1949 INSPECTION OF THE NACA LEWIS LABORATORY

TALKS ON COMPRESSOR AND TURBINE AERODYNAMICS

Part I - Compressor and Turbine Research: Scope and Methods

Speaker - Robert E. English

(See Stage Photo C-24155 and Color Photo)

The purpose of my talk is to give you an over-all picture of our research on compressors and turbines as separate engine components and to discuss our general research methods in order that you will more fully appreciate the relation between the research problems that will be discussed by other speakers. The goal of this research is to improve compressor and turbine design methods by means of which engines can be produced that have higher power and lower specific fuel consumption than our present engines, and in addition make use of inexpensive, nonstrategic materials. As was described at last year's inspection, we believe that the principal improvements in gas-turbine engine performance will result from increasing the engine pressure ratio, and increasing the maximum cycle temperature by means of turbine cooling and improved materials. The speakers on compressors and turbines will describe research completed during the past year which aids in the attainment of this goal.

This research contributes to a fundamental understanding of the operation of compressors and turbines. We plan, by using such fundamental knowledge, to produce design techniques which are almost entirely analytical so that engines can be produced after only a brief development period.

The diversity of our research on compressors and turbines is illustrated by this chart (chart 1) (see upper right of C-24155). The main divisions of our work are compressor aerodynamics, turbine aerodynamics, and turbine cooling. Our research on centrifugal compressors is divided between the radial- and mixed-flow types; this (point) is an example of a radial-flow compressor rotor and this (point) is a mixed-flow compressor rotor. The subsonic axial-flow compressor is illustrated by this 8-stage machine (point). The axial-flow and mixed-flow supersonic compressors are both receiving our attention; this (Point) is an axial-flow type of supersonic compressor rotor. The axial-flow turbine has so far received the greater share of our effort on turbine aerodynamics; research on radial-flow turbines was recently started. (Point) This is a rotor from one of our axial-flow turbines. Because the problems associated with the aerodynamics of both compressors and turbines are closely related and require the use of many identical techniques, at this exhibit we will describe our work on aerodynamics of both compressors and turbines. Our research on turbine cooling will be presented at the next stop on your itinerary.

From where you sit, you can see that this axial-flow compressor has several alternate rows of moving and stationary blades. A complete investigation of a compressor of this type requires about 100,000 separate measurements of flow conditions within the machine. To make these 100,000 measurements and then to analyze the performance in terms of the measurements is a tremendous job which is both costly and time consuming. In order to simplify this complicated problem, we have divided our research into the four steps shown on the next chart; (1) theoretical attack, (2) cascade investigation, (3) study of single-stage machine, (4) operation of multistage unit. By means of theoretical work, we divide the problems into their basic elements and study the characteristics of each element. At those points where our present theories reach their limits, we turn to fundamental experimental research in the cascades, the single-stage machines, and the multistage units. From these machines, we obtain data for the extension of our present theories. (Draw lines on chart showing relation between topics.)

A cascade is a small wind tunnel especially adapted for studies of blades. A test section of one cascade is mounted here (point). In a cascade we measure the direction and magnitude of velocity past the blades by taking pressure measurements in the flow and we observe shock-wave patterns by means of a schlieren apparatus. Cascades, with their simplicity and small number of blades, permit us to learn many fundamental details of flow in a more economical manner than by any other device. Data from the cascades are used to design single-stage and multistage units, and in turn, from the investigation of these units we learn how to apply cascade data. Individual stages from the single-stage machines are investigated in the multistage units for a correlation of these two phases of our research. In this manner, we obtain information for the aerodynamic design of compressors and turbines.

For our discussion today, our research on compressor and turbine aerodynamics was divided into topics which parallel the division of our research into its separate phases; namely, theoretical investigations, cascade studies, research on single-stage machines, and research on multistage units. The theoretical research will be described by Mr. Costello.

THEORETICAL RESEARCH

Speaker - George R. Costello

The objective of our theoretical research is to predict exactly the performance of any compressor or turbine; that is, to predict the pressure ratio, weight flow, and efficiency of a compressor or the work output and efficiency of a turbine. This requires determining exactly the paths of flow and the velocities of the fluid at every point in the machine. This problem, however, cannot be solved by known mathematical methods at this stage of our work, and simplifying assumptions based on experimental results must be made in order to handle the problem. The theoretical results so obtained are used in the following three ways:

(1) Selecting the important experimental problems, which in turn provide a basis for the extension of the theoretical research.

(2) Interpreting or correlating experimental results.

(3) In many cases, applied directly to compressor or turbine design.

Thus, the theoretical work is the foundation of all our compressor and turbine research.

In the past year, we have made marked progress in our theoretical research, and a few of our new results on the flow through a blade row will be presented in this talk. To save time, these results will be given without the complicated mathematics involved in their derivation. Our basic problem is: given a cascade of blades, such as those shown on the chart (C-24191D) to determine the exact paths of the air flow and the turning angle (indicate on chart) and the velocity at every point along these paths. Because the flow paths and the velocities change as the incoming air is changed, they must be determined for a range of entrance velocities in order to establish the performance of the compressor or turbine under these varying conditions. At the present time, we have solved the basic problem for the flow of a compressible, perfect fluid- that is, for a fluid which can be compressed but which flows over the blade with no friction.

With a real fluid, such as air, there is friction between the fluid and the blade surface which slows down the air next to the blade, so that as the air flows along the blade, a thin layer of slow-moving fluid (boundary layer) develops along the blade surface. If the velocity along the blade surface decreases rapidly, the boundary layer becomes thick and may even separate from the surface, causing the flow off the trailing edge of the blade to become highly turbulent (indicate) which would decrease the efficiency of the compressor or turbine.

A few months ago, we developed a method of solving the empirical equations which give the approximate thickness of the boundary layer at any point along the cascade blade when the perfect-fluid velocity along the blade is known. Thus, the actual flow conditions can be closely approximated by our analysis based on the flow of a perfect fluid.

To apply this analysis to the blades in an axial-flow machine, such as the one displayed on the right (point), the blades are developed on a plane surface (demonstrate by taking a spring-steel hoop on which are fastened axial-flow blades and straighten out the hoop) and the flow is assumed to be the same in all planes parallel to the base (illustrate the planes). In this manner, the three-dimensional flow problem is reduced to the problem of determining the flow through this two-dimensional cascade. Previously, in order to approximate the flow through this cascade, it was necessary to assume also that the fluid was incompressible (like water). This assumption was accurate for low velocities but was not sufficient for the high velocities used at the present time. Just recently we developed methods of computing the flow of a compressible perfect fluid through such a cascade and, what is more useful in design, we can determine the blade shape which will give the specified flow. On the next chart (C-24191-C) is shown an axial-flow turbine blade which was to have a velocity along the blade as indicated on the chart where the ratio of the actual velocity to the velocity of sound is plotted against the projected chord. The blade was built and the experimentally-determined velocities are plotted as points on the graph. The measured velocity correlates well with the prescribed velocity.

When blades designed by the foregoing two-dimensional theory were used in actual machines, it was found that the predicted turning angles (angle through which the air is turned in passing through the blade row) agreed with the experimentally-determined angles over most of the blade but, because of secondary flows, the angles did not agree at the ends of the blades (indicate on chart (C-24191-B)). A theoretical method was developed for predicting the turning angle, taking into account the effect of these secondary flows. The result of applying this theory to a set of entrance guide vanes is shown on the chart where the turning angle is plotted against the radial distance along the blade. The dashed line is the turning angle predicted by the two-dimensional theory; the solid line shows the turning angle given by the new theory. The plotted points are the experimentally-determined angles. The values predicted by the new theory are very close to the measured values.

