LEAD-OFF TALK ON COMPRESSOR, TURBINE, AND TURBINE COOLING RESEARCH

by Oscar W. Schey

For efficient operation and maximum performance the turbine inlet temperatures of turbojet engines are limited to 1300° F and the pressure ratios to 8 or 10 to 1, whereas turbine-propeller engines for optimum performance should operate at or near stoichiometric temperatures and pressure ratios of 30 and possibly as high as 60:1. Both types of engines require very efficient and reliable components and a large volume flow per unit frontal area.

The conventional turbojet or turbine-propeller engine utilizes for combustion about 1/3 of the air passing through the engine. The limiting temperature of the turbine blades, disks, burners, and other turbine parts prevents the burning of all the air. If means could be provided for cooling the parts of the turbine exposed to the hot gases as is done with reciprocating engine parts, a much larger amount of the air handled by the engine could be utilized and greatly increased power could be obtained.

The increase in specific power to be obtained by increasing the turbine inlet temperature is shown in figure 1. At a temperature of 1500° F the power is 120 horsepower, at 2000 it is 210, at 2500 it is 312 and at 3000 it is 427. This chart is for a turbine-propeller engine having compressor and turbine efficiencies of 90 percent, burner efficiency of 95 percent, and a pressure drop of 5 percent through the burner, and no heat loss. The brake specific fuel consumption corresponding to these powers and temperatures is 0.44 at 1500° F, 0.36 at 2000° F, 0.325 at 2500° F, and 0.305 at 3000° F. It will be noted that the power can be increased approximately 3 times and the fuel consumption reduced by approximately 1/3 by increasing the temperature from 1500° F to 3000° F. In order to obtain these improvements in economy and power, the pressure ratio of the engine must be increased. The pressure ratio giving maximum power at each temperature is also given on the figure. At a temperature of 1500° F, it is 8, at 2000° F it is 13, at 2500° F it is 19, and at 3000° F it is 26.

To obtain these very promising results, compressors of high pressure and turbines of high expansion ratio are essential. Much of our compressor and turbine research has been directed with this end in view. The maximum pressure ratio of any turbine-propeller engine now being built for service is about 8 and the lowest about 5. The compressors for these engines are quite bulky; the centrifugal has a large diameter and the axial is long. In considering engines of the high pressure ratios associated with these performances, one would expect them to become so bulky as to be impractical, particularly in regards to length. Fortunately, good progress has been made during the last few years in increasing the pressure ratio per stage of both axial and centrifugal. There has also been a marked increase in flow capacity and efficiency. Very encouraging results have also been obtained on supersonic compressors. The results on our turbine research indicate that very high expansion ratios can be obtained with good efficiency and that cooling can be provided for operating at temperatures much higher than used in current engines. Cooling should also improve reliability and make possible the use of non-strategic materials. The application of the results of this research to new power plants should make it possible to build engines of pressure ratios of 16 or 20:1, without any appreciable increase in bulk or weight with power output several times that of present engines and with appreciably improved economy. Incidentally, these results of this research should also enable us to build more compact turbojet engines. My associates will discuss the results obtained in our investigations on turbine cooling, turbines, and compressors that are essential in obtaining these high performances. Mr. Ellerbrock will now tell you about our work on turbine cooling or high temperature operation.

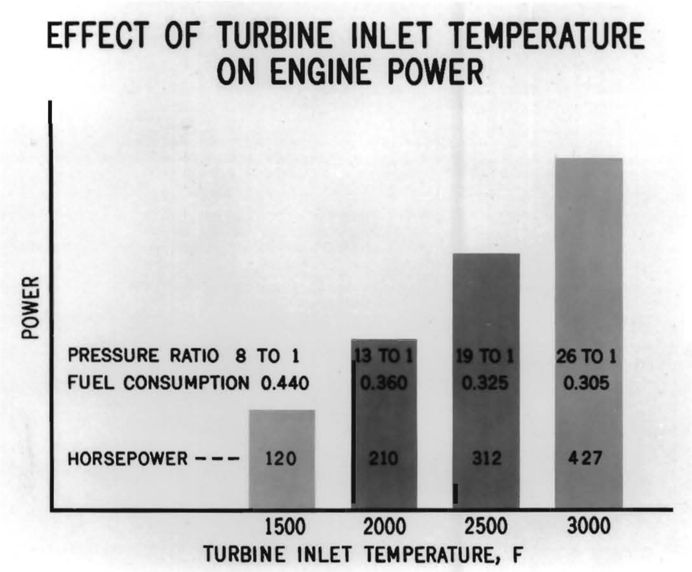
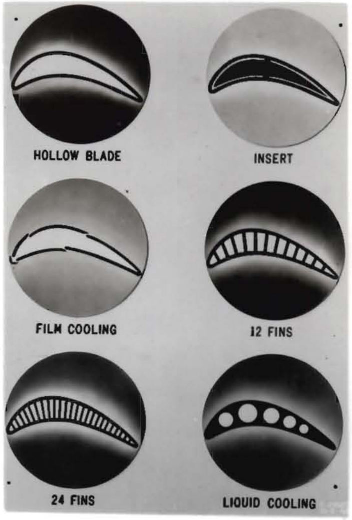


Figure 1.



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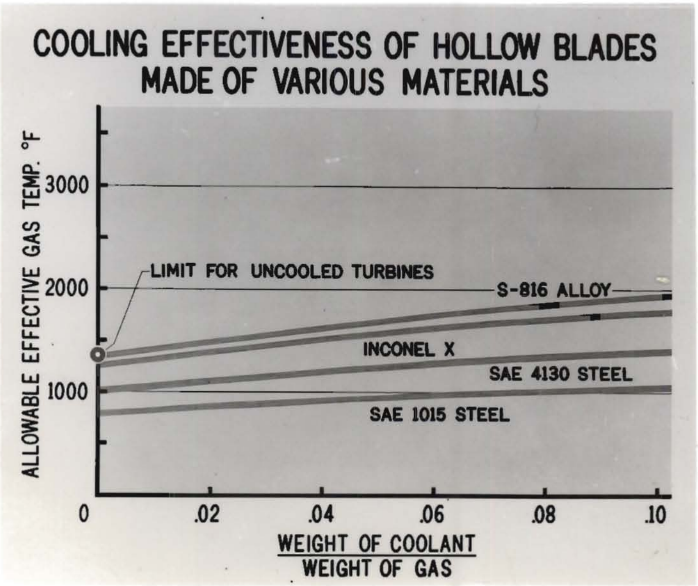

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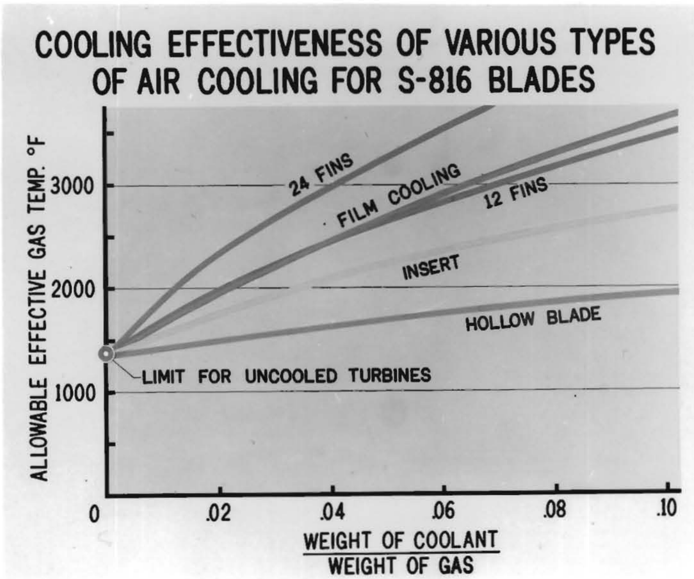

Figure 4.

