Model Tests on Factors Influencing Hot Air Recirculation in a Forced Draught Cooling System

W J Van der Elst

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CSIR

DIVISION OF AERONAUTICAL SYSTEMS TECHNOLOGY

MODEL TESTS ON FACTORS INFLUENCING HOT AIR RECIRCULATION IN A FORCED DRAUGHT COOLING SYSTEM

by

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The Steering Committee for this project consisted of the following persons:

Dr T C Erasmus	Water Research Commission (Chairman)
Mr A J Dickson	Eskom
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Dr J T D Fritz	CSIR
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EXECUTIVE SUMMARY

1 MOTIVATION

If the effectiveness of forced draught dry-cooling stacks can be improved it would boost the incentive to use such stacks in preference to conventional wet-cooling towers, and thus promote the endeavours to conserve water. This viewpoint motivated the conduction of water tunnel model studies aimed at reducing the detrimental effect of wind on hot efflux air recirculation in such dry-cooling stacks used in power stations.

2 OBJECTIVES

In order to study the factors which were thought to have a bearing on hot efflux air recirculation occurring in forced draught cooling systems, typical of a power station, the following objectives were set:

- To determine the extent of efflux air recirculation under various wind conditions, and how such recirculation is affected by the spacing between the cooling stack and the boiler house.

- To determine if there is any evidence of fan "starving" or diminishment of air inlet to the forced draught cooling fans when wind is allowed to flow freely through the stack space underneath the fans.

- To establish if, under the conditions of low wind flow, slight recirculation of efflux air occurs around the edges of certain portions of the cooling stacks (analytical two-dimensional analysis indicate that under such conditions slight recirculation does take place).

3 RESULTS AND CONCLUSIONS

It was found that if the spacing between the cooling stack and the boiler house is increased from zero to one third the cooling stack height, then recirculation of the simulated hot efflux air plume is significantly reduced. Increasing the spacing to one stack height, the only other gap studied, caused only marginal additional benefit. It is thus concluded that relatively small gaps are beneficial in reducing efflux air recirculation.

As regards the "starving" effect due to winds flowing underneath the cooling stack, quantitative measurement of the flow into the model fan inlets revealed only slight evidence of "starving". It is felt that such starving effects are negligible.

No evidence could be found of recirculation taking place around the edges of the stack at very low wind-speeds.

4 RECOMMENDATIONS

Since it was found that a gap between the power station boiler house and the cooling stack has a marked effect on efflux recirculation, it is recommended that this fact be borne in mind when designing cooling stack layout. It is furthermore recommended that more research be done on this gap effect. Virtually only two gap sizes have been considered and variation of cooling stack height has not been investigated at all. To promote optimum efficiency of dry-cooling stacks it is essential that the effect of these parameters be fully researched.

Under low wind speed conditions no evidence was found of efflux recirculation around the cooling stack edges. However, it must be pointed out that the water tunnel is not suitable for investigating recirculation at low wind speeds or under calm conditions. This is because the emitted plume of hot efflux air can strike the roof of the tunnel's working section and is then liable to give false recirculation conditions. It is recommended that further research be done by simulating calm wind conditions in a water tank to allow an extended vertical fluid height above the cooling stack.

In all model investigations thus far conducted, the cooling stack was situated downwind of the power station boiler house. It is strongly recommended that the situation where the boiler house is downwind of the cooling stack, thus allowing the wind to flow more readily in underneath the cooling fans, be carefully analysed for indications of efflux air recirculation (analytical two-dimensional studies indicate that recirculation under such wind conditions does occur).

CONTENTS PAGE

1 INTRODUCTION

This report deals with certain wind effects on recirculation of hot air in a forced draught cooling system for a typical power station.

The model used in a water tunnel, was a modified version of the one used for establishing recirculation effects in the Matimba Power Station (Ref.[1]), except that the model was somewhat simplified.

The purposes of the tests were:

- To determine the hot air recirculation varying the spacing between the forced draught cooling stack and the boiler/turbine houses, under various wind conditions.

- To determine if there is any evidence of "starving" the air inlet to forced draught cooling fans when wind is allowed to flow freely through the space underneath the fans.

- To establish if slight hot air recirculation takes place around the certain portions of the edges of the cooling stack even under conditions of no or slight wind. (Analytical determination of recirculation patterns (Ref. [2]) did indicate such a slight recirculation).

2 MODEL SIMILITUDE CONSIDERATIONS

For such a heat exchanger model to behave in the same way as a full-scale prototype, all force ratios in the model must equal the corresponding force ratios in the full-scale prototype. Such force ratios are analytically represented as follows:

Let V = a typical wind velocity.
V_e = velocity of efflux from forced draught system.
ρ = density of ambient air.
ρ' = density of hot air.
D = typical linear dimension (say the width of the bank of fans representing the forced draught heat exchanger).
g = acceleration due to gravity.
μ = viscosity of air.

The following are the significant forces encountered in the system while in operation:

(1) μVD = air viscosity forces.
(2) ρV^2D^2 = aerodynamic inertia forces of cold air.
(3) $\rho' V^2D^2$ = aerodynamic inertia forces in hot plume.
(4) $(\rho-\rho')gD^3$ = buoyancy forces on plume.

The ratios of these various forces determine the required non-dimensional numbers which must be identical in model and prototype to ensure complete similarity.

Putting $\frac{(2)}{(1)}$ gives $\frac{\rho VD}{\mu}$ (the Reynolds number)

Putting $\frac{(3)}{(2)}$ gives $\frac{\rho' V_e^2}{\rho V^2}$ (the specific impulse ratio)

Putting $\frac{(4)}{(2)}$ gives $\frac{\rho-\rho'}{\rho} \cdot \frac{gD}{V^2}$ (the densimetric Froude number)

If the values of the above non-dimensional numbers are identical in model and full-scale prototype, it is generally accepted that reliable results can be expected. Nevertheless, it is possible to reduce the specific impulse ratio and the densimetric Froude number to simpler components.

Since the hot air leaving the heat exchangers is at a temperature of 30° above the assumed ambient air temperature of 20°, the density ratio $\rho'/\rho = 0{,}91$ can readily be obtained by mixing alcohol with water. Thus, the derived simplified dimensional numbers ρ'/ρ (density ratio), V_e^2/V^2 or V_e/V, and gD/V^2 (Froude number) may be used for this test. It is of course impossible to simulate the Reynolds number $\rho VD/\mu$ in the model as is always the case for such small-scale tests. It is argued that the Reynolds number discrepancy will not be serious when testing a model power station which is essentially a bluff body and therefore relatively insensitive to a Reynolds number discrepancy. However, Reynolds number effects on actual plumes is an uncertain factor.

A linear scale of 1:1800 was selected for this test and from the Froude number it follows that the velocity scale, i.e. the ratio of full-scale velocity to model velocity is $\sqrt{1800}$, i.e. 42,4. The full scale velocity efflux from the top of the heat exchangers was assumed to be 5 m/s.

3 DESCRIPTION OF MODEL, INSTRUMENTATION AND TESTING PROCEDURE

3.1 The Model

The heat exchanger model was essentially the same system as the one previously used for the Matimba (Ref. [1]) investigation.

The heat exchanger system of Matimba power station consists of an array of 72 pillars supporting heat exchangers through which an array of 288 fans force air.

Inspection of the Matimba heat exchanger system indicated that the fans sucked air up from the lower surface of the system and discharged this air into the voids beneath the heat exchangers which simulated a discharge into a plenum chamber. From this plenum chamber, the air flowed upwards uniformly through the heat exchanger fins.

In the model, fans were therefore not required, but it was necessary to provide a system where the water could be sucked in at the bottom of the bank of simulated heat exchangers, while a different fluid (in this instance, alcohol and water) representing the less dense efflux, could be uniformly ejected out of the top surface of the heat exchanger. This heat exchanger model, illustrated in Figure 1, was constructed as follows: Each pillar, represented by a thin tube, was used to suck water away from the base of the model heat exchanger through small holes drilled in the base (see Figure 2). Thus, the inflow at the base could be simulated by allowing all the small tubes connected to the 72 pipes to discharge in the tunnel pit at a depth calibrated to give the correct flow rate for the four fans represented by the small holes surrounding each pillar.

The upper compartment of the model heat exchanger was completely separated from the lower compartment from which the water was sucked out. The upper compartment which was covered by a fine gauze contained tubes with suitable holes in the walls to ensure equally distributed efflux of the water/alcohol mixture, as is indicated in Figure 2. This water/alcohol mixture was supplied from header tanks for which the elevation was calibrated so as to give the required rate of efflux from the model heat exchangers.

The wind was simulated as a flow with a conventional vertical wind profile obtained by fitting "spires" upstream of the model and also small regularly spaced blocks in the model floor upstream of the model so as to retard flow on ground level. Only a wind direction normal to the longitudinal axis of the heat exchanger was simulated. The models of chimney stacks and all structures upwind of boiler houses were omitted and only the boiler and turbine houses were included as depicted in Figure 1.

3.2 Instrumentation and Testing Procedure

One of the advantages of water-tunnel testing is the capability of making quantitative measurements of plume dispersion and recirculation. In carrying out these quantitative measurements of recirculation taking place from the upper surface of the heat exchanger system to its lower inlet surface, the Fluorometer system used for previous investigations (Ref. [1]) was again employed. A known mass of quinine sulphate was introduced into the mixture of water and alcohol which represented the efflux plume.

For any particular test where the tunnel speed was adjusted to represent a given wind speed and where the exit plume had become stable, six samples, each coming from the six suction inlets in the model heat exchanger, were taken in small flasks. Altogether the 72 pillars in the model each sucked fluid away from the underside of the heat exchanger but 12 such suction tubes were combined thus giving six sampling points.

A Fluorometer, which can measure the quinine sulphate content in parts per million (ppm) of any sample taken anywhere in the flow field, was then used to determine the concentration of quinine sulphate at each of the six outlets. In this way, the percentage recirculation presented at any of the six sampling points could readily be determined.

The boiler houses and turbine houses could be moved away from the heat exchanger array from a position abutting the heat exchanger to gaps represented respectively by $^1/_3$ of the heat exchanger height, the full height of the heat exchanger and ultimately to a position where the boiler and turbine houses were removed and the heat exchanger was exposed to unrestricted wind. Percentage recirculation was measured for each gap setting. The test where it was required to establish whether fans were "starved" when simulated wind was allowed to flow freely underneath the heat exchanger, flow rates were simply measured by measuring the volume of fluid sucked from underneath the heat exchanger in a given time lapse. To check if any fans were "starved" a controlled experiment was made where the flow of wind underneath the heat exchanger was blocked off completely by the model boiler and turbine houses.

4 RESULTS AND DISCUSSION

4.1 The Effect on Hot Air Recirculation of the Gap Width between Turbine Housing and Heat Exchanger

Figure 3 gives perhaps the most significant illustration of the effect of a gap between the heat exchanger and the boiler housing. The graphs in this figure depicts total percentage recirculation in the whole heat exchanger plotted against wind speed. It can be seen that the percentage recirculation of the efflux alcohol plume representing hot air is drastically reduced by increasing the gap between the boiler housing and heat exchanger from zero to $^1/_3$ of the heat exchanger height. A further increase in this gap sized to one heat exchanger height hardly has any further circulation reducing effect. The rather hypothetical case where the heat exchanger is isolated from the boiler housing, allowing wind to flow unrestricted underneath the heat exchanger, is clearly the most advantageous setup. Figure 3 shows that even at rather high wind speeds of 7 m/s only slight recirculation takes place.

Figures 4 to 7 show graphs of percentage recirculation versus wind speed for each of the three pillar rows (refer to Figure 2). These graphs indicate in what regions most recirculation occurs.

Figures 8 to 21 show the distribution of the hot air recirculation along each of the three rows of pillars, the position of each pillar representing four fans. As can be seen on the graphs only six sampling points have been taken along each row of 24 pillars, which means that the fluid from the suction tubes of four pillars in each row has been combined and has been treated as one sample (see Figure 1). It is felt that six such sampling points in each row adequately illustrates the distribution of hot air recirculation. Only the two higher wind speeds are considered for the case where the turbine house is removed since lower wind speeds indicated zero recirculation.

4.2 The Possible "Starving" Effect on the Supply of Air to Fans when Wind is allowed to flow crosswise underneath the Fan Inlets

Although the water tunnel model of the heat exchanger does not faithfully represent the fans and fan inlets underneath the heat exchanger body it is argued that any decrease in pressure, owing to wind, which may starve full scale fans, will also be present when wind flow is simulated underneath the model heat exchanger. Such a decrease in pressure should then also starve the flow which is sucked out via the perforations in the lower surface of the model heat exchanger.

Extensive model tests with the model boiler - turbine housing in position against the heat exchanger thus presenting no wind flow and with the housing removed, thus allowing unobstructed wind flow, were carried out.

Table I gives the measured rates of flow of water sucked up in each pillar row at various wind speeds, both for unobstructed crossflow as well as crossflow blocked off by the presence of model boiler-turbine housing.

Table I

WIND-SPEED	BOILER-TURBINE HOUSING REMOVED. MODEL VOLUMETRIC FLOW IN mℓ/s			BOILER-TURBINE HOUSING PRESENT. MODEL VOLUMETRIC FLOW IN mℓ/s		
	ROW 1	ROW 2	ROW 3	ROW 1	ROW 2	ROW 3
3	10,121	10,282	10,002	10,322	10,493	10,174
4	10,333	10,557	10,320	10,410	10,351	10,171
5	10,090	10,228	10,149	10,314	10,408	10,163
6	10,285	10,389	10,098	10,264	10,440	10,097
7	10,116	10,353	10,012	10,289	10,284	10,137

Figure 22 shows the total flow into the lower surface of the model heat exchanger for both cases of boiler-turbine housing present and removed, this inflow being plotted versus wind speed.

Some scatter in the results are present but a slight starving of inflow underneath the heat exchanger, in some cases in the order of 1%, is evident.

4.3 Recirculation of Hot Air at Zero or Very Low Wind Velocities

Contrary to indications from theoretical analysis (Ref. [2]) no clear evidence could be found of recirculation taking place around the edges of the heat exchanger at no or very light wind velocities. In the present test series as well as in the test series reported in part I of Ref. [1], such recirculation at very low wind speeds (3 m/s or less) could not be detected.

5 CONCLUSIONS AND RECOMMENDATIONS

It was clearly shown that a gap $^1/_3$ times the height of the heat exchanger stack drastically reduced recirculation and that an increase in the gap size equal to the height of the heat exchanger stack did not further reduce the recirculation. It may be concluded that small gaps are beneficial and incorporating such gaps is recommended.

It is thought that the "starving" of air to fans when a free flow of wind underneath the heat exchanger tanks is allowed, is only slight and the gain in heat exchanger effectiveness, by reducing the hot air recirculation, outweighs any bad effects due to "starving".

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[1] van der Elst, W.J. : "Water tunnel tests on a forced-draught cooling system for a power station", Parts I, II and III, (Contract reports, ME1966/9, Nov. 1986, ME 2035, May 1987 and ME 2029/4, Nov. 1987).

[2] Kröger, D.G. : "Numerical predictions of air flow about air-cooled heat exchangers", Under preparation.

