspiration. At air temperatures in excess of body temperature, radiation and convection losses become negative, latent heat losses become insufficient (even though evaporation will continue as long as the air is not saturated), and the metabolic (oxidation) rate rises in spite of the difficulty in disposing of excess heat. Heat storage then takes place in the body, whose temperature will continue to rise until it reaches the wet-bulb temperature if this is above normal body temperature. The limit of the human system is reached when its temperature is thus increased to about 106 or 108° F (although dry-bulb temperatures considerably higher can be endured when the air is not saturated).

Conditions under which the body may maintain its thermal equilibrium have been the subject of considerable research, with the primary purpose of establishing a quantitative index combining temperature, humidity, and air velocity in such a way that sets of conditions which are equivalent in thermal effect, i.e., which produce the same feeling of warmth, may be predicted.[7] Such equivalent conditions also produce identical physiological reactions in the body. The results of tests conducted by the American Society of Heating and Ventilating Engineers have been compiled in the form of an "effective temperature" chart (Fig. 2). The effective temperature is so called because it denotes the sensory heat level as determined by the effects induced in the body by heat or cold. It is the same as the dry-bulb temperature in saturated still air (defined as air moving at an average velocity not greater than 15 to 25 fpm), and drops as the wet-bulb temperature falls and/or air velocity rises. The A.S.H.V.E. has also found the psychrometric region, called the "comfort zone," in which the majority of adults will feel comfortable. This zone lies between the effective Fahrenheit temperatures of 66° and 75° in summer, with a maximum number of the

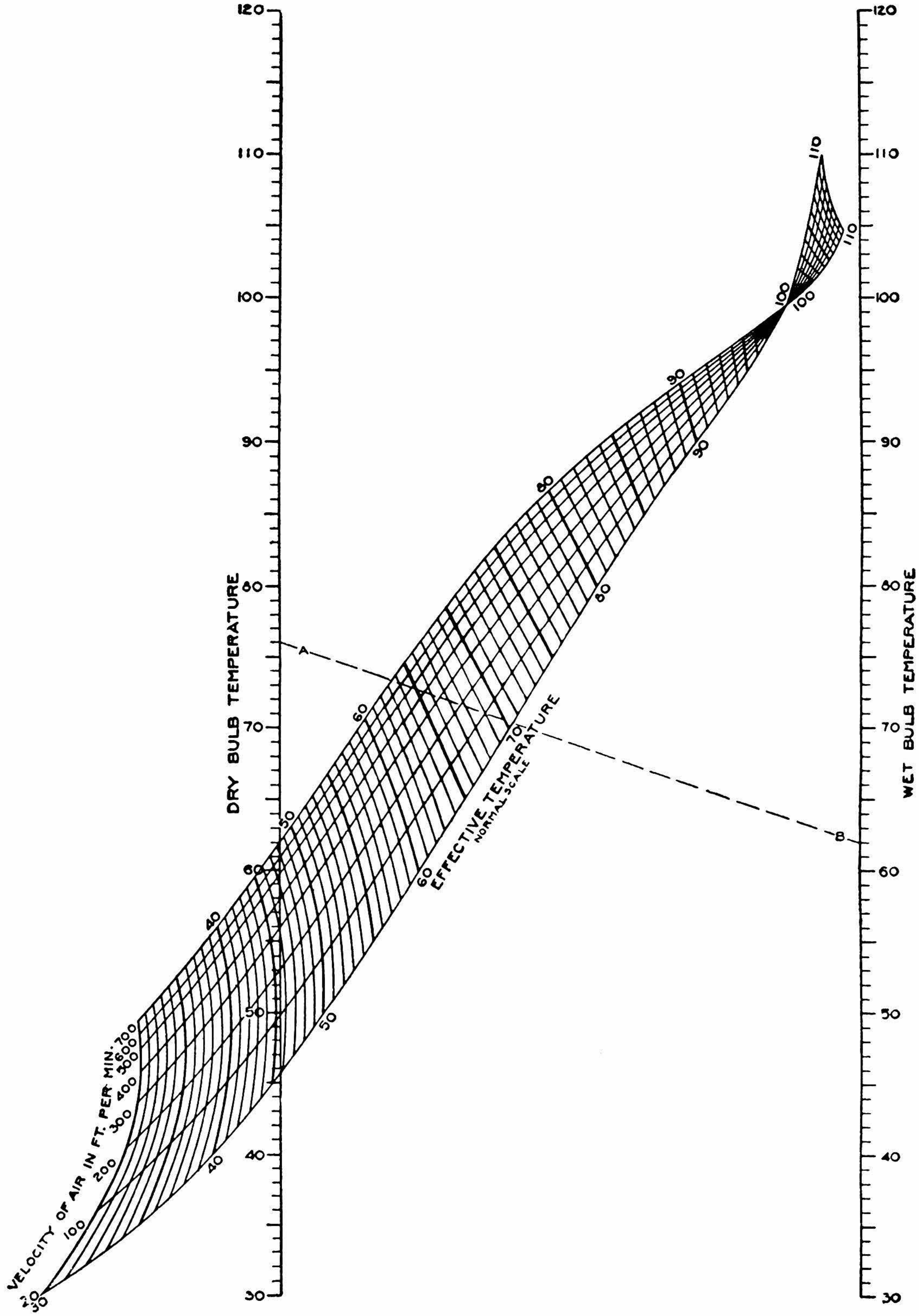


Figure 2.

The "Effective Temperature" Chart (A.S.H.V.E.)

trained test subjects comfortable at 71° ET, and between 63° ET and 71° ET in winter, with the maximum at 67° ET (Fig. 3). The preferred relative humidities are based between 30% and 70%, although comfort may be experienced somewhat outside this range. The center of the combined summer and winter comfort zones is at about 73° F dry-bulb and 50% relative humidity, with some 80% of the subjects comfortable at this point in both seasons.

In still air, moisture evaporated from the skin tends to accumulate in a "boundary layer" of warm moist air near the body surface, acting as an insulator against further latent heat loss unless removed and replaced by air which is cooler and drier.[8] Although this condition is accentuated at the higher temperature and humidity levels, where its effects are also most harmful, it obtains in some degree down to and below the comfort zone since a certain amount of perspiration is produced and evaporated even at low temperatures. In order to maintain temperature equilibrium, air movement is thus essential under all situations, at velocities varying with temperature and humidity, and also dependent on whether required for positive cooling or principally for health. Where all the air within an enclosure is at conditions above the comfort level, its velocity may be relatively high, from a minimum of 150 fpm to more than 300 fpm.[9] When cooled

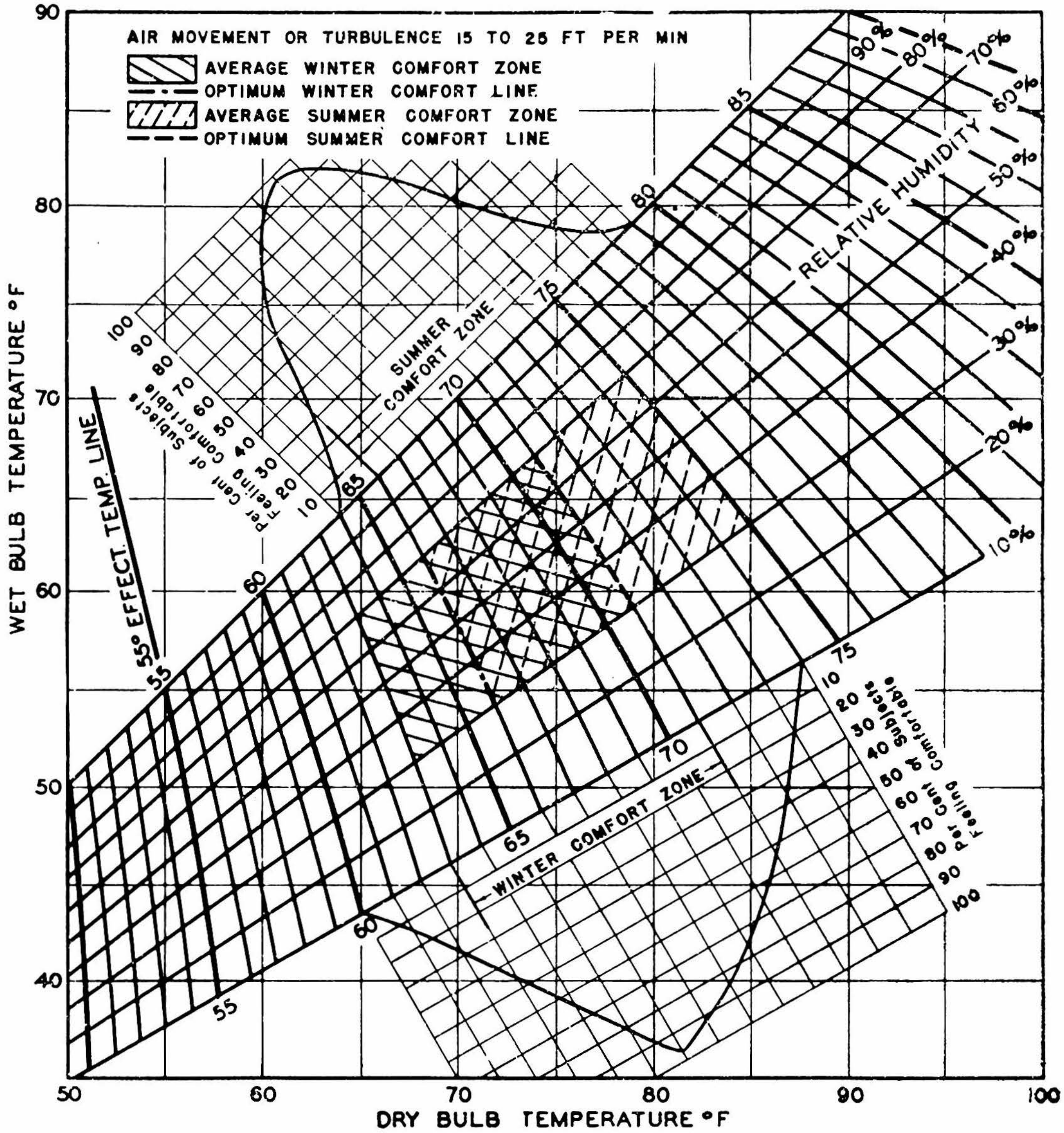


Figure 3. The Comfort Zone (A.S.H.V.E.)

air is being introduced into a warm space to reduce the overall temperature it should not, in general, strike the occupants at rates in excess of 50 fpm (although its velocity at the point of entry may be considerably higher to insure thorough mixing).[10] The reverse condition, of heated air being circulated in a cool space, permits of somewhat higher velocities, but it must be remembered that the basic effect of moving air is to raise the evaporation rate: the heat units added must exceed those lost in latent heat (as well as in radiation to cold surfaces) to produce a sensation of warmth. In this connection it should be noted that, within wide temperature limits, air moving at any detectable velocity will psychologically induce a feeling of coolness in the average subject. Even distribution of velocity and avoidance of localized air currents are important since, with accelerated evaporation, differential cooling (evidenced by the sensation of "draftiness") has a deleterious physiological effect due to the irregular demands imposed on the vasomotor system.[11]

Overall control of the atmosphere may be achieved within delimited areas by means of air conditioning, which is defined by the A.S.H.V.E. as the "simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors and toxic gases, most of which affect in greater or lesser degree human health or comfort."[12] In its practical application, the amount of air transferred between the air-conditioned enclosure and its surroundings is customarily kept as small as is commensurate with the number of occupants and the type and degree of their activity. Operating costs are thus reduced by taking advantage of the condition of the recirculated air rather than continuously modifying new supplies from the outside. In spaces where contamination of the air is due solely to respiratory processes, as little as 10 cfm of fresh air per person may be adequate, and a minimum of 15 cfm may suffice where smoking is permitted.[13] To assure that all outside air passes through the conditioner for treatment, an amount must be so introduced that the natural leakage through cracks around doors and other openings will be exceeded. In buildings of average construction sufficient air leakage will occur naturally to prevent pollution and depletion of oxygen to an extent inimical to health, so that in general no special provision need be made for controlling air purity.[14] This will hold true when the occupants are engaged only in the usual activities of home or office, and have an adequate space allotment of 100 to 300 cubic feet per person; the respective minimum outdoor air requirements here range from 30 to 10 cfm per person.[15]

Ventilation, defined as the "process of supplying or removing air, by natural or mechanical means, to or from any space," may be required in amounts greater than supplied by leakage alone to remove the larger quantities of smoke and odors arising from heavy smoking, cooking, or other sources, and to provide for high degrees of occupancy. Often natural draft through vents or relatively small window openings will provide such supplemental ventilation. In other cases a small mechanical unit installed at the source of pollution, such as a kitchen exhaust fan, may be desirable.

Total air change is desirable primarily for the purpose of temperature control, generally in cooling, where air conditioning is not provided. It may be applied to advantage whenever the air in a given enclosure is warmer than that in the surrounding space, as in hot weather when the night air is cooler than that in the house, or in a working space where exothermal processes are involved. The unit for accomplishing this air change may be a blower, which is of primary value when control of the source of the entering air is required, as for instance in subway-car ventilation when the air must be filtered for dust removal.[16] In most other cases an exhaust ventilator will be preferable. For home cooling such a unit is often installed in the attic to exhaust the hot air which accumulates there and to bring in cooler air through windows on the ground floor, particularly at night.[17] Other types are designed for installation in wall openings or windows, and even a conventional portable fan may be used effectively as an exhauster when placed about 3 feet inside a window.[18] Exhaust ventilation insures air change throughout the whole of a structure with less regard to the location of auxiliary openings than in the case of a blower. While it is axiomatic that the air which enters a given space will force the exit of a like amount (provided the space is not pressure-tight), a blower is likely to set up ineffective counterflows in corners, etc., where no egress exists. An exhauster will in general be more efficient since a direct path may be established between the inlets and the unit, through the spaces to be cooled.

In heating applications air movement must be provided not only to assure mixing of the warm air at the point of its introduction into a space, but also to reduce the temperature stratification produced as the less dense warm air rises and collects at the ceiling. In a room heated by a gravity system to 70° F at the 5-foot breathing level, the temperature at a ceiling level of 10 feet is often at least 7° higher.[19] The heat represented by this difference is not only practically unavailable to the occupants, but increases the heat loss to the outside because of the high differential in temperature. Proper air circulation will increase the effectiveness of the heat source by moving the warm air at the ceiling down to the occupied level. Where a radiator or convection heater is used, heat transmission will be increased by forcing cool air from the floor level over the hot surfaces, reducing the insulating effect of the stagnant boundary layer at these surfaces. By these two actions greater heating efficiency is attained, and may act either to raise the heat output of a given source or to reduce the amount of fuel required to maintain the desired temperature level.[20] A further contribution to comfort results from the warming of cold walls by air flow, which in raising their temperature reduces the absorption of radiant heat from the body.[21]

The foregoing discussion demonstrates that air movement is an important factor in man's atmospheric environment, and the only one which appears in all phases of control of this environment for comfort or health. It forms a part of any properly designed cooling or heating system, and as a supplement to such an installation provision for additional circulation may increase its efficiency. In comfort cooling, exhaust ventilation permits full advantage to be taken of a naturally cool source such as the night air. Where internal circulation alone must be depended upon, considerable beneficial cooling may be obtained by motion of the air at room temperature without risk of the sometimes objectionable thermal shock often experienced when entering a space cooled by conditioned air.

A device to insure air movement of the proper character and degree is thus the simplest and perhaps the most effective one which, operating on a single principle, can materially affect the conditions of the atmospheric environment for greater human comfort. Such a unit, here termed an "air circulator" rather than simply a fan to signify that the scope of its application is enlarged to include many of the uses described above, is developed in the following pages.

M E C H A N I C A L   F A N S

The general term "fan" is used to describe any device used for impelling gas or setting it in motion. Such a device may be physically very similar to those used for compressing or transferring fluids and classed as "pumps." In dealing with air or gases, the distinction is usually made on the basis of the difference between inlet and outlet pressures: units dealing with the highest pressure differentials are termed "compressors" or "vacuum pumps"; those in an intermediate group, "blowers" or "exhausters"; and where little or no pressure change is required, the name "fan" is most often employed. (In industrial applications the terms "fan" and "blower" are used, almost synonymously, to denote centrifugal machines which may operate against considerable pressure.) Again in order of decreasing pressure differential, the unit may be a positive-volume type (piston or rotary), a radial-flow or centrifugal machine, or an axial-flow (propeller) type.

Both centrifugal and propeller fans are used in ventilating and air-circulating applications. Prior to the development of the vaneaxial propeller type (a duct-mounted unit provided with guide vanes), centrifugal fans were used wherever substantial flow resistance was encountered as in ducted ventilation systems. Propeller fans, formerly limited mainly to use in free air, are now available throughout a performance range such that either a propeller or centrifugal unit may be used against low or moderate pressures.

Centrifugal Fans[22]

A centrifugal fan is made up of a rotor and a casing. The rotor is essentially similar to a paddle wheel, with a series of vanes or blades equally spaced around the circumference of a wheel and usually of an axial length greater than the radial length (Fig. 4-A). The wheel is mounted within a spiral casing (Fig. 4-B), the purpose of which is to convert as high as possible a proportion of the potential energy of velocity (measured as "velocity pressure") into kinetic energy ("static pressure"). Air flow is axial into the open center of the rotor, where the centrifugal effect of rotation changes it to a radial direction, and then is tangential from the casing outlet, so that discharge takes place in a plane perpendicular to the inlet direction. The operating characteristics of centrifugal fans, and thus the selection of the type to be used for a given application, vary with the conformation of the vanes. The simplest centrifugal fans, the "steel-plate" or "paddle-wheel" type, have six to twelve flat blades, each lying in a plane through the axis of rotation. To minimize shock and eddy losses at the entering (inner) edge of the blades the tangential velocity is kept low by small inside diameter and low rotational speed; radial acceleration, and thus pressure, is increased by long radial blade length and a progressive narrowing of

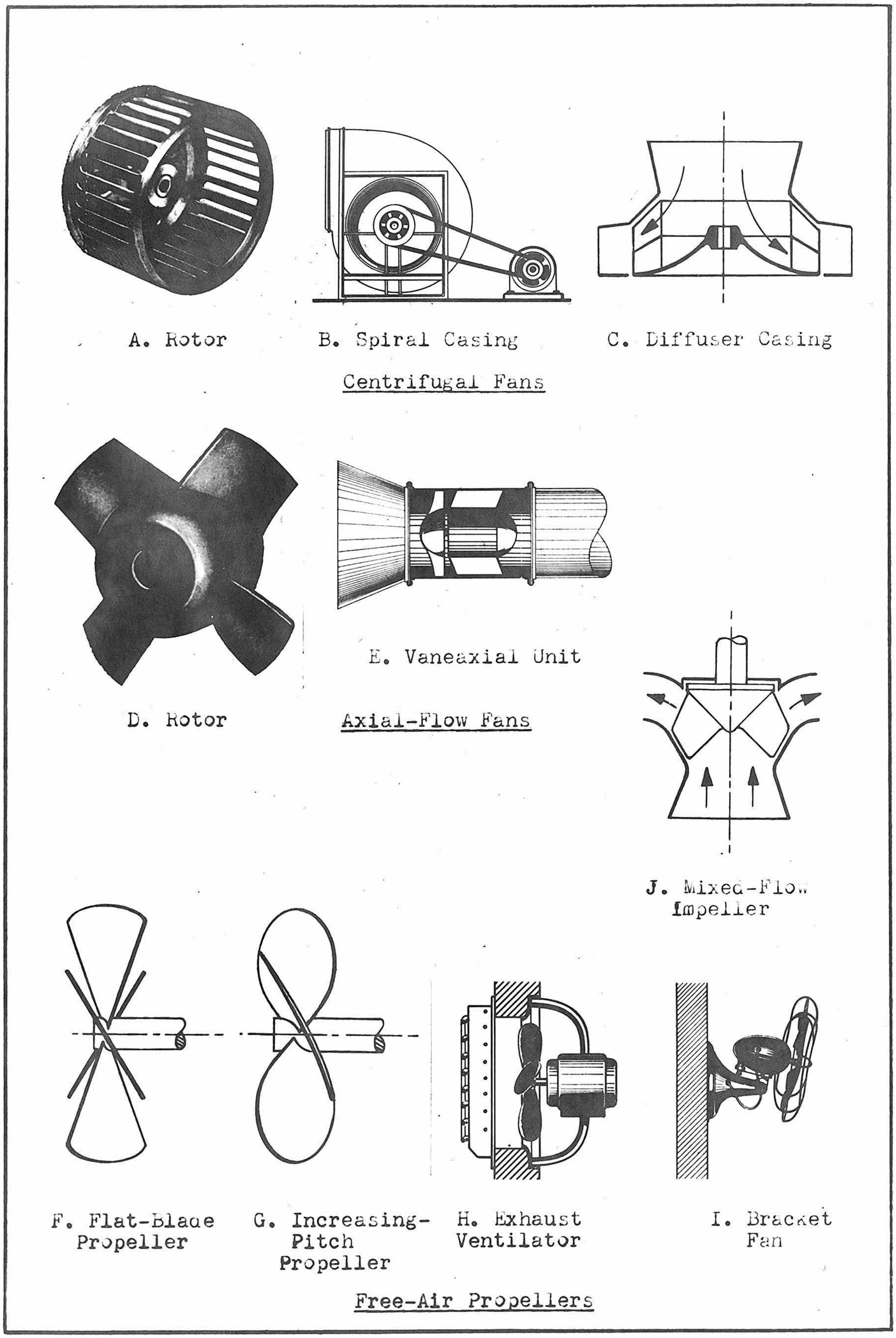


Figure 4. Impeller and Housing Types

the axial length from heel to tip. Such a wheel usually has a high ratio of diameter to depth as a result. The steel-plate fan is the simplest and oldest centrifugal type: its conception probably antedates the propeller fan, since it seems to have been developed by arranging a series of paddle fans on a shaft, and adding a casing primarily to give controlled discharge for its early use in mine ventilation.

Centrifugal wheels of more advanced design have from twenty to seventy forward- or backward-curved vanes, or they may use a combination of these two basic forms such as the radial tip or reversed curve. In all of these, the blade slants backward from the entering edge, thereby decreasing entrance shock and eddy losses and allowing higher rotative speeds such as are given by direct drive from an electric motor. In the forward-curved blade, a concave surface is presented in the direction of rotation: a wheel of this type will deliver a greater volume of gas than any other for a given size, and is sometimes referred to as a "volume" fan. The backward-curved fan has vanes with a concave leading face: at high speeds, higher pressures are attainable than with forward-curved wheels and the noise level is lower, but capacity is less for a given size and speed. Various other combinations of characteristics are obtainable from intermediate vane forms (Fig. 5).

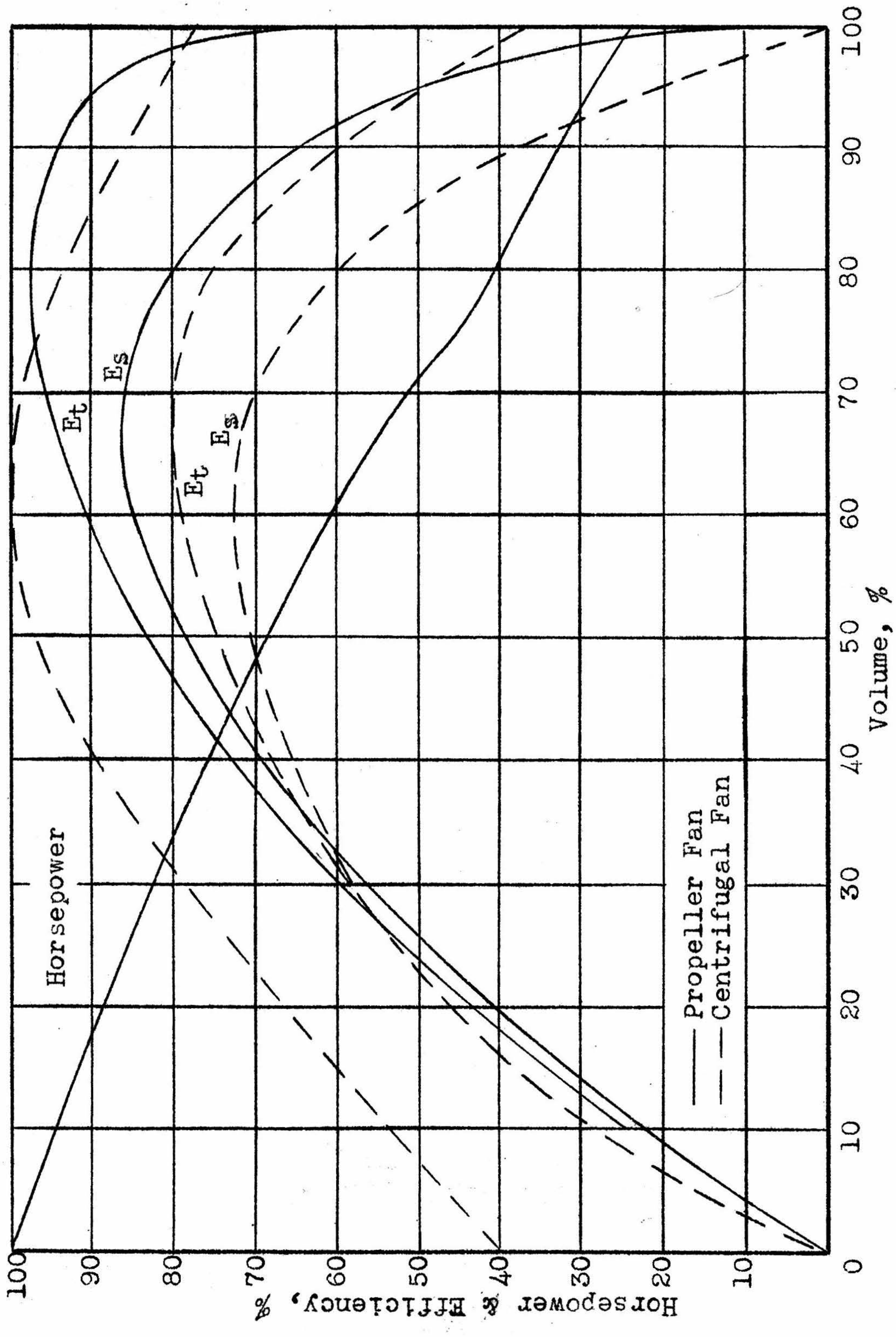


Figure 5. Comparative Performance Curves

Propeller Fans[23, 24, 25]

A propeller fan consists essentially of an inclined plane, wrapped spirally around a central axis or shaft. In practice, equal sectors of such a plane are displaced along the shaft so that their principal axes lie in a single plane perpendicular to the central axis to form blades equally spaced about it. If such blades are flat planes, each intersecting the central shaft at a fixed angle, a "disk" fan is formed (Fig. 4-F). In blades of this type the "pitch angle" (the angle at which a plane perpendicular to the shaft is intersected by a plane through two adjacent radial blade elements, measured in a plane parallel to the shaft) is at a maximum when measured at the radial blade element perpendicular to the shaft (corresponding to the line of intersection of the two planes defining the angle). The pitch angle is less when measured in any other plane parallel to the shaft, and so decreases with angular displacement to either side of the blade element defined above. Since most disk fans extend an equal distance to either side of the perpendicular radius, the angular pitch, and thus the speed of the gas particles propelled across the blade, increases from the leading edge to the center and then decreases to the trailing edge.

Because uniform gas velocity, or, more often, uniform acceleration, is desirable the plain flat blade is little used except in large diameters at low speeds, or where extreme simplicity of construction is desired. Blades formed as sectors of a helix (Fig. 4-G) inherently overcome the chief disadvantage of the flat blade in that they provide constant (rather than increasing-decreasing) pitch at any given radius. The helical surface may be generated in either of two ways. When the generating radius moves at constant angular and axial velocity, a surface will be formed which has the same lead, or axial advance per revolution, at every radial distance. If the angular velocity of the generatrix is again held constant, but axial speed varied in direct proportion to the radius, a surface with the same pitch angle at all radial distances will result. The former gives theoretically uniform gas velocity along the axis and therefore constant volume per unit of area (from Q = AV), so that total volume varies directly with the area or the square of the radius (App.III-C). The velocity for the latter varies directly with the radius, and volume with the cube of the radius. The plain helical (and flat) blades are of course the same in either direction of rotation and thus reversible. Most commercial blades are based on the constant-lead helix but have increasing pitch from the leading to the trailing edge, such as would be imparted by axial acceleration of the generatrix. By thus making the outlet pitch angle greater than that at the intake, the gas is accelerated gradually as it passes through the blade, making for improved efficiency.[26] If a constant-lead blade were carried to the central axis, where the pitch angle becomes 90°, the action of the central portion of the fan would be centrifugal. It has been found that to prevent the resulting radial flow, the blade should be discontinued where it reaches a maximum pitch angle of 45°.[27]

Comparison of Fan Types

The most apparent difference between the two principal fan types is in the flow character at the outlet, that is, axial discharge in the case of a propeller fan and tangential discharge for the centrifugal unit. While a centrifugal fan may be fitted with a diffusing casing (Fig. 4-C) if radial discharge is desired at the expense of static pressure, there is no practical method of obtaining axial discharge from this type. Reversing the direction of rotation of a forward-curved wheel in a diffusing casing would in effect merely convert it into a backward-curved type, without reversing flow direction to give axial discharge through the regular inlet. On the other hand, a propeller fan may be provided with turning vanes on the outlet side to deflect the flow from an axial to practically a radial direction. In addition it is possible to design a reversible propeller for use where flow in either direction is required.

From the engineering viewpoint where static efficiency is taken as the major criterion of fan performance, the centrifugal fan is admittedly superior. However, for a portable circulating unit operating in free air, without the pressure requirements imposed by ducting, the principal consideration is attainment of maximum capacity with minimum power and space requirements. In this case static pressure may properly account for a relatively small percentage of total pressure, and the use of a propeller fan is indicated. Other factors tending against the use of centrifugal units are: higher noise level; less convenient proportions of depth to diameter (although not necessarily greater volume for a given capacity); and higher cost, when the expense of the inherently required casing is added to that of the more complex impeller. The specific requirements of the present problem thus recommend some modification of the propeller-type fan.

While the foregoing discussion may seem unnecessarily detailed, it is included as a prelude to the calculations of Appendices II, III, IV, and V. These calculations, which provide rationalized formulae for the mathematical development of propeller fans, are felt to be an important part of this thesis. There are numerous sources detailing the theory of centrifugal fans, especially in regard to the mechanism of pressure development, and some material is available on the newer vaneaxial propeller units.[28] However, as far as the author has been able to ascertain, literature on the theoretical development of free-air propellers, with capacity determined on the basis of displacement, is nonexistent. It appears from meetings and correspondence with manufacturers of free-air propellers, and from articles written by their engineers,[29] that most of these fans have been developed empirically by bending, snipping, and twisting pieces of sheet metal until a satisfactory compromise is reached between capacity, power requirements, and noise level. It is hoped that the calculations developed here may form a useful groundwork for other designers, pending the results of further analytical and experimental work beyond the scope of this thesis.

T H E M A R K E T

In any design job, the designer must familiarize himself with existing solutions of similar problems. This is as true in the redesign of a current model as it is in cases where a new invention, or a new application or combination of known principles, is involved. While his research may have shown him theoretical methods which seem to recommend themselves, the experience of his client and of competing manufacturers will form a valuable guide in evaluating such ideas. In the highly competitive field of home appliances, it is unlikely that any valid idea has gone untried; those which are impractical or outmoded will generally be found in the record of discontinued items. The principles which may be applied to good advantage are usually demonstrated by their embodiment in products being successfully marketed, although new techniques or concepts may permit use of an idea previously considered infeasible.

An established manufacturer will be able to give the designer much of the required market information. This may include not only information such as specifications, manufacturing facilities and methods, and sales records of his own and sometimes of competitive products, but facts regarding the general characteristics of the trade as well. The latter comprise data on pricing policies, sales and advertising trends, mechanics of distribution, and so on, all of which are necessary to define completely the framework of the designer's operations. Material relative to consumer preferences and desires which is not available from the client or trade sources, as in the case of a new type of product, will have to be ascertained through a survey or similar means. In any case the designer must verify and supplement the data he assembles in this important preliminary phase of his work.