The rotation of the fluid in an actual machine produces centrifugal forces and, as a result, the fluid does not pass directly through the blades but is deflected radially. We have developed a theoretical method of computing this deflection and the results show that, for a compressor stage, the flow path is generally deflected inward on passing through the rotor blade row and deflected outward on passing through the stators, with the amount and direction depending on blade loading and the hub shape of the stage. An experimental verification of these results is shown on the next chart (C-24182-B), where we have a cross-sectional drawing of a 10-stage compressor showing the experimentally-determined flow paths. Note how these paths are deflected as predicted by the theory. Because of this radial deflection, the velocities in the machine differ considerably from those predicted by the simple theory, and so this deflection must be taken into account in designing compressors and turbines.

All the foregoing discussion applies to flows at velocities less than the speed of sound. When the flow velocities are greater than the speed of sound, the problems are much different because of the formation of shock waves and their interactions. The final chart (C-24191-A) shows a supersonic cascade. This blade is representative of those in a mixed-flow supersonic compressor and it differs radically from those of the subsonic cascades shown earlier. The velocity distribution on the surface of the blade as predicted by the theory which takes into account the shock wave is shown as a solid curve on the chart where we have the ratio of the actual velocity to the velocity of sound plotted against the projected chord. The experimentally-determined velocities are shown as plotted points. The two sets of values agree well. Note that the lowest velocity ratio is 1.35 and therefore the flow is supersonic throughout the passage.

By these few illustrative examples of our theoretical research, we have indicated some of the steps we have made toward our ultimate goal of predicting exactly the flow through a compressor or turbine and thus reducing the design of such machines to a purely analytical procedure. It is expected that by continued close correlation between experimental and theoretical research many of the remaining problems will be solved in the future. The new results we have obtained in our theoretical research are now available to industry and can be utilized immediately in the design of compressors and turbines of higher performance and in the elimination of much of the trial-and-error procedure in their development.

Our theoretical work is accompanied by extensive experimental research, and the remainder of the program will be devoted to this phase of our work. The next speaker, Mr. Hauser, will discuss experimental cascade research.

CASCADE RESEARCH

Speaker - C. H. Hauser

Mr. Costello has shown how certain experimental studies of the flow in compressors and turbines are required to supplement our theoretical work. One of the most important problems is acquiring an exact knowledge of the flow around a given blade. A detailed examination of the flow surrounding compressor and turbine blades is most easily made by use of two-dimensional cascades. (Point to model) The results of these cascade studies are essential in interpreting the performance of our rotating compressors and turbines.

As an example of the effectiveness of our compressor and turbine cascade research, I have selected the study of a limitation in the power output of turbines. On our first chart (C-24192-A) we have plotted turbine power against pressure ratio. The blue curve shown here (point) is the plot of the power available from an ideal expansion of a gas; that is, one without loss. The red curve (point) shows the power obtained from an actual expansion of a gas through a turbine. This chart shows that the actual turbine power increases in a manner similar to the ideal curve up to this region (point) where maximum power is obtained. Even though the pressure ratio is further increased, beyond this point (point) the turbine power output remains constant.

In order to determine the fundamental reason for this limitation of the power output of turbines, an investigation of the flow around a typical turbine rotor blade was carried out in a cascade. This chart

(C-24192-B) illustrates the cascade used in this investigation. High-pressure air enters the cascade of blades (point) at the angle of the leading edge of the blades, flows through the passages, (point) formed by adjacent blades, and leaves the cascade in this (point) direction. The blades are mounted between the glass plates so that schlieren photographs may be taken through the glass in order to visualize any shock waves in the flow. Pressure measurements are used to determine both the direction and magnitude of velocity throughout the cascade.

This chart (C-24192-C) illustrates the flow through the passage formed by two adjacent blades as determined in the cascade. These are two metal bars used to hold the blades in place in the cascade (point). The amount of power that the blades can develop in a turbine is determined by the velocities on the blade surface; the higher the velocities, the greater will be the power output. Thus, if with the cascade, we can find the condition for which the velocities surrounding the blade are a maximum, this will correspond to the condition of maximum power in the actual turbine. As the pressure ratio across the cascade is increased, the gas velocities through this passage (point) continue to increase until the velocity of sound is reached at this section which is the minimum or throat area. The results of our schlieren photographs, which we will show in a motion picture, have indicated that after sonic velocity is reached at the throat, the velocities upstream of this section will remain constant. Thus, any increase in power must be obtained through an increase in the velocity above the speed of sound over this portion of the blade (point). A shock wave was observed to follow the region of supersonic velocities obtained downstream of the throat (point). As the pressure ratio is further increased, this shock wave swings downstream, (point) in this manner until it leaves the passage. After the shock leaves the passage, no further increase in velocity along any portion of the blade can be obtained regardless of further changes in the flow downstream because the downstream pressure cannot be transmitted across this shock wave (point). This is the flow condition obtained in a turbine at the point of maximum power output.

The following brief motion picture taken with our schlieren apparatus shows the changing flow pattern as the pressure ratio is increased. The picture begins with low velocities around the blades and the velocity is continually increased throughout the length of the film. The picture was taken in color to show details in the flow pattern with greater clarity. (Movie begins). The air enters the cascade in this direction (point) and leaves in the direction of the wakes shown here (point). The bars which hold the blades in place can be seen here (point).

As the pressure ratio across the cascade is increased, sonic velocity is attained at the throat (point) as shown by the dark region on the film. After supersonic expansion in this region (point), a compression shock wave is formed here (point) and can be seen as a white line. A second shock wave is seen to form from the trailing edge of the blade, here. Now this shock wave (point) which was formerly normal to the flow has swung downstream. At this point where the shock wave leaves the passage, we attain the condition corresponding to maximum turbine power output. The flow conditions downstream of the blades will now be seen to change as the pressure ratio is further increased. These changes do not affect the velocity on the blade surface, however, and therefore have no effect on the turbine power. It would be impractical to carry out this expansion beyond the point of maximum power output in an actual turbine because the losses incurred in these strong shock waves (point) would absorb the energy available from the increased pressure ratio. The cascade investigation with the schlieren photographs has given us an understanding of the fundamental cause of the limitation in turbine power output. In an investigation of a single-stage turbine using this blade design, the maximum power obtained was in substantial agreement with the maximum power predicted from this cascade investigation.

In addition to supersonic flow phenomena, the performance of a turbine or compressor is affected by variations in the flow conditions over the blade height and the effect of centrifugal forces on the flow through rotor blades. In order to evaluate all these effects, we have to use actual compressors and turbines. Mr. Montgomery, our next speaker, will describe some of the work being done on single-stage experimental compressors and turbines.

SINGLE-STAGE RESEARCH

Speaker - John C. Montgomery

Two-dimensional cascade research provides us with valuable and necessary design information. However, as Mr. Costello has pointed out, the rotating fluid in an actual machine introduces additional problems which must be investigated. These problems arise from the fact that the flow in a compressor or turbine is not the same as the flow in a cascade. The figure on the left of the first chart (C-24178-C) shows the straight-through flow path over a blade in cascade, while the figure on the right shows the approximate flow path over a blade in a rotating passage. Therefore, before we can apply static cascade data to the flow in a rotating passage, we must use a rotating unit to investigate the effect of the three-dimensional flow on the blade performance.

We also use rotating units for the investigation of other design problems such as blade shape, blade finish, and blade tip clearance. A multistage unit such as the 8-stage compressor here could be used for investigations of this type, but a single-stage unit which consists of one row each of guide vanes, rotor blades, and stator blades is obviously more economical and also isolates the individual problem better. Our research program includes the single-stage investigation of the various types of compressors and turbines listed on this chart (C-24178-B). This single-stage research in conjunction with cascade research and theoretical analysis is being carried out in an attempt to achieve the optimum efficiency, pressure ratio, and weight flow in a compressor or turbine.