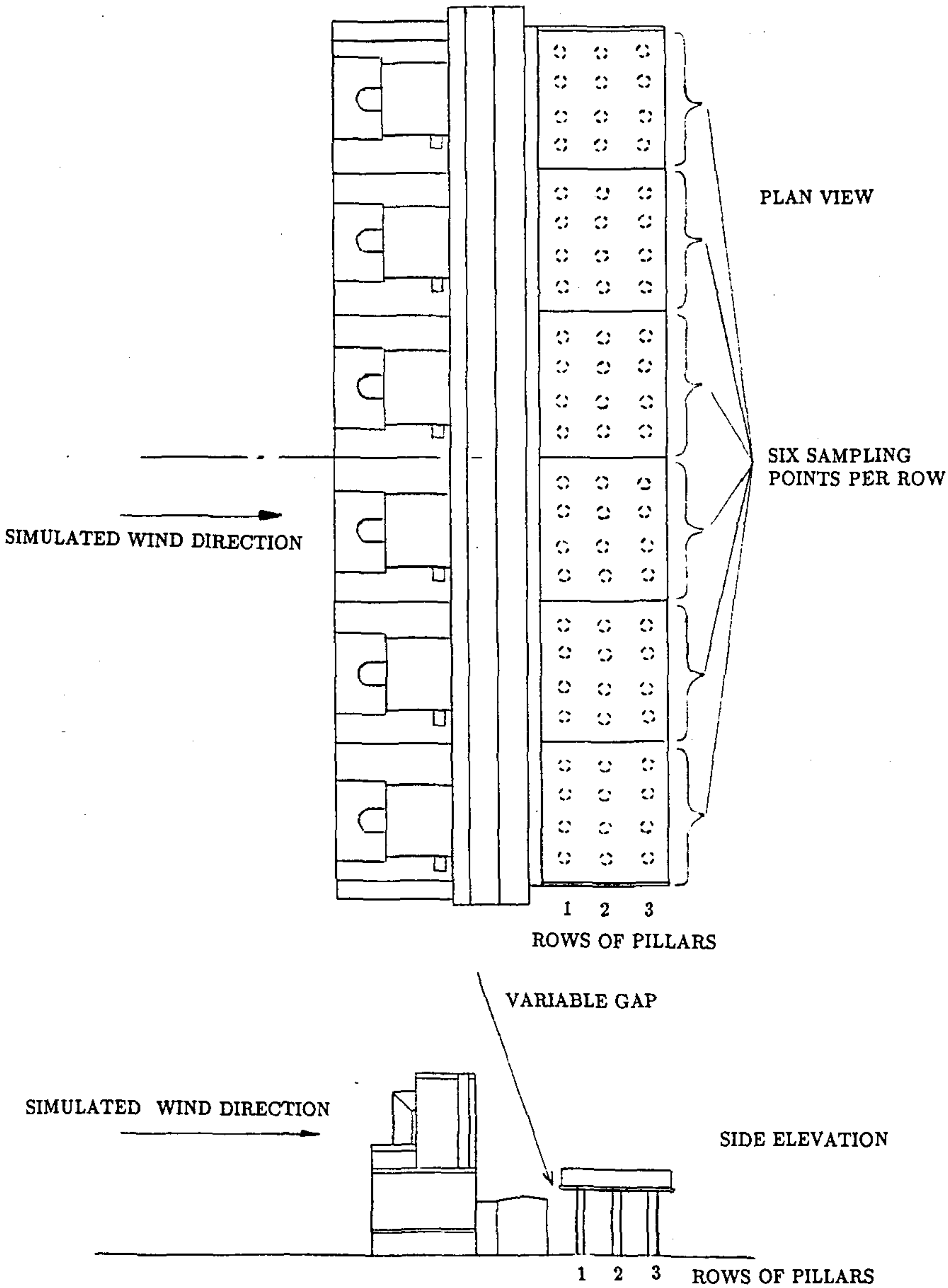

FIGURE 1: LAYOUT OF POWER STATION MODEL

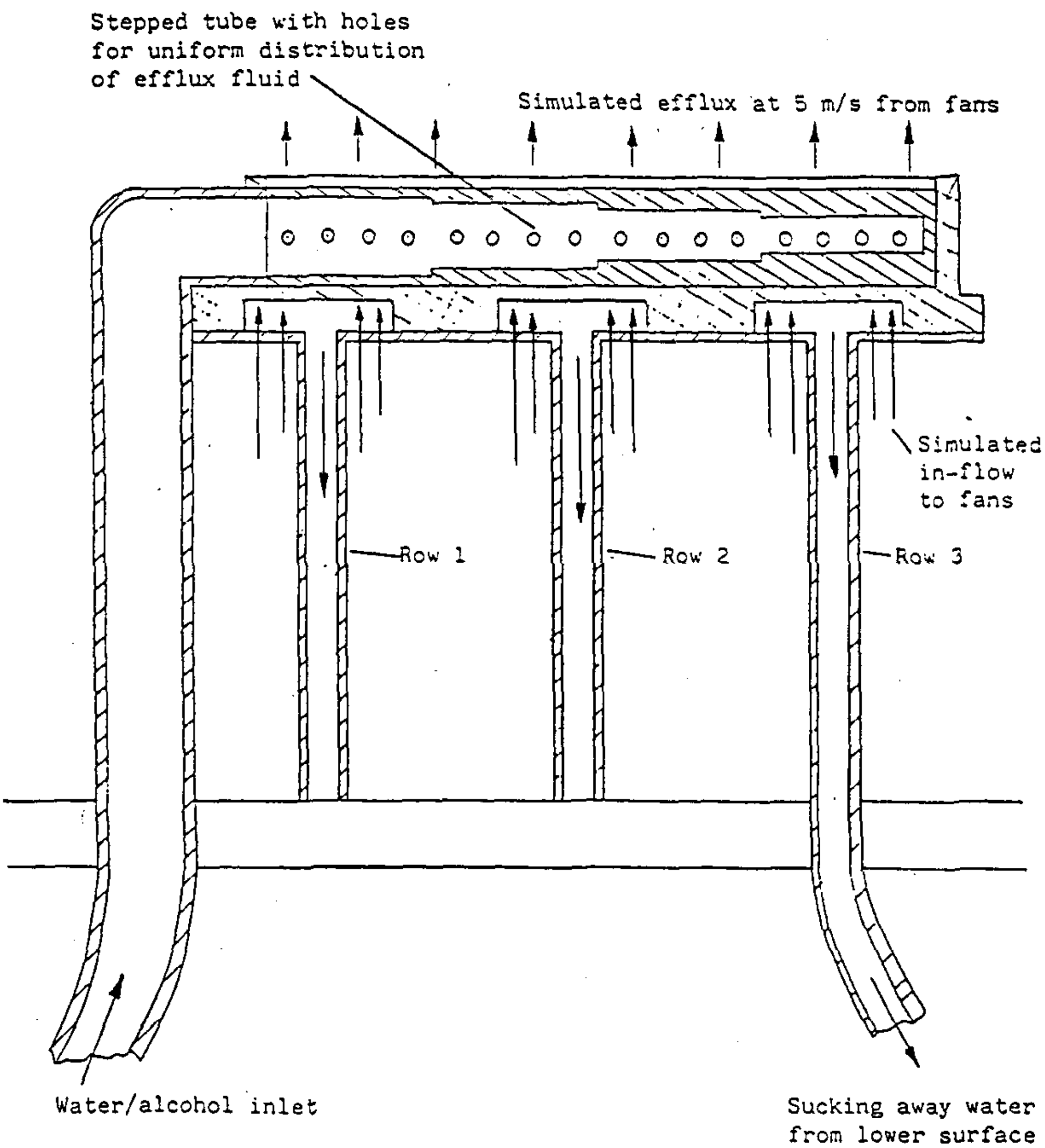

FIG. 2: CROSS-SECTIONAL SIDE ELEVATION OF MODEL HEAT EXCHANGER

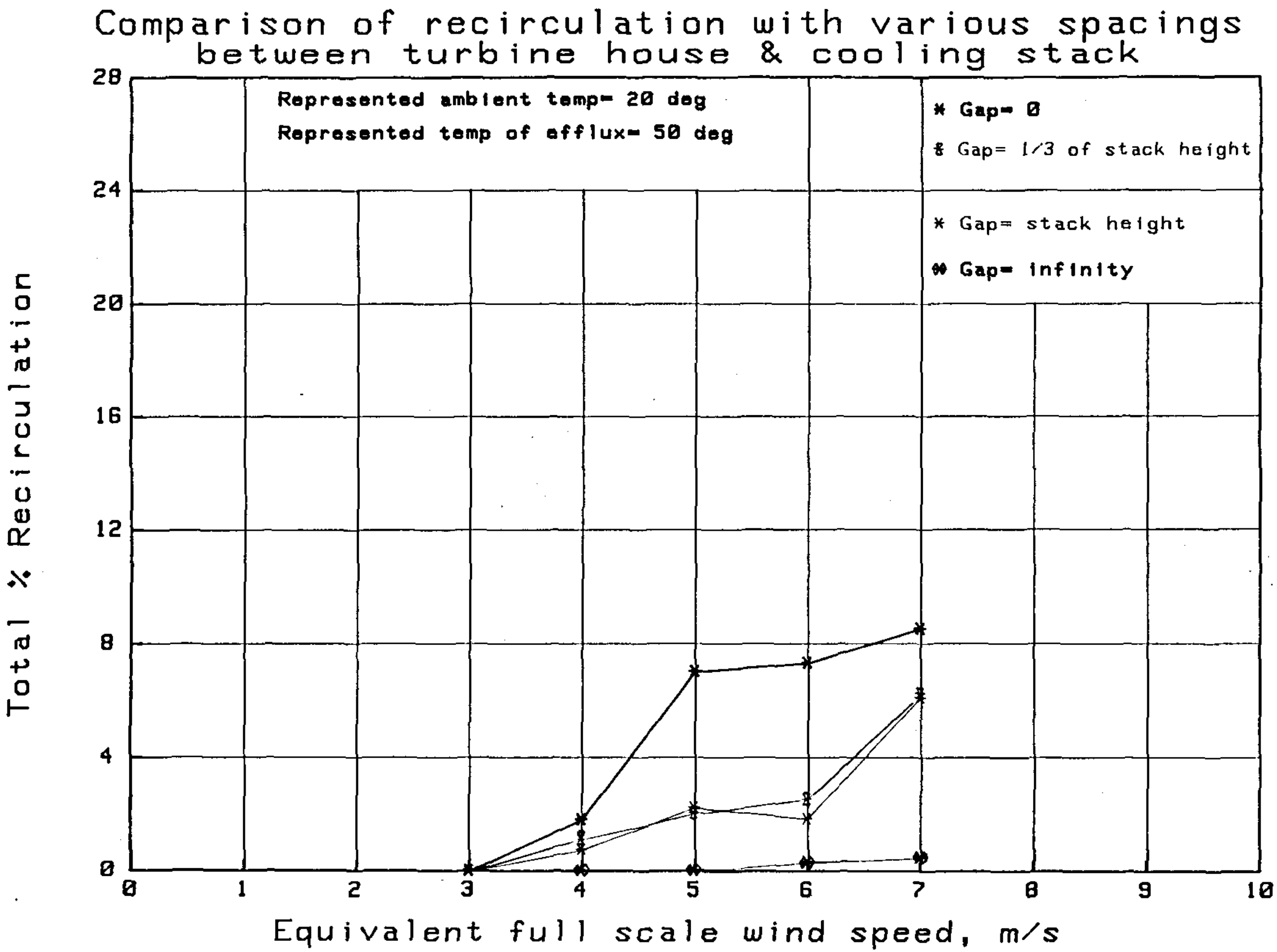

Fig 3. Recirculation showing effect of wind speed

Turbine house next to cooling stack

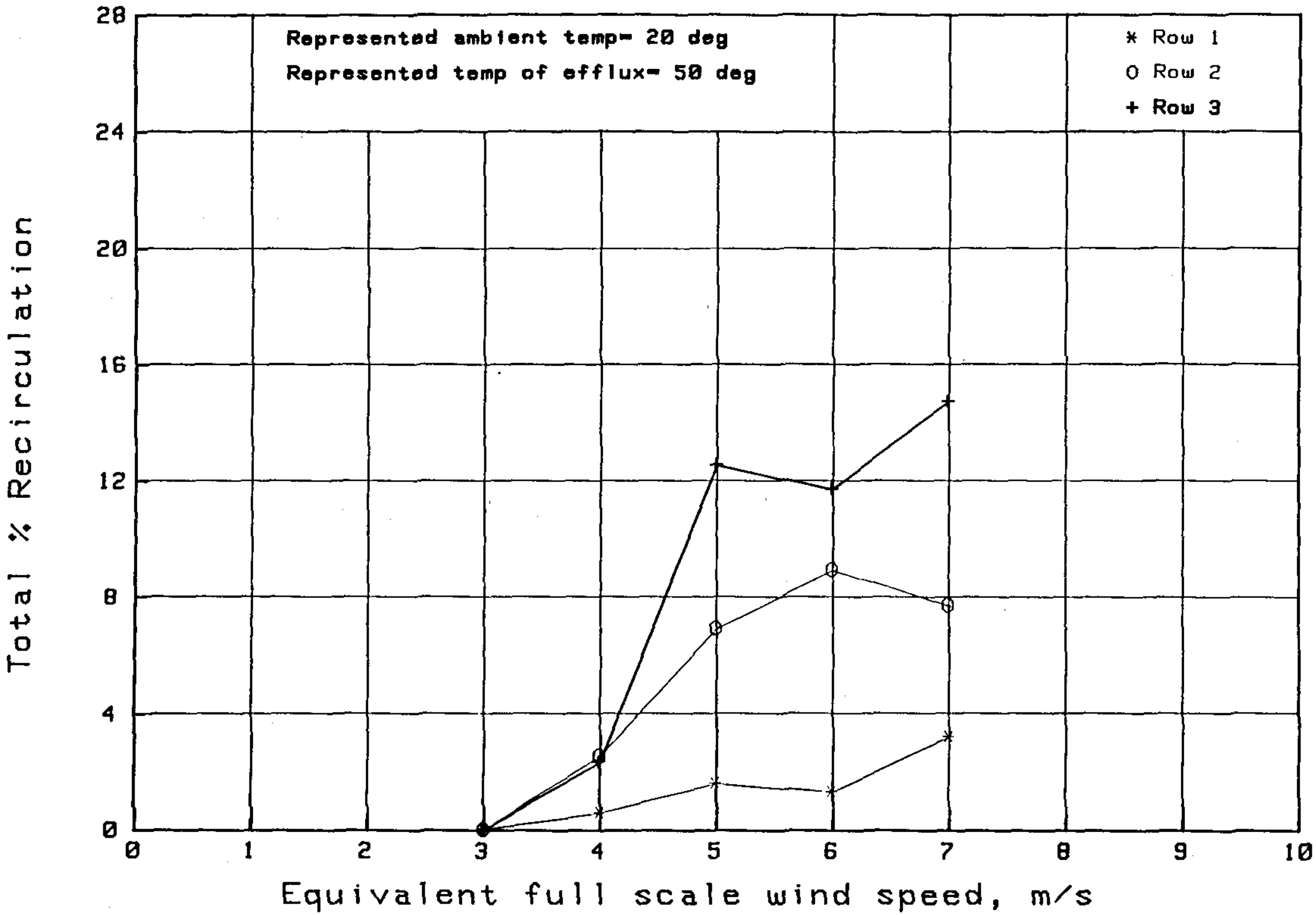

Fig 4. Recirculation showing effect of wind speed

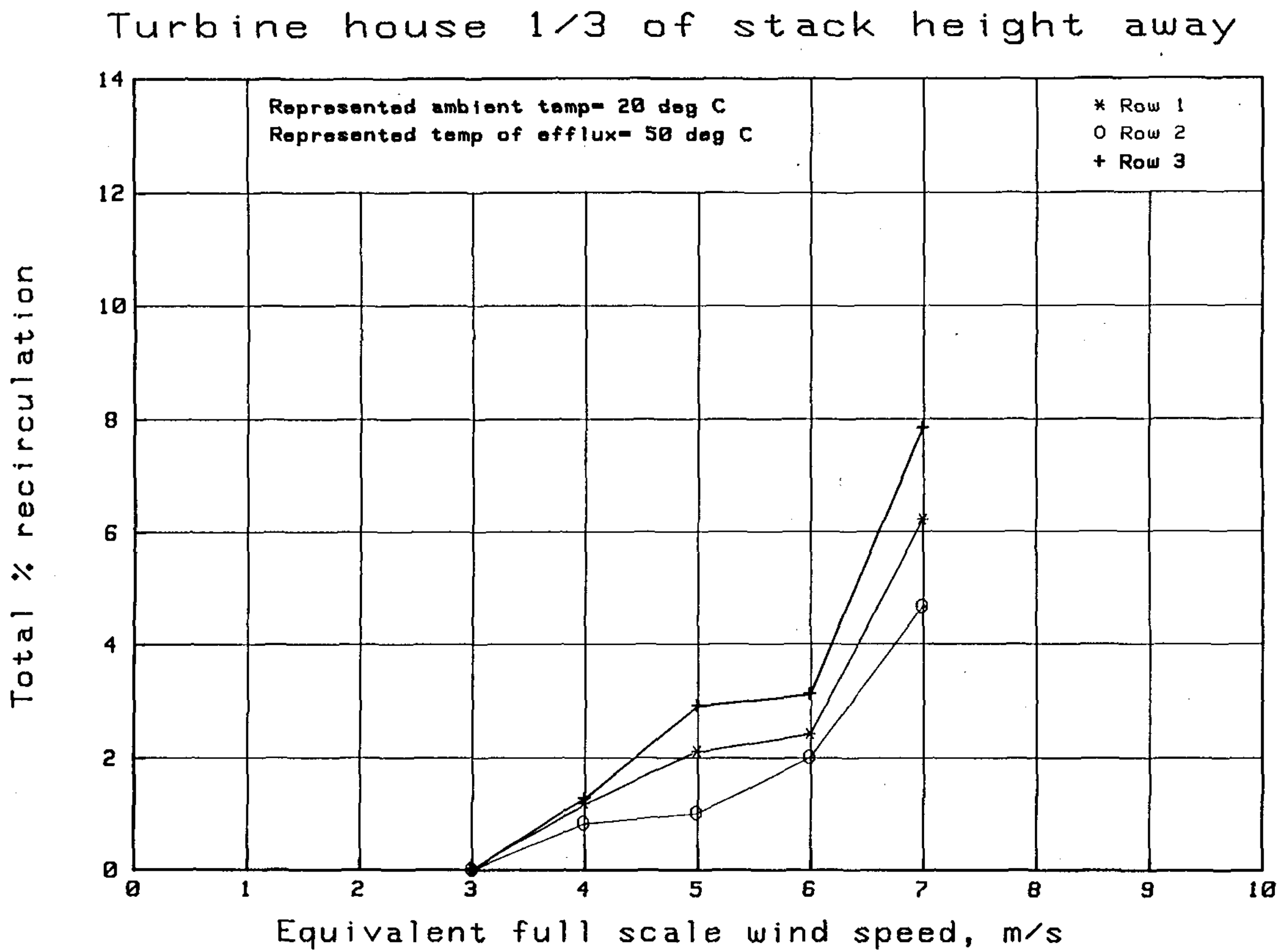

Fig 5. Recirculation showing effect of wind velocity

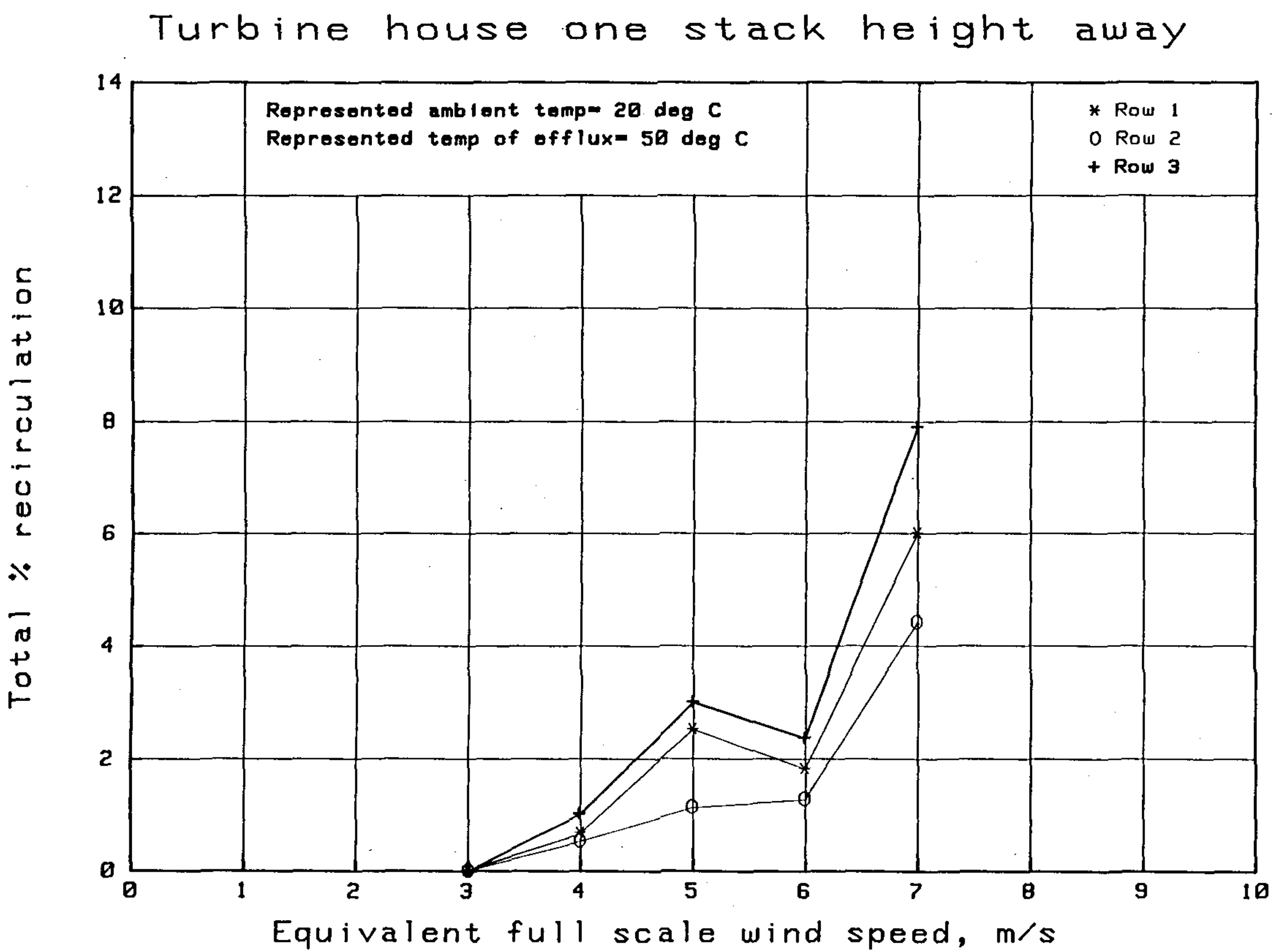

Fig 6. Recirculation showing effect of wind velocity

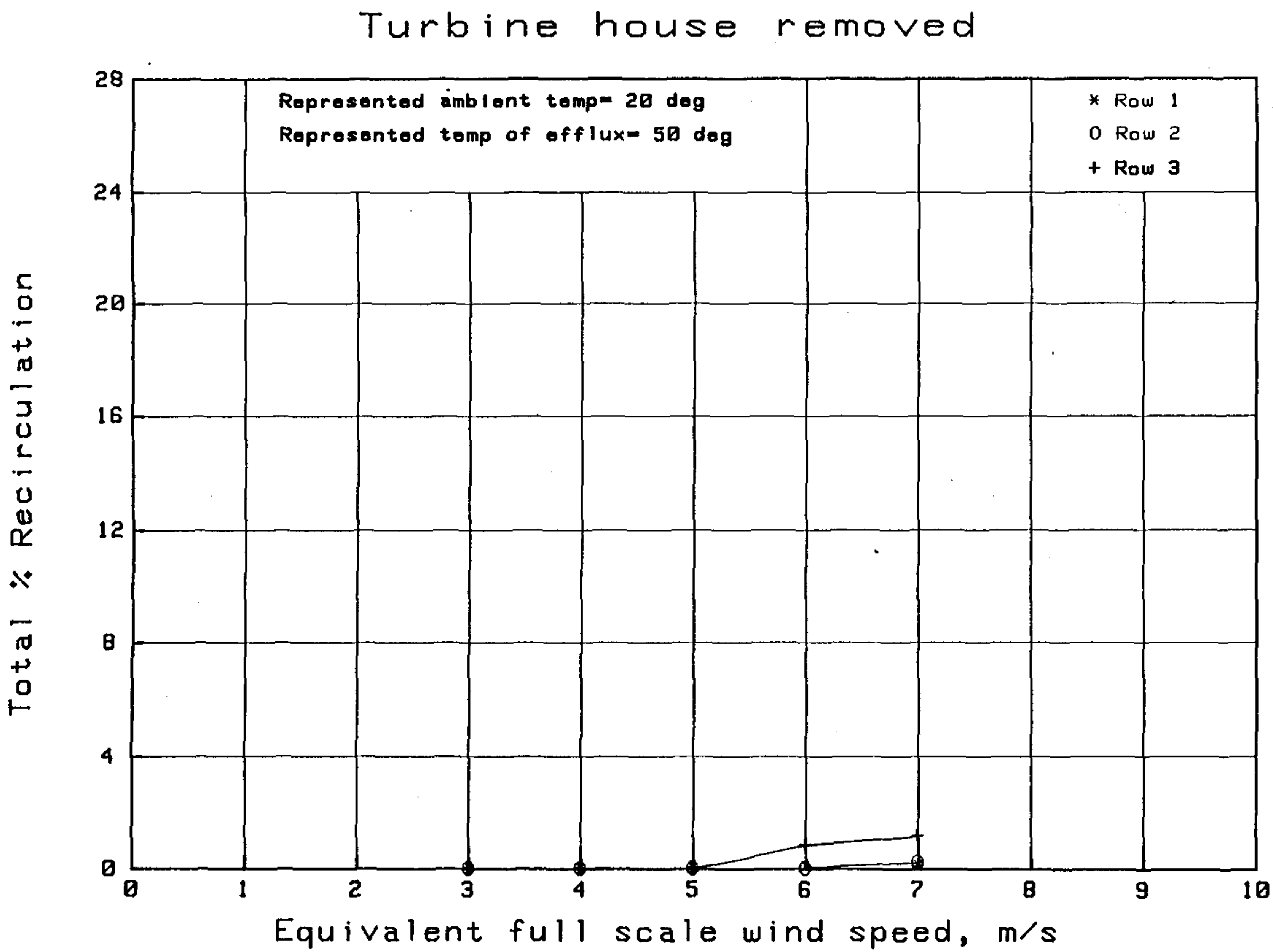

Fig 7. Recirculation showing effect of wind speed

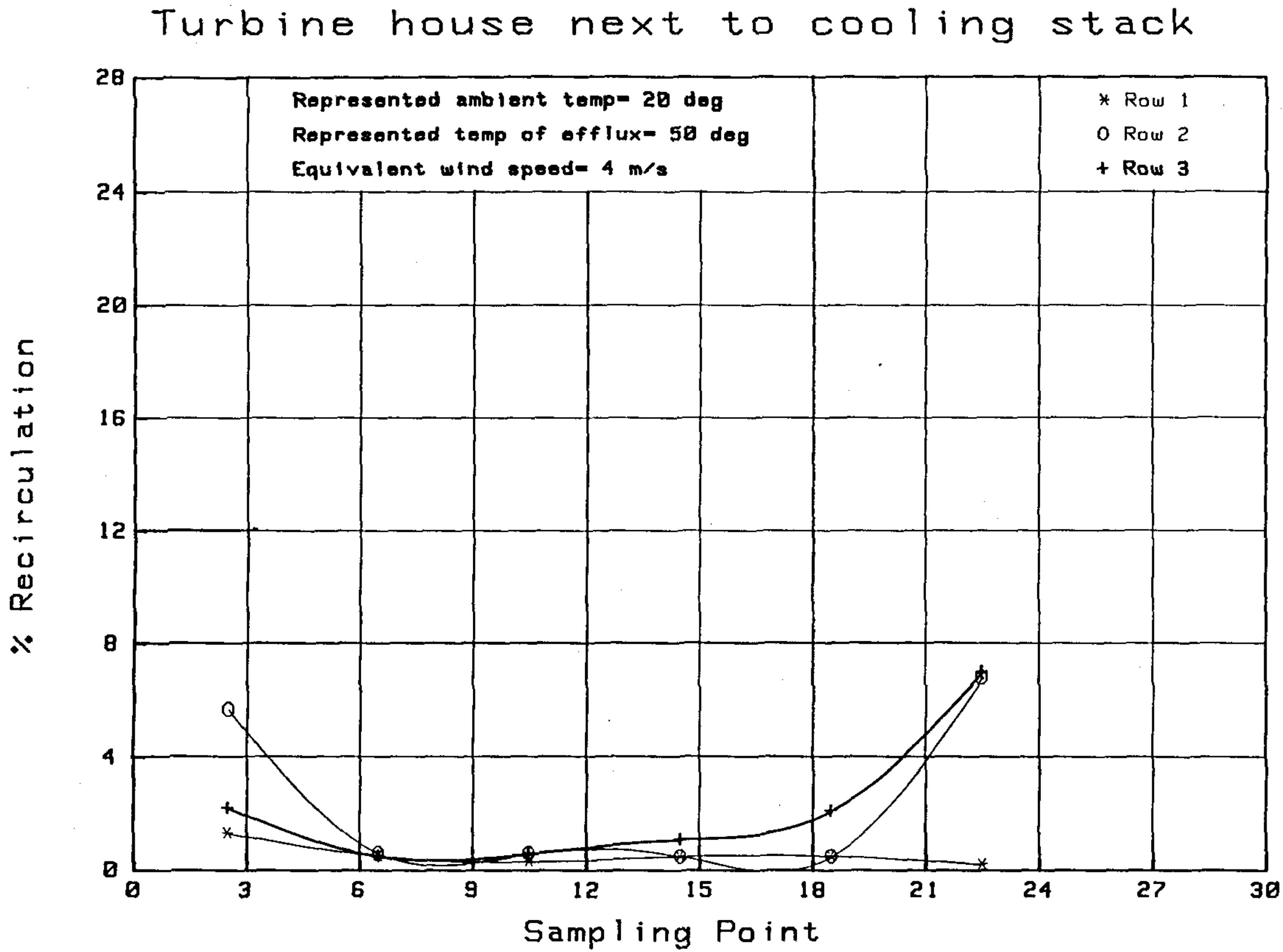

Fig 8. Hot air recirculation in heat exchanger

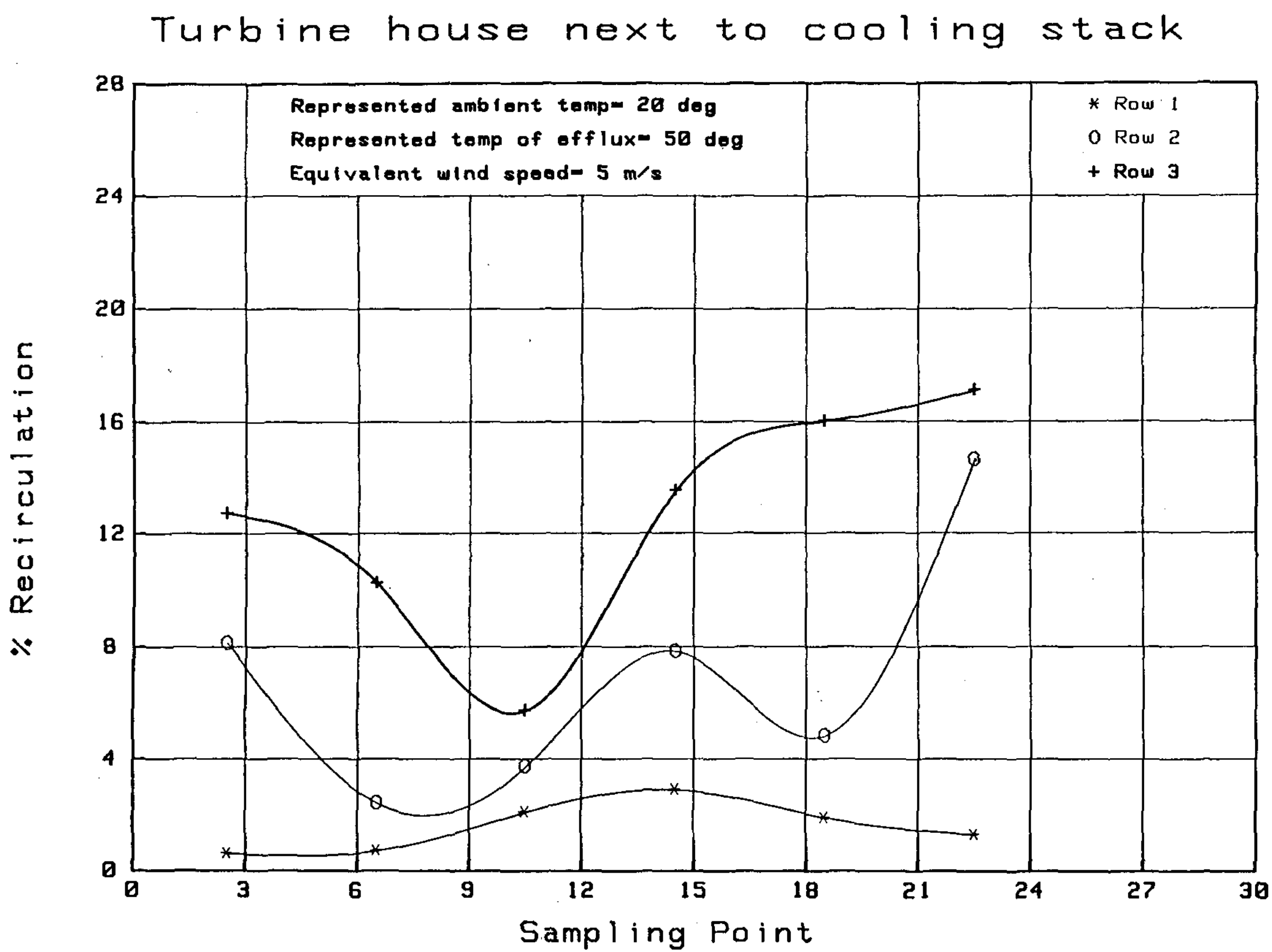

