I MIXED FLOW COMPRESSOR RESEARCH - CE-18, ERB

GENERAL DISCUSSION OF PROBLEMS ASSOCIATED WITH MIXED-FLOW COMPRESSORS

R. O. Bullock

Two fundamentally distinct classes of compressors are used in modern gas turbine engines. One type is the axial-flow compressor in which the blades are accurately formed airfoil sections. Several hundred stationary and several hundred rotating blades may be required in this class to obtain compression ratios of the order of three or four to one. In the second class of compressors this compression ratio is obtained with something like thirty rotating blades. Examples of the rotating blades, or impellers, of this class are shown by these three models on display (See fig. 1.) and by this double entry unit. For the lack of a better name, we call all compressors of this second class mixed-flow compressors principally because large axial and large radial components of velocity exist simultaneously. The blades of these impellers do not resemble airfoils and are rather long in the direction of flow path and comparatively narrow in heighth. At the tips of these blades circumferential velocities exceeding 1600 feet a second may exist and an extensive diffusion system is required to transform this kinetic energy into pressure.

Although these compressors offer the attractive advantages of mechanical simplicity and ruggedness they do not yet compare with the axial-flow compressor as far as weight flow per unit frontal area or as far as efficiency is concerned. The progress being made to improve the flow capacity of these compressors is illustrated in the first chart (fig. 2) where we compare the flow capacity of four mixed-flow compressors, each having an over-all diameter of three feet. The ordinate of this chart is weight flow in pounds per second. A compressor modeled after an early supercharger would have a flow capacity less than 20 pounds per second. One modeled after a modern commercial compressor would have a weight flow capacity of approximately 45 pounds per second. A compressor containing the experimental impeller reported in NACA Technical Note 1216 would have a flow capacity of over 60 pounds per second. An axial-discharge impeller, which is being investigated in the experimental setup in this room, would have a flow capacity of approximately 90 pounds per second. This impeller will be the subject of the third talk given in this room.

The efficiency of the mixed-flow compressors is also being improved. Six or seven years ago an efficiency of sixty percent for a pressure ratio of three to one would have been remarkable. We are now obtaining this pressure ratio with an efficiency of 78 percent.

An insight into the phenomena limiting the flow capacity and efficiency of these compressors is given in the next chart (fig. 3) where we show a typical example of the static pressure rise through a compressor. If an impeller and a vaneless diffuser are set up in the manner illustrated, and static pressures are measured along this surface, and take ratio of these pressure to that of the inlet and plot as ordinate with axial distance as abscisae, we obtain this curve for the flow at maximum efficiency. This curve is obtained for operation at maximum flow.

Now when we compare these two curves with that for ideal flow we find that a large drop in pressure occurs near the impeller inlet. At maximum flow this drop is so large that acoustic velocities are encountered and it is this phenomenon which limits the flow capacity of the impeller. Research has demonstrated that this loss in static pressure is intimately associated with the curvature of the front shroud. In general, the smaller the radius of curvature, the greater the pressure losses and the smaller will be the flow capacity of the impeller. Throughout the remainder of the impeller where compression is largely obtained by centrifugal force, the three curves are nearly parallel to each other, and the inference is that compression of this type is a rather efficient process. Some random deviations of the three curves are noted in the diffuser region, but we can obtain more information on the events here by means of the third chart (fig. 4) where we show the performance of a typical vaneless diffuser.

This figure shows the tip of the impeller and a complete vaneless diffuser. If we measure the losses between the impeller tip and some point in the diffuser passage and take the ratio of this loss to that occurring in the complete diffuser, plot that ratio as ordinate against distance from impeller tip as abscisae, we obtain this curve. Of outstanding importance is the fact that 80 percent of the losses in the diffuser occur in the first 2-1/2 inches. The cause of these high losses is the fact that the flow leaving the impeller is so highly turbulent and the distribution of velocities is so highly chaotic that large losses must occur. Obviously, the brunt of the research must be devoted to improving the flow in the impeller and thus preclude the possibility of encountering these losses. In the meanwhile, however, we must also devote some effort to insure that these mixing losses are kept to the absolute minimum. The following speaker will show how the presence of vanes in this region greatly increased the normal losses and seriously limited the flow capacity of the compressor. He will also show how this situation was remedied by a rather simple modification to the vanes in this region.

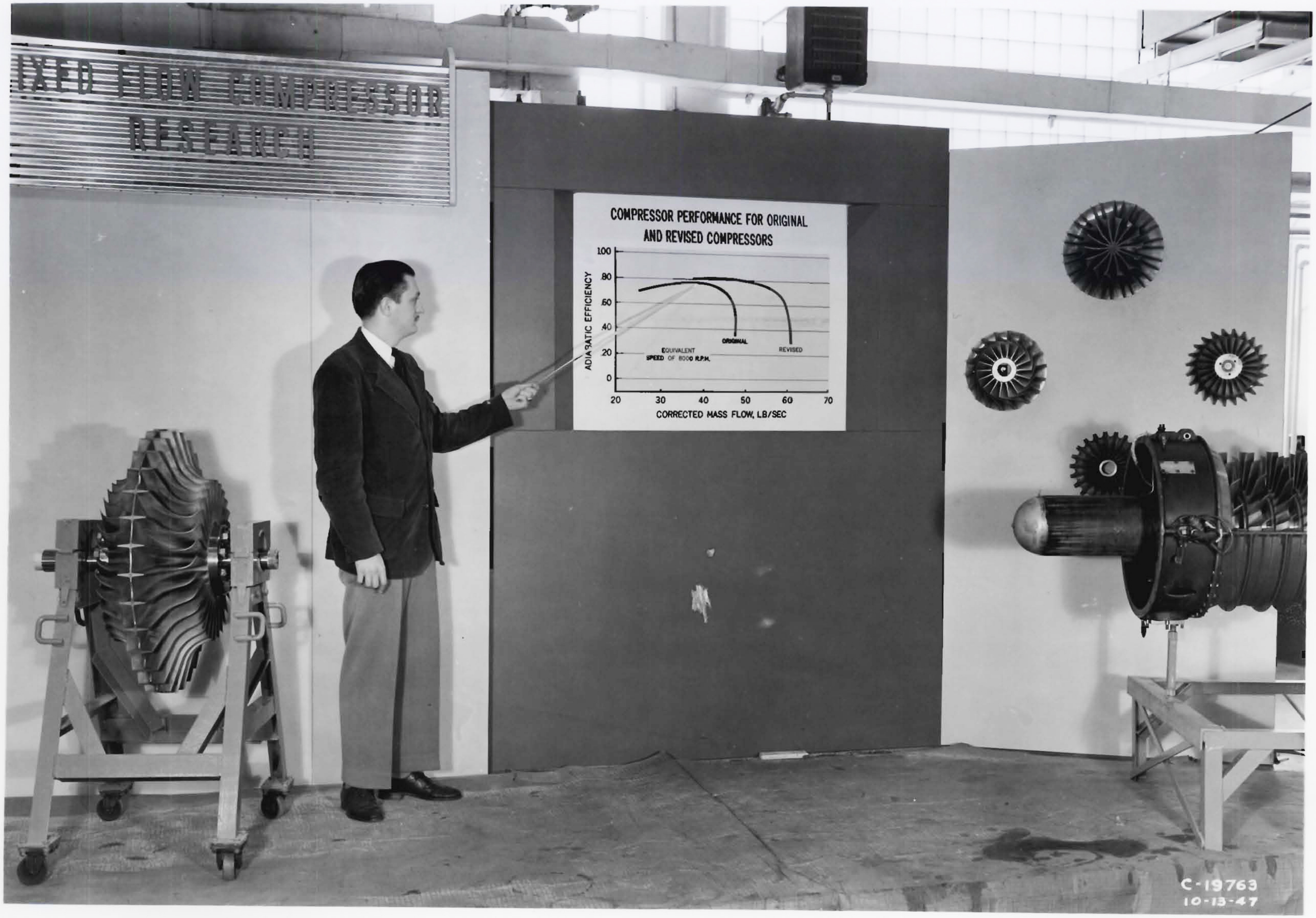

IXED FLOW COMPRESSOR
RESEARCH
COMPRESSOR PERFORMANCE FOR ORIGINAL
AND REVISED COMPRESSORS
ADIABATIC EFFICIENCY
1.00
.80
.60
.40
.20
0
EQUIVALENT
SPEED OF 8000 R.P.M.
ORIGINAL
REVISED
20
30
40
50
60
70
CORRECTED MASS FLOW, LB/SEC
C-19763
10-13-47