Competitive Products

The various types of electric fans and fan-forced convection heaters marketed during the last several years were studied in regard to their mechanical specifications, physical appearance and construction, and method of operation. This study included products of all major established manufacturers as well as of marginal firms whose contribution is principally an assembly operation. Both standard and special or novel types were covered, through inspection of products available locally, literature from manufacturers in all parts of the United States, and the advertising and "New Products" sections of American and British trade magazines.[30] Products of over one hundred firms,* most of them offering three or more models, were studied. Of these, the great majority were of the propeller type, only two portable models employing the cen-

- -

* Substantially all of the manufacturers of fans, fan-heaters, and propellers listed by the following sources were contacted, excepting those making only industrial centrifugal fans or installed ventilators, but including all members of the National Association of Fan Manufacturers (except makers of heavy-duty blowers only), the Propeller Fan Manufacturers' Association, and the Electric Fan Section of the National Electrical Manufacturers Association:

trifugal principle. Listing by method of mounting, following the standards of the National Electrical Manufacturers Association,[31] offers the best over-all distinction, although because of the innumerable possible combinations no method can be absolutely definitive.

Desk, Bracket, and Pedestal Fans--The most common type of unit, that usually referred to by the popular term "electric fan," has a propeller of two to four blades surrounded only by a wire guard without additional housing (Fig. 6-A). These sizes have the motor mounted directly on the base, and are intended for desk use or wall ("bracket") mounting (Fig. 4-I). The larger diameters are often available alternatively with a floor pedestal adjustable in height over a twenty-inch range between limits from two and one-half to six feet. The small inexpensive models are one-speed non-oscillating in operation, while the better fans have two or three speeds, with oscillation often variable

- -

Thomas' Register of American Manufacturers; Directory of Manufacturers, McGraw-Hill Pre-Filed Electrical Catalogs; Directory of Appliance and Radio Manufacturers, Electrical Merchandising News (McGraw-Hill); Index to Modern Equipment, Heating, Ventilating, and Air-Conditioning Guide (A.S.H.V.E.); List of Inspected Electrical Equipment, Underwriters' Laboratories, Inc.

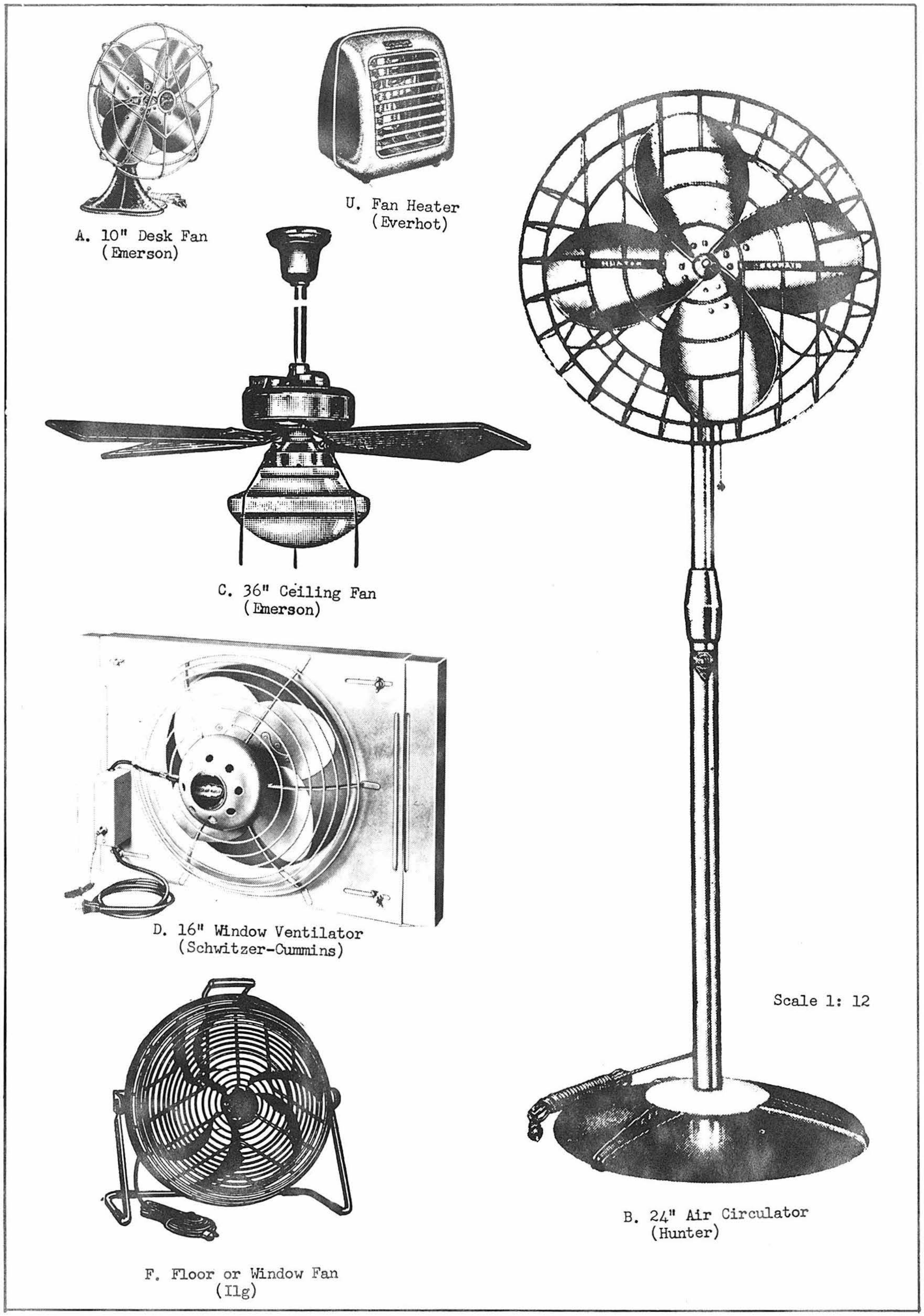
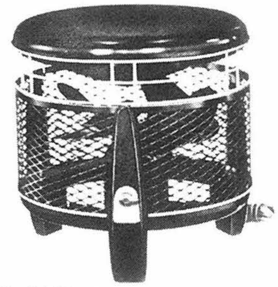


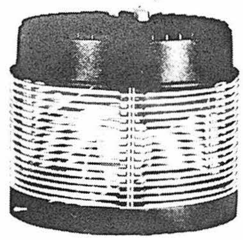
Figure 6-1.

Representative Present Day Fans

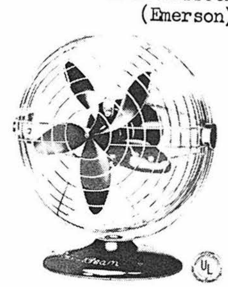
I. Straight-Flow Circulator with heating element (Mimar)



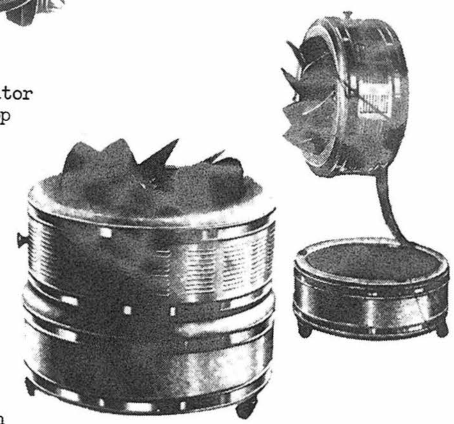
J. Diffusing Circulator with hassock top (Emerson)



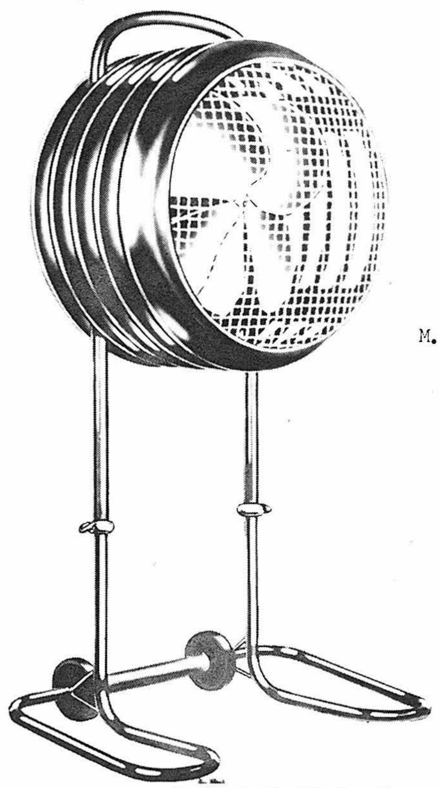
G. Vertical-Discharge Circulator with die-cast housing (Motionair)



N. Non-Oscillating Fan using plastic parts (Roto-Beam)

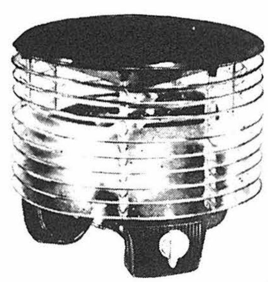


H. Floor-model Circulator with adjustable deflectors (Marsalis)

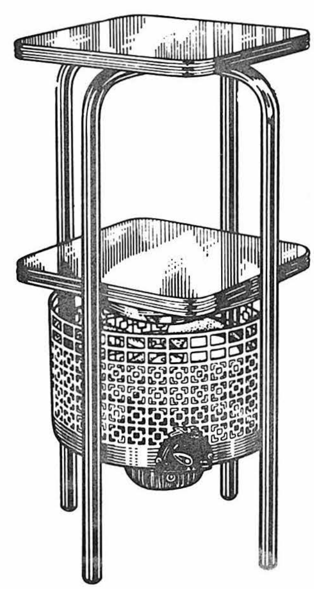


E. Portable Window Fan with injector rings (Westinghouse)

M. Straight-Flow Circulator with injector housing (Vornado)



K. Diffusing Circulator with all-plastic housing (Welch)

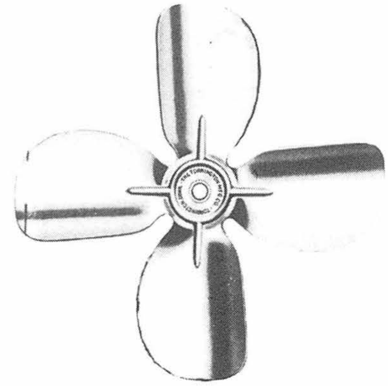


L. Diffusing Circulator combined with stand (Kisco)

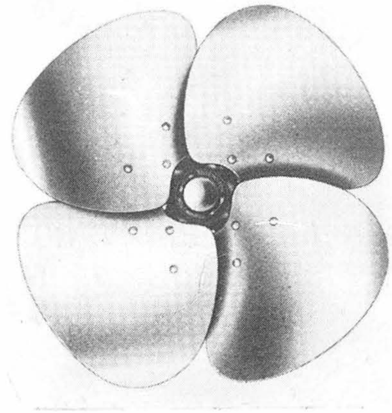
Scale 1: 12

Figure 6-2.

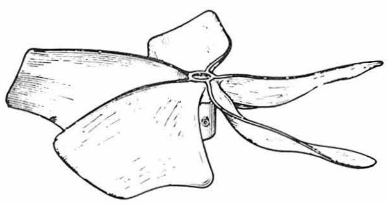
Representative Present Day Fans



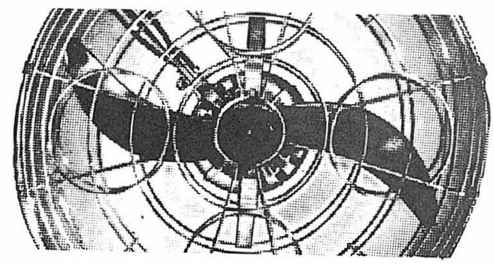
O. One-Piece Blade
(Torrington)



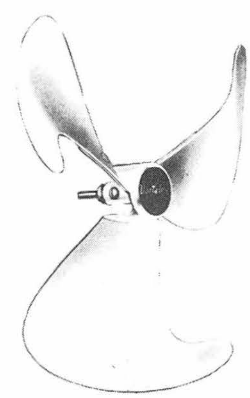
P. Overlapping-Blade
(Torrington)



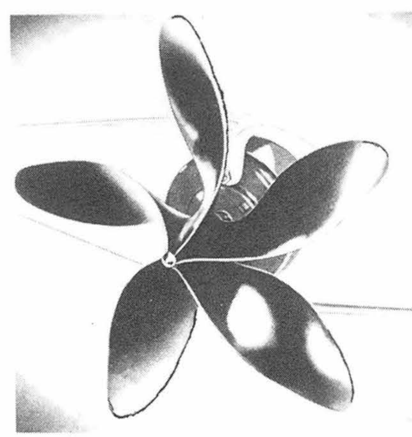
Q. Sand-Cast
(Airmaster)



R. Airplane-Type
(Mimms)



S. Quiet-Running
(General Electric)



T. Molded Plastic
(Roto-Beam)

Propellers

Figure 6 - 3.

Representative Present Day Fans

from zero to as much as ninety degrees. Vertical adjustment is usually between thirty degrees below horizontal and forty-five degrees above in desk or pedestal models (although a few can be directed vertically upward), and from horizontal to thirty degrees below when wall mounted. With the base fixed, as much as ninety degrees of horizontal adjustment may be provided. The lowest retail price found in 1949 was $2.98 for a single-speed eight-inch fan (non-oscillating), with high-quality twelve-inch units retailing at about $40.00.

Air Circulators--Larger fans designed mainly for use in commercial establishments are called "air circulators" (although this term is also being used to describe new floor types for home use, resulting in some confusion in the trade). The name properly covers units with propellers from twenty to thirty-six inches in diameter, most often on floor columns adjustable from five to eight feet in height (Fig. 6-B). Short columns for counter use and fittings for wall or ceiling mounting are also offered. These heavy-duty fans, weighing seventy-five pounds or more, are priced from $90.00 upward.

Ceiling Fans--The ceiling fan has been in common use since before the beginning of the century. When installed to blow downward in the open doorway of a store, it discourages the entrance of insects and so has earned the nickname "fly chaser." Standard diameters are thirty-six and fifty-two inches, furnishing from 3000 to 7000 cfm of low-velocity air. The four flat blades of wood or metal are sometimes adjustable in pitch to fifteen degrees in either direction, or a three-speed reversible motor may be furnished. A lighting fixture, mounted below the fan, is a common accessory and pull cords are customary for all controls (Fig. 6-C). After a long and honorable period as a standard store fixture the ceiling fan is obsolescent, as indicated by its omission from the 1949 and 1950 summaries of annual sales in "Electrical Merchandising" magazine. It is being replaced by installed ventilating or air-conditioning systems, and by air circulators which, although noisier, can move larger volumes of air at higher, more efficient velocities above the levels where objectionable drafts would otherwise result.

<u>Window Fans</u>--Fans designed for use at or near a window generally operate as exhaust ventilators (Fig. 6-D), although a few models are reversible. As "night-cooling fans" or air fresheners, they are popular where a permanent installation such as an attic fan is not practicable. Window ventilators with a frame adjustable to various window widths sometimes have glass or acrylic lights; others must be framed in. Some models are mounted on floor

stands and can also be used as circulating fans (Fig. 6-E, F). Propeller diameters range from eight to thirty inches, with as many as eight blades being used in the smaller sizes (a carryover from early ventilating fan practice). The recent rise in popularity of these units is due to increased appreciation of the benefits of forced ventilation, particularly for night cooling, and has lately brought several manufacturers of industrial ventilators into the home field.

Floor-model Circulators--The latest development which might be termed a basic fan type is that of low models designed to be placed on the floor so as to draw their supply from the cooler air at that level. Within this group there are two classes: those which discharge the air straight up for indirect circulation (Fig. 6-G, H), and those diffusing the air radially outward and upward (Fig. 6-J, K). Since in all these fans the air must be drawn in radially because of proximity to the floor, the propeller is a sufficient distance above the floor (usually about one foot) to allow ample intake area. The housing or guard extends several inches below the blades, often being carried down to the base or feet. The propeller is deeply pitched for maximum volume, and a ten- or twelve-inch diameter is the rule. None of the fans designed for floor use oscillates, and most are not adjustable as to direction although a few in the straight-outlet class may be tilted upon the base or rotated within pivots on the housing (Fig. 6-I). Few of the older manufacturers have as yet added such fans to their lines, evidently preferring to wait until demand is better established. Indeed, some of the more conservative firms have written that the floor-type fan (increasingly popular and widely approved as effective in operation) is merely a "passing fad" and will never replace the standard model which they have been producing basically unchanged for twenty years or more.

Straight-discharge fans for indirect circulation feature a high-velocity airstream which acts on the injector principle to draw in additional volume as it rises. On striking the ceiling, still at appreciable velocity, the stream is diffused outward and downward to impart motion to all the air in the room. With fan capacity properly suited to room volume, the residual velocity at occupied levels will provide thorough circulation without apparent drafts. Such units are also useful during the heating season for equalizing temperature distribution, moving the warm air at the ceiling down to raise the temperature several degrees at the occupied level. The effectiveness of the heating system is thereby increased to raise its apparent capacity or reduce fuel costs. The housing often incorporates a shroud ring around the blades, while the intake section may consist of injector rings, louvers in series, or a wire guard. One fan has an adjustable "diffuser" in the outlet. Retail prices in 1949 ranged from $20.00 to $100.00.

Diffusing circulators are similar in construction to non-adjustable units of the straight-flow class, but the upper or discharge end is closed by an inverted cone which diffuses the air. Ideally, flow should take place horizontally inward across the floor, up through the blades, and, after being deflected by the cone, outward in a horizontal plane. Practically, even if this complete reversal in flow direction could be accomplished, extensive friction losses and recirculation would result. The deflection effected beyond the propeller is thus limited to less than ninety degrees. Louvers or turning vanes are often placed above the level of the blades, and sometimes below, but even then discharge is seldom below thirty degrees from vertical. Since the cylindrical outlet of these fans is larger in area than the outlet of axial discharge units (which corresponds roughly to the blade circle area), velocities are usually considerably lower. It has been shown that stationary fans are more efficient than oscillating units,[32] and the diffusing circulator provides enough gentle air motion to give the same psychological feeling of cooling as the intermittent blast of the oscillator, without disturbing papers or the user himself. It is claimed that the user can sit comfortably close to or even on these units without experiencing objectionable drafts--at least in warm weather, when more sensible air flow is permissible. The flat cover placed over the cone is thus often advocated for use as a hassock or coffee table (Fig. 6-J). To further stress possible uses as furniture, diffusing fans have been offered with mirrored tops, or built under the lower shelf of a stand (Fig. 6-L); others may be obtained disguised as hardwood end tables. The desirability of these applications is questionable, since the other furnishings sure to be placed close to such pieces will definitely inhibit the free intake and discharge of air required for maximum effectiveness. Diffusing circulators range in price from $40.00 to $80.00, depending principally on whether large amounts of cabinetwork are involved.

Other Fan Types and Features--Many additional types, more or less unusual in construction and sometimes for special applications, are offered in the market, although seldom by old-line manufacturers whose products include a variety of home appliances in addition to fans.

The centrifugal principle was found to be used in only two portable units. One has an uncased rotor seven inches in diameter mounted on a vertical motor shaft, a removable deflector being provided as a means of directional control; operation is similar to that of radial-diffusing circulators. The other, using two five-inch rotors with motor between them, has conventional scroll casings within a wrapper. These discharge into a wide, shallow duct with its outlet in the wide side rather than in the end. It is intended for use as a circulator, blower, or window ventilator.

Most fan makers use propellers of the helical type, claiming various advantages for their own modifications. A great number of propellers have been empirically developed in efforts to attain higher efficiencies or quieter operation. Amazing forms incorporating apparently erratic bumps and hooks have resulted in some cases, although many valid contributions have been made (Fig. 6-O-T). Propellers such as the overlapping-blade type are suitable for use against moderate static pressures (Fig. 6-P). Others are designed for even velocity distribution across the diameter of the fan, or for greater capacity with the same diameter and power. Blades formed from steel or aluminum sheet are the rule in home fans, while sand-cast aluminum blades are used in some large air circulators (Fig. 6-Q). A few manufacturers of the latter still adhere to the use of airplane-type propellers with two narrow blades (Fig. 6-R), although these have the highest noise level and power requirements of any design. Plastic blades, either of laminated sheet or molded, are a relatively new development, and are quieter than metal because of the sound-damping qualities of the material. Because of their light weight, balancing is minimized, particularly where precision molding is employed (Fig. 6-T). One popular fan has flexible rubber blades, and no safety guard is required. Still another has loops of fabric ribbon in lieu of a propeller: while it is patently safe, its efficiency is questionable. Several makers feature the fact that their fans produce a "spiralling airstream" (an effect present to some extent in all propellers, indicative of a loss in efficiency). Others claim blades which move air effectively over their entire area, neglecting to state that such movement is mainly radial and of little value from the center out to the point where the pitch angle decreases to forty-five degrees.

Housing design is becoming an active field for the fan designer. While pressure fan casings have been brought to high efficiencies by theoretical analysis, no such basis seems to exist for free-air units. However, housings made up of injector rings or cones are now being incorporated in some propeller fans (Fig. 6-E, M). Two or more such rings, the first placed around the blade with the others behind it along the inlet axis, act to improve inlet conditions for increased capacity, while reducing recirculation and centrifugal-flow tendencies.

Fan-Heaters--An increasing number of electric heaters offer fan-forced convection as an improvement over the older radiant or natural-convection types (Fig. 6-U). A six- or eight-inch propeller is used to force about 100 cfm of air over the element. The fan shaft may be vertical with intake from the bottom of the metal case, or horizontal when the inlet is at the back. If the element operates at red heat it is placed so as to be plainly visible, to take advantage of the psychological effect of warmth which the glow is believed to produce. In this case, chromium-plated reflectors are often provided to direct the radiant heat, acting incidentally to increase the apparent size of the element by reflection. When a non-glowing element is used, it may be placed at the back of the case, often around the motor, and the reflectors are replaced by louvers to hide the interior. Most fan-heaters require that the fan always be used during heating to prevent overheating of the elements and motor, but at least one unit has a radiant element which may be operated independently. In several models the fan alone may be used at a higher speed for cooling; air volume is low however, because of the small propeller and the high combined air resistance of the louvers, element, guard, etc. The element is usually of nickel-chromium alloy, in wire coils or strips supported in the airstream by insulators. Glass-tube-enclosed coils or rods of conductive refractory material are employed where a self-supporting element is required, while industrial units use a finned metal tube encasing a resistance coil packed in magnesium oxide. Heating elements have been incorporated in some ten- and twelve-inch fans (Fig. 6-I), sometimes with a deflector or even the portion of the housing carrying the element removable when the fan alone is to be used. The motors for these larger models must be capable of extreme speed reduction so that air velocity and volume can be reduced to the capacity of the element. The most common rating for the elements of fan-heaters of all types is 1320 watts, with a maximum of 1650 watts based on standard 15-ampere household fusing at 110 volts. Fan-heaters are available from $10.00 to $40.00 (or more, for high-capacity 230-volt models).

Survey of Requirements and Preferences

The designer is limited and guided in the application of his knowledge by consideration of the three major groups with whose needs and desires he is concerned. These are the manufacturer, the distributor, and the consumer. Relations with the manufacturer are necessarily direct, since the designer must co-ordinate the specifications of his design with the type and extent of the facilities available for its production. The manufacturer's experience and general knowledge of the trade will supplement the designer's background to define the scope of the new design, by recording what has sold successfully in the past, and perhaps by indicating what may be advantageously marketed in the near future as well. Reputation and relative sales position in the field are factors in determining annual volume and unit price. These in turn affect materials, amount of tooling, and finish. To best simulate the designer-client relationship and to determine typical conditions which might thus be imposed, various questions were asked of the same manufacturers whose products were studied. While this multiplicity of information sources would not normally be available to the designer (except through catalogs), the method was considered the best suited to the present problem, where no specific manufacturer is involved.

Consumer Preferences and Complaints--A description of the largest-selling fan type will serve to indicate the basic requirements of the consumer. Manufacturers reported unanimously that the least expensive fan in their line was the "most popular model." Considering all firms in this branch of the appliance trade, covering the range from commercial circulators and ventilators through high-quality home fans to novelty items, the typical product is a ten-inch oscillating desk fan which may be wall mounted (Fig. 6-A). It has a die-cast base supporting a single-speed shaded pole motor, and supplies about 600 cfm of air at 1550 rpm, using a one-piece sheet aluminum propeller with four blades. The wire guard and incidental trim are chromium plated, with the base and motor housing finished in enamel. The most common color is "filing-cabinet" green, replacing black, the former standard. While bronze and ivory are also frequently offered, there is a growing trend toward more harmonious colors such as gray-green, warm gray, and tan. The most striking finish seen to any extent, and probably the least adaptable, is ice-blue wrinkle enamel, used on the very cheapest fans to attract customer attention. The current average retail price, as reported by "Electrical Merchandising" magazine, is about $18.50, and although this is for portable fans of all types including the more expensive floor-model circulators, it is typical of a medium-quality desk fan as described here.

The complaints most frequently heard from fan customers are listed here in order of importance as reported by manufacturers and distributors, with their qualifications:

1. Noisiness in operation;

2. Insufficient air output (usually due to the purchase of too small a fan in an effort to save money);

3. Appearance (the cheap fans are not as attractive as the expensive ones, and even the latter could often be greatly improved);

4. Motor overheating and subsequent failure (inexpensive fans without Underwriters' approval sometimes use a motor too small for the fan, or unsuitable for the practically continuous operation often exacted--more often the maker's explicit oiling directions are overlooked);

5. Availability of service (even though the manufacturer may give a one- to five-year service guarantee, his service agent is sometimes feckless or inaccessible);

6. Dirt collection on blades and motor (safety guards admittedly make the propellers of most fans difficult to clean, and a faulty oscillating-gear housing may leak oil to aggravate dust collection--however, servicemen testify that many housekeepers, including those who are particular about the condition of other furnishings, evidently expect electrical appliances to stay

clean and in operation with little or no care).

A listing of principal sales factors was also compiled in the same way. These are:

1. Price;
2. Appearance;
3. Manufacturer's reputation;
4. Performance;
5. Quiet operation;
6. Manufacturer's service policy, guarantee, etc.

The fact that three of the six major complaints have to do with performance, while price is by far the most compelling sales factor, is explained by the buying habits of the typical consumer. In his search for the most for his money he is usually confronted by a fan in operation: the fact that it blows air at all is proof of its "performance," i.e., that it performs; if, in addition, it oscillates so that he need not stay in the blast all the time but may be intermittently reassured by it, so much the better. When the fan is turned off it is examined to see if it is "modern" in appearance and if it has large blades, preferably four (since this is the greatest number usually offered). To verify this description it is necessary only to study the sales techniques recommended in the promotional literature of almost any aggressive fan company. As to the average buyer's criteria of performance, a survey conducted by the General Electric Company determined that a four-bladed propeller with large, relatively flat blades was preferred to the company's three-bladed design of high proven efficiency because the former "looked like" it could move more air. Apparently, in spite of constant efforts toward higher performance through scientific design, the public is not to be taken in. The fact that the less efficient fan of the buyer's choice is also noisier is not discovered until he uses it in the relative quiet of his home after the purchase.

Although appearance ranks second on the list of sales factors, the idea that their products could be more attractive does not seem to have occurred to many fan manufacturers as yet. The attitude of a large part of the industry is epitomized in a statement by an official of one of the major companies, who wrote, with considerable candor, "the matter of design on fans has not changed for several decades and there has been some talk on this subject." Such design efforts as are being made are evinced mainly in new floor-model circulators, rather than in improvement of the standard models of old-line manufacturers.

<u>Production Techniques and Facilities</u>--Further analysis of

the typical fan will serve to indicate the materials and techniques available to the designer, either in his client's own plant or from outside suppliers. The base and motor support are most often zinc die castings, although sand-cast iron, drawn steel, and molded plastic are also used. Castings and moldings are often purchased, although in some cases they are the firm's primary line. Plastic parts are also seen in control knobs, motor housings, and blades (Fig. 6-N). Most propellers are stamped aluminum, available in a wide variety of standard styles and sizes from blade specialists or other fan makers if production volume does not justify the required investment in special skills and tooling. Stamped and drawn steel parts, ranging in size from oscillator linkages to housings for window fans, are customarily made by the fan company. Propeller guards of formed steel wire, projection welded and plated in bright chromium, are almost universal. They are regarded as the best solution to the combined problems of sales appeal, ease of manufacture, maintenance, and function: while they are somewhat difficult to clean, they maintain their original appearance and provide good safety protection with little resistance to air flow.

The small amount of machining required is limited to a few screw-machine parts and to drilling holes and surfacing castings for assembly. It is usually possible to maintain ample clearances so that only in the motor is any degree of precision necessary. Motors are purchased unless they are required for an extensive line of fans or other appliances, or are themselves a primary product. (Many of the present fan manufacturers started as makers of small motors, who desired a relatively simple product as an assured outlet for their basic item.) Fan motors are almost always of the shaded-pole induction type, with two-pole (3000 rpm) construction being used in the smaller fans and fan-heaters, four-pole (1500 rpm) windings in ten- and twelve-inch fans, and six-pole (1000 rpm) in sixteen-inch fans. Because many consumers fail to follow oiling instructions, porous oil-impregnated bearings and a large oil supply are now featured in the better fan motors. Oscillating mechanisms may be assembled from purchased gears, or bought as a complete unit with the motor. Electrical components such as switches and speed controllers can readily be ordered to conform to the company's individual specifications if standard parts are not satisfactory.

Characteristics of the Trade

In most respects the market characteristics of all non-major home appliances (including fans, mixers, toasters, etc.) are similar in regard to distribution and pricing and service policies. However, the seasonal aspects of the demand for cooling and heating devices are reflected in fan and fan-heater sales and advertising.

Distribution and Pricing--The sales channel for small appliances is usually from the manufacturer through a wholesaler to retail dealers, although occasionally a jobber is involved. Small firms, particularly when newly established, often sell locally direct to retailers, and in other areas through manufacturers' agents with little or no warehouse stock. Some large companies, including those also making industrial equipment (sold direct or through jobbers and contractors), maintain district sales offices to work with their wholesalers as well as to advise industrial customers.

A wholesaler may be nationwide in scope or cover an entire geographic region of the United States, with several warehouse points, but most distributors operate in a single marketing area from one or two branches. Independent wholesalers may handle several makes of some items (often, however, in different price ranges), or may operate under agreement with a manufacturer to handle only his line, or a portion of it, in a given field. The General Electric Company and Westinghouse Electric Corporation, the two largest makers of home appliances, operate supply companies which wholesale the products of other manufacturers only when an item is not included in their own lines. The wholesaler usually has exclusive representation in his area for the products he distributes. This is seldom true of a retailer unless the product is sold under his own trade name (e.g., in the case of the large mail-order houses, auto-supply stores, and drug stores). However, the manufacturers whose products are in greatest demand commonly stipulate that other brands shall not be sold in competition with theirs, in which case an exclusive franchise may be given.

The pricing of fans, and of almost all other small appliances, follows a very uniform pattern at least as far back as the manufacturer's sales price. The dealer's net price for single purchases of an established product is usually 30% less than the fair-trade retail price, and about 35% less for larger purchases on a single order. A new item is often introduced at 40% below retail to attract dealers through higher profits, and the dealer's margin must eventually be brought into line by reducing the retail price rather than by raising the net price--in any case the dealer complains that his profit is being cut, and the period of initial price adjustments is very trying for the manufacturer. A wholesaler, with lower overhead costs than a dealer, may do business for from 5% to 20% of the net price, depending on the item and on his own sales position, but 10% to 15% is the rule. It is more difficult to generalize regarding the manufacturer's net profit, because the marginal producer must meet or better the wholesale prices of high-volume manufacturers. A 50% profit is probably not out of the question, on the basis of estimates made with the aid of quantity prices furnished by the manufacturers of motors and other components, while the minimum is about 7%. The pricing of an item retailing at $10.00 might then be broken down as follows: Dealer's net price, $6.00 to $7.00 (40% to 30% below list); Wholesaler's cost, $5.10 to $6.30 (15% to 10% below net); Manufacturer's total cost, $4.59 (10% to 27% net profit).