The aerodynamic problems of the various types of compressor or turbine are essentially the same. I will therefore restrict my discussion to the subsonic axial-flow compressor. As shown by the next chart, certain types of single-stage units are especially suited to each of the design problems that we are investigating. For instance, this first stage of a multistage compressor limits the weight flow through the unit and determines the rotational speed for all of the other stages. Therefore, a 14-inch-diameter single-stage unit typical of an inlet stage of a multistage compressor is being used to investigate the radial velocity distribution and the allowable rotational speed which will give the maximum flow capacity through the unit. A single-stage typical of a middle stage in which the blades are an average of the type of blade throughout the compressor is being used to study the blade shape and blade finish which will give the optimum efficiency and pressure ratio. Because of its small blades, a single-stage unit typical of a last stage is being used to determine the minimum relative blade length for optimum performance. This small-blade stage is also being used to investigate the effect of boundary layer; that is, the effect of the low-energy region adjacent to the compressor walls.

The investigations of the effect of blade tip clearance and three-dimensional flow require larger units to obtain measurable and accurate results. Therefore, a 30-inch-diameter single-stage unit is being used to determine the maximum permissible tip clearance for optimum performance. This unit is also being used to investigate the effect of interstaging; that is, the effect of adjacent stages on each other. The largest single-stage subsonic axial-flow compressor we have is the 72-inch-diameter compressor here to your left. It is being used to investigate the effect of three-dimensional flow on blade performance. The accurate and extensive instrumentation required to investigate the three-dimensional flow around and between the blades necessitate a unit of this size.

During the past year, we have obtained substantial improvements in the performance of subsonic axial-flow compressors. The next chart (C-24178-A) shows the total-pressure ratio and efficiency plotted against the weight flow for various tip speeds of an average stage of a multistage compressor. At a tip speed of approximately 1100 feet per second, the pressure ratio exceeds 1.6 at a reasonably high efficiency for this high pressure ratio. This represents a substantial gain over the stage pressure ratio of 1.15 typical of present-day compressors and over the stage pressure ratio of 1.4 shown here last year.

The variation of flow conditions from the inlet stage to the outlet stage of a multistage compressor make it difficult to design a multistage compressor with a high weight flow and a pressure ratio of 1.6 in each stage. However, through continual research it should be possible in the near future to design a multistage compressor with an average stage pressure ratio of 1.35. This would mean that the number of stages in a conventional multistage compressor could be cut in half, with a corresponding decrease in complexity, cost of manufacture, weight, and an increase in reliability.

As a demonstration of our single-stage technique, the operation of the 72-inch-diameter compressor and its rotor blade instrumentation will be shown. The demonstration will now begin. I would like to call your attention to this model blade section. The height of the liquid in the manometer tubes over it will represent the actual velocity distribution over the blade profile of one of the rotating blades in the compressor. This is done by transmitting the pressures from the corresponding points from one of the rotating blades through these pressure leads to the manometer tubes over the model blade. A smooth velocity profile as represented by this line indicates that the losses over the blade are a minimum. Note that as the compressor speed is increased, the velocity over the blade profile also increases. (Pause for $\frac{1}{2}$ minute during peak speed of compressor.)

The blades in this compressor are approximately 11 inches long and 5 inches wide. The compressor is driven by a 3000-horsepower motor through a gear box which reduces the speed of the compressor to $\frac{1}{2}$ the speed of the motor. In this demonstration, the compressor was turned over at approximately 400 rpm which is only 1/5 its design speed. The air drawn through the compressor was discharged directly into the room through a variable-area throttle at the rear of the unit.. The velocity distribution shown was from the midspan of one of the rotating blades. We can obtain these same data, however, from three different radii on both the rotor and stator blades. From investigations of this type, we can compare the actual velocity profile over the blade surface with cascade results and evaluate the effect of three-dimensional flow on the blade performance. In this manner we hope to obtain a correlation which will enable us to use two-dimensional cascade results to accurately predict the performance of a blade in a compressor or turbine.

The analysis of the effect of the three-dimensional flow on the blade performance does not complete the picture of the investigation of a compressor or turbine. Multistage units introduce additional problems, which Mr. Finger will now discuss.

MULTISTAGE COMPRESSORS

Speaker - Harold B. Finger

Through the use of the results of the theoretical, cascade, and single-stage investigations, we know considerably more about the flow through the multistage units than we have thus far been able to apply in multistage design. In spite of the fact that the more fundamental research techniques are far ahead of the multistage application, we must still investigate multistage units in order to check the validity of our fundamental results. We have recently investigated one of our early-model 10-stage compressors designed on the basis of theory existing several years ago. Some of the results of this investigation are shown on the first chart (C-24182-A). Here, we have the variation of stage pressure ratio throughout the compressor near the maximum pressure-ratio point. It can be seen here that from the fourth stage on, the pressure ratio is practically constant at 1.14. It is of importance to note that the application of the results of the theoretical, cascade, and single-stage investigations explained the experimental results shown on this chart. This fact indicates that ~~that~~ the effect of each blade row in "chopping up" and distorting the flow is small. As was mentioned by Mr. Montgomery, the stage pressure ratio can now be considerably increased over the value of 1.14 shown here by application of our recent fundamental results.

In this particular compressor, we also investigated the effect of radial flow on performance because of the large effect indicated in the theoretical analysis. The experimentally determined radial displacements through this 10-stage compressor were presented by Mr. Costello in the discussion of our theoretical research. As you may recall, the displacements were inward in the rotor and outward in the stator. In every case, the radial displacement was such that a lower stage pressure ratio than the design pressure ratio is obtained. Thus, the pressure ratio per stage can be increased further if the radial displacements are considered in the design procedure.

Another problem of primary importance in jet-propulsion operation is surging. The surge condition is characterized by violent fluctuations in pressures and air velocities through the compressor as shown on the next chart (C-24186-C). Here, we have an instantaneous photographic record of the variation in the pressure in the last three stages of our 10-stage compressor as the pressure ratio was increased to the surging point. It can be seen that the start of surging is characterized by violent fluctuations in pressures and air velocities through the compressor as shown on the next chart. Here, we have an instantaneous photographic record of the variation in pressure in the last three stages of our 10-stage compressor as the pressure ratio was increased to the surging point. It can be seen that the start of surging is characterized by the sudden start of violent pressure fluctuations which continue as long as the surging continues. This pressure fluctuation is accompanied by serious blade vibrations which have in some cases resulted in the destruction of the engine. Because the surging problem becomes more severe as the compressor pressure ratio is increased, as is being done in the new engines, we are devoting a considerable amount of time toward determining the exact reasons for the phenomenon in the hope that a means of avoiding it can be obtained. It should be mentioned that the surging condition must be investigated in multistage units because we have not always been able to run into surging in our single-stage investigations.

Thus far, the emphasis of our discussion has been on our multistage axial-flow compressors. I should like to now discuss two compressors which have only been mentioned previously; that is, the subsonic centrifugal and the supersonic axial-flow compressors. Using equipment and techniques similar to those discussed by the previous speakers, a considerable amount of work is being done toward improving the performance of these two types of compressors. Through the cooperative efforts of this laboratory and the manufacturer, the performance of a service-type centrifugal compressor has been improved as shown in the next chart (C-24182-C). Here we have schematically shown the increase in air flow obtained by modifying the original unit. The height of this bar repreeents the air flow through the unit. This compressor is made up of two compressors of this radial-flow type held back to back. The alteration consisted principally in modifying the blades inthe inlet section and increasing the inlet diameter. The diameter of the original compressor was limited by the maximum allowable blade speed at the inlet, but, as a result of the blade modifications, it was found that a higher blade speed could be used, making possible an increase in inlet diameter. It can be seen here that the maximum weight flow has been increased 35 percent. A 35-percent increase in thrust can be obtained from this increase in weight flow alone. For this unit, however, the increase in thrust is still higher than 35 percent because the efficiency has been increased 13 percent and the pressure ratio, 10 percent.