Fig 1

NACA

COMPARISON OF FLOW CAPACITIES OF FOUR MIXED-FLOW COMPRESSORS, OVERALL DIAMETER = 3 FT

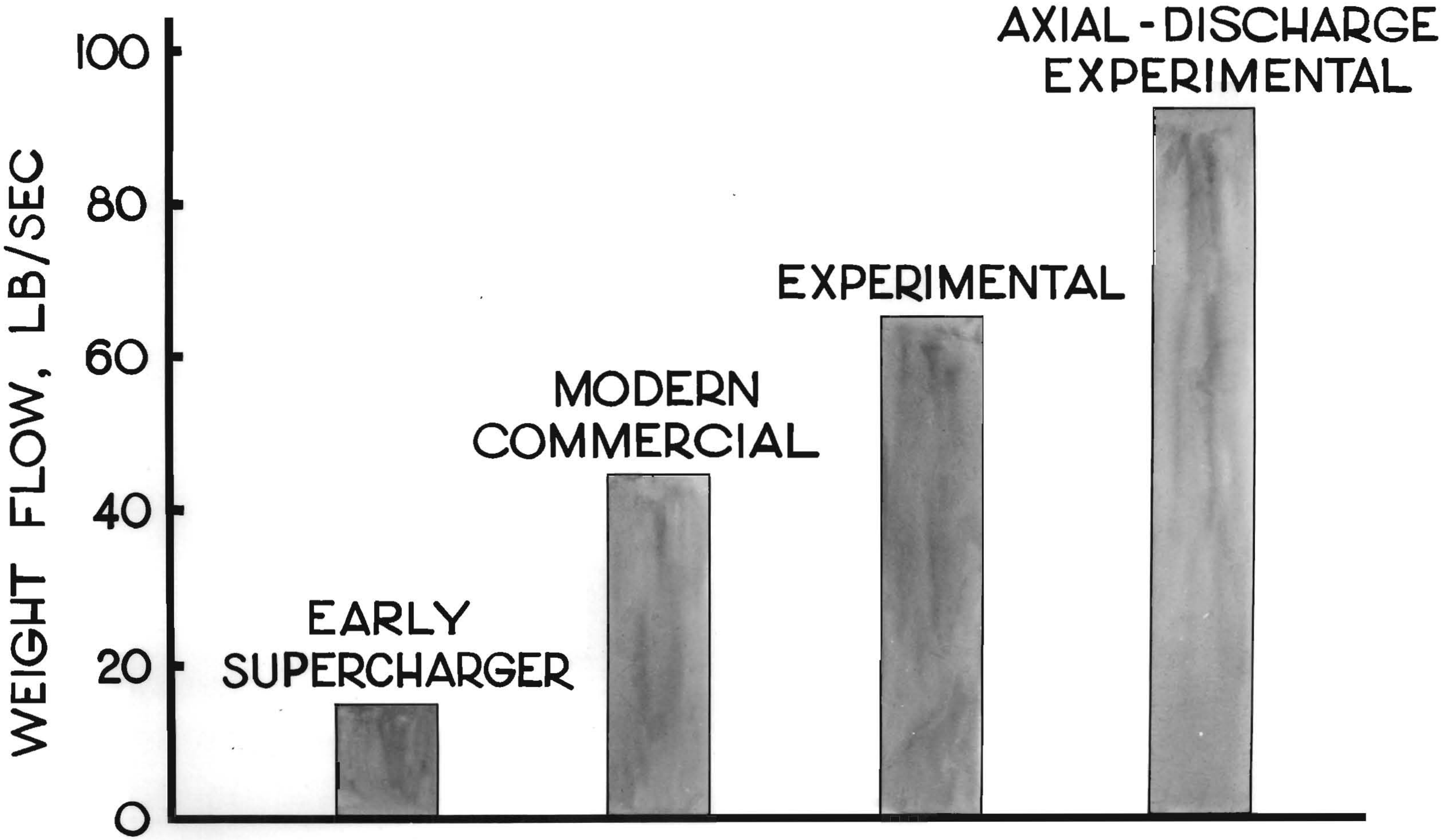
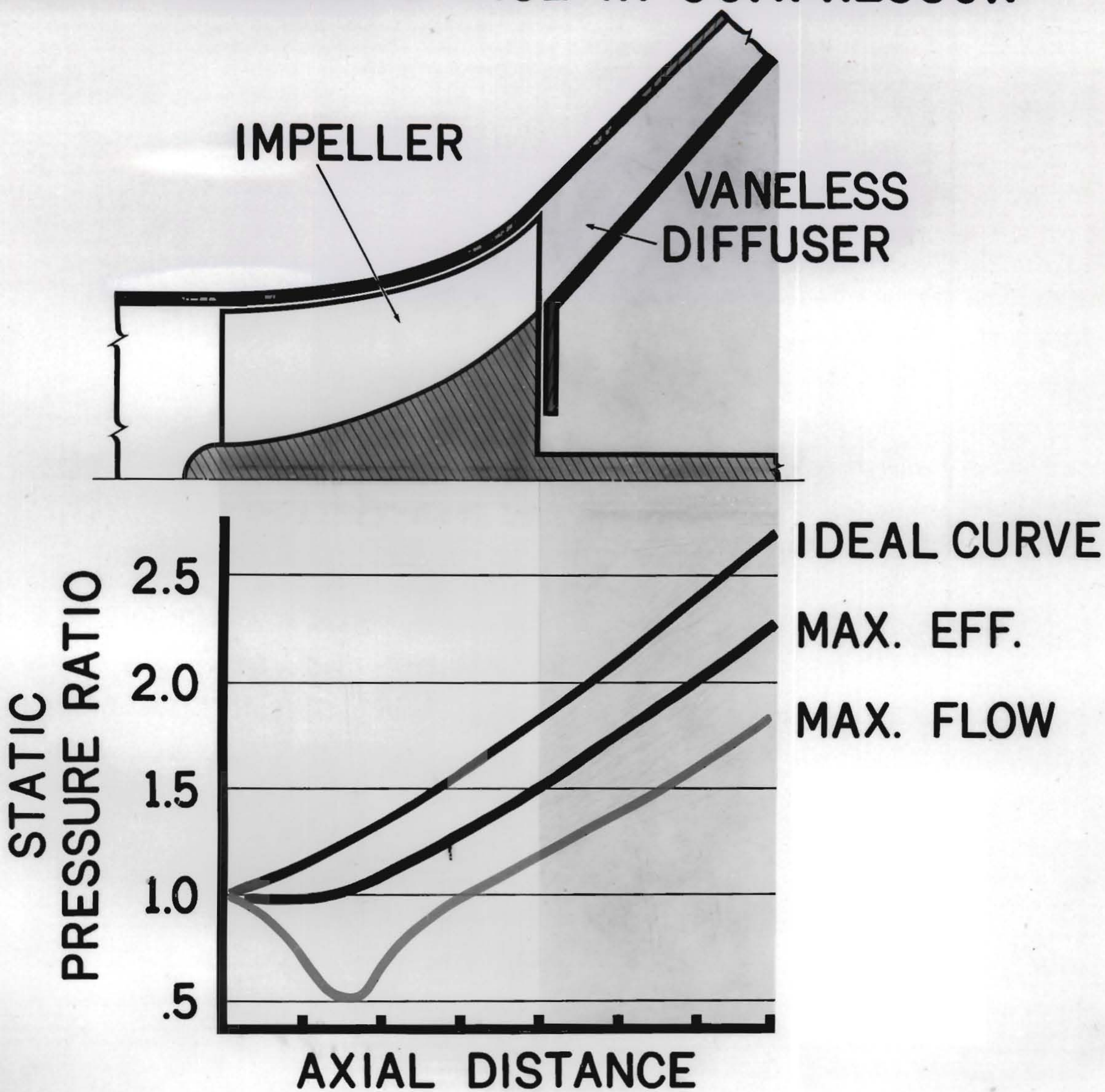


Fig 2

C- 19742
10-9-47

NACA

STATIC-PRESSURE RISE IN COMPRESSOR

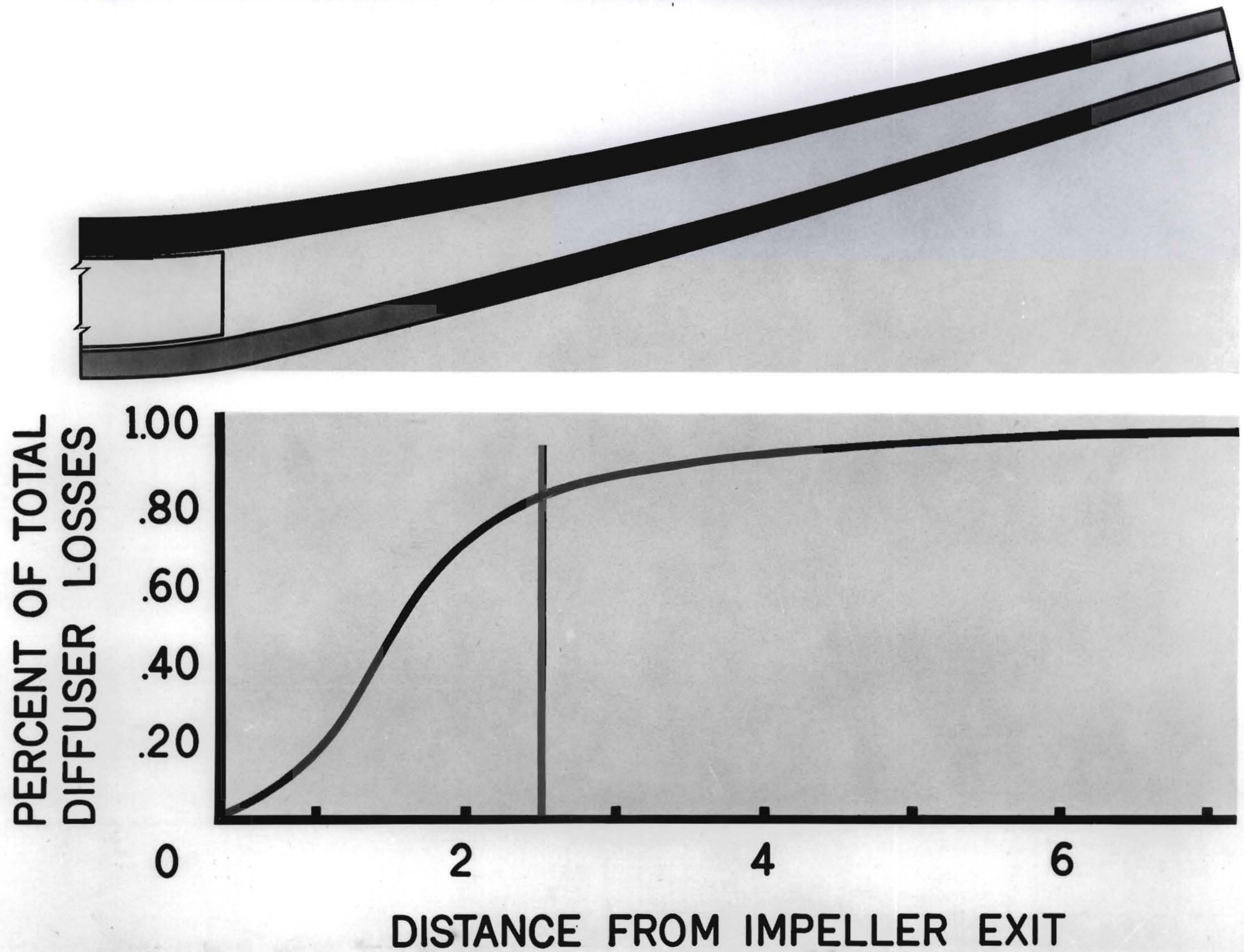


C- 19827
10- 24- 47

Fig 3

NACA

PERFORMANCE OF TYPICAL VANELESS DIFFUSER



C- 19845
10- 24- 47

Fig 4

NACA

APPLICATION OF FUNDAMENTALS TO IMPROVEMENT OF A LARGE MIXED-FLOW COMPRESSOR

A. Ginsburg

One part of the mixed-flow compressor research program concerns the performance investigation and analysis of large mixed-flow compressors.

One phase of this research program will be discussed to show how by the use of the fundamental compressor design information the performance of a large mixed-flow compressor being investigated for the Air Forces was improved considerably. The compressor from the Packard XJ-41-V turbojet engine was used in this investigation.

The accompanying chart (fig. 5) shows a schematic diagram of the compressor. The compressor has an over-all frontal diameter of 48 inches. The single-entry impeller is 32 inches in diameter and has a relatively large inlet- to-discharge tip diameter ratio which results in transonic Mach numbers at the inlet at design operating conditions. The impeller is of the mixed-flow type with a greater part of ideal compression occurring as a result of the increase of radius of rotation and a lesser amount as a result of diffusion of the relative velocity. The impeller is followed by a short vaneless diffuser section and then a vaned collector. An annular section simulates the engine burner annulus.

Because of the compressor size and compactness of design it was possible to instrument the compressor extensively and thereby enable a functional evaluation of each of the compressor components.