<u>Sales Characteristics</u>--Retail fan sales do not really begin until after the first several days of hot weather each year, and even dealers tend to delay stocking until that time. If the early part of the summer is cool, many purchases are deferred until the next season. In spite of this short and often undependable sales season, "Electrical Merchandising" magazine reported 1949 sales of 2,886,000 desk and

bracket fans compared with the prewar record of 1,985,000 units in 1941. This was in the face of a rise in average retail price from $6.90 for the ten years 1932-1941 to $18.50 in 1949 (Fig. 7). It is almost impossible to estimate the potential market for fans since commercial establishments form such a large portion of it, while other appliances can be figured on the basis of "percentage saturation in wired homes." However, some data are available to indicate the size of each manufacturer's potential share of annual sales.

There have been as many as one hundred and fifty different trade names in the fan market during a single period, but it is doubtful whether there are ever more than eighty manufacturers whose contribution to the total output of the industry is significant (considering that some of these make several brands for various outlets). Even assuming that the portable-fan market is divided equally among three hundred different models, an average of three or four per manufacturer, each would be entitled to annual sales of almost ten thousand units. Only some thirty appliance firms distribute fans nationally in any volume, and of these the dozen largest ones account for over one-half of all sales. One firm asserts that a single model, its cheapest ten-inch desk fan, accounts for one-fifth of the annual sales of this most popular type by all manu-

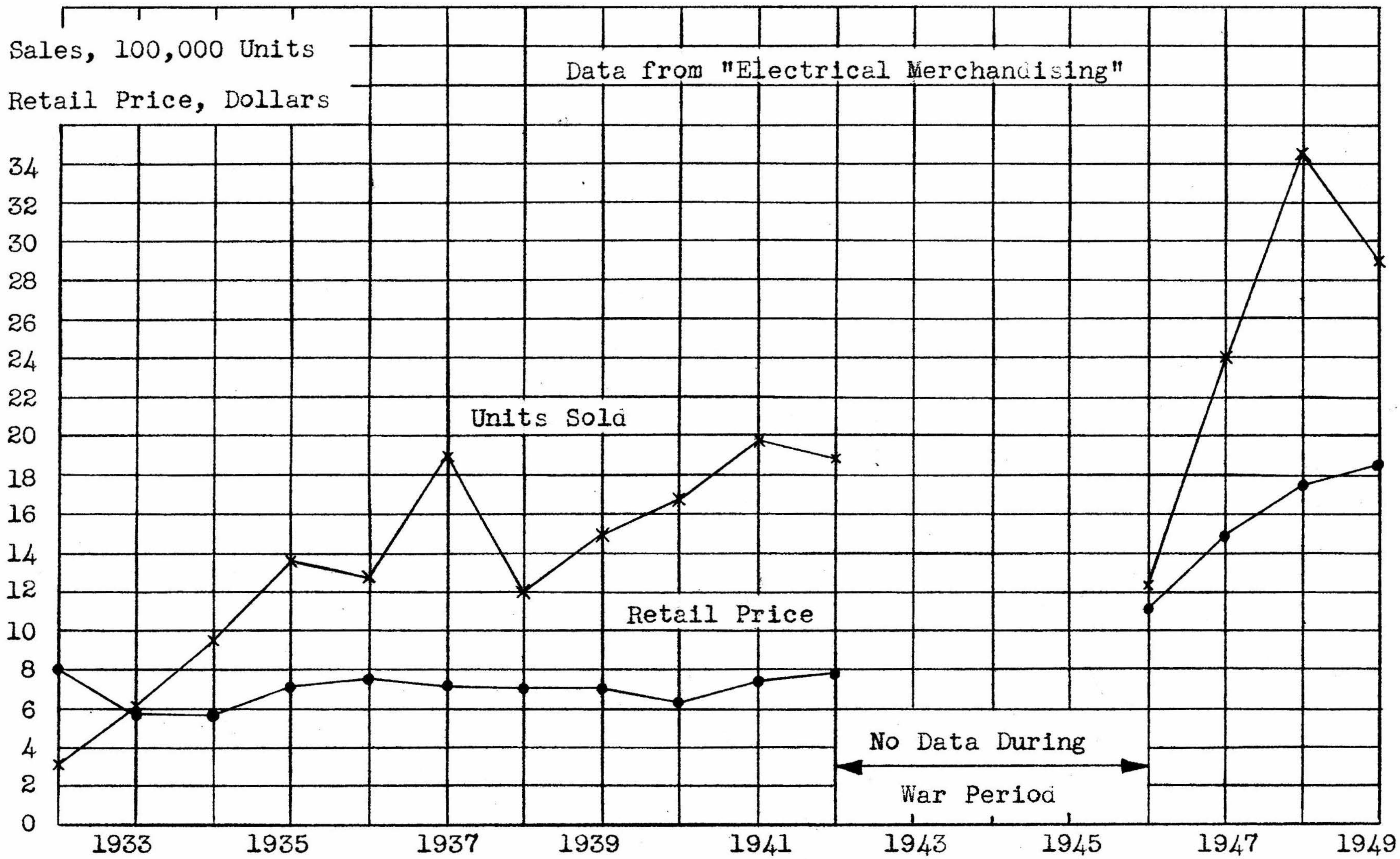


Figure 7.

Fan Sales and Average Retail Price, 1932-1949

(Portable Units Only)

facturers, or nearly one-tenth of total portable-fan sales.

About one million portable fans of other sizes and types are sold by the leading firms, which amounts to about sixteen thousand each of perhaps five models per firm. This figure includes the more expensive fans such as large floor-model circulators, and it is probable that medium-priced models may be safely estimated at twenty-five thousand annual units per firm. A new type in this range, backed by the name of a major company and by well-planned preliminary and continuing advertising, could reasonably be expected to find sales of ten thousand units a season until demand is well established.

Fan-heater sales are almost as seasonal in character as cooling fan sales, with about three-quarters of the purchases being made in the fall. The incidence of unseasonable cold weather at other periods, and the provision often made for using the fan feature alone in summer cooling, maintain some volume of business throughout the year. The suggestion of year-round utility partially counteracts the tendency to delay purchase when further warm weather is due. Fan-heaters, in common with other electric heating units, find a large part of their sales in sections of the country where central heating is not required, and are often used elsewhere at the beginning and end of the heating season when the central system is not in operation.

Sales of nearly one and one-half million electric space heaters of all types were reported by "Electrical Merchandising" magazine in its latest statistical issue, but separate figures for fan-heaters are not available. The average retail price given, $13.00, is about two dollars below that of low-priced fan-heaters. Firms desiring to capitalize on the demand for auxiliary heating equipment caused by wartime material shortages, and later by fuel shortages, raised the number of units produced to two million in 1946, or four times the prewar average. Although some of these companies have left the market, the value of portable electric heaters was effectively demonstrated to the public in spite of the many inferior products sold during that period. One out of seven families in homes with electric service now owns an electric heating unit. A survey made by the General Electric Company found increasing interest in fan-heaters, suggesting that the manufacturers of this product (40% of some seventy-five electric heater firms listed) are attaining their proportional share of sales.

<u>Sales and Advertising Trends</u>--Although 1949 sales of portable fans dropped twenty percent from the 1948 record of

three and one-half million units,* the reduction in total retail value was only fifteen percent because of the rise in average retail price from $17.50 to $18.50. The increasing consumer interest in higher-quality fans is due to a number of factors which have arisen within the last few years. One of the most important of these is the enlightened promotional policy which is now being followed by the more progressive manufacturers.

While most fan advertising is of course originated by the manufacturers, the efforts of magazine editors, home economists, and electric utility companies are being successfully enlisted to foster the more extensive use of fans for improved physical welfare. Some firms are employing independent laboratories to evaluate new or improved methods of fan use.[33] Although company research directed toward improved performance has not proved to be generally effective as a sales point (see "Survey" subsection above), the results of application research are finding acceptance by writers in consumer publications, and to an increasing extent by consumers themselves. Fan dealers are being

- -

* It should be noted that the available 1949 totals are estimates based on reports from relatively few firms (principally members of the NEMA), and additional data may cause a revision upward as it did the previous season.

trained to advise commercial customers in the efficient use of bracket fans and air circulators, with particular regard to proper location and non-oscillating operation. Many agencies are stressing the versatility of portable fans in the home, emphasizing the many ways in which a single unit can be applied.

Changes in the mode of living since the war, with lack of household help and the tremendous sales growth of installed major appliances as complementary factors, have made many Americans highly "fan conscious." Housewives doing their own kitchen and laundry work with the aid of automatic dishwashers and clothes washers and driers, are realizing that pleasanter, more comfortable conditions are attainable by using a fan to exhaust hot and moisture-laden air and cooking odors. Installed kitchen exhaust ventilators are increasingly popular, and are becoming the rule in new home construction. However, the owners of portable fans are finding them effective for the same purposes, with the additional advantage that they can be moved to any location desired. The use of fans in winter is also growing as consumers learn that fuel costs can be reduced when proper air circulation is provided. The even temperature distribution which results not only gives greater comfort but is better from the standpoint of health. Medical writers point out that a fan, correctly used for winter circulation, may be a factor in reducing the number of colds rather than in causing them, as was feared in the past. The incorporation of heating elements in several makes of straight-flow and diffusing circulators adds still more winter utility.

Another factor in fan sales is the increasing public familiarity with the benefits of air conditioning. As more commercial installations are made in temperate parts of the United States, the fan market too is expanding from its traditional centers in the East and South. There is a growing desire for fan-forced air circulation in the home, and for units which are fully adequate for the job. One manufacturer of portable air circulators, the O.A. Sutton Company, found that a surprising portion of the sales of their high-priced units was made to low-income families. Investigation showed that many manual laborers, particularly night workers who must sleep during the day, are demanding quality fans which will insure draft-free cooling during summer sleeping hours.

A complete list of fan advertising media must now include television, with broadcast "home demonstrations" making use of such new techniques as animation. Dealers are being trained by well-planned company films, instead of merely being supplied with promotional literature, so that they will be better equipped to advise customers. If the advertising and consumer-education programs are carried forward intelligently, and continue to be supplemented by the word-of-mouth advertising of satisfied users, the present upward sales trend will surely continue. It is perhaps not too optimistic to forecast that the electric fan will emerge in a new role, as the appliance with the greatest all-round utility of any in the home.

A N A L Y S I S   A N D   D E S I G N

Determination of the specific design approach to be followed is based on correlation of the user's actual requirements, plus sales and advertising trends, with the limitations imposed within the practical range of technical possibilities. These factors are, respectively, discussed in full in the preceding sections entitled "Atmospheric Factors in Human Comfort," "The Market," and "Mechanical Fans." Since the average user lacks knowledge of the criteria necessary in selection of the proper unit to best satisfy his comfort requirements under the conditions of its expected use, it is the responsibility of the designer to determine and design for optimum conditions, and to make recommendations as to the scope of operation of the completed design. His problem is similar to that of the engineer designing an air-conditioning installation, but more complex in that there are no fixed use conditions for a portable appliance which may be sold throughout the country to all types of consumers, for an endless variety of applications (and misapplications).

The designer cannot, of course, stop after a determination of "what is best for the consumer," but must integrate this with knowledge of "what is desired by the consumer." There is obviously no point in designing a unit which may functionally approach the ideal but does not attract and hold the potential buyer's primary interest at least until features of performance, construction, etc., can be demonstrated. Since this initial sensory appeal must be primarily visual, the form and appearance of the unit are of first importance. Its basic functions must be apparent to the eye, particularly in the case of a new or unfamiliar product, and these functions must be ones which the user desires, as the result either of experience or of the conditioning of advertising. It must not appear difficult to operate, nor too heavy or clumsy for portability. Shape and color must be selected with moderation if the unit is not to be exotic to the surroundings of the typical home or office--the distinction here is between modern and "moderne." To enhance, or at least sustain, the reputation of the manufacturer, the problems of convenience in actual use, dependability, ruggedness, and freedom from difficult or excessive maintenance must be met if the appliance is to live up to the impression made in the salesroom.

Formulation of the Practical Design

Studies of current sales and advertising trends and of the relative popularity of the various types of portable air circulators, ventilators, fan-forced convection heaters, and combination units have provided material on which the selection of the features of the design may be based. While the smaller oscillating fans remain the most popular, non-oscillating types embodying various improvements or new features are growing in public acceptance. Advertising and consumer education and experience are teaching the value of straight-flow fans placed on the floor, or mounted at a higher level, to set up continuous efficient circulation around a room in one direction, without oscillation. Winter uses with the unit directed upward or toward a heat source are also being featured. The employment of a circulating fan as a window exhaust ventilator is recommended, with the unit placed some distance inside the room for entrainment of a greater volume of air than is possible with the fan framed directly in the window. Diffusing circulators moving large volumes of air at low velocities are becoming popular, particularly for summer cooling where some direct sensible air flow is desired. They eliminate the draftiness caused by the intermittent blast of an oscillating fan, the only type of unit formerly available for this type of cooling. Fan-type heaters, in which forced convection distributes the heat more effectively than is possible in the older natural-convection and radiant types, are being increasingly used to supplement heating systems, and for local heating where no installed system is provided or where its operation is not desired during off-season cold periods.

The proposed design incorporates these features and uses insofar as possible in a single portable unit.

Size and Capacity of the Unit

In determination of the size of the unit, the upper limit is set by the requirement of convenient portability, while the capacity must be great enough to prove generally satisfactory throughout a wide variety of applications and surroundings. The most common use will probably be as an air circulator, which may have to care for a home living room or a small or medium-sized office space. In larger spaces any single unit would probably have to be so large as not to be easily movable, although a number of portable units could of course be used and would give more uniform circulation. The largest fans now marketed which are readily portable are propeller-type units with sixteen-inch blades, and their cubical space requirements and weight set the upper limits of the present design. These are thus about two and one-half cubic feet and twenty pounds, respectively (based on General Electric fan, Catalog No. FM16VI).

The volume of air needed for cooling a space by circulation cannot be readily calculated as in the case of an air-conditioning installation since no heat is extracted from the space, so this decision must be made on an empirical basis. The recommendations of the General Electric Company call for 1200 to 1500 cfm of fan capacity for each 5000 cubic feet of room volume, for multiple fan installations in large rooms.[34] These figures are based on the operation of several fans in combination, which may have a cumulative effect greater than the sum of their individual capacities, and are for rooms of 30,000 cubic feet and over with ceilings at least eleven feet in height. The above allowance of fan capacity per cubic foot is thus probably too low for a smaller space such as the average living room where the ratios of wall area and volume of obstructions (such as furniture) to room volume are considerably higher. The more conservative figure of 1500 cfm per 3000 cubic feet is taken here, and corresponds to one efficient twelve-inch fan (1200 cfm capacity) for a medium-sized living room (15 by 20 feet, 8-foot ceiling).

The limitations of a portable electric heating unit are rather more strictly fixed, being set by the capacity of household wiring. This is customarily taken as fifteen amperes, particularly in older construction, although certain local codes (such as that of New York) limit portable appliances to twelve amperes. The maximum rating of a fan-heater, including the motor, is thus between 1320 and 1650 watts on 110 volts. This amount of heat is, however, sufficient to hold the temperature of a medium-sized room about 30° F above the ambient in average uninsulated frame construction, when used as the sole source of heat (App. VI-C). A 1500-watt element is specific here, although this rating could easily be altered to comply with the regulations of a specific marketing territory.

Components of the Unit: Function and Operation

Combination of Straight and Diffused Flow--The first objective of this design, and the most radical departure from the features of most existing circulators, is the attainment of both straight and diffused flow in the same unit. This can be done by any one of three methods:

1. Provision of a removable deflector cone to be placed at the discharge side of a conventional fan;

2. Incorporation of attached adjustable louvers in the fan outlet;

3. Use of a housing with one end open and the other provided with a diffuser, the type of flow to be changed by reversing its direction.

The first of these alternatives requires a part which, if detachable, would need careful storing to prevent damage or loss when not in use, and would be inconvenient if not difficult to use in changing from one type of flow to the other. Such a deflector could possibly be attached to the fan by arms allowing it to be swung out of the discharge to the intake side when straight flow is desired. This would necessitate a much larger space for clearance during the conversion operation than would be dictated by the space requirements of the fan for either type of use. Another possibility would be to make the base in the form of a cone, pointing the unit straight down for diffused flow, but the full force of the airstream so close to the mounting surface would raise dust and create objectionable drafts in use on the floor, and on a desk or table would disturb papers and surrounding objects.

Adjustable deflecting louvers, used in some industrial unit heaters to direct the air, are employed as diffusers in one make of home circulator (Fig. 6-H). Where adjustments would be made frequently, as in a home unit, such louvers involve the structural and operational problems of furnishing sturdy, easily movable hinges and interlocking and adjusting devices in parts which would presumably be of sheet metal. More serious objections arise from the fact that the deflected flow would be of a swirling, tangential nature rather than radial, and that the airstream would not approach a plane perpendicular to the shaft until the louvers were nearly closed, being largely axial up to that point. Even with them wide open the normal spiral motion of the stream would be likely to cause noise in striking the louvers, and in any other position the flow area would be decreased and the air forced to turn through an abrupt angle. In consequence, increased resistance, higher velocity, greater noise, and reduced efficiency would result.

The incorporation of a fixed diffusing cone at one end of the fan in conjunction with reversible air flow would eliminate mechanical adjustments of the housing, for the convenience of the most inept user. In straight discharge, intake would be radial from the periphery of diffuser and impeller; in diffused flow, the cone, with the aid of additional louvering, would deflect the axial inflow to give radial discharge. The principal consideration in design of such a housing is to allow sufficient cylindrical area between the diffuser and the blades for unrestricted intake in straight flow, and for low velocity discharge in diffused flow. The overall dimensions of the housing would be no greater than those required with a detachable cone in place, and better performance could be attained than with movable deflecting vanes.

<u>Motor and Drive Mechanism</u>--Selecting the housing employing a fixed diffuser as the most advantageous from the standpoints of performance and user convenience, the problem of reversing the direction of air flow must next be met. Again three means are possible:

1. Rotating the motor and propeller assembly through 180° to point in the opposite direction within the housing;

2. Mechanically reversing the direction of propeller rotation with respect to the motor;

3. Reversing the motor electrically.

In the first method, with the motor and propeller assembly rotatable 180° on pivots to change the flow direction, a clear space would be needed within the housing at least equivalent to a sphere of the same diameter as the propeller. This minimum clearance would require the pivots to be placed in the central plane of the propeller, with supporting brackets carried around it from the housing to attachment points on the motor frame. The innermost point of the diffuser would then have to be at least one propeller diameter from the inside of the guard at the opposite end. If the housing itself is to be directionally adjustable, its pivots would be most conveniently placed coincident with those of the propeller assembly, and should be on a line nearly through the center of gravity of the housing and its contents. Since the weight would be concentrated in the motor, the pivot axis should pass through it. While this would simplify connection of the motor to the pivots, an even larger spherical space would be required for pivoting within the housing, because the axis would no longer pass through the plane of the propeller. This system has the advantage that a propeller of conventional (nonreversible) design could be used, but the large overall size needed merely for the purpose of reversing, as opposed to operation, is undesirable. Additional complexities arise in the design of a simple, convenient, and foolproof indexing-and-locking device to assure that the propeller is secured positively in the desired operating position. When these are considered, this method is seen to be impractical for use in a portable appliance.

Perhaps the simplest mechanical reversing device which might be used is an adaptation of the "rim drive" used for electric phonograph turntables, with a friction drive wheel mounted on the motor shaft and arranged to contact optionally the inner or outer face of a ring on the propeller. With the drive ring and motor at the fan periphery the unit would be unbalanced, however, and the fastest induction motor (3450 rpm) would require a drive wheel more than five inches in diameter to operate a twelve-inch fan at 1500 rpm. Even with the motor placed near the fan hub to utilize a smaller drive wheel, and for somewhat better weight distribution, a large portion of the flow area would be blocked by the motor, because of the eccentric mounting required and the combined longitudinal and pivotal motion occasioned in shifting. The propeller would require separate bearings instead of utilizing those of the motor as in direct-drive units, and greater rigidity (and weight) properly to support the drive ring. Although this system might be adapted to low-speed stationary fans, it is obviously not well suited to portable types.

Of the possible mechanical systems, reversible gearing offers the most promise. While most simple geared reversing mechanisms are designed with the driven shaft eccentric to the input shaft, three designs have been studied in which straight-through drive is maintained. In all of them, friction discs and wheels are used rather than true gears, to reduce lubrication requirements and simplify shifting. The first is schematically similar to the differential gearing used in computing machines, and the spider is clutched to either the output "gear" or the housing by means of a splined slider and clevis to control the direction of rotation. The second uses oppositely placed pairs of rollers and bevelled discs, moved in and out by means of a parallelogram linkage so that one pair or the other contacts appropriately formed surfaces of discs on the motor and fan shafts. The controlling mechanism would impose no drag on the shafts (as a clevis would). In the third proposal, the fan shaft is moved axially a short distance in either direction, to bring a disc at the motor end in contact with either a set of reversing rollers or a clutch surface on the motor shaft itself. In this case the fan thrust might be utilized to support at least a part of the blade weight and to augment the clutching force, relieving some of the frictional drag in the required clevis.

With any of the latter three devices the direction of blade rotation could be readily reversed (even in motion, although at the risk of excessive wear on the friction surfaces) by means of an easily operated control at the outside of the housing moving a bell crank or other simple linkage. The principal problems are their possibly excessive size and cost, and the difficulty in obtaining adequate bearing length for the output (fan) shaft. However, they are no more complex than the geared oscillating mechanisms provided even in inexpensive fans, and do not require the fine tolerances of such mechanisms even though operating at higher speeds, since wear of bearings and friction surfaces can be taken up by spring loading. In spite of their advantages, none of them could be definitely recommended for incorporation in the present design without complete mathematical and experimental investigation to determine the forces involved, and the corresponding materials and contruction required to give a long-lived unit of the smallest dimensions and least cost. The amount of work involved in such a study would form a separate project, and is beyond the scope of this thesis.

The type of drive selected, then, is the electrically reversible motor. While these are more expensive than non-reversible motors, the simplicity of operation of this method, which requires only a switch control, recommends it for the use of the average person.

Series motors can be wound for reversibility, but because they are generally built for higher operating speeds than induction motors, and require maintenance of brushes and commutator and of the filter capacitor usually included in the circuit (to reduce commutator sparking and radio interference), their use in household fans has been almost entirely discontinued.

The induction motors used almost universally are of the shaded-pole type, the simplest and cheapest of small motors. These are well suited to speed control to 60% of rated or lower speed, but considered inherently non-reversible. Where a fan application requires a reversible motor, a permanent capacitor, capacitor-start, or split-phase type is used, the latter two incorporating centrifugal switches to cut out the starting winding at about 75% of synchronous speed. The permanent split capacitor (also called capacitor-run) motor, which requires no cutout, recommends itself further in that it is available with only three leads, rather than the four required by the others. This feature is valuable in the present case where all wiring must pass through a relatively small passage from the housing to the base. Good speed regulation is obtainable down to about 60% of rated speed, providing an adequate range for cooling-fan operation.

For use with a 1500-watt heating element, it is desirable to reduce the capacity (and velocity) from the design maximum of 1200 cfm to the lowest value obtainable, in order to allow an air temperature rise great enough to obviate a possible cooling effect from the air motion. In this case, the motor speed should be as low as practicable, preferably 25% or less of the full-load speed. The shaded-pole motor best satisfies this requirement, since it is capable of extreme speed variation. The provision of a high-resistance rotor improves the speed regulation characteristics in the low range. While not a standard construction, fractional horsepower shaded-pole motors have been built with wound shading coils in pairs, one set or the other being short-circuited to reverse the motor. The reversing switch for this motor may be placed on the end cover so that the three leads required need not be led to the base with the motor and heater power wiring.

Either the reversible shaded-pole motor or the three-lead permanent split capacitor type is thus specified, dependent upon whether or not the unit includes a heating element. While exact specifications of such motors are not available, both could definitely be manufactured on order, at an estimated cost comparable with that of a standard split-phase motor. The full-load speed of either will be about the same as for a shaded-pole type, that is, approximately 1500 rpm for a four-pole motor. Of the standard motor pole numbers, four is the optimum for a 1200-cfm fan, since the 3000 rpm of a two-pole motor is too high for quiet propeller operation and the 1000 rpm of a six-pole is too low for the desired combination of fan capacity and size (App. II-E). The calculated horsepower at 1500 rpm, based on a helical blade, is .029 for 1200 cfm, and should not exceed .056 even if the actual output should prove as high as 1500 cfm (App. V-C). The next lower standard rating, one-twentieth horsepower, will give adequate power, as shown by examination of test data compiled by a leading propeller manufacturer.[35] The high cost of ball bearings is not justified in a fan-duty motor of this size; instead, oil-impregnated porous-bronze bearings, surrounded by a felt-packed oil reservoir, are specified. These must be designed for both radial and thrust loading, since the motor is to be capable of operating in any position. The addition of an internal cooling fan might be found desirable.

<u>The Propeller</u>--In a field already notably short of design data, literature on the design of reversible fans was found to be nonexistent, doubtless because of the few ap-

plications in which they are required. They are used chiefly in sidewall and window ventilators, and manufacturers' correspondence and examination of these units disclosed them to be almost invariably of the flat-blade type, perhaps with some such slight modification as a bend along one edge or at part of the tip. While these blades admittedly offer equal efficiency in both directions, this efficiency is low compared to that of helical blades, particularly of the constant-pitch type.

Since the helical propeller is best for axial discharge while the centrifugal rotor in combination with a diffuser gives radial discharge, perhaps the ideal propeller for the present design is a combination of, or compromise between, these two extremes. Mixed-flow liquid pumps are built in which the change in direction between the axial inlet and the semi-radial outlet is accomplished within the vanes (Fig. 4-J). The outlet diameter is considerably greater than the inlet diameter, however, and it is estimated that for a free-air fan built on the same principle, the rotor would have to be some eighteen inches in diameter and nine inches deep for low-velocity discharge at 1200 cfm. While this is too large from the standpoints of safety (because of the high peripheral speed), and convenience, it suggests a principle which may be employed to advantage: that is, using the blades to supplement the action of the diffuser in changing the flow direction, thus simplifying the latter by reducing the requirements imposed on it.

The propeller proposed here is a modification of the constant-lead helix, and might be termed a "semi-mixed-flow" type. In order to accomplish a part of the required change in flow (from axial to radial, and vice versa), the blades are generated within a cone, instead of in a cylinder as is a true helix (Fig. 8). The generating line, rotating at a constant angular velocity, moves perpendicular to the surface of the cone rather than to its axis, at an axial rate varying directly with the radius. The line generated on the surface of any coaxial cone with the same apex is thus a logarithmic spiral instead of a helix, and cuts all apical elements of any such cone at a constant angle. While the lead of the spiral per revolution decreases toward the apex, it is constant along the generatrix in any position. A pitch ratio of unity is selected, corresponding to a moderately high nominal pitch of 25.5°. This pitch is midway in the range of practical blade angles (from 15° to 35°), and offers good volume without excessive power requirements or great axial length. The central portion of the propeller, to the radius where the blade angle reaches 45° ($\tan^{-1} 1$ at r/π), is filled by the frustum-shaped hub, since the action of the blades

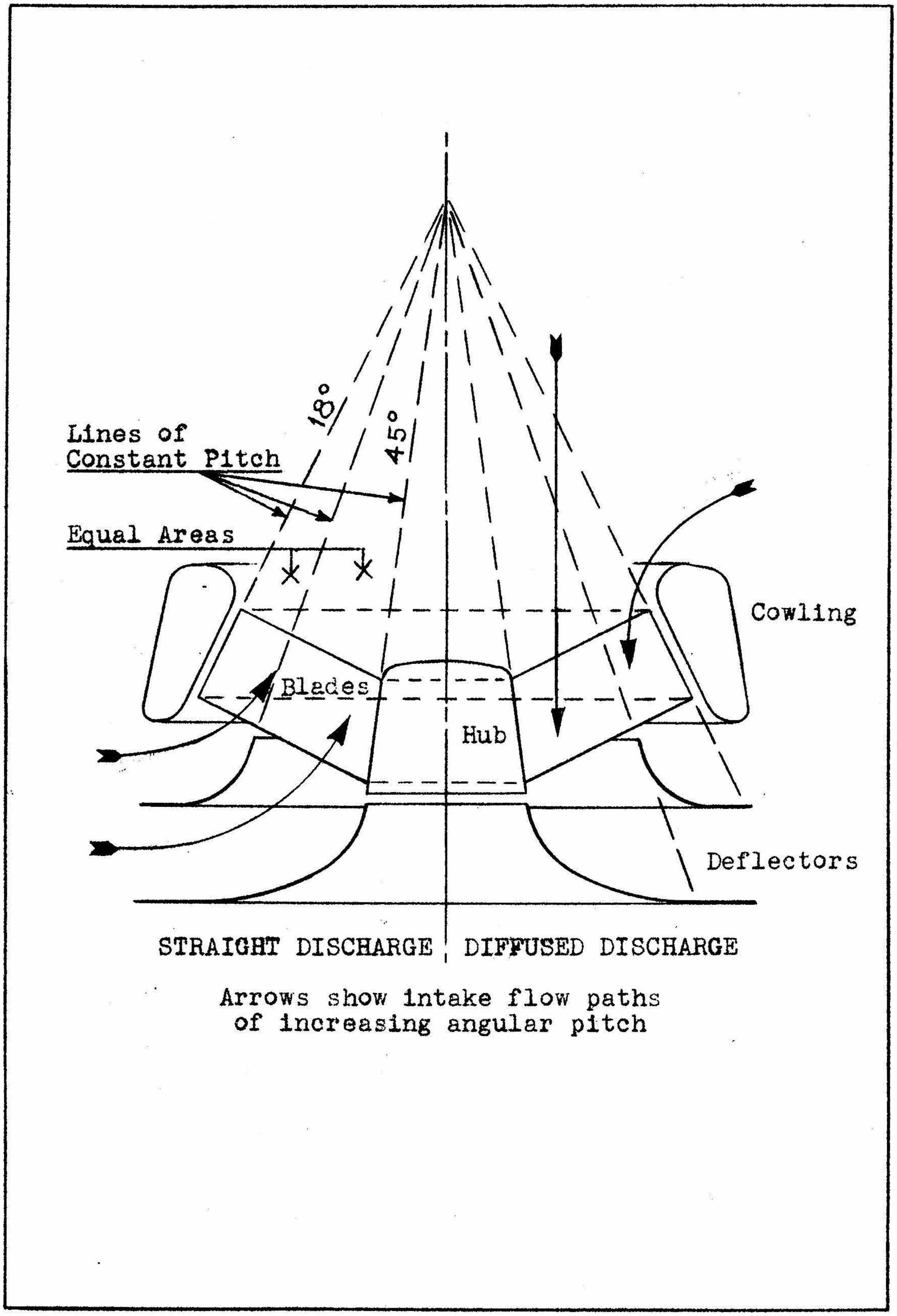


Figure 8.

Propeller and Diffuser Layout and Operation within this region would be predominately centrifugal. The motor is placed partially within the hub for space economy and better weight distribution. It is expected that in straight flow the stream passing around the hub will tend to converge near the apex of the cone, largely eliminating the "dead" area present at the center of ordinary propellers. The angle at the blade tip is 17.7° ($\tan^{-1} 1/\pi$) for the chosen pitch. The ratio of cone diameter to height is also unity, giving a large apex angle (about 53°) without placing the point of flow convergence so close to the blade as to cause excessive turbulence. The blade edges are taken along generatrices, maintaining constant pitch along the edges and bringing the central portion of the propeller close to the base of the diffuser, to shorten the flow path. Straight edges are specified, since each is both a leading and a trailing edge. A slight inward curvature of the edges might experimentally be found desirable if the blade proves noisy, although efficient (and thus quiet) operation is expected. The blade tips are slightly rounded to reduce possible vortex noise and to improve appearance.

