(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) Publication number: **0 367 079 B1**

(12) EUROPEAN PATENT SPECIFICATION

(45) Date of publication of patent specification: **20.10.93**

(51) Int. Cl.5: **F04D 29/54**

(21) Application number: **89119779.0**

(22) Date of filing: **24.10.89**

(54) Fan shroud for radiator.

(30) Priority: **31.10.88 JP 273086/88**

(43) Date of publication of application:
09.05.90 Bulletin 90/19

(45) Publication of the grant of the patent:
20.10.93 Bulletin 93/42

(84) Designated Contracting States:
DE FR GB

(56) References cited:
DE-A- 2 345 506
DE-B- 1 161 481

PATENT ABSTRACTS OF JAPAN vol. 8, no. 47 (M-280)(1484) 02 March 1984,& JP-A-58 200097 (MITSUBISHI) 21 November 1983,

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EP 0 367 079 B1

Description

The invention relates to a fan shroud according to the general clause of the claim. This type of fan shroud is generally mounted on an automotive radiator, surrounding an axial fan of a liquid cooled internal combustion engine in order to provide a safe guard against a manual touch to the fan.

The main purpose of the fan shroud is to surround the axial fan thereby to prevent peripheral air from entering into a cooled area. It also serves to increase a volume of air flow passing through the radiator thereby to promote cooling efficiency of the radiator.

Known fan configurations often bring about the problem of producing noise caused by an acceleration of air flow towards a position between a tip edge of a high speed rotating fan blade and a stationary shroud. From JP-A-58 200097 there are known wing plates disposed such that they face the top of a rotating fan on its upstream side. Their main purpose is to reduce the relative speed of fluid flow by changing the directions of air streams towards the top edges of the fan blades. This means that they serve as a deflector means for providing pre-vortex flows in advance. In short, they serve as a noise reduction means. They cannot increase volumentric flow efficiency of the fan.

Furthermore, air stream lines peel off the fan blade near an axial rear end of the fan blade so that a turbulent flow or vortex having attendency to emit high frequency noise is produced. It is believed that these vortex and turbulent flow obstruct the air flow through the radiator thereby causing a considerable decrease of volume of the air flow.

It is therefore an object of the invention to provide a high performance fan shroud which does not only achieve a substantial decrease of noise emitted to the surroundings, but even brings about a significant increase of a volume of air flow through the radiator.

According to the present invention this object is achieved by providing a fan shroud having the features of the claim.

The arrangement of the mouth portion and fin plates according to the invention has the result that irregular patterns of turbulent flow or spiral flow running against the inside surface of the shroud are controlled and transformed into regular patterns flying toward axial directions, so that air flow resistance is considerably reduced and volumetric flow efficiency of the fan is improved. According to an experiment, a remarkable improvement of 20 percent increase of air flow and 30 percent increase of volumetric flow efficiency were demonstrated. With respect to the fan noise, although it showed a slight rise of sound level, in view of the fact that the volume of air flow also increases, it has been found that a fan noise per air flow is diminished by about 10 percent. As a further advantage of the present invention, the fin plates provide an additional rigidity to the shroud, so that not only durability of the shroud is improved, but also a resonant noise is eliminated.

An embodiment of the invention will now be described by way of example with reference to the drawings, in which:

Fig. 1 is a schematic vertical sectional view illustrating an arrangement of a fan shroud according to the present invention.

Fig. 2 is a side elevational view of Fig. 1.

Fig. 3 is a cross-sectional view taken on line III - III of Fig. 1.

Fig. 4 is a graph showing curves of experimental values while changing an angular orientation of the fin plates.

Fig. 5 is a graph showing curves of experimental values while changing a vertical length of the fin plates.

Fig. 6 is a graph showing curves of experimental values while changing number of the fin plates.

Fig. 7 is a graph showing curves of experimental values while changing a distance between the fin plates and the radiator core.

Fig. 8 is a graph showing curves of experimental values while changing a midway folding angle of the fin plates.

Fig. 9 is a graph showing curves of experimental values while changing a distance between the rectangular fin plates and the fan blades.

Fig. 10 is a graph showing curves of experimental values while changing a distance between the triangular fin plates and the fan blades.

Referring to Figs. 1 to 3, there is shown a preferable embodiment of a fan shroud according to the invention. A radiator 10 comprises a core 12, a fan shroud 14, an upper tank located above the core and a lower tank located below the core. These tanks are not shown for illustrative simplification. A fan 30 comprises a plurality of fan blades 32 and a fan hub 34 for supporting the blades. The fan hub 34 is usually mounted on a fan drive shaft which is driven by an internal combustion engine. The fan shroud 14 comprises a rectangular base frame and a mouth portion 16, which extends from the base frame coaxially with the central axis of the fan 30 and terminates at a position passing over a portion of the outer circumference of the fan 30. The overlapped length is represented by "U". In these figures, the mouth portion 16 is formed in a parallel tubular form. However, it is also possible to make the mouth portion 16 in a tapered form which is rather common to a fan shroud. While the fan 30 is driven by the engine and rotated toward the direction R, cooling air is compulsorily admitted through the radiator core 12.