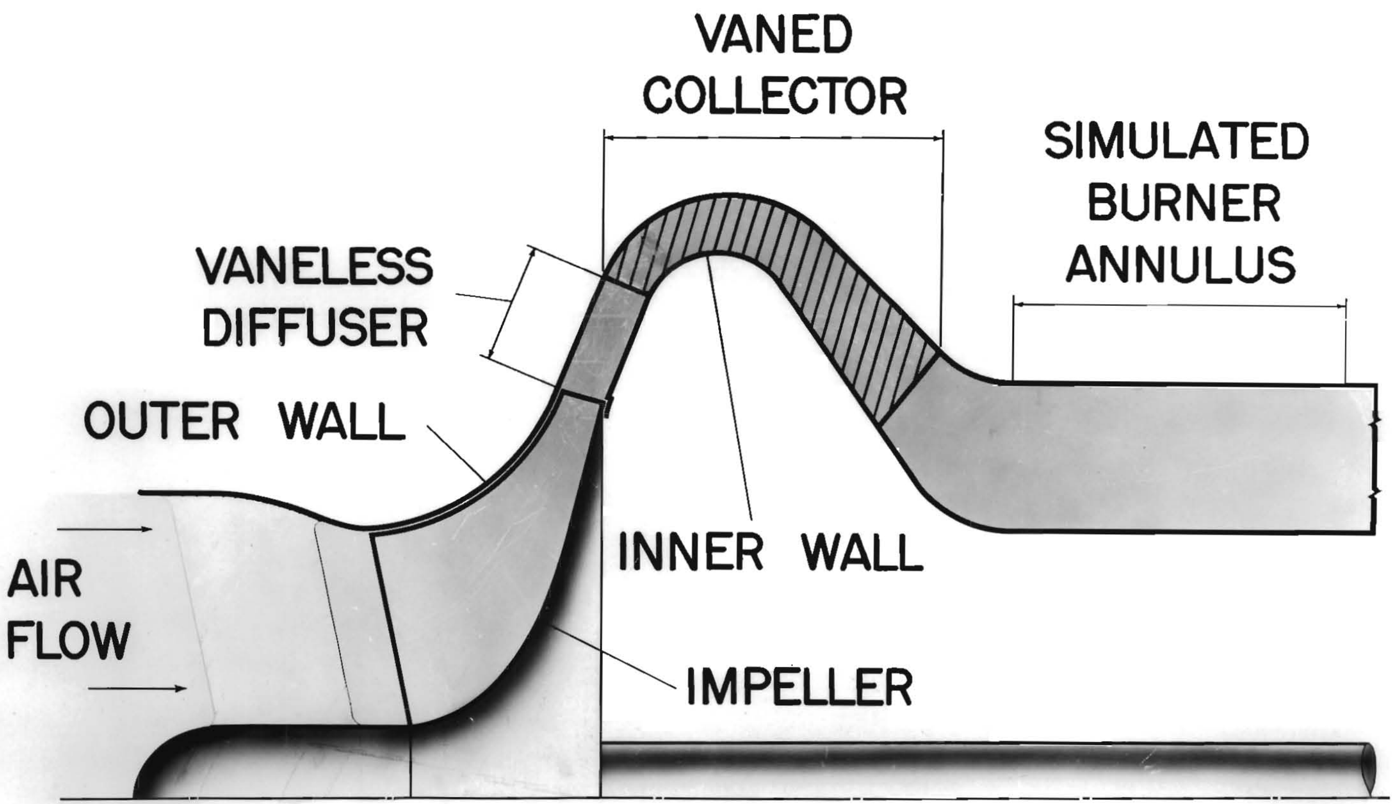
The accompanying chart (fig. 6) shows the performance of the impeller, diffuser, and compressor. Efficiency is plotted against weight flow for speed of 8000 rpm. The impeller curve shows a highly efficient impeller, the flatness indicates that the impeller pressure losses that accompany flow choking in an impeller have not occurred. The diffuser curve shows relatively low efficiencies and peaked operation over the flow range with the diffuser curve shaping the compressor curve. The sharp drop at high flows indicate large pressure losses that accompany flow choking.

To investigate further the location of the choke point in the compressor the next chart (fig. 7) shows the pressure rise along the compressor flow path. For peak compressor efficiency point a steady rise in pressure existed along the entire flow path. For maximum flow a steady rise in pressure occurred in the impeller and vaneless diffuser, however a very large pressure drop occurred at the entrance to the vaned collector, indicating the point where the flow was restricted.

Analysis of the air stream showed that Mach numbers as high as 1.5 existed in the vane entrance and that the area extrenum resulted from flow separation off the outer passage wall and off the low pressure side of the vanes. The flow separation was instigated by poor air-flow distribution in the vaneless diffuser. As a result of this analysis the compressor was revised to get possible mass flow and efficiency improvements. The vaneless diffuser diameter was increased by cutting back the vane entrance edge which resulted in a larger geometric flow area at the vane entrance and also allowed for a better stabilized air-flow condition at the vane entrance.

The next chart (fig. 8) shows compressor efficiency and weight flow before and after revision. Redesigning the vane entrance resulted in an improved efficiency from 0.78 to 0.81 and allowed an increase in mass flow of 25 percent. Thus by increasing the mass flow capacity of the diffuser there resulted a full utilization of the compressor's highly-efficient and high-capacity impeller.