Our supersonic axial-flow compressor research has resulted in new compressors which are practical units in that their blades are thicker and sturdier than the finely machined, razor-like blades of our original unit shown here. This new 24-inch compressor is representative of the results of our research. An existing unit similar to this one has given improved performance over our original compressor in addition to simplified construction. Because the newer blades are thicker than those of the original unit, the costs and problems in fabricating these blades are much less than for our original blades. The original unit, blades and all, was machined from a single solid forging, whereas this newer unit can be made with separate hub and blades fitted together. Our research has indicated that units (such as this one) can be produced which are capable not only of handling as much air as the best existing subsonic axial-flow compressors but with pressure ratios per stage approaching those of centrifugal compressors.

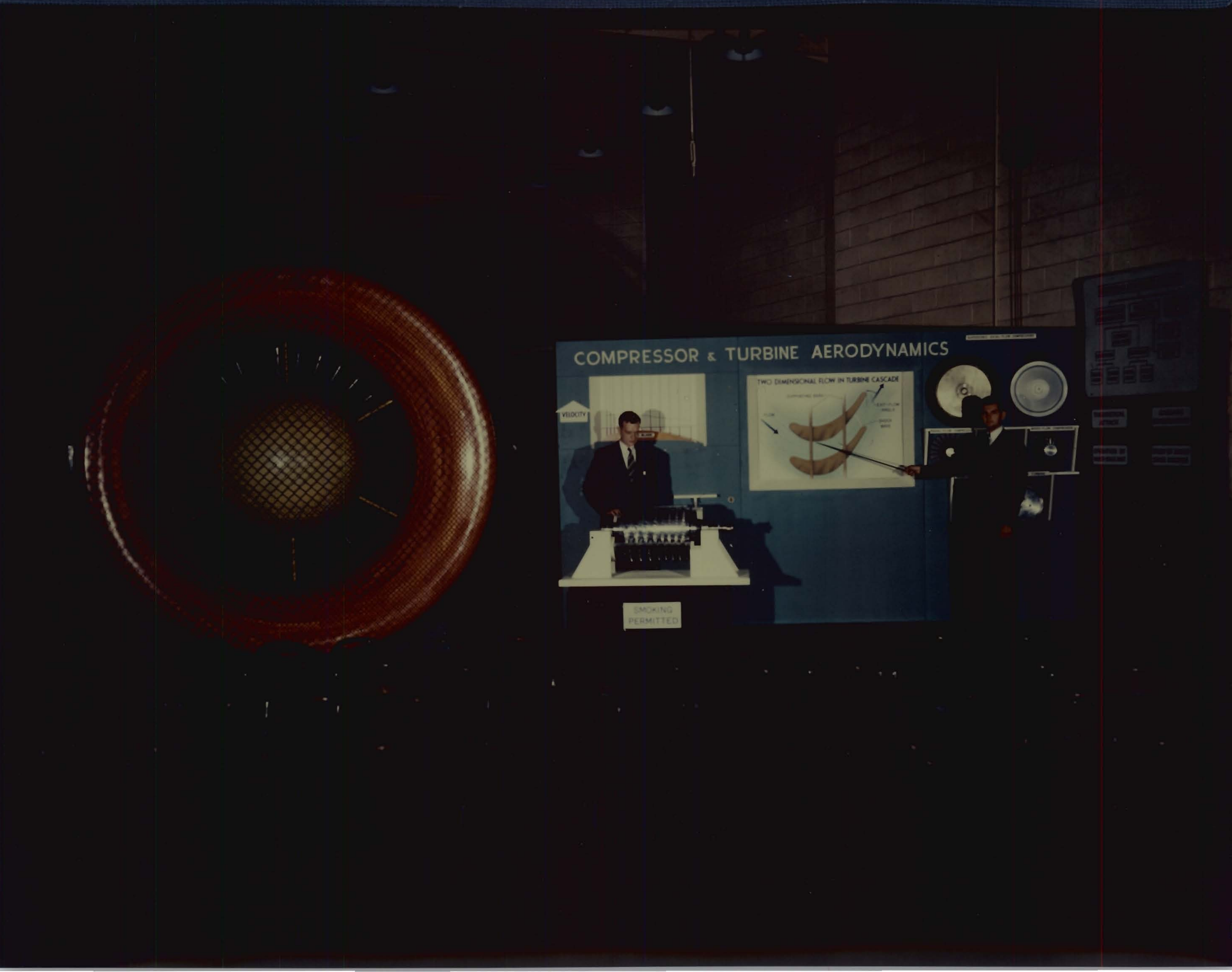
By the use of methods described in these talks, we feel that we will attain our goal of producing efficient, low-cost, light-weight units which can be designed quickly and accurately with a minimum of expensive development work. This concludes the presentation of our research in compressor and turbine aerodynamics. On your way to the next exhibit on turbine cooling, you will be shown a display of our compressor and turbine instrumentation, which is one of the major problems.