The dimensions of the propeller are calculated for a theoretical displacement of 1200 cfm at 1500 rpm. (App. V-A). While either three or four blades could be used, the latter number is chosen, with a projected area ratio of 100%, to minimize the axial depth of the propeller. The large area and greater number of blades are also preferred by the consumer. The actual displacement might be somewhat higher than the theoretical, perhaps by as much as 25% (as it is in good commercial blades--see App. IV-C). This assumption is based on the probable flow through the blades, considering the increase in angle from tip to hub. In radial flow, the incoming air will continue to flow axially or even inwardly (cf. flow through orifices) at least a part of the way through the propeller, so that the effective pitch angle increases and the air is accelerated as in a well designed standard propeller; in axial flow, the same effect will be produced as the radially entering air crosses the blades (Fig. 8). While this action will be less significant at reduced velocities, the flow will never follow an inefficient path of decreasing angularity since the pitch ratio and hub diameter are designed for minimal centrifugal action even at full speed.

<u>The Housing</u>--The two major components of the housing are the propeller cowling and the diffuser. The cowling is toroidal in form, and provides a rounded entrance in both directions of flow. While the entrance radius must be held rather small in relation to the inner diameter of the cowling, in order to maintain reasonable overall dimensions, the flow conditions will be between those for a

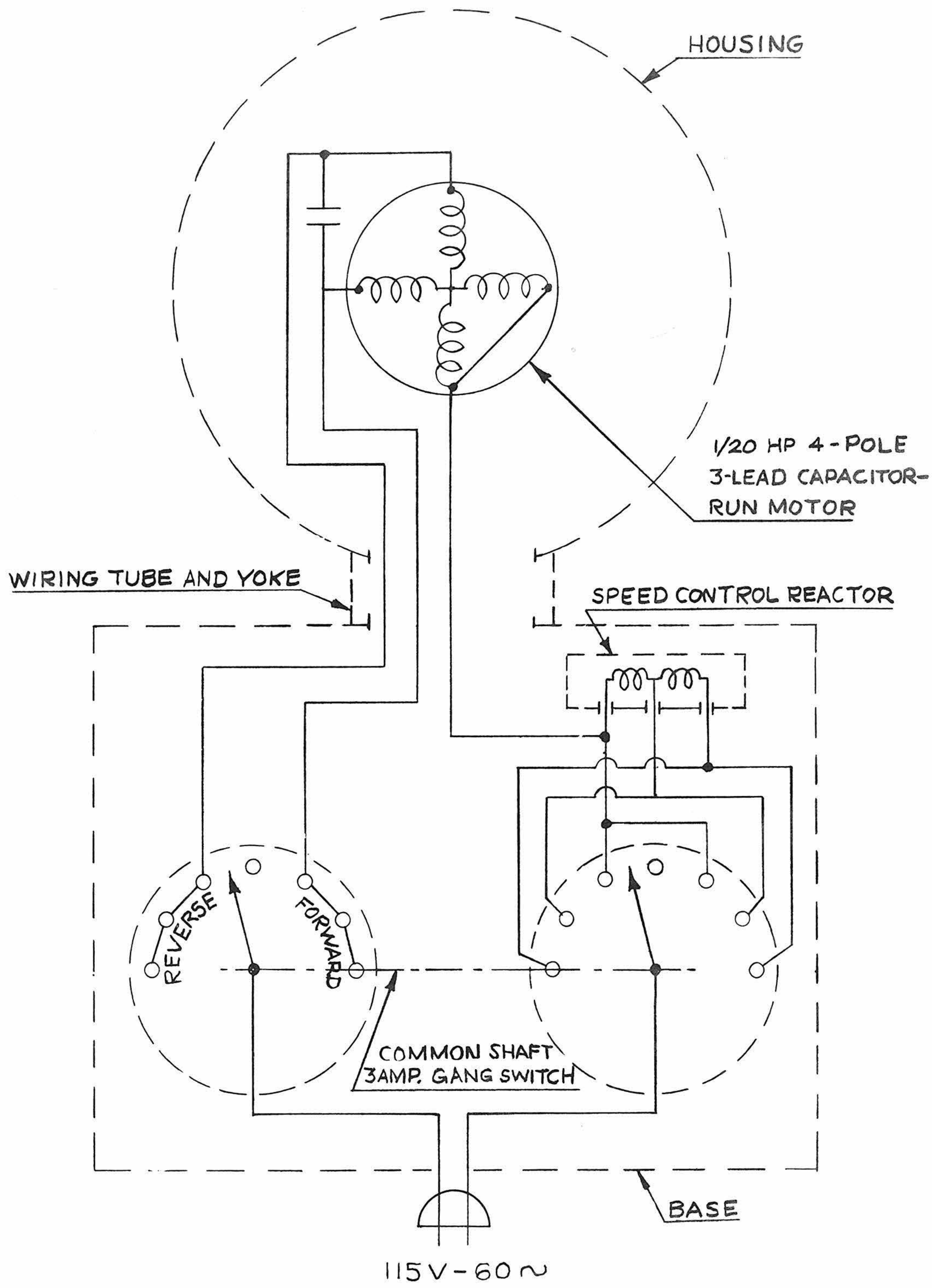
bell-mouth and a sharp-edge orifice, with a corresponding high discharge coefficient and low resistance. The inner surface of the cowling is parallel to the surface of the cone defining the blade periphery, and as close to it as assembly tolerances will permit in order to reduce vortex noise and recirculation within the housing. The outer surface is shaped to augment peripheral intake, and to carry out the form of the complete housing.

The opposite end of the housing is closed by a diffusing cone (termed the inner deflector) which curves outward and away from the propeller. It begins near the hub and tangent to it and ends tangent to a plane which is perpendicular to the axis and a sufficient distance from it for adequate intake area in straight flow, and low-velocity discharge in diffused flow. A supplemental deflecting ring (the outer deflector), also curved in section, is placed between the shroud and the diffuser, extending from a tangent to an imaginary cone dividing the blade into equal concentric areas, to a second perpendicular plane halving the peripheral area between the shroud and the inner deflector. The flow through the fan is thus divided into two streams whose areas are approximately equal at all corresponding sections, resulting in a uniform velocity distribution which is particularly desirable in diffused discharge. The outside diameters of the deflectors are made large enough to deflect all portions of the most widely divergent air stream which may pass through the propeller in diffused flow (Fig. 8). By carrying these components close to the blades, the entire airstream is deflected without the need for intermediate louver rings.

The motor is mounted inside the inner deflector and extends into the hub. The diffuser end of the housing is covered to conceal the motor and heating element wiring, and to complete the form of the housing. An annular space between the inner deflector and the cover allows the passage of cooling air over the motor, in conjunction with ports in the face of the hub. Grilles are provided at the open end of the cowling and around the housing between the cowling and the end cover for protection from the propeller and heating element.

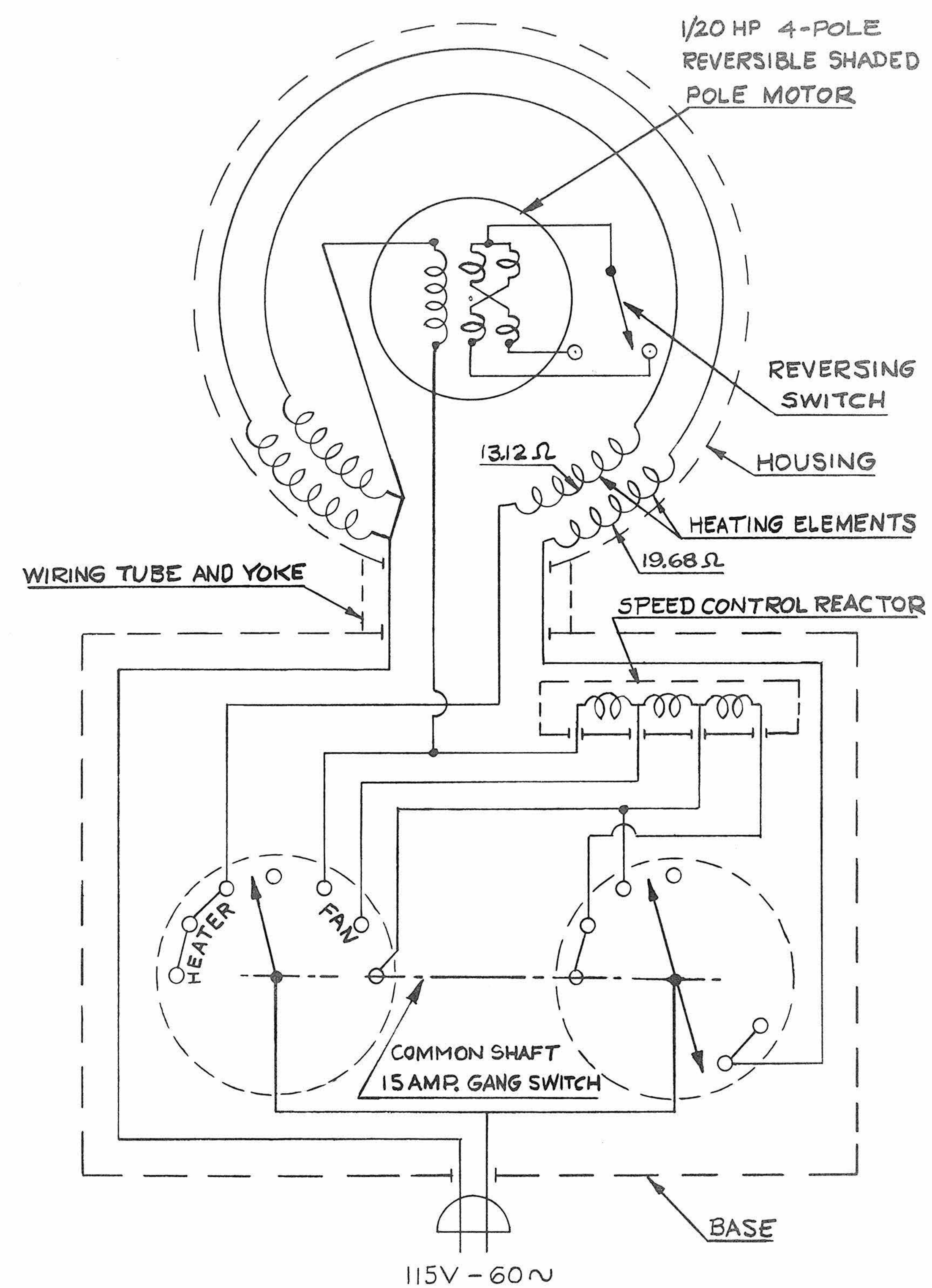
Heating Element--For proper heating of the air, the heat source should be distributed as uniformly as possible within the airstream, and the air velocity reduced to the lowest value possible by operating the motor at its minimum speed. It would perhaps be desirable to have the element on the discharge side of the propeller, at least in straight flow, in order to utilize the blades as heat reflectors. However, the difficulty in supporting it properly from the cowling, coupled with the unattractive appearance the mountings, insulators, and element would present in this readily visible location, and the extended grille required for the user's protection, more than cancel any advantage. The deflectors, in comparison, offer convenient, well protected mounting surfaces, and a portion of the element may be mounted on each of these for proper heat distribution throughout the flow area. Rigid elements would be the easiest to mount and in general give long service, since oxidation of the resistance units is low in these types. However, the cost of the finned-tube metal-enclosed type, used in industrial fan-heaters, is too high (over $15.00) for consideration, and the glass-enclosed and refractory types are too fragile for use in a highly portable appliance. An open helical coil of resistance wire, supported in insulators, is thus chosen, with the conservative operating temperature of 1100° F specified to prolong its life. A two-part element is used, one section being mounted on the outer surface of each deflector. The maximum heat output of 1500 watts is obtained by connecting the elements in parallel, with a low heat of 900 watts using only the inner portion. There is no simple and practical way to wire the element so as to heat both parts of the airstream on low heat, but this is not considered a serious disadvantage since only the lowest air velocity will be used with the 900-watt element, and a somewhat larger portion of the flow will be through the inner section of the propeller and housing at this speed. The two parts of the flow will mix shortly after leaving the housing in any case, to further assure equalization of velocity and temperature.

Control--Provision must be made for varying the motor speed to cover the range of use conditions, and to conform with consumer demand for a choice of several speeds in quality fans of this size. This can be done by winding the motor so that the number of poles may be changed, but while this maintains optimum characteristics, it is costly and limited in practice to two speeds. A reactance-type controller, which is relatively inexpensive and has a negligible effect on performance, is the alternative selected and is incorporated in the motor circuit to give three speeds below the maximum of 1500 rpm, down to the minimum practicable (about 1/4 of maximum of 1500 rpm). To protect both the unit and the user, excessive temperature rise is prevented by operation of the fan whenever the heating element is energized, at its lowest speed on low heat, and at either low or first intermediate speed on high heat (Fig. 9-A, B). At low speed, the combination of minimum volume (about 400 cfm) and maximum heat output will raise the air temperature 16° F between inlet and outlet under average conditions (App. VI-B). The fan alone may be used at first or second intermediate



PORTABLE AIR CIRCULATOR
SCHEMATIC WIRING DIAGRAM-FAN ONLY

Figure 9-A



PORTABLE AIR CIRCULATOR
SCHEMATIC WIRING DIAGRAM-FAN & HEATER

Figure 9-B

speed or at high speed, but for air circulation in the average home will probably find its greatest utility at the second speed of about 1200 rpm, giving a volume of approximately 1000 cfm. It is expected that the maximum capacity will be used in small homes mainly for exhaust ventilation. No built-in thermal overload protection is provided, since a temperature rise of about 400° F above the designed operating temperature will overload even a 20-ampere fuse, larger than should be installed in the circuit to which the fan is connected, without causing the element to burn out. The motor is afforded protection from overheating by the inner deflector, which acts as a heat-reflecting baffle, as well as by the provision made for air flow directly over it.

The variety of uses for which this fan is intended requires a greater degree of directional adjustment than is usually offered (Fig. 10). With the unit on the floor, straight flow should be obtainable at any angle between horizontal and vertical, and at a slightly greater angle of depression when placed on a table or otherwise raised. In cooling with diffused flow, the air should issue horizontally some distance above the floor to avoid drafts and most effectively to cover the occupied zone; to attain this height without greatly increasing the overall height of the unit, the intake must be at the bottom.

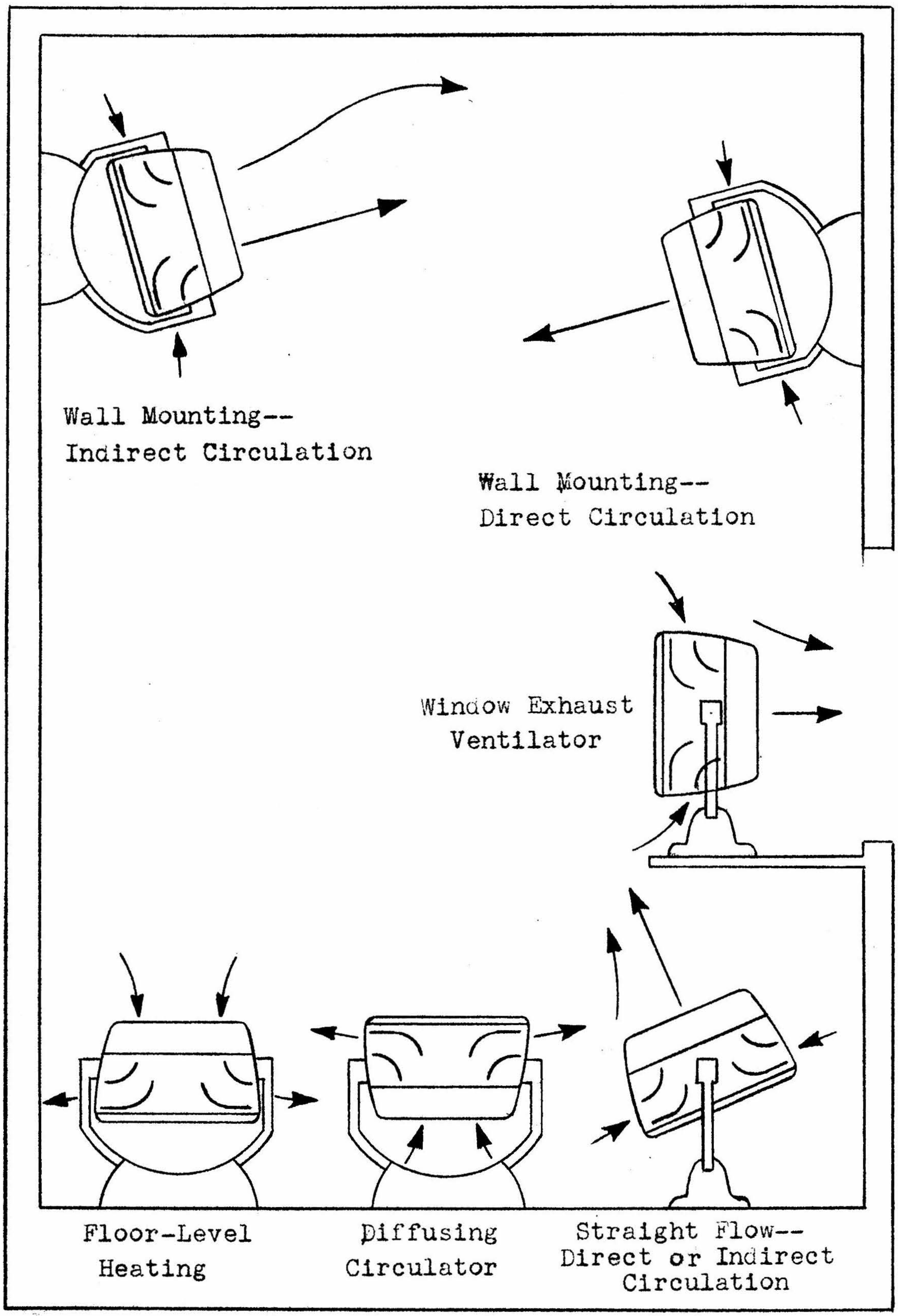


Figure 10. Operating Positions of the Unit

This is also desirable since the cooler air at the floor level will thus be drawn in, for discharge at approximately chair level. Because a complete 90-degree change in direction is seldom attained in practice except in a closed duct, the air will issue at an angle slightly above horizontal to cover the entire living area above knee height. On the other hand, when heating an area close to the unit, the lowest velocity should be used and the warm air should issue close to the floor to replace the cold air in this region and rise through the cool lower levels of the room. The position of the housing in this case is of course the same as for straight vertical flow, but with the propeller rotation reversed. To allow the required rotation of at least 180° the housing is mounted on pivots whose axis is perpendicular to that of the housing and as nearly as possible through the center of gravity of the latter. The pivots are supported from the base by a U-shaped yoke within which the housing may be rotated. Since the sides of this yoke form obvious carrying points and are structurally satisfactory for the purpose, no separate carrying handle is provided. However, the housing has adequate strength to allow the fan to be carried in any way that it might readily be lifted.

Adaptability to mounting several feet above the floor is practically a required sales feature, and is especially desirable for effective circulation in larger rooms so that high velocities can be used without disturbance to the occupants or obstruction by furniture. While pedestal mounting is offered by many nufacturers, its additional cost is a deterrent to sales, and the special features of the present unit would not normally be required at high mounting levels. The unit can, however, be readily wall-mounted by the base, and in this case the radial intake provides improved inlet conditions, particularly when discharge is nearly normal to the wall. Such a mounting is convenient in that it does not obstruct working areas such as the floor and desks, while the unit may easily be taken down for portable use--advantages not offered by pedestal mounting. The angle of depression required in wall mounting is small, since direct flow should not strike persons at close range, and only sufficient elevation is needed to circulate air across the ceiling (which may range in height from eight to twelve feet depending on the room size). An adjustment range above and below horizontal of only 15° is thus adequate at the recommended mounting height, that is, with the fan center from six to seven feet above the floor. This places the controls for ready accessibility by individuals of average height but allows the direct airstream to pass above head level. The plane of the yoke and pivots is vertical in wall mounting for greatest strength and rigidity, the vertical adjustment being obtained by sliding the yoke in an arc at the base, through an included angle of 30°. To provide horizontal adjustment to within 15° of the wall on both sides, the pivots must allow an additional rotation of 75° in conjunction with the 180° already determined, for a required total of 255°. For greater convenience, a maximum arc of 270° is provided. The pivots incorporate stops to limit motion to this angle, as well as a friction device to hold the fan at the desired angular setting without use of the usual thumbscrew.

All wiring is enclosed, running from inside the inner deflector through one pivot and the tubular yoke to the base. The yoke is held to the base at the chosen angle by a spring clamp, which provides resilience to reduce stress concentration at the point of attachment in case the fan is jarred or upset. The dimensions of the base are large enough for adequate stability, and provide mounting space for the gang switch controlling motor speed and direction and element heat, the speed-control reactance, and the yoke clamping device. The switch knob is at one end of the base under its juncture with the yoke, this to be the lower end in wall mounting, with the yoke clamp knob at the opposite end. It is recommended that the unit be also made available without the heating element, at a correspondingly lower price; the only required modifi-

Figure 10.

Color Scheme

and the yoke clamp, as well as the name plate to be mounted on the end grille, are walnut-brown solid-color plastic, giving dark accents at these points for ready identification. The name-plate lettering is in gold.

Trade Name--Since the manufacturer's trade name forms an important part of his good will, it should be selected astutely. Simplicity and distinctiveness are major requirements if the name is to be accepted by the public to the extent that they will ask for the product of a specific firm, rather than merely for a type of article. This is not to suggest, of course, that a trade name can become a "household word" if it does not represent a product of high quality.

Most products are identified by a firm name based on that of the founder (e.g., Westinghouse Electric Corporation) or one stating its products or purpose (The General Electric Company). Names of this type may also be abbreviations (Fasco Industries, formerly F. A. Smith Manufacturing Company). Other names are pure coinings (Kodak), foreign derivations (Thermos), or descriptive terms (Eversharp). A few in these groups have become convenient synonyms for all articles of the same type, regardless of make (in the case of those cited, for hand cameras, vacuum bottles, and automatic pencils). Secondary names are added to the firm name to designate the various products in a line of consumer goods. Fan manufacturers often attempt to connote some quality or function of their products by forming names from combinations of such terms as air (commonly spelled "aire"), breeze ("breez"), and flow ("flo"). Although they may be useful in advertising, real public acceptance of some of these names is questionable--it is difficult to believe that the average customer would ask for a "Kisco Handi-Table Circulair" by its full name, as an example. At any rate, most of the word combinations which are remotely applicable or even clever seem to have been claimed, so that it would be difficult to form a distinctive new term of this sort. Abstinence from the use of any such name is therefore suggested in the present case.

The design presented here has been developed to conform with the requirements which would be imposed by an established manufacturer, in regard to marketing as well as manufacturing. An old-line firm sometimes expands from the industrial-goods field by adding a line of home appliances, and adopts a new and separate name for these products. One of the proper functions of the product designer is to advise in the selection of this name, because of its close identification with the results of his work, and because his background and research are presumed to have given him special knowledge of the consumer mind.

To exemplify the considerations inherent in naming a product, the name "P H Ā R O" has been chosen. The two-syllable construction satisfies the requirement of simplicity: while the same sound could have been obtained with only four letters by using an initial "f," the diagraph "ph" is unusual in trade names, and avoids identification with the card game "faro." The heavy serif at the peak of the "a" (Ā) acts as a macron to emphasize that the pronunciation is the same as in the word "pharaoh," and heightens any exotic and aristocratic (if vague) association with ancient Egyptian royalty. A logotype with characters partaking slightly of Egyptian symbols is proposed as appropriate to the derivation of the name. The serif-A is consistent with these special characters, and may also be used readily in conjunction with standard capitals when the logotype is not available.

"PHĀRO" alliterates with the words "fair," "air flow," and "fan," for use in slogans or other advertising. Since these words are not actually spelled out in the name, it is adaptable to products other than fans (avoiding the dilemma of the Chicago Electric Manufacturing Company, which established the name "Handyhot" for its original line of heating and cooking appliances, and now makes cooling fans and ice-cream freezers as well). The present design is designated simply as a "Portable Air Circulator." This term is meant to emphasize high capacity in combination with convenient size, partially by interferring that it is unusual for an air circulator to be portable (as the twenty-inch and larger fans thus classed are not). If the line of products includes other fan models, the collective term "PHARO-Fans" (not "-Phans"!) is suggested.

Design Synthesis

The Drawings--Every part of the "PHĀRO" Portable Air Circulator is shown in the mechanical drawings (Drawings A through G2--inside back cover), and sub-assemblies have been included as needed in addition to the full-sized master assembly (Drawing H). The lettered drawings each show the parts of a major component, and the parts themselves have been numbered consecutively, essentially in the order of assembly. Part numbers followed by the letter "H" (e.g., 25H) indicate that the basic part (25) is modified when used in the optional fan-heater model. Part numbers preceded by "H" (e.g., H5) indicate a supplemental series of parts used only with the heating element, and are not incorporated in the cooling-fan model.

All necessary dimensions are given, and many of the smaller, more complex parts are reproduced full size; others are reduced to one-third size.* Most curves are dimensioned

- -

* The choice of this somewhat unusual reduction was dictated by the requirements of maximum clarity and convenient size: one-half reduction would have required too large a sheet for some of the larger parts, and is not sufficient to obviate confusion with full size, while one-fourth scale was deemed too small for clarity of detail.

only at terminal points, but their form is clearly shown. However, the shape of the side of the housing and all curves of mating parts are fully dimensioned by offsets or radii. Except in specifying standard parts or stock, decimal dimensions are used throughout. Tolerances are specified only where critical. Other tolerances would be set by the tooling and production standards of the manufacturer, in which case the general tolerances noted in the title block do not necessarily apply.

Materials and Construction--The various parts which together form a major component or sub-assembly are discussed here in groups, following the order of the lettered drawing sheets. The first such group of parts forms the Cowling.

The Cowling (Drawing A) is a toroidal shell which controls the flow of air through the propeller and separates the axial and radial regions of the airstream. It is formed from sheet aluminum in two parts, the outer and inner cowl (Parts 1 & 2), which are divided at planes normal to the direction of forming. Aluminum alloy 3S-O, with a thickness of .032", is specified for all of the larger sheet-metal parts because it is easily formed, and because it offers adequate strength with 56% less weight than the thinnest steel which could be used (24 gage, .024"). It is also less resonant than steel, tending to damp rather than amplify any sounds resulting from propeller or motor vibration. It can be given an inexpensive chemical finish for appearance and protection where the metallic color is desired. The sheet aluminum parts may be formed economically by modern semi-automatic spinning methods, where annual production ranges from fifty to fifty thousand pieces or more. No more than two spinning chucks are needed for any part in this design, and for initial production could be inexpensively made of hardwood with metal inserts. The additional cost of a trim die, needed if the parts were drawn, is eliminated since spun pieces may be trimmed with the desired cleanness and accuracy while in the spinning lathe.

After the assembly clip (4) is riveted in place in the outer cowl, the two cowls are assembled with the joint band (3) between their outer edges. The band serves an esthetic as well as a structural purpose, since it tends to stop the eye from following around to the inner face of the cowling, and thus directs emphasis to the full curve of the outside of the housing. The cowls are held together until final assembly by the push fits at the inner diameters and at the joint band. They are spray-painted with sage green enamel, which should be formulated for baking at 300° F for durability and to resist the temperature of the heating element.

The Brackets (Drawing B) are the main structural elements of the housing, acting to support all of its components as well as the housing itself. Elliptical in section, the brackets are streamlined for smooth, quiet flow. They are aluminum die castings made from Alcoa Alloy No. 218, which was selected for all die-cast parts. This alloy, containing 8% magnesium, is nearly equal to most zinc alloys in strength and weighs 62% less per unit volume. In addition to having the highest tensile strength and lowest density of any aluminum die-casting alloy, it has the greatest resistance to impact fracture: this is a valuable property in the parts of a portable home appliance which is liable to be upset or even dropped. Of the standard aluminum alloys, only Alcoa No. 218 and one other can be chemically treated to give a lustrous metallic finish of the same color as the sheet aluminum parts. Any other die-casting material except magnesium would require expensive polishing and plating. All exposed die-cast parts (excepting the base) are to be anodically finished by the Alumilite process, which builds up a protective oxide coating by off-plating in a sulfuric acid bath. A clear or "white" water-sealed finish is specified. The only finishing operations called for previous to anodizing are flash removal and a light power brushing to assure a uniform satin finish.

Although the dies for the parts designed for die casting account for the largest part of tooling costs, these parts cannot be made economically by any other method, nor would they be as attractive in appearance. Labor costs would be excessive if the same parts were sand-cast or assembled from stampings, whereas complex die-cast parts can be made in a single piece and require very little machining or finishing. The brackets (6, 7, & 8) are the only castings requiring a die of more than two sections. A third piece is needed for these parts to form the flange which mates with the cowling, and the locating pin and cored mounting holes in the flange. A special stud (9) is inserted in each of the mounting holes and a cross pin (10) is pressed into place with a plier-like tool (obtainable from the stud manufacturer). When the cowling is in place on the brackets, each stud is given a quarter-turn, engaging the cross pins in the assembly clips to lock the cowling and brackets together. The spring steel clip material gives a tight yet resilient joint which allows for any slight misalinement, and permits the extended leg of the clip to be pulled down against the inner cowl by the cross pin, holding the two parts of the cowl firmly together.

Small movable cores in the die parting plane form the screw holes for mounting the motor and the end cover, and the bearing in the pivot brackets (7 & 8). The wiring pivot bracket (7) is drilled for the extruded-aluminum wiring tube (11), which is pressed into place. This tube, also finished in Alumilite, carries the wiring to the inner part of the housing. Other machining of the brackets is limited to reaming the pivot bearings and tapping one hole in each of the pivot brackets. This is the only tapping operation required anywhere in the assembly of the unit, because all assembly screws are driven into die castings, and are of the self-tapping type which forms its own thread in the aluminum. Besides eliminating tapping, which is particularly expensive in blind holes, these screws remain tight even under vibration and thus do not require lock washers. The type 'C' screw is generally specified, since this has a standard thread and can be replaced with an ordinary screw in case of loss. Most of the screws are size number 8 with a Phillips recessed head to expedite automatic driving, but a few are button-head type 'U' drive screws which are merely pressed into place.

The Pivots (Drawing B) are screw-machine parts which fit into the pivot bracket bearings, and form journals for the rotation of the housing within the yoke. The yoke caps (17 & 18), within which the pivots are cast as inserts, fit around the top of the yoke. The caps are identical and require but one die, since they differ only as to the pivot cast in them. The direction of die parting is different from that of ejection in the caps and the cap covers (19 & 20) because of the angle at which the yoke meets the pivots. Undesirable changes in section thickness, which would result in excessive shrinkage and cracking during solidification and aging, have been eliminated by coring; and parting and ejection requirements have been taken into account in the die-cast parts. However, the provision of draft in straight-sided parts, and the location of ejector pins, gates, etc., have been left to the die designer, with the expectation that no changes in external appearance or basic design will be required.

The wiring pivot (12) is hollow to allow passage of the electric wiring from the base and yoke through the wiring pivot bracket and the wiring tube, and is curved inside to prevent abrasion of the wires. The friction pivot (13) incorporates stop pins (14) which contact the pivot retaining screw (23) in the friction pivot bracket to limit rotation of the housing to 270°. The expansion screw (16) draws the expansion plug (15) into a tapered hollow at the inner end of the friction pivot. Each quarter-turn of the screw spreads the serrated fingers of the pivot by about .002" in diameter, for adjustment of the friction in the bearing to the degree which will allow directional positioning of the housing and yet hold it at any desired setting. The thin spot provided in the cap covers is punched out in the friction pivot cap cover (20) for a snap plug (22), which gives access to the expansion screw when wear of the bearing necessitates tightening.

The Deflectors (Drawing C) are the elements of the diffuser portion of the housing which function to divert the airstream from a radial to an axial direction or vice versa. They are so proportioned that the area between the outer deflector and the cowling is approximately equal to that between the two deflectors at corresponding points. The air velocity is thus changed evenly between the blade and the outer diameter of the diffuser, to maintain uniform velocity distribution across both inlet and outlet. The deflectors are spun (or drawn) from sheet aluminum which is subsequently Alumilite-finished, and only simple dies are required to punch them for assembly. The outer deflector (25) is attached only to the brackets, by drive screws pressed into the brackets through tabs at its outer diameter. The inner deflector (26) is mounted in the same way, but has additional tabs at the inside (except where the wiring tube enters). These centralize the bracket arms and tie the housing assembly firmly together when the motor is in place.