AIR FLOW PATH THROUGH COMPRESSOR

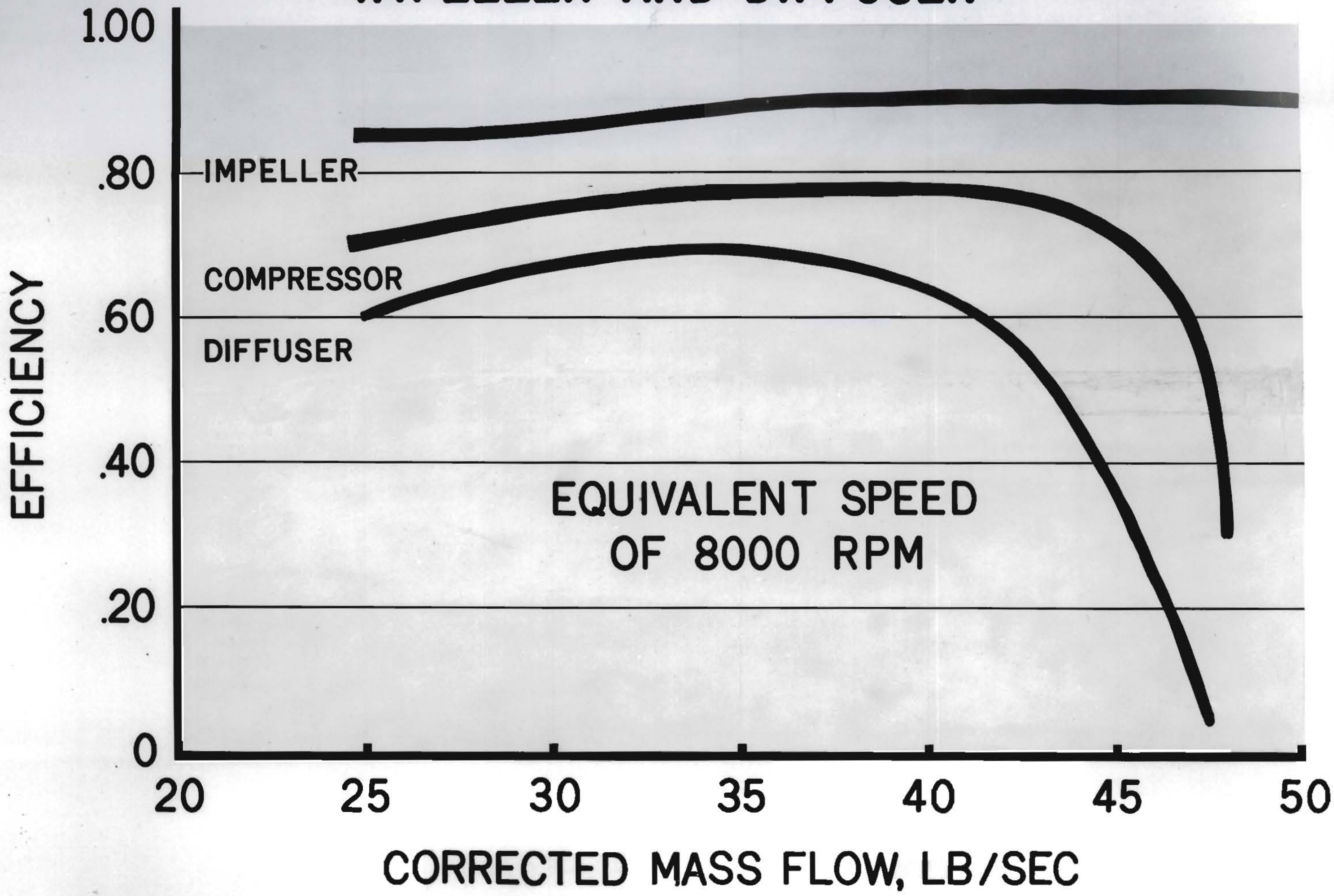


C- 19824
10-24-47

Fig 5

NACA

PERFORMANCE COMPARISON OF COMPRESSOR, IMPELLER AND DIFFUSER



C- 19828
10-24-47

Fig 6

NACA

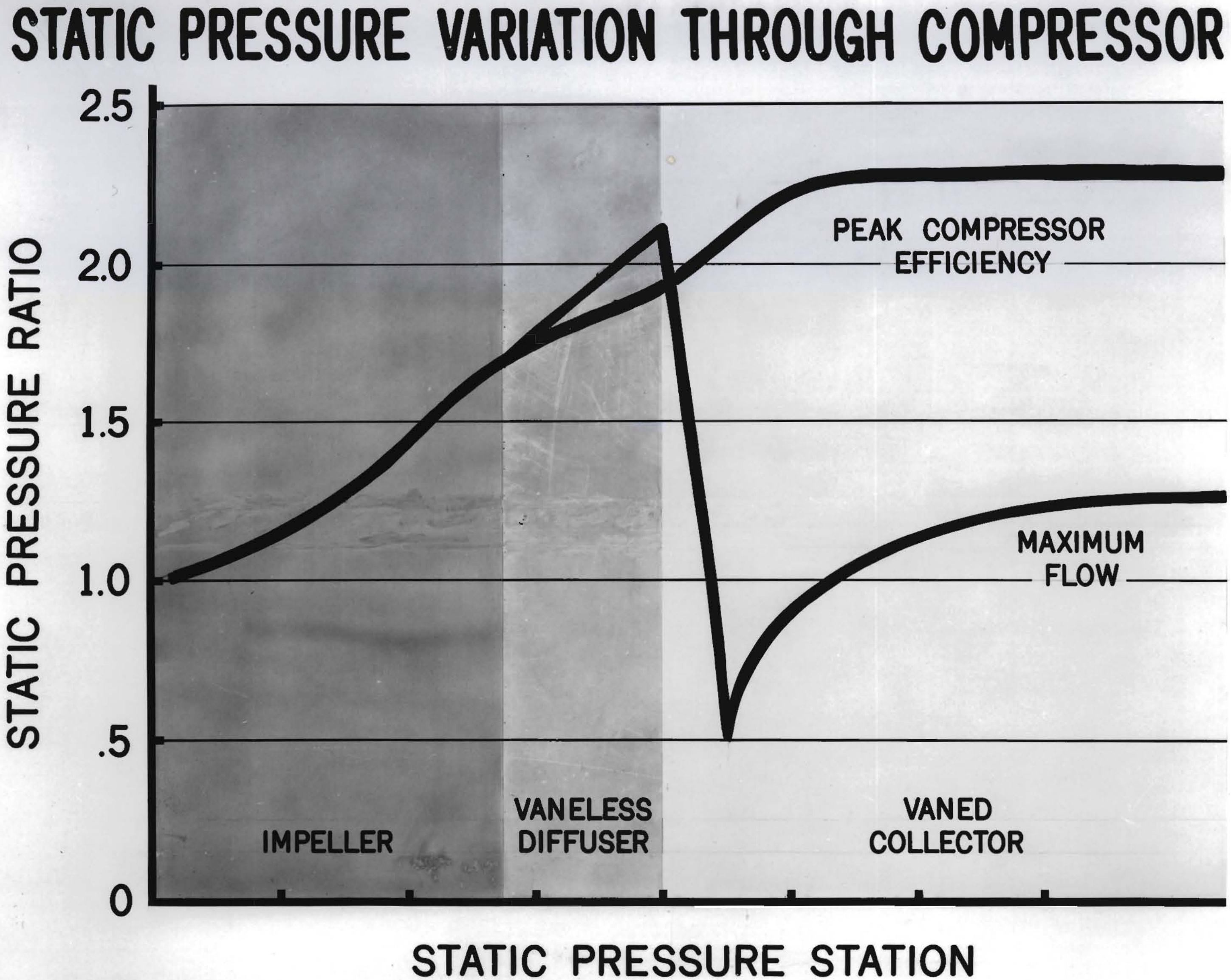
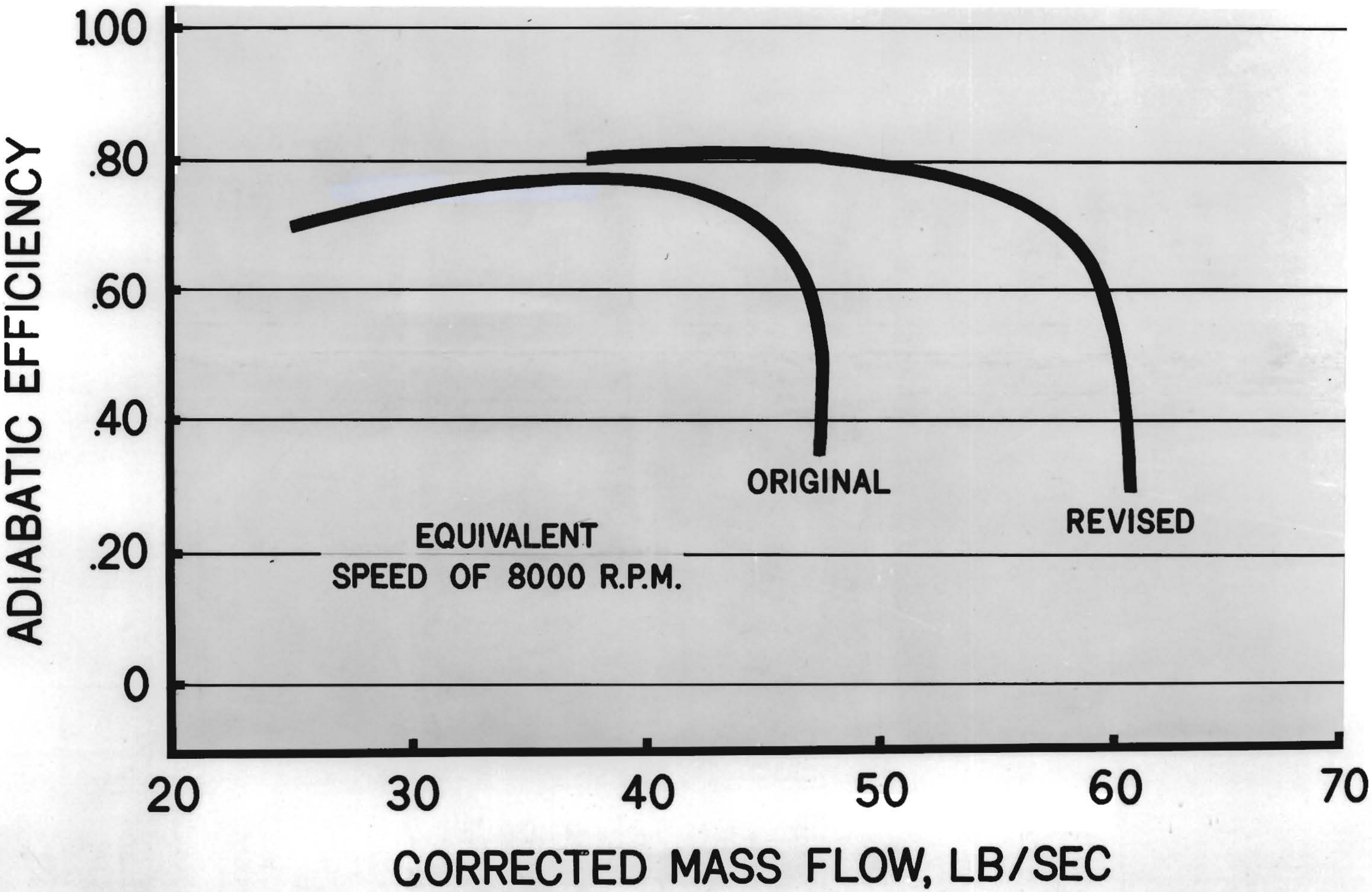


Fig 7

C- 19830
10- 24- 47

NACA

COMPRESSOR PERFORMANCE FOR ORIGINAL AND REVISED COMPRESSORS



C- 19879
10- 24- 47

Fig 8

NACA

DESIGN AND PERFORMANCE OF NACA

AXIAL-DISCHARGE MIXED-FLOW COMPRESSORS

R. Eschborn

It has been shown in the first discussion that the conventional mixed-flow impeller has a limiting flow condition which becomes more severe as the radius of curvature of the front shroud decreases. Further, as we decrease this radius, the over-all diameter or frontal area increases for a given entrance area. Then too, use of radial diffusion also increases the over-all diameter. As a result, investigation of a mixed-flow compressor with pure axial-discharge was considered a promising solution to relieve simultaneously this possible limiting flow condition and to increase the air flow to a maximum per unit of frontal area. By increasing the air flow to a maximum we obtain high jet power in a turbojet engine for minimum over-all diameter.

The decision then, was made to design an axial-discharge compressor for high mass flow and preassigned pressure ratio regardless of the shape of the resulting impeller. The impeller shape which first occurs as the most likely is as shown in part 1 of figure 9, in which the radius of curvature of the outer shroud is infinite, and in which we have maximum possible flow area for any limiting over-all diameter. Because of compression however, the entire area at the exit is not needed, so that the hub diameter is increased as shown here in part 2 of figure 9.

In part 2 the high peripheral speed at the entrance blade tip gives a low axial velocity, if the desired wheel speed is to be maintained and if the relative gas velocity entering the impeller is limited to some predetermined value.

To increase the flow with these limitations we have two possibilities:

(1) prerotate the air, by fixed vanes, so as to obtain much higher flow, and

(2) decrease the radius at the entrance

In the first case; however, prerotation cause a loss of work which the impeller imparts to the gas and in the second case the total area for flow is reduced; however, this is partly compensated for by the increase in centrifugal compression. Thus it was found that for optimum combined weight flow and pressure ratio, both prerotation and radial tip flow were needed, giving a final shape as shown in part 3 of figure 9.

This is a model of the present NACA axial-discharge impeller which can be seen installed in this test rig as you leave the room. The present impeller has a maximum diameter of 14 inches and was designed for a tip speed of 1480 feet per second, a pressure ratio of 3.5 and a weight flow of 19.7 pounds per second.

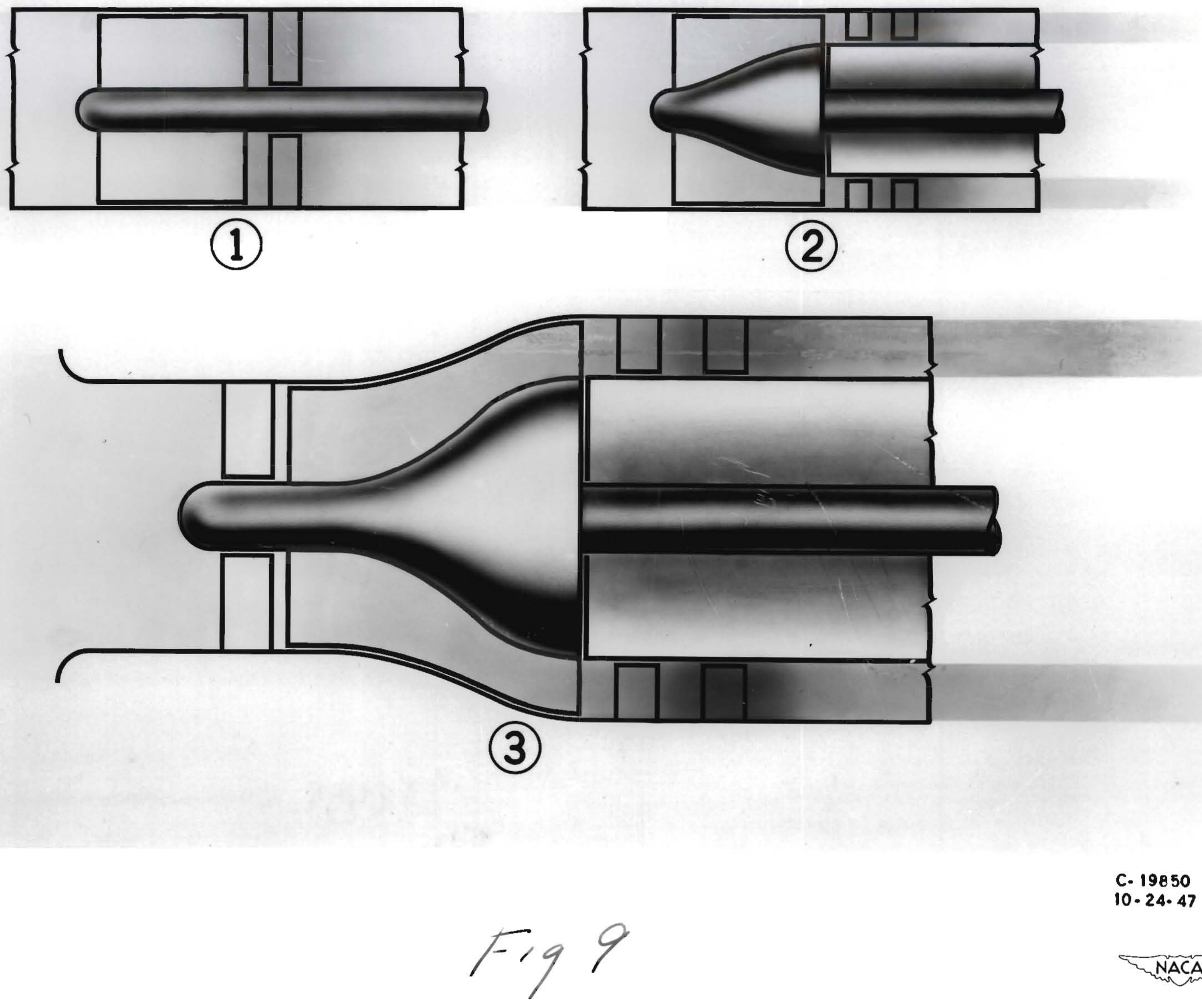
Regarding the performance of this impeller, the measured efficiencies without a diffuser are between 78 and 88 percent over the range of pressure ratios. As to the flow through the impeller, the following chart (fig. 10) showing static pressure ratio plotted against axial passage distance illustrates that we have no limiting flow condition, up to the maximum flows obtained. The maximum efficiency curve and the maximum flow curve both approach the desired flow curve. On this chart the static pressure ratio of 1.8 at the impeller exit is low when compared to the total pressure ratio of the impeller of 3.5, since the velocity components leaving the impeller are quite high.

Additional performance data on the present impeller are presented on the next chart (fig. 11) in which we have total pressure ratio plotted against weight flow in pounds per second, for impeller tip speeds from 1100 to 1480 feet per second. It should be noted that these curves are extremely flat over a wide range of weight flows. At the design speed of 1480 feet per second, we obtained a weight flow of 18 pounds per second at a total pressure ratio of 3.4. Now, this is slightly lower than the design weight flow of 19.7 pounds per second; however, this weight flow is limited by the capacity of the present test rig.

As previously pointed out on the following chart (fig. 2) for compressors of comparable diameter, this high weight flow means that we have obtained better than twice the weight flow of the conventional double entry compressor and approximately 75 percent of the weight flow of the best current axial-flow compressor of comparable diameter. Of course, the axial length of this type (axial-discharge) compressor is much less than that of the axial-flow and it has other advantages of simplicity, ruggedness, and ease of manufacture.