On the inside surface of the mouth portion 16 of the shroud 14, eight fin plates 18 are circumferentially disposed at regular intervals. These fin plates 18 project radially inwardly into the space between the fan 30 and the radiator core 12. As shown in Fig. 1, each of the fin plates 18 is formed in a triangular shape in such a manner that its vertical length decreases along the direction from the core 12 to the fan 30. Further, as shown in Fig. 2, each of the fin plates 18 is arranged in an imaginary plane which contains the central axis of the fan 30. These triangular fin plates 18 have advantages that axial overall length of the radiator 10 can be minimized. The fin plates 18 can be made without difficulty. When the shroud is made by a metallic plate, the fin plates can be made by a metallic plate and then welded to the shroud. When the shroud is made from plastic materials, the fin plates can be formed integral with the shroud through an injection moulding process.

When cooling air passes through the axial fan 30, the air is usually directed toward the direction of the arrow F. This air pressure force F can be divided into an axial component Fx and a radial component Fy. Apparently, as the axial component Fx goes up and as the radial component Fy goes down, loss of energy is saved. The radial air flow defined by the component Fy runs against the inside surface of the shroud 14, producing turbulent flow which shows a resistance to forthcoming air. If the fin plates 18 is disposed, random air flow is forced to the axial direction, whereby occurrence of a turbulence is considerably reduced. In particular, since the fin plates 18 project radially inwardly adjacent the fan blades 32, near the most efficient points of the fan blades 32 air flow is controlled and regulated, whereby the volumetric flow efficiency of the fan 30 is significantly improved. It is the basic principle of the present invention.

As an actual example of the arrangement, following dimensions are available: The outside diameter D of the fan 30 is 440mm, the width W of the fan 30 is 82mm, number of fan blades 32 is nine, the inside diameter A of the mouth portion 16 is 500mm, the axial length T of the mouth portion 16 is 120mm, the overlapped length U is 48mm, the base length of the triangular fin plate 18 is 80mm, and the vertical length of the fin plate 18 is 80mm. When the eight fin plates 18 are disposed at regular intervals as shown in Fig. 2, the most satisfactory results are obtained.

The graphs in Figs. 4 to 10 show test results achieved in accordance with the Japanese Industrial Standard B 8330 "Testing method for Turbo-Fans and Blowers". In these tests, fan speed N was constantly kept to 2200 rpm.

In Fig. 4, four curves of experimental values are shown. The first curve 1 represents a case of null fin. Other curves 2 to 4 represent respective cases while an angular orientation of the fin plate 18 is changed from zero oblique angle, that is parallel to the central axis of the fan, to 30 degrees oblique angle toward the same direction as the twist of the fan blade (see Fig. 3), and to 15 degrees oblique angle contrary to the direction of the twist of the fan blade. The experimental values are plotted in relation to the volume of air flow Q and static pressure P. The curve S represents a peculiar permeability of the radiator 10, that is a basic resistance to air flow passing through the radiator. In the test, permeability coefficient k was 0.037. Actual volume of air flow through the radiator is read from the intersecting points of the air flow curve and the permeability curve. Thus, from the curve 1 actual volume is read as 52.2 cubic meter per minute, and from the curve 2 actual volume is read as 63.2 cubic meter per minute. Therefore, according to the arrangement of the fin plates 18 at zero oblique angle, about 20 percent increase of air flow is attained. In other two cases shown in curves 3 and 4, each volume of air flow becomes less than in the curve 2. As a result, it is recognized that an oblique arrangement of the fin plate 18 cannot exhibit an advantage.

At the same time, a test of absorbing horsepower of the fan was achieved in order to measure volumetric flow efficiency thereof. As a result, it has been proved that about 30 percent increase of efficiency is attained.

In Fig. 5, five curves are shown. The first curve 1 represents a case of null fin. Other curves 2 to 5 represent respective cases while a vertical length of the triangular fin plate 18 is changed by cutting off the top portion thereof in horizontal stepwise. From these curves following tendency is recognized: As the cut-off area increases, an advantage of the fin plate is diminished.

In Fig. 6, six curves are shown. The first curve 1 represents a case of null fin. Other curves 2 to 6 represent respective cases while the number of frusto-triangular or trapezoidal fin plate is changed from four to sixteen. As a result, the curve 4 in case of eight fin plates is proved to be most advantageous.

In Fig. 7, four curves are shown. The first curve 1 represents a case of null fin. Other curves 2 to 4 represent respective cases while a distance L between the radiator core and the fin plate is changed in stepwise by cutting off the side portion of the fin plate. It is recognized that air flow and static pressure are not affected by the change of the distance L.

In Fig. 8, four curves are shown. The first curve 1 represents a case of null fin. Other curves 2 to 4 represent respective cases while a midway folding angle of the fin plate is changed from zero, to 30 degrees in clockwise direction, and to 30 degrees in counterclockwise direction. In both cases in which the fin plate is folded in midway, an advantage of the fin plate is diminished.

In Fig. 9, five curves are shown. The first curve 1 represents a case of null fin. Other curves 2 to 4 represent respective cases while a distance between the rectangular fin plate and the fan blade is changed by cutting off the side portion of the fin plate. It is recognized that an advantage of the fin plate is diminished as the area of the cut-off increases.