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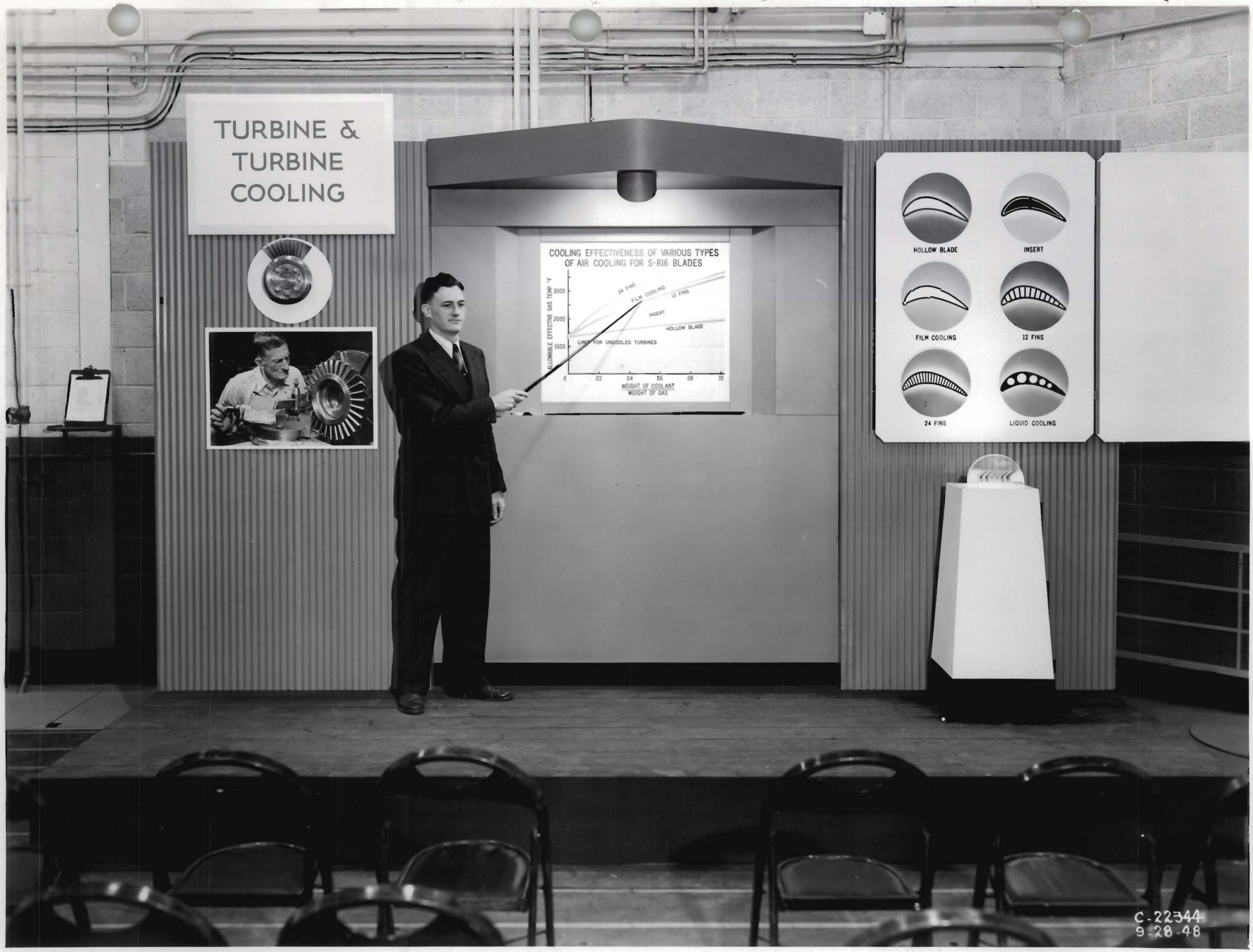
TURBINE &
TURBINE
COOLING
COOLING EFFECTIVENESS OF VARIOUS TYPES
OF AIR COOLING FOR S-816 BLADES
ALLOWABLE EFFECTIVE GAS TEMP °F
24 FINS
FILM COOLING
12 FINS
INSERT
HOLLOW BLADE
LIMIT FOR UNCOOLED TURBINES
WEIGHT OF COOLANT
WEIGHT OF GAS
HOLLOW BLADE
INSERT
FILM COOLING
12 FINS
24 FINS
LIQUID COOLING
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SOME ASPECTS OF HIGH-TEMPERATURE TURBINE RESEARCH

by H. H. Ellerbrock and J. B. Esgar

The great importance of operating turbines at high inlet temperatures was discussed by the previous speaker. Vigorous efforts are being made by the NACA to find means of building turbines that can withstand gas temperatures up to 3500° F, the approximate temperature limit that can be obtained with present hydrocarbon fuels.

The ideal method of doing this would be to use materials capable of withstanding the necessary stresses at high temperatures. One aspect of research by the NACA on this method has been the development of ceramic blades for turbines especially those for missile engines. This is a sample of the ceramic blades. A turbine with blades of Bureau of Standards Body 4811 material, which is mostly a beryllium oxide, has been operated at tip speeds up to 840 feet per second at turbine inlet temperatures up to 1800° F and the turbine has been run at 2000° F at reduced tip speed. Heat-shock and blow-out tests have been made where the gas temperature dropped from 1800° to 300° F in as little as 4 seconds without harming the blades. Research on this promising method of obtaining turbines capable of operating at high temperatures is continuing. Three reports have been published on the results of these studies to date.

Another method of approach to which much effort has been directed to obtain high-temperature turbines is to cool the turbine blades which are the most critical parts of the turbine because they are in direct contact with the hot gas. Its purposes are twofold: First, to extend the gas temperature range of turbine operation and, second, to utilize nonstrategic alloys where high-temperature alloys are now used. The utilization of nonstrategic alloys is especially important for turbojet engine application where very high temperatures are not necessary.

Analytical studies have been made of several methods of cooling turbine blades. A solid blade can be cooled by the conduction of heat along its length to the cooled rim of the turbine rotor - this is called rim cooling. Blades can also be cooled by the heat being removed by air or water flowing through passages inside the blades. Consider this large scale model of a hollow, air-cooled turbine blade, the cooling air flows through the blade and is then ducted into the exhaust jet of the engine. A sketch of this blade cross section is illustrated on ~~this~~ figure 2. The figure also shows some other typical cross sections of air-and water-cooled turbine blades. If an insert is installed in the blade passage, the air is forced to flow between the insert and the blade wall, which increases both the air velocity and the cooling effectiveness. The provision of fins in the blade passage along its entire length, as shown on the sketches of the 12 fin and 24 fin blades, increases the heat-transfer surface in contact with the cooling air. If slots are cut in the walls of a hollow blade, it allows the cooling air to flow through these slots and out over the blade surface, forming a cool, insulating film of air between the blade and the gas stream - this is called film cooling. Water-cooled blades have this general configuration. It is necessary that the holes through which the coolant flows be round because of stress considerations from the water pressure at high rotative speeds. The cooling water is continuously recirculated through a radiator similar to a liquid-cooled reciprocating engine application.

The results of analytical studies comparing the effectiveness of the different methods of turbine blade cooling will be presented. These results are comparative only because of assumptions made in the analysis for simplifying purposes.

The first method of blade cooling to be investigated was rim cooling. The studies showed that the allowable increase in turbine inlet temperature is only about 150° to 200° F with present turbine blade materials but blade life can be increased with small decreases in the rim temperatures.

The next step in the turbine-cooling investigation was to determine the effectiveness of air cooling hollow blades. Some results of these analytical studies are shown in figure 3 illustrating the cooling effectiveness of hollow blades made of various materials. The allowable effective gas temperature, which is the temperature effecting heat transfer, is plotted against the ratio of weight of coolant flow to weight of gas flow for the case of air flowing through hollow blades made of four different alloys. This ratio of coolant flow-to-gas flow will be denoted as dilution hereafter in the discussion of air cooling. Cooling-air temperature at the blade inlet was assumed to be 300° F and limiting blade temperature was based on stress-rupture data. The circle point on the figure shows the present limit for uncooled turbines having S-816 alloy blades. (About 1330° F effective gas temperature, which corresponds to about 1500° F turbine inlet temperature.) With 5 percent dilution, the tentative value considered permissible at present for air cooling, this analysis shows that turbines with blades made of the best alloy evaluated, S-816, can be operated at a gas temperature about 350° F higher than that of the uncooled turbine. S-816 alloy, however, is now considered a strategic material and substitute metals are being sought. The metals shown on the figure were evaluated as possible substitutes; inconel is less strategic than S-816, SAE 4130 has less than 1 percent of any strategic metal, and SAE 1015 is ordinary automobile fender steel. The importance of being able to use nonstrategic materials must not be overlooked due to the acute shortage of metals used in high-temperature alloys. At a dilution of 5 percent, the results show that with hollow air-cooled blades of the substitute metals the allowable effective gas temperature for the turbine will be lower than that for present-day uncooled turbines with S-816 blades. It is known from more rigorous analyses that when cycle pressure ratio and turbine inlet temperature are increased simultaneously to achieve high performance the amount of heat transfer is increased and greater dilutions than those shown here are required for adequate blade cooling.