These comparisons then, based on our preliminary results are very encouraging and indicate that this type compressor (axial-discharge) has excellent possibilities in the future.

DEVELOPMENT OF A MAXIMUM-FLOW IMPELLER DESIGN



C- 19850
10-24-47

Fig 9

NACA

STATIC PRESSURE VARIATION THROUGH COMPRESSOR

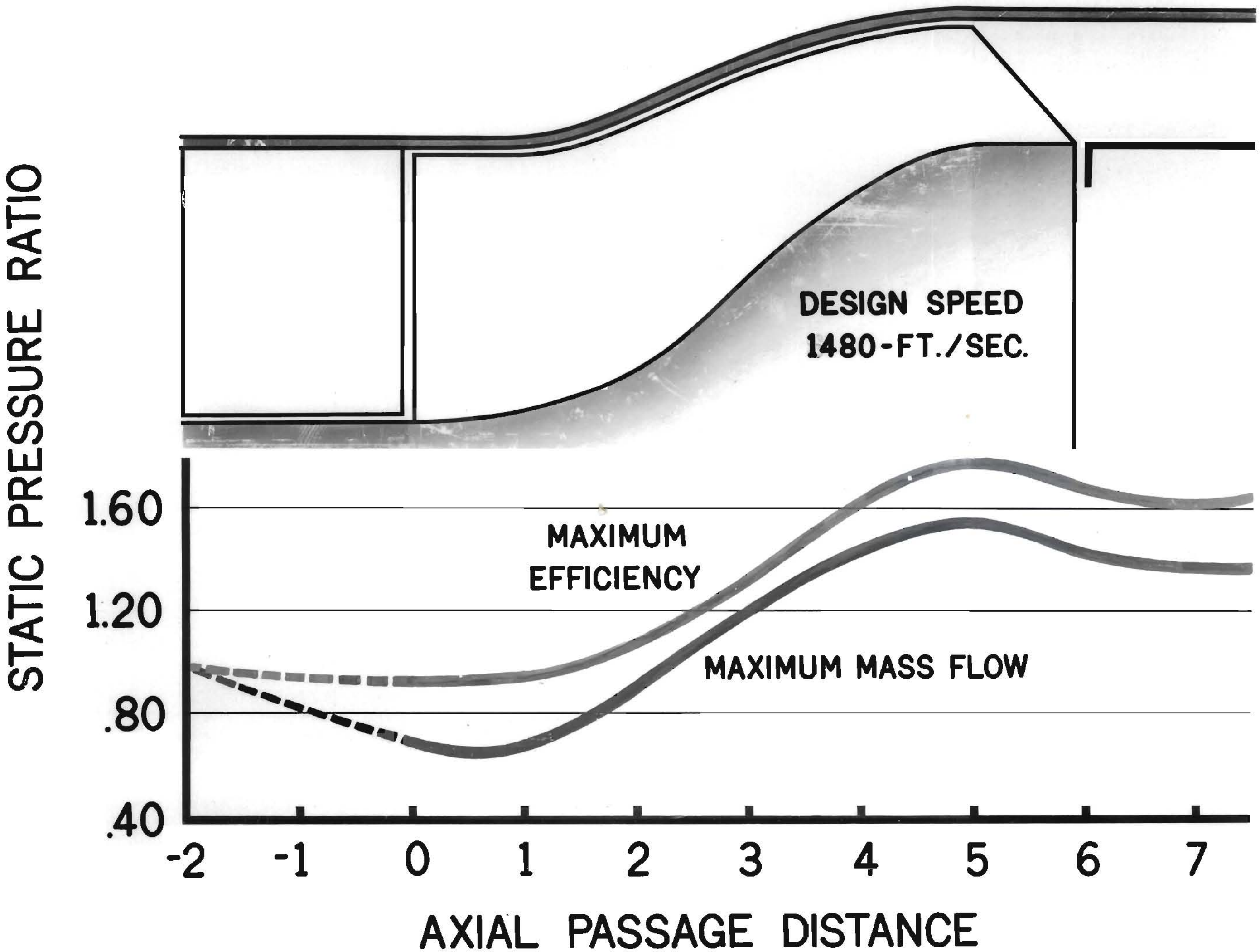
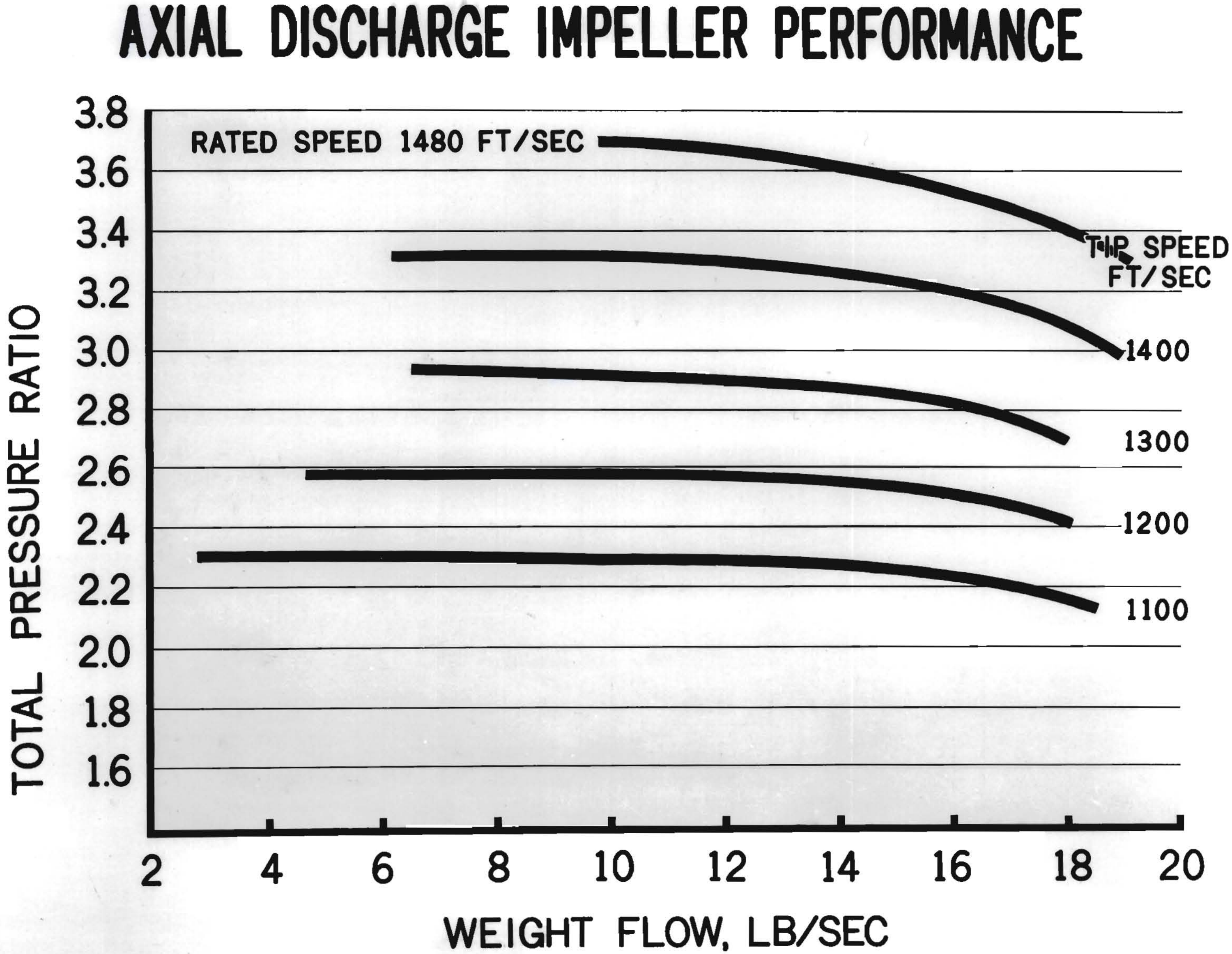


Fig 10

C-19887
10-24-47

NACA



C- 19823
10- 24- 47

Fig 11

NACA

DISCUSSION OF RANGE OF AXIAL-FLOW COMPRESSORS

J. T. Sinnette

We will describe here a few representative examples of the research on axial-flow compressors. First, I would like to point out the general objectives of the research and some of the methods used to achieve these objectives.

The efficiency of the compressor is obviously important because it is one of the main factors affecting the fuel consumption of the turbojet or turbo-propeller type of engine. Although the efficiency of the modern axial-flow compressor near its design operating condition is high compared to other types of compressors, further important gains may be expected as a result of the intensive research now being conducted. Even more important, however, is the improvement of the performance at other than design operating conditions where the efficiency of the axial-flow compressor may be quite low. This off-design performance is important for the starting and rapid acceleration of the engine and for augmented power required at take off and in combat maneuvers.

One very effective method of increasing the high-efficiency range of axial-flow compressors consist of turning the stator blades so as to give favorable angles of attack at any desired operating condition. The first chart (fig. 12) shows the effectiveness of this method for increasing the flow range at a fixed compressor speed. The measured pressure ratio and compressor efficiency are shown plotted against the air flow for three different stator blade settings. The entire air-flow range that can be obtained with any one blade setting is quite limited, but by using different stator blade settings the possible range of operation can be extended from this amount to the entire range covered by the chart, about a seven-fold increase in flow range, and high efficiency can be maintained over the extended range.

The same method can also be used to increase the efficiency over a range of compressor speeds as is shown by the second chart (fig. 13). The peak efficiency is plotted against the compressor Mach number, which is a measure of the compressor speed for the same three blade settings as those in the previous chart. The lower curve shows the efficiency with the blades set for design operating conditions and the two other curves the efficiency for the two stator blade resettings. Both resettings were for 75 percent of design speed but for different air flows. An increase in efficiency as high as 7 percent is noted at the lower speeds. Thus it is evident that the use of adjustable stator blades, although somewhat complicated, is a very effective method of increasing the high efficiency range with respect to both air flow and compressor speed.