In Fig. 10, six curves are shown. The first curve 1 represents a case of null fin. Other curves 2 to 6 represent respective cases while a horizontal distance between the trapezoidal fin plate and the fan blade is changed. It is recognized that an advantage of the fin plate is diminished as the distance increases.

In connection with a fan noise, an additional experiment was achieved under the conditions that the fan speed is 1500 rpm, and the noise is measured at a high position of vertical distance one meter from the central axis of the fan. As a result, in a case of null fin 84.2 decibel was measured by scale A, and in a case of eight triangular fin plates each having a base length 110mm and a vertical length 60mm, 86.9 decibel was measured by scale A. Although the overall noise shows an increase of 3 percent, in view of the fact that the volume of air flow also increases, it has been found that a fan noise per volume of air flow is diminished from 1.6 to 1.4, thereby effecting a decrease of 12.5 percent.

Improvements and modifications may be made to the present invention without departing from the scope of the appended claim.

Claims

1. A fan shroud for guiding air flow around an axial fan (30) of a liquid cooled internal combustion engine having a radiator (10) through which air is drawn by the fan (30), and guide vanes which are disposed in a manner of facing the top of the fan (30) on its upstream side, **characterized** in that said shroud includes a mouth portion (16) which extends coaxially with the central axis of the fan (30) in a parallel or tapered form and terminates at a position passing over at least a portion of the fan (30), and that the guide vanes are a plurality of flat fin plates (18) circumferentially disposed at regular intervals on the inside surface of said mouth portion (16) and projecting radially inwardly into the space between the fan (30) and a core of the radiator (10), whereby each of said fin plates (18) is formed in a triangular or trapezoidal shape such that the vertical length of the shape decreases along the direction from the core of the radiator (10) to the fan (30), and whereby each of said fin plates (18) is arranged in an imaginary plane containing the central axis of the fan (30).

Patentansprüche

1. Lüfterzarge zum Leiten eines Luftstroms um einen axialen Lüfter (30) eines flüssigkeitsgekühlten internen Verbrennungsmotors mit einem Kühler (10), durch den die Luft mittels des Lüfters (30) gezogen wird, und mit Leitflügeln, die derart angeordnet sind, daß sie der Oberseite des Lüfters (30) auf seiner stromaufwärts gerichteten Seite gegenüberstehen,
   dadurch gekennzeichnet,
   daß die Zarge einen Mündungsabschnitt (16) aufweist, der sich koaxial zur Mittelachse des Lüfters (30) in einer parallelen oder konischen Form erstreckt und an einer Stelle endet, an der er sich wenigstens über einen Abschnitt des Lüfters (30) erstreckt, und daß die Leitflügel aus einer Vielzahl von flachen Leitblechen (18) bestehen, die in Umfangsrichtung in regelmäßigem Abstand auf der Innenfläche des Mündungsabschnitts (16) angeordnet sind und radial nach innen in den Zwischenraum zwischen dem Lüfter (30) und einem Kern des Kühlers (10) ragen, wobei jedes Leitblech (18) derart dreieckig oder trapezförmig geformt ist, daß seine vertikale Länge entlang der Richtung vom Kern des Radiators (10) zum Lüfter (30) abnimmt, und wobei jedes Leitblech (18) in einer imaginären Ebene angeordnet ist, welche die Mittelachse des Lüfters (30) enthält.

Revendications

1. Capot de ventilateur pour guider un flux d'air autour d'un ventilateur axial (30) d'un moteur à combustion interne refroidi par un liquide, comportant un radiateur (10) à travers lequel l'air est aspiré par le ventilateur (30), et des aubes de guidage qui sont disposées de manière à être tournées vers la partie supérieure du ventilateur (30), sur son côté amont, caractérisé en ce que le capot comporte une bouche (16) qui s'étend coaxialement avec l'axe central du ventilateur (30), sous une forme parallèle ou convergente et qui se termine dans une position passant par-dessus au moins une partie du ventilateur (30) et en ce que les aubes de guidage sont constituées par une pluralité d'ailettes minces et planes (18),

disposées circonférentiellement, à des intervalles réguliers, sur la surface interne de la bouche (16) et faisant saillir radialement vers l'intérieur dans l'espace compris entre le ventilateur (30) et un coeur du radiateur (10), chacune des ailettes minces (18) ayant une forme triangulaire ou trapézoïdale telle que la longueur verticale de la forme aille en diminuant dans la direction allant du coeur du radiateur (10) vers le ventilateur (30), chacune des ailettes minces (18) étant disposée dans un plan imaginaire contenant l'axe central du ventilateur (30).