It is readily seen that something better than hollow blades are needed if high gas temperatures are to be obtained with air cooling. This is especially true if nonstrategic metals have to be used. Consequently, studies were made to evaluate the cooling effectiveness of the air-cooled configurations in figure 2 which were previously described; these are the blades with the insert, 12 fins, and 24 fins in the cooling passage and the film-cooled blade. Some results with these blade configurations made of S-816 are shown in figure 4 which illustrates the cooling effectiveness of various types of air-cooling for S-816 blades. Here again the allowable effective gas temperature is plotted against dilution. The colors of the lines correspond to the colors on these sketches of blade configuration. All configurations theoretically provided a higher allowable effective gas temperature than that for the hollow, air-cooled blade. Operation at 3500° F is made possible with 24 fins in this particular blade at a dilution of about 6 percent. This value is slightly more than that now considered allowable, but the actual limiting value of dilution will have to be determined experimentally.

The blade with 24 fins was so promising that further calculations were made to determine the allowable effective gas temperatures for this configuration using metals shown on the first figure plus an aluminum alloy. Figure 5 shows the cooling effectiveness of blades made of various materials and having 24 fins. The results indicate a nonstrategic alloy like SAE 1015 steel at a dilution of over 3 percent is capable of withstanding gas temperature higher than for the uncooled turbine and that 3500° F operation is possible with the high temperature alloys. The aluminum alloy would be of little use at a cooling air temperature at the blade inlet of 300° F, the temperature used in the calculations for this figure; but if the air temperature could be dropped to 100° F, operation at 1500° F effective gas temperature with this configuration would be possible at a dilution of about 9 percent.

In order to extend the gas temperature limit above that obtainable by air cooling, studies were made of the effectiveness of water cooling with this blade configuration. Results

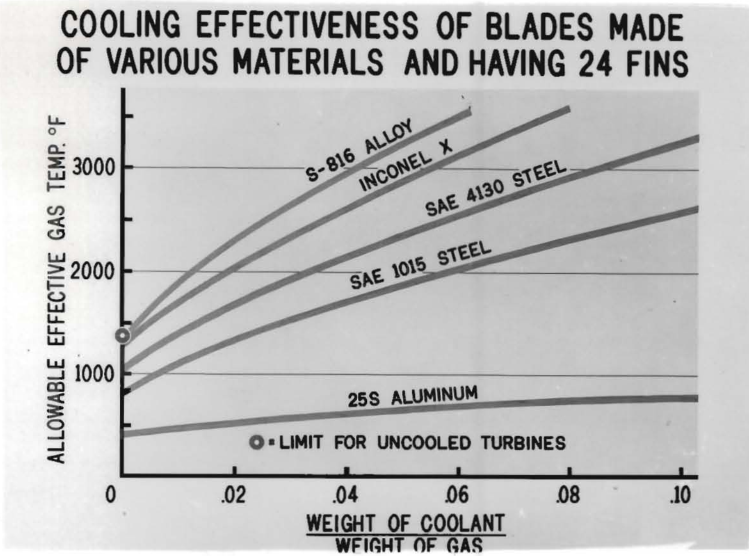

Figure 5.

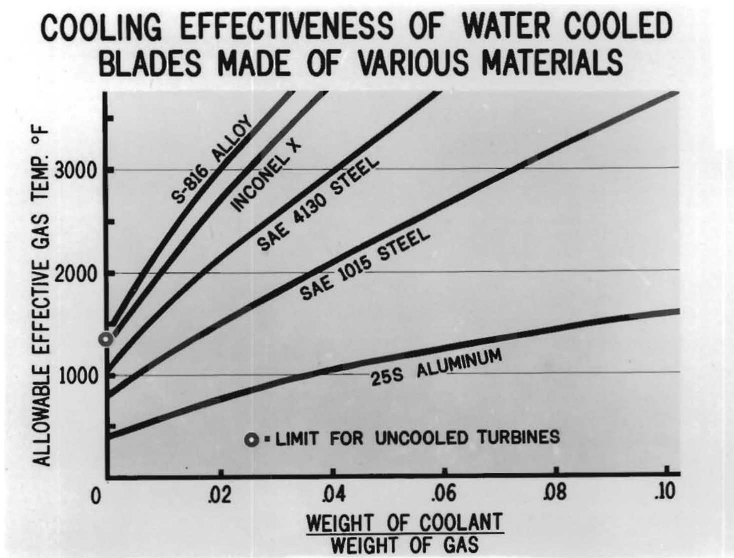

Figure 6.

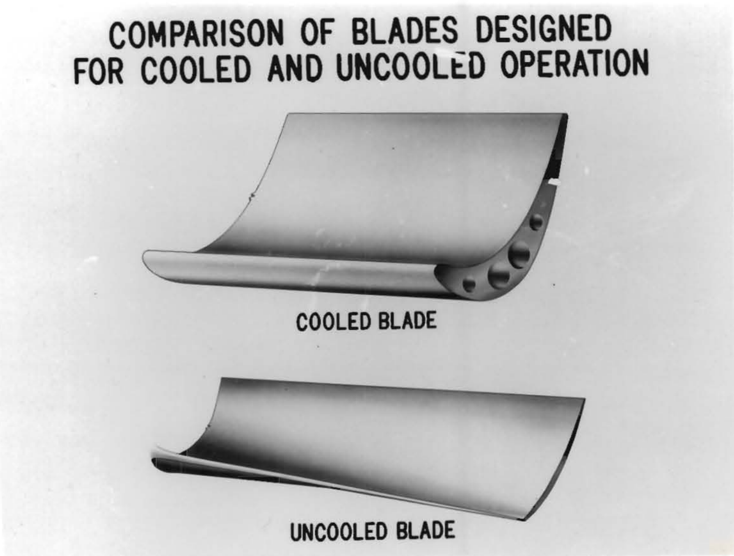

Figure 7.

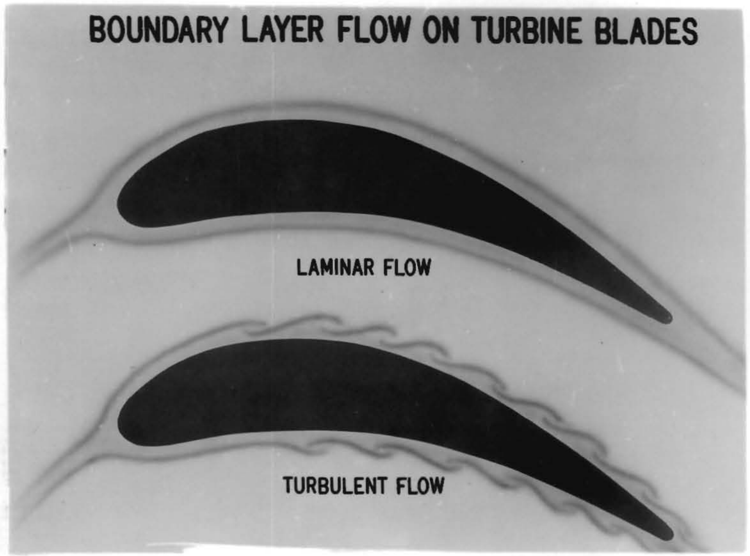

Figure 8.

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of the analytical studies are illustrated **in figure** 6 showing the cooling effectiveness of water-cooled blades made of various materials. This figure is similar to the figure just discussed; the same metals are evaluated and the coolant temperature was assumed to be 200° F. With this type cooling it would be possible to operate at 3500° F gas temperature at coolant flows of 3 to 9 percent of the gas flow with all alloys shown except aluminum, and even aluminum could be used at temperatures above the operating limit of uncooled turbines. For instance, with the coolant flow 10 percent of the gas flow the allowable effective gas temperature is 1600° F which corresponds to about 1800° F turbine inlet temperature. These calculations have been verified to some extent in that an experimental water-cooled aluminum turbine, designed at this laboratory has been successfully operated to date at turbine inlet gas temperatures up to 2200° F. The fact that this turbine was able to operate at a temperature 400° higher than the maximum shown on the figure was made possible by the cooling water temperature being 100° F instead of 200° F; and in addition, the amount of coolant flow was considerably higher than for these calculations. Liquid cooling could probably be used most successfully in long range ty e of aircraft where long engine life and fuel economy are quite important.

To sum up, the results in general show that two great advantages can be obtained by turbine cooling: (1) it is possible to use nonstrategic metals in cooled turbines while maintaining the gas temperature at levels as high or higher than is now standard practice in uncooled machines, and (2) using heat resistant alloys the gas temperature range of operation can be increased so that large increases in power can be obtained. The cooling of turbine blades dictates the use of slightly thicker blade shapes than those in current use. The aerodynamic aspects for obtaining suitable blade shapes will be discussed by the next speaker, Mr. English.