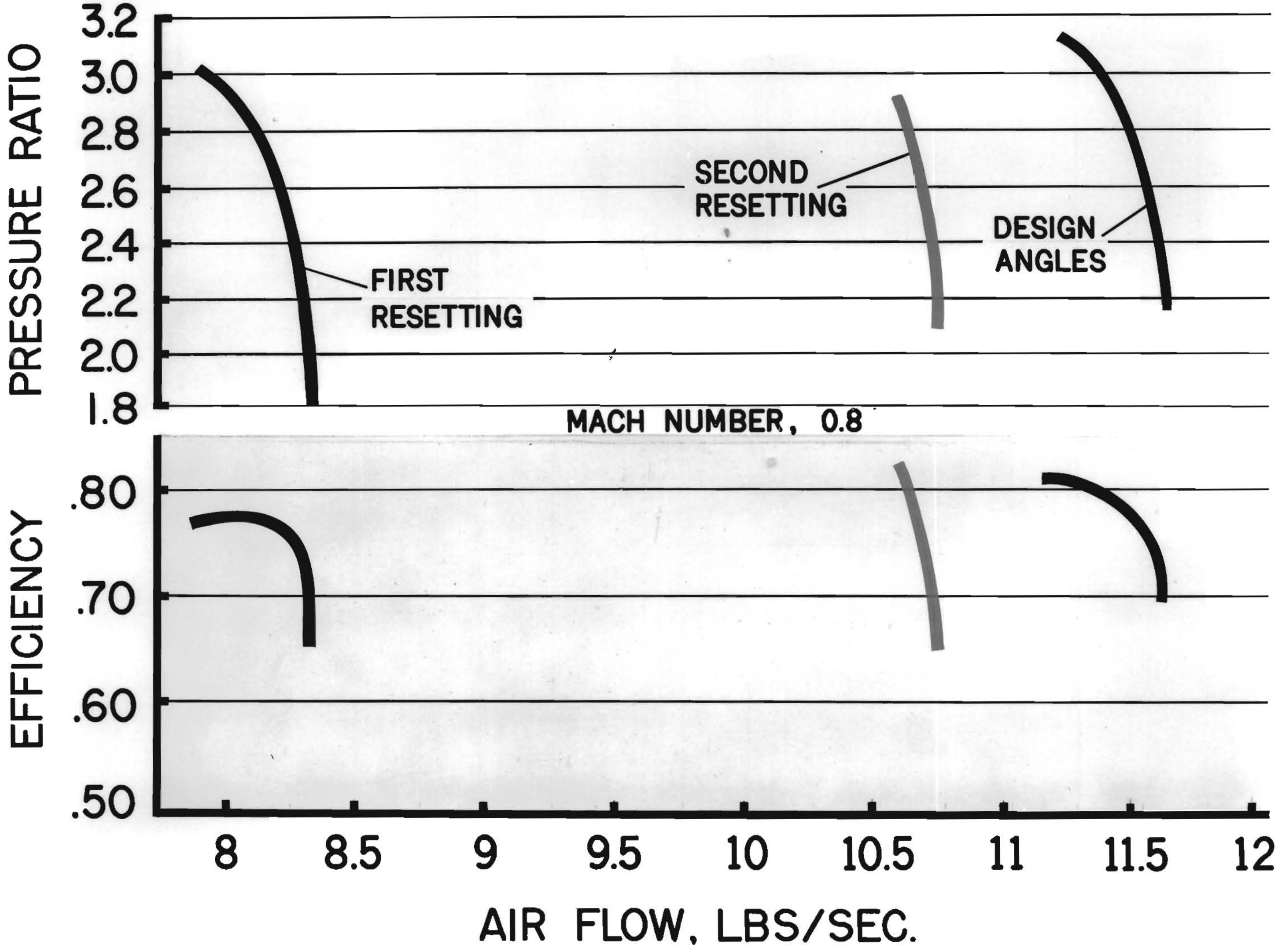
For aircraft applications it is not only important that the efficiency be as high as possible over the required range of operation, but it is also very important that the compressors be as light and compact as possible for any given job. Small diameter is dependent upon high air flow per unit frontal area and short length is dependent upon high pressure ratio per stage so as to reduce the number of stages required. One of the most important variables affecting both the pressure ratio per stage and the air-flow capacity is the Mach number relative to the blades. As a general rule, the higher the Mach number, the higher are the pressure ratio and air flow that can be obtained. In conventional designs, however, it has been found that large losses in efficiency result whenever the relative Mach number appreciably exceeds one anywhere over the blades. Consequently practically all of the compressors that we have today are designed for subsonic velocities, that is, Mach numbers less than one. Extensive analytical and experimental investigations are being conducted by the NACA to improve this type of compressor. In order to accomplish this aim, it is necessary to obtain detailed information on the flow processes taking place within the compressor. Most of this required information is being obtained by carefully planned measurements on compressor components representing different sections of a multistage compressor.

Mr. Burtt will now describe the research on a typical component of this type.

(Mr. Burtt's talk follows, after which the following introduction is made to Mr. Johnsen's talk on supersonic compressors.)

In addition to this research to improve the performance of subsonic axial-flow compressors, a new approach to compressor design has recently been made which appears very promising. The large gains in pressure ratio that are theoretically obtainable by increasing the Mach number has led the NACA to re-examine the necessity of restricting the design to subsonic velocities. The analyses indicated that the occurrence of supersonic velocities should not necessarily result in a loss in efficiency if the blades are properly designed for supersonic flow. As a result of these analyses, the NACA has initiated a research program on supersonic compressors, a few high lights of which will now be described by Mr. Johnsen.

FLOW RANGE INCREASE BY STATOR BLADE RESETTING

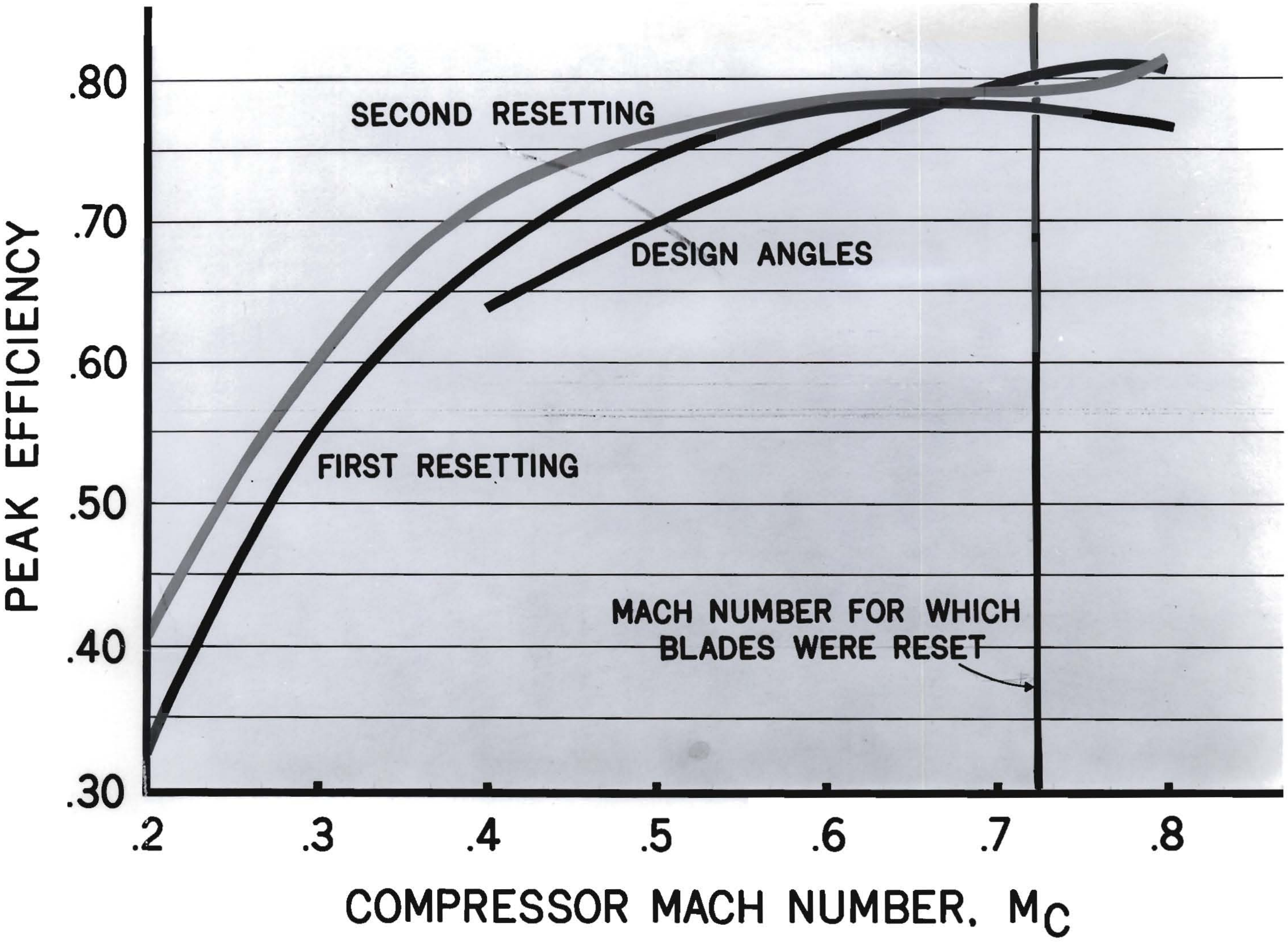


C-19853
10-24-47

Fig 12

NACA

PEAK PERFORMANCE WITH STATOR BLADE RESETTING



C- 19852
10- 24- 47

Fig 13

NACA

SINGLE-STAGE, VARIABLE-COMPONENT, AXIAL-FLOW COMPRESSOR RESEARCH

J. R. Burtt

In order to accurately predict the flow capacity and pressure ratio of a multistage axial-flow compressor and to design for peak efficiency, a thorough knowledge of the flow processes through the compressor is required. The exact flow pattern at the exit of each blade row must be known so that the following blade row can be designed for optimum performance. To obtain a more detailed picture of the flow characteristics in axial-flow compressors, investigations are being conducted on three single-stage units. These units are similar to the mock-up on display (See fig. 14.) here except for blade length. One is representative of a typical entrance stage, one a middle stage and the third an exit stage of a multistage compressor.

In compressors of this diameter, instrumentation is critical. The lack of space between blade rows necessitates the use of very small measuring devices. Note the extremely small instrument head, an enlarged view of which is shown nearby, (fig. 14). This is a remote controlled instrument mount used for survey work.

The data I will show you today is a sample taken from the entrance stage compressor. The rotor used is similar to the one on display on the back-drop and is the first of nine designs built to investigate radial distribution of axial and tangential velocities.

Chart No. 1 (fig. 15) is typical of curves presenting over-all data.

This curve shows efficiency and pressure ratio versus air weight flow. Note that at design speed these blades have a peak pressure ratio of 1.27 at an efficiency of about 90 percent. You can see that operation in the lower range of weight flow for a given speed is undesirable from a standpoint of efficiency. If higher weight flows are used, the efficiency remains fairly high but the pressure ratio is decreasing. The manufacturer will probably have to compromise between air flow and pressure, ratio depending on the purpose for which his engine is being designed.