Fig 9. Hot air recirculation in heat exchanger

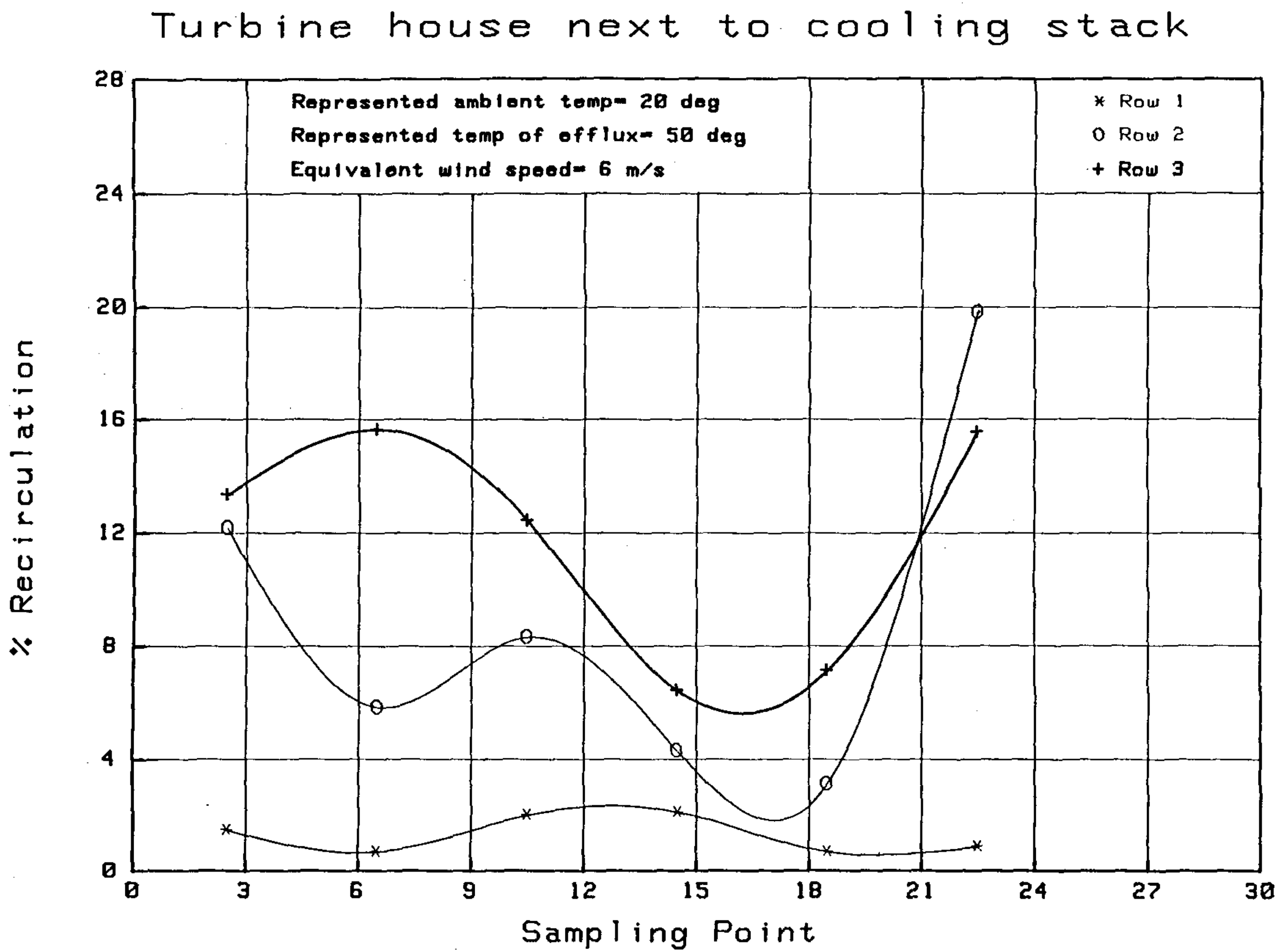

Fig 10. Hot air recirculation in heat exchanger

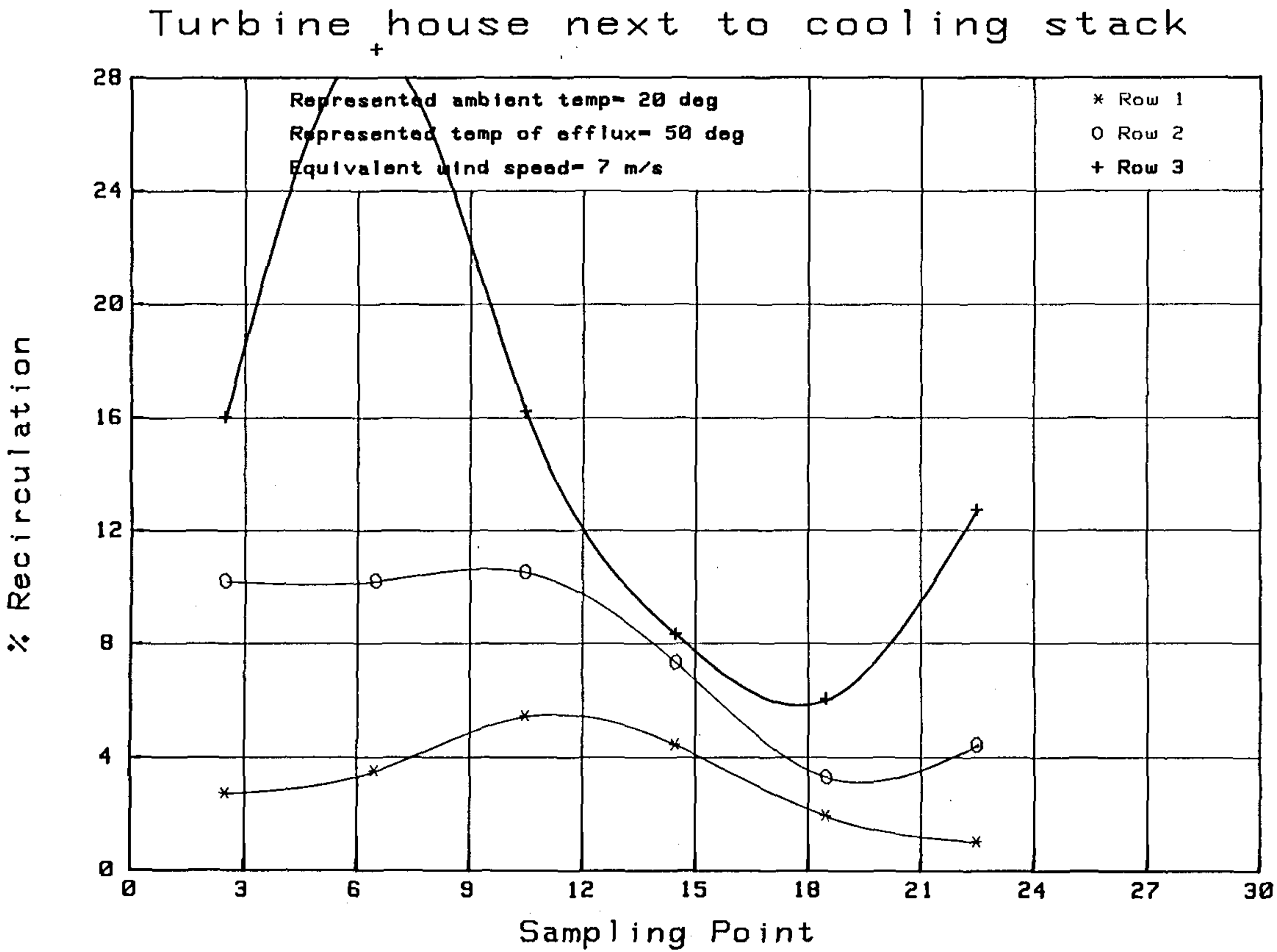

Fig 11. Hot air recirculation in heat exchanger

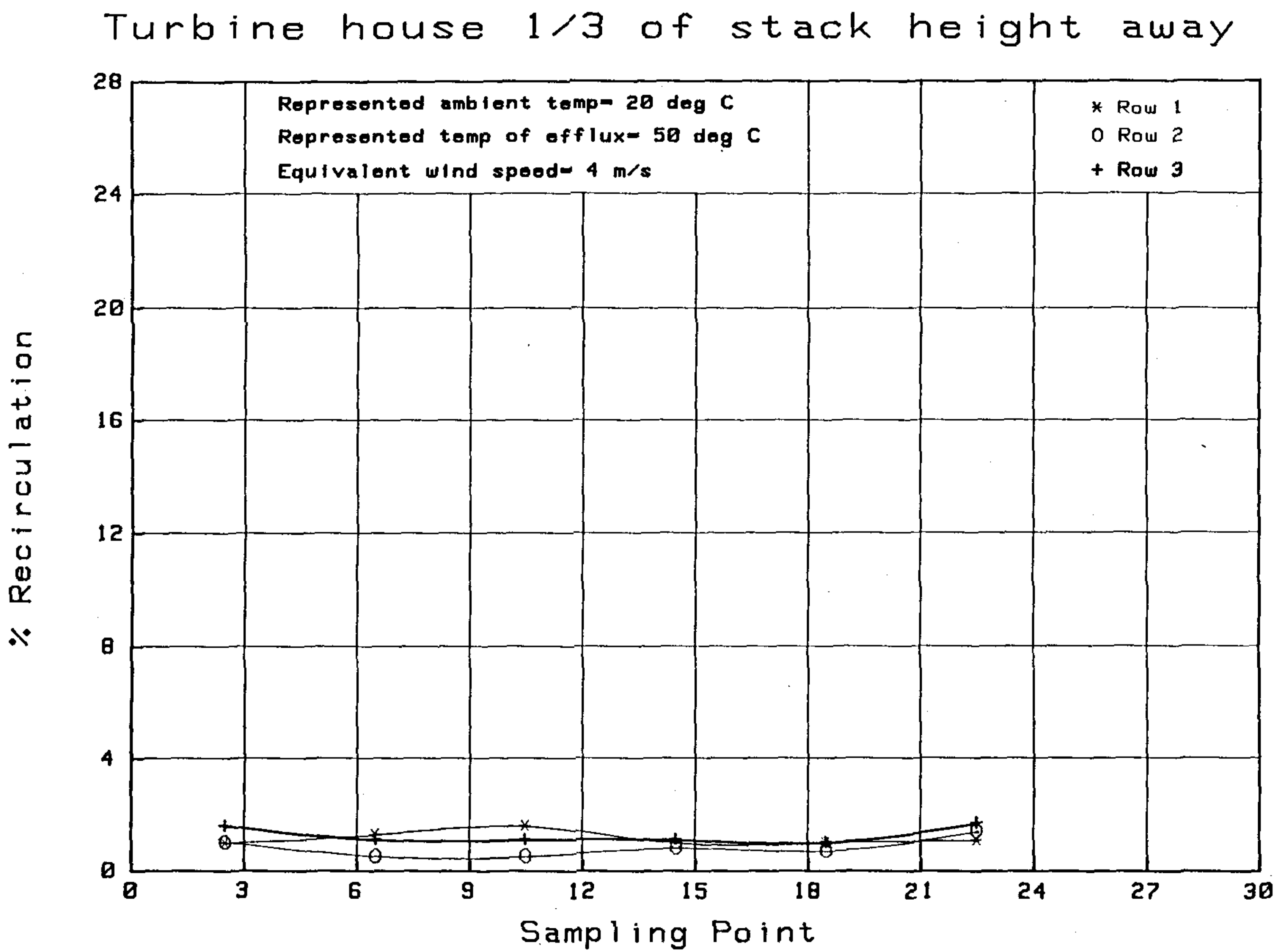

Fig 12. Hot air recirculation in heat exchanger

Turbine house 1/3 of stack height away

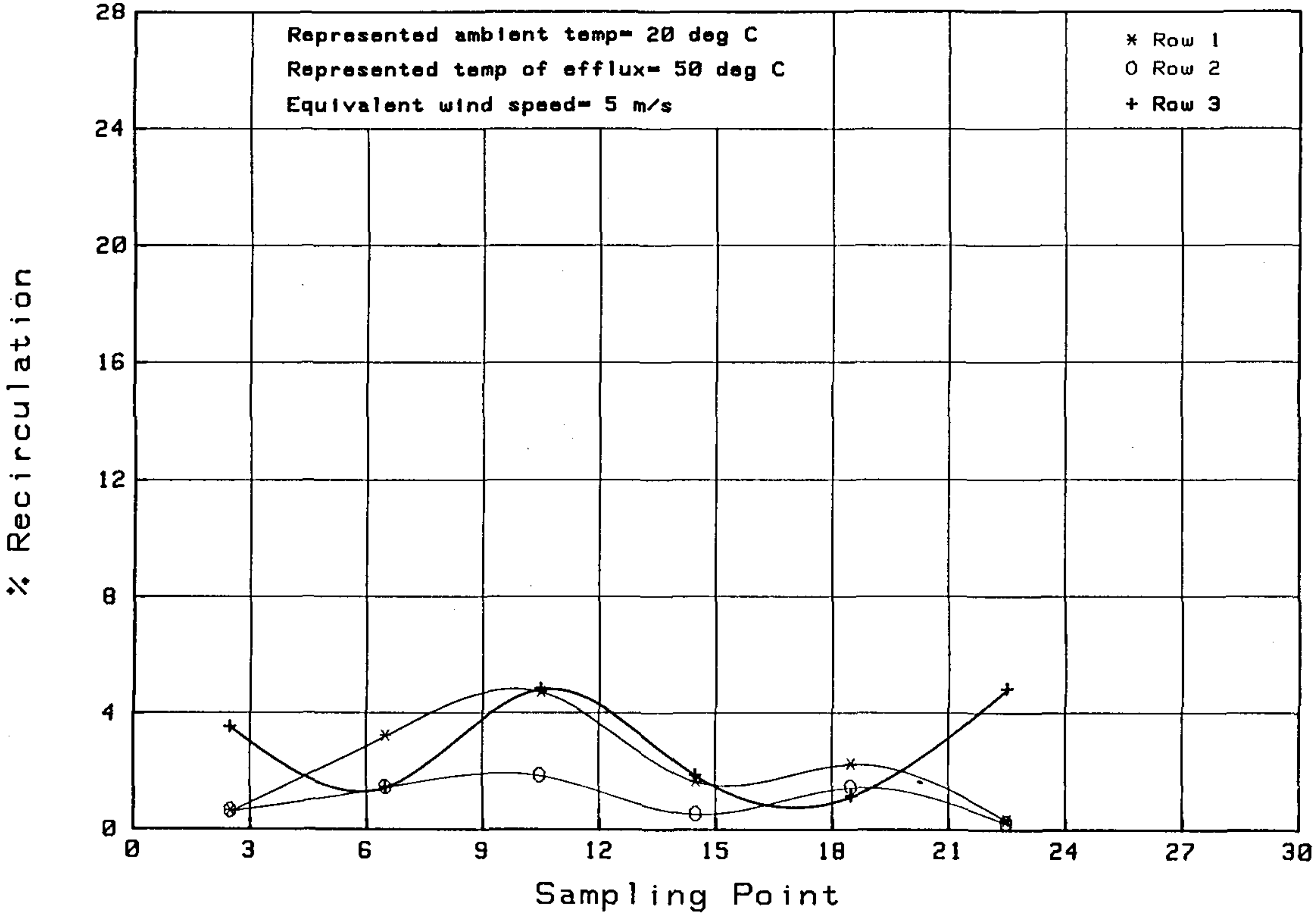

Fig 13. Hot air recirculation in heat exchanger

Turbine house 1/3 of stack height away

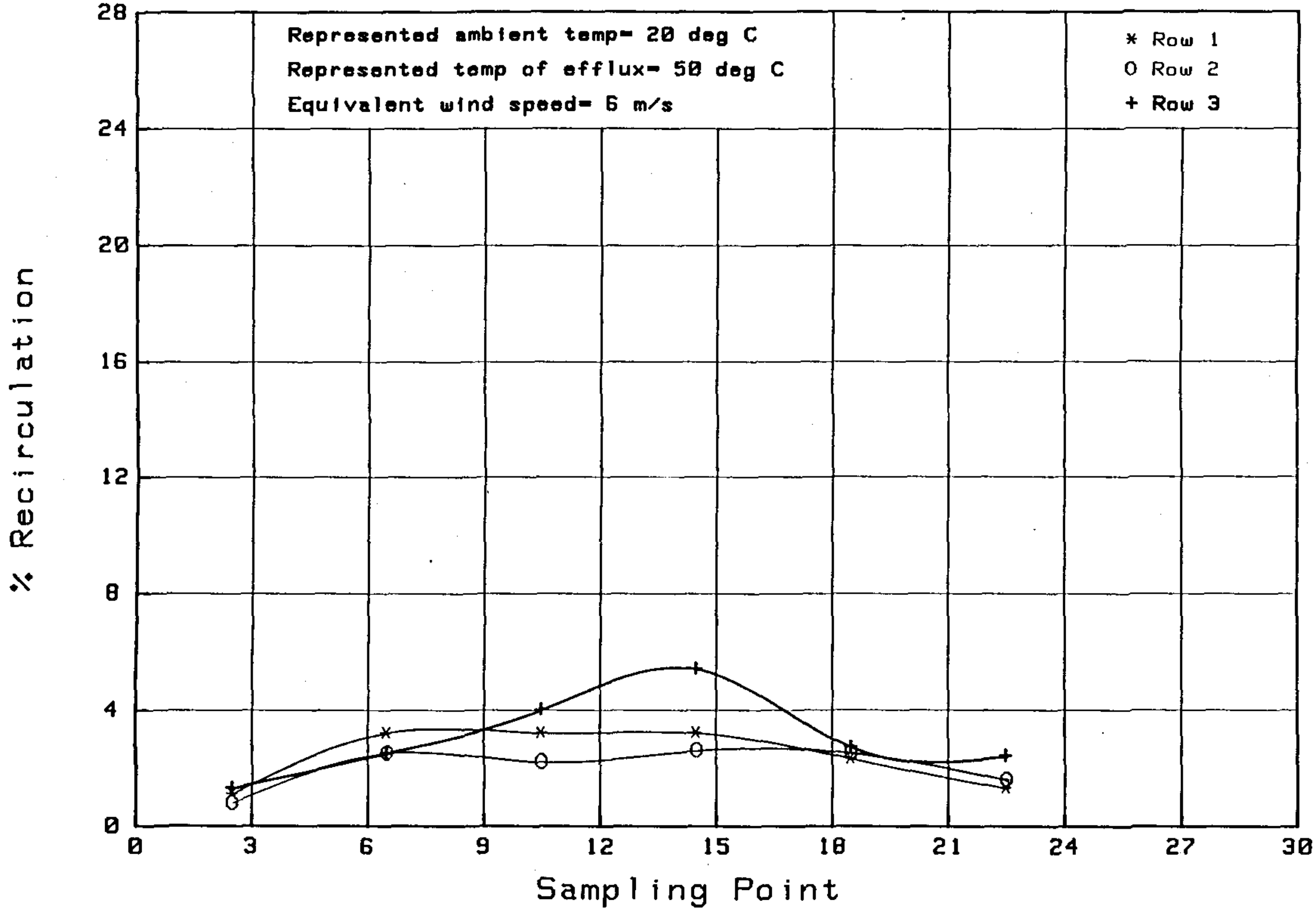

Fig 14. Hot air recirculation in heat exchanger

Turbine house 1/3 of stack height away

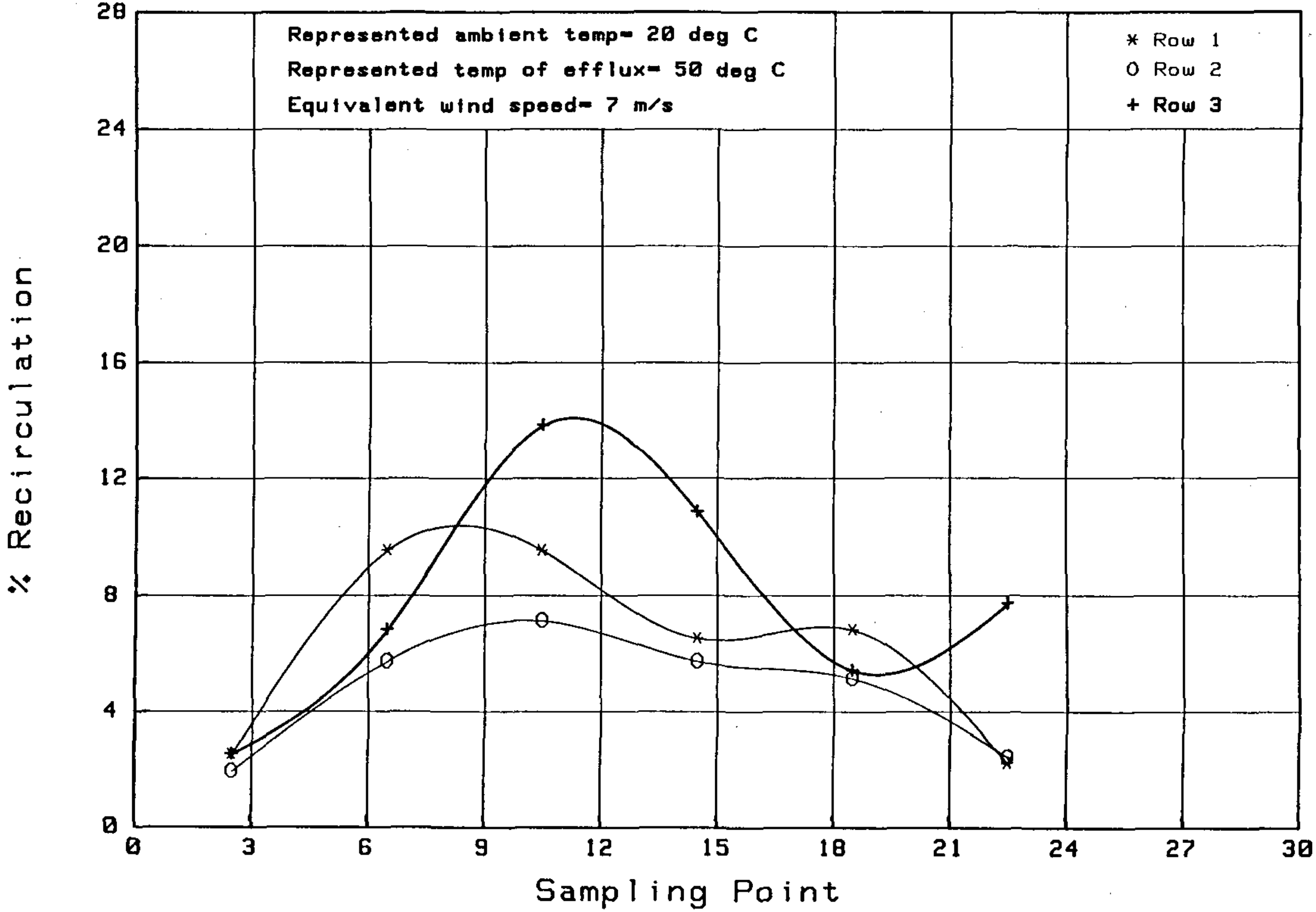

Fig 15. Hot air recirculation in heat exchanger

Turbine house one stack height away

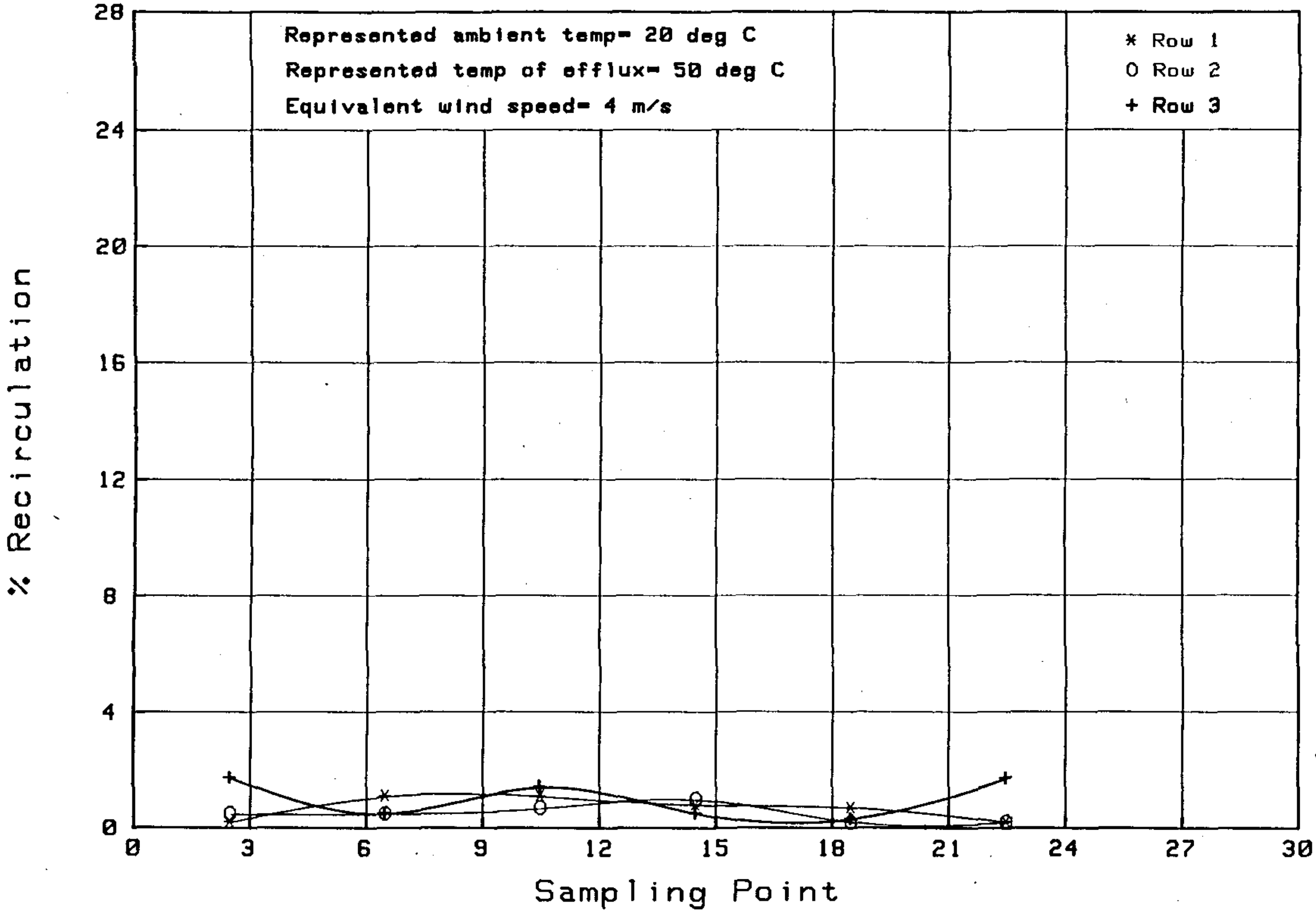

Fig 16. Hot air recirculation in heat exchanger