TURBINE AERODYNAMICS

by Robert E. English

In advancing the turbine-type of engines major improvement in engine performance will result from the solution of two research problems: (1) increasing the turbine inlet temperature with the aid of turbine cooling, and (2) raising the engine pressure ratio. The research on turbine aerodynamics is so directed that the turbines will satisfactorily meet these changing requirements. Three general turbine characteristics have been set as the goals toward which the turbine research is directed:

~~Figure I~~

(1) Blades shaped to contain the required cooling passages but still aerodynamically efficient

(2) Velocity distributions on the stator and rotor blades that keep the transferred heat to a minimum

(3) High pressure ratio per stage with high stage efficiency

Goal 1. - If turbine cooling is to be used, the turbine blades must necessarily be thick enough to permit cooling passages to be built into the blades. This problem is illustrated on the next figure (Fig. 7). This blade was designed to accommodate cooling passage and may easily be cooled. This blade was designed for no cooling and is typical of the blades for which design information is available. The blade is not only thin but twisted, both of which complicate the process of constructing cooling passages. To design thick, untwisted blades requires careful specification of the velocities in the turbine, and it is toward this end that some of our research is directed.

Goal 2. - To minimize the heat transferred from the hot gas to the turbine blades requires knowledge of the boundary-layer flow on the blades (Fig. 8). If the boundary-layer flow is laminar, smooth, streamline flow takes place along the blade surface. This permits the boundary layer that is cooled near the blade nose to remain in contact with the blade and form an insulating blanket to protect the rest of the blade. If the boundary layer is turbulent, smmoth, streamline flow is not obtained and the scouring action of the turbulent flow replaces the cooled gas near the blade with hot gas from the main stream. With a laminar boundary layer the heat-transfer rate is less than half of the rate with a turbulent boundary layer.

If the heat-transfer rate is low, few cooling passages are required. Also, if a laminar boundary-layer blankets the region near the tail, the last cooling passage need not be so far back in that slender region. Because a laminar boundary layer is a great aid to turbine cooling, a portion of our research is directed toward determining the design conditions that will produce laminar boundary layers.

In working toward the third of our three goals, this laboratory has recently improved the pressure ratio and efficiency of its turbine stages. These higher pressure ratios are desirable because the number of turbine stages may then be reduced, thereby decreasing both the engine weight and the required amount of cooling. In this work we use two cold-air turbines, one of which is here; the other is just behind you. A partially bladed rotor from this turbine is displayed here. The performance of one set of blades is shown in the next figure (Fig. 9). (Describe.) With this design both high pressure ratio and high efficiency were obtained.

Although we obtain a great deal of information from these turbines concerning the over-all performance of the various designs, measurement of the flow conditions within the rotor-blade passages presents a difficult instrumentation problem which has not yet been solved for machines of this size. To determine the flow conditions within the rotor-blade passages, we resort to a two-dimensional approximation for the three-dimensional conditions in the turbine by mounting untwisted blades of uniform section in a cascade rig. This is just like placing an airplane wing in a wind tunnel. The cascade is schematically shown in the next figure (Fig. 10). The blades are shown mounted in a rectangular duct through which the air flows in this direction. Holes drilled in the blade surface are used to measure static pressure on the blades. This instrument downstream of the blades determines the angle of flow and the losses. In this rig we can compare the actual and theoretical velocity distributions on the blades. One of these comparisons is presented on the following figure (Fig. 11). (Describe.) The velocities agree everywhere except near the tail, where a normal shock occurs within the blade channel to invalidate the design assumptions. The occurrence of the shock may be confirmed by looking at a schlieren photograph, which permits the shock waves to be discerned. A schlieren photograph at this same operating condition is presented in the following figure (Fig. 12). This shows the cascade with the flow through the blades in this direction. Here is the normal shock within the blade channel as the pressure measurements indicated. This is the shock pattern at the blade exit that accompanies high pressure ratios. These blades have a channel width that continuously decreases from entrance to exit so that the blades choke at the exit. This shock pattern is obtained by expanding from sonic speed to supersonic speeds at the exit from the blade row.

Blades of this type have the advantage of operating satisfactorily over the entire range of subsonic velocities and also well in the supersonic range. As part of the program of raising the pressure ratio per stage, studies are being made of the shock patterns at the blade exit.

To summarize, the blades must be shaped to contain the cooling passages; the boundary-layer flow must keep the transferred heat to a minimum; the turbine stages must have high pressure ratio and high efficiency. By means of objective research such as this, this laboratory is producing methods for turbine design that will satisfy the changing requirements of turbines for aircraft.

As an example of the NACA's research on turbines one of the experimental water-cooled turbines is on display in the next room. The turbine is running at a tip speed of 10,000 rpm. Instruments above the turbine indicate the inlet gas temperature and the temperature of a representative turbine blade. See photograph C-22349, next page.

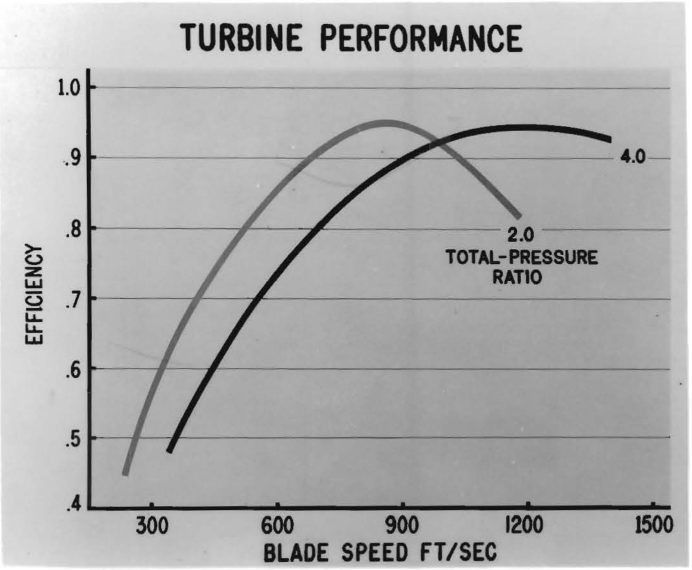

Figure 9.

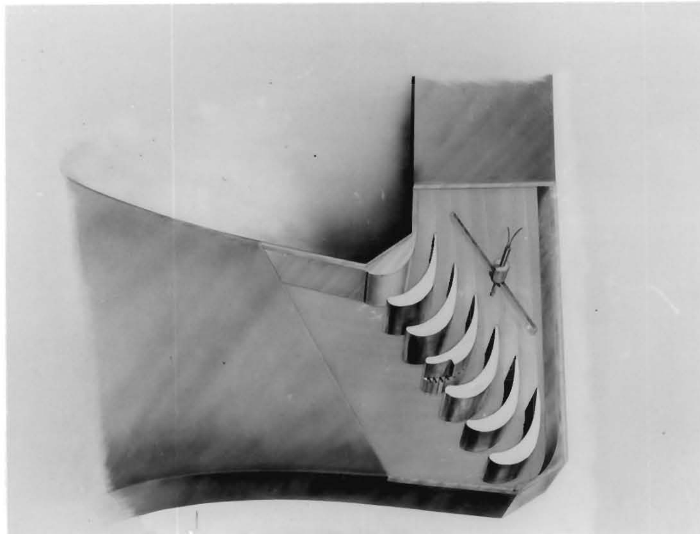

Figure 10.

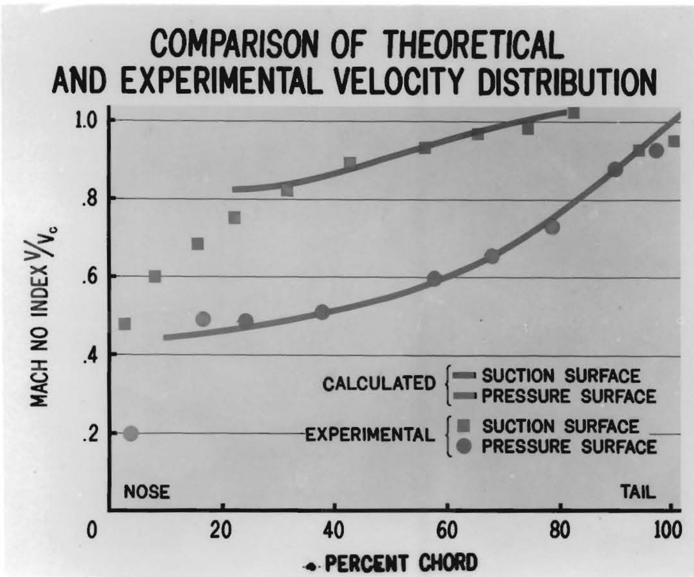

Figure 11.