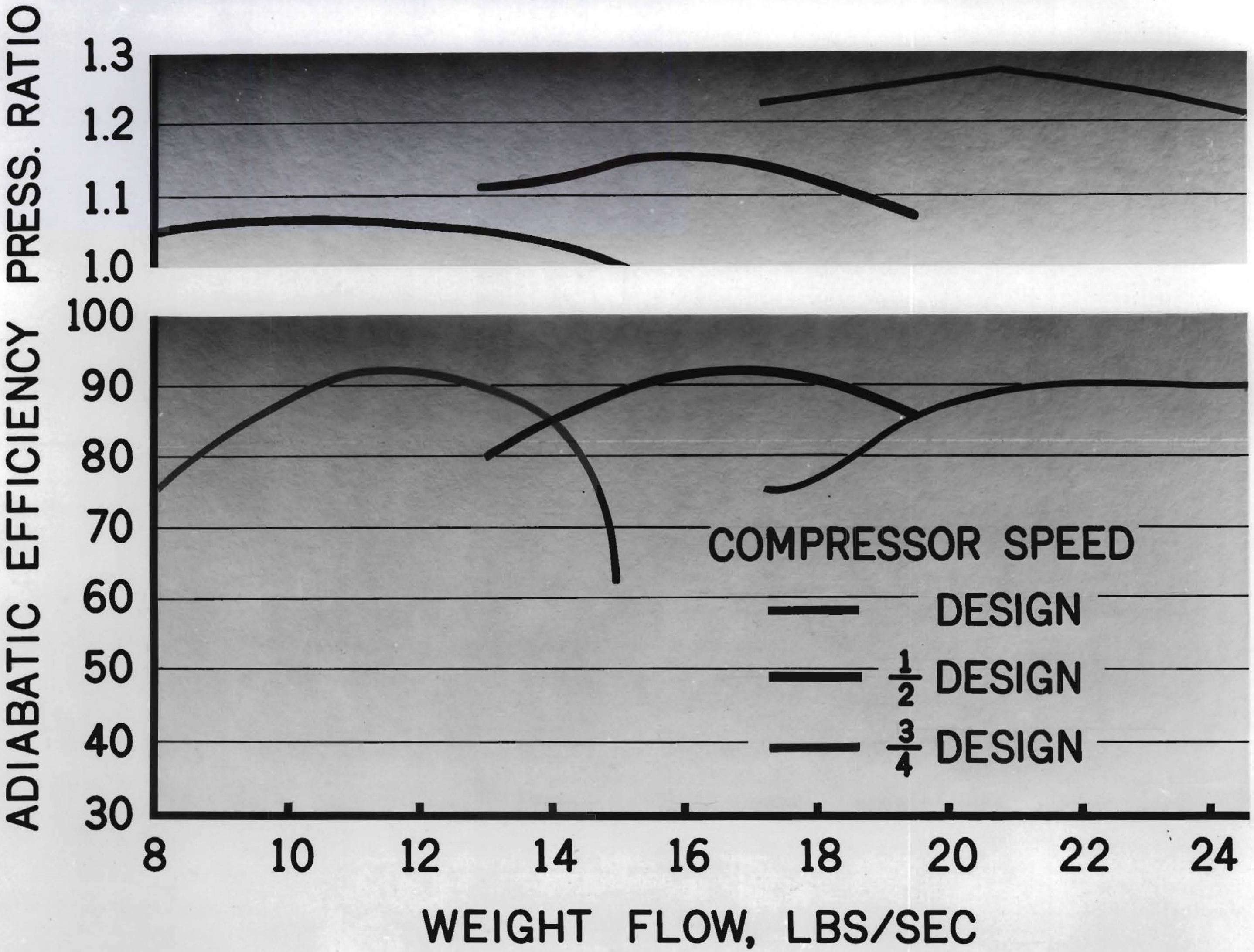
Since this particular compressor is being used for velocity distribution work, angles of attack and turning angles were obtained with interstage survey instruments. The next chart (fig. 16) shows turning angles versus angles of attack at 3/4 design speed for 4 radial positions across the passage. An inspection of the data at the tip shows that as the weight flow is decreased corresponding to an increase in angle of attack, a critical flow is reached below which the turning angle drops off. This shows us that the rotor is stalled at the tip and accounts for the rapid loss in efficiency at the low flows. With a stall condition at the tip, a further decrease in weight flow causes the stall to progress down the blade toward the hub. However, a study of the data shows that this stall near the tip leaves the flow conditions near the hub relatively unchanged. Obviously, if this were the initial stage of a multistage compressor, the flow would be disrupted through the entire unit. Therefore, extreme care should be taken to stay away from operation in this flow range.

From correlation of data of this type, multistage compressors can be designed in which each element will operate at its optimum performance and thus assure maximum possible performance for the compressor as a whole.


AXIAL FLOW COMPRESSOR RESEARCH
OVERALL PERFORMANCE CURVES-DESIGN I
PRESS. RATIO
COMPRESSOR SPEED
DESIGN
WEIGHT FLOW, LBS/SEC
YAW AND TOTAL
RESSURE PROBE
REMOTE CONTROLLED
INSTRUMENT MOUNT
STATIC PRESSURE
PROBE
THERMOCOUPLE RAKE
F&L
C-1975
10-13-

Fig 14

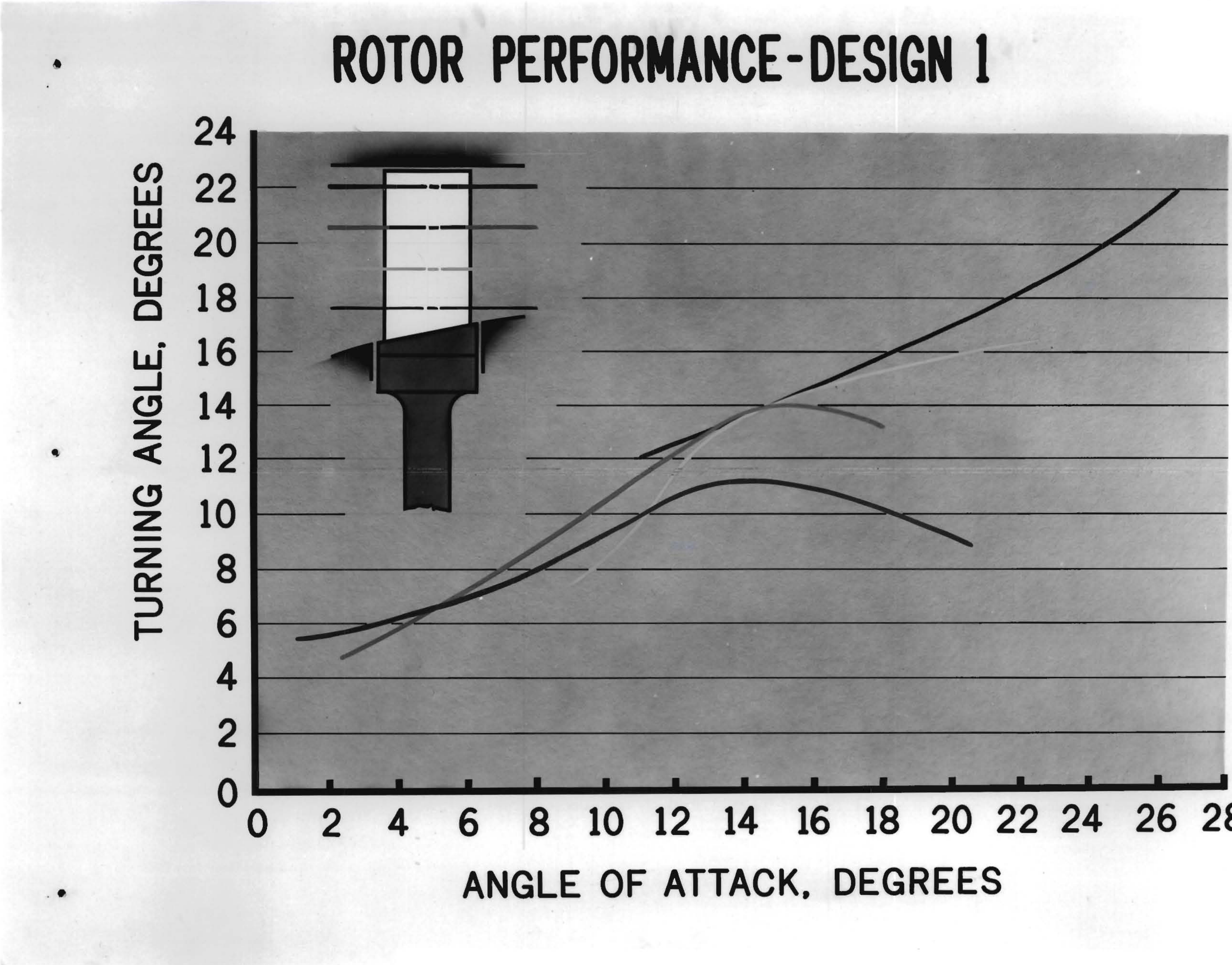
OVERALL PERFORMANCE CURVES-DESIGN 1



C- 19836
10- 24- 47

Fig 15

NACA



C-19859
10-24-47

Fig 16

NACA

NACA SUPERSONIC AXIAL-FLOW

COMPRESSOR RESEARCH

I. A. Johnsen

The supersonic type of compressor has the inherent advantage of a very high pressure ratio per stage and high mass flow. On the basis of results already obtained by the NACA, the supersonic compressor shows promise of being more compact by a factor of about four than a comparable subsonic machine performing the same function. These scale models show the relative sizes of a supersonic compressor and an equivalent multistage subsonic compressor. Note the short axial length of the rotating stage and the large saving in size and weight which can be made through the use of this type of compressor.

The Langley Field and Cleveland laboratories of the NACA are cooperating on an extensive program of research on the supersonic type of compressor. The first step in this program was accomplished at the Langley Field laboratory. As the result of supersonic diffuser and cascade studies made there, blading was developed which could be applied to a compressor design. The rotor was built and run with Freon -12 as the testing medium, chosen because the velocity of sound in Freon is approximately half that in air, thus permitting operation with relatively low speeds and centrifugal stresses.

The results of this initial investigation were very promising; so the second phase of the program is being carried out at this laboratory in which an equivalent compressor is studied in air. The objectives of this program are to: First, demonstrate that the idea of supersonic compression can be successfully applied to a compressor running in air, second, gain an understanding of supersonic compression so that by systematic research we may obtain compressors that are superior to this first design.

This model shows a scaled-up cross-section typical of one of the blades of the rotor used in this investigation. Note the thin blade, the sharp leading and trailing edges, and this bump, which I will discuss later. The first chart (fig. 17) is a corresponding sectional view showing the rotor blading.

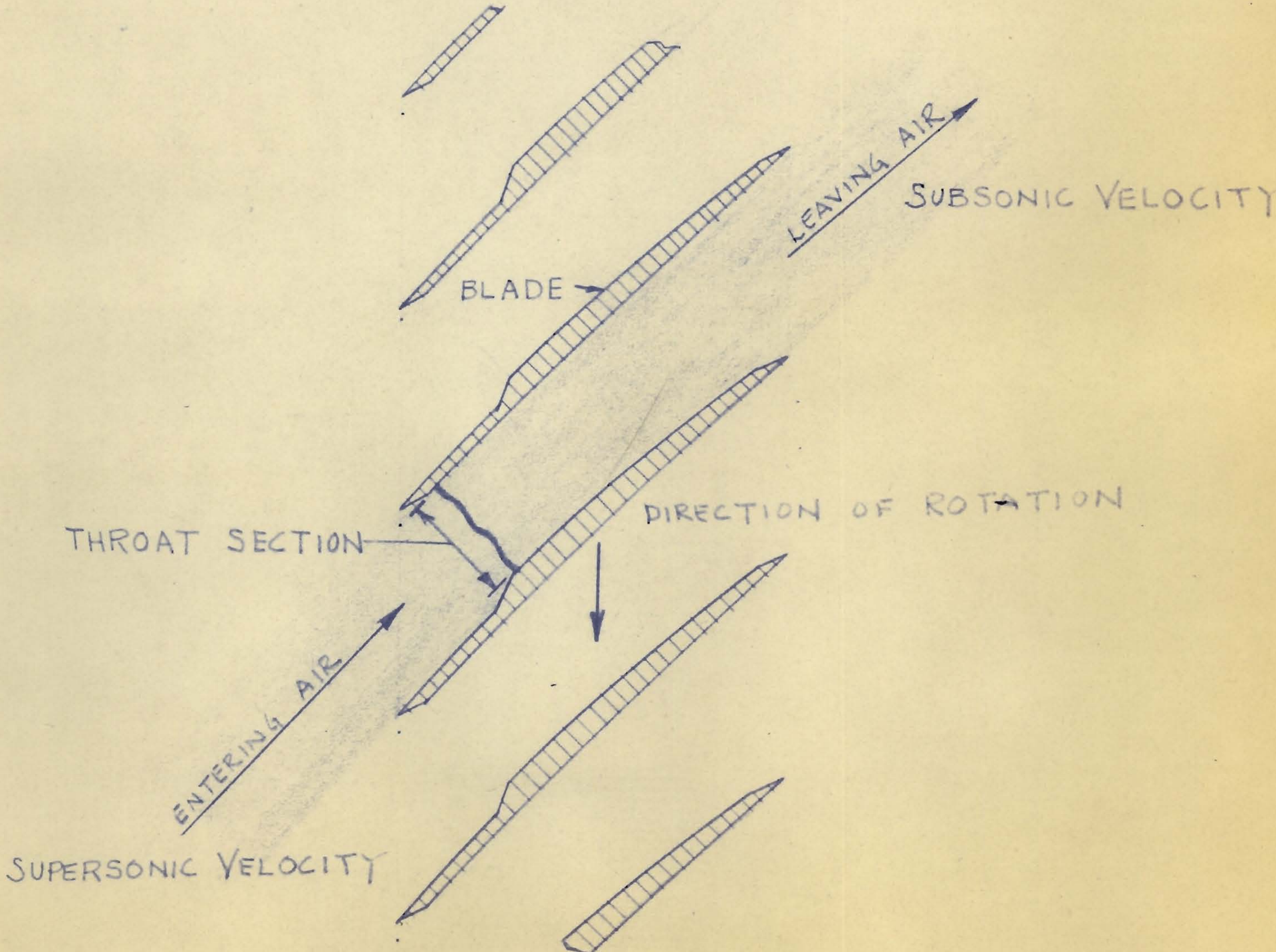
Let us consider the nature of flow through a passage formed by two adjacent blades. As you all know, the formation of shock waves has proved a serious limitation in increasing the speed of aircraft. In this compressor, however, these same shock waves are controlled and made to do useful work in compressing the air.