The Heater (Drawing C) comprises resistance-wire coils (H3 & H4) supported in ceramic insulators (H1). The resistance alloy specified, which contains 16% of iron in addition to nickel and chromium, is satisfactory for operation as high as 1700° F, and will give long service at the designed operating temperature of 1100° F. The deflectors are modified (25H & 26H) to provide mounting holes for the insulators, which are attached with spring clips (H2) rather than rivets for ease of assembly and to prevent strains that might crack the ceramic material. The ends of the elements are brazed in the head slots of ordinary machine screws (H5); these screws are tightened against spacers (H8) in insulators of the same design as used elsewhere, to form terminals for the heater lead wires (H10). Where the leads pass through the deflectors to the inner end of the wiring tube, aluminum eyelets (H9) prevent abrasion of the insulation.

The Propeller (Drawing D) is designed in accordance with specifications which were developed theoretically (App. V) and compared with actual performance data (App. IV-C). The four blades (29) are formed from bright aluminum and do not require further finishing. The tabs for riveting them to the hub are placed on the less visible side of the blades; this placement also avoids excessive stresses where the tabs are bent down since the required angle is under 90°, with less likelihood of weakening the metal. The stiffness of the high-strength alloy used, and the compound curvature where the blades meet the hub, together give the blades great rigidity to resist forces tending to deform them during operation. The tabs and rivets thus need only to be strong enough to counteract the relatively low centrifugal stresses. In addition to the rivet holes, the tabs have larger holes for balancing weights (31). While fairly good static balance may be obtained by grouping blades of equal weight into sets, a dynamic balancing operation is recommended to equalize weight distribution in the axial direction. The weight holes have been separated as far as possible axially, and each of them will accommodate up to four of the lead discs, which are pressed in where required.

The hub (28) might have been built up from a drawn shell and a screw-machine shaft socket, but die casting was selected because of its simplicity and other advantages. The greater thickness of the casting gives better sound absorption, and allows the vent holes in the end to be formed so as to augment air flow over the motor. Since the shaft is not directly accessible with the motor in place, the customary setscrews are not used (although screws could be reached if large screwdriver holes were provided in the outside of the hub). A further consideration in the present design, where the direction of propeller rotation is reversible, is that setscrews would tend to loosen and the propeller might fly from the shaft. A simple mounting method has been developed to obviate these difficulties, and permits the propeller to be merely pressed into place on the shaft. This method has equal resistance to axial forces in either direction, and incidentally transmits less sound than rigid attachment by setscrews. A standard insulating grommet (32), made of synthetic rubber to resist deterioration by aging or by oil from the motor bearing, is placed in the counterbored shaft socket of the hub. The metal at the end of the socket is then spun over the grommet to hold it in place, and the propeller assembly is pushed onto the motor shaft until the grommet snaps into the knurled groove provided. It was found by test that this assembly requires only about five pounds of pressure for installation on a chamfered shaft, but when once in place will withstand a direct pull of nearly twenty pounds. The propeller when installed passes close to the deflectors and cowling to prevent recirculation; the somewhat greater spacing from the bracket arms, together with their streamlined form, obviates turbulence at these points.

The different motors used with the fan (33) and the fan-heater (33H) are to be manufactured in accordance with the electrical and mechanical specifications given in the preceding subsection, "Components of the Unit." All dimensions which would be required by an outside vendor are shown, including shaft and mounting details and clearance dimensions. Standard grommets (35) are inserted in the motor mount angles (34) spot-welded to the motor case, to provide a resilient mounting which prevents motor vibration from being transmitted to the housing. Wide tabs on the deflector at three points and large screw heads keep the grommets from being crushed against the thin brackets; at the fourth bracket, the stamped tube clip (37) fits into the belled end of the wiring tube to hold it in position and to provide a rounded surface for the wires. The motor shaft should be alined with the housing axis during assembly by further tightening of one or more of the mounting screws against the grommets. During this operation the shaft can be extended by a spindle to indicate when alinement is reached.

A capacitor-run motor is used when the heating element is not incorporated in the unit. The capacitor should be of the oil-filled type (rather than electrolytic) for long life, and is clamped to the motor case in the space between the inner deflector and the housing end cover. To minimize the number of wires which must be led between the housing and base when the heater is included, a separate rotary reversing switch (H11) is mounted on the end cover. The reversing switch knob (H12) is large in diameter so that it can be grasped readily even when the fan is positioned so that the knob is invisible to the user. The design of the knob conforms to the shape of the end cover so as not to attract unwarranted attention, and harmonizes in form and material with the knobs used on the base.

The Grilles (Drawing E) are fabricated from round steel wire by projection welding, and plated in bright chromium. Short extensions of the wires are snapped into holes in the cowling, and held by notches which engage with the sheet metal of the cowls as the grilles spring back to their original form. The end grille (38) is attached to the axial-flow end of the cowling at six points spaced approximately 60° apart, and has an open center to promote air flow through the holes in the hub face. The name plate (39) which mounts on the end grille is an open-centered piece designed to be intermediate in form between the straight lines of the grille and the curves of the housing. It is injection-molded from heat-resistant polystyrene plastic to resist deformation by the heat of the element. The thermoplastic material has sufficient elasticity to allow the molding of a special groove where-by the nameplate is snapped onto the grille and held without additional clips or screws. Gold powder is fused into the raised design to provide wear-resistant coloring.

The side grille (40) follows the curvature of the brackets so as to fit them snugly. The rings are connected by cross wires placed every 30° except at the brackets, and two gaps are left in the smallest ring to allow space for the pivots and access to the retaining screws. The number of rings used is felt to be a proper compromise between openness and solidity, and should be adequate to prevent draperies, etc., from being drawn into the heating element. However, a strict interpretation of the "Standard For Electric Heating Appliances" (Underwriters' Laboratories, Inc.) would require cross wires at each seven-eighths of an inch, and the present design would probably be subjected to special tests before approval.

The End Cover (Drawing F, 41) closes the end of the housing opposite the cowling and forms a third point of attachment for the brackets, to complete the housing structure. The cover is finished in sage green enamel to match the cowling. These painted parts of the housing act as visual terminal planes to emphasize the air-flow qualities of the lighter-colored open diffuser section. The small internal flange which finishes the edge of the cover can be formed by use of a sectional internal spinning chuck; or a rolled bead, made with the part in a separate female chuck, may be substituted for the flange. An annular space is left between the cover and the inner deflector for the passage of cooling air over the motor. By removing the four screws which secure the cover, the wiring connections are exposed for servicing, and the capacitor is accessible for replacement if required.

The yoke (43) is made from extruded aluminum tubing with a hollow rectangular section. The alloy selected, 61S, is the only readily weldable wrought alloy which can be given a light-colored Alumilite finish. (The presence of silicon in other aluminum alloys causes darkening during chemical treatment.) The light metallic finish, matching the other anodized aluminum parts, emphasizes th the free suspension of the housing within the yoke, and the inward curve toward the pivot points further accentuates this quality. While the section is shallow to avoid a heavy appearance, it has sufficient rigidity to prevent excessive deflection when the unit is wall-mounted (see Stress Calculations, App. VII). Additional strength is obtained by forming the yoke on a stretcher-bender machine, in which the tubing is clamped and hydraulically pulled around a template. This operation strain-hardens the aluminum and raises its yield point above that of the as-extruded ("F") condition.

The yoke material must be suitable for welding so that the yoke stops (44) can be attached without mechanical fasteners, which would protrude into the wiring passage. The wiring from the base enters the hollow yoke through a hole between the stops. The stops are placed off-center so that the wires will not interfere with the internal components of the base as the yoke is adjusted.

The Base (Drawing G1-2) provides stability for the unit regardless of the position of the housing, and incorporates the principal controls. The base proper (46) is a die casting which requires only a two-section die without movable cores, in spite of its large size and somewhat complex design. The use of aluminum rather than zinc saves about four pounds in weight. Alcoa Alloy No. 218 is specified because its high impact strength is particularly valuable in the base (although its anodizing characteristics are not required here since the part is enamelled). A more free-flowing alloy such as Alcoa No. 13 might be substituted if uniform density and maintenance of detail cannot otherwise be assured. The base is so designed that the suggested wall thickness of .075" can be decreased or increased (to as much as .125") to suit casting requirements, without affecting the function or appearance of the part.

Screw holes for the speed controller (53) and the baseplate (47) are cored, as is the slot for the cord bushing (56); other holes (for the control shafts and the escutcheon screws) are not in line with the direction of die parting, and so are drilled. The yoke fits in the groove at the top of the base with only sufficient side clearance to allow freedom of adjustment. The chordal height of the groove arc and the snug fit of the yoke keep the latter from tipping when its clamping device is loosened. The wiring slot at the bottom of the groove is wide enough to clear the wires and the yoke stops, but retains a bearing surface along both sides. The stops contact the ends of the slot when the limits of yoke adjustment, 15° to either side of the central position, are reached. At the end of the base nearest the wiring slot, a flat is provided for the yoke-clamp control. An internal flat at the other end forms a mounting surface for the power switch (52), which controls the motor speed and the operation of the heating elements.

The baseplate (47) performs several functions in addition to its primary one of enclosing the bottom of the base. It supports the unit in wall-mounting, by means of keyhole slots so that the fan can readily be taken down for portable use. The woodscrews (72) furnished to fit the slots are adequate to support the weight of the unit, and are driven in the wall at points conveniently spaced six inches above and below the desired mounting center. So that the switch will be readily accessible at the lower end of the base, the unit should be centered between five and one-half and seven feet from the floor, depending on the size of the room, the activities of its occupants, and the arrangement of furniture. To assure that the baseplate will always be replaced properly (with the large ends of the screw slots toward the switch), one of the holes for the baseplate attaching screws is placed asymmetrically A small notch in the plate is then also positioned correctly to retain the cord bushing in the base. The baseplate is heavy-gage steel, which provides some additional weight at the lowest point of the unit in portable use, and gives strength at the keyhole slots so the wall-mount screws will not tend to pull through. Four feet riveted to the corners of the baseplate support the unit far enough from any mounting surface to prevent damage without impairing the stable appearance of the base. Polystyrene plastic is used for the feet instead of the harder phenolic type often employed. The thermoplastic material is tough and long-wearing, and its smoothness and greater resilience allow the unit to be moved without marring finished surfaces. All manufacturer's data (such as the model and patent numbers, Underwriters' approval seal, electrical data, and servicing instructions) are incorporated in a decalcomania (49) applied to the baseplate, rather than in some other, unduly prominent location. Operating instructions and other information for the user should be furnished in the form of a small, easily read booklet.

The yoke is clamped in the desired position relative to the base by a flat steel spring (57), formed from pre-tempered stock, and finished in brushed chromium plate to match the color of the yoke. The clamping method used gives a resilient attachment which can be tightened easily and positively and prevents damage to the base if the housing is jarred. It is so designed that the yoke cannot come entirely free from the base, a danger which would be present if an external thumbscrew or other conventional fastening were used. Arms on each side of the clamp spring hold it in position where they enter the base through slots at the sides of the yoke. The ends of the arms are notched to engage studs (60) in the ends of two die-cast bell cranks (59L & 59R). Once the studs are in place, the clamp can be loosened only to a point which allows free adjustment of the yoke, since further movement is prevented by contact of the studs with the inside of the base. The bell cranks pivot in bearings formed integrally with the base casting. When the clamp is tightened, the inclined surfaces at the pivot points act to hold the cranks outward against guide faces in the base, and alinement is maintained by the clamp nut (61) which spaces the crank ends opposite the studs at the proper distance. The crank arms are proportioned to give a mechanical advantage of 4/3 for ease in tightening. The longer crank arms are perpendicular to the axis of the clamp screw (63) at the medial position of the clamp assembly, so that the screw axis does not shift appreciably as it is turned in the nut. A brass bearing (71) around the screw shaft at the outside of the base fits under the clamp knob (69) to reduce friction in tightening. The length of the screw between the knob and the nut-retaining ring (64) is so calculated that the crank pivots cannot drop clear of their bearingseven when there is no tension on the clamp spring. After the point of initial tension is reached, a spring deflection of .04" is sufficient to secure the unit firmly to the base when wall-mounted; only slightly more than one full turn of the knurled knob is required. Knobs of the same design are used for both the clamp and the control switch, to maintain the symmetrical form of the base. The switch knob (68) is modified only by a small groove which is filled with gold enamel to indicate the switch position, and it is held in place by a retaining spring which bears against a flat on the switch shaft. The switch knob is provided with a dog-point setscrew in place of the spring.

The point of the screw acts as a shear pin to secure the knob, so the utmost tightening is not required (the more commonly used cone-point screw might strip the threads of the phenolic knob before exerting sufficient pressure for secure attachment).

Basic operating directions are given on the knob escutcheons (65 & 66). These are of anodized aluminum, and the etched lettering is filled with walnut-brown enamel to match the knobs. Since the ends of the base are small, the size of the escutcheons is limited, while the lettering must be large enough for easy legibility (hence the abbreviation "strait" in the indication for straight flow, and the word "spread" instead of "diffused"--although the former possibly describes the function more clearly to the average user). Drive screws are used, rather than ordinary escutcheon pins, to draw the escutcheons firmly to the curved end surfaces of the base.

<u>Size and Cost</u>--The complete fan-heater is made up of eighty-five numbered parts, comprising some two hundred and fifty separate pieces; the model without the heater requires seventy-three numbered parts, or about one hundred and eighty pieces. The weight of the unit is less than fifteen pounds because of the extensive use of aluminum in its construction. The minimum over-all space

required is about one and one-half feet wide, one foot deep, and two feet high, with a volume of two and nine-tenths cubic feet. While the cubical limit established in the subsection "Size and Capacity of the Unit" is exceeded by about one-eighth, the weight is only two-thirds of the maximum which was based on a large portable fan of conventional design.

The unit performs to best advantage when placed on the floor, where ample space can be provided in the average home; for office use, wall mounting may be preferable. The space actually required for the base of the unit measures only eight and one-half by sixteen inches, which is not excessive even for table use. The proportions of the circulator are such that it is easily carried when lifted by the yoke, and oonsidering the many functions combined in this single appliance, its size should not be a disadvantage. Weight is a greater factor in portability in this case, and the unit is light both in appearance and in fact, yet without sacrificing the effect of solidity and strength.

It is difficult to form a valid estimate of the cost of the unit without knowledge of the specific facilities by which it is to be produced and marketed. The degree of tooling, the types of parts to be purchased from outside vendors, initial production quantities, scope and manner of distribution--these factors are typical of the ones which enter into a cost estimate for an appliance of this type. Some information is available to indicate the price range in which it will be sold, however.

The greatest tooling expense will be for the die castings. Molds for these parts will cost about $35,000. Other tooling, including spinning chucks, extrusion dies, punches, and drill and assembly jigs, should not exceed $10,000. Amortized over a three-year period, on an average annual production of 15,000 units, tooling will thus amount to $1.00 per unit. The approximately three pounds of aluminum die castings can be purchased for about $2.00 at current prices, and raw materials including sheet metal, wire, etc., for $3.00 additional. Although relatively few appliance manufacturers produce their own die castings, many of them have facilities for making small motors. Assuming that this is the case, a base cost of $3.50 can be assigned to either motor required for this design. The high-current switch required for use with the heater should be obtainable for $3.25, and other purchased parts, such as the reversing switch and the control knobs, for $1.50. Assuming that labor and overhead each account for as much expense as the material costs for parts made by the principal firm, and doubling the cost of purchased parts and tooling (since most of the labor and part of the overhead is already included), the total manufacturing cost is \$35.50. On the basis developed in the subsection "Characteristics of the Trade," the retail price would be approximately \$77.00. Without the heating element, the unit would sell for about \$11.00 less, because of the less expensive switch required, and the elimination of some seventy pieces which reduces material and labor costs.

Evaluation

The purpose of this thesis, namely "to develop the complete design of a portable air-circulating unit," has been fulfilled. The "PHĀRO Portable Air Circulator" combines a greater number of functions than is found in any similar appliance available today, in order to make the maximum contribution to health and comfort at moderate cost. While the unit is high in price compared with the typical fan of conventional design, there are many more expensive fans on the market which do not offer comparable performance or utility. In view of the growing public demand for high-quality units that can be used throughout the year and in a variety of ways, this design should find good acceptance.

All details of construction have been developed, with regard to the requirements of actual production. Although too many variables are involved to permit the performance of any air-impelling unit to be exactly predicted, the radical propeller design presented here is expected to approach closely, if not exceed, its theoretical capacity. Most of the mechanical features have been tested in a full-scale prototype, assuring that their operation in the lighter and stronger production model will be satisfactory. The arrangement of electrical components (particularly the heater and the wiring from the base into the housing) is of course subject to examination before approval by Underwriters' Laboratories, Incorporated; this step is required, not only as a matter of responsibility on the part of the manufacturer, but because the consumer has learned to expect the Underwriters' seal on the better electrical appliances.

The controls have been so planned that operation is simple and easily learned, in spite of the many ways in which the unit may be used. It is felt to be pleasing in form and finish, and is suitable for use in almost any home or office from the standpoint of appearance as well as function.

APPENDIX I

A. References

1. "Fans." Encyclopedia Americana (1947), vol. 11, p. 11.
2. "Propeller." Encyclopedia Britannica (1946), vol. 20, p. 513.
3. "Henry, Joseph." Encyclopedia Britannica (1946), vol. 11, p. 444.
4. "Electric Motor." Encyclopedia Britannica (1946), vol. 8, p. 198.
5. "Heating, Ventilating, and Air Conditioning Guide." American Society of Heating and Ventilating Engineers (1946), p. 215.
6. Ibid., pp. 219-221.
7. Ibid., p. 226 ff.
8. Ibid., p. 221 ff.
9. Morse, H.F., "Physical Comfort." The General Electric Company (1943), p. 3.
10. "Code of Minimum Requirements for Comfort Air Conditioning." American Society of Heating and Ventilating Engineers (1938), sect. VI, par. 2.
11. Loc. cit., ref. 8.
12. Op. cit., ref. 5, p.1.
13. Op. cit., ref. 10, sect. V, par. 6.
14. Op. cit., ref. 5, p. 216.

15. Op. cit., ref. 5, p. 217 (Table 1).

16. "Pressure Ventilating Equipment for Subway Cars." Westinghouse Electric Corporation (1948), p.1.

17. Badgett, W.H., "The Installation and Use of Attic Fans." Bulletin of the Agricultural and Mechanical College of Texas, fourth series, vol. 11, no. 7 (July 1, 1940), p. 4 ff.

18. Smith, E.G., "The Efficient Use of Portable Fans." Bulletin of the Agricultural and Mechanical College of Texas, fifth series, vol. 1, no. 10 (September 15, 1945), p. 10 ff.

19. Op. cit., ref. 5, p. 262.

20. "Winter Use of Electric Fans." The General Electric Company (1945), p. 2 ff.

21. Op. cit., ref. 5, p. 231.

22. Baumeister, Theodore, Jr., "Fans." McGraw-Hill Book Company, Inc. (1935), pp. 1-41.

23. Baumeister, op. cit., pp. 221-236.

24. "The Development of Quiet Propeller Fans." General Electric Review, vol. 37 (February, 1934), p. 82 ff.

25. Bleier, Frank P., "Design, Performance, and Selection of Axial Flow Fans." Heating and Ventilating, vol. 43, no. 10 (October, 1946), p. 83 ff.

26. Dowell, M.F., "Study of Air Movement Through Axial-Flow Free-Air Propellers." General Electric Review, vol. 42 (May, 1939), pp. 210-217.

27. "Designing Air Impelling Units." The Torrington Company (1948), p. 5.

28. Loc. cit., ref. 25.

29. Upson, Walter L., "Some Features of Propeller Fan Performance." Refrigerating Engineering, vol. 43, no. 6 (June, 1942), pp. 343-346.

30. Trade Magazines consulted (years 1937-1949):

"Consumers' Research Bulletin," Consumers' Research Inc. (Washington, New Jersey).

"Edison Electric Institute Bulletin," Edison Electric Institute (New York).

"Electrical Merchandising," McGraw-Hill Publishing Company (New York).

"Electrical West," McGraw-Hill Co. of Calif. (San Francisco).

"Electrical World," McGraw-Hill Publishing Company (New York).

"Electrical Review," Electrical Review (London).

"Heating and Ventilating," The Industrial Press (New York).

"Heating, Piping and Air Conditioning," Keeney Publishing Company (Chicago).

"Modern Plastics," Modern Plastics, Inc. (New York).

"Printers' Ink," Printers' Ink Publishing Company (New York).

"Product Engineering," McGraw-Hill Publishing Company (New York).

"Refrigerating Engineering," American Society of Refrigerating Engineers (New York).

31. "Standards for Fans." National Electrical Manufacturers Association (1947), pp. 7-9.

32. Morse, op. cit., p.3.

33. Smith, op. cit., p.1.

34. "Fan Installations in Large Rooms." The General Electric Company (1939), pp. 1-3.

35. "N.E.M.A. Free Air Test and N.A.F.M. Code Test Performance Data." The Torrington Company (1948).

B. Bibliography

Baumeister, Theodore, Jr., "Fans." McGraw-Hill Book Company, Inc. (New York, 1935).

Hagen, H.F., "Centrifugal and Propeller Fans," Marks' "Mechanical Engineers' Handbook." McGraw-Hill Book Company, Inc. (New York, 1941).

Hill, Leonard, & Campbell, Argyll, "Health and Environment." Edward Arnold & Co. (London, 1925).

Hobbs, Douglas B., "Aluminum." The Bruce Publishing Company (Milwaukee, 1938).

"Die Casting For Engineers." The New Jersey Zinc Company (New York, 1942).

"Heating, Ventilating, and Air Conditioning Guide." American Society of Heating and Ventilating Engineers (New York, 1946).

"Tool Engineers Handbook" (American Society of Tool Engineers). McGraw-Hill Book Company, Inc. (New York, 1949).

APPENDIX II. FAN LAWS

For constant air density with operation at the same point of performance.

Abbreviations: capacity in cfm, Q; horsepower, hp; speed in rpm, S; diameter, D; pressure or head, H.

A. For a Given Fan ($D = k$)[a*]

1. $Q_1:Q_2::S_1:S_2$, or $Q \propto S$
2. $hp \propto Q^3$
3. $hp \propto S^3$
4. $H \propto S^2$
5. $H \propto Q^2$
6. $hp \propto H^{3/2}$

B. For a Series of Geometrically Similar Fans at Constant Speed ($S = k$)[a]

1. $Q \propto D^3$
2. $hp \propto D^5$
3. $hp \propto Q^{5/3}$
4. $hp \propto H^{5/2}$

---

* Superior letters indicate references listed at end of Appendix II.

5. $H \propto D^2$
6. $H \propto Q^{2/3}$

C. For a Series of Geometrically Similar Fans at Constant Volume ($Q = k$)[b]

1. $S \propto 1/D^3$
2. $hp \propto 1/D^4$
3. $hp \propto S^{4/3}$
4. $hp \propto H$
5. $H \propto 1/D^4$
6. $H \propto S^{4/3}$

D. Overall Relationships for Similar Fans[c]

1. $Q \propto D^3S$
2. $hp \propto D^5S^3$
3. $H \propto D^2S^2$

E. Example of the Use of the Fan Laws

The operation of the fan laws in the selection of a fan for a given duty may be exemplified as follows:

Assume that 1200 cfm are required, and that this capacity may be obtained from a fan 12" in diameter, operating at 1500 rpm with a 1/25 hp motor.

1. A 10" fan of the same form would then need to operate at a speed determined by the relation $S \propto 1/D^3$ (from C.1) and would require a horsepower given by $hp \propto 1/D^4$ (from C.2), or

$$rpm_{10} = 1500 \cdot 12^3/10^3 = 2590 \text{ rpm}$$

$$hp_{10} - 1/25 \cdot 12^4/10^4 = .0892 \text{ hp (about 1/12).}$$

2. For a similar 14" fan,

$$rpm_{14} = 1500 \cdot 12^3/14^3 = 944 \text{ rpm}$$

$$hp_{14} = 1/25 \cdot 12^4/14^4 = .0216 \text{ hp (about 1/46).}$$

Thus, for a decrease or increase in diameter of 1/6, the respective speed and horsepower requirements are approximately doubled or halved.

- -

a. Baumeister, Theodore, Jr., "Fans." McGraw-Hill Book Company Inc. (New York, 1935), p. 55 ff.

b. Derived from Parts A & B.

c. Marks, Lionel S., "Mechanical Engineers' Handbook." McGraw-Hill Book Company Inc. (New York, 1941), p. 1948.

III. THEORETICAL DISPLACEMENT OF PROPELLER FANS

A. Constant-Pitch Helix

A helical surface whose advance per revolution, or pitch, is the same at any radius may be formed by a generatrix of length r_o perpendicular to an axis of rotation, moving at constant axial and angular velocity (Diagram A). The end of the generatrix defines a cylinder of radius r_o. If the curve formed on the surface of the cylinder crosses the elements of the cylinder at an angle a, the axial distance travelled per revolution of the generatrix is defined as the pitch,

$$P = 2\pi r_o \tan a = k^* \qquad (1).$$

Since the pitch is constant, the angle b at any smaller radius r_1 is

$$b - \tan^{-1} (P/2\pi r_1) \qquad (2),$$

which becomes 90^o at the central axis.

If the helical surface is fixed axially and rotated, it acts as an Archimedean screw to displace in one revolution a volume equivalent to that of a cylinder of height P and radius r_o,

$$q = 2\pi r_o \tan a \cdot \pi r_o{}^2 = 2\pi^2 r_o{}^3 \tan a \qquad (3).$$

A constant-pitch propeller of N blades may be formed by

- -

* The letter "k" is used throughout the calculations to signify various constants, which are not to be taken as equivalent.

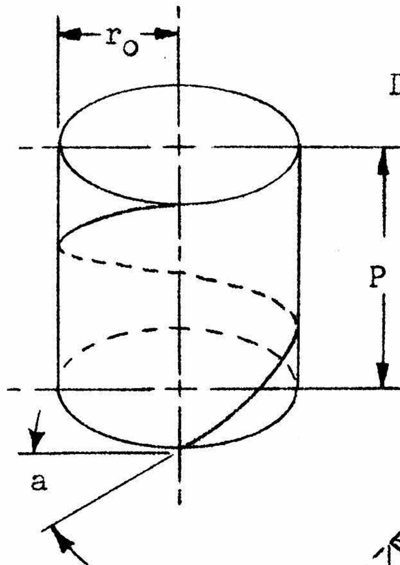


Diagram A

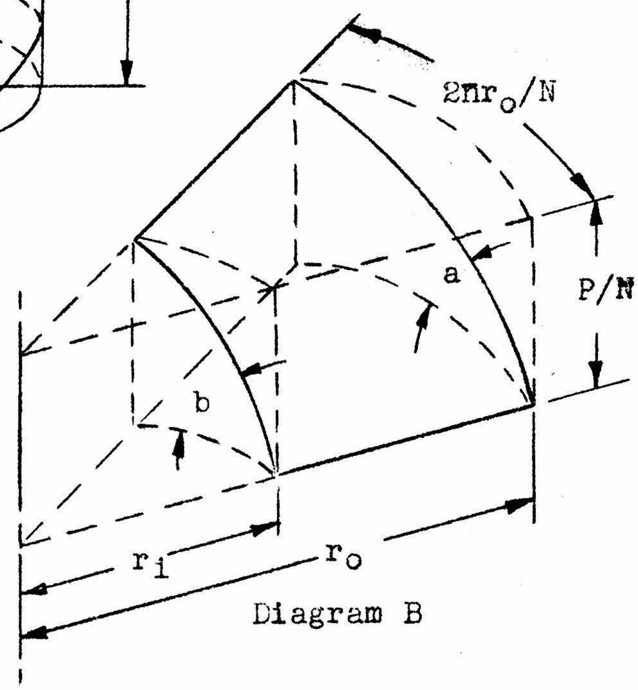


Diagram B

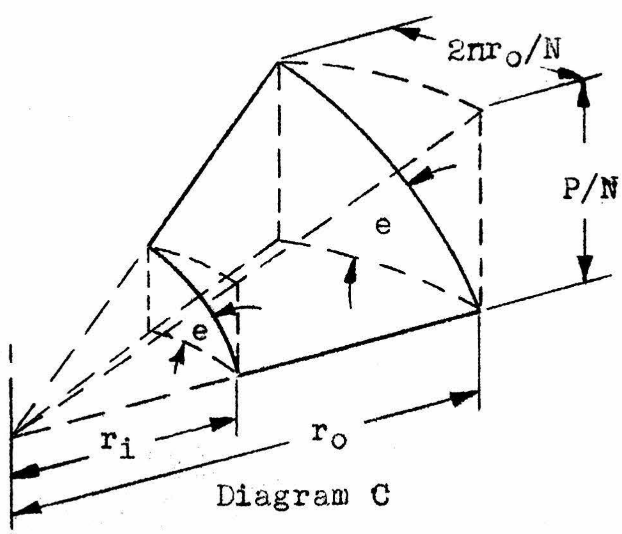


Diagram C

dividing the surface into N equal sectors, each of which will have an axial length P/N and displace a volume q/N per revolution. These blades may be arranged axially to lie between two planes perpendicular to the axis and separated by a distance

$$B = P/N \qquad (4),$$

which is the axial depth of the propeller (Diagram B).

B. Constant-Angle Helix

A helical surface which maintains a constant angle e with the axis at all radii may be divided into N equal sectors to form a propeller which resembles the flat-blade type (Diagram C). The pitch of such a propeller must vary with the radius since the angle is constant, or

$$\tan e = P_r/2\pi r = k \qquad (5).$$

The axial depth at the center is thus zero ($P = 0$ at $r = 0$), while the depth at the periphery is

$$P_{r_o} = 2\pi r_o \tan e \qquad (5a).$$

The displacement per revolution of the complete propeller of N blades is equivalent to the volume of a cylinder of height P_{r_o} and radius r_o, minus the volume of a cone of the same dimensions, or, from Equation 5a,

$$q = \pi r_o^2 P_{r_o} - (1/3)\pi\, r_o^2 P_{r_o} = (2/3)\pi r_o^2 \cdot 2\pi r_o \tan e$$
$$= (4/3)\pi^2 r_o^3 \tan e \qquad (6).$$

C. Comparison of Propeller Types

In comparing and predicting propeller performance, two additional relations may be required. The first is the pitch ratio, defined as the ratio of the pitch to the outside diameter, or, from Eq. 1,

$$pr = P_{r_o}/2r_o = \pi \tan a \qquad (7).$$

The second is the projected-area ratio, which is the fraction of the area of the circle of radius r_o which would be covered by an axial projection of the blades; the effect of a ratio other than unity is to vary the axial lead per revolution so that it is not equivalent to the pitch. The displacement will be affected in direct proportion, which may be determined by the sum of the axial depths of the individual blades divided by the theoretical pitch based on the blade angle (both of which factors must be measured at the same radius in the case of a constant-angle propeller), so that

$$ar = BN/P \qquad (8).$$

A comparison may be made of the constant-pitch and constant-angle types on the basis of equal volumes:

$$\frac{q\ (P = k1)}{q\ (e = k_2)} = \frac{2\pi^2 r_o^3 \tan a}{(4/3)\pi^2 r_o^3 \tan e} = 1, \text{ or}$$

$$\tan a = (2/3) \tan e \qquad (9).$$

The axial depth at the periphery of a constant-angle blade is thus 3/2 that of an equivalent constant-pitch blade. The radius at which the angle of a constant-pitch blade must equal that of a constant-angle blade for equivalent displacement is determined (using Eq. 2, 3, and 6) from the relation,

$$\frac{q\ (p = k_1)}{q\ (e = k_2)} = \frac{\text{pitch} \cdot \text{area}}{(4/3)\pi^2 r_o^3 \tan e} = \frac{2\pi r_1 \tan b \cdot \pi r_o^2}{(4/3)\pi^2 r_o^3 \tan e} = 1,$$

where tan b = tan e, so that

$$r_1 = (2/3) r_o \qquad (10).$$

The specified pitch angle, or "nominal pitch," of commercial blades, as well as of aircraft propellers, is measured at this radius, so that in estimating the capacity of a commercial propeller, the displacement per revolution, from Eq. 3, may be taken as

$$q = (4/3)\pi^2 r_o^3 \tan b \qquad (11),$$

where b is the nominal pitch.