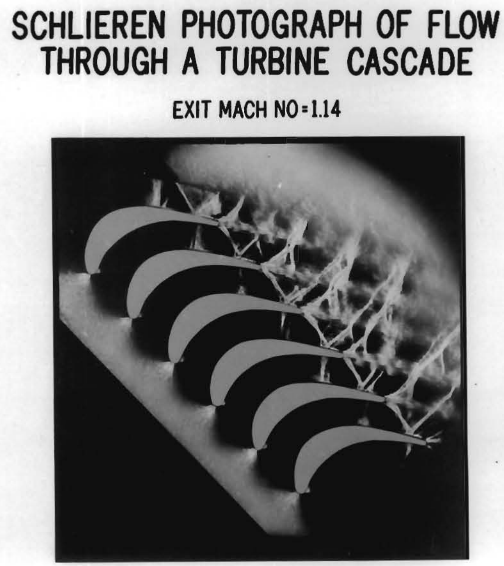

Figure 12.

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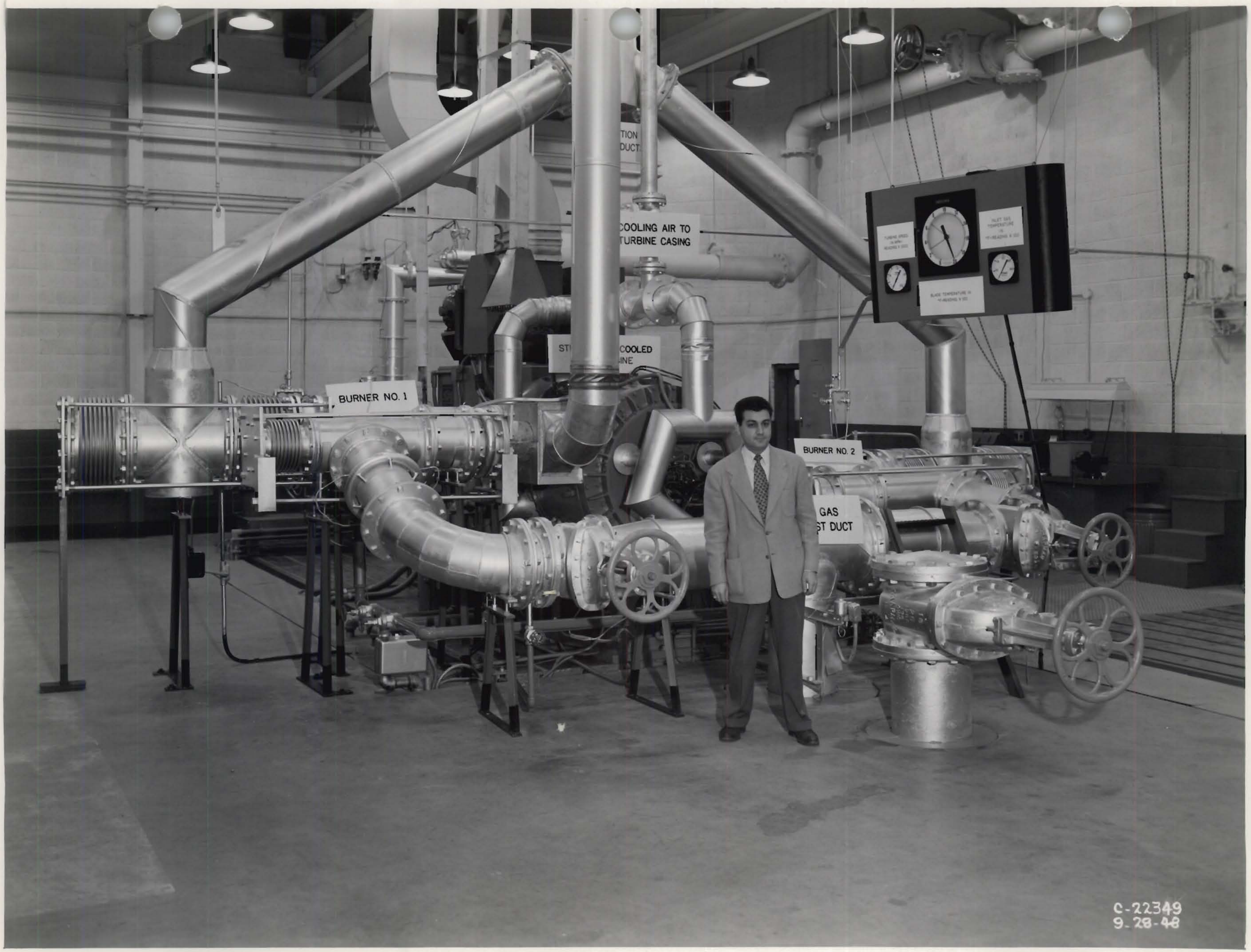
COOLING AIR TO
TURBINE CASING
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CENTRIFUGAL COMPRESSOR RESEARCH

by John D. Stanitz

The introductory speaker at the previous demonstration pointed out the need for small, efficient compressors with high pressure ratios. This need has resulted in the following objectives for compressor research ~~(fig.)~~.

OBJECTIVES OF COMPRESSOR RESEARCH

(1) High efficiency

(2) Small frontal area (for a given air-flow rate)

(3) High pressure ratio per stage (to obtain a compressor with the minimum number of stages)

(4) Wide flow range (at satisfactory efficiency)

(5) Simplicity (which in general leads to reliability, low cost, and low weight)

At the present time three compressor types appear most promising with regard to these objectives. The three types are: (1) the centrifugal, (2) the axial-flow, and (3) the s upersonic compressor. Each of these types will be discussed briefly at this demonstration. This concludes the general introduction to compressor research. The centrifugal type compressor will now be considered.

Advantages of the centrifugal compressor are: (1) high pressure ratio per stage, (2) wide flow range, and (3) simplicity. The simplicity of the centrifugal compressor is especially important at the present time because the resulting ease of manufacture enables low cost, quantity production. On the other hand, two major problems of centrifugal compressors are: (1) large frontal area, and (2) low efficiency compared to the axial-flow compressor.

The first major problem, the large frontal area, is being attacked by the NACA on axial-discharge centrifugal compressors. The impeller of such a compressor is exhibited on the stage. The flow enters and leaves the impeller in the axial direction and centrifugal compression is obtained by the increased radius at which the flow leaves. A discussion of this work was given at the annual inspection last year and two reports have since been issued. The frontal area of this compressor is less than half that of conventional centrifugal compressors and approaches the low frontal area of axial-flow compressors. However, the efficiency is 5 to 10 points lower than the efficiency of axial-flow compressors. This is the case for all centrifugal compressors and leads to the second major problem, i.e., the relatively low efficiency.

At the present time the design of centrifugal compressors is primarily an art. The design is not based upon detailed knowledge of flow conditions within the impeller. If these flow conditions can be determined and controlled, then the efficiency of centrifugal compressors can be as high as that of axial-flow compressors.

As a first step in this direction analytical methods have been developed whereby the flow conditions within the compressor can be computed for compressible, nonviscous flow. A result of the analysis is shown on figure 13. This plot shows the streamlines for flow through an impeller with 20, straight, radial blades. The impeller rotates in the clockwise direction and the air enters the impeller near the center and flows outward between the impeller blades and into the vaneless diffuser along the streamlines indicated on the chart. For the flow rate and rotational speed of this example an eddy forms on the driving face of the blade as shown. From this streamline configuration velocities and pressures can be determined and in this example the velocities were found to decelerate rapidly along the trailing face of the blade . For a real, viscous fluid this deceleration leads to separation of the flow from the blade surface and results in low compressor efficiency.

These phenomena, associated with real, viscous fluids, are not accounted for by the ideal theory. Therefore, to investigate these phenomena and to provide a basis for the correlation of the ideal theory with experimental fact the NACA has designed and built a research compressor which is sufficiently large to accommodate complete instrumentation of the impeller. The impeller of this compressor is exhibited on the platform and a cross-sectional diagram of the assembled rig is shown in figure 14 . The impeller diameter is 48 inches and the collector diameter is 100 inches. Air flows into the compressor at the hub and outward through the impeller and vaneless diffuser into the collector. The readings of 168 static and total pressure taps on the impeller are transmitted to manometers through the pressure commutator as shown in figure 14. The experimental information thus obtained will be correlated with the theoretical pressures and velocities and from this correlation between experiment and analysis the what and the why for flow conditions in centrifugal compressors will be obtained. From this knowledge methods of design can be determined for centrifugal compressors, including the axial-discharge compressor, with higher aerodynamic efficiency and better over-all performance.

This concludes the discussion on centrifugal compressor research. The next speaker, Mr. Sinnette, will discuss the research on axial-flow compressors.