Air enters the passage with a relative velocity that is greater than the speed of sound. At the design condition, shock occurs in this throat or minimum area section which is formed by this bump on the blade. You will notice the shock wave is completely contained within the passage. This confining of the shock is essential to the success of this type of compressor, since large losses occur when shock waves extend ahead of the rotor. In passing through the shock, the pressure of the air increases and the velocity becomes subsonic. A further increase in the pressure of the air takes place in the subsonic diffusion after the shock, and the air leaves the rotor with a velocity that is less than the speed of sound.

The next chart (fig. 18) shows the supersonic compressor which has been investigated in air. The rotor is machined from a stell forging, is 24 inches in diameter, has 62 blades, three inches high, and is 1½ inches in depth. Several mechanical problems arose in the design of this compressor. In order to obtain a high entrance Mach number it is necessary to operate at a high rotational speed. In this particular supersonic compressor design, it is also necessary to use very thin blades, sharpened to razor-blade thicknesses at the leading and trailing edges in order to allow the entry of shock. Running these thin blades at high speeds without tip support is impractical from a vibrational point of view, so shrouding of the rotor is necessary. Now this shroud, which is needed for blade support, is objectionable from a stress consideration. For example, the centrifugal stress in a free shroud rotating at the design speed is of the order of 240,000 pounds per square inch. Therefore this design was evolved in which the blades reduce the centrifugal stresses of the shroud and the shroud supports the blades in vibration.

This compressor has been investigated in this variable-component unit located to your left. The next chart (fig. 19) shows pressure ratio and efficiency vs weight flow at the design tip speed of 1600 feet per second. In a single stage, a maximum pressure ratio of 1.93 was obtained with an adiabatic efficiency of 80 percent at a weight flow of 59 pounds per second. This pressure ratio is roughly equivalent to the pressure ratio obtained in the first four stages of a conventional subsonic compressor.

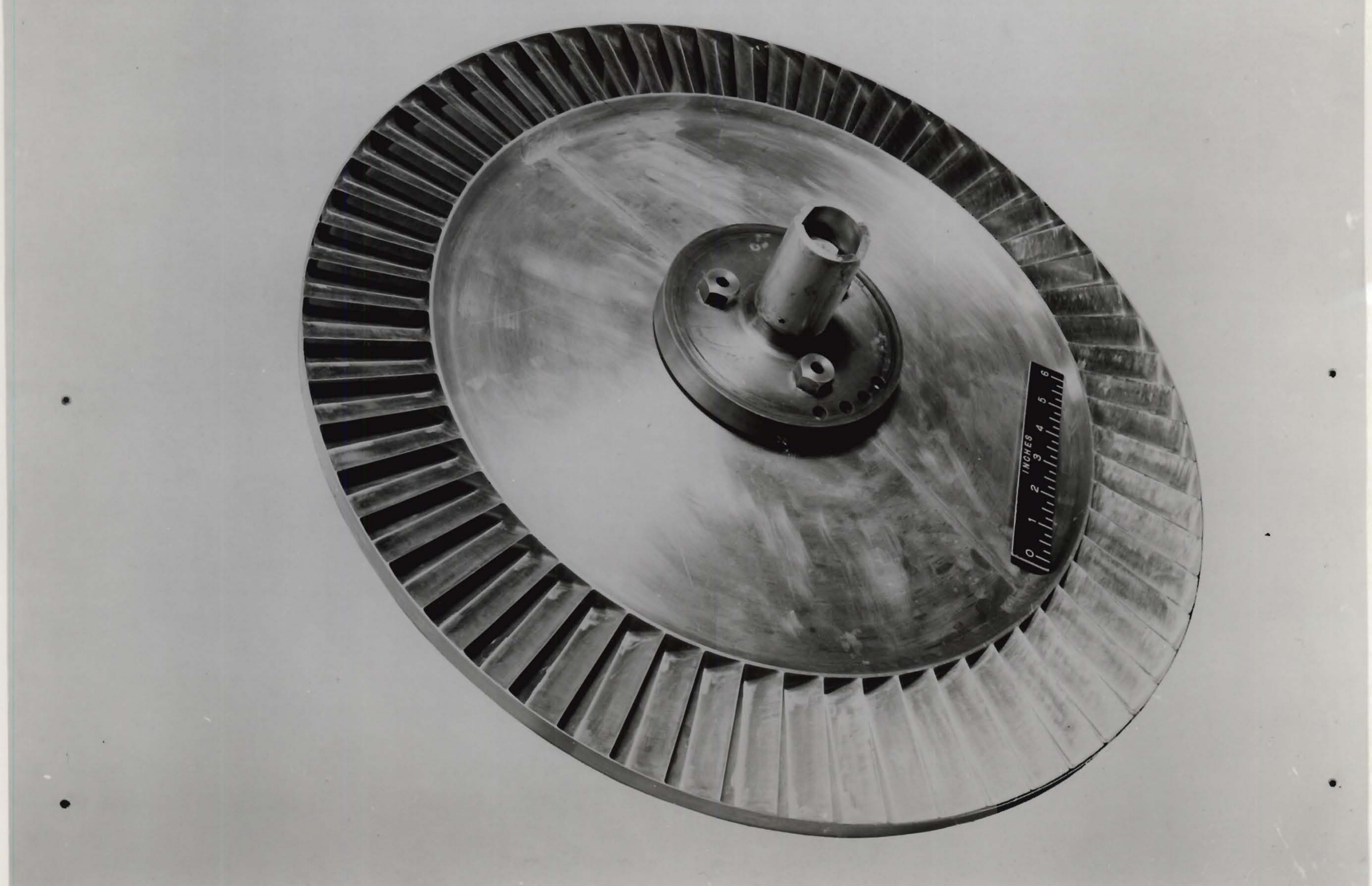
These data have been obtained on a preliminary design of supersonic compressor. More recent analyses made by the NACA indicate that this type of compressor may be made more practical by using thicker blades and eliminating the shroud; at the same time increasing the pressure ratio and efficiency. These excellent characteristics indicate that the supersonic type of compressor has considerable promise for aircraft power plant application.



SUPERSONIC COMPRESSOR BLADING

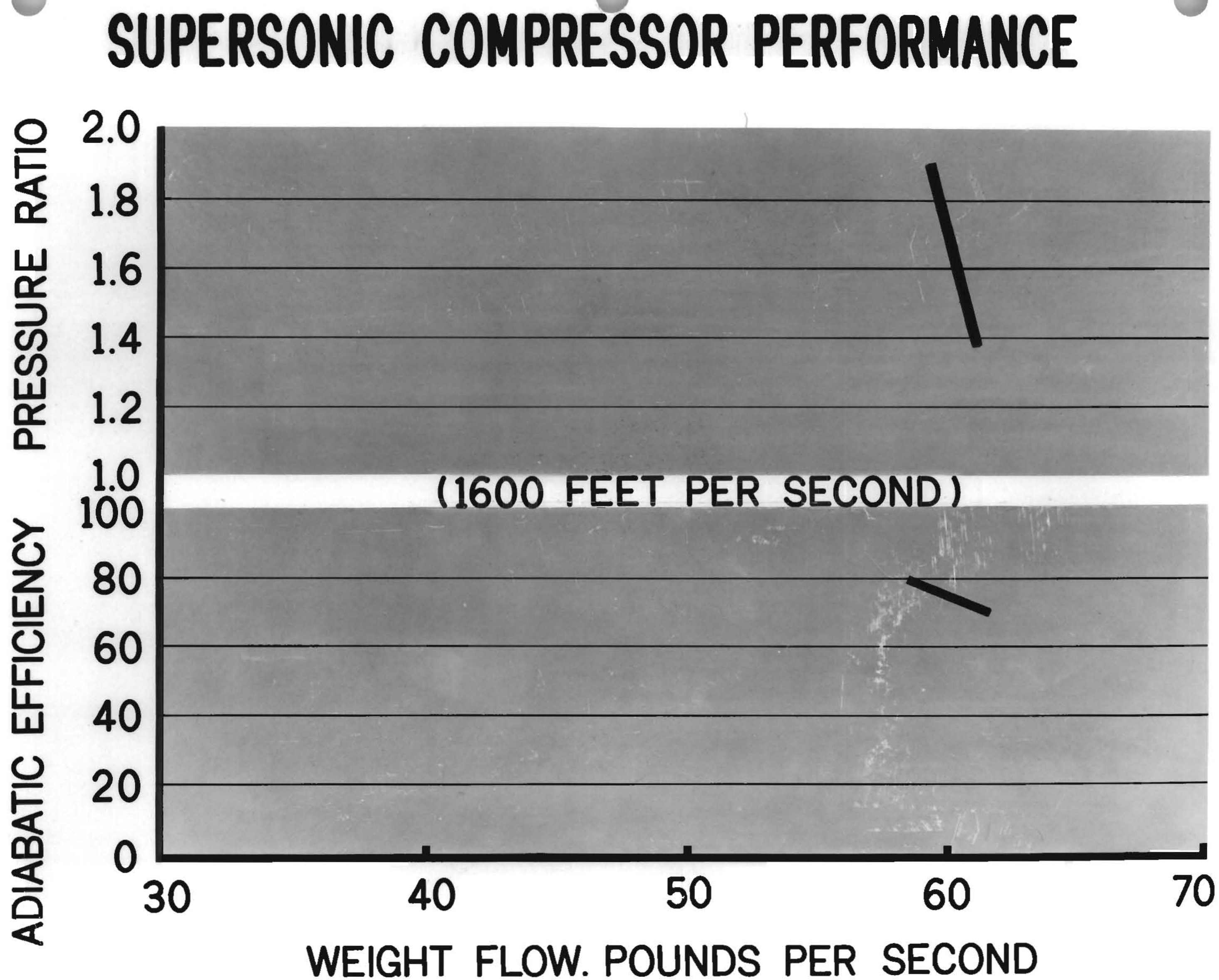
FIG:-17

SUPERSONIC ROTOR



C- 19848
10- 24- 47

Fig 18



C- 19835
10- 24- 47

Fig 19

NACA