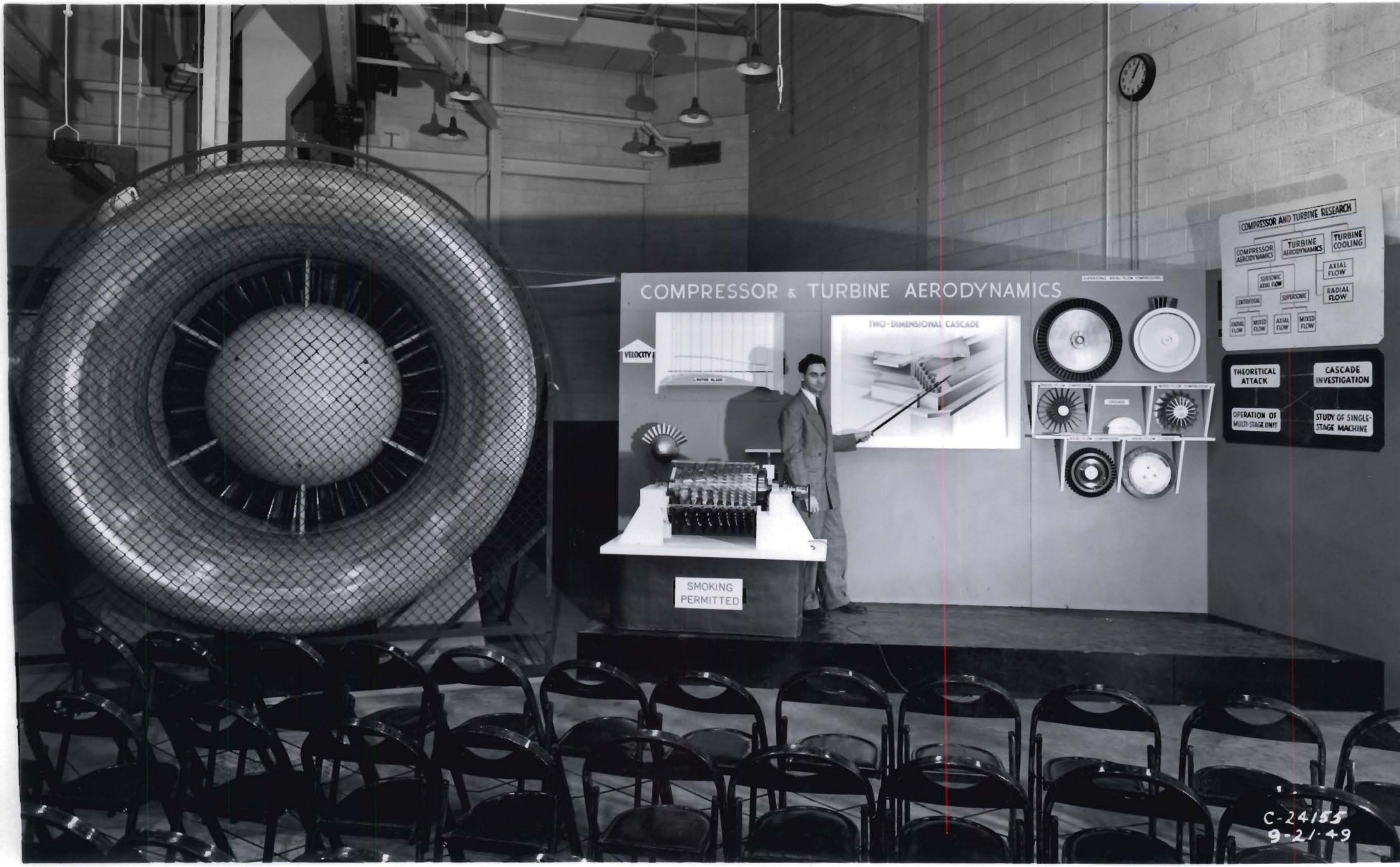
B COMPRESSOR & TURBINE AERODYNAMICS

Stage	Color Print	Compressor & Turbine Aerodynamics	
Stage	Color Print (B&W)	" " "	C-24155
Chart 1	Costello	Flow Paths and Boundary Layer	C-24191-D
Chart 2	Costello	Blade Design for Specified Conditions	C-24191-C
Chart 3	Costello	Compressor Guide Vane Turning Angle	C-24191-B
Chart 4	Costello	Streamlines in Axial-Flow Compressor	C-24182-B
Chart 5	Costello	Surface Velocities in Supersonic Cascade	C-24191-A
Chart 1	Hauser	Pressure Ratio Affects Turbine Power	C-24192-A
Chart 2	Hauser	Two-Dimensional Cascade	C-24192-B
Chart 3	Hauser	Two-Dimensional Flow in Turbine Cascade	C-24192-C
Chart 1	Montgomery	Variation in Flow Path	C-24178-C
Chart 2	Montgomery	Typical Design Problems	C-24178-B
Chart 3	Montgomery	Single-Stage Compressor Performance	C-24178-A
Chart 1	Finger	Stage Pressure Ratios Thru Compressor	C-24182-A
Chart 2	Finger	Compressor Surging Affects Engine Safety	C-24186-C
Chart 3	Finger	Modifications Increase Airflow	C-24182-C

TURBINE COOLING

Stage	Color Print	Turbine Cooling	
Stage	Color Print (B&W)	Turbine Cooling	C-24157
Chart 1	Arne	Higher Temperatures Increase Power	C-24181-C
Chart 2	Arne	Cooling Method Affects Power	C-24181-A
Chart 3	Arne	Blade Material Affects Power	C-24181-B
Chart 1	Rossbach	Passage shape Affects Heat Transfer	C-24185-B
Chart 2	Rossbach	Gas Flow Around Turbo Blades	C-24185-A
Chart 3	Rossbach	Cascade Heat Transfer Checks Theory	C-24186-A
Chart 4	Rossbach	Cascade and Single Stage Heat Transfer	C-24186-B


COMPRESSOR & TURBINE AERODYNAMICS
TWO DIMENSIONAL FLOW IN TURBINE CASCADE
VELOCITY
SMOKING
PERMITTED


COMPRESSOR & TURBINE AERODYNAMICS
VELOCITY
TWO-DIMENSIONAL CASCADE
COMPRESSOR AND TURBINE RESEARCH
COMPRESSOR AERODYNAMICS
TURBINE AERODYNAMICS
TURBINE COOLING
AXIAL FLOW
RADIAL FLOW
SUPERSONIC
CENTRIFUGAL
RADIAL FLOW
MIXED FLOW
AXIAL FLOW
MIXED FLOW
THEORETICAL ATTACK
CASCADE INVESTIGATION
OPERATION OF MULTI-STAGE UNIT
STUDY OF SINGLE-STAGE MACHINE
SMOKING PERMITTED
C-24155
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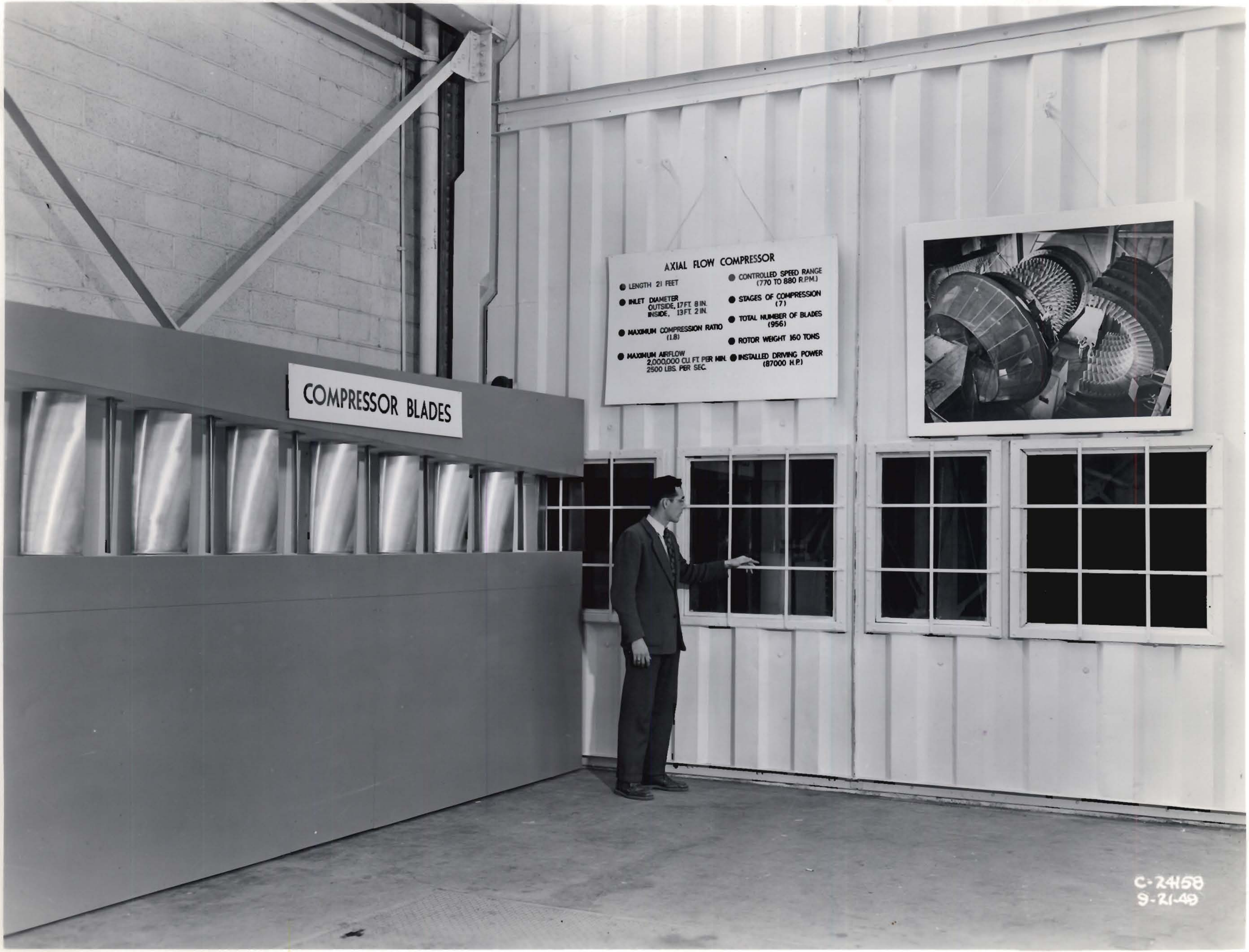
COMPRESSOR BLADES
AXIAL FLOW COMPRESSOR
LENGTH 21 FEET
INLET DIAMETER
OUTSIDE, 17 FT. 8 IN.
INSIDE, 13 FT. 2 IN.
MAXIMUM COMPRESSION RATIO
(1.8)
MAXIMUM AIRFLOW
2,000,000 CU. FT. PER MIN.
2500 LBS. PER SEC.
CONTROLLED SPEED RANGE
(770 TO 880 R.P.M.)
STAGES OF COMPRESSION
(7)
TOTAL NUMBER OF BLADES
(956)
ROTOR WEIGHT 160 TONS
INSTALLED DRIVING POWER
(87000 H.P.)
C-24158
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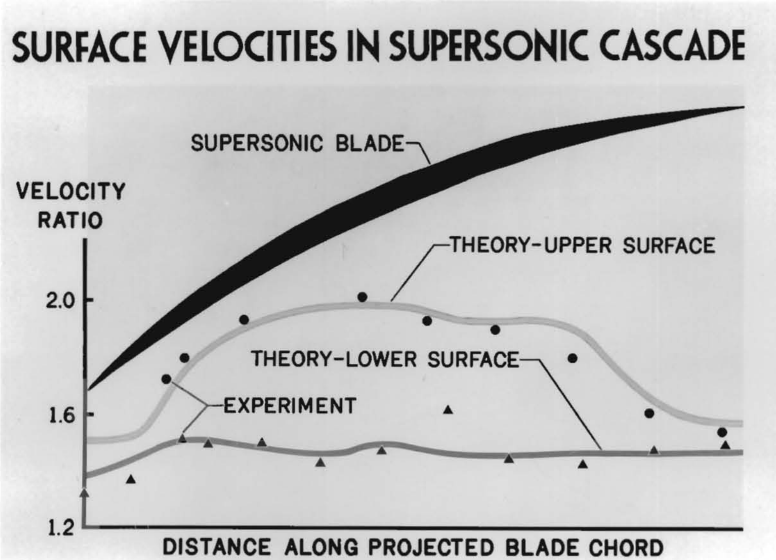