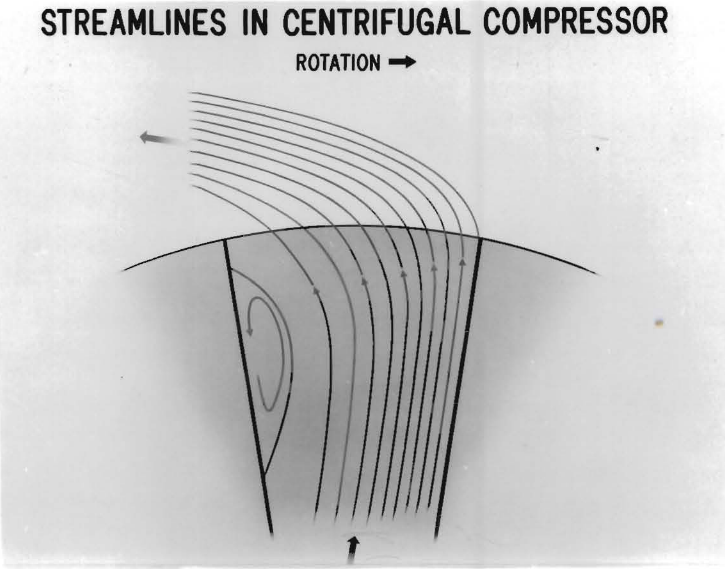

Figure 13.

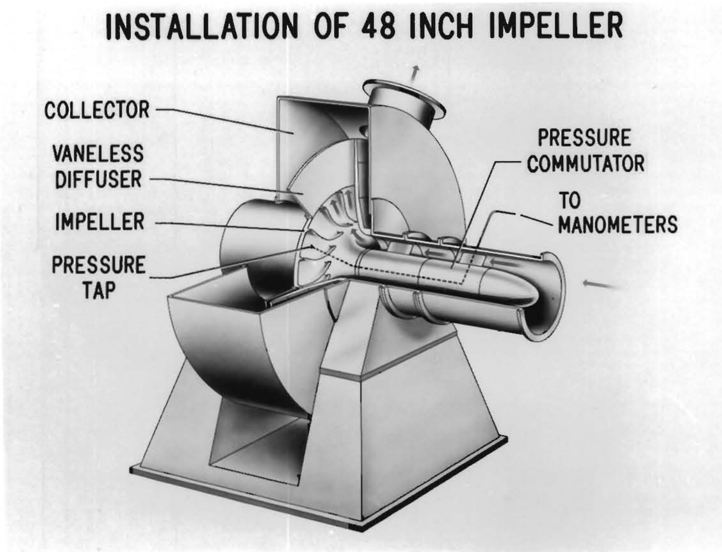

Figure 14.

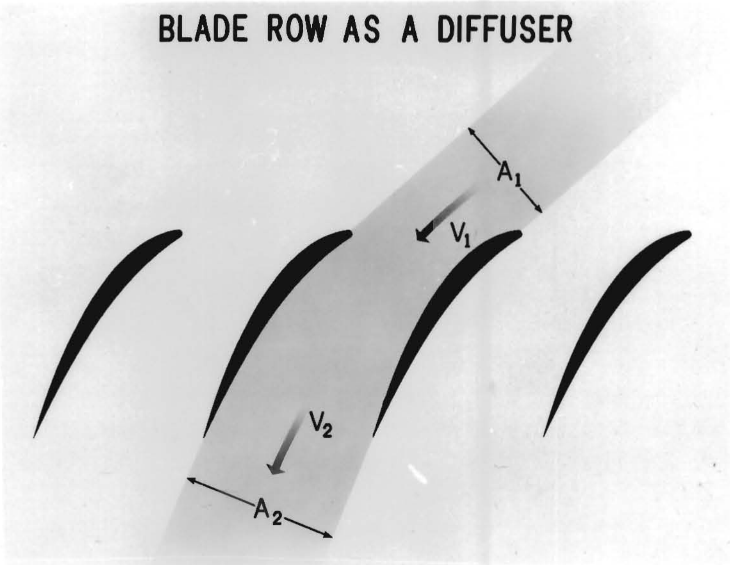

Figure 15.

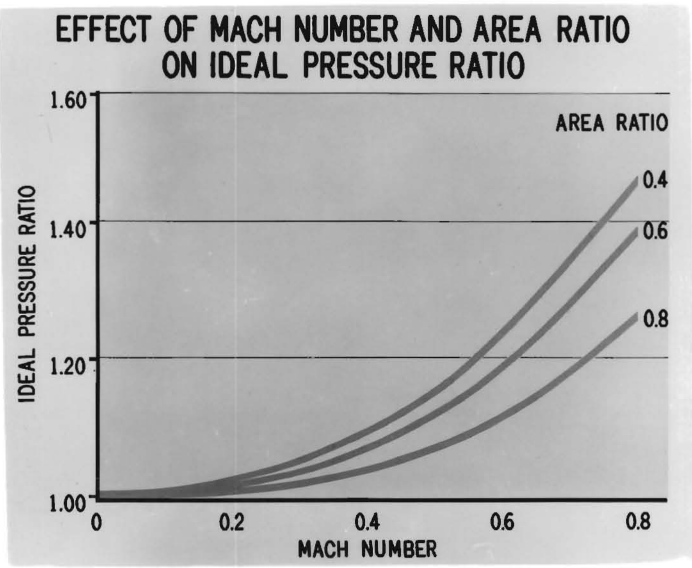

Figure 16.

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SUBSONIC AXIAL-FLOW COMPRESSOR RESEARCH

by John T. Sinnette, Jr.

Mr. Stanitz has pointed out the advantages and disadvantages of the centrifugal as compared to the axial-flow compressor. The present-day axial-flow compressor is undoubtedly much more complex, expensive, and difficult to manufacture than the centrifugal compressor. This fact is strikingly shown by a comparison between the thousand or so blades required in an axial-flow compressor with the few dozen rugged blades of a centrifugal compressor.

In spite of these handicaps, however, the modern trend in turbojet and turbine-propeller engines is toward the axial-flow compressor, because of its intrinsically high efficiency, outstanding air-flow capacity per unit frontal area, and ease of staging to obtain high pressure ratio. The necessity of being able to produce these compressors in large quantities in any national emergency makes it essential that every effort be directed toward minimizing any undesirable features.

The cost and difficulty of manufacturing large numbers of precision blades is a problem of utmost importance. Research is therefore being directed towards reducing the number of blades required and simplifying the shape so as to reduce the cost per blade. The cost of the blades can also be reduced considerably if the tolerances on blade accuracy can be relaxed somewhat. Although little is known at present about the effects of manufacturing tolerances on compressor performance, research is now being directed towards determining these effects.

One of the most effective methods of reducing the number of blades is by increasing the pressure ratio per stage so as to reduce the number of stages required. This increase in pressure ratio per stage is also of considerable value in reducing the over-all length of the compressor.

The method of obtaining pressure rise in an axial-flow compressor is shown in figure 15, which illustrates a typical row of blades. As is well known, this type compressor may be considered as a series of diffusers wherein the pressure rise is obtained by a reduction in the relative velocity through each blade row. If the entering velocity is subsonic, diffusion is obtained by increasing the effective area accomplished by turning the air in the axial direction. The pressure rise that is obtained in this diffusion process depends on the entering Mach number, the ratio of inlet to outlet area, and the efficiency of diffusion. The ratio of the inlet to the outlet area will be referred to hereafter as the area ratio and for compressors will be less than 1.

The pressure ratio that would be obtained in an ideal diffuser of 100 percent efficiency is shown in figure 16, where the pressure ratio is plotted against the entering Mach number for several different area ratios. The rapid rise in pressure ratio with increasing Mach number indicates the importance of using as high a Mach number as possible to obtain high pressure ratio per stage. The pressure ratio can also be increased by decreasing the area ratio as is seen by comparing the different curves. This decrease in area ratio is accomplished by turning the air through a larger angle. The method of obtaining larger turning of the air is illustrated in figure 17. On the left we see conventional blades designed to produce small turning of the air. The blades have little camber or curvature of the mean line. If larger turning is desired it is necessary to go to blades with more camber as shown at the center. However, if the turning is too large, or the Mach number too high, the flow may separate from the blades because of the high blade loading. This flow separation can generally be avoided by placing the blades closer together as shown at the right. But closer spacing means more blades per row and higher skin friction. Thus the determination of the optimum combination of camber, spacing, and Mach number is of considerable importance, particularly since larger turning causes the blades to stall at a lower Mach number.