The difference in the velocity and volume characteristics of the two basic propeller types may be readily demonstrated. From the equivalent relations for capacity in cfm (by which it is equal either to the product of the flow area and the velocity, Q = AV, or to displacement per revolution times revolutions per minute, Q = qS) at constant speed, $V \propto q/A$. For any portion of a constant-angle blade to a radius r_1,

$$V_{av} \propto \frac{(4/3)\pi^2 r_i{}^3 \tan e}{\pi r_i{}^2}$$, and since tan e is a constant,

$$V_{av} \propto r_i \qquad (12).$$

Further, since the average velocity over the whole blade is proportional to

$(4/3)\pi^2 r_o{}^3 \tan e \;/\; \pi r^{\,2}$, which is equivalent to the expression

$P_{av} \cdot A \;/\; A$, the radius at which the velocity is equal to the average may be found, using Eq. 5:

$(4/3)\pi^2 r_o{}^3 \tan e - 2\pi r_{av} \;\; \tan e \;\; \pi r_o{}^2$, or

$$r_{av} = 2r_o/3 \qquad (12a).$$

For a portion of a constant-pitch blade,

$$V_{av} \propto \frac{2\pi r_i \tan b \cdot \pi r_i{}^2}{\pi r_i{}^2}$$, and since $2\pi r_i \tan b$ is

a constant equal to the pitch P, at all radii

$$V_{(e=k_2)} = k \qquad (13).$$

In a similar way the relative displacement of various portions of the blade, which might be termed the "volume distribution," may be compared in the two blade types. For a portion of a constant-angle blade between radii r_1 and r_2,

$$q_{1-2} \propto r_2{}^3 - r_1{}^3. \qquad (14).$$

For a similar portion of a constant-pitch blade,

$$q_{1-2} \propto r_2^2 - r_1^2 \qquad (15).$$

The displacement per unit of area may also be compared, as an indication of the relative amounts of "work" performed. For a constant-angle blade,

$$(q/A)_{1-2} \propto (r_2^3 - r_1^3)/(r_2^2 - r_1^2) \qquad (16).$$

For a constant-pitch blade,

$$(q/A)_{1-2} = k \qquad (17).$$

These latter two functions of course have the same form as the velocity functions, Eq. 12 and 13.

If a propeller of each type were divided into annular portions of equal radial width, the relative effectiveness of such portions would be found to vary widely (here r_o is taken as unity):

Annular portion	Area, both types	Volume Distribution	
		$e = k$	$P = k$
$r_o - 2r_o/3$	5/9 (55.6%)	19/27 (70.4%)	5/9 (55.6%)
$2r_o/3 - r_o/3$	3/9 (33.3%	7/27 (25.9%)	3/9 (33.3%)
$r_o/3 - 0$	1/9 (11.1%)	1/27 (3.7%)	1/9 (11.1%)

IV. COMPARISON OF THEORETICAL AND ACTUAL PERFORMANCE OF COMMERCIAL PROPELLERS

A. General Equation for Capacity

A generalized formula for the capacity of a helical propeller of constant pitch may be developed from Eq. 3 (Appendix III) for the displacement per revolution, which may be written

$$q = P \cdot A \qquad (18).$$

From this, the capacity may be determined by incorporating the speed, the projected area ratio, and a factor which accounts for the effect of the number and form of the blades on actual output, which may be considered as the volumetric efficiency:

$$Q = q \cdot S \cdot ar \cdot E \qquad (19).$$

For pitch P and radius r in inches, area A in square inches, speed S in rpm, and the projected area ratio $ar = BN/P$ as a decimal (where B is the axial blade depth in inches and N the number of blades), the capacity Q in cfm for a constant-pitch propeller is given by

$$Q = 2\pi r_o \tan a \cdot \pi r_o^2 \cdot S \cdot (BN/P) \cdot E \,/\, 1728 \qquad (19a).$$

Where the nominal pitch is given in terms of the angle b measured at $r_1 = 2r_o/3$, and $P = 2\pi r_o \tan a = 2\pi(2r_o/3) \tan b$, (see Eq. 10 and 11),

$$Q = E \cdot (4/3)\pi^2 r_o^3 \tan b \cdot S\ (BN/P) \,/\, 1728$$

$$= E \cdot \pi r_o^2 \cdot S \cdot BN / 1728 \qquad (20).$$

If the area within which the blade angle is greater than 45^o is omitted (since it is practically ineffectual and largely occupied by the hub) its radius as determined from

$$P = 2\pi(2r_o/3) \tan b = 2\pi r_{45} \tan 45^o \quad \text{is}$$

$$r_{45} = (2r_o/3) \tan b \qquad (21).$$

Subtracting the area represented by this radius,

$\pi(2r_o/3)^2 \tan b$, from that given by Eq. 20,

$$Q = E \cdot (4/3)\pi^2 r_o^3(1 - 4 \tan^2 b/9) \tan b \cdot S\ (BN/P)/1728$$

$$= E \cdot \pi r_o^2(1 - 4 \tan^2 b/9)\ S \cdot BN / 1728 \qquad (22).$$

B. General Equation for Horsepower

The power is the product of the volume and the pressure. In a free-air unit the static pressure may be disregarded since it is normally very low, and is not a factor in the usual applications of such units, so the pressure may be based on velocity alone, or

$H \propto V^2/2g \propto (Q/A)^2/2g$. For pressure in pounds per square foot at standard air density ($d = .075\ lb/ft^3$), with Q in cfm, A in in^2, and g in ft/sec^2,

$$H = .075\ Q^2 / (A/144)^2 \cdot 2g \cdot 60^2$$

$$= .075\ Q^2 \cdot 12^4 / 2g\pi^2 r_o^4 \cdot 60^2 \qquad (23).$$

For power in horsepower, from Eq. 20 and 23,

$$hp = QH/33{,}000$$

$$= (E\pi r_o^2\ SBN/12^3)^3\ .075 \cdot 12^4 \ /\ 2g\pi^2 r_o^4 \cdot 60^2 \cdot 33{,}000$$
$$= E^3\pi r_o^2\ S^3B^3N^3\ .075\ /\ 2g \cdot 12^5 \cdot 60^2 \cdot 33{,}000$$
$$= 1.239 \cdot 10^{-16}\ E^3 r_o^2 S^3 B^3 N^3 \qquad (24).$$

Using Eq. 22 and 23, the same derivation gives, for a propeller in which the hub area to the point where $b = 45°$ is omitted,

$$hp = 1.239 \cdot 10^{-16}\ E^3 r_o^2 (1 - 4\ \tan^2 b/9) S^3 B^3 N^3 \qquad (25).$$

Since BN ($= P = 2\pi r_o \tan a$) is proportional to the diameter D, and $r_o = D/2$, Eq. 24 and 25 are of the general form given by the fan laws, hp $\propto S^3 D^5$ (Appendix II-A, 4-b).

The horsepower may also be expressed in terms of the capacity Q. From Eq. 22 and 24,

$$hp = Q^3 .075 \cdot 12^4 \ /\ 2g\pi^2 r_o^4 \cdot 60^2 \cdot 33{,}000$$
$$= 2.061 \cdot 10^{-8}\ (Q^3/r_o^4)\ (1 - 4\ \tan^2 b/9)^2 \qquad (26).$$

C. Comparison of Theoretical Values with Test Data

The theoretical capacity and horsepower for a group of propellers of various diameters, pitches, and styles, operating at various speeds, are tabulated on the following page for comparison with the results obtained in NEMA and NAFM tests as listed in the 1949 catalog of the manufacturer, The Torrington Manufacturing Company, Torrington, Connecticut. The equations used were numbers 22, 25, and 26. The results indicate that within the capacity range of the present design, the NEMA (free-air) capacity rating may be predicted approximately on the basis of 125% of the value calculated by Eq. 22, and the horsepower based on this higher capacity value, by Eq. 26, should be adequate.

Table C. Comparison of Theoretical Values With Test Data

Blade Description							Capacity, cfm					Horsepower			
Series & Type	Diameter, "	Nom. Pitch, °	Pitch Ratio	$1 - 4\tan^2 b/9$	Blade Depth B, "	No. Blades N	Speed, rpm	Calculated, Eq. 22	NAFM Rating	E, % NAFM/Calc.	NEMA Rating	E, % NEMA/Calc.	Calculated, Eq. 25	NAFM Rating	NEMA Rating	Calc., Eq.26, NEMA cfm
E "Hi-Eff"	10	18	.682	.953	1-13/16	4	3450	1084	980	90	1390	128	.046	.062	.083	.097
Y Free-Air	12	25	.976	.913	2-23/32	3	2400	1170	1070	92	1650	141	.031	.092	.10	.086
Standard Free-Air	12	27	1.068	.839	2-13/16	4	1500	977	848	87	1190	122	.015	.031	.032	.034
P Pressure	14	23	.889	.920	3 - 9/32	4	1725	1856	1682	91	1990	107	.065	.12	.083	.080
P Pressure	16	33	1.361	.812	5	4	850	1610	1645	102	2100	130	.032	.060	.058	.071

APPENDIX V. DESIGN AND PERFORMANCE OF PROPELLER FOR THESIS DESIGN

A. Geometrical Construction

The propeller is a modification of the constant-pitch helical type, having four blades, with a projected-area ratio of 1 and a pitch ratio of 1. The blade form is generated within a right circular cone of height H and radius H/2 by a generatrix of length s, rotating at a constant angular velocity about the axis of the cone and remaining always perpendicular to its surface (Diagram D). The axial velocity of the generatrix is such that the curve traced by it on the surface of the cone cuts the elements of the latter at a constant angle a. By Eq. 7 (App. III),

$pr = \pi \tan a = 1$, so that

$\tan a = 1/\pi$, and

$a = 17.66^\circ$.

The propeller hub is a truncated portion of a coaxil, coapical cone of such proportions that the curve traced on its surface by the generatrix cuts its elements at 45°, which is thus the maximum blade angle. Since the pitch is constant at all points on the generatrix, the generatrix intercept s_1 at the inner cone may be found by Eq. 1 and 2:

$P = 2\pi s_o \tan a = 2\pi s_1 \tan 45^\circ = k$, or

$s_1 = s_o/\pi$. (Note: While the 45° angle thus defined is in a plane normal to the generatrix rather than tangent to the hub cone, the discrepancy is small, and calculation is greatly simplified.)

The nominal pitch angle b measured at $2s_o/3$ (see Eq. 10) is similarly determined by Eq. 1:

$s_o \tan a = (2s_o/3) \tan b$, or

$\tan b = 3/2\pi$, and

$b = 25.5^\circ$.

From Eq. 19 and 20, the capacity may be expressed as

$Q = (q \cdot ar)\ S \cdot E / 1728 = (\pi r_o^2\ BN)\ S \cdot E / 1728$, or

$q \cdot ar = \pi r_o^2\ BN$.

The term $\pi r_o^2\ B$ is equivalent to the volume displaced by one blade in one revolution. In this design, where a constant, r_o, is replaced by a variable, s_o, the term corresponds to the volume between the surfaces which would be formed if s_1 and s_2 (Diagram D) were rotated without moving axially. Designating this volume as d,

$q \cdot ar = d \cdot N$.

Rewriting Eq. 20,

$Q = d \cdot N \cdot S \cdot E /1728$.

For the desired theoretical capacity of 1200 cfm at 1500 rpm,

$d = Q \cdot 1728 / N \cdot S \cdot E = 1200 \cdot 1728 / 4 \cdot 1500 \cdot 1$

$= 1728/5$.

The volume d is generated by rotation of an area defined by the shaded portion of Diagram E. The calculation of d is based on the difference of two volumes as indicated by shading in Diagram F, the larger of which has a dimension $R_o = R_1$, and the smaller, $R_o = R_2$. Each of these volumes may be considered as the sum of the volumes of two right circular cones of radius h and heights X and Y minus the volumes of two cones of radius j and heights x and y. By similar triangles,

$$j = h\ ((s_1/\pi)/s_1) = h/\pi.$$

The volume of a cone being equal to 1/3 the product of the height and the area of the base, the volume of rotation in Fig. C is

$$\pi h^2(X + Y)/3 - \pi(h/\pi)^2(x + y)/3.$$

From Diagram E, $\tan f = 1/2 = \tan 26.57°$. By the Pythagorean theorem,

$$X + Y = x + y = (R_o^2 + (R_o/2)^2)^{1/2} = \sqrt{5}R_o/2.$$

Also, $X = 2h$, and

$$R_o = (h^2 - \overline{2h}^2)^{1/2}, \text{ or}$$

$$h = R_o/\sqrt{5}.$$

The total volume of the portion shaded in Diagram F is thus

$$(\pi/3)(R_o/\sqrt{5})^2\sqrt{5}R_o/2 - (\pi/3)(R_o/\sqrt{5}\pi)^2\sqrt{5}R_o/2$$

$$= (\pi/3)(R_o^2/5)(\sqrt{5}R_o/2)(1 - 1/\pi^2) - (\pi^2 - 1)\sqrt{5}R_o^3/30\pi.$$

The volume per blade per revolution is then

$$d = (R_1^3 - R_2^3)(\pi^2 - 1)\sqrt{5}/30\pi = 1728/5.$$

R_o must also vary in such a way that the elements of the outer cone are cut at a constant angle a by the end of the generatrix s. If the surface of the cone is considered as a plane, it forms a circular sector of radius R_o and circumference $2\pi h = 2\pi R_o/\sqrt{5}$ (Diagram G). The central angle is

$$c_o = 2\pi\ (2\pi R_o/\sqrt{5})/2\pi R_o = 2\pi \sin f = 2\pi/\sqrt{5} \text{ radians}$$
$$= 360^o/\sqrt{5} = 160.99^o.$$

The end of the generatrix, in forming one blade by rotating through 90^o, covers 1/4 of the sector, the corresponding angle between R_1 and R_2 being

$$(\theta_1 - \theta_2)_o = c_o/4 = \sqrt{5}\pi/10 \text{ radians}.$$

To satisfy the requirements for intersection of the elements, or polar radii, at a uniform angle, the path on the surface of the cone must be an exponential spiral, with the form

$$R = pe^{k\theta}$$

In Diagram H, tan a = $dR/Rd\theta$ is represented schematically. Differentiating R with respect to θ,

$$dR/d\theta = kpe^{k\theta}, \text{ or}$$

$$\tan a = dR/Rd\theta = (dR/d\theta)/R = kpe^{k\theta}/pe^{k\theta} = k, \text{ and}$$

$$R = pe^{(\tan a)\theta} = pe^{(1/\pi)\theta}$$ (cf. Marks' "Handbook," Fourth Edition, p. 155).

If $\theta_2 = 0$ at R_2, then $\theta_1 = \sqrt{5}\pi/10$ at R_1. Thus

$$R_2 = pe^{(1/\pi)0} = p, \text{ and}$$

$$R_1 = pe^{(1/\pi)(\sqrt{5}\pi/10)} = pe^{\sqrt{5}/10}, \text{ so that}$$

$$R_1{}^3 - R_2{}^3 = (pe^{\sqrt{5}/10})3 - p^3 = p^3((e^{\sqrt{5}/10})^3 - 1).$$

The equation for the volume d (per blade per revolution) may then be rewritten

$$d = p^3((e^{\sqrt{5}/10})^3 - 1)(\pi^2 - 1)\sqrt{5}/30\pi = 1728/5$$

Solving for $p = R_2$,

$$R_2 = [(1728/5)30\pi/((e^{\sqrt{5}/10})^3 - 1)(\pi^2 - 1)]^{1/3}$$

$$= 11.977 \text{ inches, and}$$

$$R_1 = [R_2e^{\sqrt{5}}]^{10} = 14.979 \text{ inches.}$$

The generatrix lengths s_o are

$$s_1 = R_1 \tan f = 14.979/2 = 7.49 \text{ inches and}$$

$$s_2 = 5.99 \text{ inches.}$$

The hub intercepts s_i on these lines are

$$s_1/\pi = 2.38 \text{ inches and}$$

$$s_2/\pi = 1.91 \text{ inches.}$$

By similar considerations, the equation may be found for the spiral at the hub cone. The length R where s has rotated through any portion of its 90° movement per blade, or traced the same portion of the $\sqrt{5}\ \pi/10$ radian segment of the spiral may also be found. For example, in 45° of rotation of s,

$$R_m = pe^{(\sqrt{5}/10)/2} = 11.977\ e^{.118} = 13.39 \text{ inches.}$$

B. Velocity in Straight and Diffused Flow

The minimum area and thus the maximum velocity of discharge

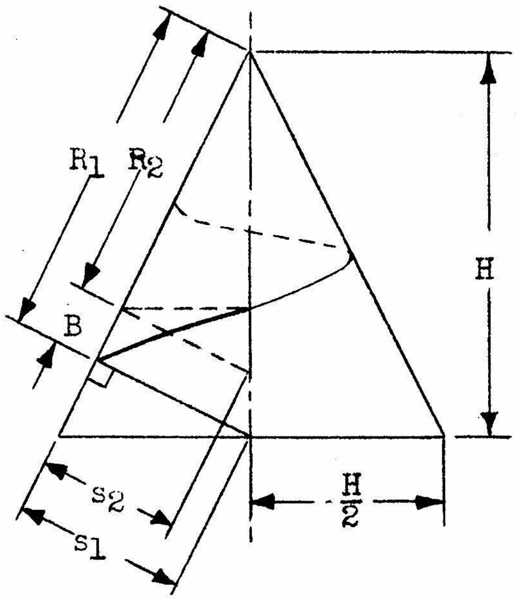


Diagram D

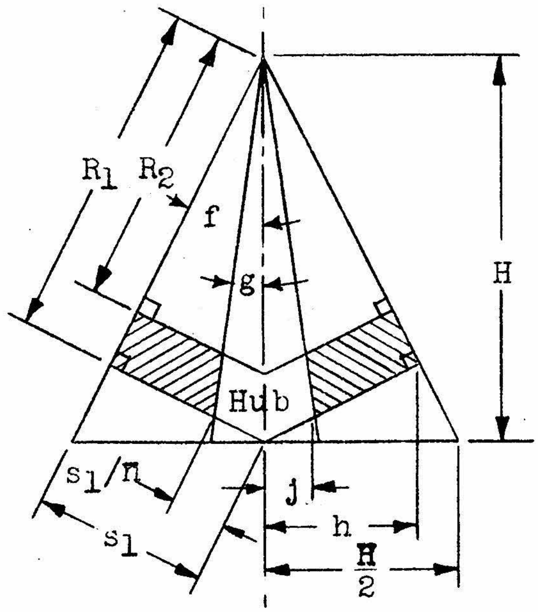


Diagram E

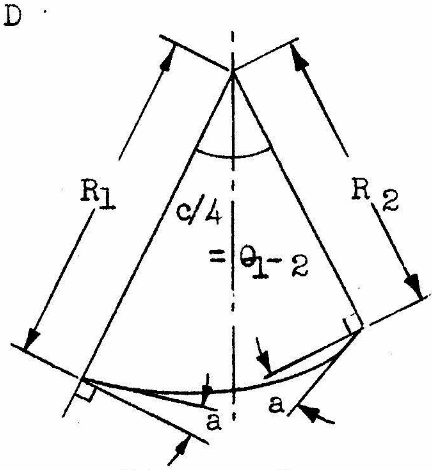


Diagram G

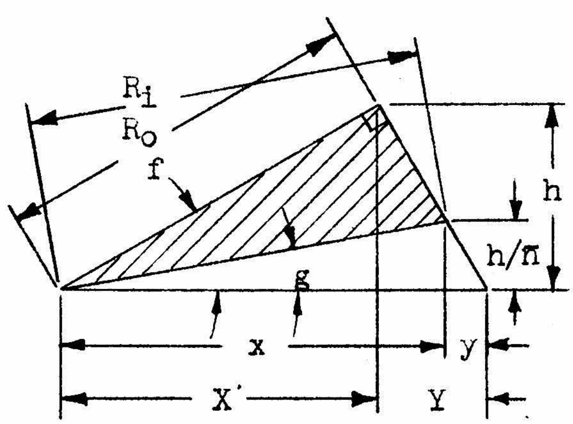


Diagram F

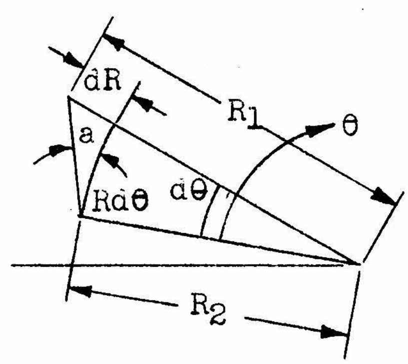


Diagram H

in straight flow are determined by the smallest inside diameter of the cowling, which is 11 inches (mechanical drawings, Assembly A, Part 1). Then, at a flow of 1200 cfm,

$$V = Q/A = 1200/(\pi 11^2/4 \cdot 12^2) = 1820 \text{ fpm.}$$

The discharge area in diffused flow is determined by the distance between the cowling and the end of the diffusing section, and the average diameter of the housing at this section. These dimensions are, respectively, 5 inches and 16.5 inches (Assembly E, Part 40). At 1200 cfm,

$$V = 1200/(\pi\, 16.5 \cdot 5/12^2) = 670 \text{ fpm.}$$

C. Horsepower Required

In calculating the horsepower required, it is convenient to determine a radius equivalent to that of a conventional constant-pitch propeller with the same capacity. From Eq. 18 and 19,

$$Q = P \cdot A \cdot S \cdot ar \cdot E, \text{ where}$$

$$A = \pi\, r_o^2(1 - (1/\pi)^2).$$

For $Q = 1200$ cfm, $S = 1500$, $ar = 1$, $E = 1$, and $P = 2r_o$ (r_o in inches),

$$PA/1728 = Q/S, \text{ or}$$

$$2r_o \pi r_o^2((\pi^2-1)/\pi^2) = 1728 \cdot 1200/1500, \text{ so that}$$

$$r_o = (1200 \cdot 1728\, \pi \,/\, 1500 \cdot 2\, (\pi^2-1))^{1/3}$$

From Eq. 25,

$$hp = 1.239 \cdot 10^{-16} E^3 r_o^2((\pi^2-1)/\pi^2)S^3(2r_o)^3.$$

Substituting for r_o,

$$hp = .0288 \quad \text{(about 1/35)}.$$

If the efficiency should be higher, say 125% (see Appendix IV-C),

$$hp = .0562 \quad \text{(about 1/18)}.$$

APPENDIX VI. DESIGN AND OPERATION OF HEATING ELEMENT

A. Element Characteristics

The appliance is rated at 15 amperes on 110 volts. Assuming that the motor draws 1 ampere, the resistance of the heating element at high heat must not be more than:

$$R = \frac{110}{15-1} = 7.86 \text{ ohms.}$$

On low heat the nominal rating is six-tenths of the maximum, and since for a given voltage, $R \propto 1/I$, the resistance of the smaller element is:

$$R_1 = 7.86\ (10/6) = 13.1 \text{ ohms}$$

The resistance of the second element which must be connected in parallel with R_1 for high heat is:

$$R_2 = \frac{R_2 \cdot R}{R_2 - R} \quad \frac{13.1 \cdot 7.86}{5.14} - 20.0 \text{ ohms.}$$

The elements are in coil form, mounted on the outer and inner deflectors (Assembly C, Parts 25H & 26H), the respective lengths being 40" and 27". The coil lengths are approximately in proportion to their resistances. To give the recommended extension of two to three times the closed length, #19 wire in .375" diameter coils is used. This has a resistance of 1.28 ohms per inch of closed length.*

- -

* "Nichrome and Other High Nickel Electrical Alloys." Driver-Harris Co. (1947), table II.

The required extension in this size is, for both R_1 and R_2,

$$L/(R/(r/in.)) = 2.6.$$

The selection of this size thus gives equivalent amounts of heat per unit length in both sections of the diffuser (on high heat, when both elements are used).

B. Air Temperature Rise

The rise in air temperature can be calculated using the specific heat of air, .24 Btu/16/^{o}F, the average specific volume, 13.4ft^3/16, and the conversion factor Btu/kwh = 3413. Substituting values for the operation of the fan-heater at high heat and one-fourth speed,

$$\frac{14 \cdot 110}{1.000} \text{ kwh} \cdot 4/1200 \text{ cfm} \cdot \frac{3413 \cdot 13.4}{.24 \cdot 60}$$

$$= 16.5^{o}F$$

C. Heating Effectiveness

Assume that a bedroom of average size (10' by 12', 8' ceiling) is to be held 30^{o} F above the ambient temperature, and that losses take place through walls of average frame construction and through four windows 2-1/2" by 4', with two air changes per hour. Using the formula,

$$Btu/hr = (.02\ NC + 1.13\ G + KA)(t - t_o)^*$$

where N = changes/hour, C = ft ^{3}/change, G = glass area, K = .25 for frame construction, and A = exposed wall and ceiling area,

$$t - t_o = \frac{1.54 \cdot 3413}{.02 \cdot 2 \cdot 960 + 1.13 \cdot 40 + .25 \cdot 432}$$

$$= 27.4^\circ.$$

---

* "Electric Heaters and Heating Devices." The General Electric Company (1948), p. 55.

APPENDIX VII. STRESS CALCULATIONS

Stress analyses at critical points are shown in the following calculations. Weight calculations have not been included.

A. Brackets

The brackets (Assembly B, Parts 6, 7, & 8) are elliptical at the smaller section between the motor and cowling (Diagram I). The section properties are:

$$A = \pi ab = .1241''^2$$

$$I_{x-x} = \frac{A}{4} a^2 = .00246''^4$$

$$I_{y-y} = \frac{A}{4} b^2 = .00062''^4$$

The maximum bending moment occurs as shown in Diagram J (propeller imbalance acts in line with cowling attachment). The maximum fiber tensile stress is:

$$f = \frac{Mc}{I} + \frac{P}{A} = \frac{5.21 \cdot 3.1}{.00246} \cdot .281 \quad \frac{+ 5.21}{.1241} \cos 18^{\circ}$$

$$= 1886 \text{ psi}.$$

The material, Alcoa aluminum alloy no. 218, with a yield strength of 23,000 psi. The factor of safety is:

$$\frac{23,000}{1886} = 12.2.$$

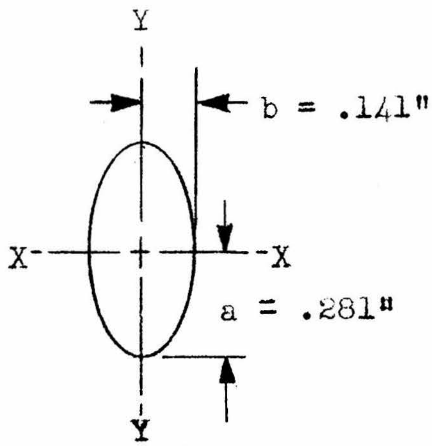


Diagram I

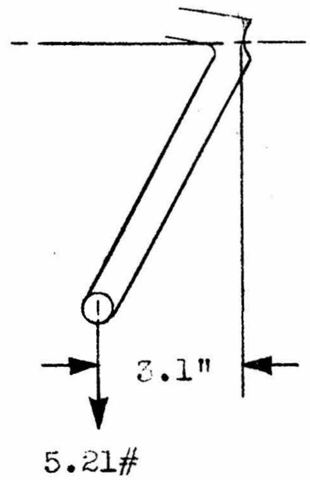


Diagram J

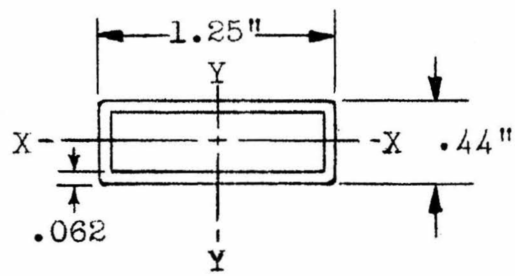


Diagram K

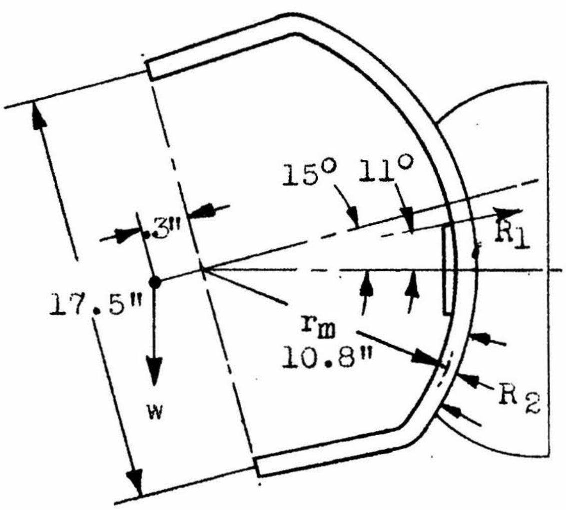


Diagram L

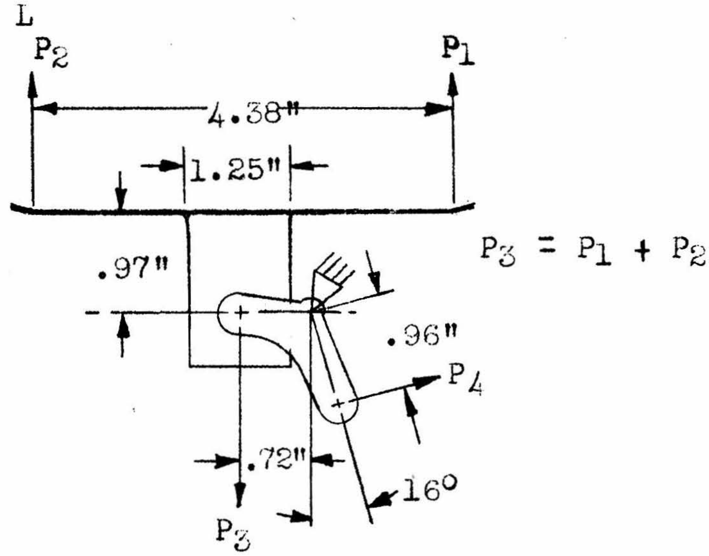


Diagram M

B. Yoke

The yoke (Assembly F, Part 43) is a rectangular tube (Diagram K). The section properties about the x-x axis are:

$A = .1951''^2$

$c = .1875''$

$Ac^2 = .00551''^4$

$I_x = .00008''^4$

$I_y = .00032''^4$

$I_T = .00591''^4$

The yoke is loaded as shown in Diagram L, with reactions at R_1 and a uniformly distributed load R_2. The loads total 11.36# at w, .3" outside the pivot axis. It is conservatively assumed that the upper arm of the yoke is clamped at R_1, and that half the load is taken by each arm. The maximum bending moment, at R_1, is:

$$M_m = \frac{11.36}{2} \cdot 12.98 = 73.9''\#$$

The material is aluminum alloy 6 1 S which has a yield point of 39,000 psi. The maximum tensile stress is:

$$f = \frac{Mc}{F} = \frac{73.9}{.00591} \cdot .218 = 2730 \text{ psi.}$$

The factor of safety is

$$\frac{39,000}{2730} = 14.3$$

The deflection of the top arm is the sum of the moments of the M/EI curve about the end point of the arm. This is

$$d = \frac{1}{59{,}100}(400 + 4020 + 1050)$$

$$= .093".$$

C. Yoke Clamp

The yoke-clamping device comprises a clamp spring and bell-crank linkage (Assembly G2). The load on the spring is a moment around R_1, Diagram L. Assuming that R_2, Diagram L, acts as a concentrated load 18° from the horizontal, the value of P_1, Diagram M, can be obtained. The distance between P_1 and P_2 is:

$$2.19 + 10.8 \tan 18^\circ = 5.69"$$

The moment where the lower arm of the yoke leaves the base is:

$$M = 5.68 \cdot 6.25 = 35.5"\#. \quad \text{Then}$$

$$P_1 = P_2 = \frac{73.9 + 35.5}{5.69} = 19.3^{\#}$$

The clamp spring (Part 57) is S.A.E. 1070 steel, with an allowable stress of 75,000 psi for light (non-repetitive) loading. The actual deflection of each leg is:

$$d = \frac{4PL^3}{Ebt^3}, \text{ where } P = \text{load} = 19.3\#,$$

L = length = 1.62", $E = 30 \cdot 10^6$ psi , b = width = 1.12", t = thickness = .062".