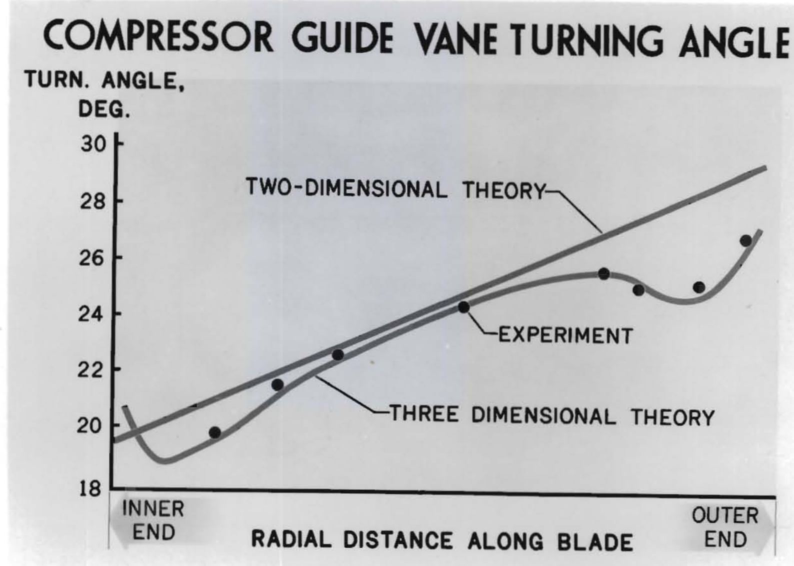

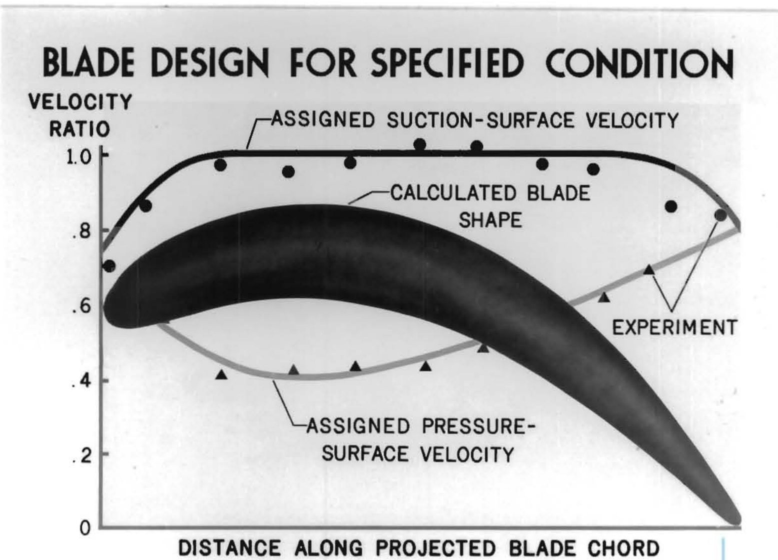

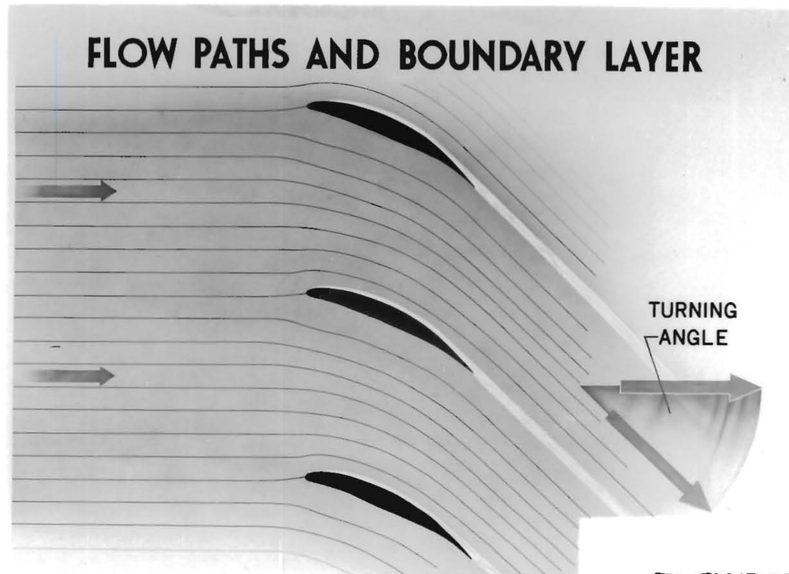

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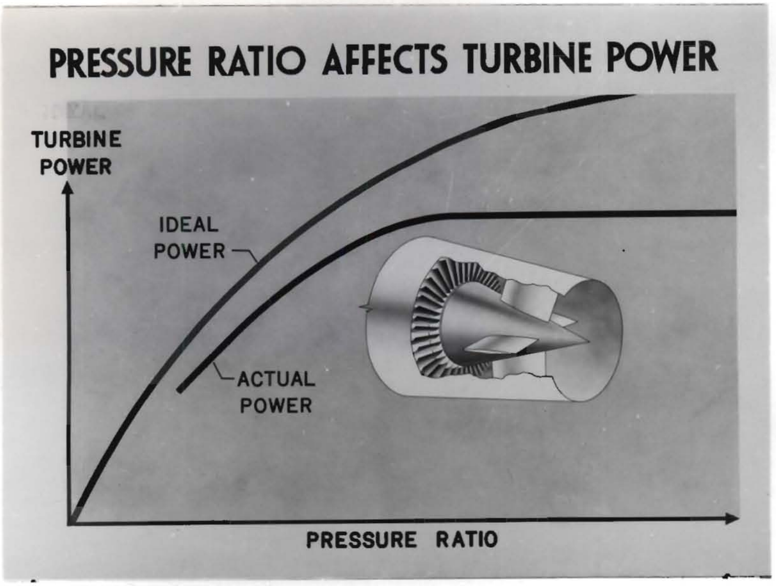