Figure 18 shows some of the more promising results to date of the NACA research directed towards obtaining the optimum combination of these factors. The red curves show the stage total pressure ratio and efficiency plotted against the relative entering Mach number as determined on a 14-inch experiment compressor with a hub-tip ratio of 0.8. The corresponding pressure ratio of an ideal compressor of 100 percent efficiency is shown for comparison by the blue curve. As can be seen, the experimental pressure ratio does not continue to rise with Mach number as does the ideal pressure ratio but reaches a maximum beyond which it drops off rapidly. The peak in the pressure ratio curve is a result of the rapid drop in efficiency beyond a certain Mach number. This rapid drop in efficiency is associated with the occurrence of local supersonic flow regions over the blades and the production of shock waves and flow separation. Although the occurrence of this phenomenon has long been recognized, it is only recently that sufficient detailed knowledge has been obtained on the variation of efficiency with Mach number to permit accurately designing compressors to obtain the maximum practical pressure ratio per stage. At a relative entering Mach number of 0.8, we note that the stage pressure ratio exceeds 1.4 at a reasonably high efficiency. This represents a substantial gain over the stage pressure ratio of 1.15 typical of present-day compressors. Although it may not be practical to obtain the same high pressure ratio in all the stages of a multistage compressor, the present results nevertheless indicate that, with proper design, it should be possible to reduce the number of blades in a multistage compressor by approximately 35 percent with a corresponding decrease in compressor length and an increase in reliability.

In contrast to compressors with low pressure ratio per stage, where the design is not very critical, the design of compressors with maximum pressure ratio compatible with high efficiency and high air-flow capacity is quite critical and makes it imperative that we obtain more accurate and detailed information on the flow processes within the compressor.

For this purpose, several 14-inch experimental compressors have been used extensively at this laboratory to obtain such valuable information as Mach number and turning angle limitations with different types of blades. The experience with these compressors, however, has shown that much larger compressors are necessary for investigating the details of the complicated flow processes within axial-flow compressors. A 30-inch compressor has been built and is installed in the setup to investigate these processes in compressors having up to four stages. But even this compressor is not large enough to investigate such important problems as the boundary-layer flow and pressure distribution about rotating blades. To satisfy this need, this 72-inch compressor has been built and will be ready for use in the near future.

In order to obtain the most out of this equipment, it is necessary to supplement the experimental work with extensive theoretical investigations. To accomplish this end, a separate section is devoted to fundamental analytical investigations on problems relating to compressors and turbines.

In summarizing, we may say that a substantial increase in pressure ratio per stage has already been obtained as a result of the investigations directed toward determining the optimum combination of Mach number, blade camber, and blade spacing; and, that the incorporation of these principles in multistage compressors should result in a substantial reduction in the number of blades and the axial length, and an increase in the reliability of future compressors.

So far we have been concerned only with the so-called subsonic axial-flow compressor. We have seen that large theoretical gains in pressure ratio are obtainable by increasing the Mach number, and that the main reason for restricting ourselves to subsonic flow is the rapid drop in efficiency that has been experimentally found to occur with conventional airfoil blades as the inlet Mach number approaches one.

Mr. Graham will now discuss what the NACA has done to develop special blades that will efficiently diffuse supersonic velocities and thus obtain much higher stage pressure ratios than are possible with subsonic compressors.

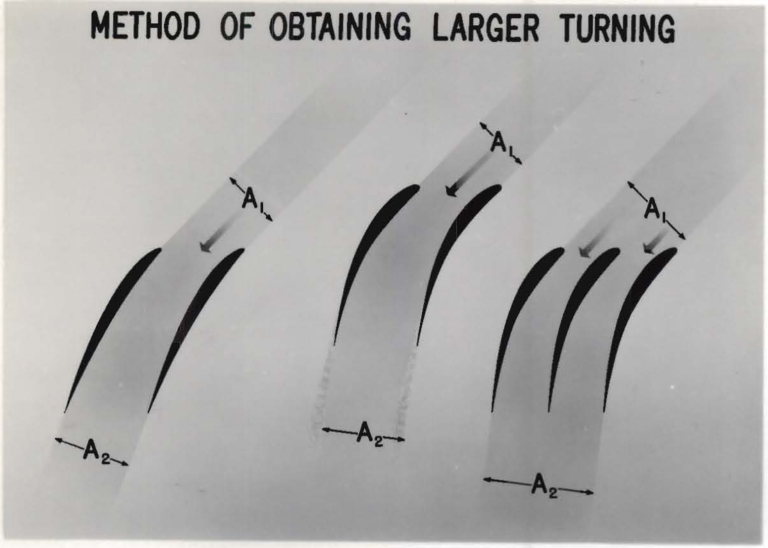

Figure 17.

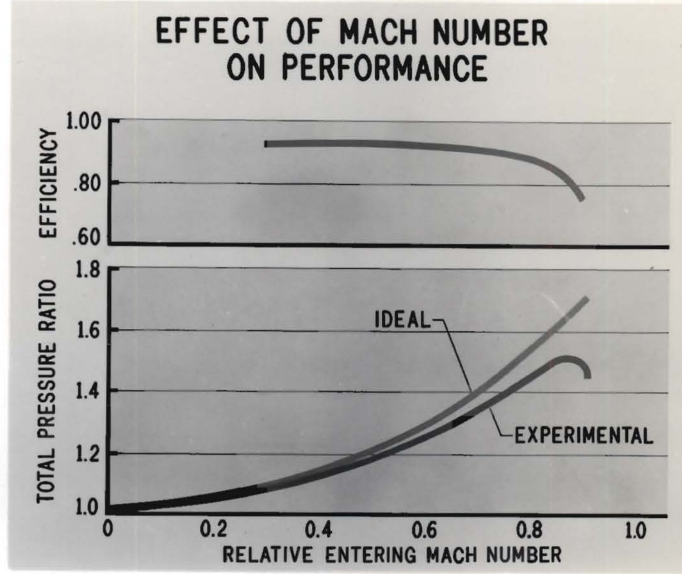

Figure 18.

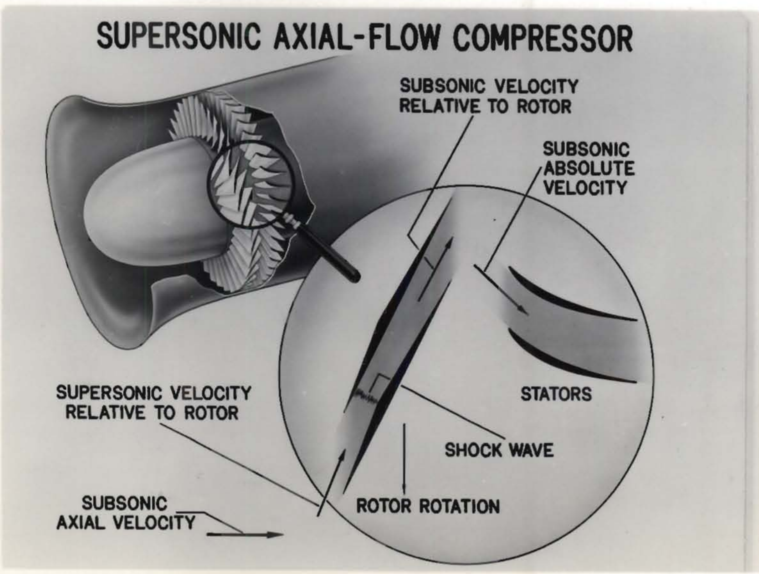

Figure 19.

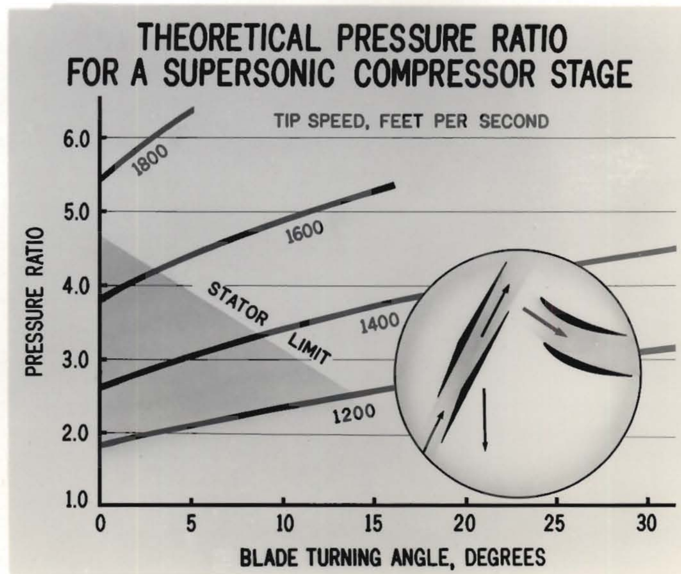

Figure 20.