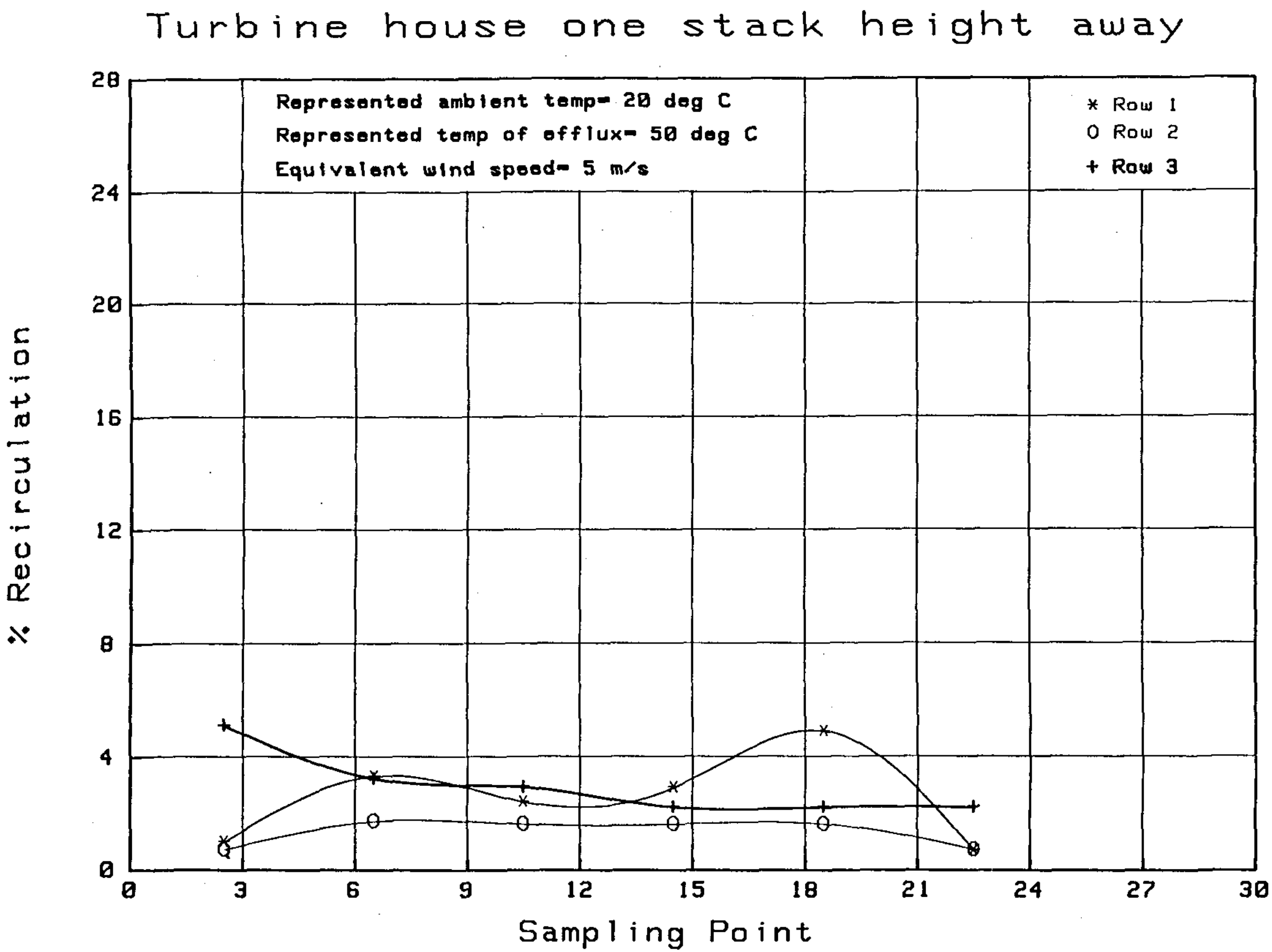

Fig 17. Hot air recirculation in heat exchanger

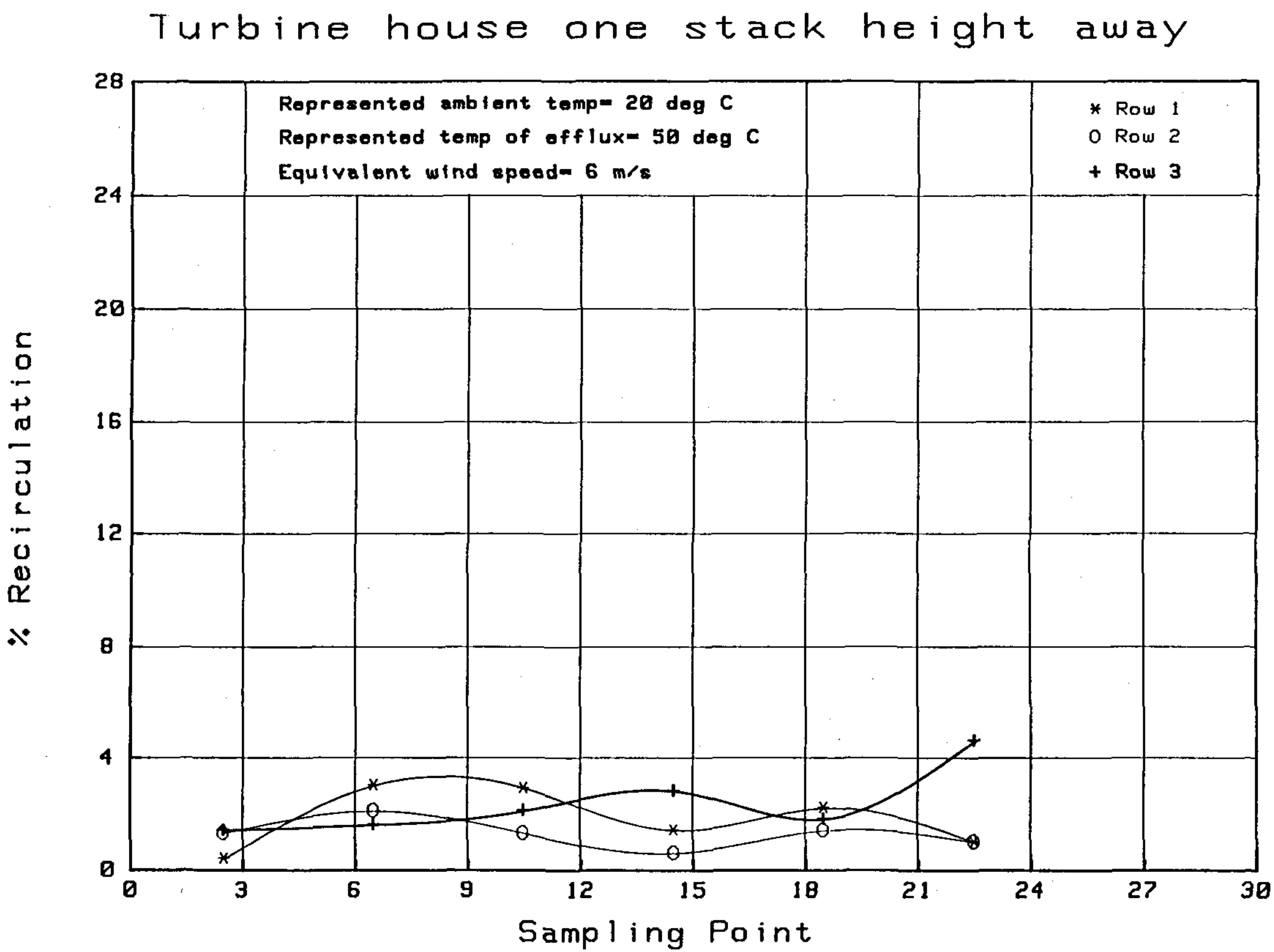

Fig 18. Hot air recirculation in heat exchanger

Turbine house one stack height away

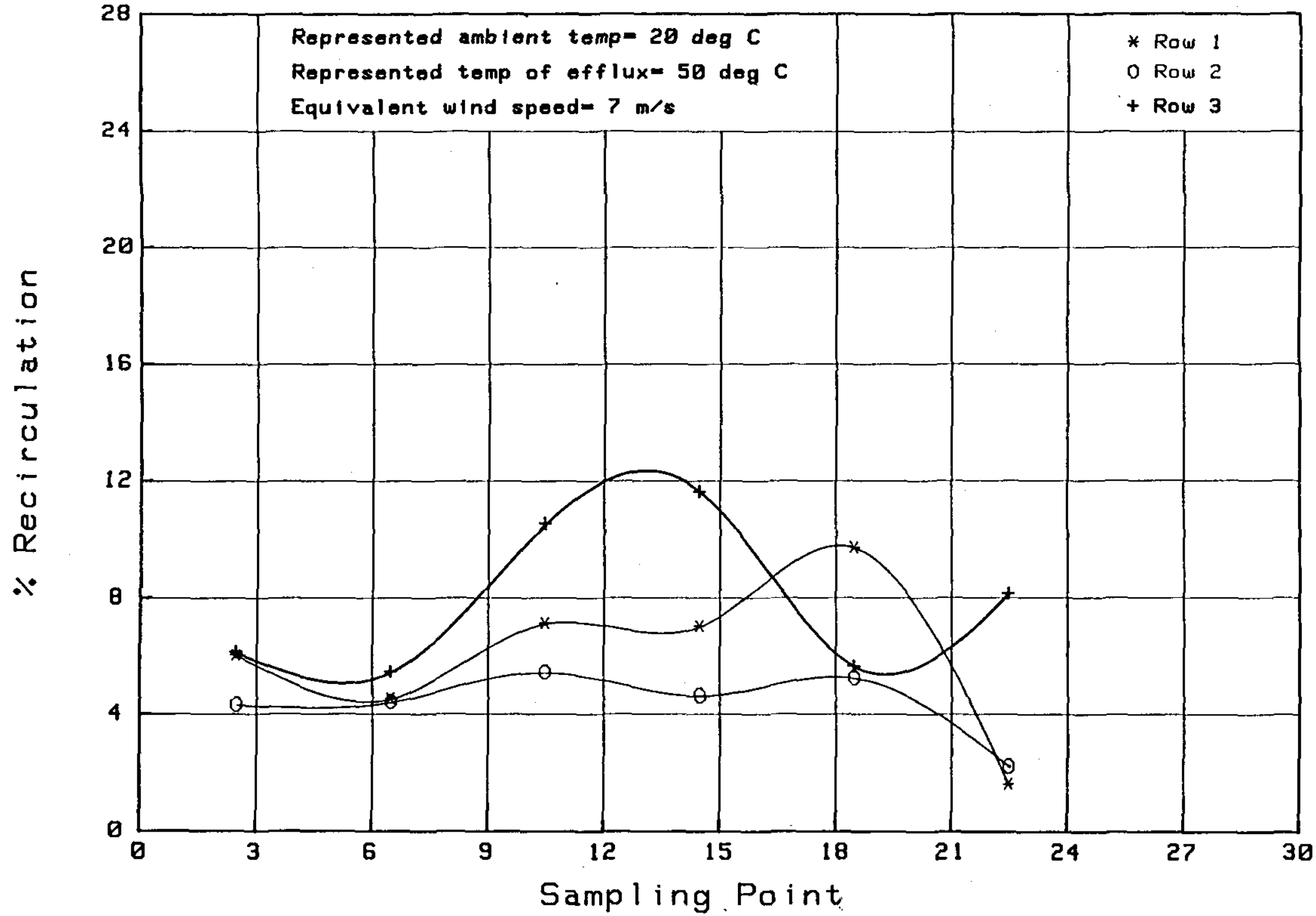

Fig 19. Hot air recirculation in heat exchanger

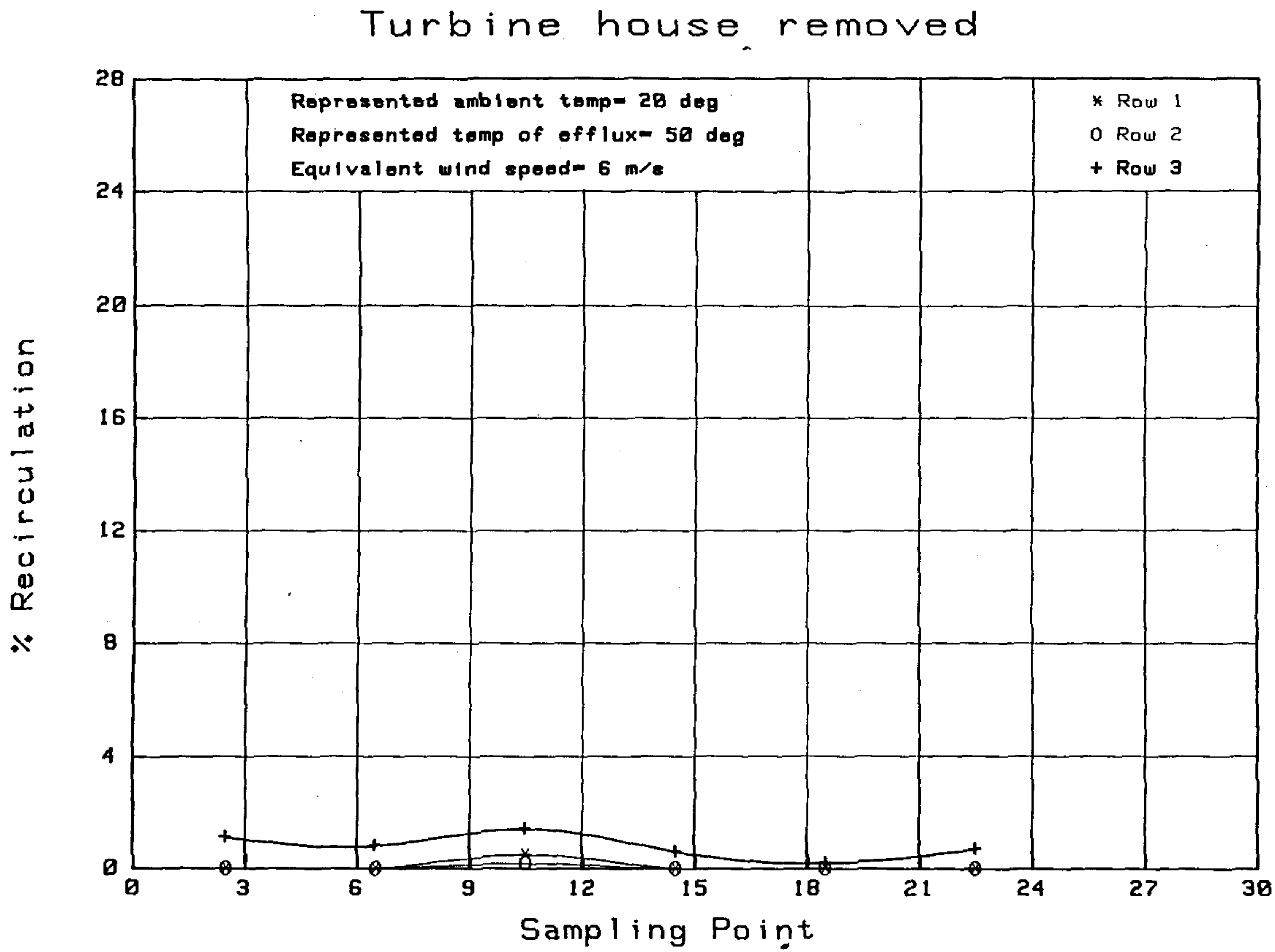

Fig 20. Hot air recirculation in heat exchanger

Turbine house removed

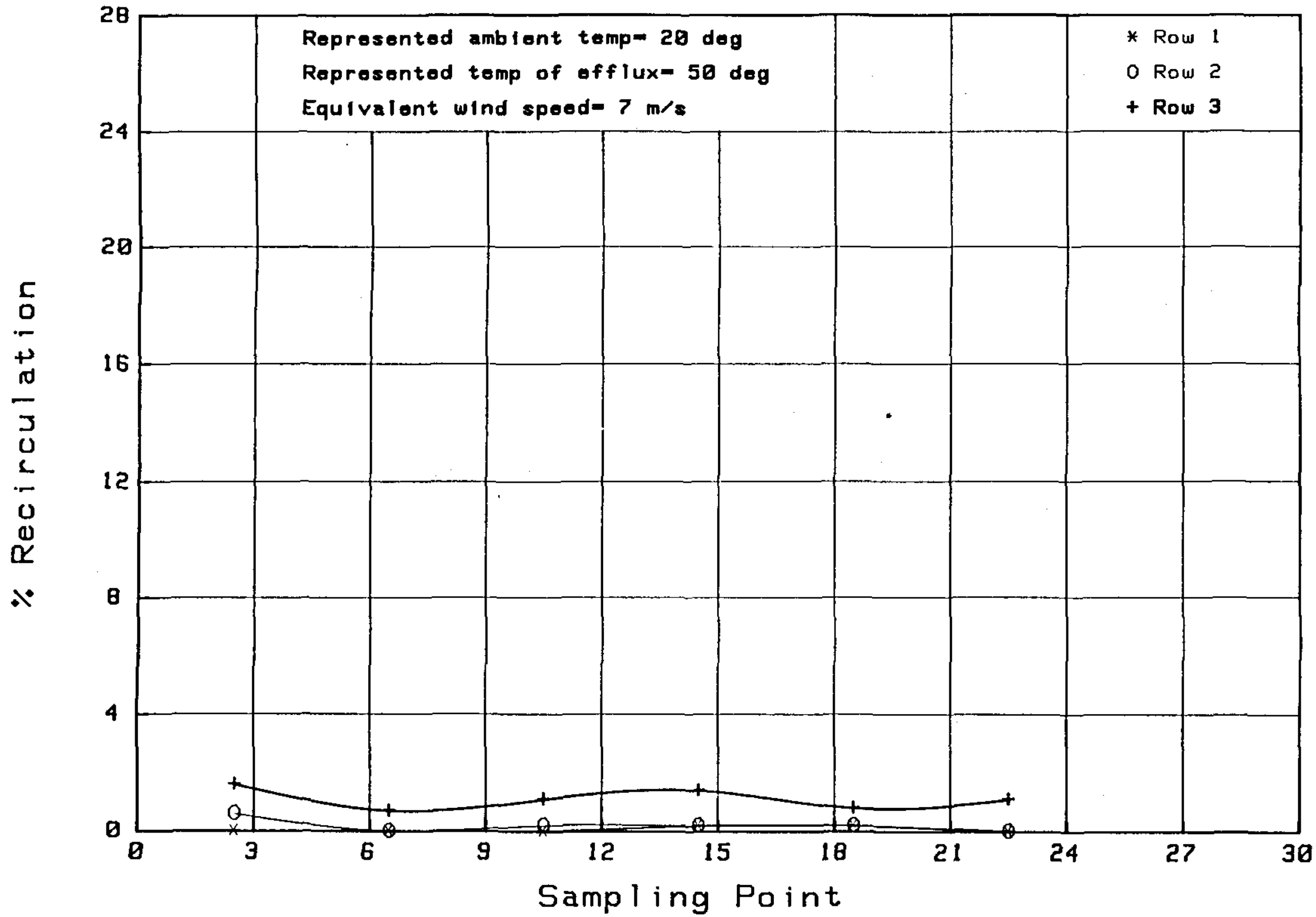

Fig 21. Hot air recirculation in heat exchanger

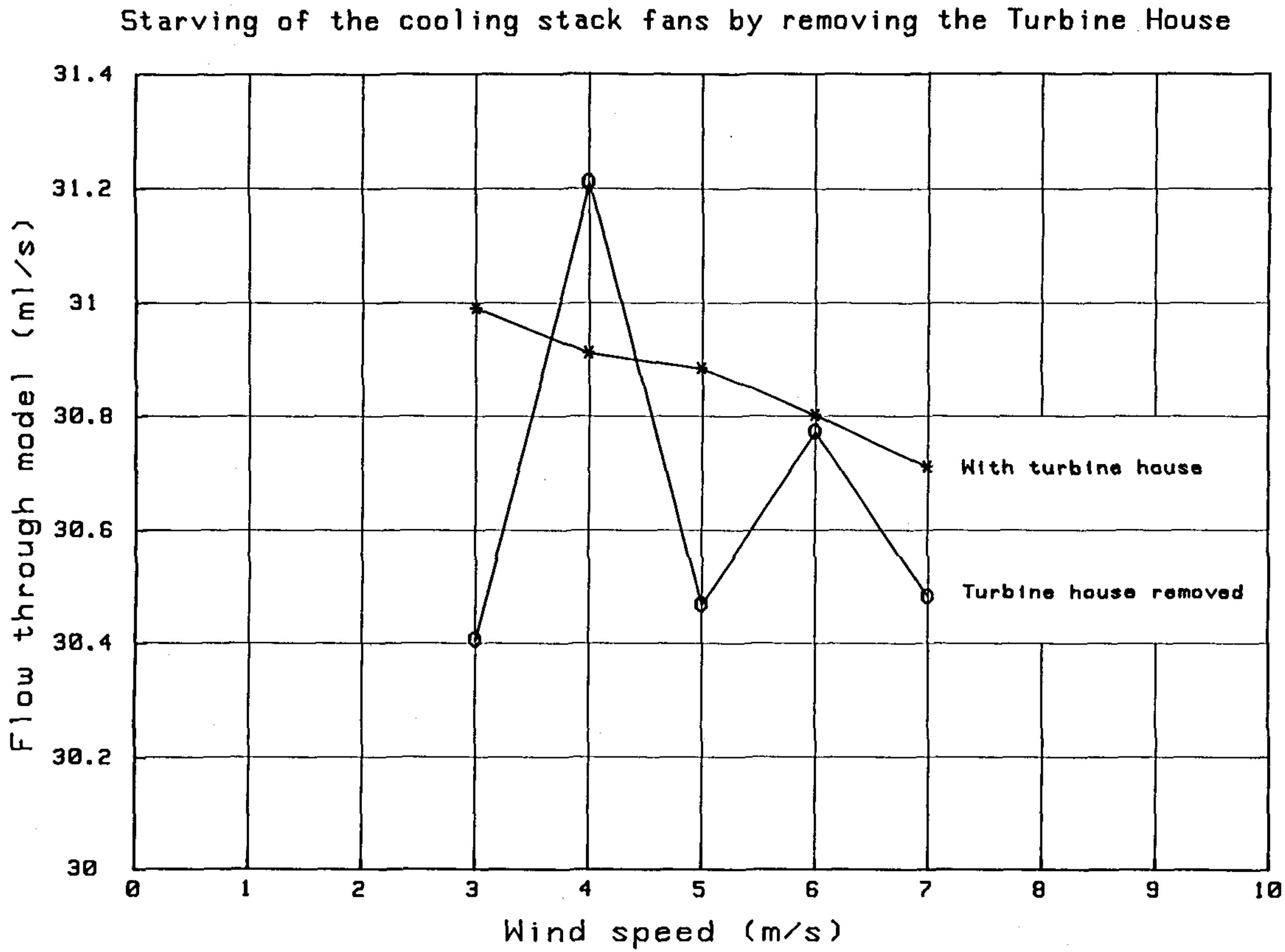

Fig 22. Wind speed vs flow through model

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The model used in a water tunnel, was a modified version of the one used for establishing recirculation effects in the Matimba Power Station (Ref.[1]), except that the model was somewhat simplified.