Fig. 1

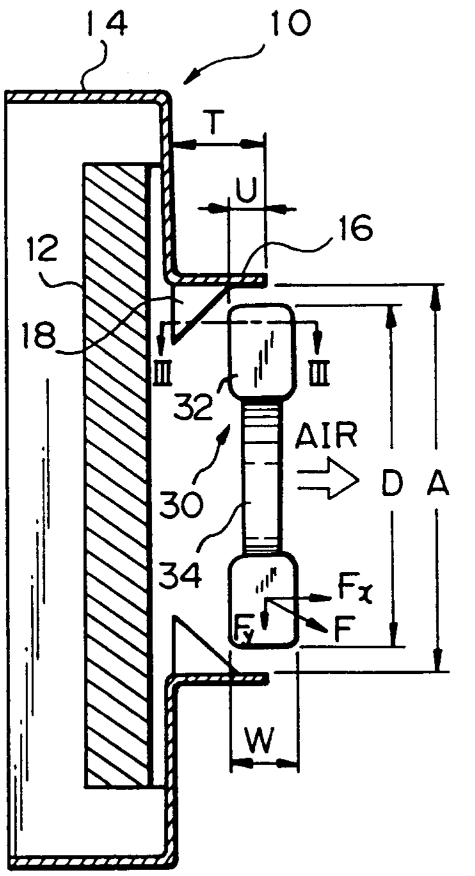


Fig. 2

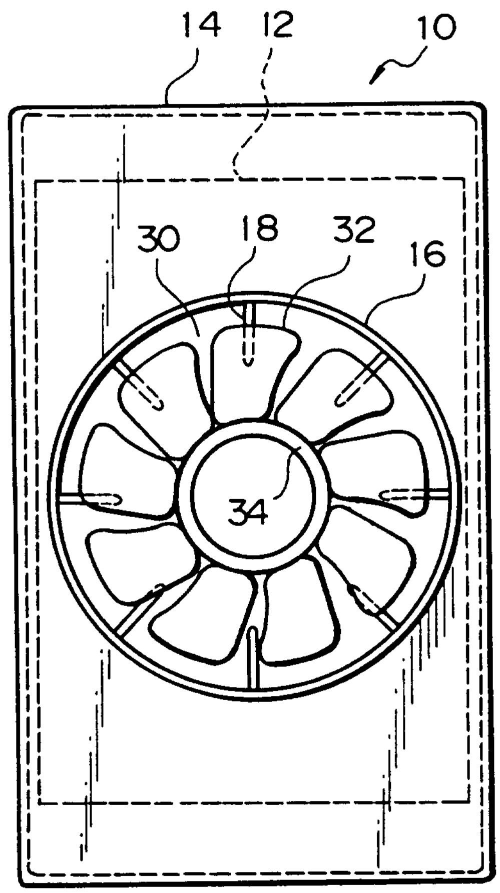


Fig. 3

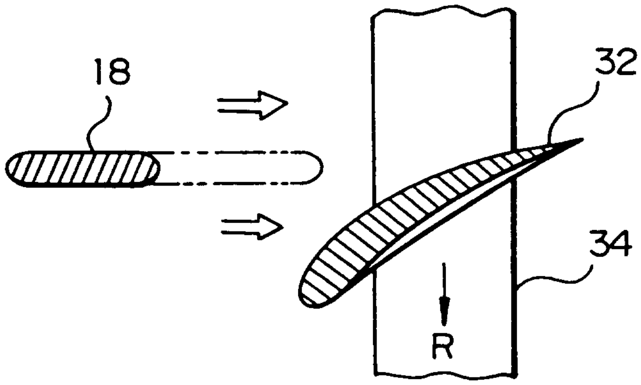
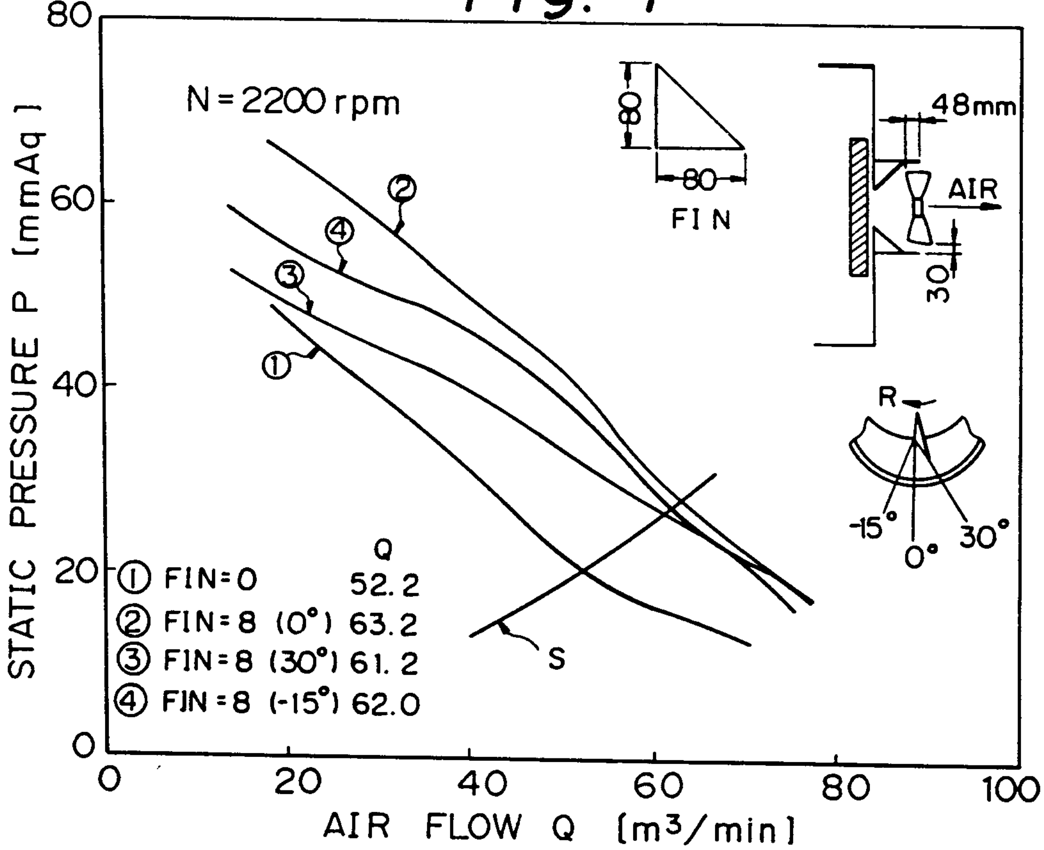
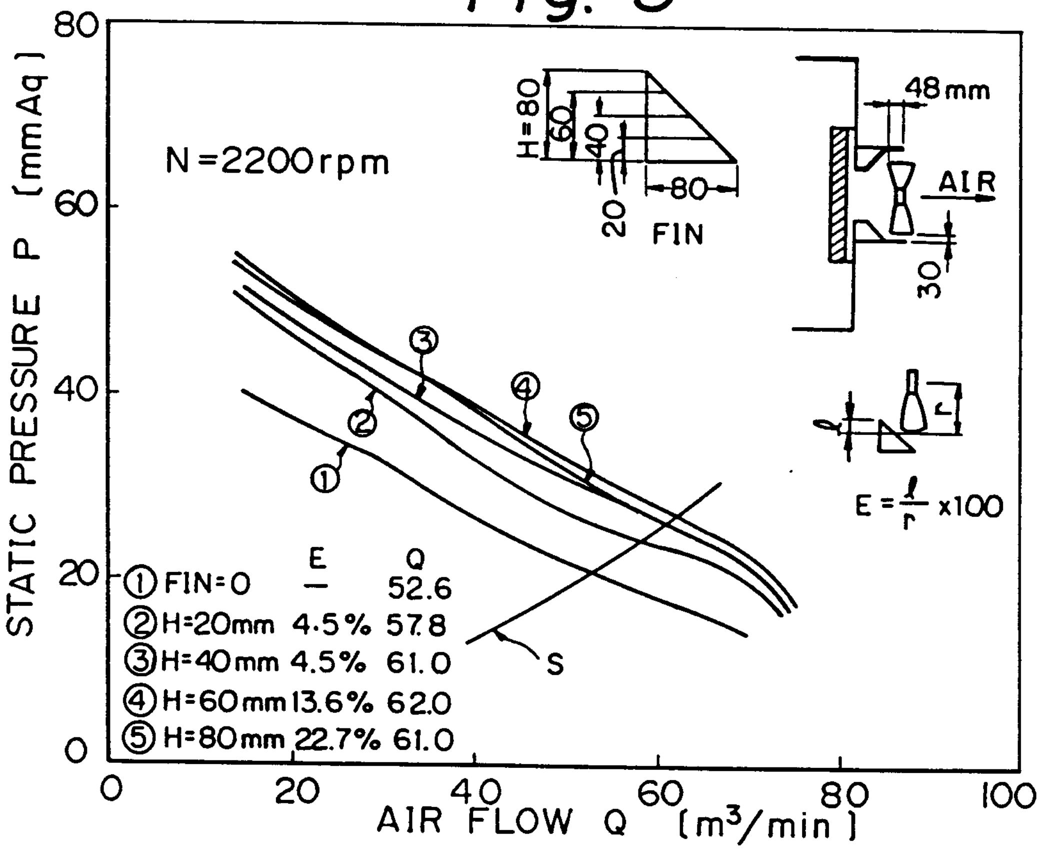


Fig. 4



N=2200 rpm
STATIC PRESSURE P [mmAq]
AIR FLOW Q [m3/min]
80
80
FIN
48mm
AIR
30
R
-15°
0°
30°
Q
① FIN=0 52.2
② FIN=8 (0°) 63.2
③ FIN=8 (30°) 61.2
④ FIN=8 (-15°) 62.0
S

Fig. 5



N=2200rpm
STATIC PRESSURE P [mmAq]
AIR FLOW Q [m3/min]
H=80
60
40
20
80
FIN
48mm
AIR
30
E = l/r x100
E Q
① FIN=0 — 52.6
② H=20mm 4.5% 57.8
③ H=40mm 4.5% 61.0
④ H=60mm 13.6% 62.0
⑤ H=80mm 22.7% 61.0
S