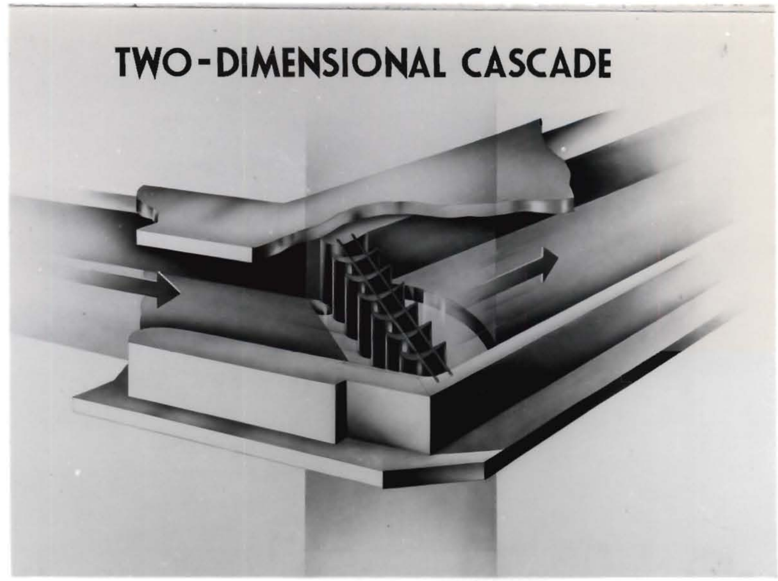

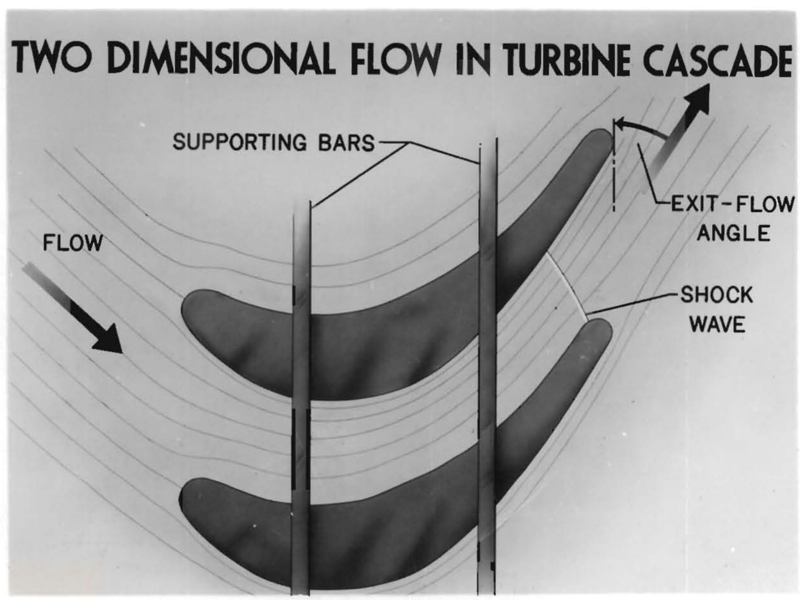

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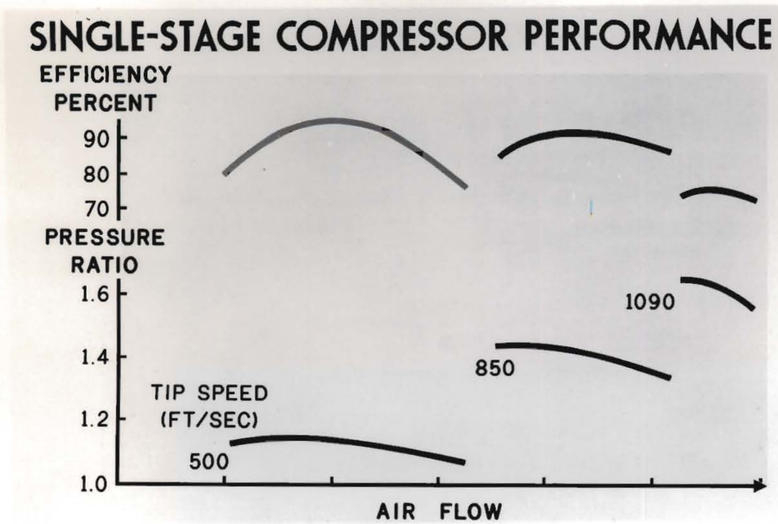