The purposes of the tests were:

- To determine the hot air recirculation varying the spacing between the forced draught cooling stack and the boiler/turbine houses, under various wind conditions.

- To determine if there is any evidence of "starving" the air inlet to forced draught cooling fans when wind is allowed to flow freely through the space underneath the fans.

- To establish if slight hot air recirculation takes place around the certain portions of the edges of the cooling stack even under conditions of no or slight wind. (Analytical determination of recirculation patterns (Ref. [2]) did indicate such a slight recirculation).

2 MODEL SIMILITUDE CONSIDERATIONS

For such a heat exchanger model to behave in the same way as a full-scale prototype, all force ratios in the model must equal the corresponding force ratios in the full-scale prototype. Such force ratios are analytically represented as follows:

Let
- V = a typical wind velocity.
- V_e = velocity of efflux from forced draught system.
- ρ = density of ambient air.
- ρ' = density of hot air.
- D = typical linear dimension (say the width of the bank of fans representing the forced draught heat exchanger).
- g = acceleration due to gravity.
- μ = viscosity of air.

The following are the significant forces encountered in the system while in operation:

(1) μVD = air viscosity forces.
(2) ρV^2D^2 = aerodynamic inertia forces of cold air.
(3) $\rho' V^2D^2$ = aerodynamic inertia forces in hot plume.
(4) $(\rho-\rho')gD^3$ = buoyancy forces on plume.

The ratios of these various forces determine the required non-dimensional numbers which must be identical in model and prototype to ensure complete similarity.

Putting $\frac{(2)}{(1)}$ gives $\frac{\rho VD}{\mu}$ (the Reynolds number)

Putting $\frac{(3)}{(2)}$ gives $\frac{\rho' V_e^2}{\rho V^2}$ (the specific impulse ratio)

Putting $\frac{(4)}{(2)}$ gives $\frac{\rho-\rho'}{\rho} \cdot \frac{gD}{V^2}$ (the densimetric Froude number)

If the values of the above non-dimensional numbers are identical in model and full-scale prototype, it is generally accepted that reliable results can be expected. Nevertheless, it is possible to reduce the specific impulse ratio and the densimetric Froude number to simpler components.

Since the hot air leaving the heat exchangers is at a temperature of 30° above the assumed ambient air temperature of 20°, the density ratio $\rho'/\rho = 0{,}91$ can readily be obtained by mixing alcohol with water. Thus, the derived simplified dimensional numbers ρ'/ρ (density ratio), V_e^2/V^2 or V_e/V, and gD/V^2 (Froude number) may be used for this test. It is of course impossible to simulate the Reynolds number $\rho VD/\mu$ in the model as is always the case for such small-scale tests. It is argued that the Reynolds number discrepancy will not be serious when testing a model power station which is essentially a bluff body and therefore relatively insensitive to a Reynolds number discrepancy. However, Reynolds number effects on actual plumes is an uncertain factor.

A linear scale of 1:1800 was selected for this test and from the Froude number it follows that the velocity scale, i.e. the ratio of full-scale velocity to model velocity is $\sqrt{1800}$, i.e. 42,4. The full scale velocity efflux from the top of the heat exchangers was assumed to be 5 m/s.

3 DESCRIPTION OF MODEL, INSTRUMENTATION AND TESTING PROCEDURE

3.1 The Model

The heat exchanger model was essentially the same system as the one previously used for the Matimba (Ref. [1]) investigation.

The heat exchanger system of Matimba power station consists of an array of 72 pillars supporting heat exchangers through which an array of 288 fans force air.

Inspection of the Matimba heat exchanger system indicated that the fans sucked air up from the lower surface of the system and discharged this air into the voids beneath the heat exchangers which simulated a discharge into a plenum chamber. From this plenum chamber, the air flowed upwards uniformly through the heat exchanger fins.

In the model, fans were therefore not required, but it was necessary to provide a system where the water could be sucked in at the bottom of the bank of simulated heat exchangers, while a different fluid (in this instance, alcohol and water) representing the less dense efflux, could be uniformly ejected out of the top surface of the heat exchanger. This heat exchanger model, illustrated in Figure 1, was constructed as follows: Each pillar, represented by a thin tube, was used to suck water away from the base of the model heat exchanger through small holes drilled in the base (see Figure 2). Thus, the inflow at the base could be simulated by allowing all the small tubes connected to the 72 pipes to discharge in the tunnel pit at a depth calibrated to give the correct flow rate for the four fans represented by the small holes surrounding each pillar.

The upper compartment of the model heat exchanger was completely separated from the lower compartment from which the water was sucked out. The upper compartment which was covered by a fine gauze contained tubes with suitable holes in the walls to ensure equally distributed efflux of the water/alcohol mixture, as is indicated in Figure 2. This water/alcohol mixture was supplied from header tanks for which the elevation was calibrated so as to give the required rate of efflux from the model heat exchangers.

The wind was simulated as a flow with a conventional vertical wind profile obtained by fitting "spires" upstream of the model and also small regularly spaced blocks in the model floor upstream of the model so as to retard flow on ground level. Only a wind direction normal to the longitudinal axis of the heat exchanger was simulated. The models of chimney stacks and all structures upwind of boiler houses were omitted and only the boiler and turbine houses were included as depicted in Figure 1.

3.2 Instrumentation and Testing Procedure

One of the advantages of water-tunnel testing is the capability of making quantitative measurements of plume dispersion and recirculation. In carrying out these quantitative measurements of recirculation taking place from the upper surface of the heat exchanger system to its lower inlet surface, the Fluorometer system used for previous investigations (Ref. [1]) was again employed. A known mass of quinine sulphate was introduced into the mixture of water and alcohol which represented the efflux plume.

For any particular test where the tunnel speed was adjusted to represent a given wind speed and where the exit plume had become stable, six samples, each coming from the six suction inlets in the model heat exchanger, were taken in small flasks. Altogether the 72 pillars in the model each sucked fluid away from the underside of the heat exchanger but 12 such suction tubes were combined thus giving six sampling points.

A Fluorometer, which can measure the quinine sulphate content in parts per million (ppm) of any sample taken anywhere in the flow field, was then used to determine the concentration of quinine sulphate at each of the six outlets. In this way, the percentage recirculation presented at any of the six sampling points could readily be determined.

The boiler houses and turbine houses could be moved away from the heat exchanger array from a position abutting the heat exchanger to gaps represented respectively by $^1/_3$ of the heat exchanger height, the full height of the heat exchanger and ultimately to a position where the boiler and turbine houses were removed and the heat exchanger was exposed to unrestricted wind. Percentage recirculation was measured for each gap setting. The test where it was required to establish whether fans were "starved" when simulated wind was allowed to flow freely underneath the heat exchanger, flow rates were simply measured by measuring the volume of fluid sucked from underneath the heat exchanger in a given time lapse. To check if any fans were "starved" a controlled experiment was made where the flow of wind underneath the heat exchanger was blocked off completely by the model boiler and turbine houses.

4 RESULTS AND DISCUSSION

4.1 The Effect on Hot Air Recirculation of the Gap Width between Turbine Housing and Heat Exchanger

Figure 3 gives perhaps the most significant illustration of the effect of a gap between the heat exchanger and the boiler housing. The graphs in this figure depicts total percentage recirculation in the whole heat exchanger plotted against wind speed. It can be seen that the percentage recirculation of the efflux alcohol plume representing hot air is drastically reduced by increasing the gap between the boiler housing and heat exchanger from zero to $^1/_3$ of the heat exchanger height. A further increase in this gap sized to one heat exchanger height hardly has any further circulation reducing effect. The rather hypothetical case where the heat exchanger is isolated from the boiler housing, allowing wind to flow unrestricted underneath the heat exchanger, is clearly the most advantageous setup. Figure 3 shows that even at rather high wind speeds of 7 m/s only slight recirculation takes place.

Figures 4 to 7 show graphs of percentage recirculation versus wind speed for each of the three pillar rows (refer to Figure 2). These graphs indicate in what regions most recirculation occurs.

Figures 8 to 21 show the distribution of the hot air recirculation along each of the three rows of pillars, the position of each pillar representing four fans. As can be seen on the graphs only six sampling points have been taken along each row of 24 pillars, which means that the fluid from the suction tubes of four pillars in each row has been combined and has been treated as one sample (see Figure 1). It is felt that six such sampling points in each row adequately illustrates the distribution of hot air recirculation. Only the two higher wind speeds are considered for the case where the turbine house is removed since lower wind speeds indicated zero recirculation.

4.2 The Possible "Starving" Effect on the Supply of Air to Fans when Wind is allowed to flow crosswise underneath the Fan Inlets

Although the water tunnel model of the heat exchanger does not faithfully represent the fans and fan inlets underneath the heat exchanger body it is argued that any decrease in pressure, owing to wind, which may starve full scale fans, will also be present when wind flow is simulated underneath the model heat exchanger. Such a decrease in pressure should then also starve the flow which is sucked out via the perforations in the lower surface of the model heat exchanger.

Extensive model tests with the model boiler - turbine housing in position against the heat exchanger thus presenting no wind flow and with the housing removed, thus allowing unobstructed wind flow, were carried out.

Table I gives the measured rates of flow of water sucked up in each pillar row at various wind speeds, both for unobstructed crossflow as well as crossflow blocked off by the presence of model boiler-turbine housing.

Table I

WIND-SPEED	BOILER-TURBINE HOUSING *REMOVED. MODEL* VOLUMETRIC FLOW IN $m\ell/s$			BOILER-TURBINE HOUSING *PRESENT. MODEL* VOLUMETRIC FLOW IN $m\ell/s$		
	ROW 1	ROW 2	ROW 3	ROW 1	ROW 2	ROW 3
3	10,121	10,282	10,002	10,322	10,493	10,174
4	10,333	10,557	10,320	10,410	10,351	10,171
5	10,090	10,228	10,149	10,314	10,408	10,163
6	10,285	10,389	10,098	10,264	10,440	10,097
7	10,116	10,353	10,012	10,289	10,284	10,137

Figure 22 shows the total flow into the lower surface of the model heat exchanger for both cases of boiler-turbine housing present and removed, this inflow being plotted versus wind speed.

Some scatter in the results are present but a slight starving of inflow underneath the heat exchanger, in some cases in the order of 1%, is evident.

4.3 Recirculation of Hot Air at Zero or Very Low Wind Velocities

Contrary to indications from theoretical analysis (Ref. [2]) no clear evidence could be found of recirculation taking place around the edges of the heat exchanger at no or very light wind velocities. In the present test series as well as in the test series reported in part I of Ref. [1], such recirculation at very low wind speeds (3 m/s or less) could not be detected.

5 CONCLUSIONS AND RECOMMENDATIONS

It was clearly shown that a gap $^1/_3$ times the height of the heat exchanger stack drastically reduced recirculation and that an increase in the gap size equal to the height of the heat exchanger stack did not further reduce the recirculation. It may be concluded that small gaps are beneficial and incorporating such gaps is recommended.

It is thought that the "starving" of air to fans when a free flow of wind underneath the heat exchanger tanks is allowed, is only slight and the gain in heat exchanger effectiveness, by reducing the hot air recirculation, outweighs any bad effects due to "starving".

REFERENCES

[1] van der Elst, W.J. : "Water tunnel tests on a forced-draught cooling system for a power station", Parts I, II and III, (Contract reports, ME1966/9, Nov. 1986, ME 2035, May 1987 and ME 2029/4, Nov. 1987).

[2] Kröger, D.G. : "Numerical predictions of air flow about air-cooled heat exchangers", Under preparation.