Fig. 6

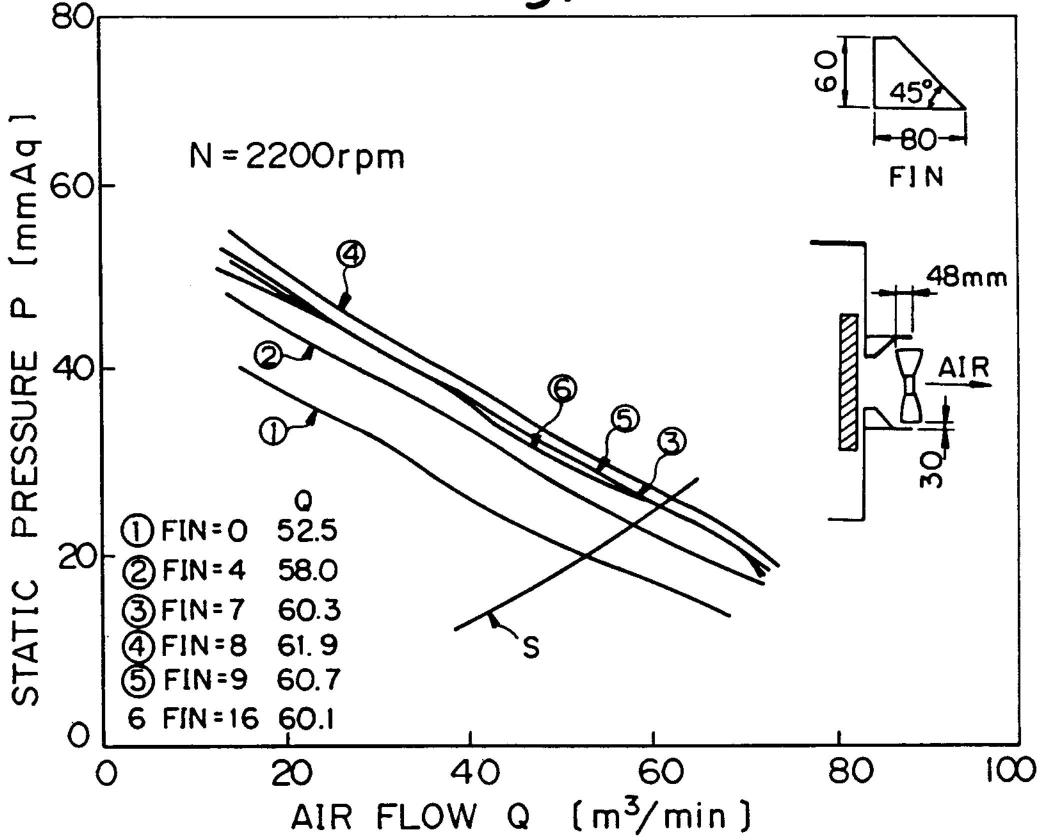

N = 2200rpm
60
45°
80
FIN
48mm
AIR
30
STATIC PRESSURE P [mmAq]
AIR FLOW Q [m3/min]
Q
① FIN=0 52.5
② FIN=4 58.0
③ FIN=7 60.3
④ FIN=8 61.9
⑤ FIN=9 60.7
6 FIN=16 60.1
S

Fig. 7

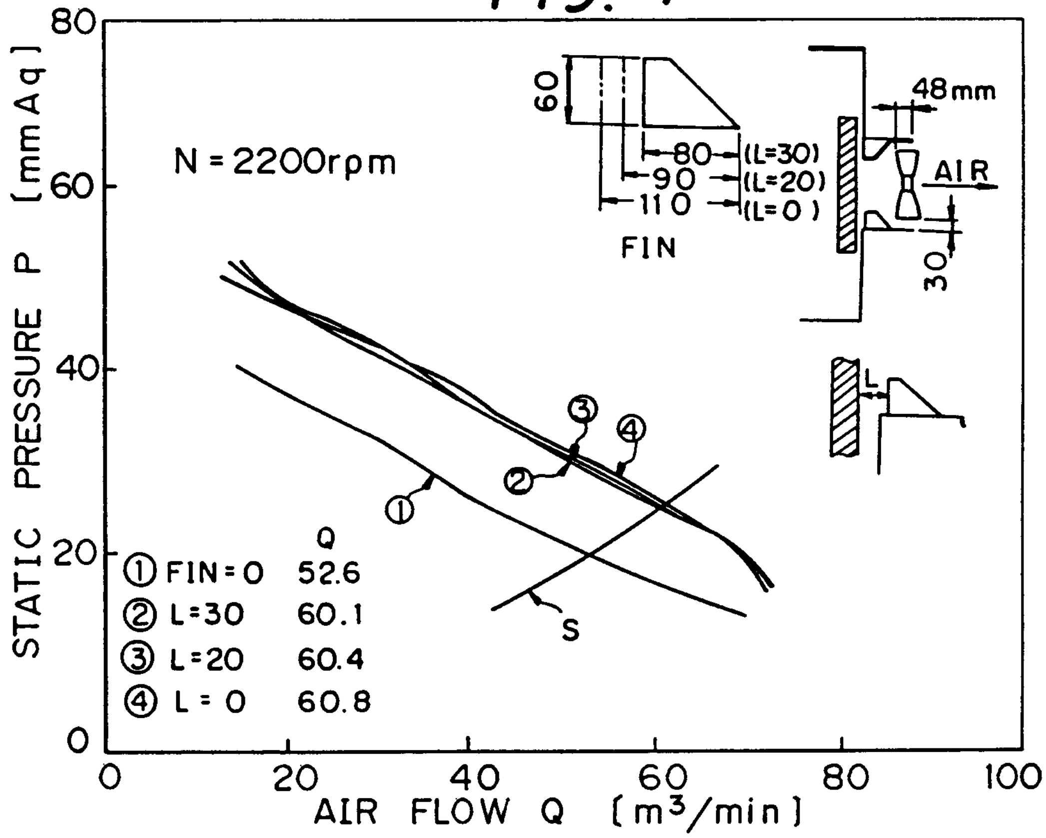

N = 2200rpm
60
80 (L=30)
90 (L=20)
110 (L=0)
FIN
48mm
AIR
30
L
STATIC PRESSURE P [mmAq]
AIR FLOW Q [m3/min]
Q
① FIN=0 52.6
② L=30 60.1
③ L=20 60.4
④ L = 0 60.8
S