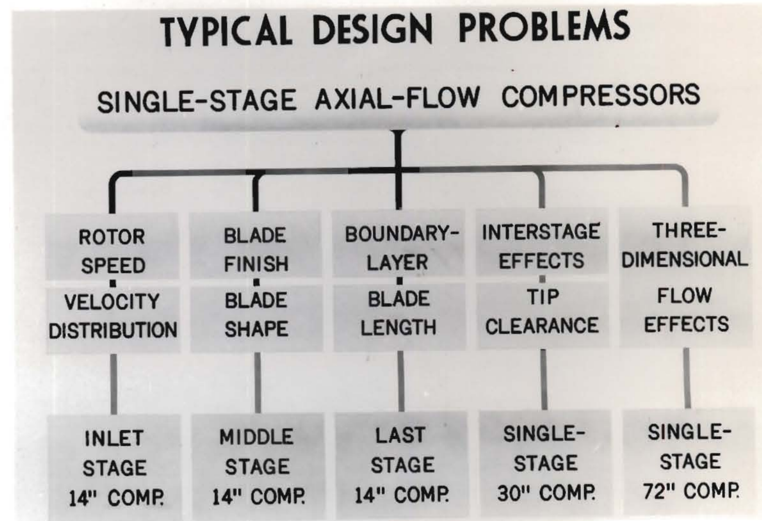

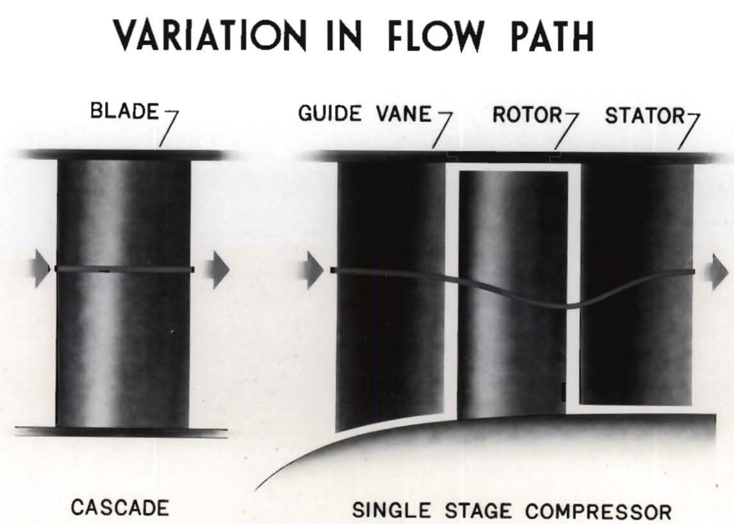

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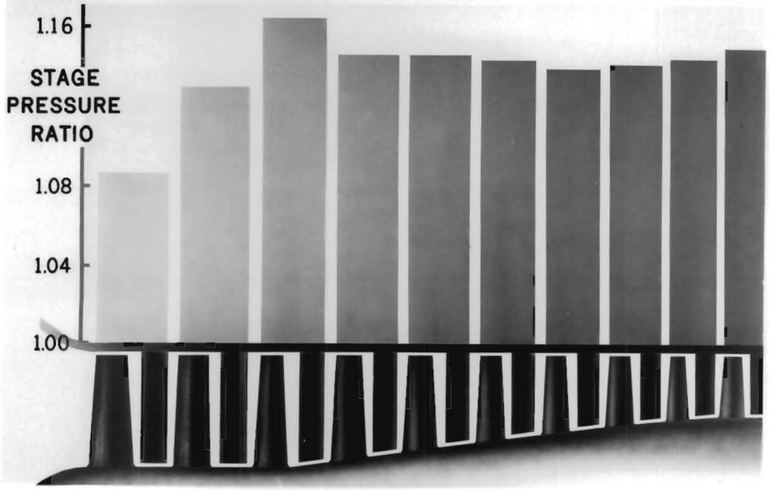
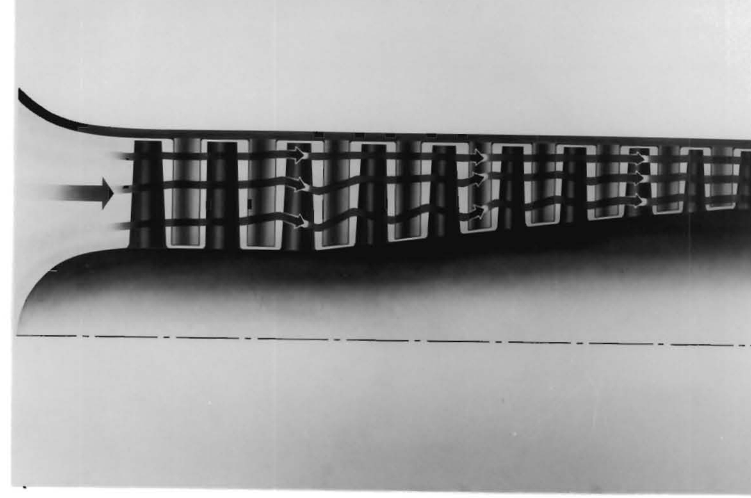
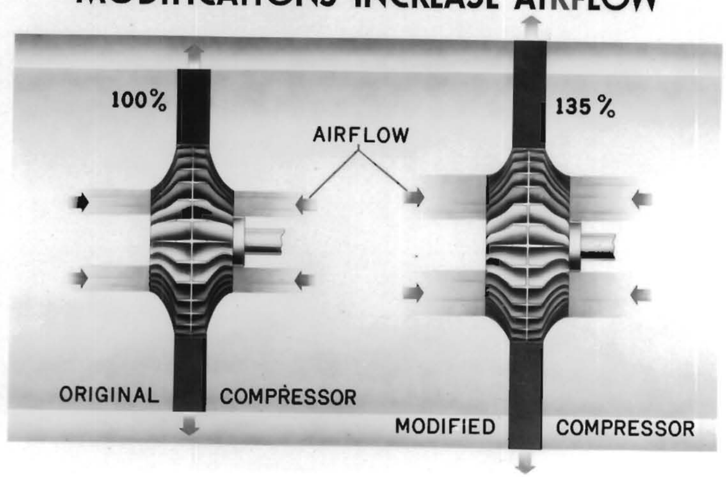
STAGE PRESSURE RATIOS THRU COMPRESSOR



STREAMLINES IN AXIAL-FLOW COMPRESSOR



MODIFICATIONS INCREASE AIRFLOW



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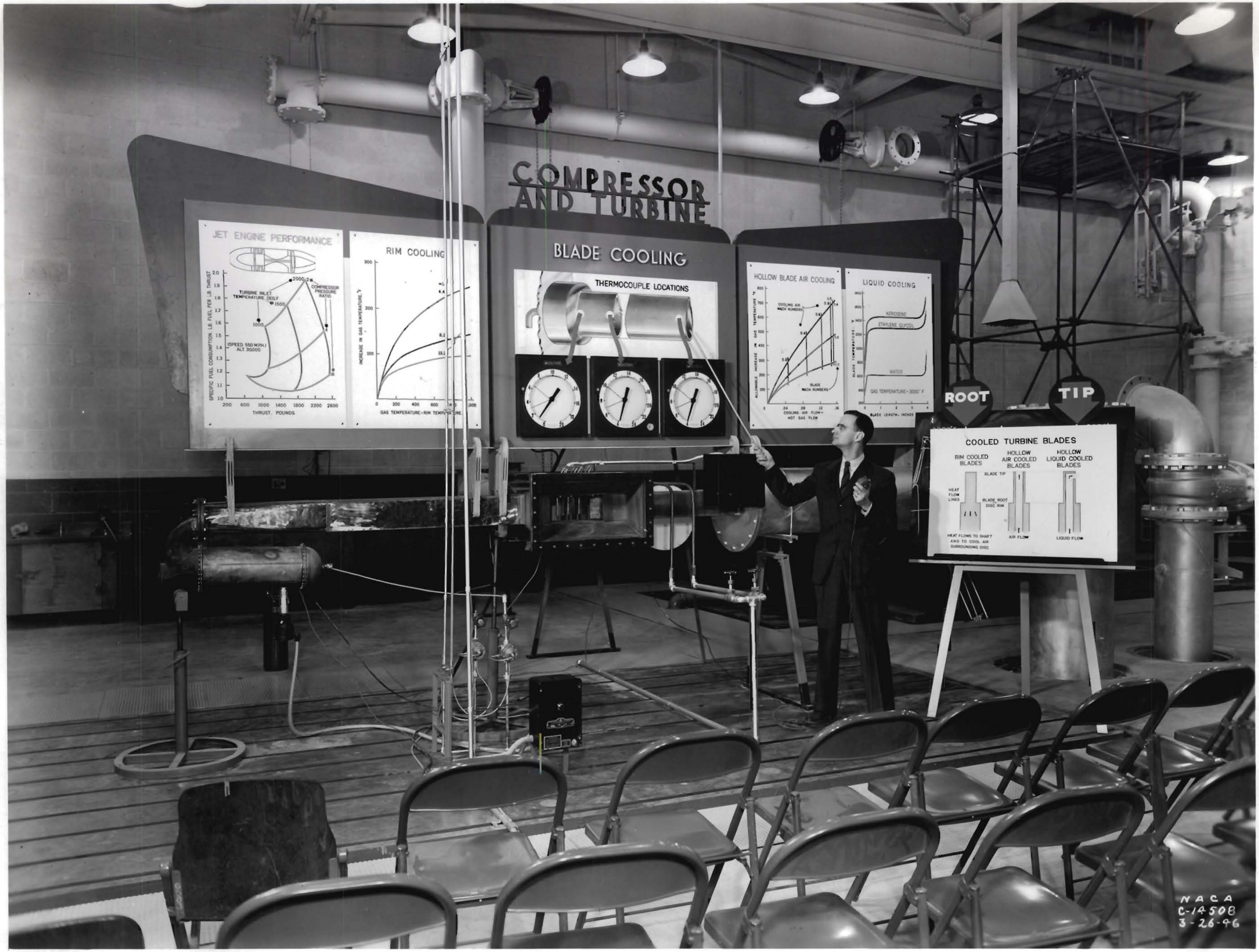

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