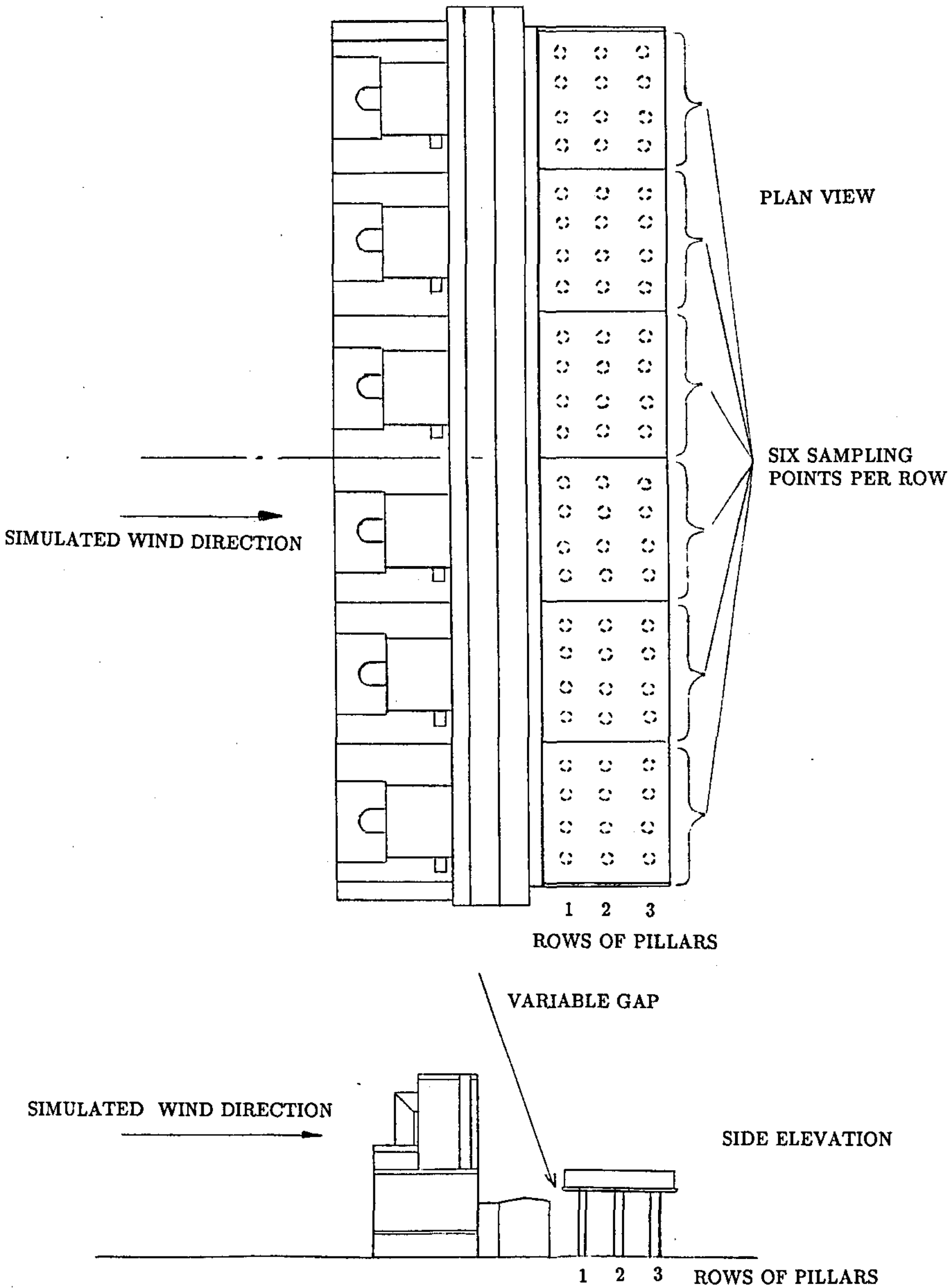

FIGURE 1: LAYOUT OF POWER STATION MODEL

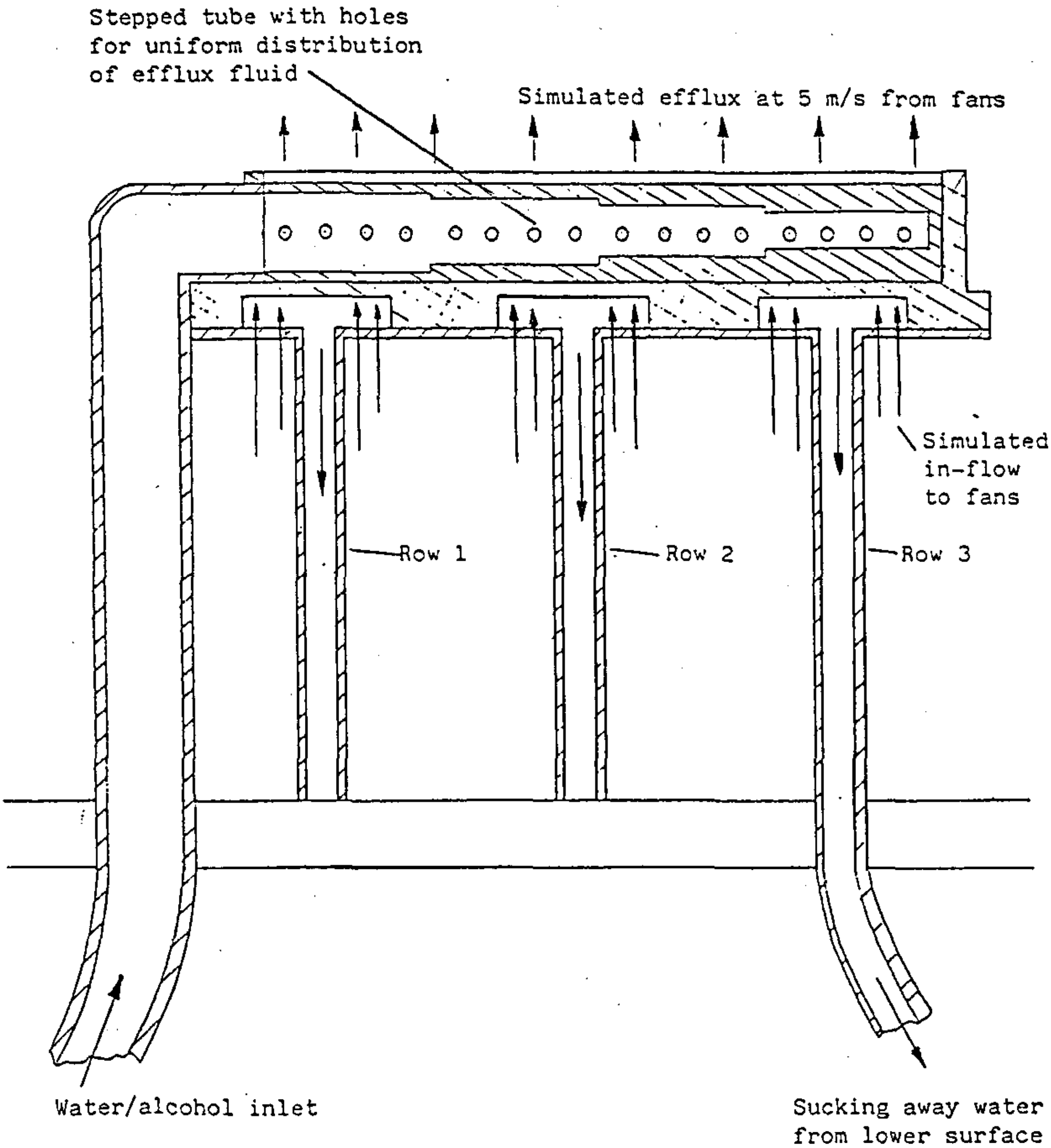

FIG. 2: CROSS-SECTIONAL SIDE ELEVATION OF MODEL HEAT EXCHANGER

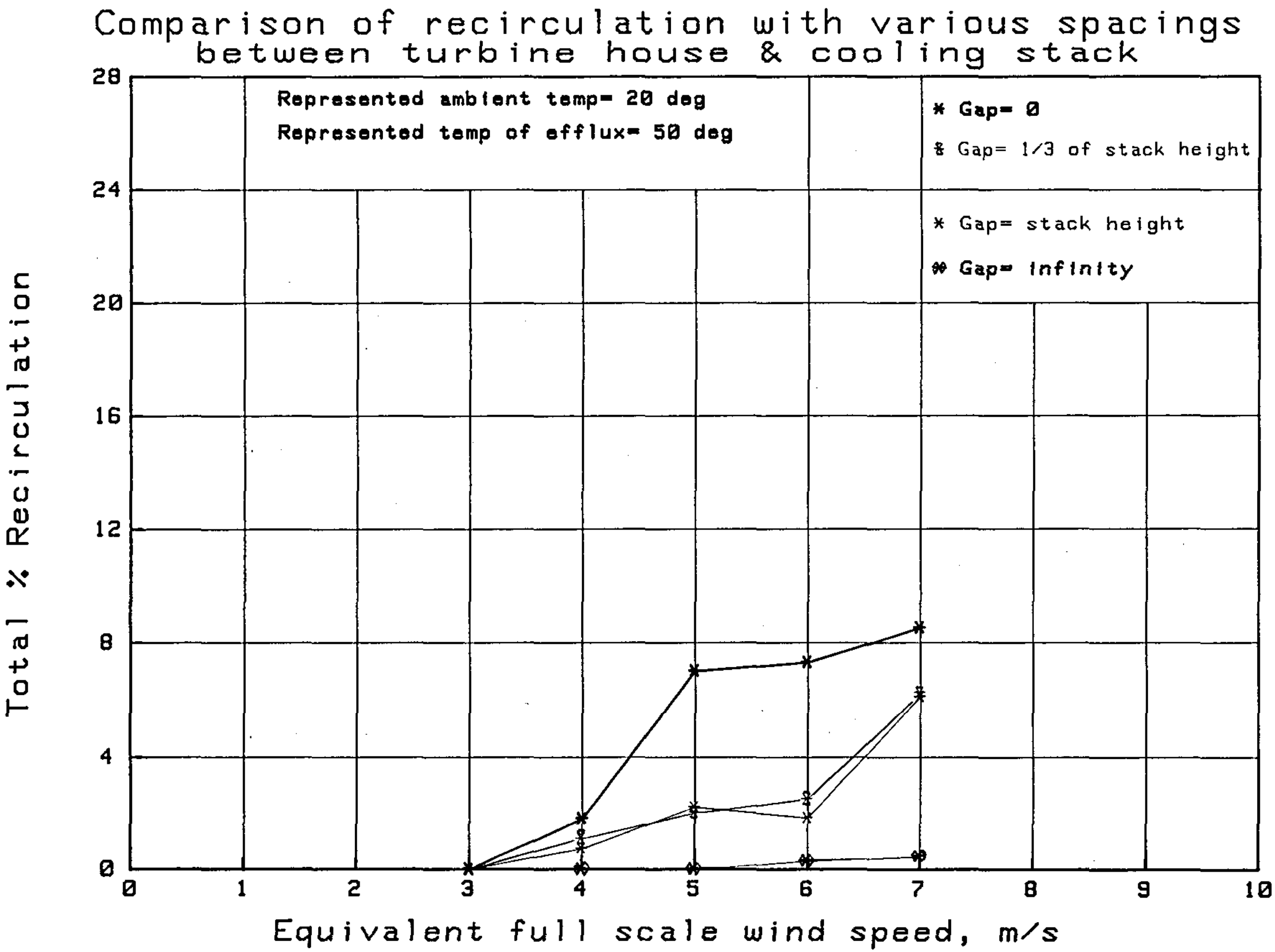

Fig 3. Recirculation showing effect of wind speed

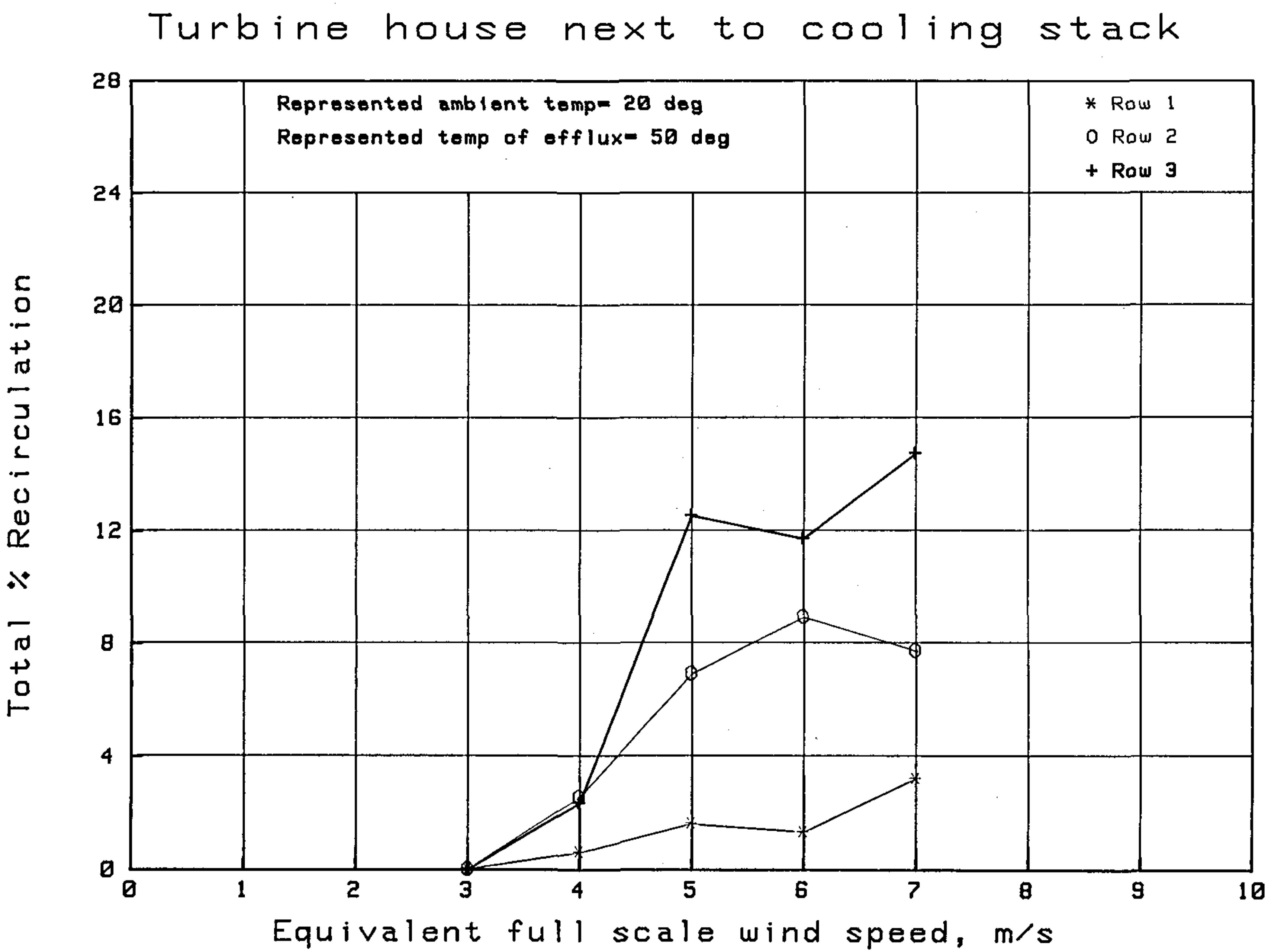

Fig 4. Recirculation showing effect of wind speed

Turbine house 1/3 of stack height away

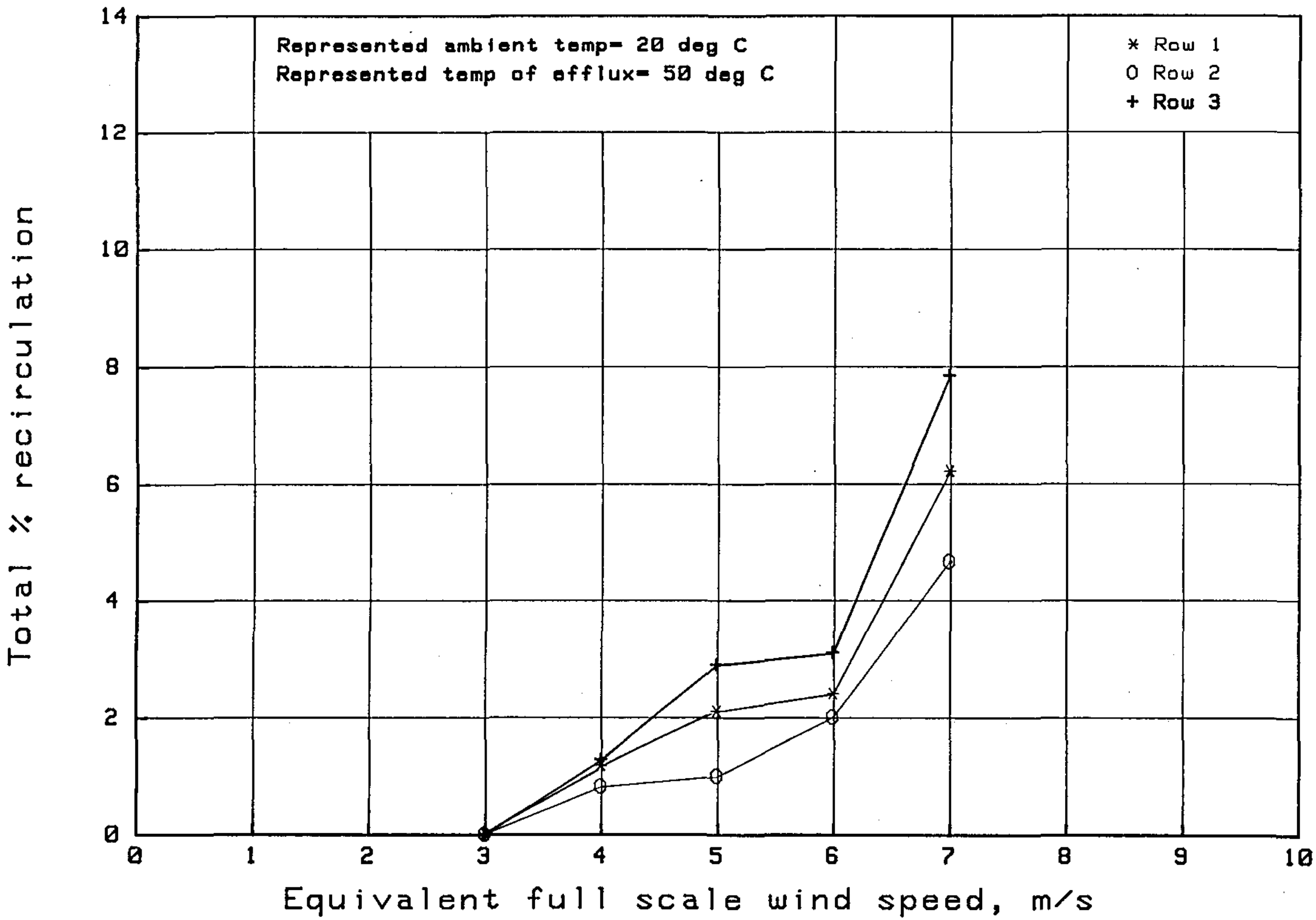

Fig 5. Recirculation showing effect of wind velocity

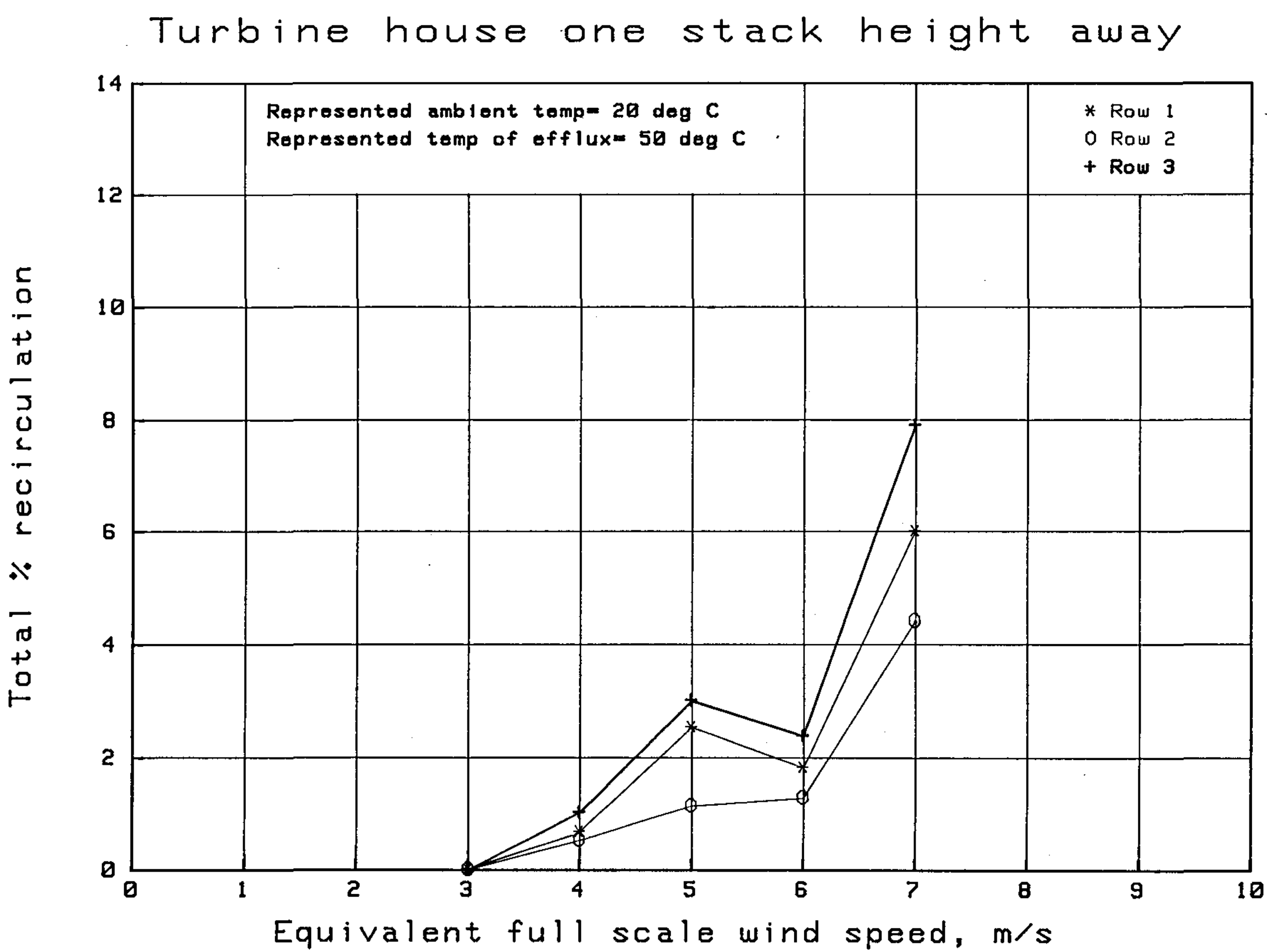

Fig 6. Recirculation showing effect of wind velocity

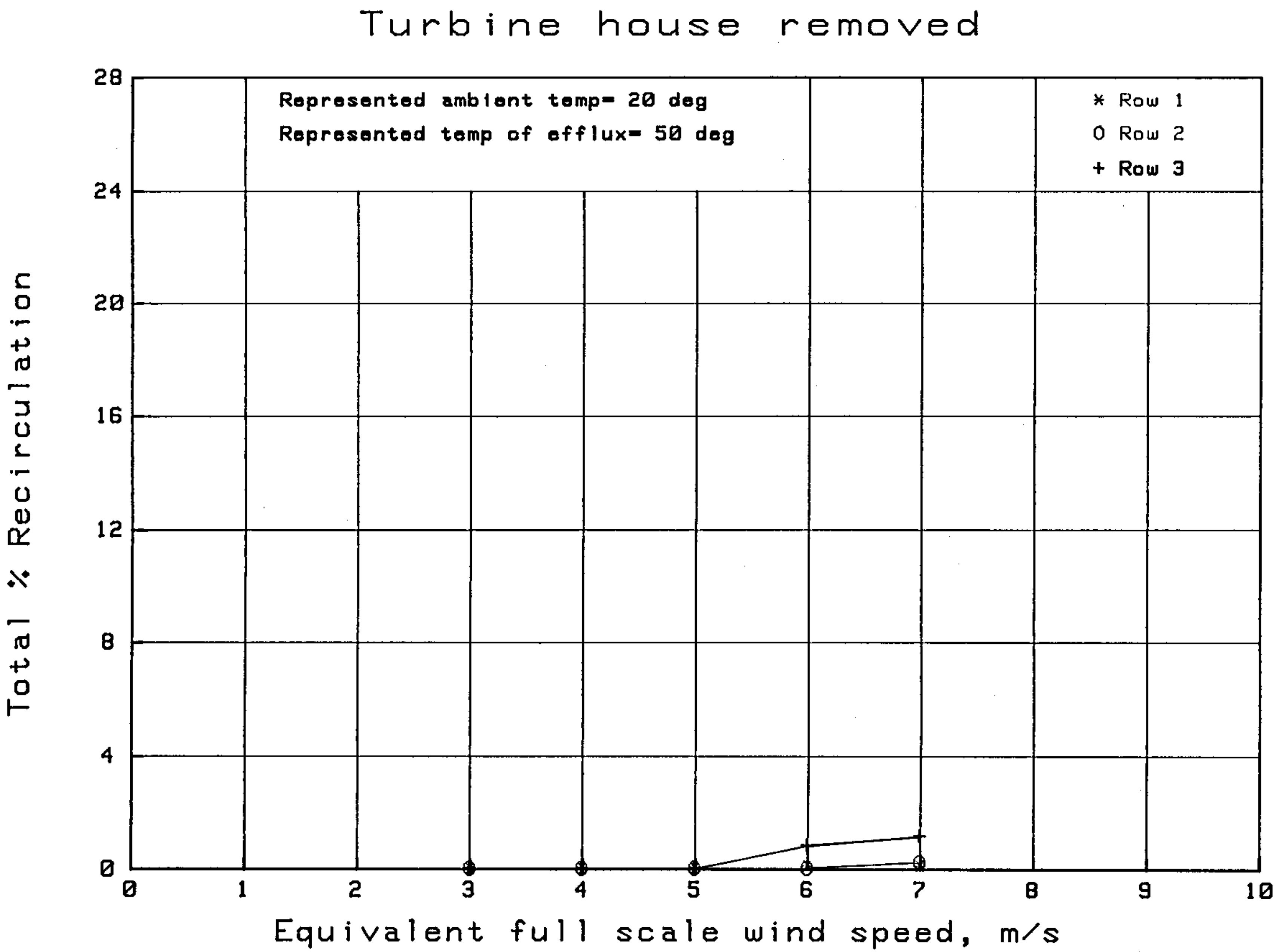

Fig 7. Recirculation showing effect of wind speed

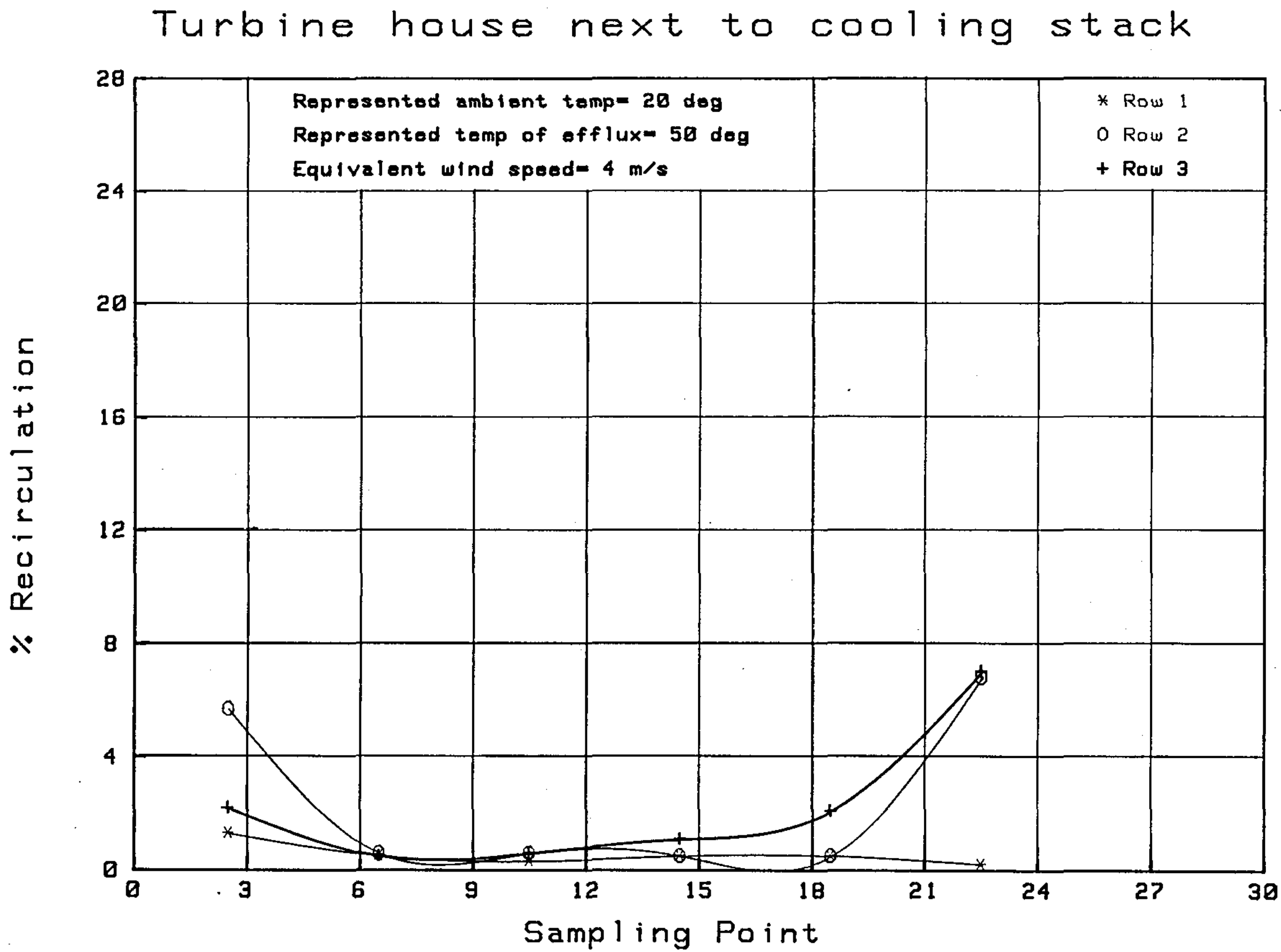