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SUPERSONIC COMPRESSORS

by Robert C. Graham

As pointed out by Mr. Sinnette, one of the major factors limiting the pressure rise attainable in axial-flow compressors is the Mach number relative to the blades. An analysis of the problem indicated that if high relative velocities could be used, the pressure ratio could be increased considerably. The NACA, therefore, initiated an investigation of special blade shapes over which the velocities are greater than the speed of sound, and to which the conventional limits of pressure ratio do not apply.

Although supersonic compressors are in the early stage of development, results already obtained have shown that it is an outstanding compressor type, and that it has inherent advantages of high pressure ratios per stage and high mass flow.

One method of utilizing supersonic velocities in a compressor is illustrated in figure 19. The compressor consists of a row of specially shaped rotor blades which revolve at a very high speed followed by a row of conventional subsonic stator blades which are stationary. The air enters the compressor in an axial direction and with a velocity close to the speed of sound, thereby providing a high mass flow. The insert shows an enlarged view of the rotor and stator blade passage. The subsonic axial velocity combined with the high rotational velocity of the rotor results in a supersonic velocity relative to the moving blades.

The rotor passage may then be considered a supersonic diffuser. At design conditions, a normal shock wave is formed at the point of minimum area. As the air passes through this shock, of course, the pressure is increased and the velocity becomes subsonic. A further increase in pressure is obtained from the diffusion of the air in the passage following the shock. The air then leaves the rotor passage in this direction relative to the moving blades. Now, again applying the rotative speed of the rotor, which tends to carry the air around with it, the absolute velocity of the air entering the stators is subsonic and is in this direction. The operation of the stators is, therefore, the same as in a conventional subsonic compressor.

Figure 20 shows the theoretical pressure-producing capacity of this type of supersonic compressor. Pressure ratio for a single stage is plotted as a function of blade turning angle and compressor tip speed. Blade turning angle is defined as the angle through which the air is turned in passing through the rotor. For purposes of this figure the pressure ratio is calculated on the basis of two-dimensional flow, considering only normal shock losses.

These theoretical curves show that this supersonic compressor is capable of producing extremely high pressure ratios in a single stage. Pressure ratio increases with turning angle and tip speed. However, as pointed out by Mr. Sinnette, there is a limiting entrance Mach number into subsonic stator blades, thereby restricting the performance to the shaded area. It may be possible to design stators for supersonic flow also, but since this creates complex problems of stability, the NACA has, to date, restricted its research to compressors with conventional subsonic stator blades. When losses are taken into consideration, it is probable that this type of supersonic compressor with subsonic stators will be limited to pressure ratios of $3\frac{1}{2}$ or 4 to 1.

The attainment of these pressure ratios in a single stage will, of course, be of great significance in the gas turbine field.

Shown ~~on figure~~ at the upper right on the display board (see photograph C-22346) is the rotor used in the initial investigation of the supersonic compressor in air at this laboratory. The rotor blades are very thin and therefore had to be supported at the tip to prevent destructive vibrations. In order to obtain the required structural strength, it was found necessary to machine this rotor from a solid steel forging. The machining was therefore a complex, time consuming, and costly process.

At the 1947 inspection, the performance of this compressor was shown for a single operating speed. At this time I would like to present the performance over the entire speed range, and point out a few of the more important results obtained from the analysis of the data.

Figure 21 shows the over-all operational characteristics of the compressor, with pressure ratio plotted as a function of weight flow for a range of tip speeds. Before these results were obtained, it was felt that the compressor might not operate satisfactorily until high tip speeds were reached, thus creating problems of low-speed operation and acceleration. However, no such problems were encountered. As can be seen, the performance at 800 feet per second is comparable to a present-day subsonic stage, with a wide range of weight flow and a peak pressure ratio of 1.18. As the speed is increased, through the transonic range, the peak pressure ratio increases and the characterestic curve becomes more vertical. At a tip speed of 1765 feet per second, the compressor reached a pressure ratio of 2.08 and operated at the nearly constant weight flow that was predicted from supersonic theory. This pressure ratio is, however, less than that for which the compressor was designed.

A more complete analysis of operation, based on detailed flow measurements, indicated two methods of improving this performance. The first method can be illustrated by reference to figure 22 which shows a profile view of a blade passage. On the basis of the two-dimensional theory used for this design, air e ntering the blade passage at a given radius was expected to flow straight through the rotor. Instead it was found that forces were created in passing through the shock which deflected the air inward in this manner. This resulted in concentration of flow at the hub, with the tip section of the passage being practically ineffective. As a result the expected diffusion in the passage was not obtained and the pressure ratio was therefore lower than that predicted. On the basis of this study, it became apparent that the three-dimensional aspects of flow must be taken into account and that the blades must be designed to eliminate the forces which cause the deflection of the stream lines.

The second method of improving the supersonic compressor is to increase the blade turning angle. This increased turning is accomplished by oblique shock waves in the passage ahead of the normal shock. There are two advantages to this supersonic turning: (1) the pressure ratio is substantially increased as was shown on a previous chart and (2) the blades can be made considerably thicker than those used in the initial compressor.

Shown on the display board is a second rotor ~~Figure — shows a blade~~ designed on this basis. Since the blades are ~~it is~~ approximately three times as thick as the thin initial blades, it is possible to eliminate the shroud and use blades which are machined separately and inserted in the rotor hub. The significance of this improvement in compressor design from a manufacturing standpoint is evident from a comparison of this relatively simple rotor with the complex rotor used in the initial investigation.

In order to fully realize the potentialities of the supersonic compressor, the NACA is conducting an extensive research program. Results already obtained, on which I have touched briefly, have been released in the form of three reports. The objectives in this program are the same as those of over-all compressor research; that is, to obtain high pressure ratio per stage, small frontal area, and good efficiency; at the same time to provide a compressor that is simple, and easy to manufacture. The attainment of these general objectives on the centrifugal, axial-flow, and supersonic types of compressors is essential for maximum performance of gas turbine power plants for aircraft.


COMPRESSOR
RESEARCH
48 INCH CENTRIFUGAL IMPELLER
SUPERSONIC AXIAL-FLOW COMPRESSOR
1st
SUPERSONIC
ROTORS
2nd
AXIAL DISCHARGE
IMPELLER
72 INCH AXIAL FLOW ROTOR
C-22346
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NACA

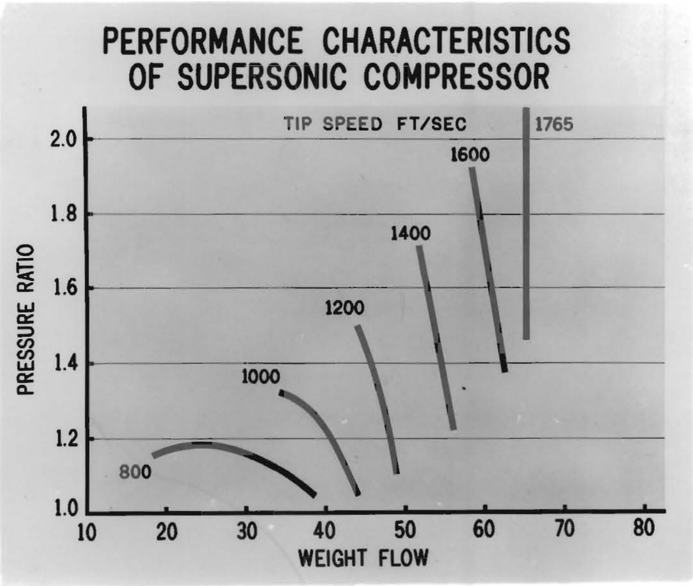

Figure 21.

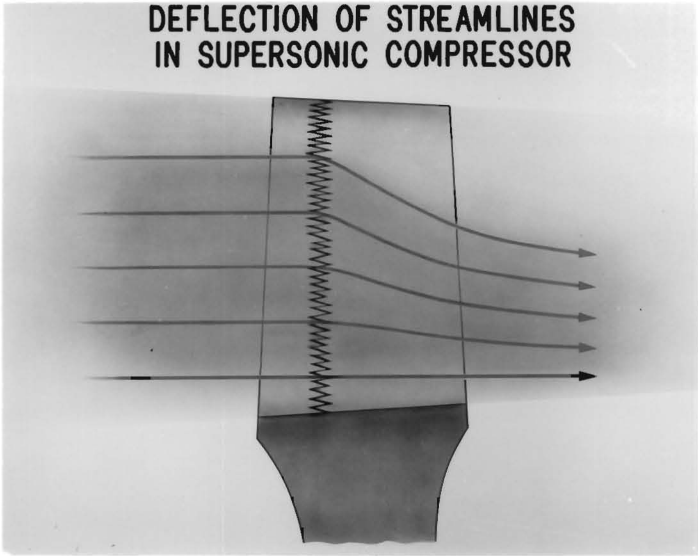

Figure 22.

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