Fig. 8

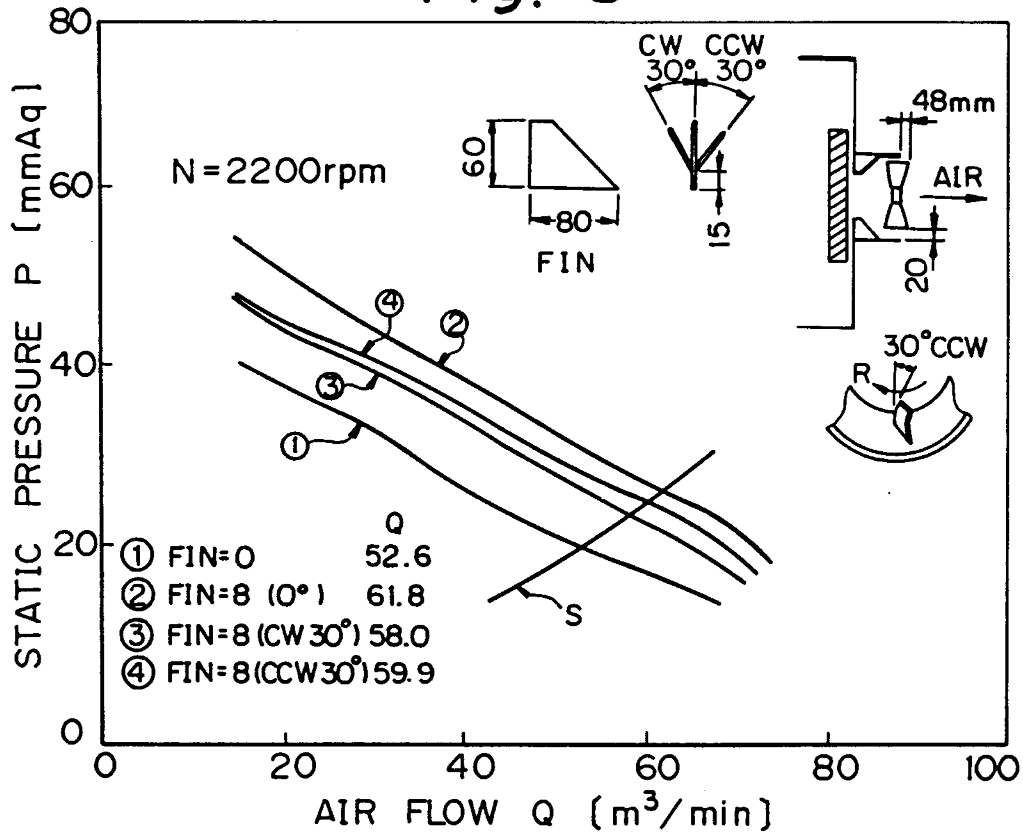

N=2200rpm
FIN
CW CCW
30° 30°
48mm
AIR
30°CCW
Q
① FIN=0 52.6
② FIN=8 (0°) 61.8
③ FIN=8 (CW30°) 58.0
④ FIN=8 (CCW30°) 59.9
S
STATIC PRESSURE P (mmAq)
AIR FLOW Q (m³/min)