Fig 8. Hot air recirculation in heat exchanger

Turbine house next to cooling stack

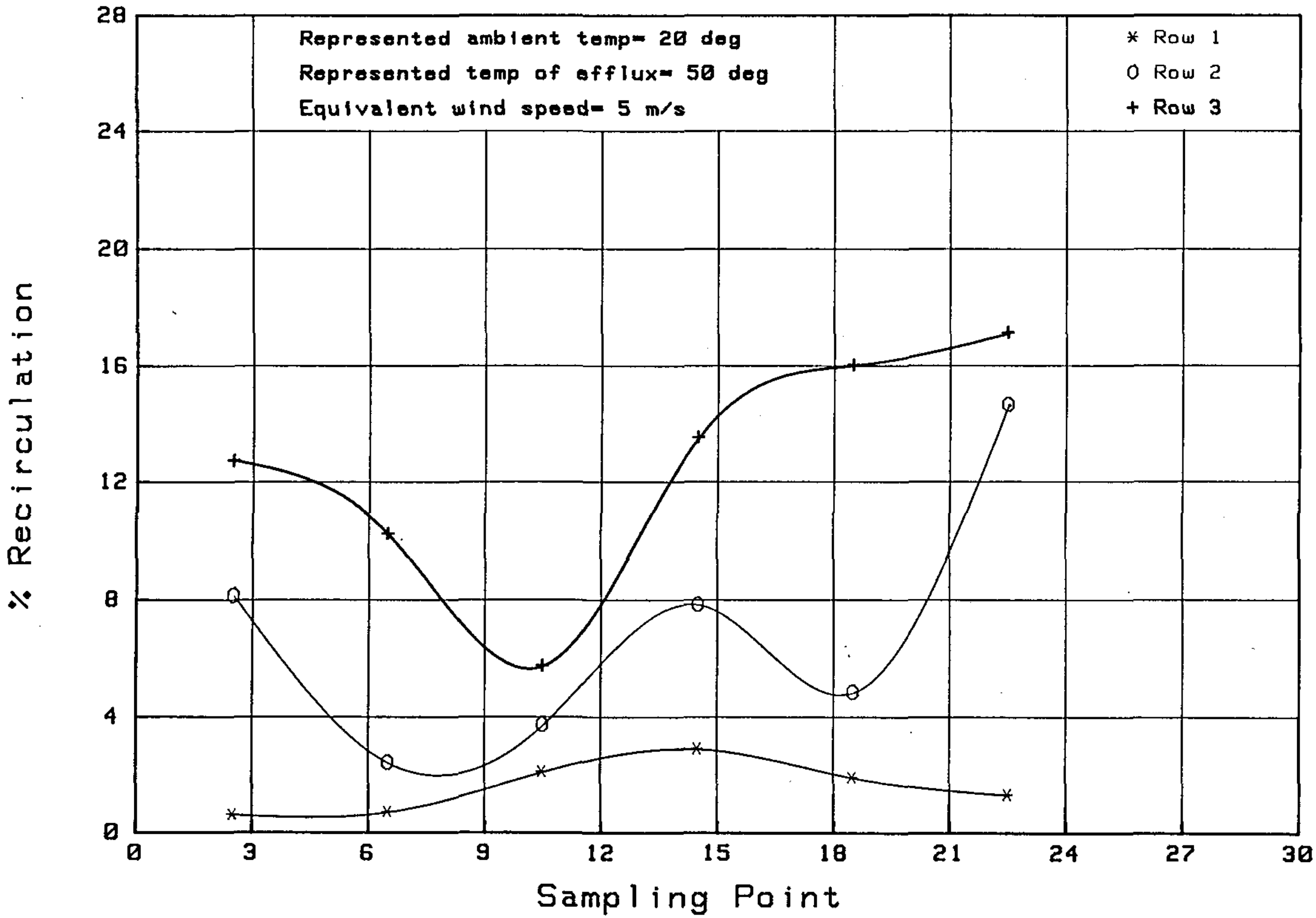

Fig 9. Hot air recirculation in heat exchanger

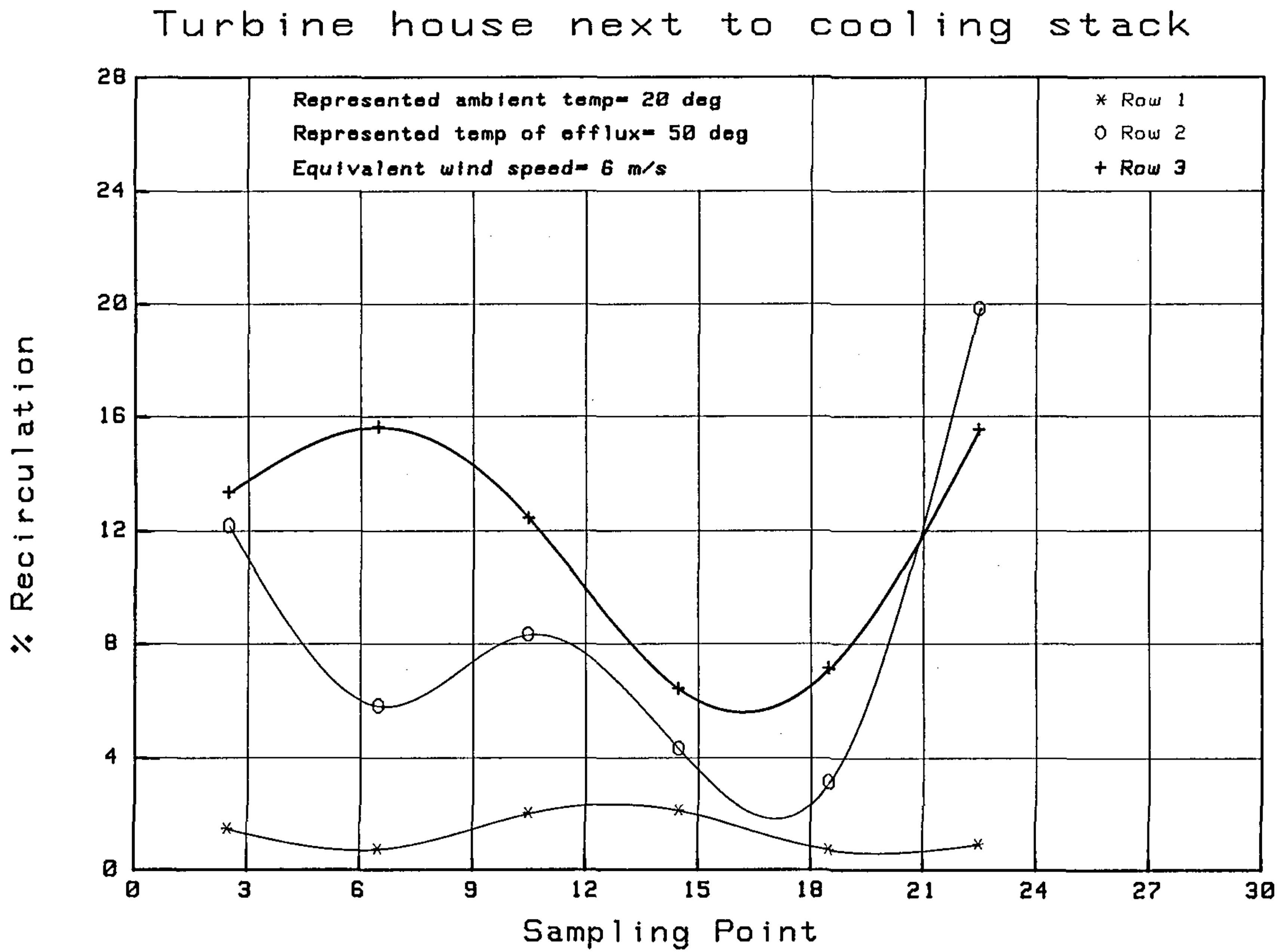

Fig 10. Hot air recirculation in heat exchanger

Turbine house next to cooling stack

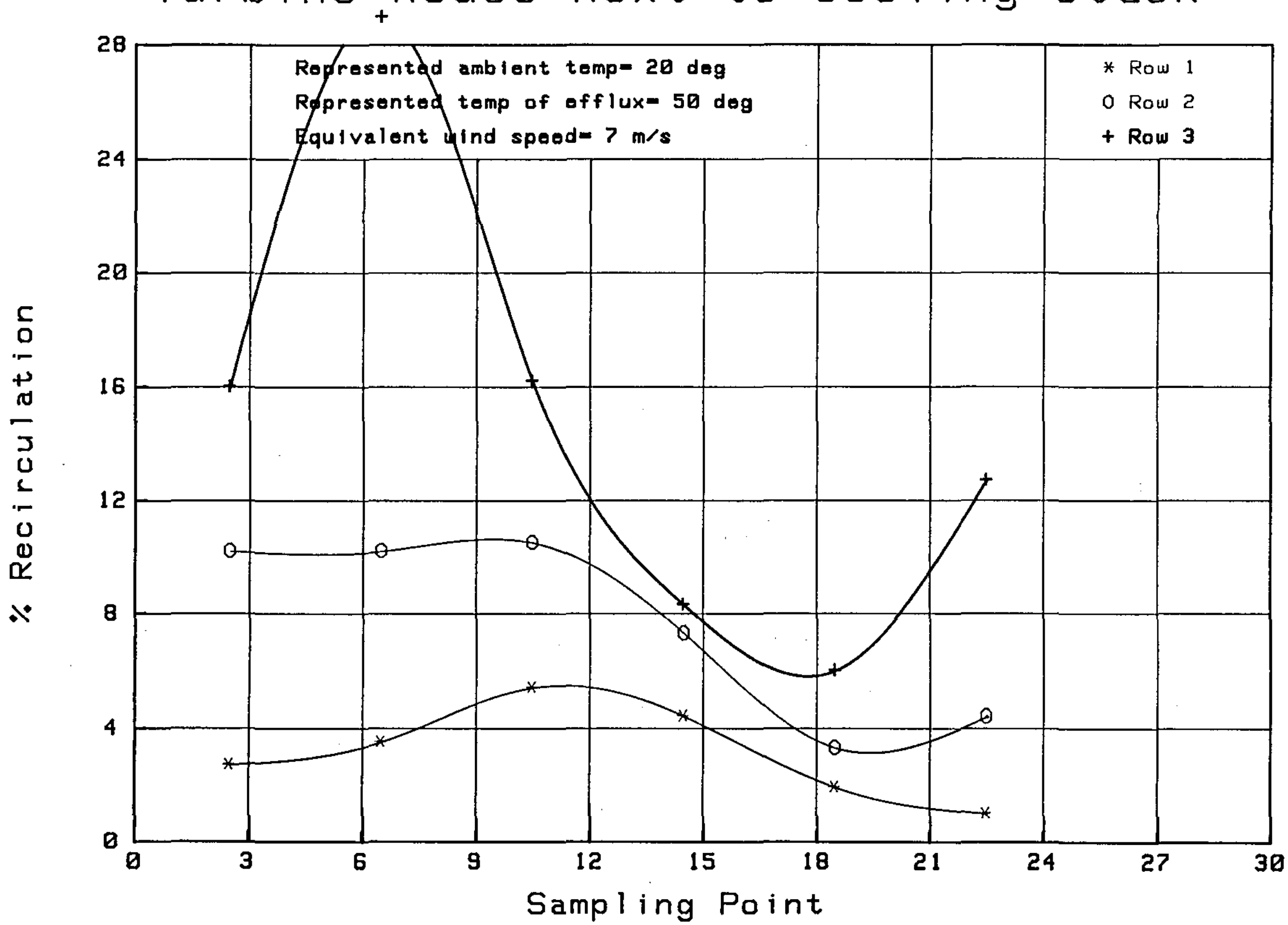

Fig 11. Hot air recirculation in heat exchanger

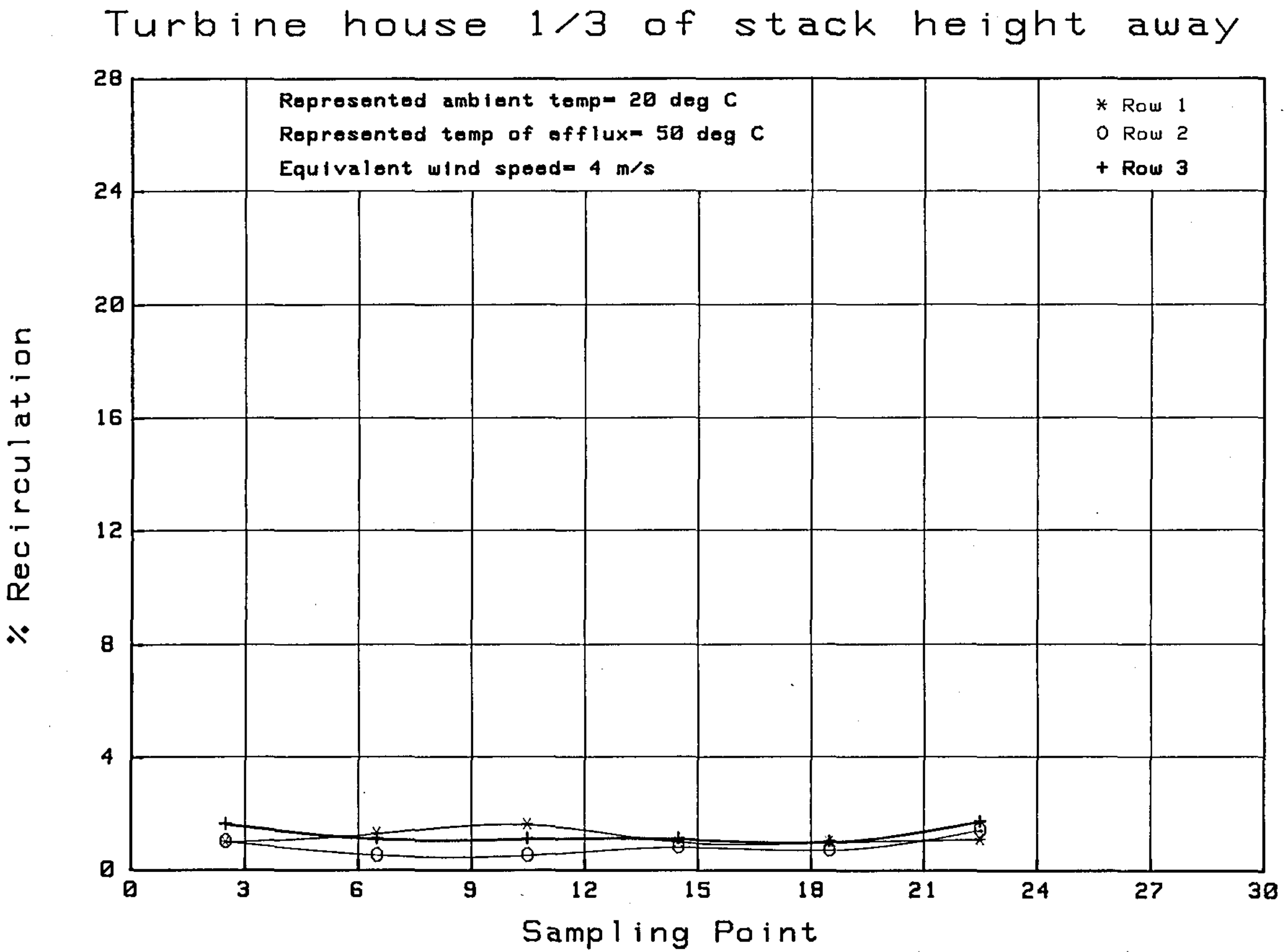

Fig 12. Hot air recirculation in heat exchanger

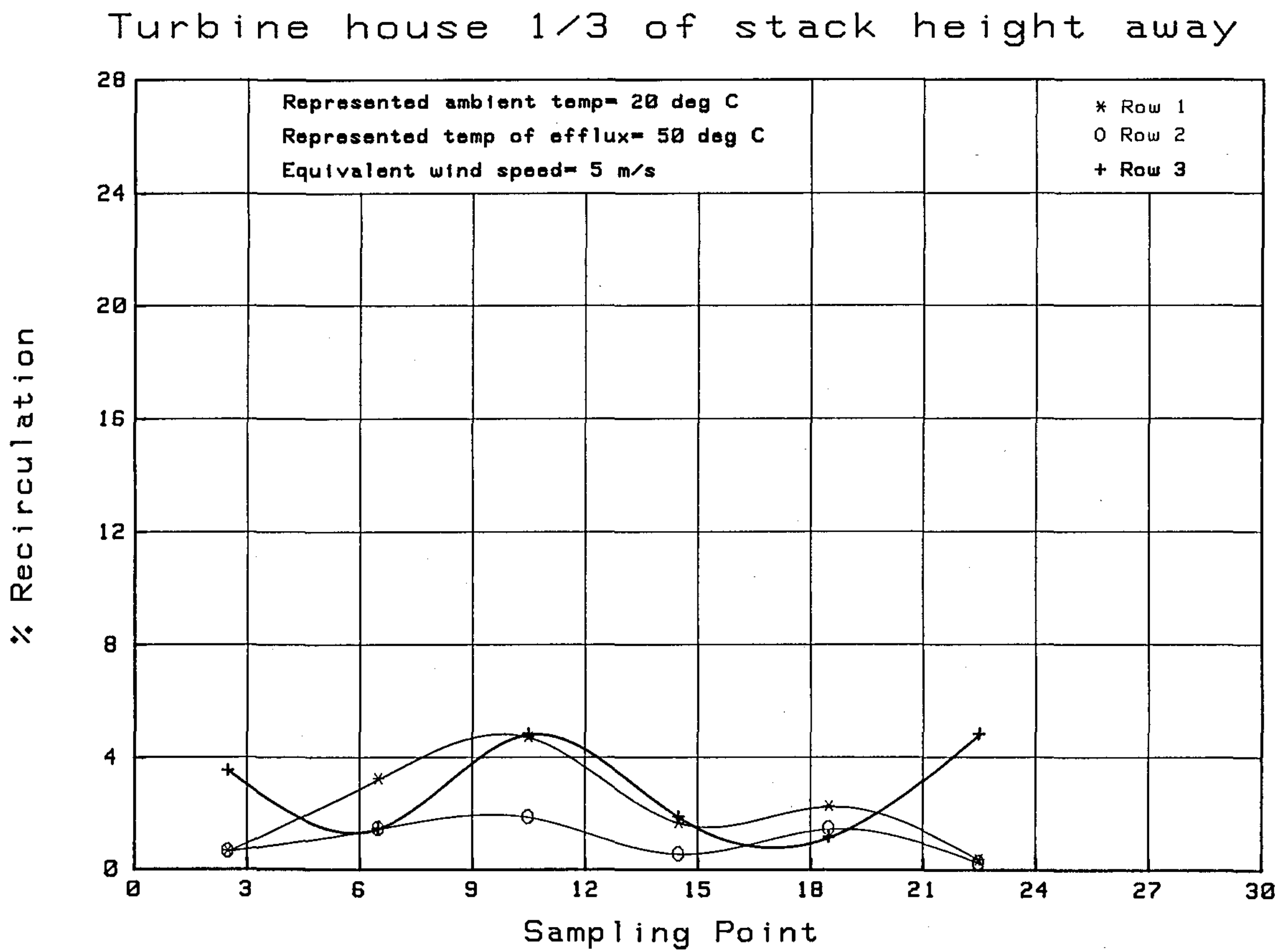

Fig 13. Hot air recirculation in heat exchanger

Turbine house 1/3 of stack height away

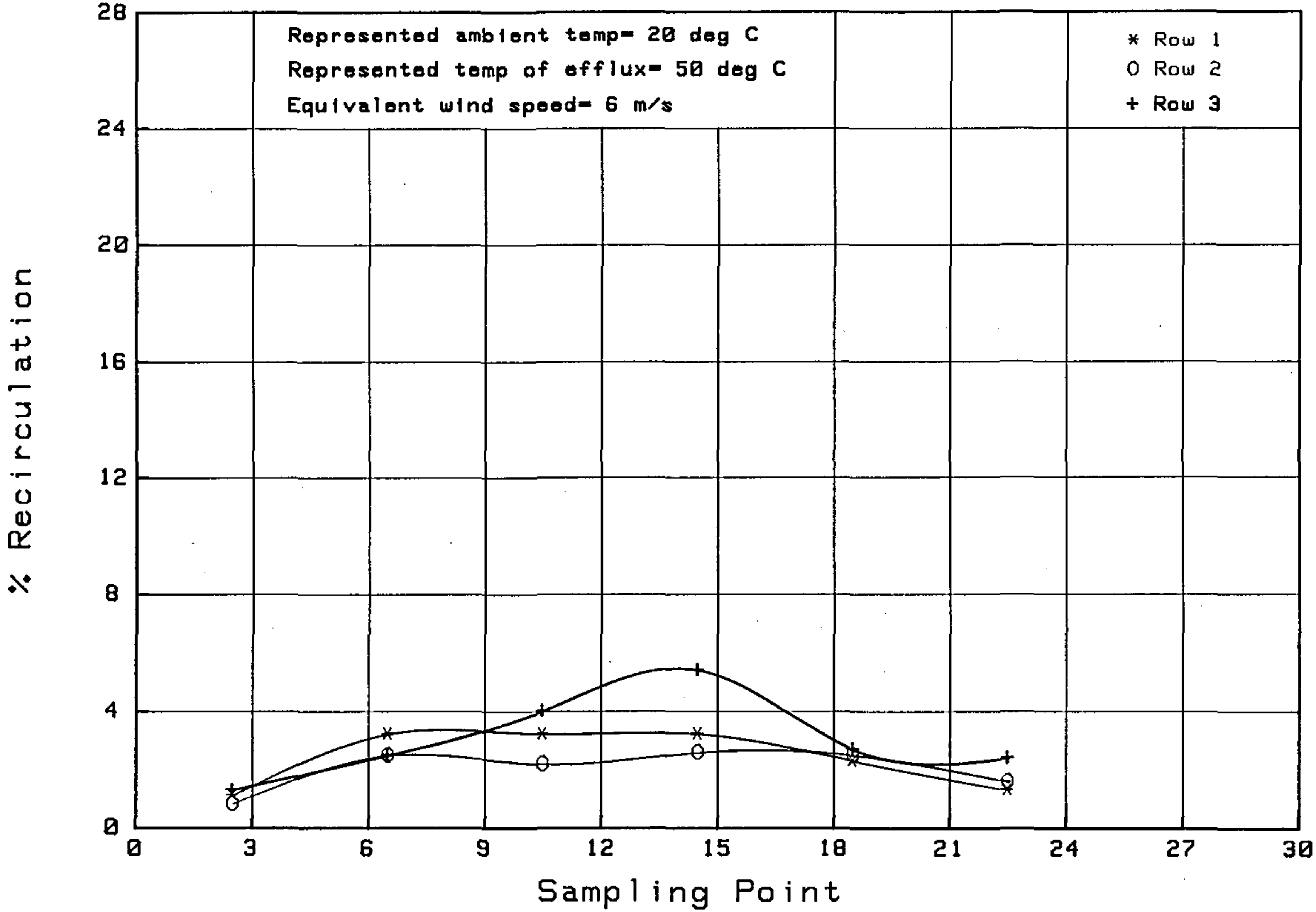

Fig 14. Hot air recirculation in heat exchanger

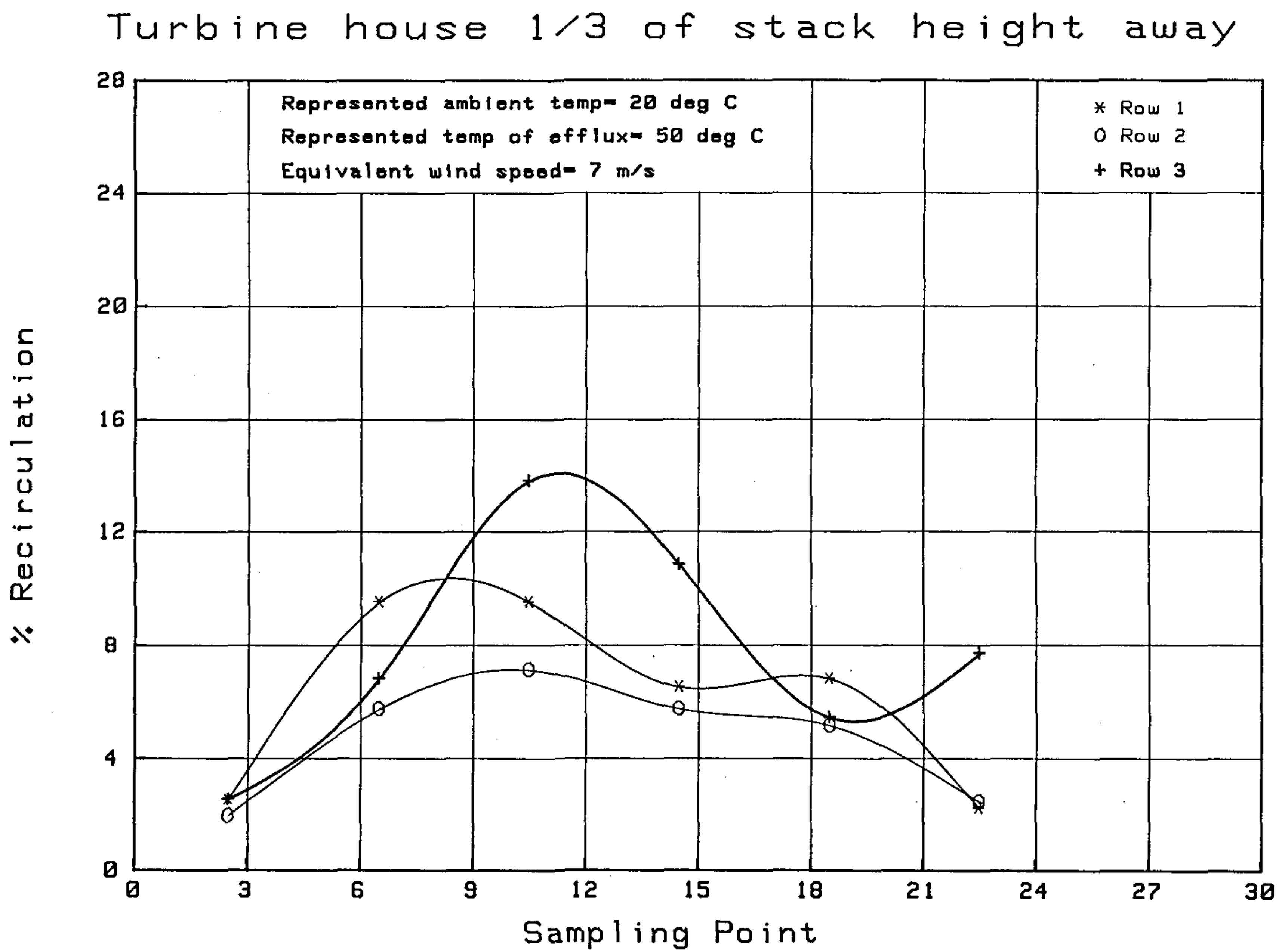