$$d = \frac{4 \quad 19.3 \quad 1.62^3}{30 \cdot 10^6 \cdot 1.62 \cdot .062^3} = .0401''.$$

The maximum allowable deflection is

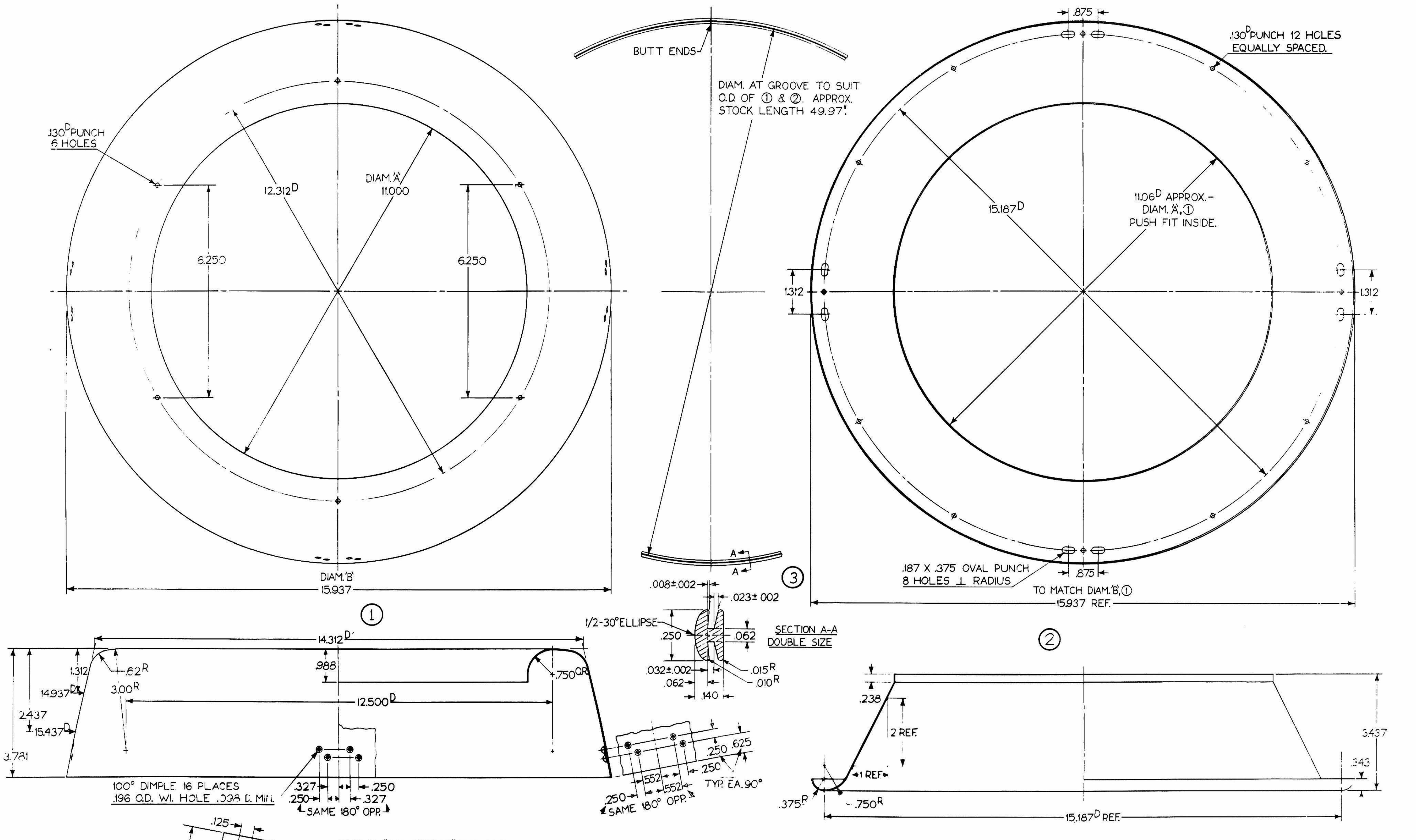
$$d_a = \frac{2\ S_b\ L^2}{3\ Et}, \text{ where } S_b = \text{allowable stress.}$$

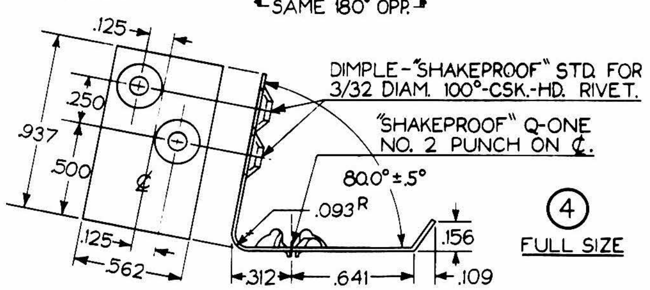
$$d_a = \frac{2 \cdot 75{,}000 \cdot 1.62^2}{3 \cdot 30 \cdot 10^6 \cdot .062} = .0704''.$$

The factor of safety is 1.75.

The load P_4 transmitted to the clamp screw is

$$\frac{(19.3 - 19.3)\ .72}{.96} = 29.0^{\#}.$$



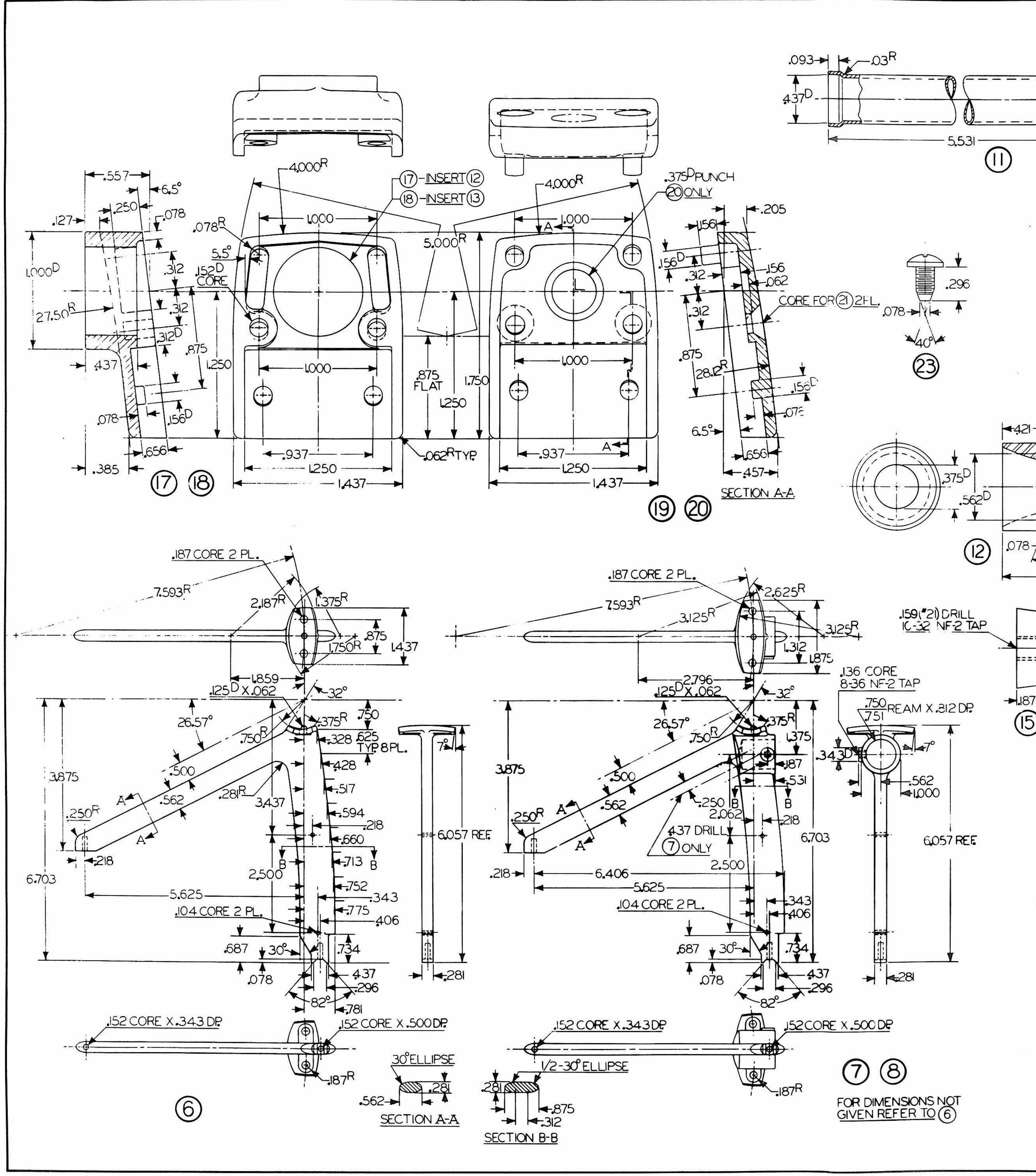


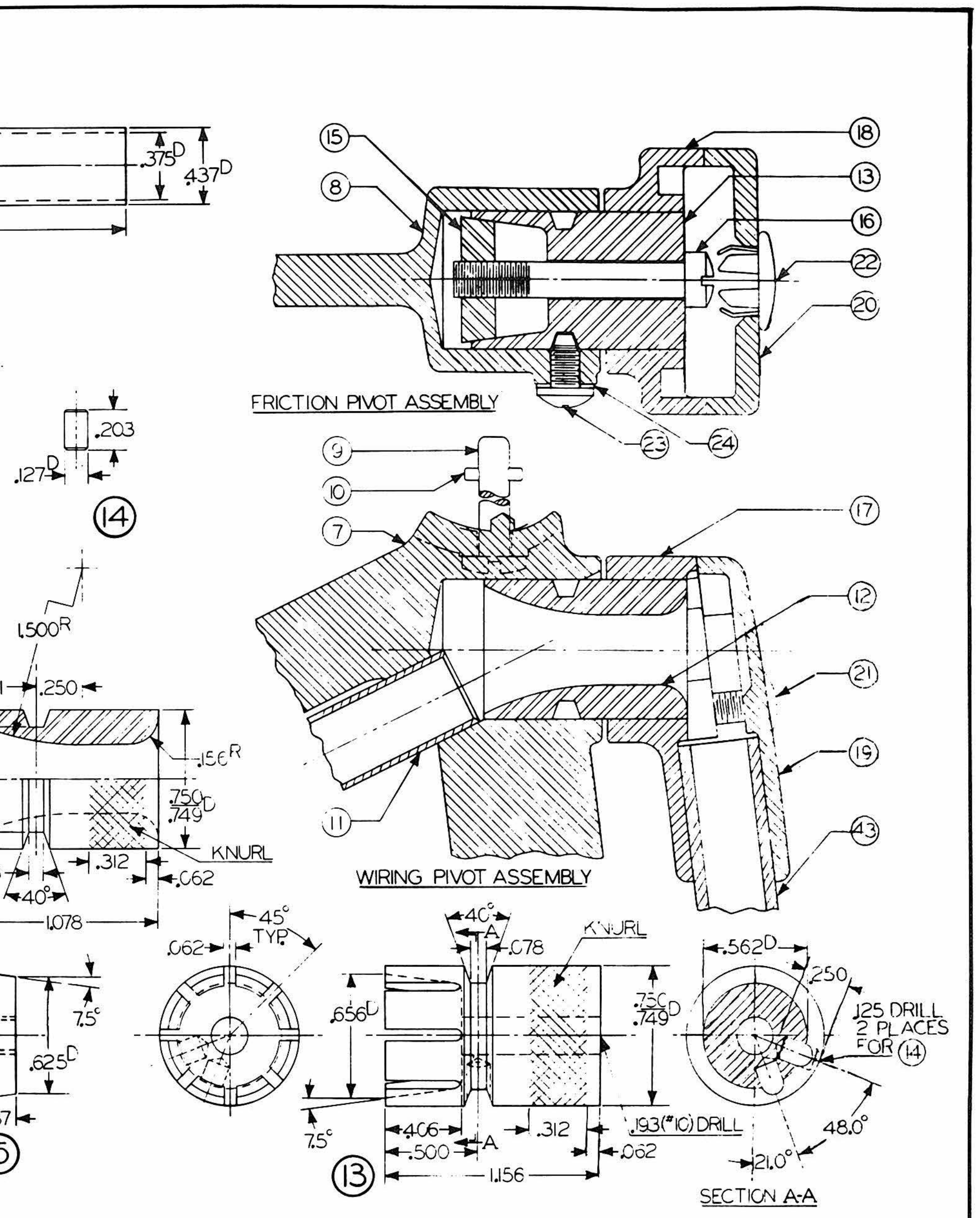
NO.	PART NAME	QTY	MATERIAL & SPECIFICATIONS	FINISH
1	OUTER COWL	1	.032 AL. 3S-O SPUN OR DRAWN	PAINT AFTER ASSEM.
2	INNER COWL	1	"	
3	JOINT BAND	1	AL. 3S-F EXTRUDED	
4	ASSEMBLY CLIP	8	.021 SPRING STEEL "SHAKEPROOF" (SPECIAL)	—
5	CLIP RIVET	16	3/32D. X 3/16 AL. 3S-F 100°-CSK. HD.	—

PORTABLE AIR CIRCULATOR | ASSEMBLY A | COWLING

SCALE 1/3 EXCEPT AS NOTED. LINEAR DIMENSIONS IN INCHES. TOLERANCES UNLESS SPECIFIED: 2 DECIMAL PLACES. ±.01"; 3 PLACES. ±.005".

Harry L. Masser | Industrial Design Section, California Institute of Technology | 1949-1950



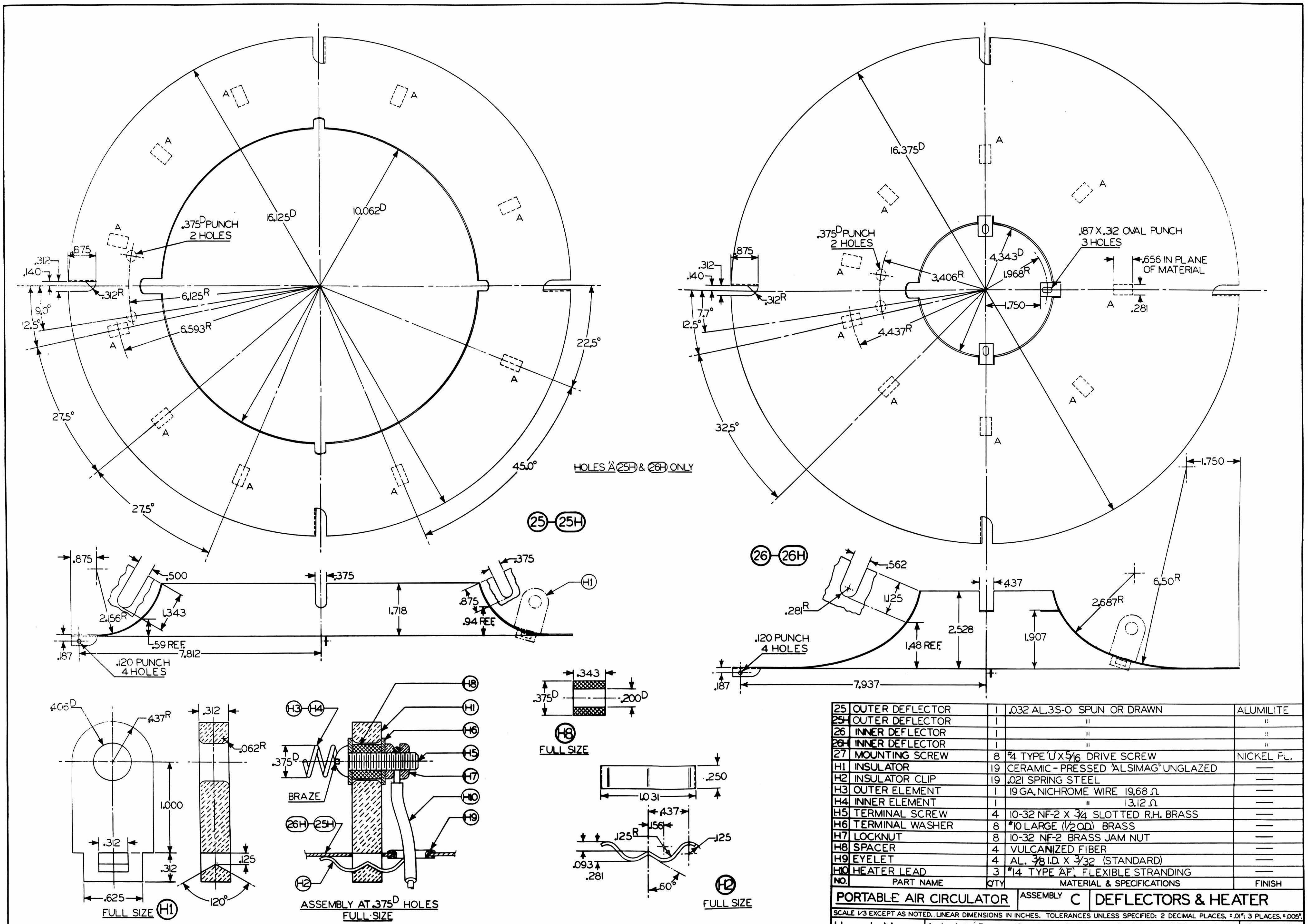


NO.	PART NAME	Q'TY	MATERIAL & SPECIFICATIONS	FINISH
6	BRACKET-PLAIN	2	AL. "ALCOA 218" (8%MG.) DIE CAST	ALUMILITE
7	BRACKET-WIRING PIVOT	1	"	"
8	BRACKET-FRICTION PIVOT	1	"	"
9	STUD	8	"SHAKEPROOF" #98-0S-2-53	NICKEL PL.
10	CROSS PIN	8	"SHAKEPROOF" #98-2-CP	CAD. PLATE
11	WIRING TUBE	1	AL. 3S-F EXTRUDED	ALUMILITE
12	PIVOT-WIRING	1	FREE-CUTTING STEEL	——
13	PIVOT-FRICTION	1	"	——
14	STOP PIN	2	#30 (.127) DRILL ROD	——
15	EXPANSION PLUG	1	FREE-TURNING BRASS	——
16	EXPANSION SCREW	1	10-32 NF-2 X 1-1/4 SLOTTED ROUND HD. STEEL	CAD. PLATE
17	YOKE CAP-WIRING	1	AL. "ALCOA 218" (8%MG.) DIE CAST	ALUMILITE
18	YOKE CAP-FRICTION	1	"	"
19	CAP COVER-WIRING	1	"	"
20	CAP COVER-FRICTION	1	"	"
21	CAP SCREW	4	8-32 TYPE "C" X 1/2 PHIL. OVAL HD. SELF-TAP.	NICKEL PL.
22	SNAP HOLE PLUG	1	3/8D. X 7/32-STANDARD	"
23	PIVOT RETAINING SCREW	2	8-36 NF-2 PHIL. BINDING HEAD STEEL	"
24	LOCKWASHER	2	"SHAKEPROOF" #1208	CAD. PLATE

PORTABLE AIR CIRCULATOR | ASSEMBLY B | BRACKETS & PIVOTS

SCALE 1/3 EXCEPT AS NOTED. LINEAR DIMENSIONS IN INCHES. TOLERANCES UNLESS SPECIFIED: 2 DECIMAL PLACES, ±.01"; 3 PLACES, ±.005".

Harry L. Masser | Industrial Design Section, California Institute of Technology | 1949-1950

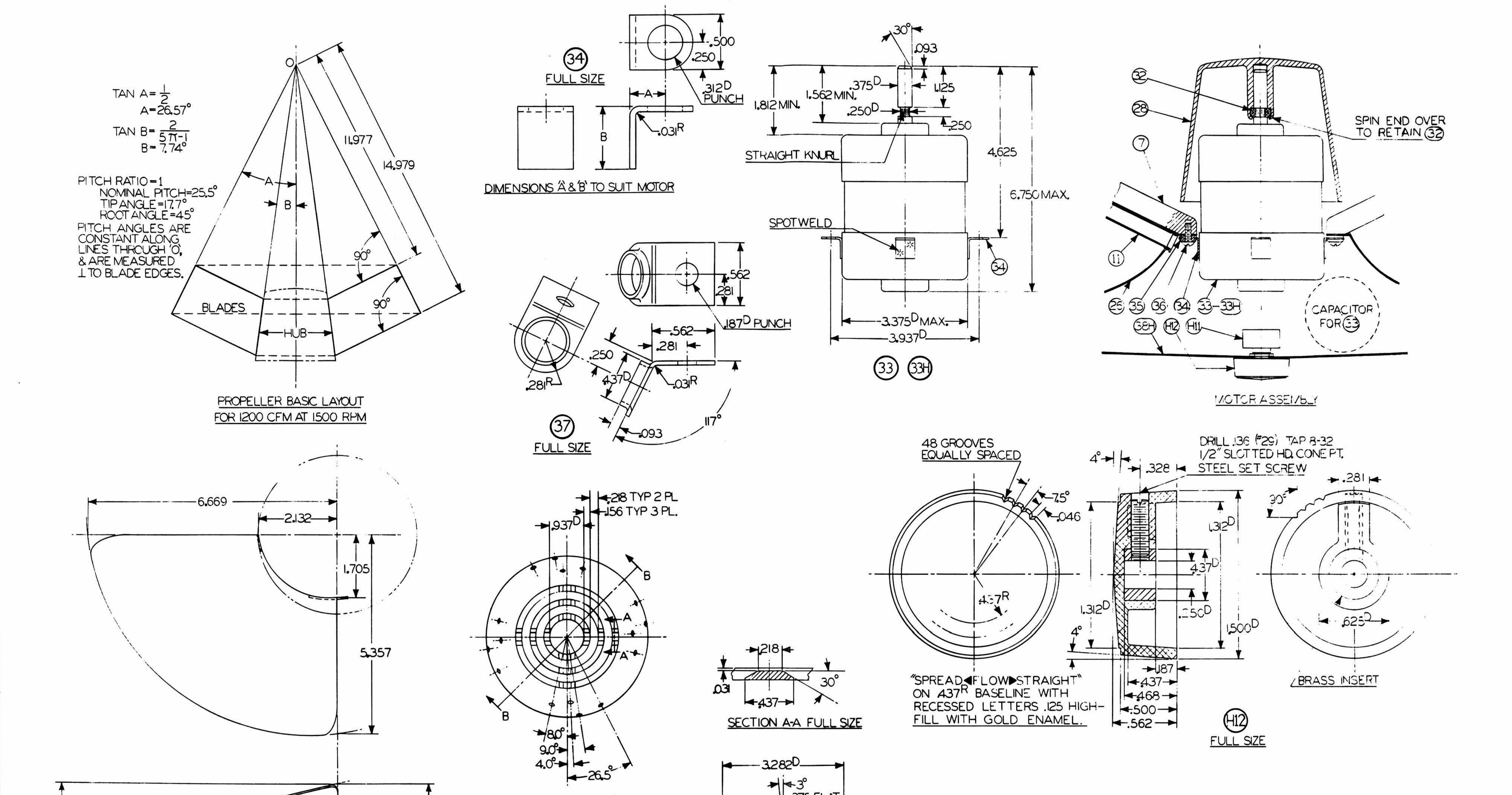


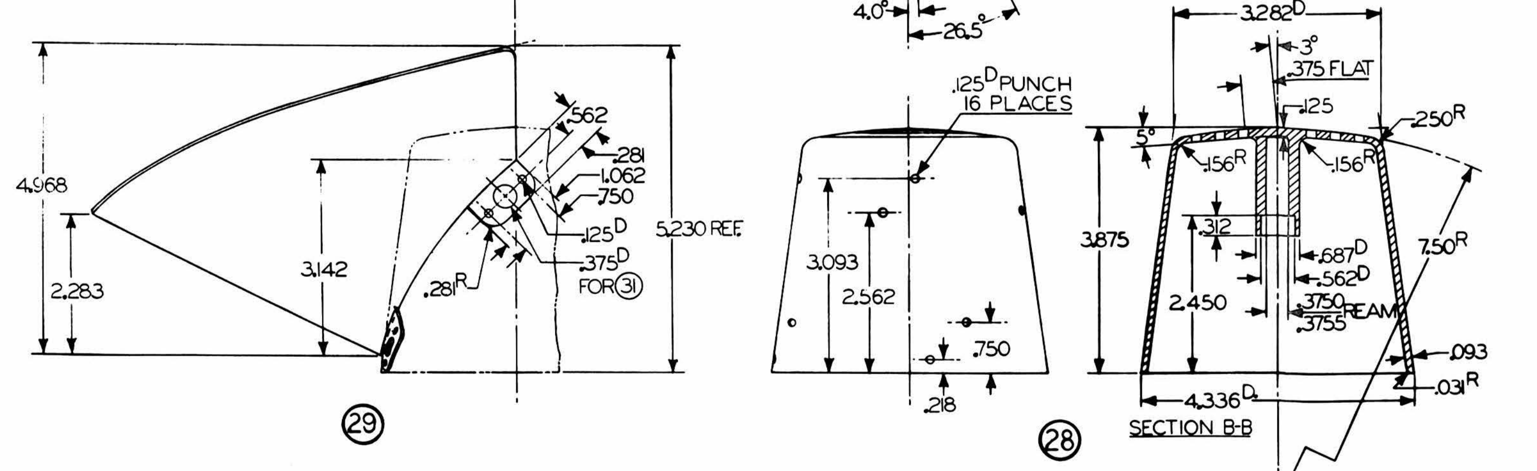
NO.	PART NAME	QTY	MATERIAL & SPECIFICATIONS	FINISH
25	OUTER DEFLECTOR	1	.032 AL. 3S-O SPUN OR DRAWN	ALUMILITE
25H	OUTER DEFLECTOR	1	"	"
26	INNER DEFLECTOR	1	"	"
26H	INNER DEFLECTOR	1	"	"
27	MOUNTING SCREW	8	#4 TYPE 'U' X 5/16 DRIVE SCREW	NICKEL PL.
H1	INSULATOR	19	CERAMIC - PRESSED "ALSIMAG" UNGLAZED	—
H2	INSULATOR CLIP	19	.021 SPRING STEEL	—
H3	OUTER ELEMENT	1	19 GA. NICHROME WIRE 19.68 Ω	—
H4	INNER ELEMENT	1	" 13.12 Ω	—
H5	TERMINAL SCREW	4	10-32 NF-2 X 3/4 SLOTTED R.H. BRASS	—
H6	TERMINAL WASHER	8	#10 LARGE (1/2 O.D.) BRASS	—
H7	LOCKNUT	8	10-32 NF-2 BRASS JAM NUT	—
H8	SPACER	4	VULCANIZED FIBER	—
H9	EYELET	4	AL. 3/8 I.D. X 3/32 (STANDARD)	—
H10	HEATER LEAD	3	#14 TYPE "AF" FLEXIBLE STRANDING	—

PORTABLE AIR CIRCULATOR | ASSEMBLY C | DEFLECTORS & HEATER

SCALE 1/3 EXCEPT AS NOTED. LINEAR DIMENSIONS IN INCHES. TOLERANCES UNLESS SPECIFIED: 2 DECIMAL PLACES, ±.01"; 3 PLACES, ±.005".

Harry L. Masser | Industrial Design Section, California Institute of Technology | 1949-1950



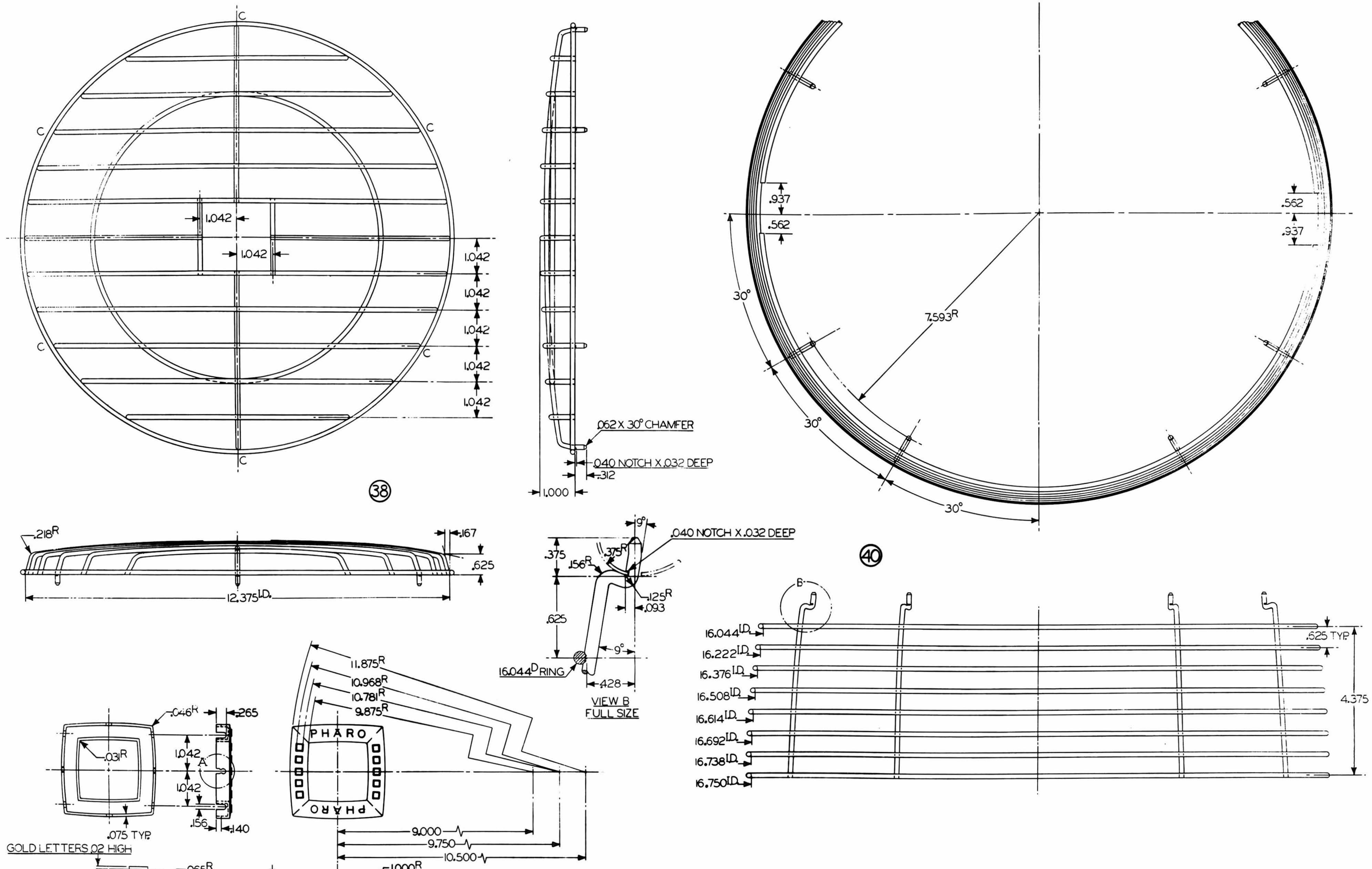


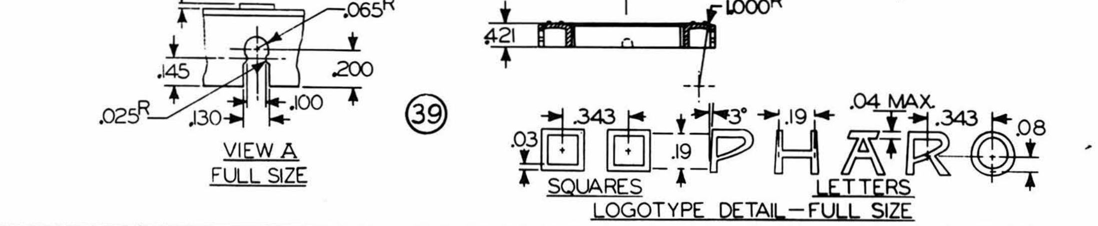
NO.	PART NAME	QTY	MATERIAL & SPECIFICATIONS	FINISH
28	HUB	1	AL. "ALCOA 218" (8%MG.) DIE CAST	ALUMILITE
29	BLADE	4	.064 AL. 24S-T6 COMMERCIAL BRIGHT 2 SIDES	
30	BLADE RIVET	16	AL. 3S-F 1/8"D. X 1/4", ROUND HEAD	—
31	BALANCE WEIGHT	—	1/64" LEAD SHEET X 3/8"D.	—
32	DRIVE GROMMET	1	NEOPRENE 1/4"I.D. X 9/16"O.D. (STANDARD FOR 3/8" HOLE)	
33	MOTOR	1	1/15 HP 4 POLE 3 LEAD, 5 MFD. CAPACITOR	MFR'S STD.
33H	MOTOR	1	1/15 HP REVERSIBLE SHADED POLE	MFR'S STD.
H11	REVERSING SWITCH	1	S.P.D.T. ROTARY 3 AMP., 1/4" SHAFT, 3/8" BUSHING & NUT	
H12	SWITCH KNOB	1	NUT BROWN MOLDED PHENOLIC WITH SET SCREW	
34	MOTOR MOUNT ANGLE	4	18 GA. C.R. STEEL — BY MOTOR MFR.	AS MOTOR
35	MOTOR MOUNT GROMMET	4	RUBBER-STANDARD FOR 5/16" HOLE	—
36	MOTOR MOUNT SCREW	4	8-32 TYPE "C" X 1/2" PHIL. TRUSS HD. SELF TAP.	CAD. PLATE
37	TUBE CLIP	1	22 GA. C.R. STEEL	CAD. PLATE

PORTABLE AIR CIRCULATOR	ASSEMBLY D	PROPELLER & DRIVE

SCALE 1/3 EXCEPT AS NOTED. LINEAR DIMENSIONS IN INCHES. TOLERANCES UNLESS SPECIFIED: 2 DECIMAL PLACES, ±.01"; 3 PLACES, ±.005".