Fig. 9

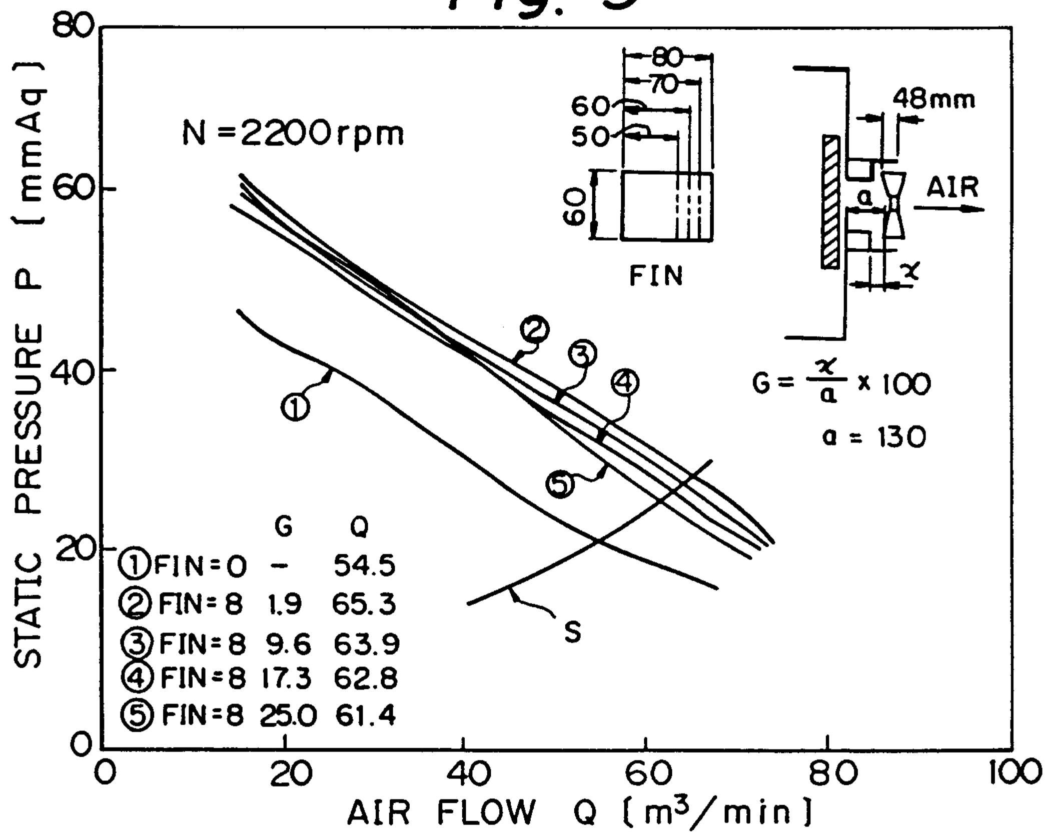
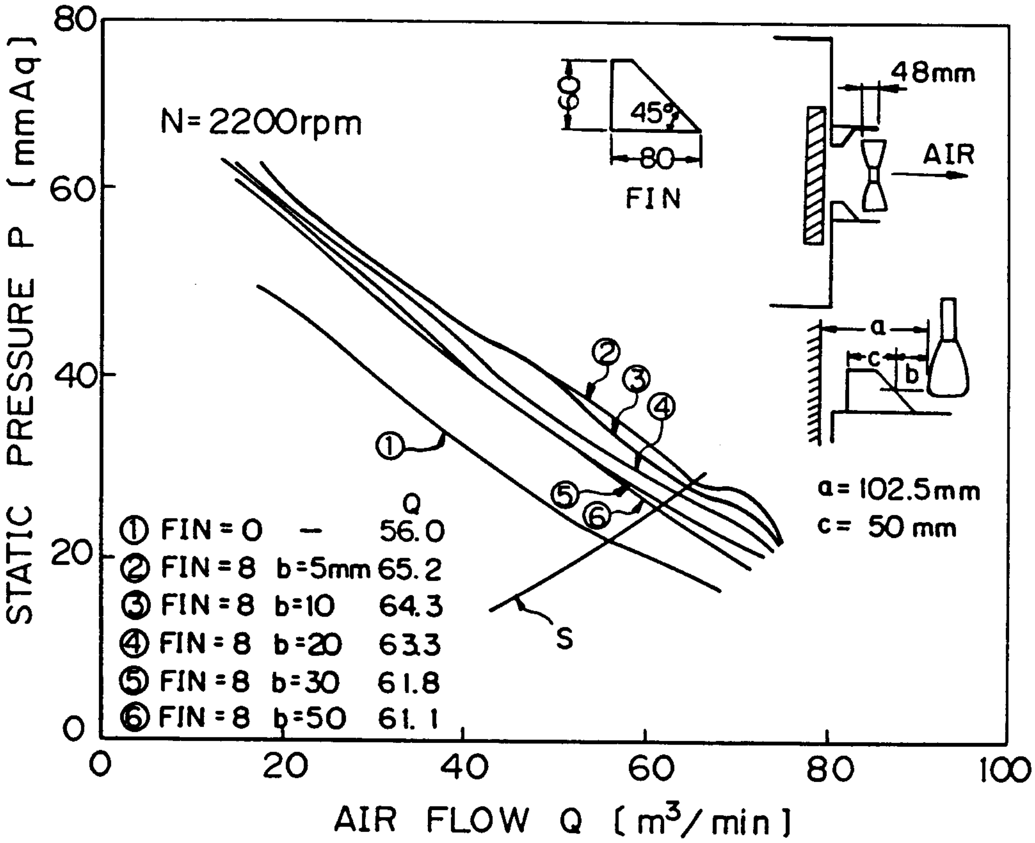

N=2200rpm
FIN
48mm
AIR
G = x/a × 100
a = 130
G Q
① FIN=0 - 54.5
② FIN=8 1.9 65.3
③ FIN=8 9.6 63.9
④ FIN=8 17.3 62.8
⑤ FIN=8 25.0 61.4
S
STATIC PRESSURE P (mmAq)
AIR FLOW Q (m³/min)

Fig. 10


N=2200rpm
STATIC PRESSURE P (mmAq)
AIR FLOW Q (m³/min)
0
20
40
60
80
100
FIN
45°
60
80
48mm
AIR
a
b
c
a= 102.5mm
c= 50 mm
S
Q
① FIN=0 — 56.0
② FIN=8 b=5mm 65.2
③ FIN=8 b=10 64.3
④ FIN=8 b=20 63.3
⑤ FIN=8 b=30 61.8
⑥ FIN=8 b=50 61.1