Fig 15. Hot air recirculation in heat exchanger

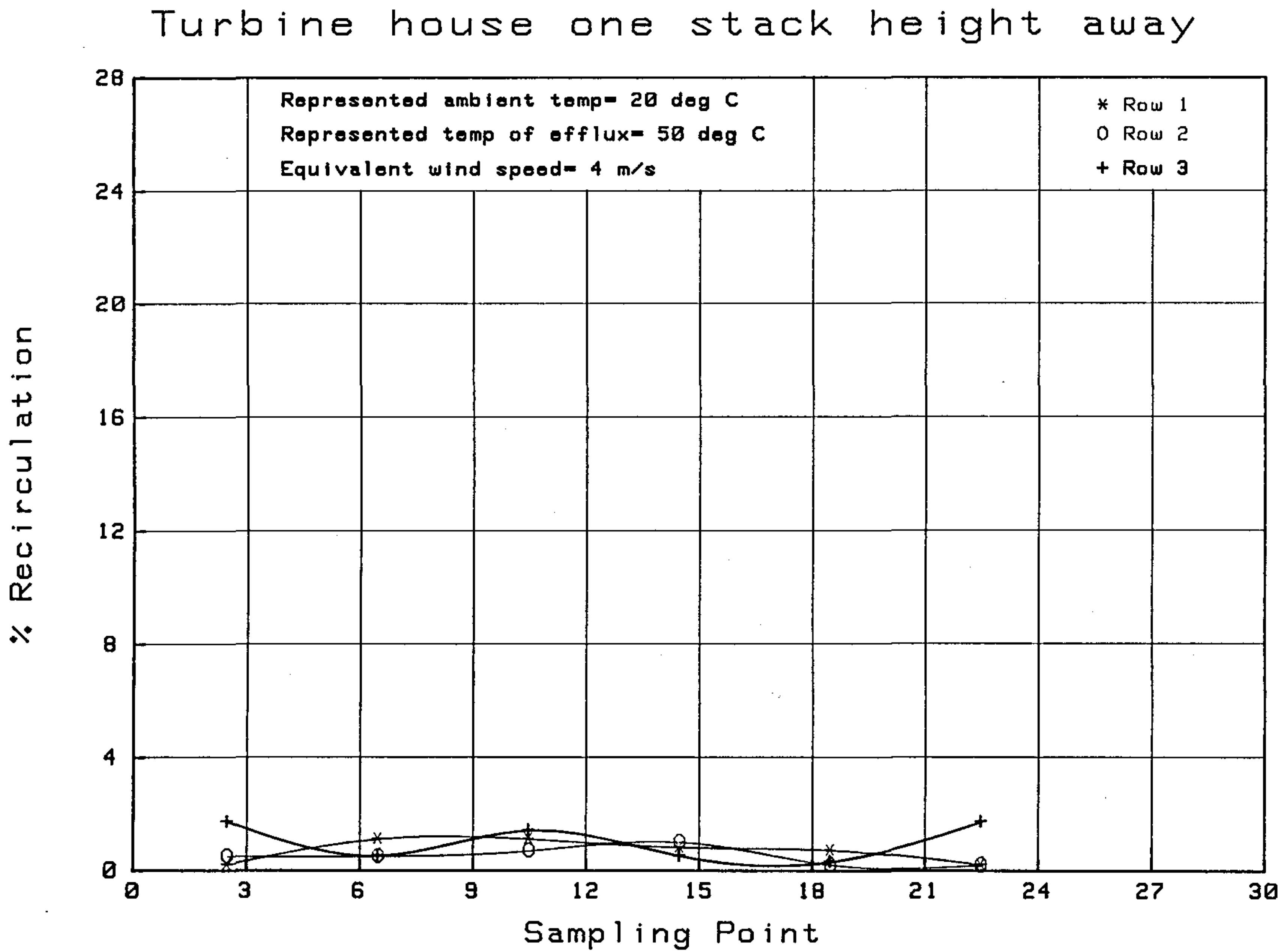

Fig 16. Hot air recirculation in heat exchanger

Turbine house one stack height away

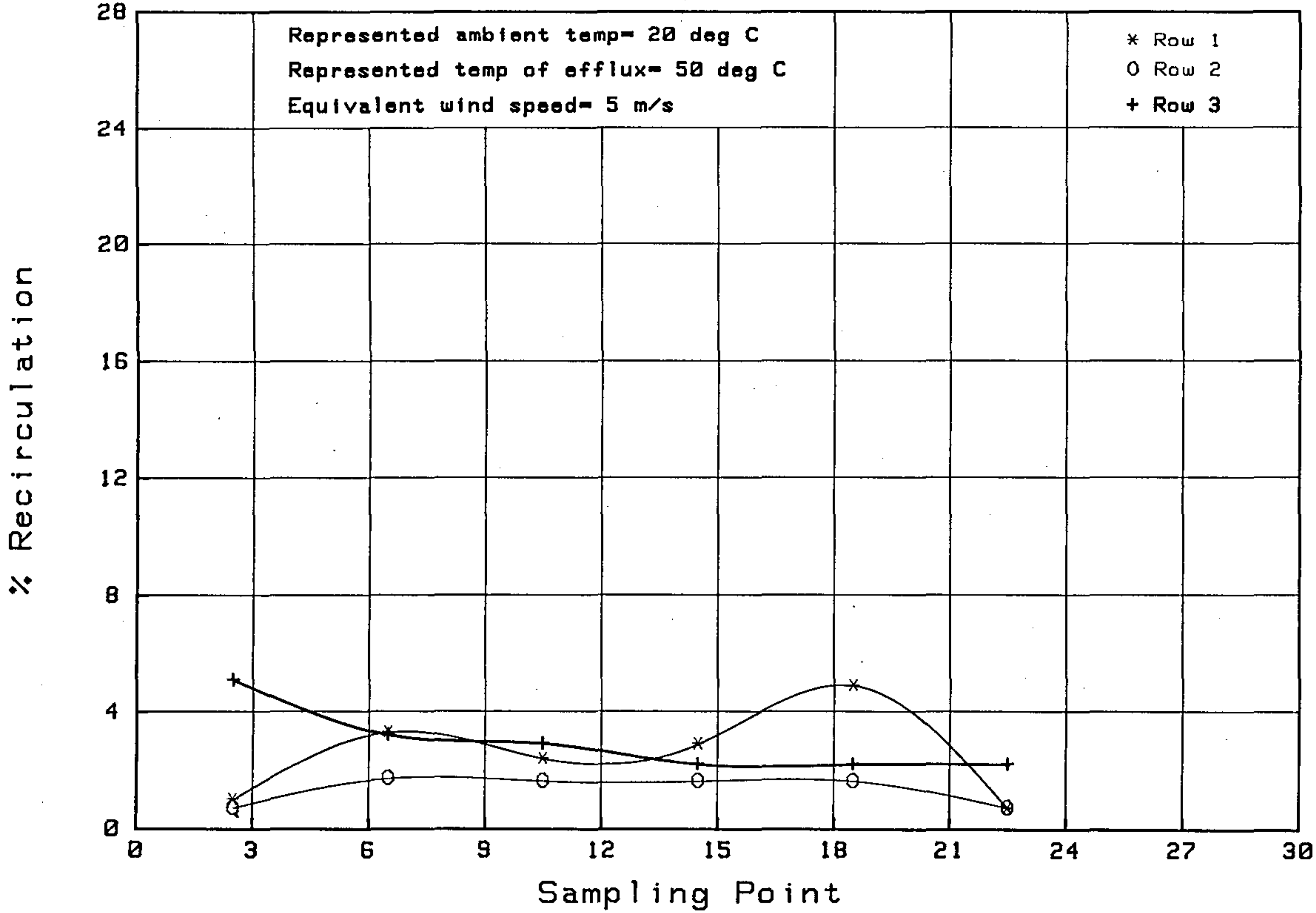

Fig 17. Hot air recirculation in heat exchanger

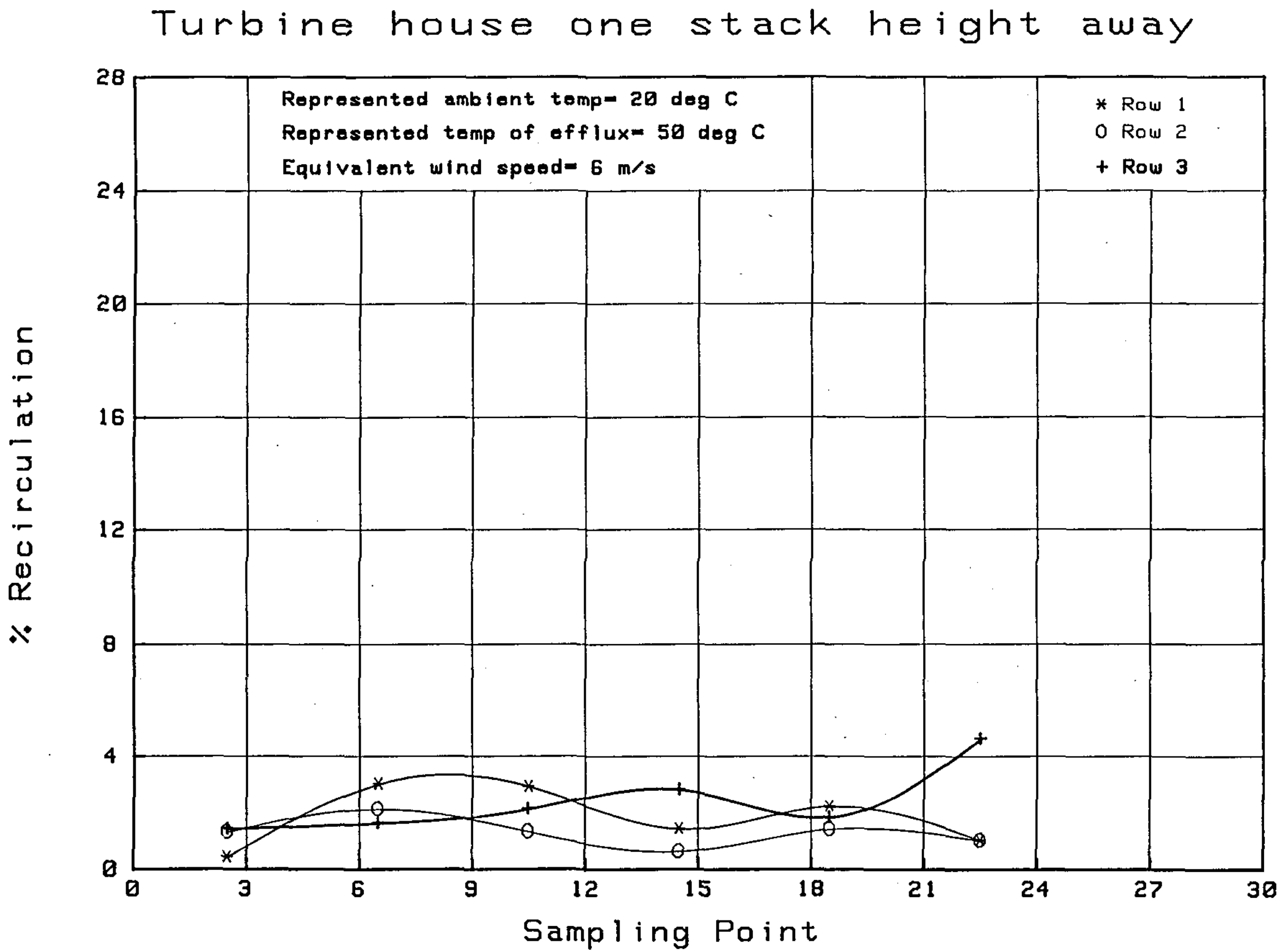

Fig 18. Hot air recirculation in heat exchanger

Turbine house one stack height away

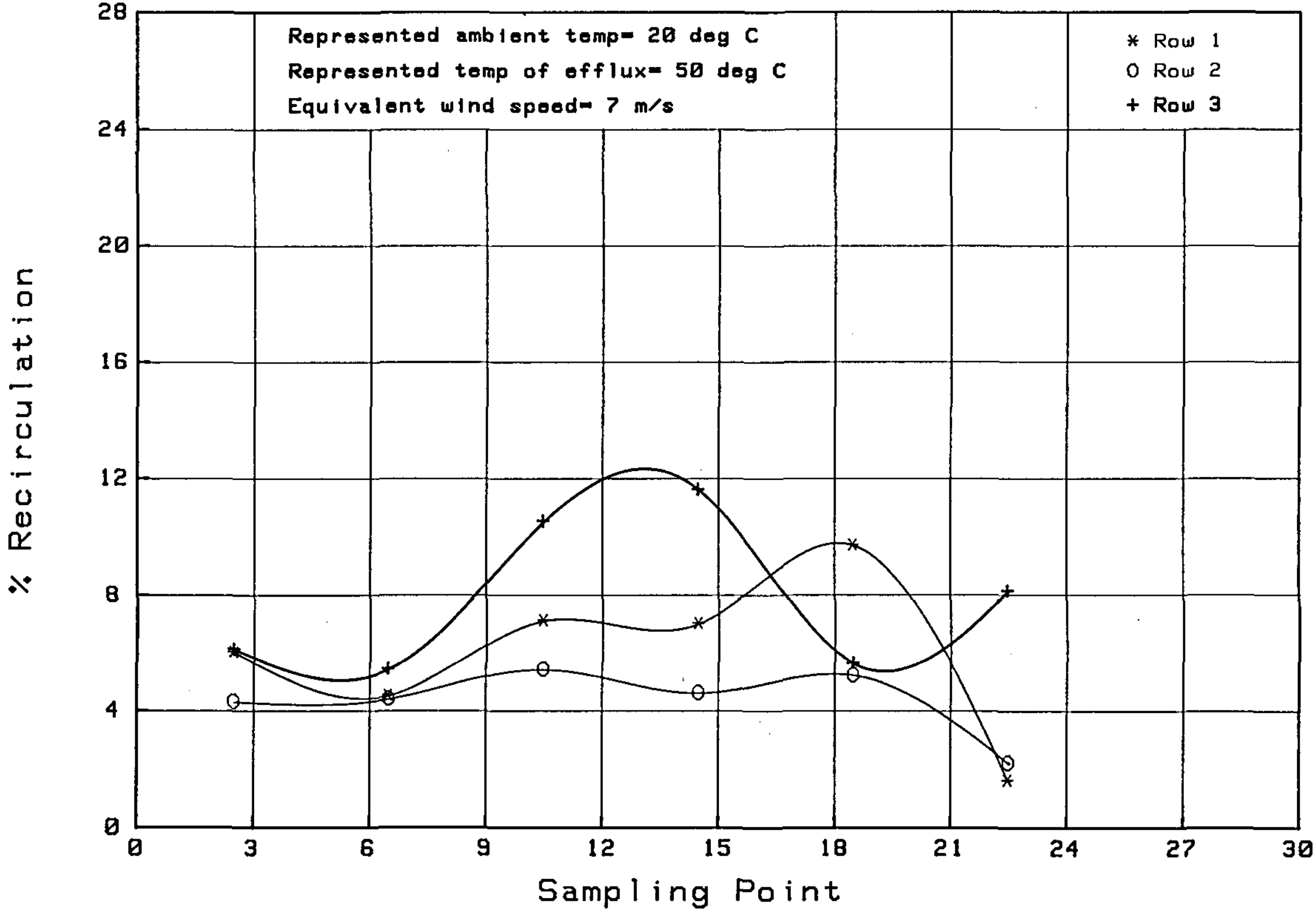

Fig 19. Hot air recirculation in heat exchanger

Turbine house removed

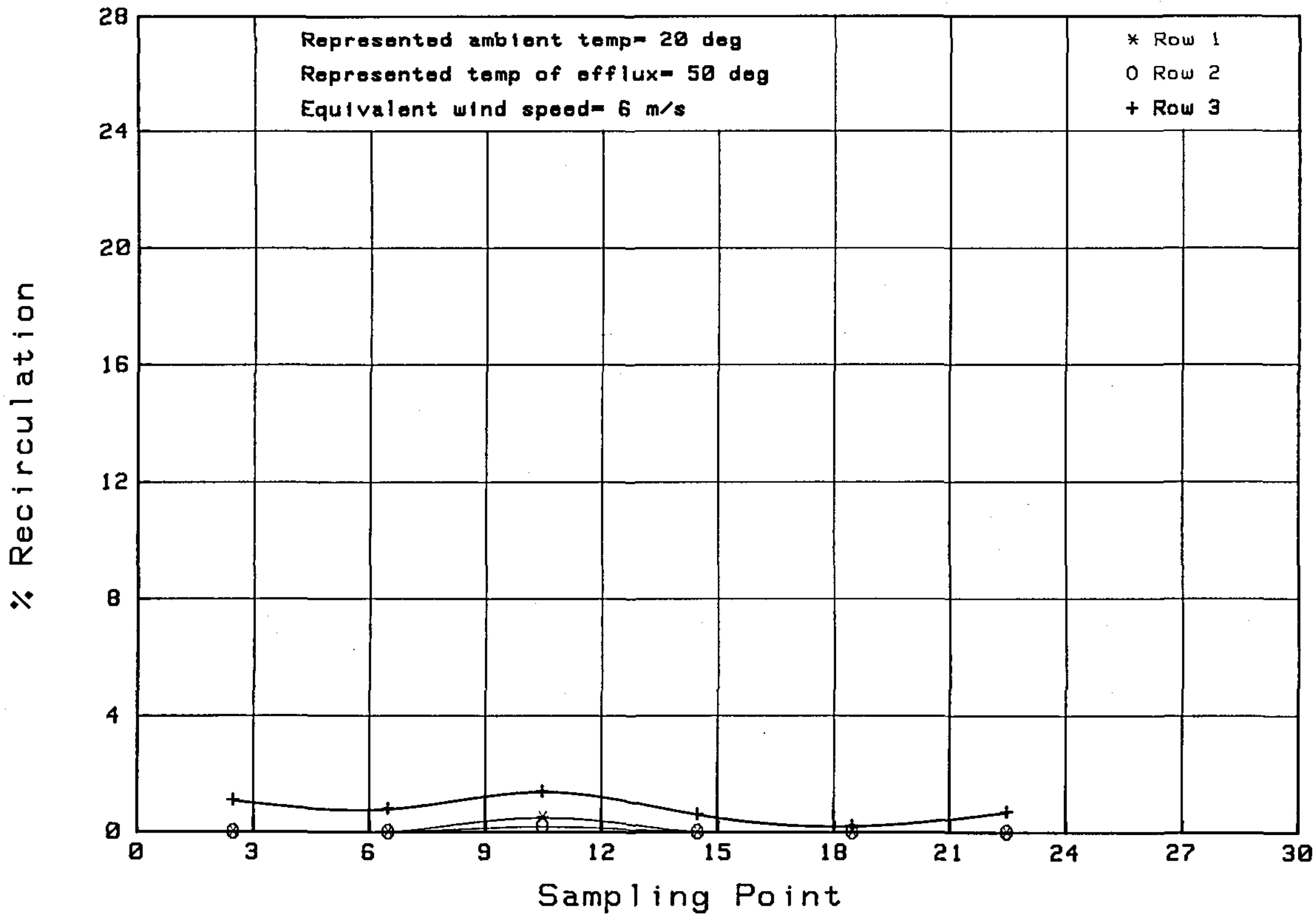

Fig 20. Hot air recirculation in heat exchanger

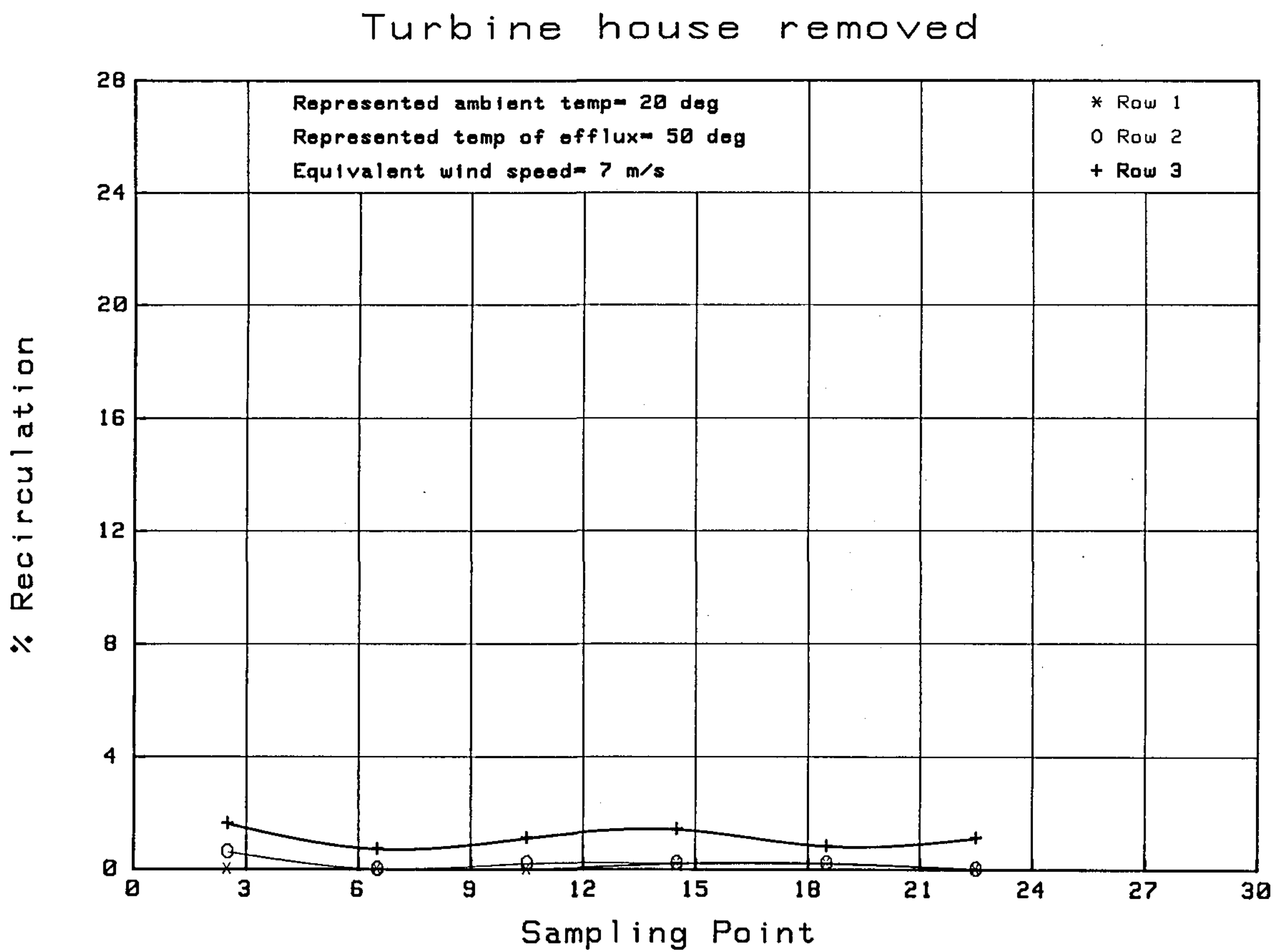

Fig 21. Hot air recirculation in heat exchanger

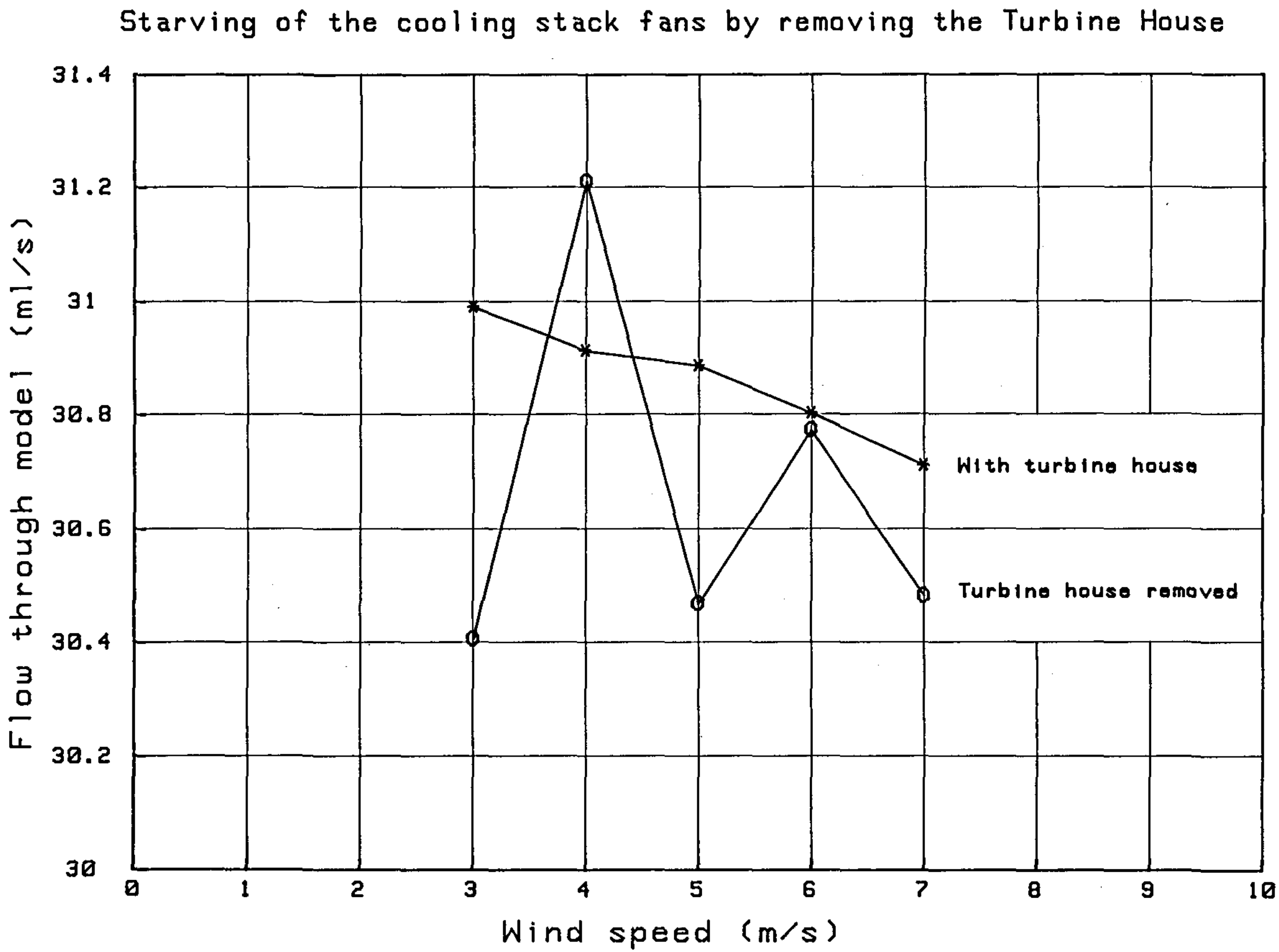

Fig 22. Wind speed vs flow through model