Harry L. Masser	Industrial Design Section, California Institute of Technology	1949-1950





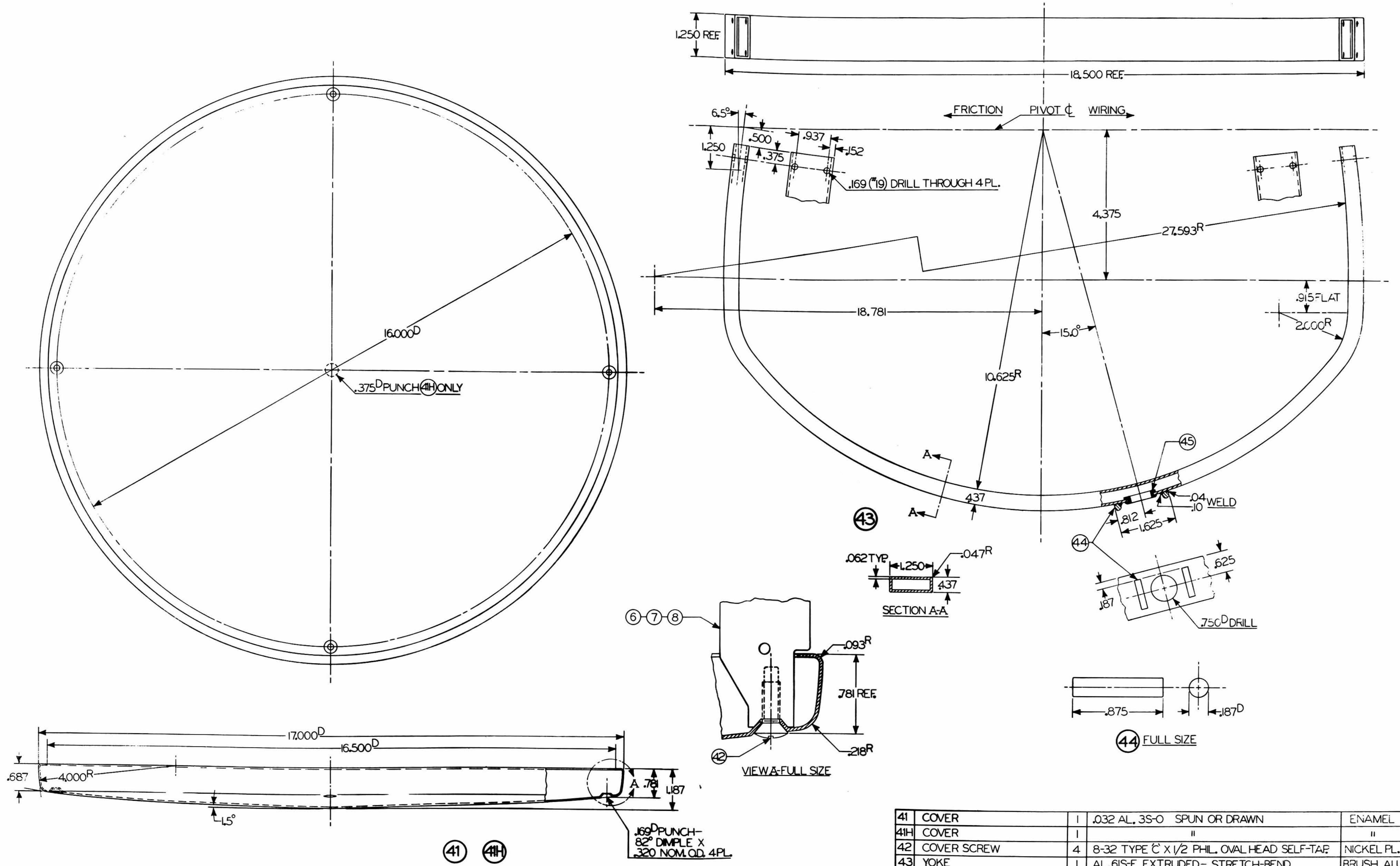
38 40 ALL INTERSECTIONS EXCEPT 'C' PROJECTION WELDED

NO.	PART NAME	QTY	MATERIAL & SPECIFICATIONS	FINISH
38	END GRILLE	1	1/8 RD. STEEL WIRE COMMERCIAL BRIGHT	CR. PLATE
39	NAME PLATE	1	WALNUT-BROWN MOLDED "LUSTREX LX" STYRENE	
40	SIDE GRILLE	1	1/8 RD. STEEL WIRE COMMERCIAL BRIGHT	CR. PLATE

PORTABLE AIR CIRCULATOR | ASSEMBLY E | GRILLES & NAMEPLATE

SCALE 1/3 EXCEPT AS NOTED. LINEAR DIMENSIONS IN INCHES. TOLERANCES UNLESS SPECIFIED: 2 DECIMAL PLACES, ±.01"; 3 PLACES, ±.005".

Harry L. Masser | Industrial Design Section, California Institute of Technology | 1949-1950

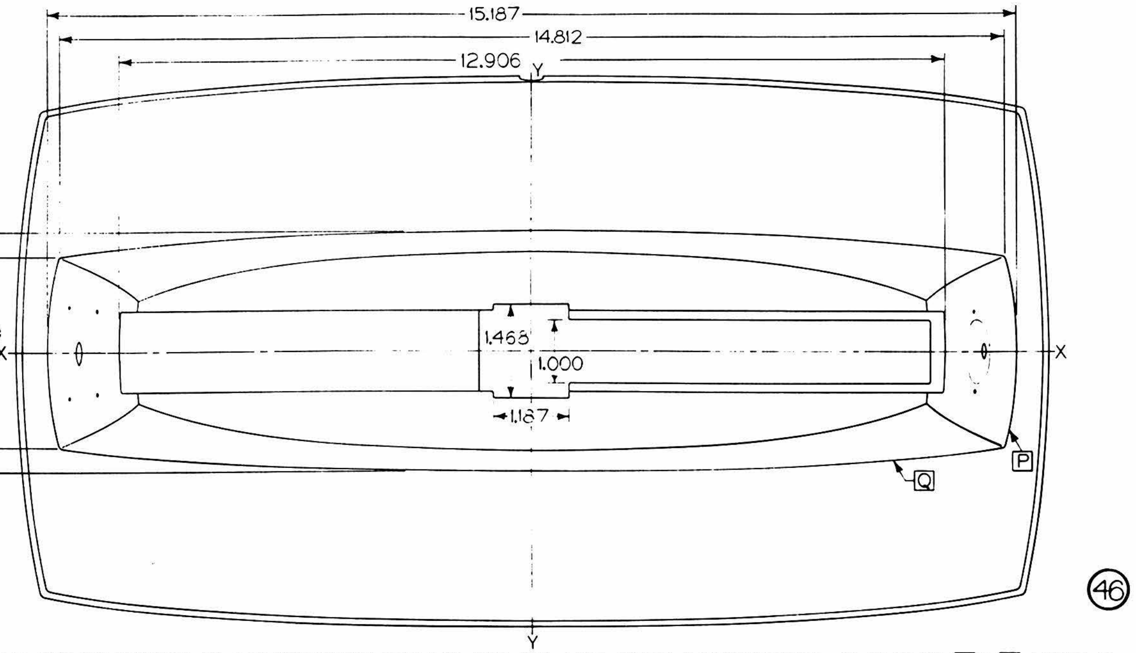


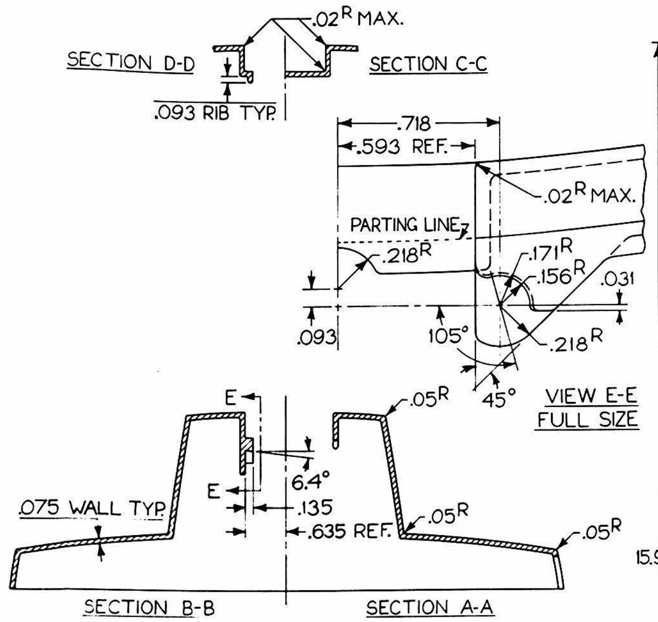
41	COVER	1	.032 AL. 3S-O SPUN OR DRAWN	ENAMEL
41H	COVER	1	"	"
42	COVER SCREW	4	8-32 TYPE 'C' X 1/2 PHIL. OVAL HEAD SELF-TAP.	NICKEL PL.
43	YOKE	1	AL. 61S-F EXTRUDED- STRETCH-BEND	BRUSH ALUM.
44	YOKE STOP	2	AL. 3S-F ROD	———
45	YOKE GROMMET	1	RUBBER-STANDARD FOR 3/4" HOLE	———
NO.	PART NAME	Q'TY	MATERIAL & SPECIFICATIONS	FINISH

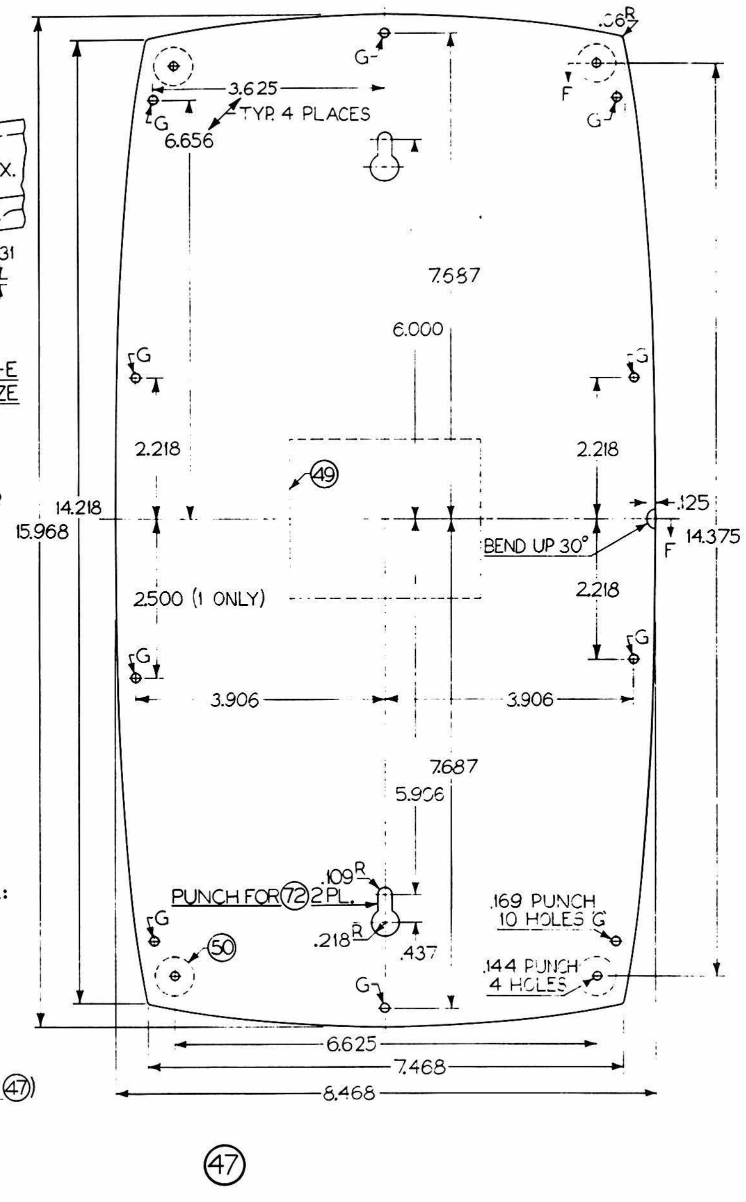
PORTABLE AIR CIRCULATOR | ASSEMBLY F | **END COVER & YOKE**

SCALE 1/3 EXCEPT AS NOTED. LINEAR DIMENSIONS IN INCHES. TOLERANCES UNLESS SPECIFIED: 2 DECIMAL PLACES, ±.01"; 3 PLACES, ±.005".

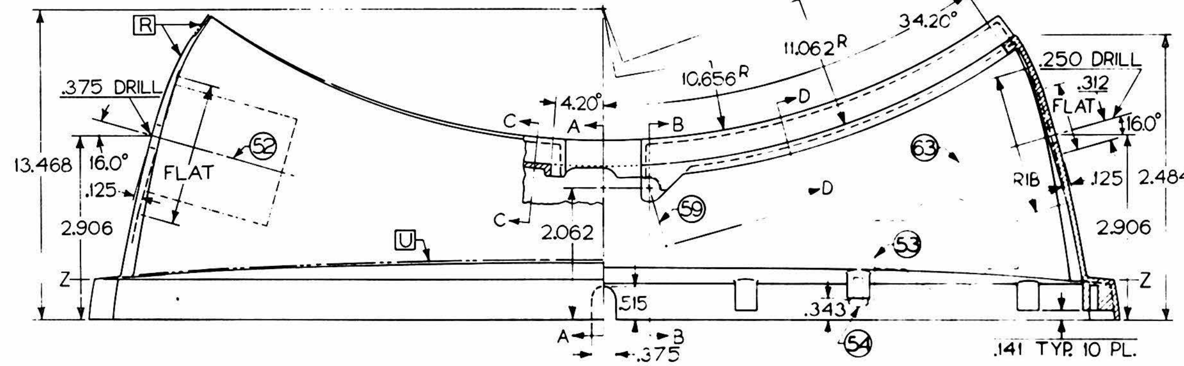
Harry L. Masser | Industrial Design Section, California Institute of Technology | 1949-1950

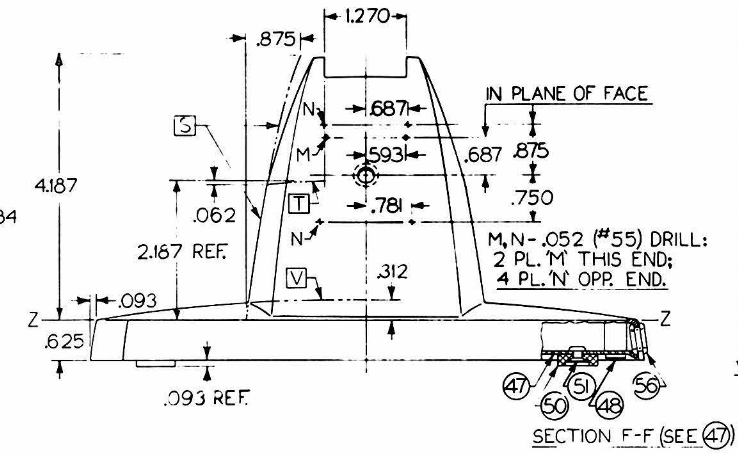


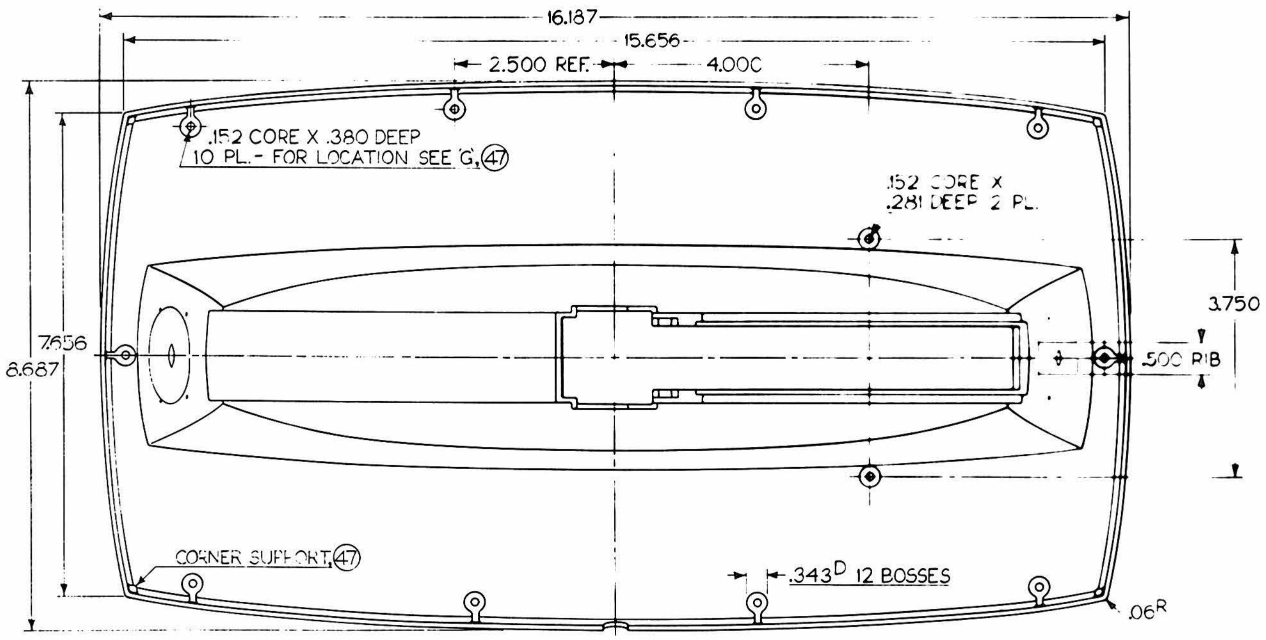


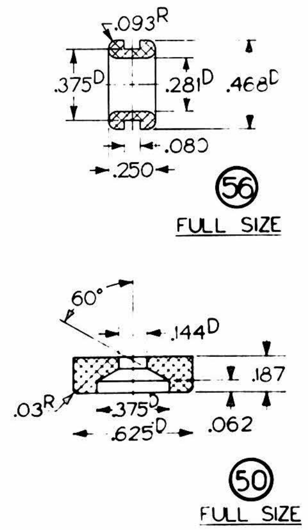


1. CURVES [P] & [Q] SHOWN AT INTERSECTION WITH PLANE Z-Z, NOT TRUE INTERSECTION. 2. CURVES [R] & [S] MOVE IN VERTICAL PLANES ⊥ [P] & [Q]. CURVE [T] MOVES IN PLANE HORIZONTALLY AND RADIALLY ⊥ TOP ARC OF (46). 3. CURVES [R], [S] & [T] CONSTANT IN FORM FROM BASELINES (P, Q & ARC, RESPECTIVELY) OUT TO MUTUAL INTERSECTIONS. 4. CURVE [U] ABOVE ℄ X-X & [V] ABOVE ℄ Y-Y ARE .312" ABOVE PLANE Z-Z AT CENTER AND BLENDED WITHOUT INTERSECTIONAL RIDGES TO LEVEL OF Z-Z AT PERIPHERY.







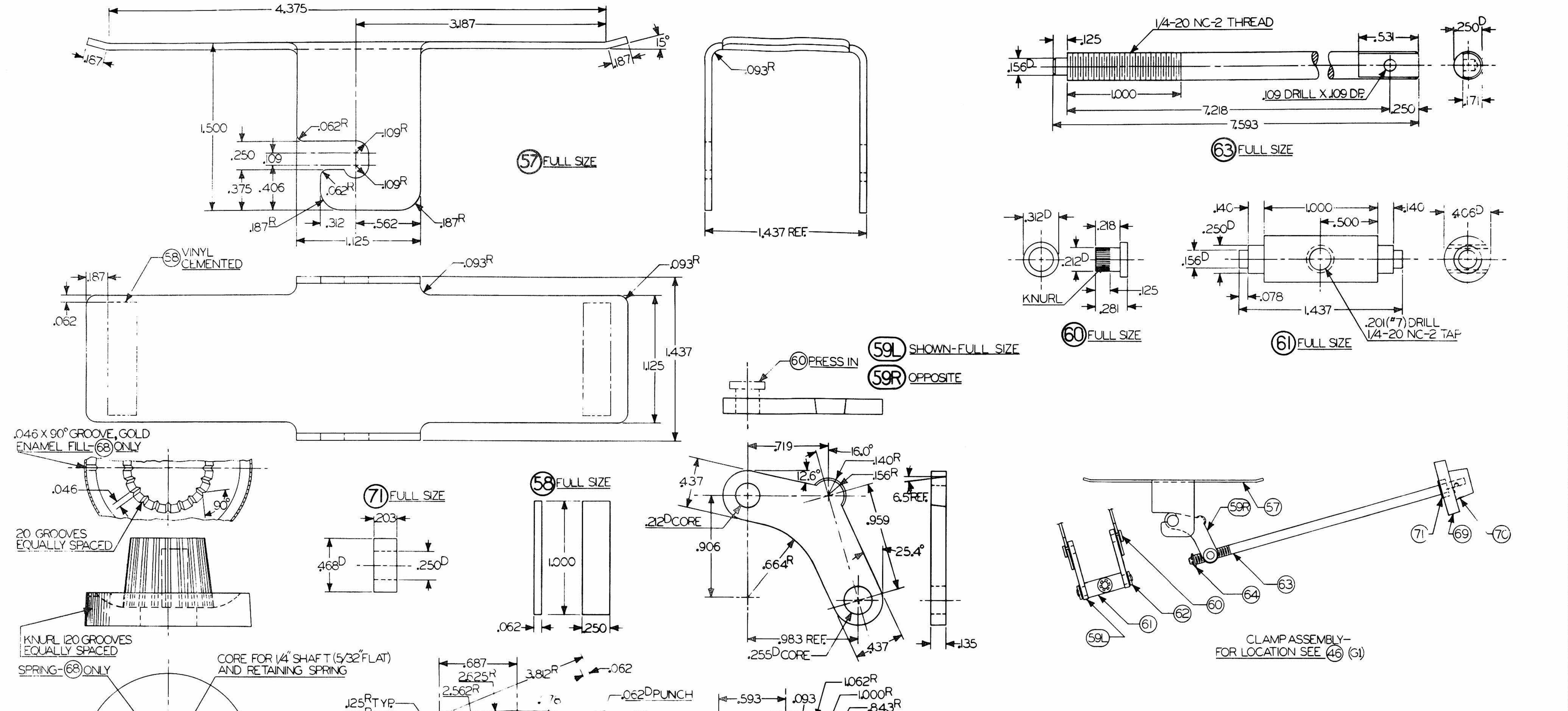


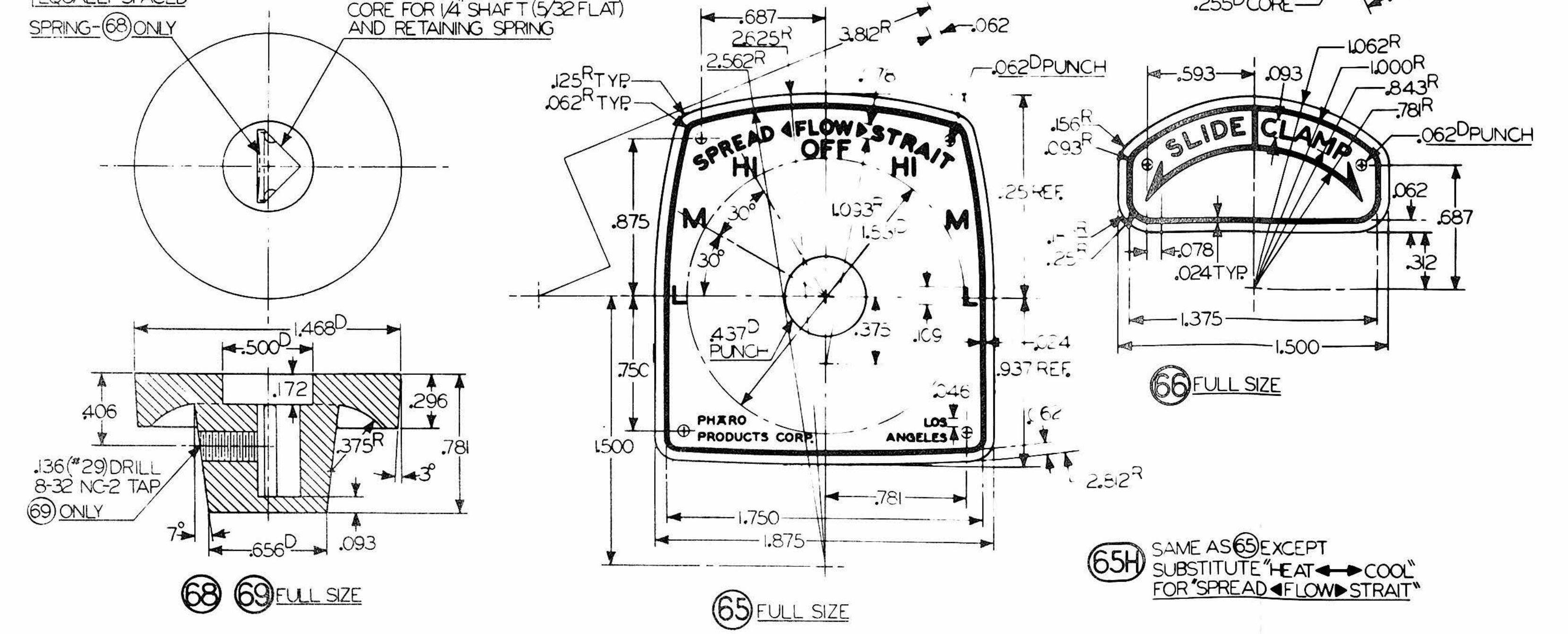
NO.	PART NAME	Q'TY	MATERIAL & SPECIFICATIONS	FINISH
46	BASE	1	AL. "ALCOA 218" (8% MG.) DIE CAST	ENAMEL
47	BASEPLATE	1	18 GA. C.R. STEEL	ENAMEL
48	BASEPLATE SCREW	10	8-32 TYPE 'C' X 3/8 PHIL. BINDING HD. SELF TAP.	CAD. PLATE
49	DECALCOMANIA	1	FOR MANUFACTURER'S DATA	—
49H	DECALCOMANIA	1	"	—
50	FOOT	4	NUT BROWN MOULDED "LUSTREX LX" STYRENE	—
51	FOOT RIVET	4	9/64 D. X 1/4 AL. SEMI-TUBULAR (5/16 D. 120° HD.)	—
52	MOTOR SWITCH	1	3 AMP. ROTARY 7 POS. (30° EA.) 1/4 D. SHAFT WITH FLAT.	
52H	MOTOR SWITCH	1	15 AMP. 3/8 D. BUSHING & LOCKNUT - SEE WIRING DIAGRAM	
53	SPEED CONTROL REACTOR	1	2 TAP (80% & 60% FULL SPEED)	MOUNTING BRACKET
53H	SPEED CONTROL REACTOR	1	3 TAP (80% & 60% FULL SPEED & MINIMUM)	TO SUIT PART NO. 46
54	CONTROLLER MOUNT SCREW	2	8-32 TYPE 'C' X 1/4 PHIL. BINDING HD. SELF TAP.	CAD. PLATE
55	CORD SET	1	#18 POSJ-32 8 FT. WITH PLUG	BROWN
55H	CORD SET	1	#14 "G.E. PREEN-X" 8 FT. WITH PLUG	BROWN
56	CORD BUSHING	1	OLIVE GREEN VULCANIZED FIBER	—

PORTABLE AIR CIRCULATOR | ASSEMBLY G1 | BASE-BODY & ELECTRICAL

SCALE 1/3 EXCEPT AS NOTED. LINEAR DIMENSIONS IN INCHES. TOLERANCES UNLESS SPECIFIED: 2 DECIMAL PLACES. ±.01"; 3 PLACES. ±.005".

Harry L. Masser | Industrial Design Section, California Institute of Technology | 1949-1950





57	CLAMP SPRING	1	S.A.E. 1070 STEEL PRETEMPERED	BRUSH CR. PL.
58	SPRING PAD	2	SHEET FELT – HARD WHITE	———
59L	BELL CRANK	1	AL. "ALCOA 218" (8% MG.) DIE CAST	———
59R	BELL CRANK	1	"	———
60	CRANK STUD	2	FREE-CUTTING STEEL	CAD. PLATE
61	CLAMP NUT	1	"	"
62	WASHER	2	1/8" S.A.E. STANDARD (3/8" O.D.)-STEEL	"
63	CLAMP SCREW	1	1/4" ROUND C.R. STEEL	"
64	RETAINING RING	1	TRUARC #5105-15	———
65	SWITCH ESCUTCHEON	1	.020 AL. 2S-O. ETCH DESIGN AND	———
65H	SWITCH ESCUTCHEON	1	FILL WITH WALNUT-BROWN ENAMEL,	———
66	CLAMP ESCUTCHEON	1	ALUMILITE BACKGROUND.	———
67	ESCUTCHEON SCREW	6	#00 TYPE 'U' X 1/8 DRIVE SCREW	NICKEL PL.
68	SWITCH KNOB	1	WALNUT-BROWN MOLDED PHENOLIC-WITH RETAINING SPRING	
69	CLAMP KNOB	1	" -WITHOUT SPRING	
70	SETSCREW	1	8-32 X 3/8 FULL DOG PT. SOC. HEAD STEEL	CAD. PLATE
71	CLAMP KNOB BEARING	1	FREE-TURNING BRASS	———
72	WALL MOUNT SCREW	2	#11 X 1-1/2 SLOTTED RD. HD. STEEL WOODSCREW	CAD. PLATE
NO.	PART NAME	Q'TY	MATERIAL & SPECIFICATIONS	FINISH

PORTABLE AIR CIRCULATOR | ASSEMBLY G2 | BASE-FITTINGS

SCALE 1/3 EXCEPT AS NOTED. LINEAR DIMENSIONS IN INCHES. TOLERANCES UNLESS SPECIFIED: 2 DECIMAL PLACES, ±.01"; 3 PLACES, ±.005".

Harry L. Masser | Industrial Design Section, California Institute of Technology | 1949-1950